November/December 2008

GEAR TECHNOLOGY

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The Journal of Gear Manufacturing

State of the Gear Industry

- Results of Our Annual Survey
- Gear Success Stories Behind the Numbers

Technical Articles

- High-Contact-Ratio Spur Gears
- Induction Hardening of Gears, Part II
- Shaft Misalignment Influence on Bending Stresses

Plus

- Addendum: It's That Time of Year Again

THE GEAR INDUSTRY'S INFORMATION SOURCE

GEAR TECHNOLOGY

November/December 2008

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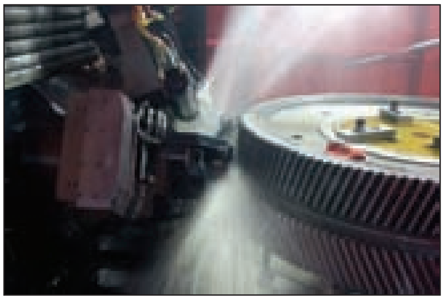
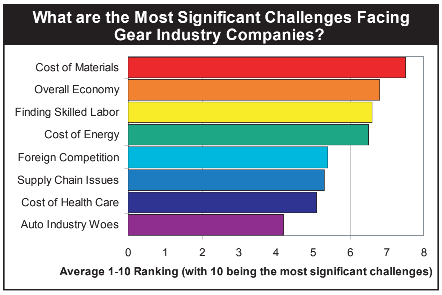
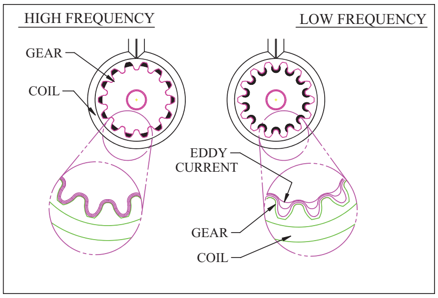
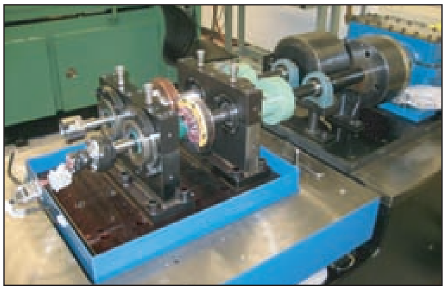
STATE OF THE GEAR INDUSTRY

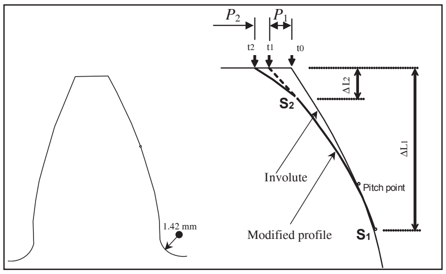




TECHNICAL ARTICLES







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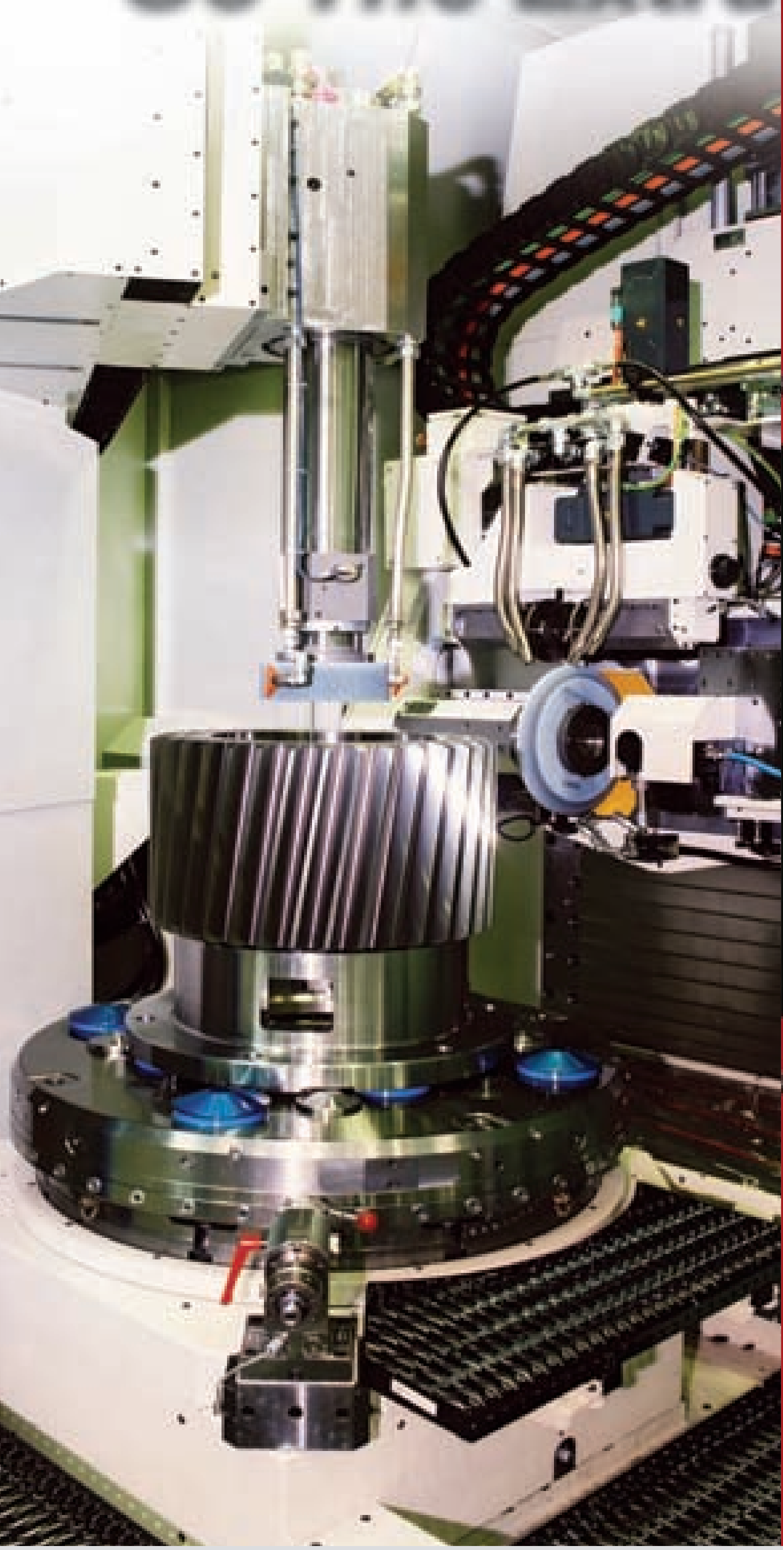

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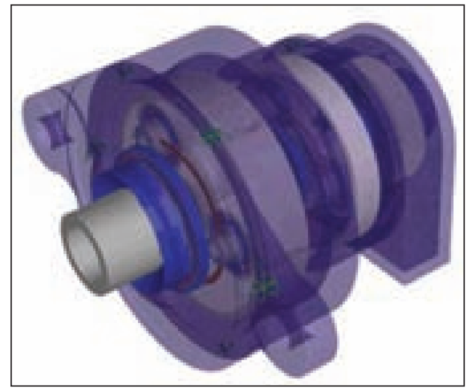

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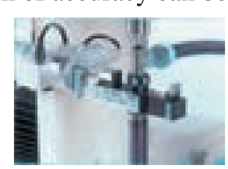

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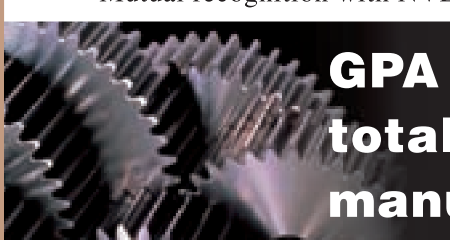

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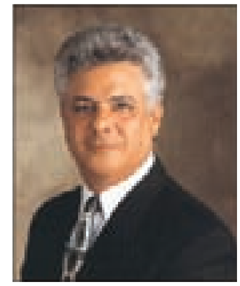

Don't Panic

The world economy is in turmoil. A year ago, the Dow Jones industrial average was more than 14,000. As I write this, after eight straight days of massive losses and a week of wild up-and-down swings, the average sits at about 8,900. Just watching the stock market's fluctuations—let alone thinking about what it means for your own personal investments—is enough to make your stomach churn.

On top of all that, we're three weeks away from the U.S. presidential election.

With all this uncertainty in the air, it was a heck of a time for us to send out our annual "State of the Gear Industry" survey. With pundits proclaiming gloom and doom, and headlines frightening everyone from Wall Street to Main Street and around the world, we were apprehensive about what kind of results our survey would reveal.

But as it turns out, the gear industry seems to be still going strong and is expected to continue to do so. We sent the survey October 8–9, in the middle of one of the worst weeks in U.S. stock market history (the Standard and Poor's index fell farther than in any week during the last 75 years). You'd think that the overall economy would have had a greater effect on our results. You'd think that watching their retirement accounts dwindle by the hour would have left people depressed about everything. But the mood among many respondents was still fairly positive, and their responses indicate expectations of continued growth for the gear industry in 2009. Eighty-five percent of our respondents reported that they're optimistic about their companies' abilities to compete over the next five years. Last year, when the stock market was soaring, that number was 88 percent.

More than half of respondents (56 percent) expect their company's sales volume to be higher next year than in 2008. Similarly, 57 percent expect production volume to be more next year. Sure, those numbers are down from last year, when 75 percent expected a sales volume increase and 79 percent expected a production volume increase. But with more than half the industry expecting revenue and production growth, the outlook is still positive.

In fact, the overall economy wasn't even the gear industry's No. 1 concern. According to our results, gear manufacturers are more concerned about the price and availability of steel. They're also worried about the cost of energy and their ability to find skilled labor.

Don't get me wrong—they're worried about the economy, too. A number of respondents recognized that an economy in recession will eventually affect more parts of the industry. In some cases, we know, the economic crisis is already having a big effect on manufacturers. For example, I've heard some cases of gear industry projects stalling due to the lack of available credit. The work is there, but companies are having a harder time today getting the financing they need to buy machine tools than they did a year ago.

Sitting here in mid-October, there's no telling which way the market is going to go. There's no way to be sure whether the government's various economic bailout packages will have the desired result, or if they work, how long we have to wait.

But as my longtime friend Iain Exeter, whom I call my Scottish rugby player/philosopher, is fond of telling me, you shouldn't worry about things over which you have no control; the people who do control those things don't ask your opinion anyway. Just go to work every day and figure out how to produce more gears, at a better quality and lower cost. That, you can control. When the economic crisis is over, you'll be well rewarded for your efforts.

We took the gear industry's pulse at what was arguably the worst possible moment. The fact that a wide range of respondents are still optimistic shows that at least the gear industry is still going strong and expects to keep doing so.

Michael Goldstein,
Publisher & Editor-in-Chief

P.S. Thank you to all who participated in our survey. If you'd like to make any comments or suggestions about the survey itself or the results, please e-mail *publisher@geartechnology.com*. We'd love to hear from you.

Mystery Solved (?!)

The September/October 2008 Addendum column about the mystery gear on the mountain created quite a stir among gear industry conspiracy theorists. We're sure the truth is still out there, but you'll have to judge for yourselves whether any of these readers and Addendum fans hit the mark.

Dear Editors:

I saw your article on "Mystery Gear on the Mountain" in *Gear Technology* of September/October 2008. You have requested our explanation for this mystery.

I failed to see the scale of this picture. Therefore I had difficulty in visualizing the gear's size. But comparing both pictures gave me the feeling that some smart photographer has manipulated this picture to throw off all engineers from their analysis efforts. It seems too good to believe the picture portrays reality, with all the rocks arranged precisely and the background of snow-covered mountains.

This is a face gear shown in the picture. It seems to be machined rather than cast, as would be true of a more ancient gear. It has a reduced-weight design for lower horsepower, as evidenced by the thin web and rim on the gear. Therefore your explanation that this gear may have been used for mining purposes doesn't fit very well.

The size of the gear looks like anyone could have carried it to the flat bed of a quarry and arranged all the rocks around it. The photographer may have put in the snow-covered mountain during his/her creativity efforts to fool engineers.

Regards,
Vallabh Vaghani, staff engineer
Hamilton-Sundstrand Aerospace
Rockford, IL

(*The Addendum staff has unilaterally taken offense at the suggestion that the photographs presented last issue were anything but untainted originals. In fact, they assure the editors that the photographer is a well-known acquaintance whose knowledge and reputation in the obscure is without doubt.*)

Good morning,

Let me first state that I do not know why the gear is there.

The picture appears to have been taken well above the tree line. The peak called "Mt. Baldy" in New Mexico is near the same latitude as Roswell. The height of Mt. Baldy is around 7,400 feet or so, and it is not in the Cimarron mountains.

Baldy Mountain, on the other hand, is at the same longitude as Roswell, but it is a couple hundred miles to the north (in the Sangre de Cristo mountain range). This Baldy Mountain is over 12,400 feet and is supposed to be the second highest peak in New Mexico. I haven't been on Baldy Mountain, but I have been on Mount Phillips 15 miles to the south. Phillips is above the tree line at 11,900 feet. (The tree line is somewhere around 11,200–11,500 feet). If the gear is on Baldy Mountain, then it is on Philmont Scout Reservation (as is Mount Phillips).

Having been to Philmont three times, I would suspect that the gear has been hauled to the top of the mountain by a couple of rangers (or mischievous scouts), just to make for an interesting discussion. There are several skits and dramas presented at the staffed camps to embellish the rugged history of the area. The staff camp at French Henry is well down from the summit on Baldy Mountain, but was supposed to be one of the better camps when I got to go there. The gear is probably tied to their evening program, likely as an incentive to get the scouts up to the top of Baldy Mountain. 12,400 feet is a tough climb with a backpack if you aren't acclimatized to the high elevation.

Sincerely,
Kevin Connor, H-1 senior drive systems engineer
Drive and Propeller Systems Tech Team, U.S. Navy
Cherry Point, NC

(*The Addendum staff confirms that the pictures were indeed taken on the Philmont Scout Reservation. We thank Mr. Connor for correcting us about the true name of the peak—Baldy Mountain, not Mt. Baldy. The Addendum staff neither confirms nor denies any of the other details outlined above*).

Dear Editors:

During the recent earthquakes in California, the center of the earth may have shifted, causing huge cracks to surface to the outer crust of the earth. The power of these earthquakes, steam, flowing lava and back pressure can, in fact, expel matter from the center of the earth to the earth's surface.

These gears therefore prove that the earth's axis is made of high-strength, high-temperature gears. And that gears are, in fact, the center of the universe.

Gary Matosian, manufacturing engineer
Moog, Inc.
Torrance, CA

(*"Interesting" was the only response the editors could get from the Addendum staff. But readers should be aware that an Addendum correspondent has been dispatched to investigate this theory.*)

Another Lean Tool

MAKING HEAT-TREAT INROADS

Typical hardness capability using HighLight direct diode laser system		
Material	**Maximum Hardness (Rc)**	**Maximum Depth (mm)**
Carbon Steels		
1080	68	2
1075	68	2
1045	60	1.5
1030	50	0.75
Heat Treatable Alloys		
4140	68	2
4340	68	2
Heat Treatable Stainless Steel		
420	65	1.6
410	50	0.5
Cast Irons		
Gray	65	1
Ductile	55	0.75

Anyone who attended the recent IMTS or National Manufacturing Week shows will tell you that lean manufacturing was on display in a big way. With lead time becoming one of the most important issues in getting and retaining business, companies are jumping into the lean pool in hopes of getting their products made faster and shipped earlier than their competitors.

And the heat treating sector is no exception, which is where the relatively unknown diode laser heat treat method enters the picture. CO_2 laser is the more commonly known method, but it has its drawbacks (more on that later). Bottom line, however, laser heat treating is a method that—while it isn't suitable for all applications—can eliminate two time- and cost-consuming extra steps in the heat treat process.

But first some background.

"CO_2 lasers are cheaper to buy but very expensive to run due to CO_2 cost," says Sri Venkat, business unit manager/direct diode systems, for Coherent, Inc., a seller of diode lasers. "You have to frequently maintain them, and the chemical and maintenance costs are not cheap; you have to pay a lot more to keep them going. They are also very 'tweaky' in that you have to have a laser specialist on your shop floor who knows how to use them. It's like buying a car that's prone to breaking down, and you almost have to have a mechanic in the family."

In addition, says Venkat, "CO_2 is more expensive to run than diode laser due to lower wall-plug efficiency.

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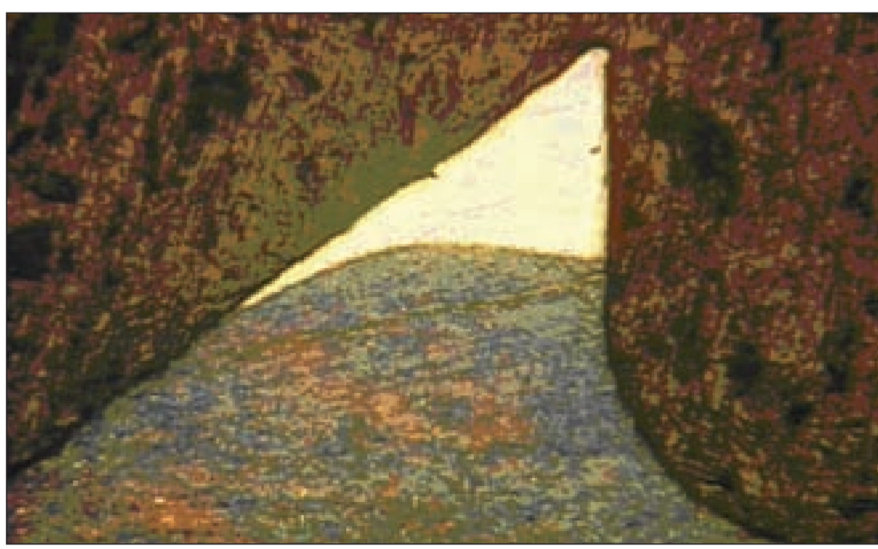

A saw tooth tip selectively hardened by a diode laser.

Typical heat treating results with the HighLight laser.

Diode lasers have much higher wall-plug efficiency."

Diode lasers, according to Venkat, are something altogether different. He uses the analogy of coal vs. solar power to make his point.

"Diode lasers are more state-of-the-art; they're a completely different kind of technology. Diode is solid state—there's no gas in there."

He goes on to explain that diode was a process first used for fiber optic cabling and later in CD and DVD player applications. However, as is still the case with CO_2 lasers, diode lasers carried a big price tag as well. But things changed after the telecom bubble burst. Now, the cost has come down, making the process attractive for specific types of part heat treating and other uses. The medical industry, for instance, is a huge market for laser; printing is another. And now, industrial manufacturing, according to Venkat. He explains that a major (unnamed) manufacturer of very large, heavy equipment approached his company some years ago with an application issue regarding heat treating of parts. The short version of the story is that Coherent was successful in meeting their needs and it's been a glass half-full situation ever since.

Which brings us to the lean part of the story. Venkat explains:

"They wanted to implement lean manufacturing, and they wanted to get away from having a large inventory of parts. Because when you have to process carburized heating, it relies on a batch of parts being put in a furnace, and then you put them on a shelf, so you have inventory buildup."

The company looked at both methods: CO_2 and diode. But, as Venkat points out, "One of the problems (in addition to unit cost) with CO_2 is that you have to pre-coat the part with black oxide—an extra step—whereas with the diode laser you don't need to do that."

In addition, "They were looking for a much more compact heat source and the diode laser is very small—you can literally mount it directly onto a robot for heat treating."

And as for gear-specific applications, "Gear hardening requires localized heat treating without distortion," he says. "Diode lasers are ideally suitable for both these attributes due to their small beam size and low thermal input to achieve required hardness."

Another important advantage of laser is that there is "no post grinding or surface modification, period," according to Venkat. Also, it is easy to use, with few learning curve issues.

It is useful to point out that laser heat treating is designed for case hardening only. Also, it is for part-by-part, not bulk, hardening. But if you have parts with complex geometry and profiles, diode heat treating can be effective.

The other caveat is that laser heat treating is uniquely conducive to automated manufacturing. Obviously, heat treating onsies-twosies manually is not an option. But if your part handling is automated, laser is a viable option. Automotive comes immediately to mind, among other high-volume, automated processes.

For more information:

Coherent Inc.
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The OLS Model 1100 deburring machine from McNeil Industries features various options for users to tailor the machine to their specific finishing needs, including compatible tooling choices and control and power variations. With a 52 x 47 inch platform design, the Model 1100 is well-suited to lower-volume, hand-loaded finishing of a range of gears, sprockets and other parts with diameters up to six inches.

The basic version of Model 1100 uses 110 VAC power and pneumatic cutting pressure with two tooling heads, which adapt to a range of quick-change pneumatic brushes, grinders, cutters or other tools. A 240/480 VAC power

continued

option is available for applications that involve more horsepower and tooling force. The high-power upgrade cutting pressure is supplied by electric motors.

Other options to choose from include movable chucks or air-actuated collets that accommodate different part clamping requirements and can make loading and unloading easier for the operator. The machine's functions and part profiles are customized per order, including various programming languages. The machines are PLC-controlled, and users opt for the PLC model to best suit their plant systems and equipment needs.

The Model 1100 has special application features that include a highly efficient dust collection system for stand-alone or plant-wide system integration operations. A sump-based liquid cooling system is available to deal with high-temperature applications.

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AUTOMATICALLY LOADS PARTS

The SL-20APL from Haas Automation is capable of automatically picking up parts from an external tray and loading them directly in the lathe. The result is turning or part flips without an operator for double-ended machining.

The maximum cutting capacity of the SL-20 APL is 10.3" x 20", and the maximum swing is 23" over the front apron with a 2" standard bar capacity. The machine uses a 20 hp vector dual-drive spindle with 4,000 rpm providing 154 ft-lb of torque. Constant surface-feed cuts are achieved by fast wye-delta switching that provides a wide constant-horsepower band.

The SL-20APL has an A2-6 spindle nose, an 8" hydraulic three-jaw chuck and a 10-station tool turret. If long-part machining is necessary, Haas offers a programmable hydraulic tailstock as an extra option.

Two gripper configurations are standard with the lathe: a shaft gripper deals with parts up to 5" diameter, while a disc gripper handles parts up to 7" diameter. Each configuration is capable of parts up to 15 pounds. The automated parts loader (APL) loads raw pieces and removes the complete parts; it also flips pieces for second operations and lingers until long shaft pieces are machined before bringing them back to the table.

The APL, programmed through the lathe's control, operates in the background of the lathe operations, so the machine functions unattended and almost continuously.

For more information:
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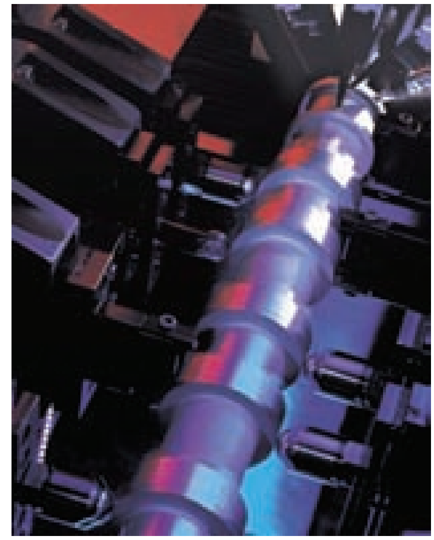
micron accuracy measuring camshafts for rise error and chattermark detection. It features top loading or through-feed bottom loading and is capable of measuring up to 200 parts-per-hour.

The Adcole 1310 is fully automatic and uses separate measuring heads for every journal and cam lobe. The gage has a 0.1 micron resolution and takes one data point each 1/10th degree or 36 data points per revolution. Adjustable followers allow quick changeover for inspecting different camshafts. It can measure 10 or more parameters that include radius, profile, taper, crown, timing angle, diameter, velocity,



acceleration, runout and roundness and concentricity.

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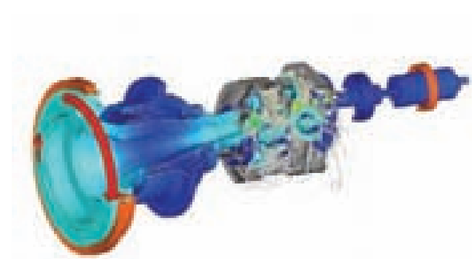
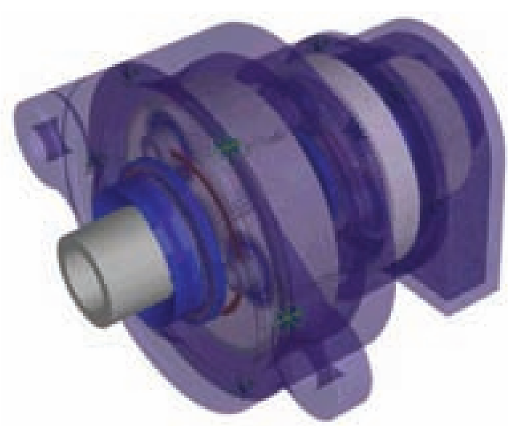


Romax Technology has issued the first service pack for *RomaxDesigner*

R12.6. The release contains two new modules as well as updates and extensions to existing modules. An advanced spherical roller bearing module takes into consideration the profiling effect of the bearing rollers and raceways. The second is a new gearbox efficiency module that includes gear drag analysis used to determine the power losses from individual gears.

The implementation of gear mesh alignment contribution allows the contribution of shafts, bearings, gears and housing to the overall misalignment of a gear mesh to be reported. An analysis setting "Use gear mesh rotation mismesh analysis" allows gears to be rotated on their mounting shaft away from the nominal calculated rotations to take into account gear initial rotation errors.



"The additional features and updates will greatly enhance a user's experience," says Nick Lewin, product development manager. "Working with our clients has helped us develop the product further, and we believe these additions and enhancements will make *RomaxDesigner* an invaluable tool."

For more information:

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Bates

EXPANDS PROMATIC HONE TOOLING LINE

Bates Technologies has expanded its ProMatic line of production-style hone tooling to include an economy version (E-line) designed to fit and function on many common small job shop hone machines. The tooling utilizes the maximum number of abrasive ledges (stones) that may be required to meet cylindricity and roundness requirements. The design features of the line stem from more than 80 years of development and manufacturing of tooling technology used to hone engine cylinder bores, connecting rods and transmission gears for several OEM engine and powertrain manufacturers.

"ProMatic E-line has been designed for those facilities that are looking to improve upon their existing two- and four- stone hone heads but do not have a requirement for high-production, high-precision OEM style hone tooling," says David Chobany, vice president of Bates Technologies.

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The XTractor metal separator from Bunting Magnetics Company features a self-cleaning magnetic drum separator that allows for continuous separation of ferrous and non-ferrous materials for recycling in any manufacturing operation. Scrap metal prices have risen since 2003, and recycling facilities offer a higher value for clean, separated material.

"In today's economy, you need a competitive edge in every part of your manufacturing operation. Bunting Magnetics Co. and the XTractor can help your company have a huge, positive impact on your bottom line," says Charles Whitt, Bunting's director of marketing. "Prices have skyrocketed for scrap and the XTractor allows our customers to take advantage of those market prices and turn their scrap into cash. In fact, one company has seen their investment pay for itself in just six months."

The XTractor is custom made for customers' specific processing rate requirements. The standard units have a 12" diameter x 12" wide drum surface capable of processing up to 1,500 lbs. per hour, or an 18" diameter x 24" wide surface that is capable of 2,500 lbs. per hour. A custom made work stand for processing scrap is available as an option in addition to a rake for funneling scrap into the separator.

For more information:
Bunting Magnetics Co.
500 S. Spencer Ave.
P.O. Box 468
Newton, KS 67114-0468
Phone: (800) 835-2526
bmc@buntingmagnetics.com
www.buntingmagnetics.com

Bore Gage

IMMUNE TO FLUCTUATIONS IN SHOP FLOOR VARIABLES

The PG-Series bore gages from Sunnen are designed for process control in a lean manufacturing setting, and they feature lab-precise accuracy of ±0.0006 mm (0.000025 inch) with a mechanical design. They do not require electrical or air connections, meaning they are unaffected by fluctuations in line voltage or air pressure.

A PG gage is capable of measuring a range of part diameters and costs

continued

less than dedicated devices like air gages, according to the company's press release. They do not need plug gages, mikes, jo blocks, master rings or micrometers because they can be mounted directly to a machining station. The bore gage allows machine operators to confirm part size with a visual analog scale, which resembles a speedometer and provides readouts down to 0.001 mm (0.000040 inch) to inspect machining operations or quickly sample large vendor lots.

The PG-Series bore gages examine an entire bore for diameter, taper, barrel, bell mouth, out-of-round and lobing. Bores are scanned from end-to-end by employing a sliding faceplate with adjustable stops. A floating-reed system is responsible for locking gage fingers at a requested diameter size against drift or lag. This removes friction and wear, and accuracy is not affected by temperature, moisture, air pressure or vibration. Accuracy is guaranteed by a calibration ring.

The PG-Series bore gages can be purchased or leased on a monthly basis from Sunnen. This includes maintenance and system updates. Available models include gage IDs from 0.090 to 4.310 inches or 2.0–109.47 mm.

For more information:

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sales@sunnen.com
www.sunnen.com

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HAVLIK INTERNATIONAL MACHINERY

WE'VE INCREASED OUR GRINDING CAPACITY.

HAVLIK GEAR HAS AQUIRED ONE OF THE LARGEST GRINDERS IN NORTH AMERICA.

KAPP NILES

The ZP 40 grinding machine features the ultimate in design construction with true hydrostatic guideways and rotary table. The work table is direct drive. Nodular cast iron is used for the machine beds, columns, and all sub-assembly housings, for stability. The machine accuracy is therefore assured for a long lifetime.

The ZP40 is designed to profile grind spur and helical external and internal gears using dressable corundum or sintered grinding wheels. An integrated measuring device further increases machine's flexibility. The ZP machine can be set up to accommodate application-specific requirements for maximum flexibility, high precision and optimum productivity.

It is specifically designed to accommodate shaft pinions, mating bull gears and internal ring gears.

Extended capacity for gears and profiles is:

- External grinding capability to 50 module (0.5 DP)
- Internal grinding capability to 25 module (1.0 DP)

KAPP NILES

NILES ZP 40

Tip diameter max.	4,000 mm (157.48 in.)
Distance rotary table – grinding wheel min. / max.	435 / 2.260 mm (17.13 / 88.98 in.)
Number of teeth	any
Profile depth external	100 mm (3.94 in.)
Module-External (at tooth height 2,25 x mod. max.	50 mm (0.5 DP)
Module-Internal (at tooth height 2.25 x mod)	25 mm (1.0 DP)
Helix angle max.	40 deg.
Stroke length max.	1,550 mm (61.02 in.)
Rotary table- load max.	40,000 kg (88,000 LBS)
Table diameter	2,250 mm (88.58 in)
Bore diameter x depth	1.365 x 265 mm (53.74 x 10.43 in.)
Grinding wheel- diameter max.	500 mm (19.69 in.)
Dressable diameter min.	220 mm (8.66 in.)
Width max.	130 mm (3.94 in.)
Speed max.	2,200 RPM
Driving power max.	40 kW

A bumper crop in wind farming will continue in 2009, especially with the recent extension of the energy tax credit by Congress. (Photo courtesy of Vestas.)

CLOGGED SUPPLY CHAIN HAS GEAR COMPANIES IN

"HURRY-UP-AND-WAIT" MODE

Lead Time Remains Crucial, but Steel is Precious

Jack McGuinn, Senior Editor

Never have so few served so many. That, in essence, describes gear makers and the role they play in our world. Think of it—although the gear cutting industry represents much less than one percent of the global workforce—the gears it produces are what make things run in practically every industry and profession imaginable. From bulldozers to Rolexes, gears are an integral part of the mix.

With that—and the scary Wall Street headlines in mind—*Gear Technology* posed some questions to a number of players in the industry to get their take on how things went in 2008 and, where possible, what they see for 2009.

Many of the major issues facing the industry in 2007 remained a factor this year and will continue to do so in the next. Supply chain kinks plague the makers of very large gears for wind turbine, mining and off-road heavy equipment, for example, as demand far outstrips supply. (American Wind Energy Association figures for example show that the U.S. will increase the installed capacity of wind turbines from 17,000 MW in 2007 to 25,000 MW by year's end.) But the high-grade steel necessary to produce them remains a rare commodity indeed. Reports of lead times for wind turbines of up to four years are not uncommon, as players around the world seek ways to secure the precious steel needed to make them for their own countries. In reaction to this and other applications, companies are striving to introduce new manufacturing capabilities and processes to their production lines in hopes of knocking off a chunk of that lead time that has become the difference-maker in gaining a competitive edge. And in the grand tradition of the squeaky wheel getting the grease, the larger gear companies are closer to the front of the line by virtue of their ordering power. As a result, smaller job shops and the like must wait hat-in-hand for their turn.

Concerning alternative energy developments, Congress finally renewed—for the usual one year—the energy tax credit as part of the Wall Street bailout package in October. Had it not been passed, the wind energy industry and attendant investment in ongoing development would have dried up quickly.

And then there's the price of oil and energy. Never-before-seen prices are driving up the cost of most everything needed to run a business, especially in the shipping department. The cost of shipping is now at record highs, causing some gear makers to rethink their selling efforts overseas, especially to China and other Asian markets.

Also troubling is the continued shortage of skilled workers needed to keep those production lines humming. Anecdotally—supply chain issues aside—a number of gear makers are sitting on orders they can't fill because they don't have the experienced personnel to keep up.

And last, there's the Wall Street blow-up and attendant credit crunch in effect as this goes to press. The bailout that emerged from Congress will be key in determining where all this leads us.

And yet, given the following responses, business remains good for the gear industry. It's slowing down some, to be sure—recent reports show U.S. manufacturing down in general—but it is hard to find any gear companies on life support. And of course slowdowns are relative; as one veteran in the machine tool industry recently answered when asked how his business was doing, "Let's just say we're catching up," thus acknowledging a drop-off while alluding to yet-unfilled orders that will take them through the remainder of 2008 and beyond.

Catching up or not, 85 percent of respondents—compared to 88 percent last year—to our annual gear industry survey are "optimistic about their ability to compete over the next five years."

Some other breakouts from the survey (*with previous year's numbers in parentheses*):

- Thirty-eight percent of gear industry respondents work at locations where employment increased in 2008 (55 percent last year).
- Thirty-six percent of

continued

gear industry respondents expect employment at their location to increase in 2009 (61 percent last year).

- Sixty-six percent saw production volumes increase in 2008 (71 percent last year).
- Fifty-seven percent expect production volume to increase in 2008 (79 percent last year).
- Sixty-five percent saw sales volume increase in 2008 (71 percent last year).
- Fifty-six percent expect sales volume to increase in 2009 (75 percent last year).
- Forty-two percent work at locations where capital spending increased in 2008 (48 percent last year).
- Thirty-three percent expect capital spending at their locations to increase in 2009 (44 percent last year).

As for other issues affecting the industry, material cost, the economy, finding skilled labor, energy costs and foreign competition headed the list of concerns.

(*Ed.'s note: For more specifics, turn to our annual gear survey results beginning on page 34.*)

With those numbers in mind, let's get to our experts for their take on things, relative to how their own companies are doing. Only gear cutters (excepting one plastic gear molder) were approached for this article, thus providing some insights from those on the very front lines of the gear industry. Following are excerpts from their responses to a series of questions.

"If the news media would stop trying to scare the hell out of everybody, 2009 should be as good as 2008 for us here at Arrow Gear." —Joe Arvin

Forest City Gear's Fred Young says 2008 is "on track to be about 10 percent ahead of 2007 in sales. As for 2009, "While I see the overall economy slowing in 2009, I believe those of us nimble enough and not afraid to reinvest will be able to continue to secure new business. And regarding sales to China: "We are continuing to sell a modest amount in China, though I suspect their rate of growth will slow along with the world economy. In fact, we see some items returning from overseas sourcing due to an inability to deliver quality product on time. Unfortunately, overseas pricing has in some cases been wildly optimistic to try to gain market share. The Europeans spend a lot of money tooling up for jobs delaying payback by burying start-up costs. This has led to some unrealistic price expectations when the buyer seeks to resource in the U.S."

At **Schafer Gear**, company president **Bipin Doshi** says "2008 is a good year and we expect 5 percent or more growth in revenue. Unless the present economic news has a significant negative effect on the economy, we are expecting 10-plus percent growth in 2009 and an additional 20 percent growth in 2010."

Circle Gear's Michael McKernin says 2008 was—in a word—"Great."

Karl Seitz, vice president of **Seitz Gear Corp**., says "Our domestic sales are about 10 percent above budget and our bookings of new future business are 18 percent above budget. China has some new and exciting programs that will take off in early 2009." As for '09, "We see about a 10 to 15 percent growth in sales domestically and our new programs in our Seitz facility in China will make for a promising year in 2009."

Arrow Gear's Joe Arvin points out that "First off, 2008 is not over; however, my prediction is that 2008 will be our best year since 2001. We have the largest backlog in the history of the company. He adds, "If the news media would stop trying to scare the hell out of everybody, 2009 should be as good as 2008 for us here at Arrow Gear."

Milwaukee Gear's president, **Rick Fullington**, says 2008 was "a good year with both shipments and hours produced up 7 percent. At the same time, quoting levels have been strong. Prior to the (Wall Street) events of the last week, we felt that (business) would be level with our 2008 activity."

With their work in wind turbine gears, **Cam Drecoll** of **Brad Foote** (now a subsidiary of Broadwind Energy, Inc.) is optimistic as well. "Brad Foote continues to grow well above industry levels and has for the second year in a row. 2009 will be another record year for the company."

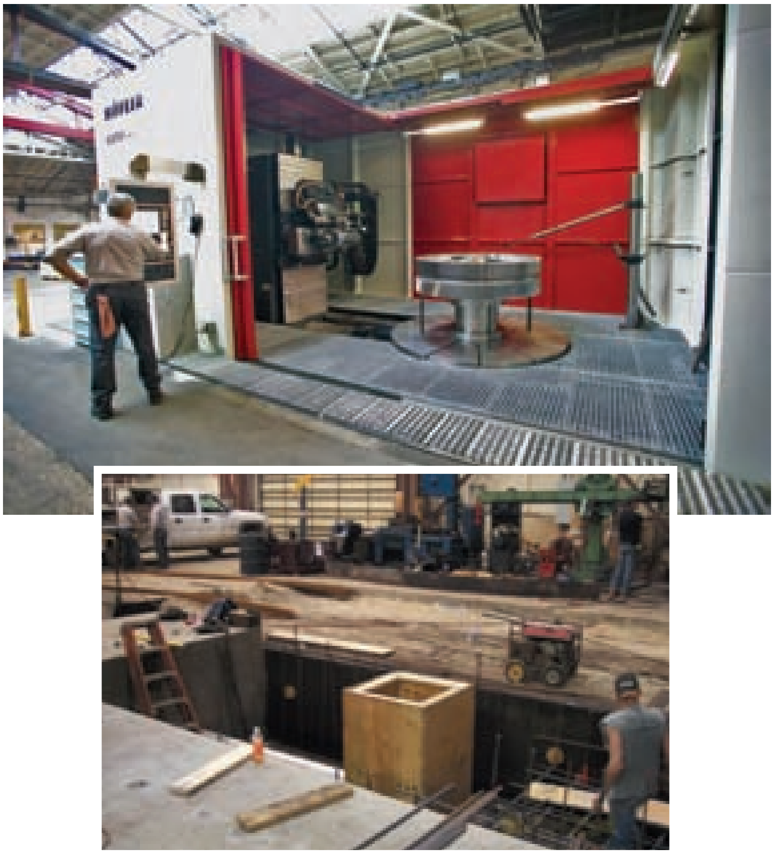
And speaking of records, **HMC**—another maker of outsize gears—just

A Koepfer MZ 130 is just one of the new machinery additions at Forest City Gear. (Courtesy Forest City Gear.)

enjoyed another one, says sales manager **John Schnarr**. “HMC has recorded another record year for both sales and shipments in 2008, with the final quarter yet to come.” As for 2009, “We expect to see a continued demand in several markets that we consider key to our success. Beginning in 2006, our president and CEO, Robert J. Smith III, committed to investing heavily in new plant expansion and machine tool technology. By the end of 2008, our Phase I plan will be providing HMC dramatically increased capacity and capability to better serve our current customer base and expand our market horizons. We look forward to 2009 and the increased opportunities it holds for our company.”

Ron Wright, corporate sales engineering manager for **Gear Motions** (**Nixon/Oliver Gear**), says “We are seeing about a five to 10 percent increase in sales.”

Lastly, **Frank Romans** at **Chicago** (**now Overton Chicago** [see Industry News]) **Gear**, says “All reports and interaction we see is that 2008 was a booming year for the gear market. In one word, energy—i.e., oil, gas mining and (wind) sources.”



(Top photo) Big gears and those who make them will continue to be a mainstay in 2009. (Courtesy Brad Foote Gear Works.) (Bottom photo) HMC personnel preparing the way for the Hofler 600 form grinder, the world’s largest gear grinder. (Courtesy HMC.)

“All reports and interaction we see is that 2008 was a booming year for the gear market. In one word, energy—i.e., oil, gas, mining and (wind) sources.”
—Frank Romans

Gear Technology then asked these gentlemen what they perceived as the most significant development in the gear industry in 2008.

For FCG’s Young, it’s in the wind. “The most significant development is the realization that wind energy is a real thing for at least five to 10 years and deserving of investment. Those late to jump on the bandwagon are finding a lengthy time to acquire gear equipment. This has caused some to buy their way into the industry. As an adjunct, some folks are also cognizant that growth opportunities are more prevalent in higher-quality applications and that one has to have the equipment on the floor and operating to take advantage of work that comes knocking.” Young also mentions FCG’s recent acquisition of ISO 13485 certification for medical device manufacturing as a significant development.

For Schafer Gear’s Doshi, it was “significant buyouts by investment companies of several gear manufacturers and a focus on wind energy markets.”

Karl Seitz cited “more efficiency and the latest technology in inspecting plastic gears.”

At Arrow Gear, “There are three things,” says Arvin. “One, we stopped taking LTA’s (long-term agreements) with year-over-year price reductions. We had given our largest customer a 49 percent price reduction over the past 10 years, due to their LTA which required (price deductions). Second, we raised our prices for the first time in 10 years. Third, we have more business (backlog) than we produce.”

For Fullington, “It tended to be the opportunity—both directly and indirectly—associated with anything energy related. The weak dollar certainly helped and more international gear companies looked for investments in the U.S.”

Brad Foote’s Drecoll believes that “The changes in the gear industry continue to be more evolutionary. Tooth grinding technology continues to get better; non-destructive testing continues to evolve, allowing for better detection of flaws. The level of required quality continues to be pushed upwards. Anyone can now do a single gear to the old AGMA 15, but the challenge is to compete in serial production at DIN 3.”

And echoing Young, HMC’s Schnarr says “The emergence of wind energy as a viable alternative, and the increased demand it has placed upon manufacturers, has had a dramatic

continued

effect on changing the complexion of the entire gear industry."

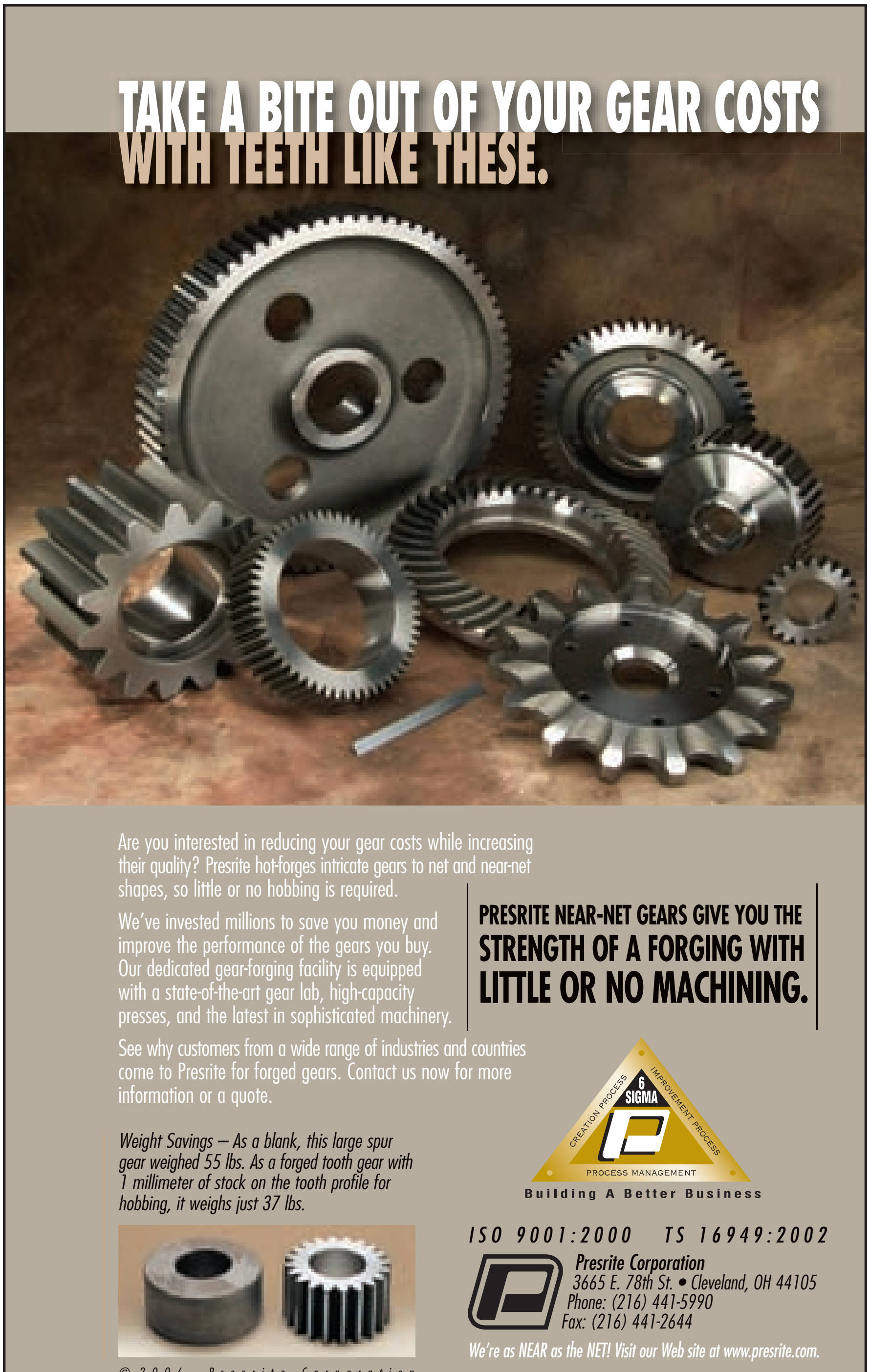
"For our company," says Overton's Romans, it was "the consolidation of gear companies by either larger companies or investment companies. No doubt the wind energy market has been the key to the increased value of gear companies.

We then asked if they foresee what the "next big thing" will be in 2009.

"Energy of all kinds," says McKernin.

"The biggest issues for 2009 are the world economy and the price of natural gas and oil," says Romans.

"Business will continue to slow," says Young, "in some areas, especially related to home building and automotive, but opportunities will be available even in those areas where people are trying to develop more fuel-efficient cars requiring higher-quality gears that come in smaller, more compact, high-power density. Make that gear ground rather than shaved, for instance. This trend will continue in other industries where people understand we have to become more efficient at producing higher-quality products.



"The most significant development is the realization that wind energy is a real thing for at least five to 10 years and deserving of investment. Those late to jump on the bandwagon are finding a lengthy time to acquire gear equipment. This has caused some to buy their way into the industry."—Fred Young

For Doshi, the trend will be "continued focus on energy- and commodity-related markets.

Says Fullington, "The U.S. credit crisis and its global impact will be the story in 2009.

And Drecoll—"Look for robotic Barkhausen effect to move into the marketplace on a more widespread basis.

Schnarr says HMC "will continue to see strong demand from all markets related to energy. That includes the old standard sources of oil, gas and coal, as well as developing wind and nuclear generation." In addition, "Increased development of emerging nations will continue to increase demand for energy resources and raw materials. The building of their commercial infrastructures will pace the increased demand for minerals (coal, copper, iron and aluminum) and affect requirements at both ends of the spectrum, from mining to processing."

As for Arvin, he's taking the practical view: "I will be able to tell you this time next year."

And last, given the good year the industry enjoyed in '08, we asked who was investing in new machinery.

"On $10,000,000 sales in '07 we have added about $6,000,000 worth of new gear toys in the past one-and-a-half-years, which we are in the process of digesting," says Young. "We have integrated a new-technology chamfering machine in our deburring area. We've also added a new, more compact, vertical generating grinder (with form grinding ability as well), which will operate off our (new) centralized filtration and chiller system, allowing us to recapture some more valuable (shop floor) real estate. Also due in is a replacement for our Swiss hobbing machine, which features very high speed for gears and threads." Young also mentions the possible addition of a video inspection system "for burrs in very obscure locations on parts. My bankers willing, I have a yen to acquire some more gear grinding equipment (in 2009) and possibly upgrading some of our older gear hobbers."

Schnarr says new machinery acquisitions and a new facility were a big part of HMC's 2008. "In 2008, we moved into a new, high-bay (11,250 sq. ft.) welding and fabrication building (if you build it, big-gear customers will follow), and added several new machine tools: Höfler 6000 form grinder; Höfler Rapid 6000 form grinder; and a Hankook-VTC-3040Y vertical turning center."

"The changes in the gear industry continue to be more evolutionary. Tooth grinding technology continues to get better; non-destructive testing continues to evolve, allowing for better detection of flaws. The level of required quality continues to be pushed upwards. Anyone can now do a single gear to the old AGMA 15, but the challenge is to compete in serial production at DIN 3." —Cameron Drecoll

As for Brad Foote, "BFGW announced earlier this year the addition of 14 Höfler grinders and hobber/gashers for installation in 2008," says Drecoll. "Brad Foote will have 18 large carburizing furnaces running by year's end. 2009 will bring further growth to support our customer needs."

And at Seitz Gear, "We have added a new 300-ton electric machine," says Seitz, "with more to follow in '09. We also plan to increase our capabilities in inspecting our gearing and component parts."

Back at Gear Motions, Wright says "We have bought new equipment this year and will likely buy more next year."

At Milwaukee Gear, says Fullington, "We'll include a form grinder and gear checker," among other equipment."

Circle Gear is taking a wait-and-see approach regarding capital investment for 2008 and beyond, for more than one reason. "Cost is the main factor why we are not buying new equipment, and a lack of market consistency," says McKernin. "New equipment requires years (in payment), and none of our customers are placing orders out past six months. It's hard to commit to a multiyear payment schedule without a long-term commitment from a client."

And "Yes," says Arvin, "we have

continued

added gear grinding and gear cutting equipment in '08 and will be adding more gear cutting equipment in '09."

So there you have it—an unscientific look at some success stories in 2008 to help round out the state of the gear industry in a manner that goes beyond the hard numbers. All in all, a very good year is almost in the books, with seemingly more to come in 2009.

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GEAR INDUSTRY

Results of Research on Trends in Employment, Outsourcing, Machine Tool Investment and Other Gear Industry Business Practices

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

The respondents considered here all work at locations where gears, splines, sprockets, worms and similar products are manufactured. They work for gear manufacturing job shops (47 percent), captive shops at OEMs (50 percent) and shops manufacturing gears for maintenance, spares and their own use (3 percent).

The survey covers gear manufacturing around the world, with 58 percent of respondents working in the United States, and 42 percent outside the United States.

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

continued

What Factors Are Presenting Significant Challenges to Your Business?

"Availability of capital."
—Corporate executive at a U.S. manufacturer of replacement gearing

"Available heat treatment capacity in the U.S. Collaborative supply chain partners that understand the alternative energy market and know how to support growth."
—Purchasing professional at a U.S. gearbox assembly plant

"Business insurance."
—Corporate executive at a U.S. manufacturer of marine transmission units

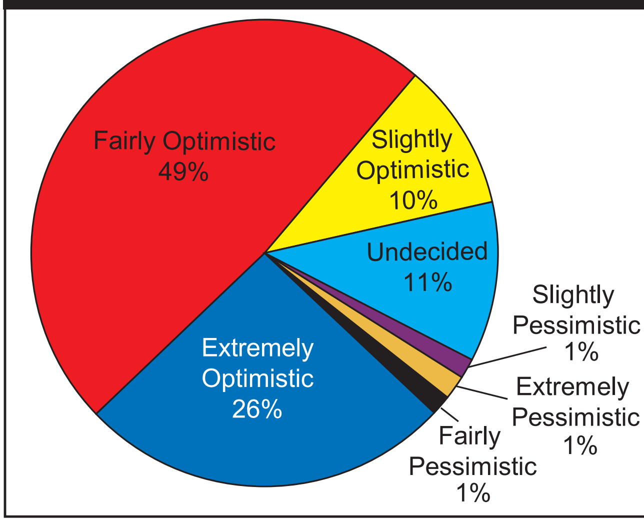
"Commercial viability of OEM customer(s)."
—Chief engineer at a U.S. manufacturer of pump shafts for construction/off-road equipment

"Congress. Stock market. Drugs."
—Corporate executive at a U.S. manufacturer of oilfield gears

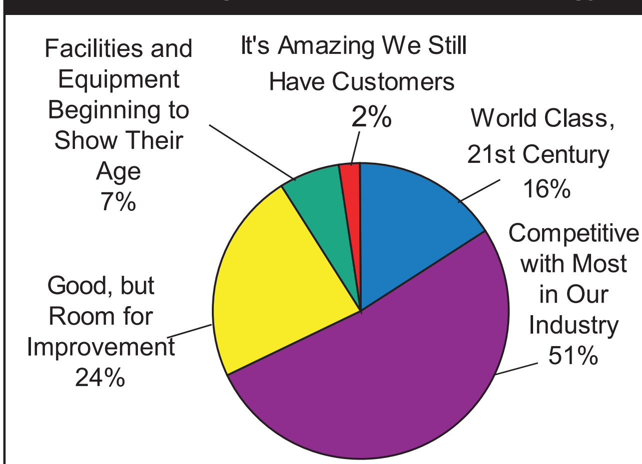
"Continuous product updating."
—R&D supervisor at an Italian speed reducer manufacturer

"Cost reduction initiatives."
—Production worker at an Indian automotive gearbox manufacturer)

85% of Gear Industry Respondents are Optimistic About their Ability to Compete over the Next Five Years



How Do Respondents Describe Their Manufacturing Operations and Technology?



What Factors Are Presenting Significant Challenges to Your Business?

"Credit crunch and therefore sponsorship of race teams."
—Engineering manager at a British motorsport transmission manufacturer

"Customer inability to compare quality difference between low-cost Far East and domestic products."
—Corporate executive at a Canadian manufacturer of workholding devices

"Difficulty in finding training center for special gear technology (marine and wind turbine)."
—Design engineer for a Korean manufacturer of speed reducers

"Exchange rate fluctuations."
—Sales manager at a British job shop

"Financing facilities for wind parks and infrastructure projects."
—Indian gear manufacturing consultant

"Fuel cost."
—Design engineer for a U.S. manufacturer of diesel engines

"Good quality, better pricing, scheduled delivery on time. To become a continuous supplier."
—Production worker at a Turkish manufacturer of gears and transmissions

"High cost of materials, increase of fuel prices, decline of U.S.-based manufacturing."
—Quality manager at a U.S. manufacturer of automotive gearboxes

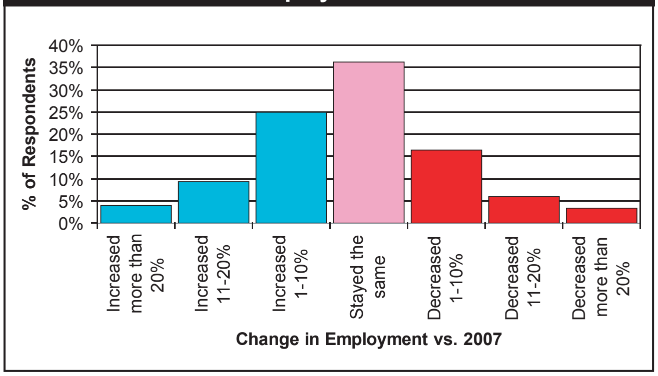
"Increasing currency U.S. Dollar against new Turkish Lira and Euro against new Turkish Lira."
—Manufacturing engineer at a Turkish manufacturer of gear pumps

"India."
—Design engineer at a German manufacturer of forged bevel gears

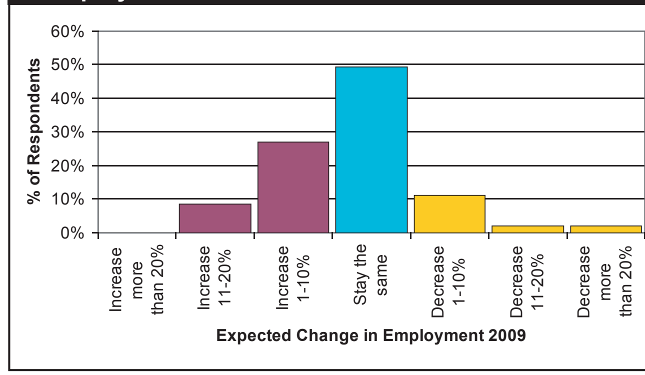
"Internal logistical problems with one major foreign vendor."
—Corporate executive at a U.S. gearbox repair shop

"Lack of capital."
—Manufacturing engineer at a U.S. manufacturer of rack and pinion steering

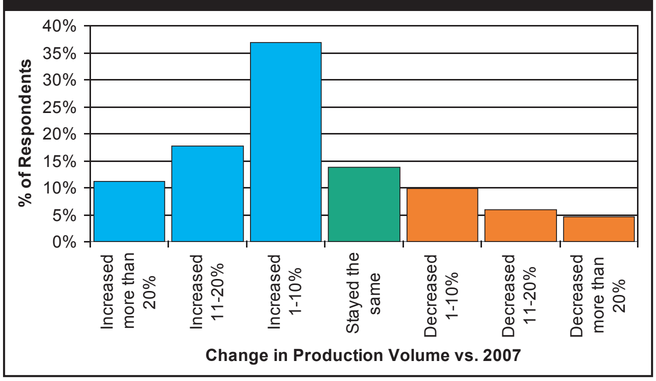
38% of Gear Industry Respondents Work at Locations where Employment Increased in 2008



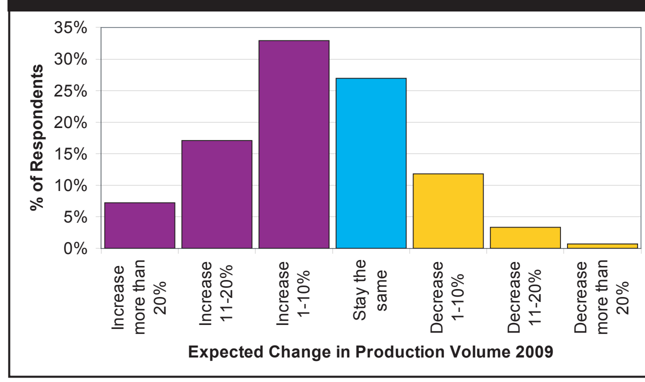
36% of Gear Industry Respondents Expect Employment at their Location to Increase in 2009



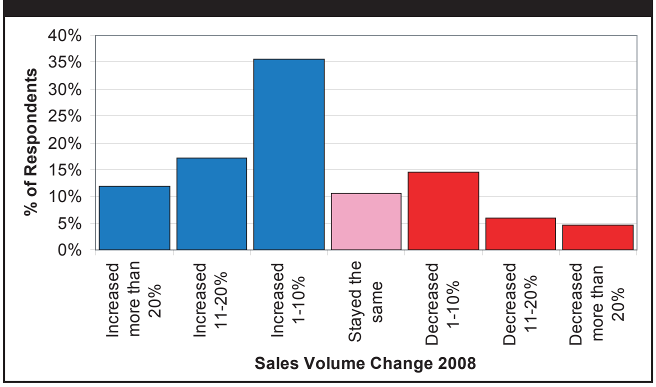
66% Saw Production Volumes Increase in 2008



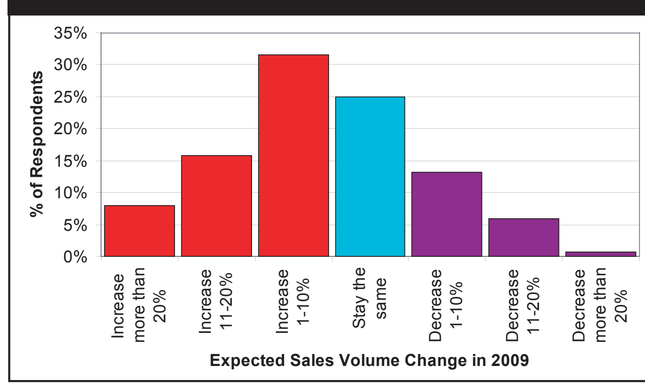
57% Expect Production Volume to Increase in 2009



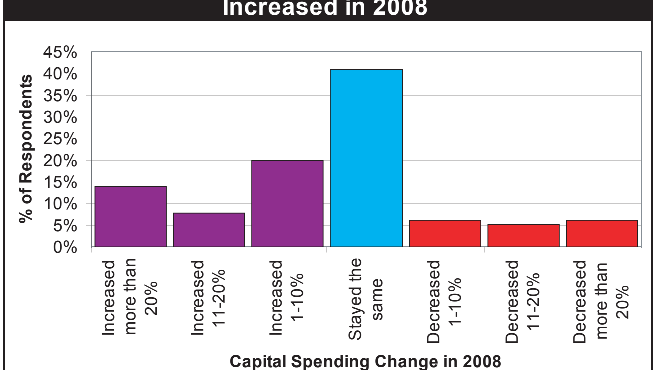
65% Saw Sales Volume Increase in 2008



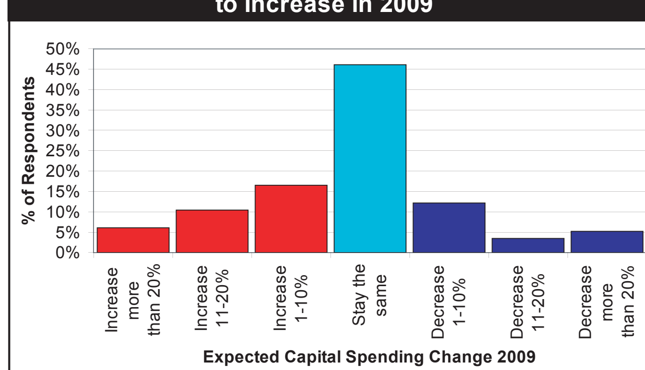
56% Expect Sales Volume to Increase in 2009



42% Work at Locations where Capital Spending Increased in 2008



33% Expect Capital Spending at their Locations to Increase in 2009



What Factors Are Presenting Significant Challenges to Your Business?

"Long development lead times; pre-financing of development costs."
—Corporate executive at a German manufacturer of automotive actuators

"Long lead times and limited supplier capacity for raw material such as large steel forgings, which is driven by worldwide demand in heavy industry. Finding domestic gear suppliers who have the capability to produce high quality, large gears and are willing to do so at low volumes."
—Quality control worker at a U.S. manufacturer of diesel engines

"Low prices from competitors. Rising technology requirement."
—Sales manager at a U.S. powder metal gear manufacturer

"Managed by sales and financial executives that have little to zero understanding in the manufacturing of high quality gears."
—Manufacturing engineer at a U.S. manufacturer of enclosed gear drives

"Mature market with minimal growth."
—Design engineer at a U.S. manufacturer of water treatment drives

"New (and old) management."
—Production worker at a U.S. manufacturer of custom gears.

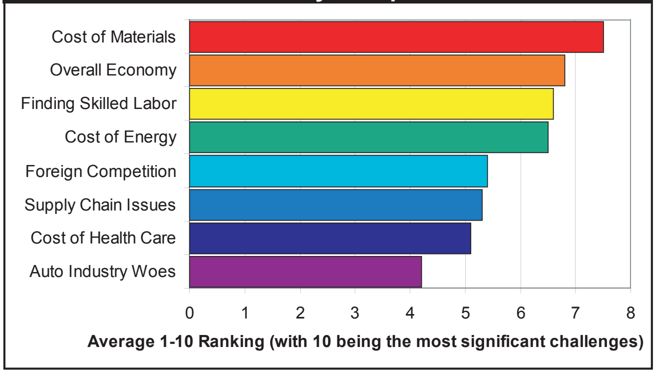
"New competitors."
—Design engineer at a U.S. manufacturer of manual transmissions

"None."
—Corporate executive at a U.S. manufacturer of precision gears

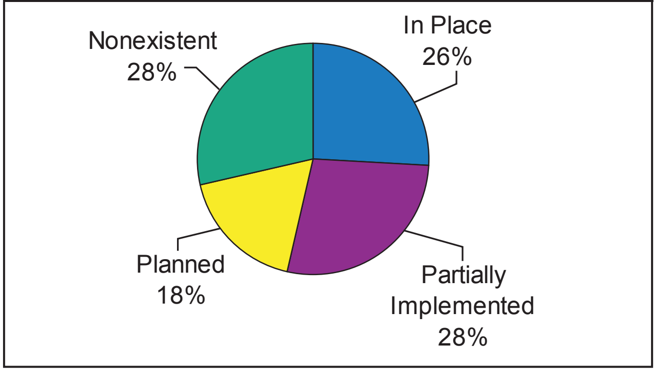
"Not much, but there are chances of some depression in business due to poor condition in USA."
—Manufacturing engineer at an Indian gear rack manufacturer

"Power shortage, lack of availability of skilled labor."
—Manufacturing engineer at an Indian manufacturer of automotive gears

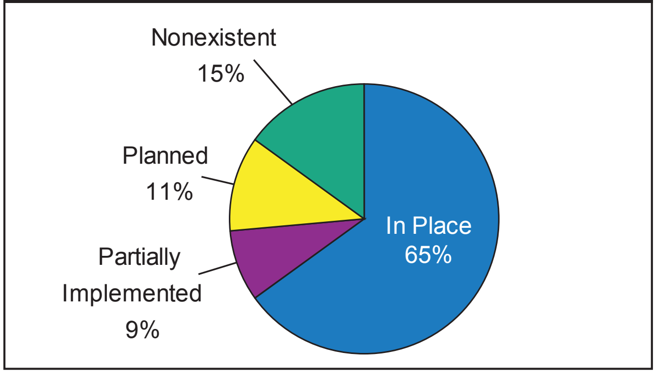
What are the Most Significant Challenges Facing Gear Industry Companies?



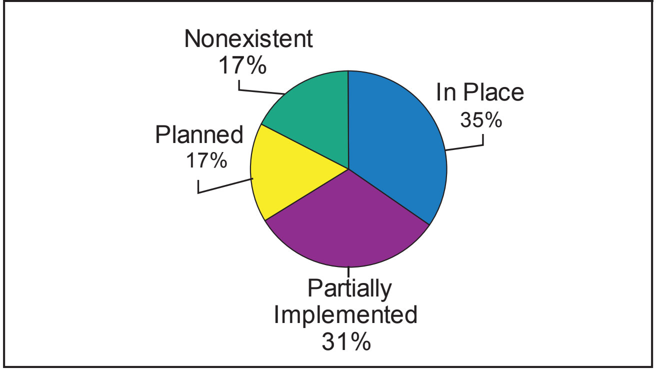
Green or Sustainable Manufacturing Implementation



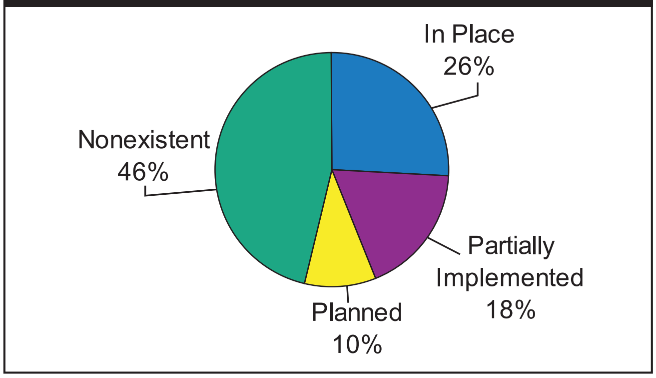
ISO 9000 Implementation



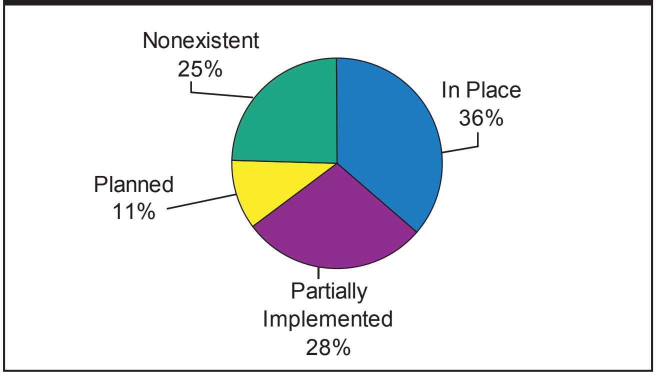
Lean Manufacturing Implementation



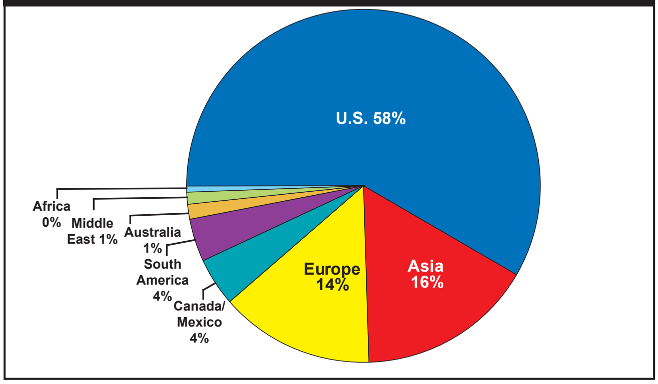
Six Sigma Implementation



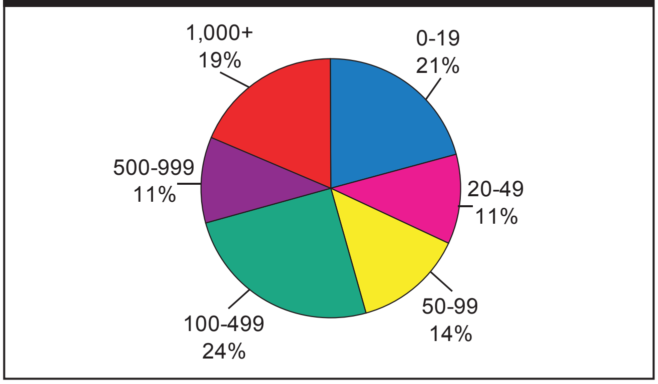
Statistical Process Control (SPC) Implementation



Location of Respondents



Number of Employees



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What Factors Are Presenting Significant Challenges to Your Business?

"Rise in raw material prices has resulted in unstable product cost, and it is a challenge to make the end user understand this unstable product cost."
—Engineer at an Indian manufacturer of spiral bevel gears

"Quality issues especially with regard to raw material inputs. Migration of skilled manpower to other industries."
—Design engineer at an Indian manufacturer of gas turbines

"R&D capital and expansion capital."
—Corporate executive at a U.S. manufacturer of aircraft engines

"Raw material delivery time and costs."
—Design engineer at a U.S. manufacturer of custom equipment

"Raw material for gears (steel grade)."
—Engineer at an Indian engine and transmission assembly plant

"Recession."
—Engineer at a British gearbox manufacturer.

"Rising cost of fuel and materials."
—Design engineer at a job shop in the Philippines

"Rising rates of raw material and non availability of gear making machines and equipment are major factors."
—Engineer at an Indian gear manufacturing job shop

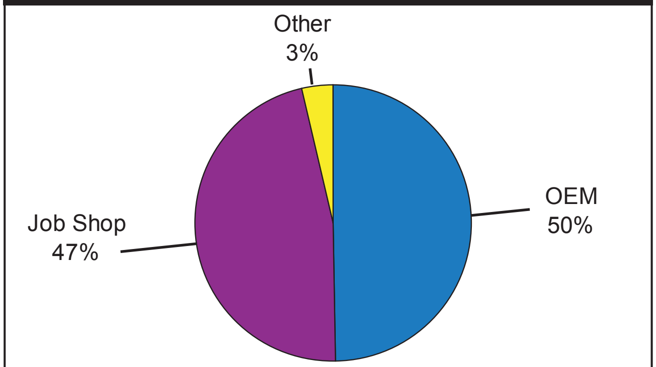
"Skilled labor is the primary issue. Our current issue as of 10/11/08 is the uncertain economy. Current changes in the market are going to affect us. The question is when and how bad will the market affect the gear industry?"
—Sales manager at a U.S. gear manufacturer

"Skilled labor, skilled labor, skilled labor."
—Design engineer at a U.S. manufacturer of aerospace gearing

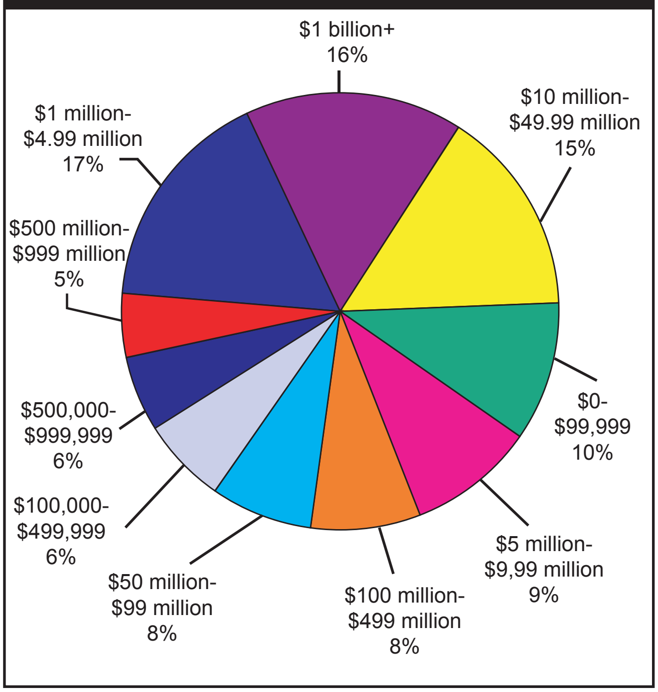
"Starting up engineering in Asia."
—Design engineer at a manufacturer of office equipment in the Netherlands

"Steel cost."
—Technical manager at a marine gear manufacturer in Colombia

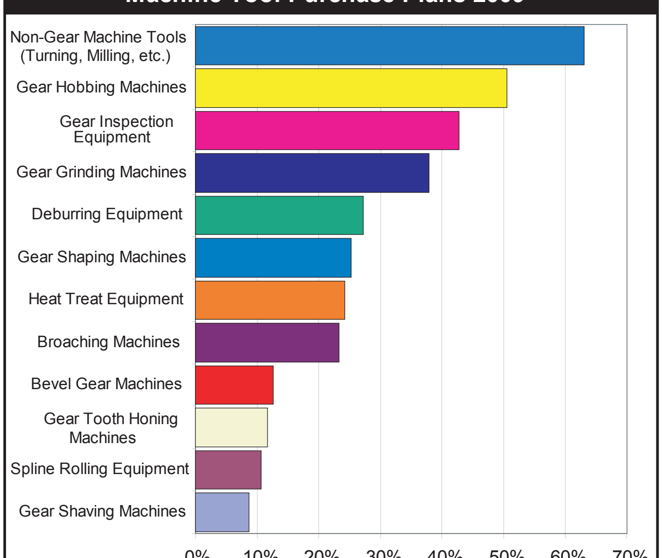
Type of Operation



Annual Sales Volume of Company

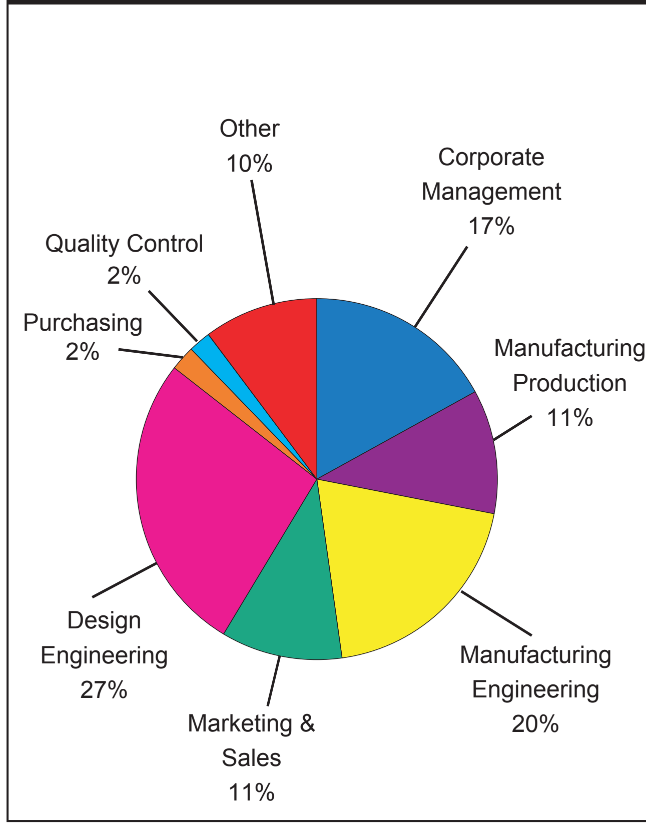


Machine Tool Purchase Plans 2009

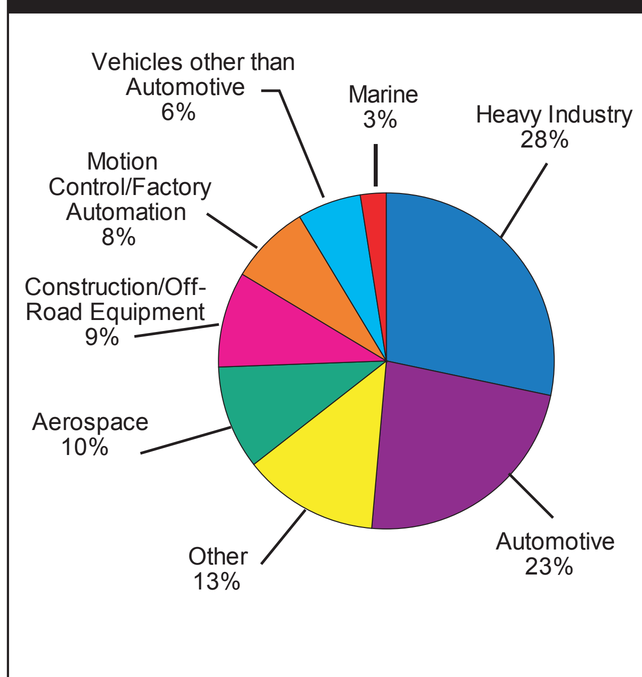


Of Those Planning Capital Purchases in 2009, the % of Respondents Intending to Buy Each Machine Type

Job Title/Function of Respondent



Prime Industry of Respondent



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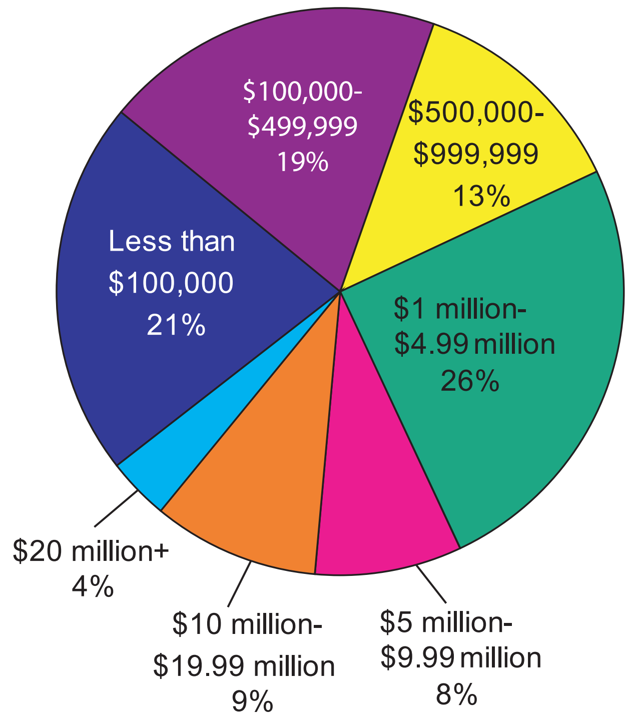
Capital Spending for 2008



What Factors Are Presenting Significant Challenges to Your Business?

"Supply of bearings."
—Manufacturing engineer at a U.S. manufacturer of speed reducers

"The basic challenge is the economic downturn of the industrial climate."
—Corporate executive at an Indian gearbox manufacturer

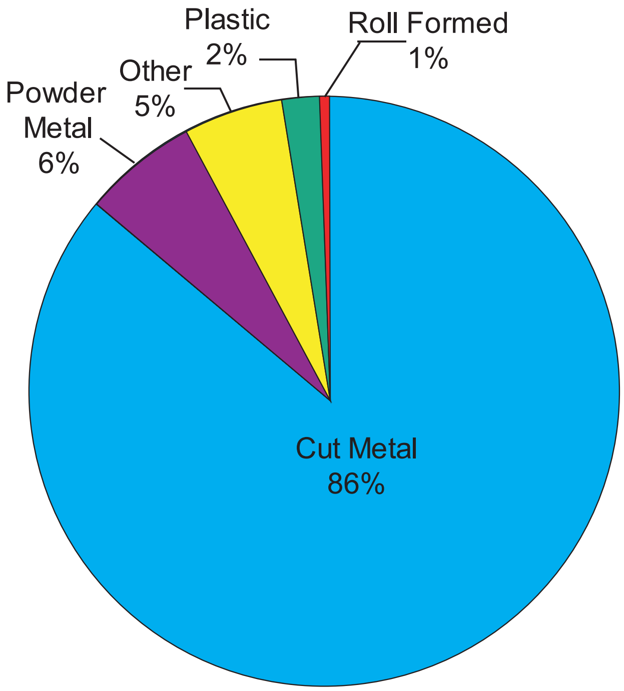
"The world economy."
—Production worker at a Mexican manufacturer of gearboxes for heavy industry

"To meet future fuel economy and emission targets."
—Design engineer at an Indian manufacturer of two- and three-wheeled vehicles

"To minimize lead time for new component development."
—Engineer at an Indian manufacturer of motorcycles

"Unable to get cheaper gears with same performance. It may be done by raw material change, but no one is trying to do so, and we are losing lots of money due to increasing raw material costs day by day."
—Purchasing agent at an Indian manufacturer of commercial vehicles

Primary Method of Manufacture



What Factors Are Presenting Significant Challenges to Your Business?

"We have customers looking for 7-year, long-term contracts and others implementing PO specs that require us to either be compliant with ISO9000 or face a $500.00 in-house inspection charge."
—Corporate executive at a U.S. manufacturer of precision aerospace gears

What are Your Company's Greatest Manufacturing/ Engineering Challenges for 2009?

"Accommodating volume fluctuations and adapting to lower volume work."
—Chief engineer at a U.S. manufacturer of pump shafts

"Aging workforce retiring, taking tribal knowledge away—lack of engineering manpower to capture tribal knowledge in work instructions."
—Engineer at a U.S. manufacturer of speed reducers

"Attempt to increase the efficiency and throughput of our engineering department."
—Design engineer at a U.S. manufacturer of custom equipment

What are Your Company's Greatest Manufacturing/ Engineering Challenges for 2009?

"Being able to stay competitive with pricing yet still being able to finance very expensive CNC gear cutting equipment. Also being able to find adequate skilled labor."
—Corporate executive at a U.S. manufacturer of precision aerospace gears

"Being capable of meeting the scheduled delivery of new mining shovels."
—Manufacturing engineer at a U.S. manufacturer of mining equipment

"Cash Flow."
—Corporate executive at a U.S. gear manufacturing job shop

"Competition."
—Design engineer at a machine shop in the Philippines

"Cost competitiveness without affecting the quality."
—Purchasing agent at an Indian manufacturer of commercial vehicles

"Cost pressures. Technology upgradation."
—Purchasing agent at an Indian manufacturer of machinery

"Cutting tools/hob availability in the USA. Rolled ring forgers with rolling mills large enough to forge ring gears with sizes greater than 63 inches OD."
—Purchasing agent at a U.S. gearbox assembly plant

"Defects below 100 ppm."
—Design engineer at an Indian manufacturer of two- and three-wheel vehicles

"Defining profitable markets with short-term business opportunities."
—Corporate executive at a German manufacturer of automotive actuators

"Developing engines with less fuel consumption."
—Design engineer at a U.S. manufacturer of diesel engines

"Energy costs."
—Maintenance manager at a Croatian manufacturer of paper products

"Engineering for lower cost and fewer parts, manufacturing and assembly in Asia. And also to start up engineering in Asia."
—Design engineer for a manufacturer of office equipment in the Netherlands

continued on page 71


“So, essentially I need to be


a little bit more jolly then?”
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Induction Hardening of Gears and Critical Components Part II

Dr.Valery Rudnev

Management Summary
Induction hardening is a heat treating technique that can be used to selectively harden portions of a gear, such as the flanks, roots and tips of teeth, providing improved hardness, wear resistance and contact fatigue strength without affecting the metallurgy of the core and other parts of the component that don't require change. This article provides an overview of the process and special considerations for heat treating gears. Part I, which was published in the September/October 2008 issue, covered gear materials, desired microstructure, coil design and tooth-by-tooth induction hardening (For the online version, visit *http://www.geartechnology.com/issues/0908/)*. Part II covers spin hardening and various heating concepts used with it.

Gear Spin Hardening (Using Encircling Inductors)

Spin hardening is the most popular induction gear hardening approach, and it is particularly appropriate for gears having fine- and medium-size teeth (Figure 8). Gears are rotated during heating to ensure an even distribution of energy. Single-turn or multi-turn inductors that encircle the whole gear can be used (Ref. 1).

When applying encircling coils, there are five parameters that play major roles in obtaining the required hardness pattern: frequency, power, cycle time, coil geometry and quenching conditions. Figure 9 illustrates a diversity of induction hardening patterns that were obtained with variations in heat time, frequency and power (Ref. 1).

As a rule, when it is necessary to harden only the tooth tips, a higher frequency and high power density should be applied

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Figure 8—Examples of gears that use spin-hardening techniques.

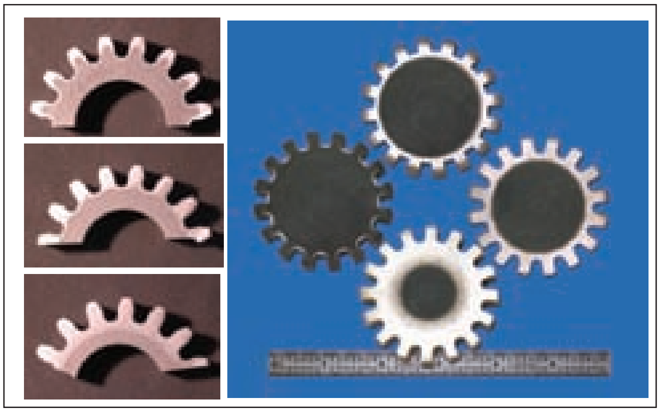

Figure 9—Diversity of hardness patterns obtained with induction spin hardening.

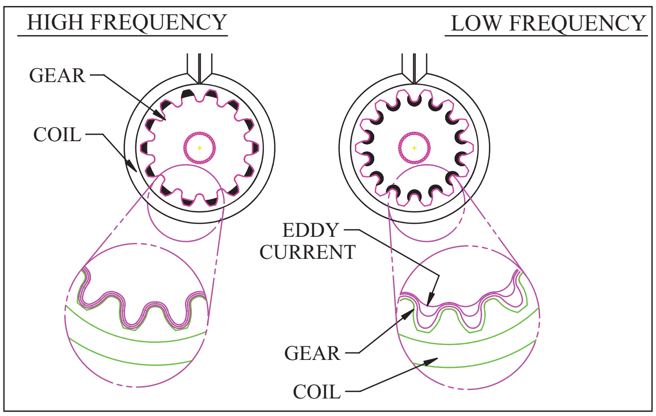

Figure 10—Frequency influence on eddy current flow within the gear when using an encircling inductor.

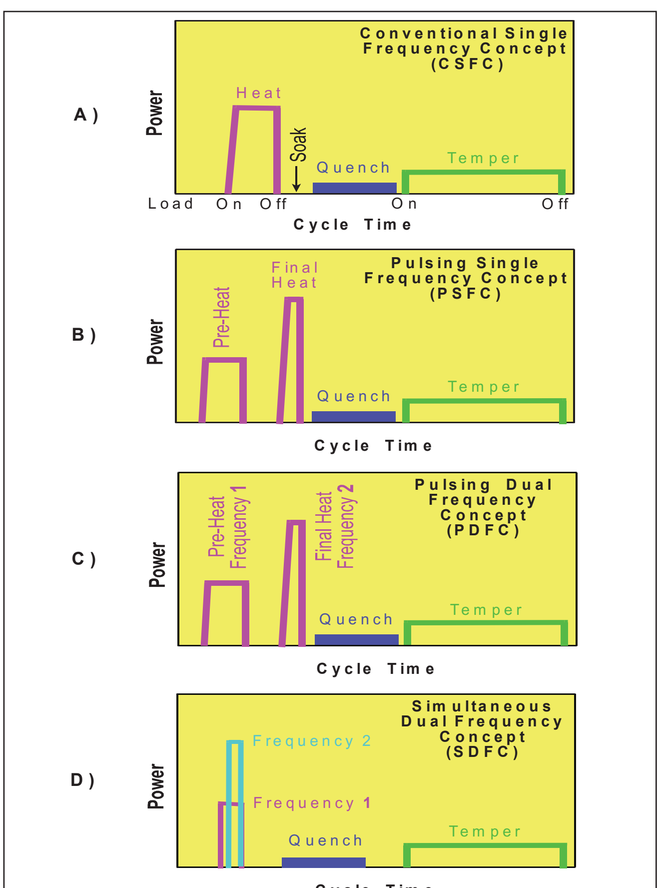

Figure 11—Induction gear hardening concepts: A) CSFC; B) PSFC; C) PDFC; D) SDFC.

in combination with short heat time (Fig. 10, left). To harden the tooth roots, use a lower frequency (Fig. 10, right). A high power density generally gives a shallow pattern, while a low power density will produce a deep pattern with wide transition zones. Hardness pattern uniformity and repeatability depend strongly on the relative positions of gear and induction coil and the ability to maintain the gear concentric to the coil.

Gear Quenching

Induction hardening is a two-step process: heating and quenching. Both stages are important. There are three ways to quench gears in spin hardening applications (Ref. 1):

- Submerge the gear in a quench tank. This technique is particularly applicable for large gears.
- Quench "in place" using an integrated spray quench. Small- and medium-size gears are usually quenched using this technique.
- Use a separate, concentric spray-quench block (quenchring) located below the inductor.

Note that the widely published, classical cooling curves that represent the three stages of quenching—vapor blanket, boiling and convection heat transfer—cannot be directly applied in spray quenching (Ref. 7). Due to the nature of spray quenching, the first two stages are greatly suppressed. At the same time, cooling during the convection stage is more severe.

Tooth geometry and rotation speed are other factors that have a pronounced effect on quench flow and cooling severity during gear quenching.

It is also important to avoid both eccentricity of the inductor and quench system relative to the gear and gear wobbling. Even with gear rotation, gear wobbling will cause a specific part of the gear to be hotter during heating, because regardless of rotation it will always be closer to the coil. Besides non-uniform heating, wobbling also causes uneven quenching, leading to additional hardness non-uniformity and gear shape distortion.

It has been reported that more favorable compressive stresses within the tooth root were achieved with the gear spin hardening technique than with the "tooth-by-tooth" or "gap-by-gap" approaches.

Heating Concepts

There are four popular heating concepts used for the induction spin hardening of gears that employ encircling-type coils (Fig. 11): the conventional single-frequency (CSFC), pulsing single-frequency (PSFC), pulsing dual-frequency (PDFC) and simultaneous dual-frequency (SDFC) concepts (Refs. 1-4, 8). All four modes can apply either a single-shot or scanning approach. The choice of heating concept depends upon the application and equipment cost.

Conventional single-frequency concept (CSFC). The conventional single frequency concept (Fig. 11a) is typically used for hardening gears with medium and small teeth. Often with this technique, the tips of teeth are through hardened. As an example, Figure 12 shows an induction gear-hardening machine that applies this concept. The gear being heat treated in this application is an automotive transmission component with helical teeth on the inside diameter (ID) and large teeth on the outside diameter (OD) for a parking brake. Both the inside diameter and outside diameter require hardening (Figure 12). The hardening of the inside diameter gear teeth requires a higher frequency than the outside diameter. Therefore, a frequency of 10 kHz was chosen for OD heating and 200 kHz was chosen for ID heating. After heat is off, quenchant is applied to the hot gear in place; that is, no repositioning is required. This practically instantaneous quench provides a consistent metallurgical response. Quenching reduces the gear temperature to the quenchant temperature or temperature suitable for gear handling. Gears are conveyed to the machine, where a cam-operated robot then transfers them to the hardening station.

Parts are monitored at each station and accepted or rejected based on all the major process factors that affect gear quality. This includes energy input into the part, quench flow rate, temperature, quench pressure and heat time. An advanced control/monitoring system verifies all machine settings to provide confidence in the quality of processing for each individual gear. Precise control of the hardening operations and optimized coil design minimize gear distortion and provide the desirable residual stresses in the finished gear. The hardened gear is inspected and moved to the next operation.

Although the conventional single-frequency concept (CSFC) is primarily suitable for small and medium size gears, there are cases when this concept can be successfully used for large gears as well. As an example, Figure 13 shows an induction machine for hardening large gears. A multi-turn encircling inductor is used for hardening gears with a major diameter of 27.6" (701 mm), root diameter of 24.3" (617 mm) and thickness of 3.125" (79 mm). In this case, it was in the customer's best interest to harden and temper gears using the same coil and power supply. In other cases it might not be the best solution.

Quite often, in order to prevent problems such as pitting, spalling, tooth fatigue and endurance, it is required to harden the contour of the gear (contour hardening). In some cases, this can be a difficult task for CSFC due to the difference in current density (heat source) distribution and heat transfer conditions within a gear tooth. Two main factors complicate the task of obtaining the contour hardness profile using the CSFC approach. The first factor is that with encircling-type coils, the root area does not have as good of an electromagnetic coupling with the inductor compared to the coupling at the gear tip. Therefore, it is more difficult to induce energy into the gear root. Second, there is a significant heat sink located under the gear root (below the base circle, Figure 10).

Pulsing single-frequency concept (PSFC). In order to overcome these difficulties and to be able to meet customer specifications, the pulsing single-frequency concept (PSFC) was developed (Figure 11b). In many cases, PSFC allows the user to avoid the shortcomings of CSFC and obtain a

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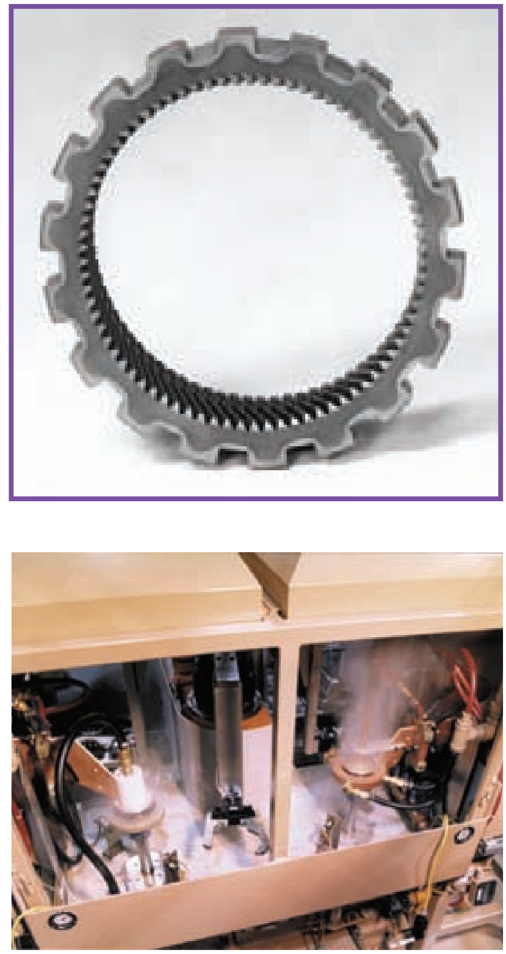

Figure 12—Induction hardening of an automotive transmission component with smaller, helical teeth on the ID and large teeth on the OD.

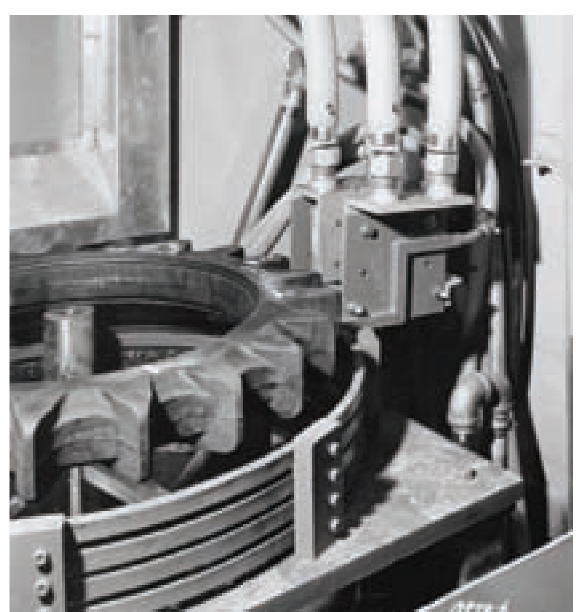

Figure 13—A multi-turn encircling inductor for hardening large gears.

pattern close to a contour hardening profile. Pulsing provides the desirable heat flow towards the root of the gear without noticeable overheating of the tooth tip. A well-defined, crisp, hardened profile that follows the gear contour can be obtained using high power density at the final heating stage.

An induction machine can be designed to provide gear contour heat treatment (including preheating, final heating, quenching and tempering) with the same coil using one high-frequency power supply. Figure 11b illustrates the process cycle with moderate power preheat, soaking stage, short high-power final heat and quench followed by low-power heat for temper.

Preheating ensures a reasonable heated depth at the roots of the gear, enabling the attainment of the desired

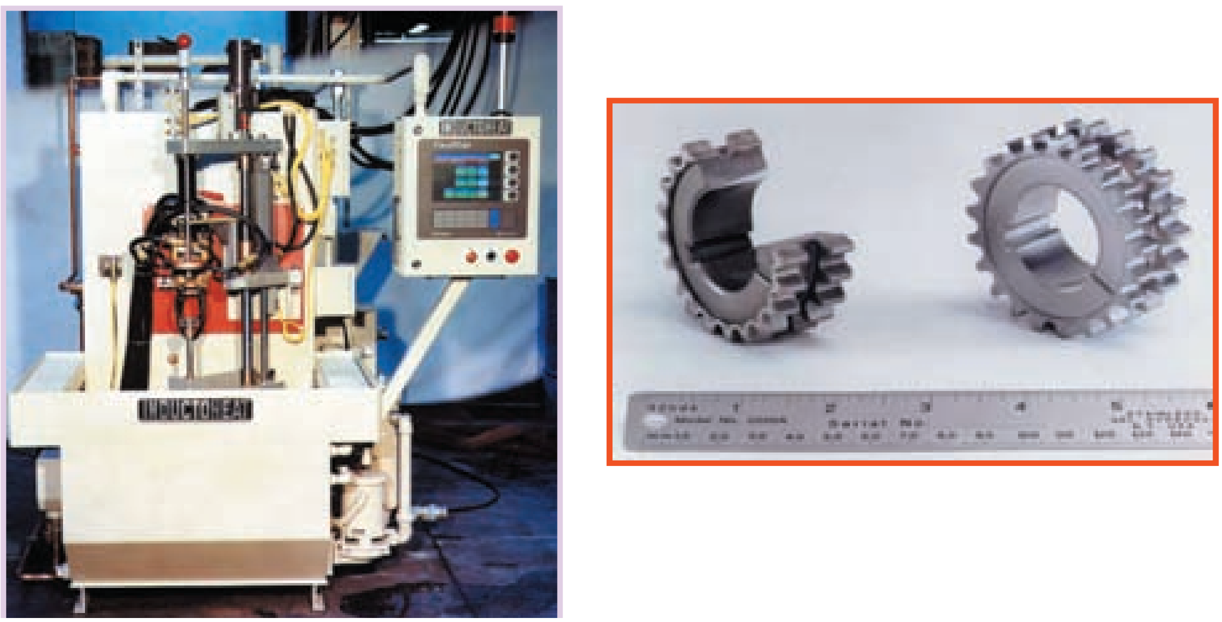

Figure 14—A unitized induction system capable of providing both CSFC and PSFC gear hardening.

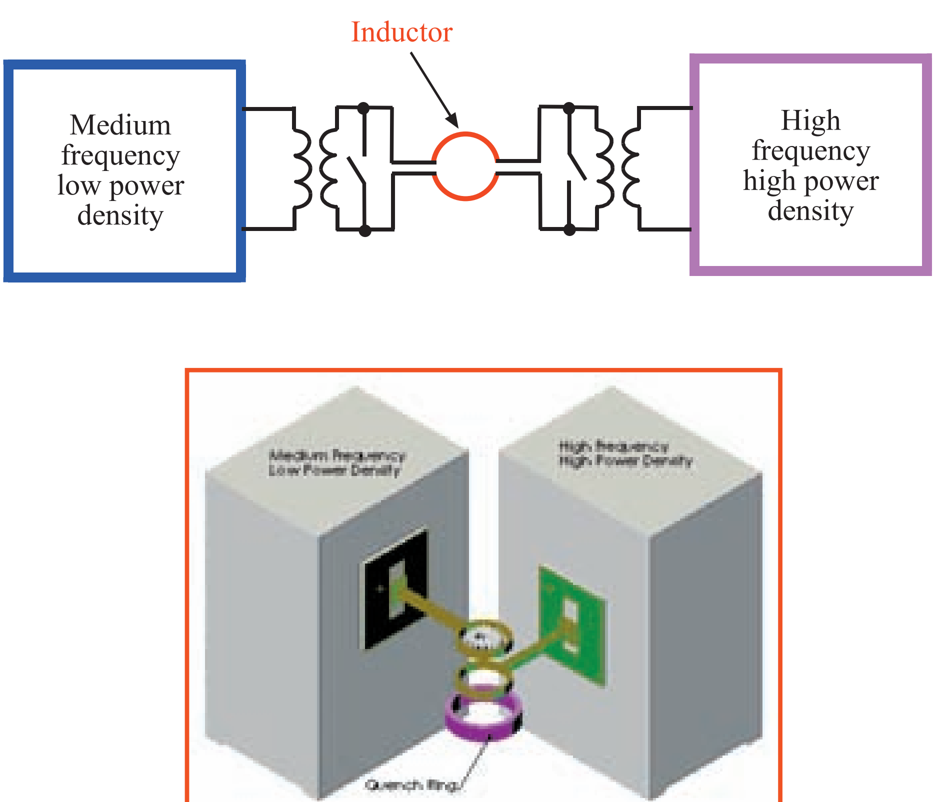

Figure 15—Dual frequency induction hardening utilizing a single inductor (top) vs. two inductors (bottom).

metallurgical result and decreasing the distortion in some cases. Preheat times are typically from several seconds to a minute, depending on the size and shape of the gear teeth. Obviously, preheating reduces the amount of energy required in the final heat and decreases thermal stresses as well.

After preheating, there might be a soak time ranging from two to 10 seconds to achieve a more uniform temperature distribution across the teeth of the gear. Depending upon application, preheat can consist of several stages (preheat pulses). Final heat times can range from less than one second to several seconds.

As a general rule, for both CSFC and PSFC techniques, the higher frequency is used for finer-pitch gears, which typically require a smaller case depth. Figure 14 shows a double sprocket, hardened utilizing a unitized induction hardening system capable of providing both CSFC and PSFC gear hardening.

Pulsing dual-frequency concept (*PDFC*). A third concept is the pulsing dual-frequency concept (PDFC). The idea of using two different frequencies to produce the desired contour pattern has been around since the late 1950s. This concept was primarily developed to obtain a contour hardening profile for helical and straight spur gears.

According to PDFC (Fig. 11c), the gear is preheated within an induction coil to a temperature determined by the process features. This temperature is usually 350ºC–100ºC below the critical temperature Ac1. Preheat temperature depends upon the type and size of the gear, tooth shape, prior microstructure, required hardness pattern, acceptable distortion and the available power source. The higher the preheat temperature, the lower the power required for the final heat. However, high preheat temperatures can also result in increased distortion.

As in previous concepts, PDFC can be accomplished using a single-shot mode or scanning mode. The scanning mode is applied for wider gears.

Typically, preheating is accomplished by using medium frequency (3–10 kHz). Depending on the type of gear, its size and material, a high frequency (30–450 kHz) and high power density are applied during the final heat stage. For the final heating stage, the selected frequency allows the current to penetrate only to the desired depth.

Depending upon the application, single-coil design or two-coil design arrangements can be used (Fig. 15).

Quenching completes the hardening process and brings the gear down to ambient temperature. In some cases, dual-frequency machines produce parts with lower distortion and have more favorable distribution of residual stresses compared to other techniques.

As mentioned above, when applying high frequency (i.e. 70 kHz and higher), it is important to pay special attention to gears with sharp corners. Due to the electromagnetic edge effect, high frequency has a tendency to overheat sharp edges and corners. This could result in weakened teeth due to decarburization, oxidation, grain growth and sometimes

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Figure 16—A contour hardened gear.

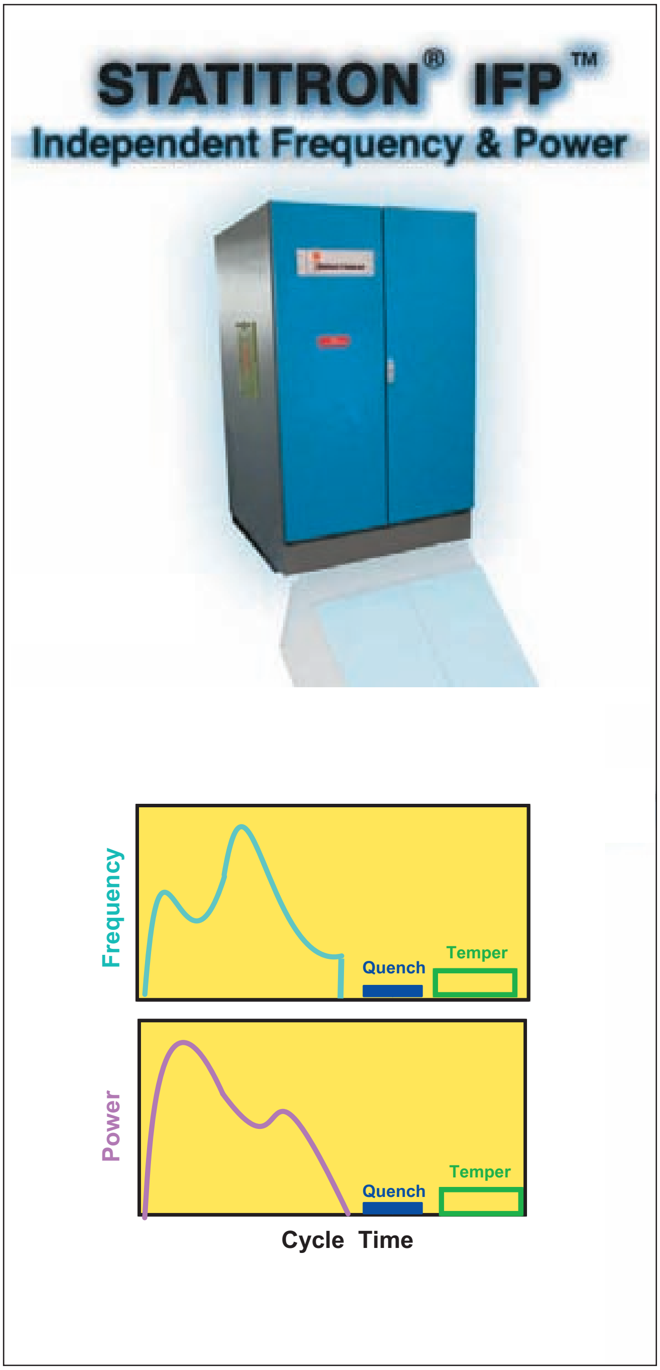

Figure 17—Independent frequency and power control.

grain boundary liquidation or even local melting of sharp edges. Therefore, in order to improve the life of a gear, the sharp edges and corners should be broken and generously chamfered. A four-inch spur gear induction contour hardened using the PDFC approach is shown on Figure 16. As one can see from Figure 16, the hardness pattern of an induction-hardened gear is quite similar to the hardness pattern obtained using carburizing. At the same time, the induction contour hardening process is accomplished in a much shorter time, with much simpler processing procedures using environmentally friendly processes.

Simultaneous dual-frequency concept (SDFC). The main drawback of the PDFC process is its relative complexity and high cost, since it is necessary to have two different power supplies and a fast lift-rotating devise. In some cases, it is possible to use a simultaneous dual-frequency concept (Figure 11d). This concept implies the use of a simultaneous dual-frequency power supply instead of two single-frequency inverters; however, the cost of these variable frequency devices is still quite high.

Independent frequency and power concept (IFPC). The ability to independently change the frequency and power of an induction heating system (Fig. 17, left) represents the long-awaited dream of commercial induction heat treaters, since that type of setup would provide the greatest process flexibility. This was the reason for the introduction of the STATITRON IFP (Independent Control of Frequency and Power) inverter, which allows for independent change of frequency in a 5–40 kHz range and power in a range of 10–75 kW in a single-module system (Fig. 17, right) (Ref. 8). This concept (IFPC) substantially expands heat treat equipment capabilities for processing parts by programming power and/or frequency changes on the fly, maximizing heating efficiency while heating different part sizes and/or optimizing performances of both hardening and tempering while utilizing the same power supply more development is taking plase to increase the power output of IFPC inverters.

Computer Modeling

Computer modeling is a major factor in the successful design of induction heating systems, providing the ability to predict how different factors and process parameters may influence transitional and final heat treating conditions (Refs. 1, 2). Modeling delineates what must be accomplished in the design of the system and/or process recipe to improve the effectiveness of the heat treatment and guarantee that the required results are obtained.

By its very nature, induction heating is characterized by a close relationship to the physical properties of the metal being heated (Ref. 1). Some physical properties strongly depend upon the temperature of the metal and its microstructure, while others are functions of magnetic field intensity and frequency as well. During the heating cycle, significant changes occur in such important physical properties as thermal conductivity,

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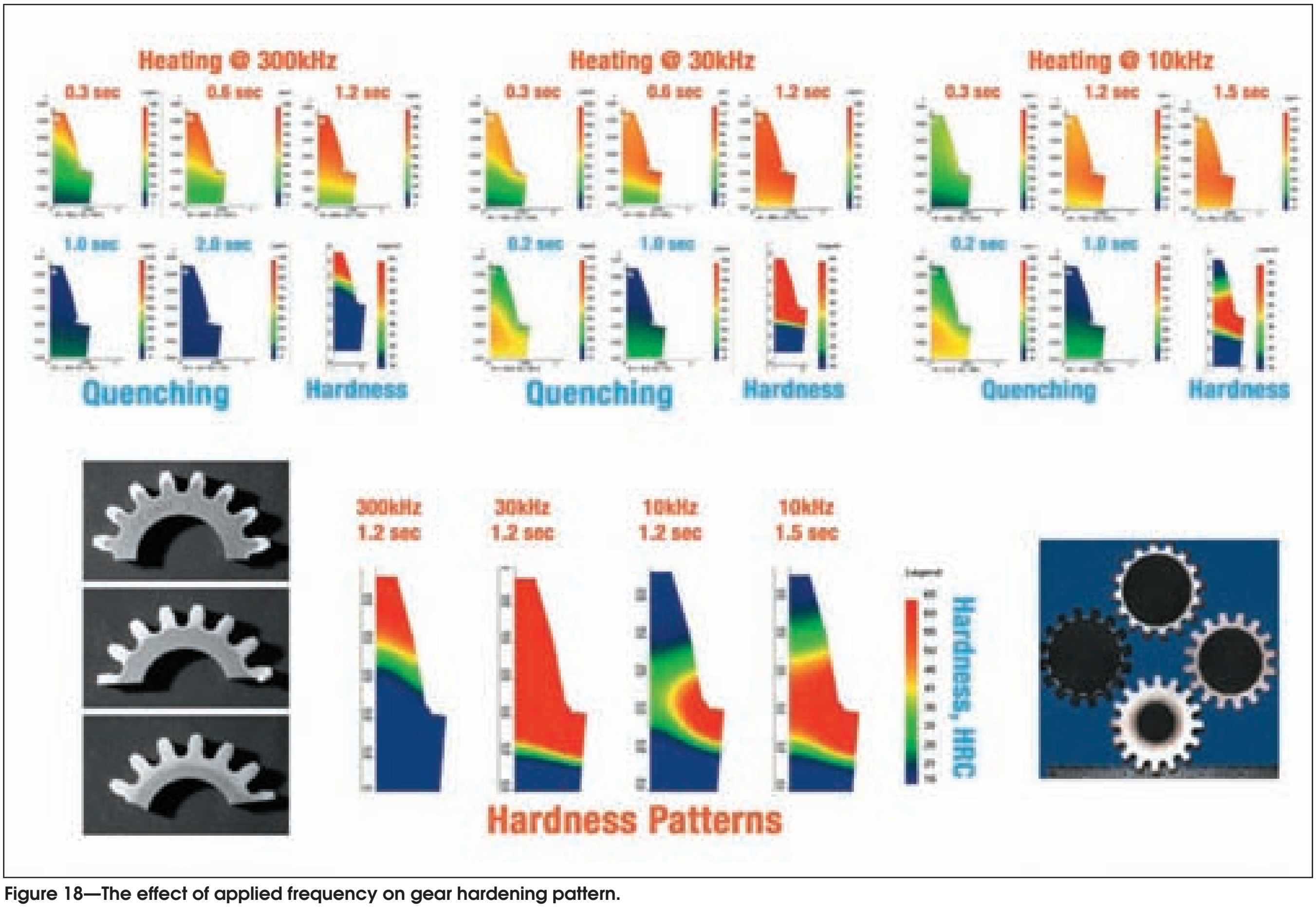

Figure 18—The effect of applied frequency on gear hardening pattern.

specific heat, magnetic permeability and electrical resistivity. Variations in magnetic permeability and electrical resistivity can result in an increase in the current penetration (heat source) depth of up to 16 times during the heating cycle from ambient to austenitization temperatures. Such a dramatic change leads to a considerable three-dimensional redistribution of the heat induced within the gear during the heating cycle (Ref. 1). This explains why variation in physical properties should be carefully taken into consideration when computer modeling an induction gear hardening process.

As an example, Figure 18 (top) shows the dynamics of temperature distribution during heating and quenching of a fine-pitch gear using different frequencies (300 kHz, 30 kHz and 10 kHz) (Ref. 2).

High frequency. As expected, when an RF frequency of 300 kHz is applied, an eddy current induced in the gear follows the contour of the gear (Fig. 10, left). Since the highest concentration of current density will be in the tip of the tooth, there will be a power surplus in the tip compared with the root (Fig. 18, top-left). Also taking into account that the tip of the tooth has the minimum amount of metal to be heated, compared with the root, the tip will experience the most intensive temperature rise during the heating cycle. In addition, from the thermal perspective, the amount of metal beneath the gear tooth root represents a much larger heat sink compared with that beneath the tooth tip.

Another factor that contributes to more intensive heating of the tooth tip is better electromagnetic coupling—the so-called proximity effect—between the inductor and tooth tip, vs. its root. Higher frequency has a tendency to make a proximity effect more pronounced (Ref. 1).

A combination of these factors provides rapid austenitization of the tooth tip, which, upon quenching, produces a martensitic layer on the tip.

Low frequency. When a low frequency, such as 10 kHz, is applied for heating fine-tooth gears, the eddy current flow and temperature distribution in the gear tooth will be quite different (Fig. 18, right).

A frequency reduction from 300 kHz to 10 kHz noticeably increases the eddy current penetration depth in the steel—from 1 mm to 5.4 mm—particularly at temperatures above the Curie temperature. In a fine-tooth gear, such an increase in penetration depth results in a current cancellation phenomenon in the tooth tip and pitch line area. This makes it much "easier" for induced current to take a "short" path, following the base circle or root line of the gear instead of the tooth profile (Fig. 10, right). The result is more intensive heating of the root fillet area compared with the tip of the tooth (Fig. 18, top-right), and the development there of martensite upon quenching.

Hardening patterns. An example of how the gear hardening pattern varies with applied frequency is shown in Figure 18 (bottom). The results of modeling support the experimentally obtained hardening patterns shown in Figure 9 and confirm the previous explanation of the physics of the electromagnetic-thermal processes that occur during induction spin hardening of gears using different frequencies.

It is important to remember that the terms "high frequency" and "low frequency" are not absolute. For example, depending upon gear geometry, a frequency of 10 kHz might be considered low when heating fine-tooth gears, but would be considered as high frequency when hardening large gears having coarse teeth. Similarly, a frequency of 300 kHz could act as a very low frequency for certain gear tooth geometries, and be able to harden only the root of the tooth and unable to properly harden its tip.

Conclusions

Induction hardening of gears includes a number of process concepts that can be applied depending on the part geometry and hardness profile required. Spin hardening techniques, wherein the coil encircles the part, are most often used for small and medium size gears.

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An Investigation of the Influence of Shaft Misalignment on Bending Stresses of Helical Gears with Lead Crown

M.A. Hotait, D. Talbot and A. Kahraman

Management Summary

In this study, the combined influence of shaft misalignments and gear lead crown on load distribution and tooth bending stresses is investigated. Upon conclusion, the experimental results are correlated with predictions of a gear load distribution model, and recommendations are provided for optimal lead crown in a given misalignment condition.

Introduction

Gears are highly engineered machine elements that must be designed not only to meet the torque, speed, life and noise requirements under nominal conditions, but also to compensate for adverse effects due to manufacturing errors, variations and elastic deformations of the support structures. In addition to the elastic deformation of the support and structures, manufacturing errors in the gears, shafts and housing must also be considered in the design. Assuming that perfectly aligned shafts support a gear pair in which the shafts, bearings and the housing are all rigid might lead to severe wear and noise problems.

A reasonably accurate gear pair, with a limited amount of tooth surface manufacturing errors and operated under ideal (no error and no deflection) support conditions, can be expected to exhibit a good load distribution along its face width. If the shafts are mounted on bearings with position errors, or the shafts, bearings and housing deflect under load, then the rotational axes of the gears will no longer be parallel to each other. These conditions cause a mismatch of meshing teeth, resulting in a non-uniform load distribution along the face width, with perhaps very little or no load on one side and a larger edge load on the other. Such poor load distribution conditions might result in contact and tooth bending stresses at the overloaded side that are higher than the allowable (designed) limits, thus triggering premature bending or contact fatigue failures. Such poor load distribution also accelerates the rate of wear at the gear tooth surfaces.

One widely accepted, practical solution to edge loading due to misalignments caused by manufacturing errors and shaft/bearing deflections is to machine the gear tooth profiles so that additional material is removed from the edges to form a convex tooth surface in the face width (lead) direction, or lead crown. A gear pair having a certain amount of lead crown when there are no shaft misalignments would reduce the stresses near the edges of the tooth, while increasing the stresses at the tooth center. In case of a certain amount of shaft misalignment, the load distribution is modified so that the excessive edge loading due to shaft misalignments is prevented.

Predicting gear load distribution and its resultant stresses has been a major research topic. A number of theoretical and computational studies (Refs. 1–10) were carried out, considering both static and dynamic loading conditions. These studies mostly used finite element or boundary element models of varying complexity and size to predict load distribution and gear stresses at the tooth root region. Some of these models allowed misalignments to gears in a certain direction to predict the resultant changes in the load distributions (Refs. 11–15). For instance, the study by Wagaj and Kahraman (Ref. 15) has shown that the tooth profile and lead modifications influence the contact pattern, load distribution and contact stresses significantly, and that the amount and shape of the tooth modifications requirements vary with misalignments.

Most of the computational studies listed above focused on the modification of the tooth in the involute direction to reduce gear transmission error, a common gear noise excitation. Therefore, the primary effort in validation of these models was applied to their predictions of static transmission error; experimental studies by Kahraman and Blankenship (Ref. 16) provided such data for validations of these models. They also showed experimentally that gear involute contact ratio and profile modifications can be adjusted to minimize such excitations. The same experiments were also used to relate the dynamic stress factors to gear transmission error (Ref. 17). Other experimental studies, starting with Kubo (Ref. 18), provide experimental root stress data for spur gears under both static and dynamic conditions. Houser (Ref. 19) provided a comprehensive database of measured root strains for gears having tooth spacing errors. Oswald and Townsend (Ref. 20) compared analytical and experimental data for dynamic tooth load and fillet stresses of spur gears. And more recently, Baud and Velex (Ref. 21) published static and dynamic helical gears strain data to validate the root stresses of a finite element model.

A review of previous work reveals that, while influence of the profile modifications is well-studied—especially for spur gears—the number of experimental studies on the influence of the lead crown on helical gear pairs subjected to misalignments is quite limited. There is very little experimental data available on the relationship between the gear lead crown and shaft misalignments. The models cited above lack any validation in terms of their predictions of root stresses under misaligned conditions. In many practical applications, the amount of lead crown is still determined based on the trial-and-error method or past field experiments. It has been reported that applying lead crown increases root stresses in the middle of the gears, while eliminating edge loading. Any excessive amount of lead crown employed to eliminate the negative effects of shaft misalignments has the potential to increase the stresses at the center of the teeth beyond the stress levels caused by edge loading. Therefore, the relationship between the lead crown and shaft misalignment under varying torque values must be investigated, both experimentally and theoretically.

Investigation of the impact of shaft misalignments on the root stresses of gears with or without lead crown is the main focus of this study. Its first objective is to develop an experimental test program that will yield a root strain database of tightly controlled experiments for helical gears with misalignments and lead crown, thus quantifying the relationship between shaft mounting errors and gear lead crown modifications in terms of gear bending stresses. Wide ranges of shaft misalignments and magnitudes of lead crown will be considered in these experiments, as well as a wide range of transmitted torque. An existing gear contact model will be used to simulate these test conditions in order to describe the behavior observed in the experiments. The accuracy of the root strain prediction of the contact model will then be assessed, and design guidelines on lead crowning of helical gears with misalignment will be provided.

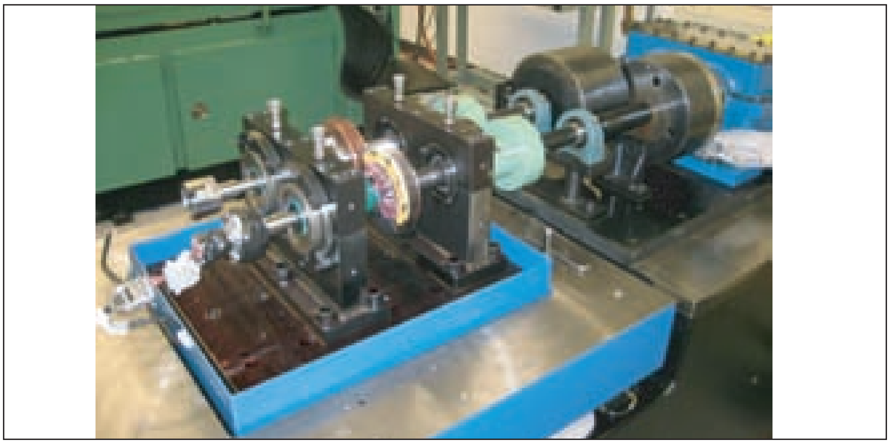


Figure 1—Gear test machine used in the study.

Table 1—Basic design parameters of the test gear pairs (all the dimensions are in mm unless specified).

Parameter	Drive Gear	Driven Gear
Number of Teeth	62	62
Normal Module	2.04	
Normal Pressure Angle (deg)	16	
Helix Angle (deg)	32.5	
Pitch Diameter	150.0	
Base Diameter	142.02	
Major Diameter	153.74	153.24
Minor Diameter	142.02	
Circular Tooth Thickness	2.46	
Lead Crown (λ)	Variable	
Involute Crown	0.012	0

Experimental Setup

A power circulatory-type gear dynamics test machine—shown in Figure 1—was used for the experiments. In this arrangement, the test gear pair and a reaction gear pair are connected by flexible shafts and elastomer couplings, ensuring full isolation of the reaction gear pair from the test side. The test pair—a driving gear and a driven gear—were mounted on parallel shafts, which are supported by a pair of spherical roller bearings. The bearings are housed by bearing caps that are held by rigid bearing pedestals. The bearing caps of varying eccentricity can be clocked in any direction to obtain shaft misalignments of various magnitudes in any desired direction. A split coupling is used to manually hang calibrated weights through a torque arm to achieve constant gear torque values up to 500 Nm, or a mesh force up to 6,600 N. Test gears and support bearings were jet-lubricated to minimize adverse frictional effects.

Four helical test gears (named *g*1, *g*2, *g*3, and *g*4) were used in these experiments. Table 1 lists basic design parameters of the test gears. Gear g1 of the right-hand is the strain-gauged driving gear, while left-handed gears *g*2, *g*3 and *g*4 form the driven gears. Driving gear *g*1 has a nominal tooth profile crown modification of 12 μm in the involute direction but has no lead modification. The driven gears have no

continued

Figure 2—Test gears used in this study.

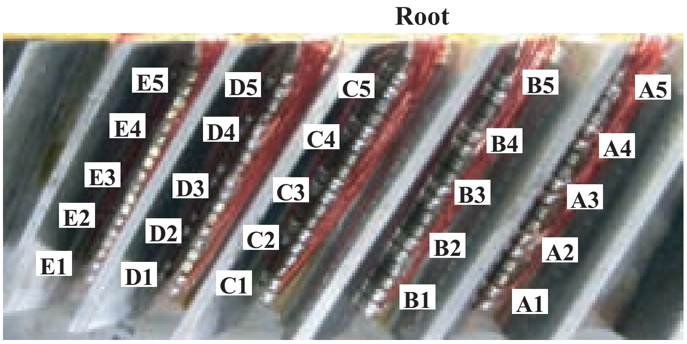
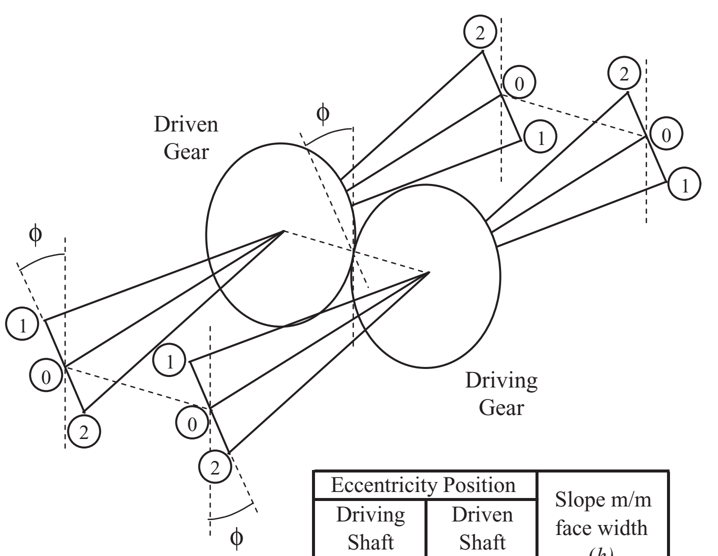


Figure 3—A view of the strain gauges on gear *g*1.



Eccentricity Position		Slope m/m face width (h)
Driving Shaft Bearing	Driven Shaft Bearing	
1	2	-0.002
0	2	-0.001
0	0	0
0	1	0.001
2	1	0.002

Figure 4—Definition of shaft misalignments.

modifications in the involute direction, yet they have varying amounts of circular lead crown λ (λ = 0) for gear $g2$, λ = 12 μm for gear $g3$ and λ = 25 μm for gear $g4$). These desired modification amounts were obtained by precise grinding of the gear profiles after case hardening. This way, gear pairs $g1$–$g2$, $g1$–$g3$ and $g1$–$g4$ have the same total involute crown value of 12 μm, and three levels of total lead crown (λ = 0, 12 and 25 μm). The test gears are shown in Figure 2.

Five consecutive teeth of gear g1 were strain-gauged as shown in Figure 3. At the root of each of these five teeth (A through E), five gauges were placed and equally spaced along the face width direction, bringing the total number of gauges to 25. Gauge strips (measurement group EA-06-031PJ-120) were used in order to ensure precise alignment of the gauges, slightly below the start of active profile, at a radius of 72.22 mm (roll angle). Only five of the gauges on each strip were activated, resulting in gauges at 2.7, 6.1, 9.6, 13.0 and 16.5 mm along the face width, measured from one edge.

With the concentric bearing caps (the outside diameter and the bore of the caps are concentric), the test gear shafts are positioned parallel to each other, without misalignments. That is because caps of a certain bore eccentricity forces the shafts to be misaligned at the bearing locations by the same amount, and thus a shaft misalignment is initiated. A set of bearing caps having an eccentricity of 125 μm is used for the misalignments. When two of these caps are used on a shaft with the eccentricities in opposite direction, a total misalignment of 250 μm is achieved over a span of 250 mm, resulting in a misalignment of h = 0.001 m/m. Pairing this shaft with one having no misalignment results in a total gear misalignment of h = 0.001 m/m. If this misaligned gear shaft is matched with another shaft having the same amount of misalignment in the opposite direction, the gear pair misalignment is doubled to h = 0.002 m/m. Figure 4 shows how the two levels of shaft misalignment in the direction of line of action are obtained in both directions to achieve total slopes of $h = \pm 0.001$and $h = \pm 0.002$ m/m, in addition to the case of zero misalignment. For example, when the directions of eccentricities on the driving shaft are at position 1, and those on the driven shaft are at position 2 (as in Figure 4), one obtains h = –0.002 m/m face width. Position 2 on the driving shaft and position 1 on the driven shaft, meanwhile, represent a misalignment of $h = 0.002$ m/m. Table 2 shows the test matrix considered in this study. Gear pairs $g1$–$g2$, $g1$–$g3$ and $g1$–$g4$ were tested under the five shaft misalignment values of $h = 0$, ± 0.001 and ± 0.002 m/m, resulting in a total of 15 tests. Each of these tests was performed at discrete torque values of 100, 200, 300, 400 and 500 Nm, bringing the total number of tests to 75.

All of the tests were performed at a constant rotational speed of 200 rpm in order to avoid any dynamic effects. Gauges on tooth C (C1 through C5) were considered as the primary gauges, and data from these five gauges were collected and analyzed simultaneously to capture the variation of root strains along the facewidth. Additional tests were also performed with active gauges of A3, B3, C3, D3 and E3 to ensure that the tooth-to-tooth variability of the measured strain signals is minimal. The strain signals were taken out of the shaft through a slip ring, put through a multi-channel strain-gauge conditioning unit (Vishay 2300), acquired by a NI PXI–4472 dynamic signal acquisition module integrated into the multi-purpose NI PXI–1042, and finally processed using a *LabView* program.

Experimental Results

The experimental data for the case of no misalignment and no lead crown will be presented as the baseline condition. As such, gear pair $g1$–$g2$ is used with no shaft misalignments; i.e., $\lambda = 0$ and $h = 0$. Figure 5 shows the strain-time histories of the gauges of tooth C gauges throughout one loading cycle—at four discrete torque levels of 100, 200, 300 and 400 Nm. Several observations can be made from Figure 5; first, each gauge exhibits only tensile strains, indicating that efforts to move the gauges to a location as close to SAP as possible was successful. Gauges C1 and C5 measure slightly less load than gauges at the middle section of the tooth, with C3 having the highest strains. Each gauge is loaded for slightly longer than two complete mesh cycles. That is in line with the value of the theoretical total contact ratio of the gear pair. The measured strain signal for gauge C1 leads the other gauges since that side of the contact is initiated at that side of gear $g1$. The root strains measured along the face width indicate that the load distribution is quite uniform. Moreover, the increase in strain levels is almost linear with torque transmitted,

continued

Table 2. Test matrix considered in this study.

Gear Pair	Total Lead Crown λ(μm)	Misalignment *h* (m/m)	Torque *T*(Nm)
1	0	0	100–500
		+0.001	
		–0/001	
		+0.002	
		–0.002	
2	12.5	0	100–500
		+0.001	
		–0/001	
		+0.002	
		–0.002	
3	25	0	100–500
		+0.001	
		–0/001	
		+0.002	
		–0.002	

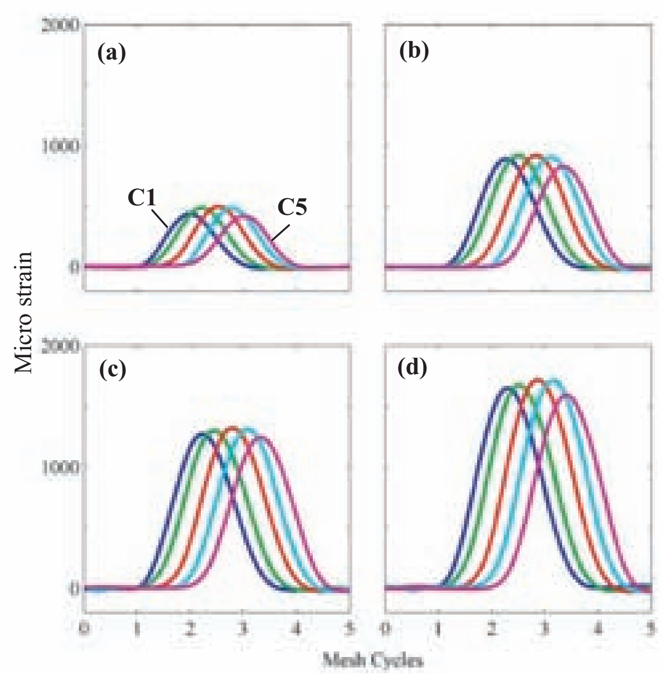


Figure 5—Measured strain time histories for gear pair $g1$–$g2$ having $\lambda = 0$ at $h = 0$—(a) $T = 100$ Nm; (b) $T = 200$ Nm; (c) $T = 300$ Nm; and (d) $T = 400$ Nm.

indicating that gear tooth deformations are linearly proportional to the gear mesh force.

Next, four different misalignments of $h = \pm 0.001$ m/m and ± 0.002 m/m are applied to the same gear pair $g1$–$g2$ at $T = 200$ Nm, and the resultant changes in root strains were compared to the baseline case of $h = 0$. Figure 6 shows the measured strain signals from gauges of tooth C for different h. A dramatic change in the root strains is observed with h. When $h < 0$, load is shifted to the edge near C1, resulting in significantly larger strains measured by this gauge in Figure 6 (a) and (b), while gauge C5 is very lightly loaded. The opposite is true for $h > 0$, where C5 bears the maximum strain while C1 records the lowest strain values. Both Figure 6 (a), for $h = -0.002$ m/m, and Figure 6 (e), for $h = 0.002$ m/m, represent severe edge-loading conditions that must be eliminated through lead crown. In these extreme cases, the measured maximum strains were about 45–60% higher than the maximum strain value measured by gauge C3 in Fig. 6 (c) for h = 0.

Figure 7 shows the measured strain data from gear pair $g1$–$g3$ at the same load and misalignment levels as in Figure 6; this gear pair has a total lead crown of $\lambda = 12$ µm. As a result, Figure 7 (c), for $h = 0$, indicates that the strains at the edges are reduced somewhat, while gauge C3 measures more strain than that of Figure 6 (c). This allows compensation of edge loading in Figure 7(e) for $h = 0.002$ m/m, while the load sharing in Figure 7(a) and (b) for $h < 0$ is still rather poor. Similarly, Figure 8 shows the same data, but now for gear pair $g1$–$g4$, which has a total lead crown of $\lambda =$ µm. In this figure, no sign of edge loading is evident, even for extremes of h. In Figure 8 (a), for $h = -0.002$ m/m, the maximum strain is at gauge C2, while it is gauge C4 for $h = 0.002$ m/m in Figure 8 (e). Both suggest that the amount of λ was sufficiently large for this gear pair under such misalignment conditions to prevent edge loading. The same behavior observed in Figures 6–8 for $T = 200$ Nm was seen in other tested torque levels as well.

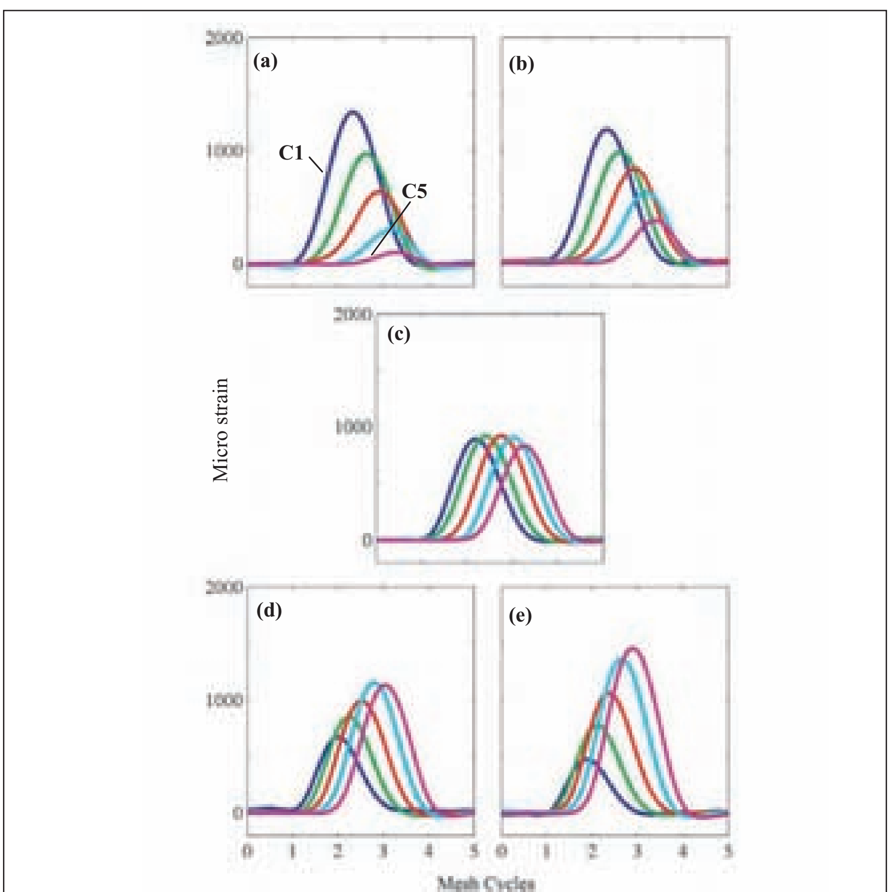


Figure 6—Measured strain time histories for gear pair $g1$-$g2$ having $\lambda = 0$ at $T = 200$ Nm—(a) $h = -0.002$ m/m; (b) $h = -0.001$ m/m; (c) $h = 0$; (d) $h = 0.001$ m/m; and (e) $h = 0.002$ m/m.

Figure 9 provides a direct comparison of the variation of maximum root strains along the gear face width between λ and h. In Figure 9 (a), for gear pair g1–g2 ($\lambda = 0$), the slope of the measured maximum strains is positive for $h > 0$ and negative for $h < 0$. Given this data, one would reasonably expect the curves for various h values to intersect on the mid-plane of the gears—at 10 mm from the tooth edge—while in fact they cross at about 8 mm from the edge. This is perhaps due to unavoidable shaft misalignments within the test machine tolerances. In Fig. 9 (a), an allowable misalignment tolerance band of $h = \in [-0.002, 0.002]$ results in nearly 50% more strain near the edges. Considering that gauges C1 and C5 were two to three mm away from the tooth edges, the strain values at the edges should be even higher. In fact, the maximum strain distributions in Figure 9 (b), for gear pair $g1$–$g4$ with $\lambda = 25$ µm, introduce a convex-shaped distribution of the maximum root strain along the face width.

Comparison to Predictions

In this section, an existing gear contact model will be used to describe the trends exhibited by the experimental data. The model first computes the load distribution by using an existing load distribution model called LDP (Ref. 22), and then employs a three-dimensional finite element model to predict the root strains resulting from the computed load distribution (Ref. 22). LDP makes its prediction by first predicting the contact zone between the mating parts; next, for each point along this zone, the elastic deformation of each part is computed in the form of a compliance matrix. The elastic deformations considered include bending and shear deflections, base rotation and translation, as well as the local contact deflections of the teeth. Continuity and equilibrium conditions are then enforced. The continuity conditions assume initial separations and introduce a slack variable that tracks whether a given point is in contact or not. The initial separations considered in this study include micro-geometry modification of both parts and misalignment. The load distribution is then calculated from these conditions for each time-step through a rotation of one base pitch and using a modified, Simplex-type algorithm (Ref. 22). The load distribution predicted in LDP is then used as the impetus for the finite element model to calculate the predicted root stress. The 3D finite element model (Ref. 23) employs 20-noded isoparametric elements in order to capture bending and shear effects. Because the contact zone and the load distribution are previously calculated, the stiffness matrix only needs to be factored once, and different nodal solutions for individual time-step loading case can be calculated simultaneously. This is advantageous because factoring the stiffness matrix is the most time-consuming step. After applying the load distribution and obtaining the nodal solutions, principal stress solutions are obtained for the full model and, most importantly, for the root.

The model employs user-defined radial and face width positions to interpolate strains from the finite element shape functions.

Figure 10 shows the predicted strain data for gear pairs $g1$–$g2$ (λ = 0) and $g1$–$g4$ (λ = 25 μm) at T = 200 Nm. Figures 10 (a), 10 (b) and 10 (c) refer to misalignment values of h = – 0.002, 0 and 0.002 m/m face width, respectively. These predictions are in qualitative agreement with the experimental strain-time histories presented earlier in Figures 6 and 8. The results for λ = 0 show discrepancies with the experimental results, primarily in the edge gauges; this is caused by the prediction of edge loading. Also, the incongruities observed in these predictions are primarily due to edge loading. Predictions for λ = 25 μm, on the other hand, exhibit a better correlation with the experimental results and show no incongruities resulting from the removal of predicted edge loading due to lead crown.

Figures 11 and 12 show the predicted maximum contact stress and maximum normal root stress distributions (on gear $g1$) for (λ = 0 and 25 μm) at T = 200 Nm. In both figures, the plots for λ = 0 demonstrate excessive edge loading and corner contact effects for h = – 0.002 and 0.002 m/m, which are eliminated in gear pair $g1$–$g4$ case, due to the addition of lead crown. The contact stress and root stress follow a similar trend as both are similarly affected by the same predicted load distribution.

Finally, Figure 13 compares the experimental maximum root strain curves of Figure 9 with the predicted values from Figure 12 at the locations of gauges specified earlier. In these figures the predicted and measured maximum strain measurements for λ = 0 and 25 μm are compared for h = –0.002, 0 and 0.002 m/m face width, individually. Here, regardless of the value of h, the maximum strain is measured by one of the gauges in the middle, suggesting that edge loading is eliminated. It is also noted that the maximum strains of a gear pair having λ = 25 μm for h = –0.002 and 0.002 m/m are comparable to the maximum strain value found in the gear pair having no lead crown. The difference is the location where this maximum strain is measured; under no-misalignment conditions, the maximum strain of the gear pair having λ = 25 μm is nearly 40% higher than that of the gear pair having no lead crown. This clearly demonstrates that, while eliminating adverse edge-loading effects, lead crown increases the bending stresses rather significantly. Therefore, any design that uses lead crown to accommodate shaft misalignments must also account for this increase in the root stresses. It is also noted in Figure 13 that the model predictions are in reasonably good agreement with the measured values. This model can therefore be used with confidence in determining the optimum amount of lead crown required to compensate for a given misalignment condition.

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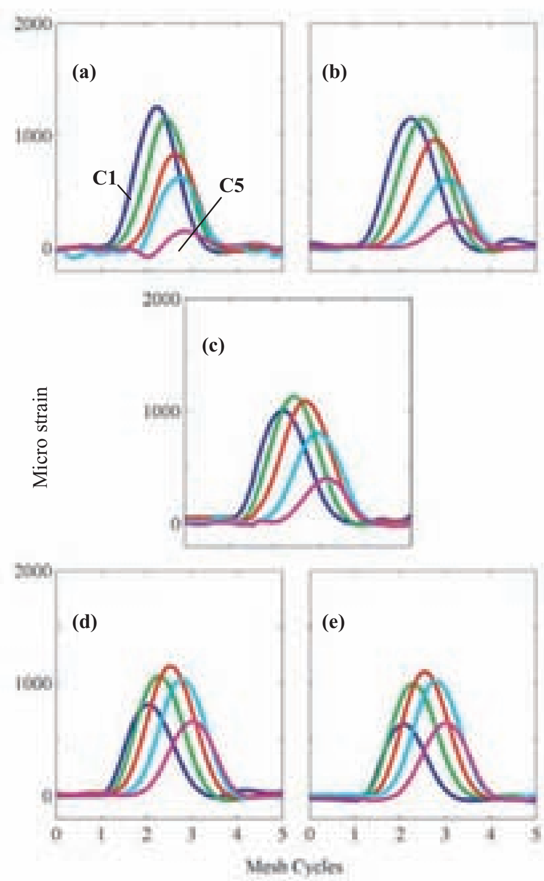

Figure 7—Measured strain time histories for gear pair $g1$–$g3$ having λ = 12 at T = 200 Nm; (a) h = -0.002 m/m; (b) h = -0.001 m/m; (c) h = 0; (d) h = 0.001 m/m; and (e) h = 0.002 m/m.

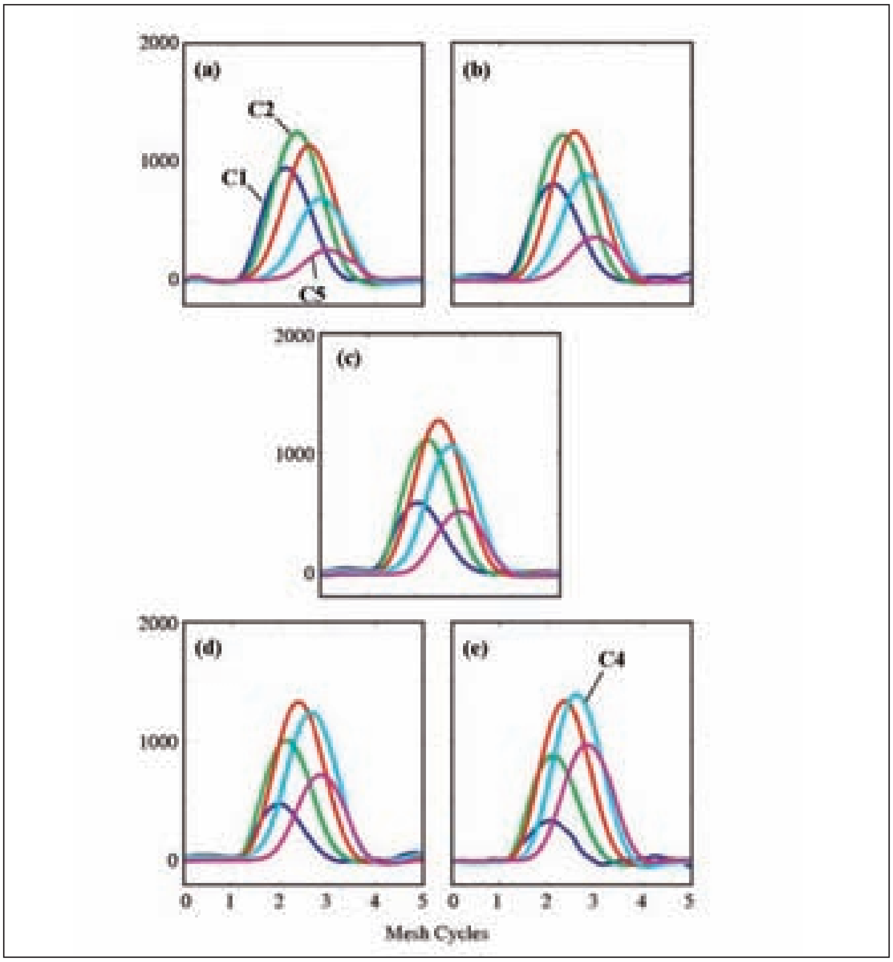

Figure 8—Measured strain-time histories for gear pair *g*1–*g*4 having λ = 25 at *T* = 200 Nm—(a) *h* = –0.002 m/m; (b) *h* = –0.002 m/m; (c) *h* = 0; (d) *h* = 0.001 m/m; and (e) *h* = 0.002 m/m.

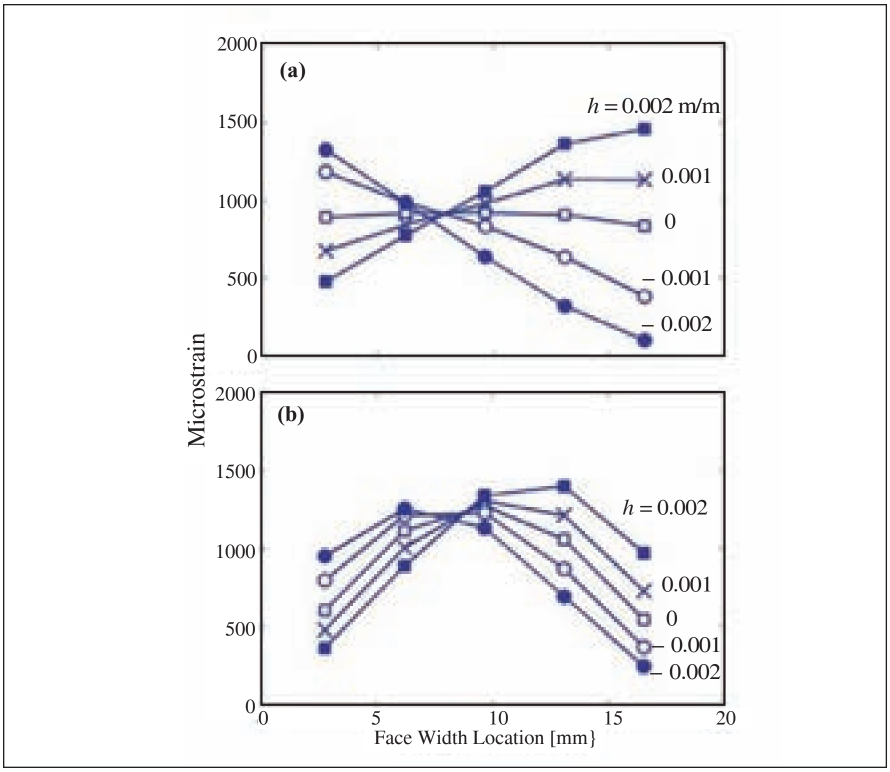

Figure 9—Variation of maximum root strain along the face width with *h* at *T* = 200 Nm for (a) λ = 0 and (b) λ = 25 μm.

Conclusions

The combined influence of shaft misalignments and gear lead crown on the load distribution and tooth bending stresses was investigated experimentally. An experimental study was performed by using a set of helical gear pairs having various amounts of lead crown. Gears were operated under tightly controlled shaft misalignments introduced in the direction of the line of action; the distribution of the root stresses was measured for various lead crown values. The experimental results were then compared to the predictions of a gear load distribution model to demonstrate good correlation. It was also shown that the amount of lead crown must be defined as a function of expected gear misalignment conditions. While edge loading was eliminated, excessive lead crown values were shown to increase maximum root and contact stresses.

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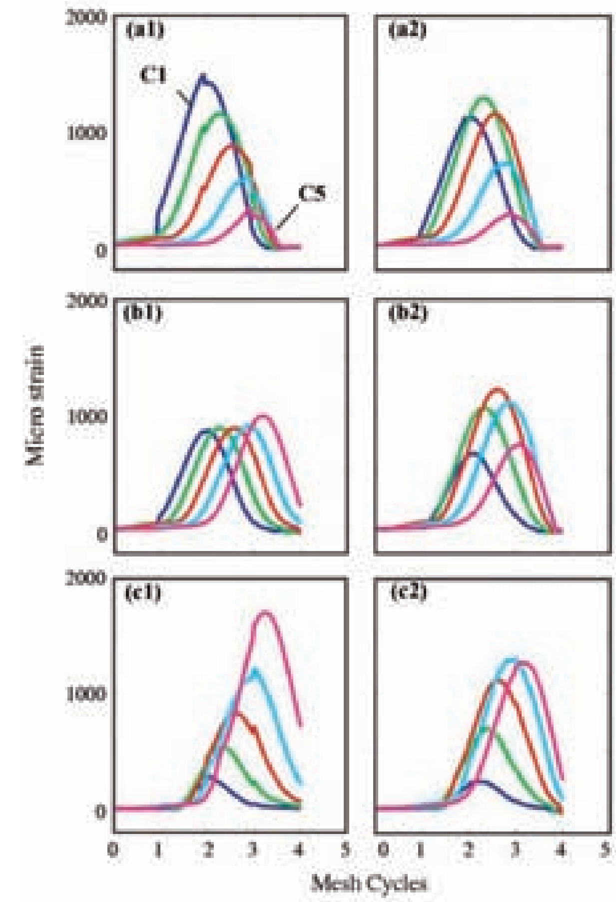

Figure 10—Predicted strain-time histories at T = 200 Nm—(a1) h = –0.002 m/m and λ = 0; (a2) h = –0.002 m/m, λ = 25 μm; (b1) h = 0 and λ = 0; (b2) h = 0 and λ = μm; (c1) h = 0.002 m/m and λ = 0; and (c2) h = 0/002 m/m and λ = 25 μm.

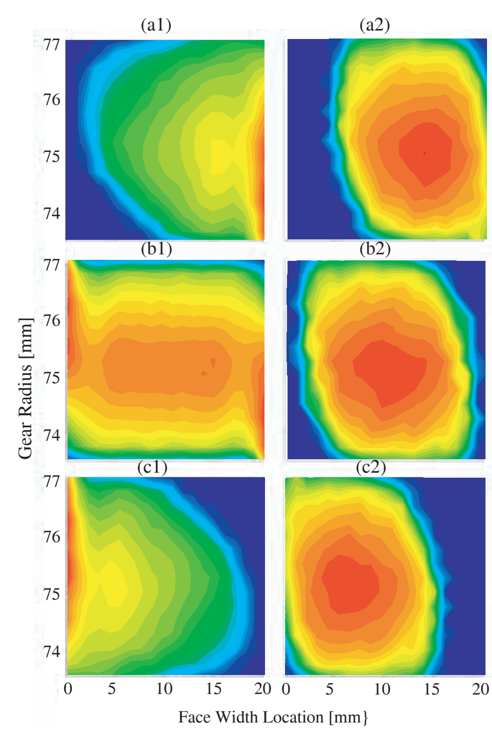

Figure 11—Predicted maximum contact stress distribution at T = 200 Nm—(a1) h = –0.002 m/m and λ = 0; (a2) 0.002 h = -0.002 m/m and λ = 25 μm; (b1) h= 0 and λ = 0; (b2) h = 0 and λ = 25 μm; (c1) h = 0.002 m/m and λ = 0; and (c2) h = 0.002 m/m and λ = 25 μm.

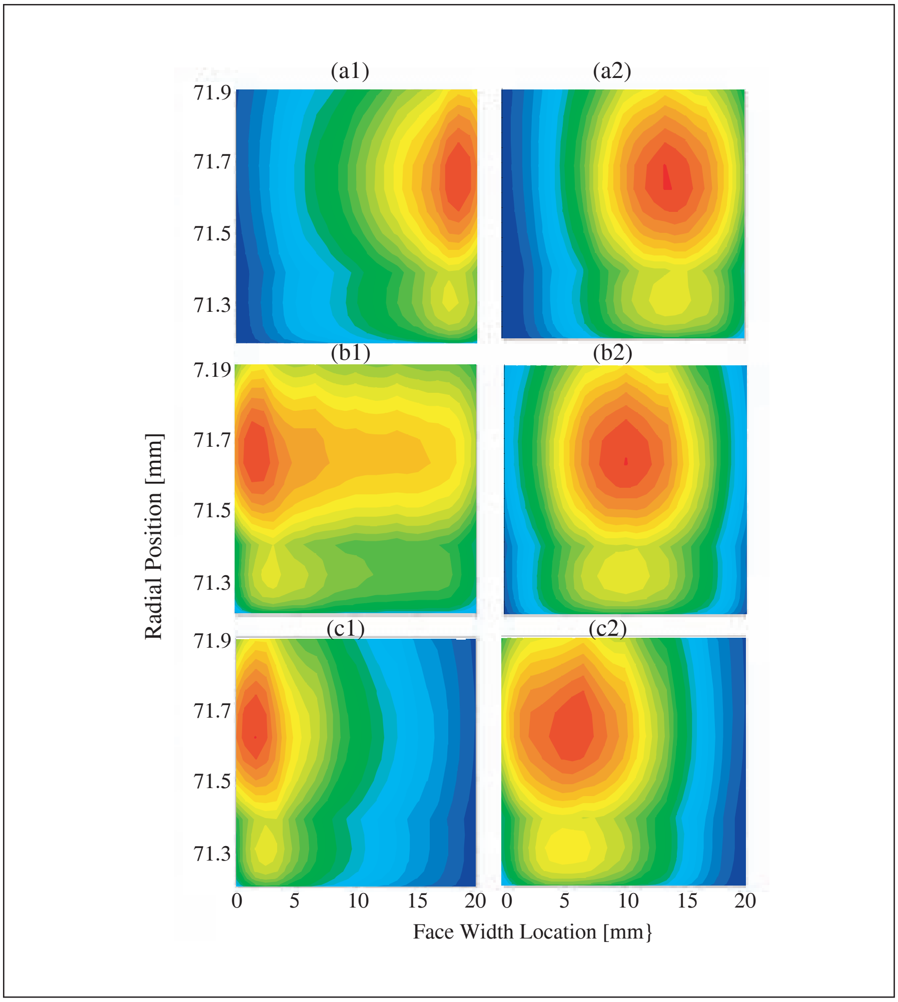

Figure 12—Predicted maximum root stress distribution at *T* = 200 Nm—(a1) *h* = -0.002 and λ = 0; (a2) *h* = -0.002 m/m and λ = 25 μm; (b1) *h* = 0 and λ = 0; (b2) *h* = 0 and λ = 25 μm; (c1) *h* = 0.002 m/m and λ = 0; and (c2) *h* = 0.002 m/m and λ = 25 μm.

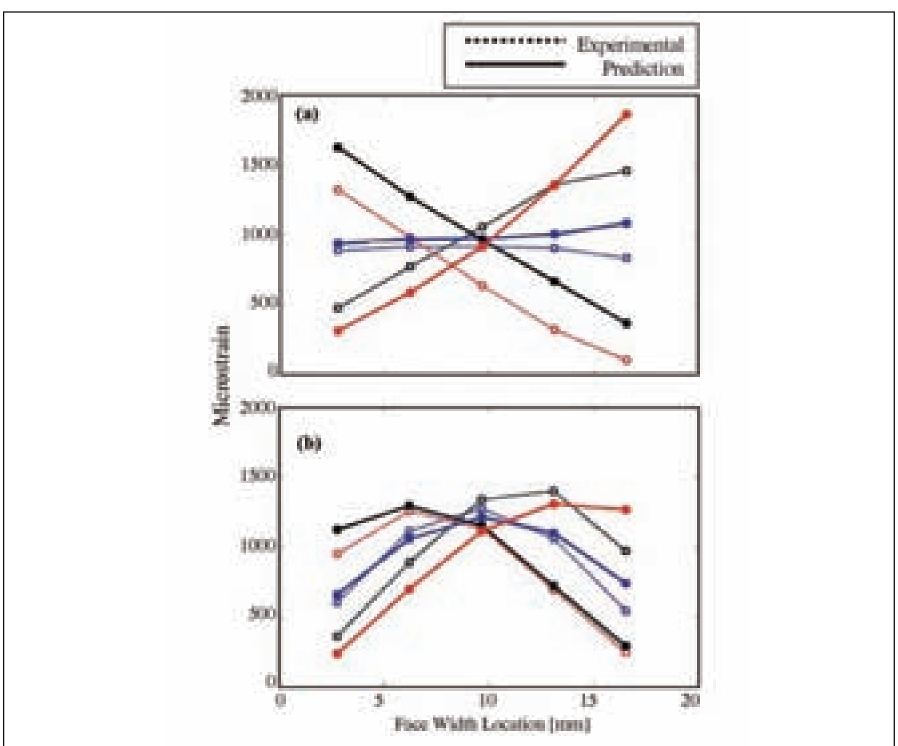

Figure 13—Comparison of measured and predicted maximum root strain along the face width at *T* = 200 Nm—(a) λ = 0 and (b) λ = 25 μm.

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A Further Study on High-Contact-Ratio Spur Gears in Mesh with Double-Scope Tooth Profile Modification

Jiande Wang and Ian Howard

(Proceedings of 10th ASME International Power Transmission and Gearing Conference PTG 2007, September 4–7, 2007, Las Vegas)

NOMENCLATURE	
O_1	Driving gear hub center
O_2	Driven gear hub center
HCRG	High-Contact-Ratio Gears
LCRG	Low-Contact-Ratio Gears
BTR	Bulk Tooth Rotation
T.E.	Transmission Error
LSR	Load-Sharing Ratio
Km	Combined torsional mesh stiffness, Nm/rad
P, P_1, P_2	Magnitude of profile modification, mm
ΔL_1, ΔL_2	Length of profile modification, mm
AT	Approach tooth (contact)
RT	Recess tooth (contact)
MT	Mid-tooth (between AT and RT)
SEQV	von Mises stress, MPa
S_1	First principal stress, MPa
S_3	Third principal stress, MPa
B	Base point
S, S_1, S_2	Relief starting point
t_0	Involute tooth tip
t_1, t_2	Modified tooth tip
σ	Profile rotation angle
e	Extra extension (the amount below base point).
ε_α	Profile contact ratio
P_b	Base pitch
T'_1, T'_2	Limits of the plane of action ($T'_1T'_2 = \frac{1}{2}\ \varepsilon_\alpha\ P_b$)

Management Summary

This paper will demonstrate that, unlike commonly used low-contact-ratio spur gears, high-contact-ratio spur gears can provide higher power-to-weight ratio, and can also achieve smoother running with lower transmission error (TE) variations. The research presented in this paper concentrates on providing proofs and verifications on the topic by using modern numerical methods and comprehensive analysis. Additionally, a general bulk tooth rotation (BTR)-type tooth profile modification is introduced and applied to the high-contact-ratio spur gears in demonstration of improved tooth profile design.

Introduction

Research on high-contact-ratio spur gears with various tooth profile modifications was previously conducted using experimental (Refs. 1–2) and numerical methods (Refs. 2–3). Research by Yildirim (Ref. 2) has provided more details of the double-scope tooth profile modification, and his analytical results have shown the advantages of bringing the two reliefs together, resulting in an overall superior performance in terms of peak-to-peak transmission error value, maximum tooth load value, tooth load sharing ratio and smoother TE curves. More recently, Ajmi (Ref. 3) continued researching the double-scope tooth profile modification with modern numerical simulation methods. It was further confirmed that the relief with two slopes was superior to a conventional short or long, linear relief. Furthermore, the analysis presented here was conducted with the tooth loading conditions and dynamic conditions in order to assess some potentially interesting profile modifications for future consideration.

However, the above analyses can also be seen as having distinct shortcomings. First, Yildirim's analytical (or geometrical) results (Ref. 2) do not fully represent the characteristics of elastic bodies under load, though under very light loads it may be close enough to reality. Second, the predicted transmission error of Ajmi (Ref. 3) doesn't appear to satisfy the principle of solid mechanics, where the width of single, double and triple contact zone was shown to be far too rigid. In the case of common low-contact-ratio gears in mesh, the width of the single contact zone shrinks and the double contact zone expands while the load increases (Refs. 4–5). For the tip-relieved, high-contact-ratio gears in mesh, the width of single, double and triple contact zone will change dramatically while the load increases (Refs. 6–7). Obviously, it is vital to overcome those shortages in order to realize a unique, optimum design of high-contact-ratio spur gears. Based on the results of previous research (Ref. 6) and the use of advanced modeling techniques such as adaptive meshing and an element birth and death option, a satisfactory simulation result can be achieved on the high-contact-ratio spur gears in mesh with double-scope tooth profile modification. In order to compare previous results and published experimental tests, the original high-contact-ratio gear model (Ref. 8) was used, and for further reference, this study was also conducted using another classical high-contact-ratio spur gear model, which has been used by many researchers (Refs. 1, 7 and 9–11).

The FE Model and the Profile Modification

The FE model of the high-contact-ratio spur gear pair has parameters as shown in Table 1, based on the set ratio 1:1 of the test rig used by Munro for the transmission error measurements (Ref. 8). In the test rig, the gear center distance was set to 203.2 mm, which is slightly greater than the theoretical value of 202.9 mm. The FE model and details are shown in Figure 1, where the theoretical value 202.9 mm was used for the center distance.

The FE model was built with the following major considerations:

1) 2D plane stress quad elements were used for TE evaluations, combined torsional mesh stiffness and tooth load sharing ratio; and also for the primary calculations of tooth contact stress and tooth root stress. Over the mesh cycle, the majority of components were calculated with acceptable errors (Ref. 12), except for the higher-order components such as the tooth contact stress. 3D brick element models were then used for a better evaluation of the tooth contact stress over the flank face.

2) For computation efficiency and solution accuracy, adaptive meshing was used, and the necessary element sizing (Ref. 6) was performed.

3) When the tooth profile modification was applied, the element birth and death option was used, which has proven to be efficient for solving the model when rigid body motion was incorporated into the mesh cycle (Ref. 7). Finally, all considerations were incorporated into several looping programs, running automatically, that also included post-processing, as each mesh cycle contained over 100 calculation points. According to the previous research results (Ref. 2, 3 and 13), and the "MAAG Gear Book" (Ref. 13), the parameters of the profile modification—as shown in Figure 2—have the

continued

Table 1. Gear Parameters used in the Analysis.

Material	Steel
Friction coefficient	0.06
Number of teeth	54
Nodule M, mm	3.738
Pressure angle, deg	18
Addendum, mm	1.26
Dedendum, mm	1.66
Profile shift coefficient	0.143
Center distance, mm	202.9*
Face width, mm	11
Theoretical contact ratio ε_a	2.36
Tip fillet radius, mm	0.2
Hub radius, mm	25.4
Design (max) load, Nm	700

* When the profile shift coefficient is zero, the center distance should be the standard of 201.852 mm.

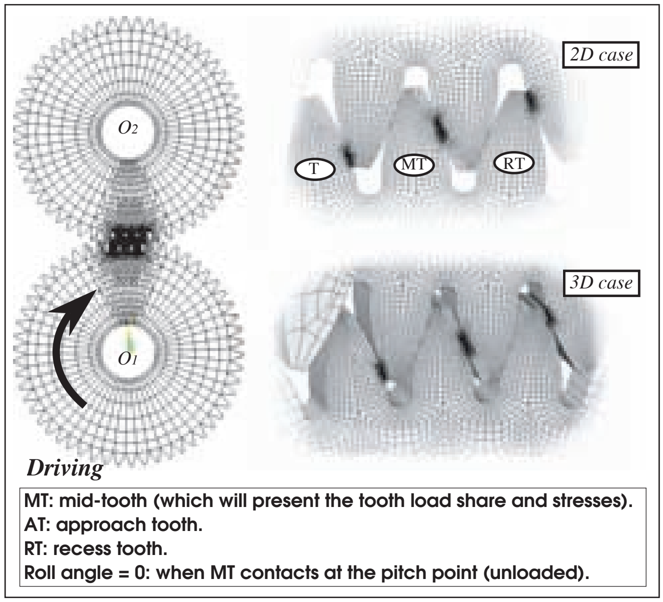

Figure 1—FE model of the gears in mesh (ratio 1:1) and its mesh adaptation with contact.

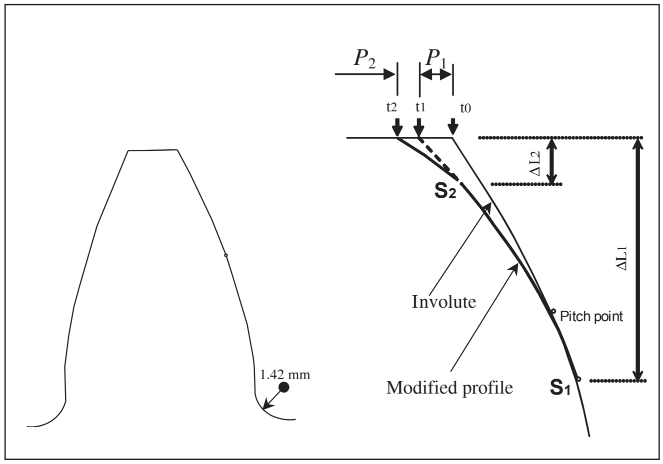

Figure 2—Details of profile relief.

following magnitude of linear type relief:

$$P_1 = |t_0 t_1| = 20\mu m, \quad (1)$$

$$P_2 = |t_1 t_2| = 10\mu m, \quad (2)$$

where: P_1 and P_2 are the amplitudes of relief, and t_0, t_1 and t_2 are the tooth tip positions.

The first and second relief extent are given by the following,

$$\Delta L_1 = 0.5 T_1'T_2' \quad (3)$$

$$\Delta L_2 = 0.075 T_1'T_2' \quad (4)$$

and

$$T_1'T_2' = \varepsilon_\alpha P_b = 13.18 \text{ mm} \quad (5)$$

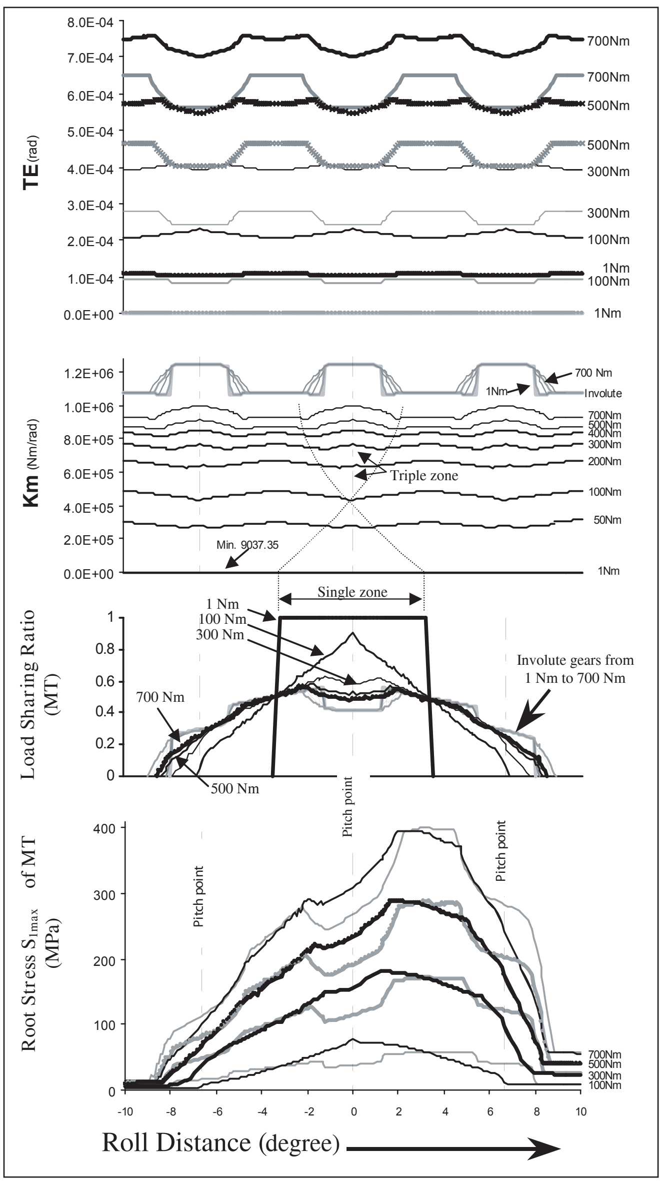

Figure 3—The changeover process under various input loads of the involute gears (light-color curves) and the profile-modified gears (dark-color curves).

Analysis with Tooth Profile Modifications

The analysis was conducted according to comprehensive methods (Refs. 4, 6 and 14) in which the techniques of adaptive mesh and element birth-death options were used to ensure accuracy and to compensate for the rigid body motion resulting from the profile modifications. The transmission error, combined torsional mesh stiffness (Km), load-sharing ratio and tooth root stress were produced over the complete mesh cycle simultaneously, as shown in Figure 3.

It can be seen that each of the components has also been compared with that of unmodified gears in order to clearly demonstrate the meshing characteristic variations. The contact stresses of unmodified and modified gears are presented separately (Fig. 4).

In summary, the double-scope profile modification achieved the following results:

- A primary rigid body motion occurred due to the long relief extent, which was causing a significant, unloaded transmission error.
- Transmission error variation was reduced, but no smooth TE curve was found near the design load of 700 Nm. The expectation of two smooth TE areas in moderate load range was not produced.
- Combined torsional mesh stiffness results were reduced significantly with lighter load and tended to be smoother.
- Tooth root stress was relatively high, with the lighter input load below the design load of 700 Nm.
- Contact stresses were significantly reduced, compared to unmodified gears, especially for loads under 300 Nm. Contact stresses tended to be smooth with only small stress irregularities at the relief starting point. However, the relatively high stress, especially for the contact recess case in Figure 4 (b), has shown that premature contact occurs near the tooth tip when the input load increases. The less significant, high stresses occurred at the contact approach case and at the relief starting point $\mathbf{S}_2$. Increasing the magnitude of profile modification P_1 or P_2 could avoid high stresses near the tooth tip, but it cannot avoid high stresses occuring at the relief starting point(s).

Comparison with previous transmission error research results can be made as shown in Figure 5, where it can be seen that the effect of primary rigid body motion has been removed in (a), (b) and (c) so that the absolute value of TE under each load was reduced. The results in (a) and (b) also show some zero TE under zero load. Experimental measurement (a) has shown large TE variations under zero and lighter loads. This usually is due to manufacturing or geometric errors, and corresponding errors larger than elastic deformation are common in many experimental tests.

Long tooth profile modification can also cause large zero (or lighter) load TE variations if the primary rigid body motion is comparatively small, or if the primary rigid body motion is not accounted for.

High-contact-ratio gears with long profile modification will result in single tooth contact under light loads. The single contact zone in the mesh cycle can quickly disappear when the load is increased to become a triple contact zone that will then expand in zone width as the load is further increased. However, these characteristics have not totally been shown in the results in Figure 5 (b) or 5 (c).

A Novel Profile Relief Based on Tooth Deformation Reconsideration

The numerical study of tooth deformation included the use of common low-contact-ratio gears in mesh (Ref. 7). For the complete gear pair in mesh, the tooth local deformation is shown in Figure 6.

Five significant deformations have been highlighted in Figure 6, where 1 and 2 represent the tooth root bulk rotation, 3 is the Herzian (contact) deformation, 4 is due to the local shear stress and 5 is due to tooth bending (and tension).

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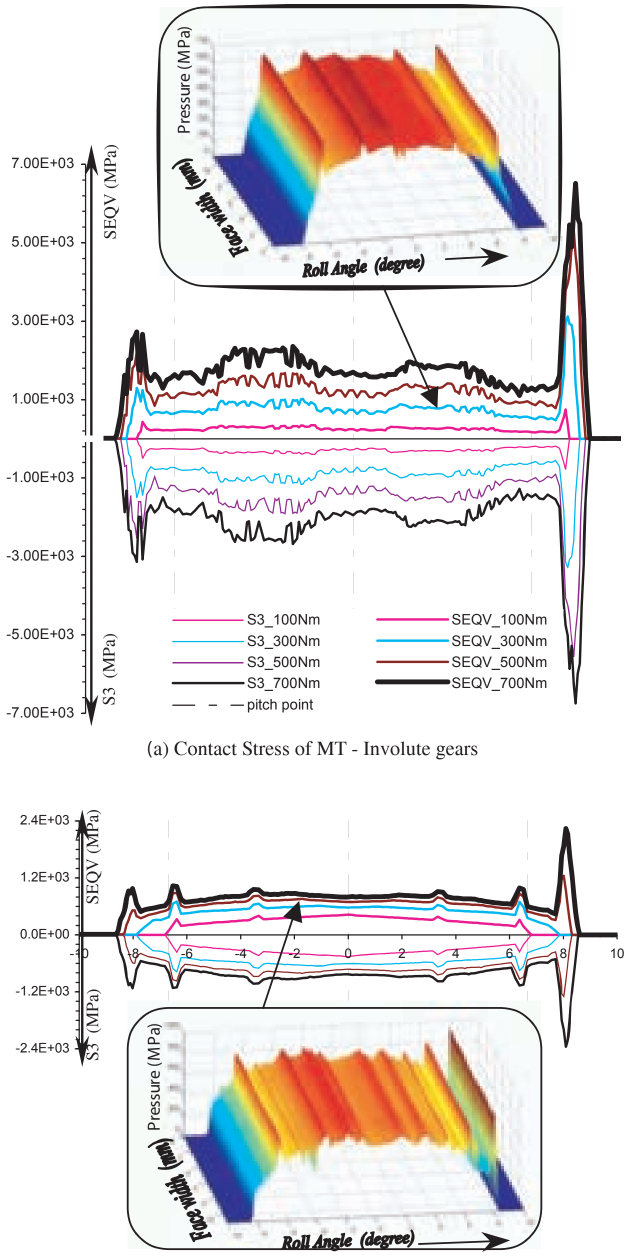
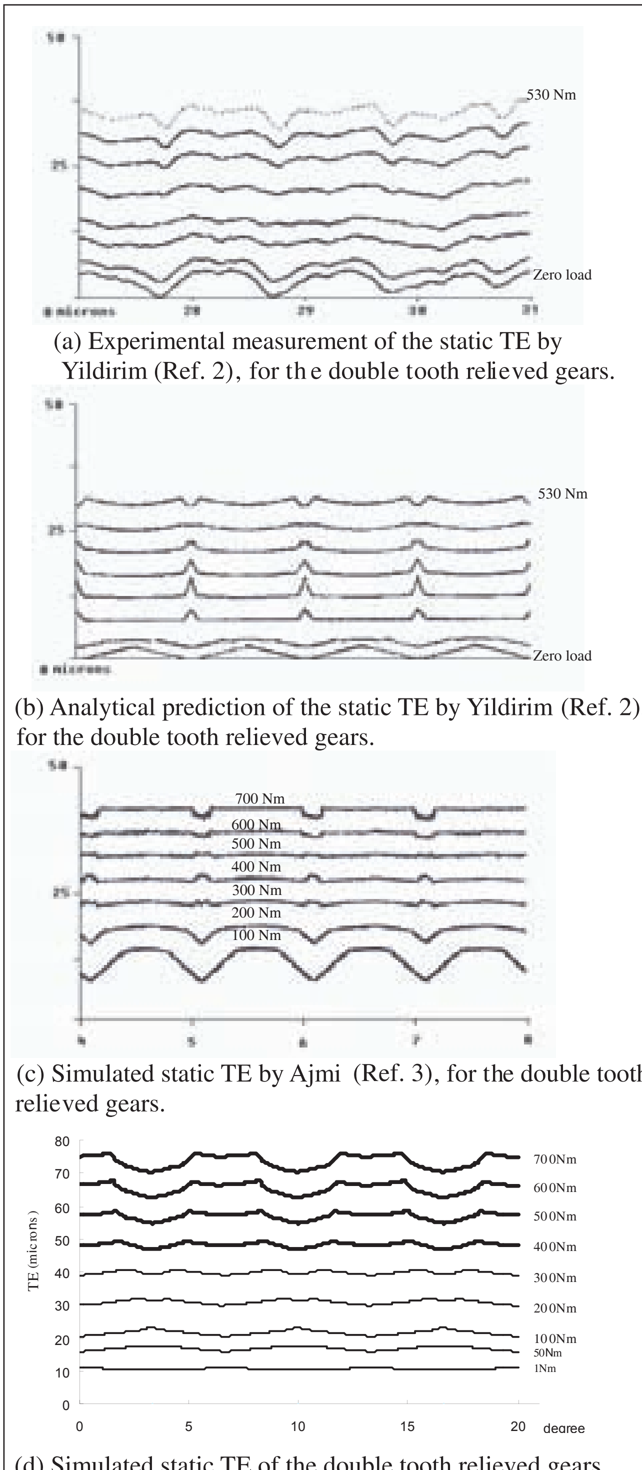


(a) Contact Stress of MT - Involute gears

(b) Contact Stress of MT - Profile relieved gears

Figure 4—Contact stresses of mid-tooth over the mesh cycle under various loads.



(a) Experimental measurement of the static TE by Yildirim (Ref. 2), for the double tooth relieved gears.

(b) Analytical prediction of the static TE by Yildirim (Ref. 2), for the double tooth relieved gears.

(c) Simulated static TE by Ajmi (Ref. 3), for the double tooth relieved gears.

(d) Simulated static TE of the double tooth relieved gears in this research.

Figure 5—Experimental and analytical results of TE predictions.

The rotation of the unloaded tooth profile to the deformed tooth demonstrates that the tooth bulk rotation is the largest amount out of the total tooth deformation. Based on the above information, a bulk rotation tooth profile (about tooth root) type relief was tested, called the bulk tooth rotation (BTR)-type tooth profile relief or modification. The modified tooth profile starts from point S, which is below the base point B with a (small) distance e called extra extension. The modified tooth profile S t_1 was produced by rotating the involute profile

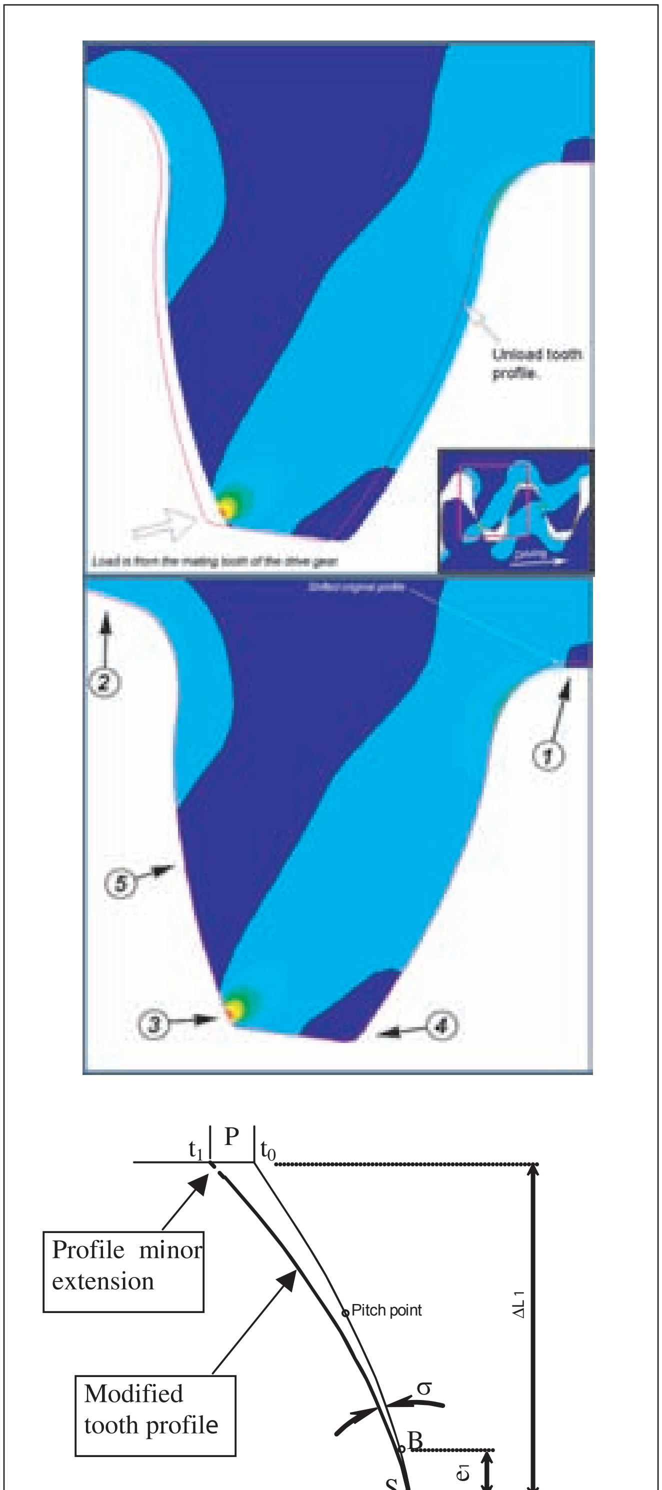


Figure 6—Local tooth deformation and profile modification.

S t_0 through an angle σ with a minor profile extension near t_1. Compared to the current application recommendations—BS (Ref. 15) and ISO (Ref. 16)—the tooth bending-based standards, bulk tooth rotation-type tooth profile relief is production-friendly, especially for the double relief.

Analysis with Bulk Tooth Rotation (BTR)-Type of Tooth Profile Modification

Bulk tooth rotation-type tooth profile modification was applied on the high-contact-ratio gears with a relief magnitude of 70 μm . The relief starting point was positioned at the profile rotating point, just under the base point. Thus the modified tooth profile simulated 1.6 degrees of bulk tooth rotation about the root. The results of FE simulation of BTR-type tooth profile modified gears over the mesh cycle are shown in Figure 7.

It can be seen that a smoother transmission error has been achieved for both the lighter load of 50 Nm and the higher load (design load) of 700 Nm, and these characteristics are also shown in the curves of combined torsional mesh stiffness. More significant is that the tooth contact stresses have been optimized over the load range. To confirm the 2D analysis results, the analysis was also confirmed with 3D modeling. The 3D results are compared as shown in Figure 8. Significant differences between 2D and 3D results were found in the tooth contact stresses, where the 2D stress for SEQV (von Mises stress) may be overstated compared to the 3D results, and the 2D third principal stress may be underestimated. However, the contact stress optimizations have been achieved. The major disadvantage of the BTR-type relief has been found in the tooth root stress as shown in Figure 9, where the tooth root stress could be increased up to 28% from that of involute gears when the input load is 300 Nm. However, there is only a minor increase of 4% at the input design load of 700 Nm.

Conclusions

Double-scope tooth profile modifications are attracting interest, as they show great potential for improving gear designs for applications including reduced noise and vibration. However, the research in this paper shows that neither previous nor current results can provide smooth transmission error curves at both design and light loads. Also, the tooth loading condition— especially the tooth contact stress—was still subject to high fluctuations over the mesh cycle. Tooth-loading condition is a critical parameter in high-contact-ratio gear applications, especially the tooth contact stress (or pressure) which is critical to the wear, pitting and operating temperature which was studied by Cornell (Ref. 9). For now, the application of double-scope tooth profile modifications still requires extensive study to ensure its effectiveness and its manufacturing possibilities.

An alternative tooth profile modification for high-contact-ratio gear applications—the bulk tooth rotation-type—has been introduced. Documentation in this paper has shown that BTR-type tooth profile modification can provide smooth transmission error pass in both light-load (cruise running) and full, heavy-load conditions (design load). Moreover, the tooth

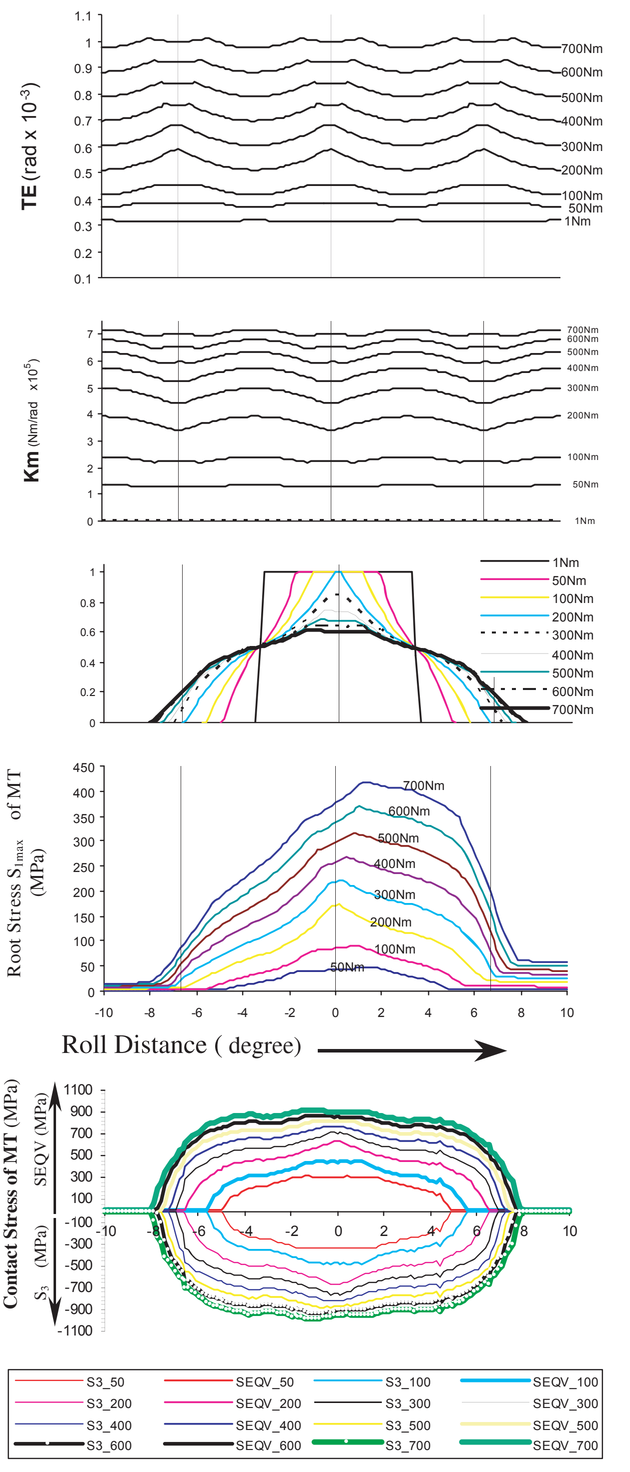


Figure 7—The changeover process under various input loads of bulk tooth rotation-type, relieved HCR gears (2D model).

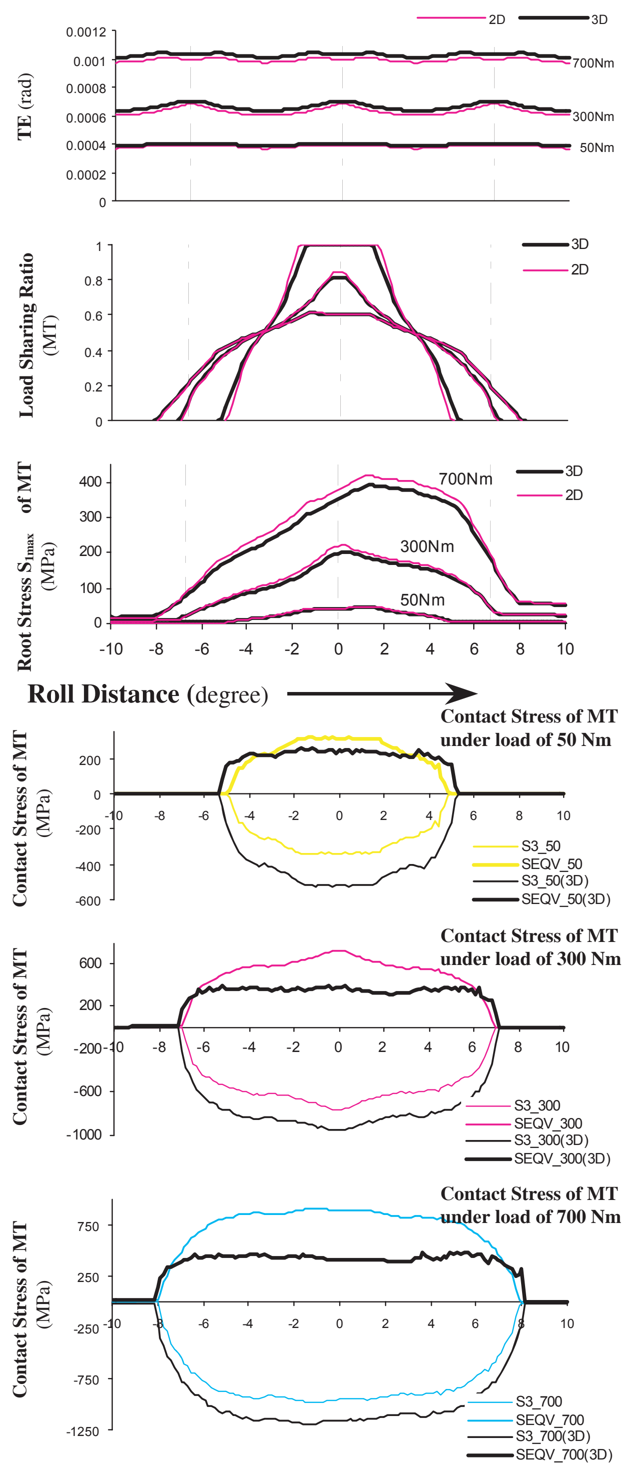


Figure 8—The changeover process under various input loads of bulk tooth rotation-type, relieved HCR gears (2D and 3D model).

pressure (contact stresses) has been smoothed out over the mesh cycle.

BTR-type tooth profile modification has general benefits for both low-contact-ratio gears and high-contact-ratio-gears. However, its major drawback—especially for high-contact-ratio gear applications—is slightly increased tooth root stress when the input load is below the design load. To that extent, this could affect the gears' fatigue life.

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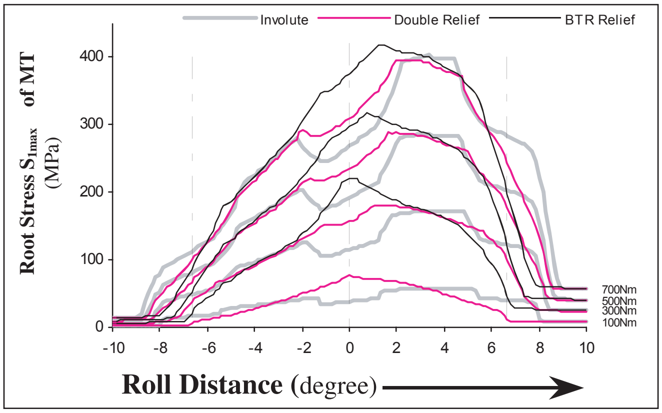

Figure 9—Tooth root stress comparisons.

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What are Your Company's Greatest Manufacturing/Engineering Challenges for 2009?

"Engineering help."
—Corporate executive at a U.S. precision gear manufacturer

"Euro to dollar conversion. Asian competition."
—Design engineer at a U.S. manufacturer of planetary gearboxes

"Expansion costs for going after large projects. Financing large new projects."
—VP of engineering at a U.S. manufacturer of rotary dryers and kilns

"Face new investments to commit with market demands."
—Manufacturing engineer at a British gearbox manufacturer

"Far East competition."
—Corporate executive at a Canadian manufacturer of workholding

"Finding new products."
—Corporate executive at a German gearbox manufacturer

"Finding qualified, highly technical employees who have experience in our technology."
—Corporate executive at a U.S. manufacturer of PVD coating equipment

"Finding skilled people."
—Corporate executive at a U.S. manufacturer of mining equipment

"Follow up new product lines."
—Corporate executive at an Italian manufacturer of speed reducers

"Gear tooth hard finish process implementation for most of our products."
—Engineer at a U.S. manufacturer of automotive gearboxes

"Get lean, improve quality."
—Consultant to Indian gear manufacturers

"Growing demand."
—Quality control worker at a Polish manufacturer of locomotives

"Hold our gain until the economy rises again."
—Quality manager at a U.S. manufacturer of automotive gearboxes

"Improvement of technologies and skills."
—Design engineer at a French manufacturer of gas turbines

"Improving manufacturing techniques to increase throughput."
—Engineering manager at a British manufacturer of motorsport transmissions

"Increase product efficiency and incorporating electronics."
—Design engineer at a gearmotor manufacturer in the Netherlands

"Insurance cost, taxes. Skilled employees. Drugs."
—Corporate executive at a U.S. manufacturer of oilfield gears

"Integrating additional new products into our existing facility and making all the product lines more productive."
—Quality control worker at a U.S. manufacturer of diesel engines

"Lack of skilled labor. Engineers that are able to adapt in a fast-paced job shop situation."
—Sales manager at a U.S. gear manufacturer

"Lean manufacturing implementation."
—Manufacturing engineer at a U.S. manufacturer of aircraft

"Long new equipment lead times."
—Sales manager at U.S. gear manufacturing job shop

"Developing low-cost, heavy rotary actuators."
—Design engineer at a manufacturer of motion systems in Israel

"Maintaining capital spending to meet future growth in a very challenging business climate."
—Design engineer at a U.S. manufacturer of automotive chain drives

"Maintaining turnover, fighting exchange rates, fighting rising energy costs, increasing UK turnover in particular."
—Sales manager at a British gear manufacturing job shop

"Material costs and product quality."
—Design engineer at U.S. manufacturer of water treatment drives

continued

What are Your Company's Greatest Manufacturing/Engineering Challenges for 2009?

"New assembly line."
—Engineer at an Indian engine and transmission assembly plant

"None."
—Production worker at an Indian manufacturer of defense equipment

"Obtaining the money to improve the production and replace antiquated machines with more *versatile and less 'skilled operator' machines."*
—Manufacturing engineer at a U.S. manufacturer of enclosed gear drives

"Old equipment, no maintenance person, too much dead weight personnel-wise, management not earning their paycheck."
—Production worker at a U.S. gear manufacturing job shop

"Process improvement."
—Design engineer at a German manufacturer of forged bevel gears

"Productivity enhancement and reduce overhead."
—Production worker at a Pakistani manufacturer of transmission shafts and gears

"Reduce costs and adjust to new types of products."
—Engineer at a U.S. manufacturer of rack-and-pinion steering

"Revamping internal systems to meet customer demand for on-time delivery while dealing with rapid growth on an already strained scheduling system."
—Design engineer at a U.S. aerospace gear manufacturer

"Rising cost of material."
—Design engineer at a Korean speed reducer manufacturer

"Rising energy cost."
—Engineer at an Indian gear manufacturing job shop

"Becoming self-sufficient in design and development facilities."
—Engineer for an Indian automobile manufacturer

"Significant quality improvements in manufactured goods, expanding gear grinding capability."
—Corporate executive at a U.S. manufacturer of replacement gearing

"Start new projects in low-cost locations."
—Process design manager at a Belgian manufacturer of engine timing gears

"Staying competitive in technology."
—Sales manager at a U.S. manufacturer of powder metal gears

"The purchase of gear hobbing machines."
—Production worker at a Mexican manufacturer of industrial gearboxes

"To develop qualified alternative sources with a prime focus on cost reduction."
—Design engineer at an Indian manufacturer of gas turbines

"To find more abroad customers for complete gearboxes, transmissions for agricultural tractors, trucks and other off-road vehicles."
—Production worker at a Turkish manufacturer of gears and transmissions

"To keep inventory costs down and to increase productivity."
—Production worker at an Indian automobile manufacturer

"To manufacture products at the most economical price."
—Corporate executive at an Indian gearbox manufacturer

"Training and reduction of rework."
—Engineer at a Canadian manufacturer of mechanical presses

"Upgrade of machine tools and inspection equipment."
—Production worker at a U.S. manufacturer of custom gears

"We don't find any challenges."
—Engineer at an Indian manufacturer of spiral bevel gears

A "Whodunnit?!"

IN GEARBOX FAILURE

Ray Drago engages a student in discussion.

Forensics isn't just for tough-talking crime-busting scientists—most commonly found on your television; the tactic also holds the key to successful gearbox design and manufacture. At AGMA's Gearbox CSI: Forensic Analysis of Gear and Bearing Failures—Useful Tools for Optimizing Gearbox Design, find out just how to follow the integrated process of identifying gearbox failure and determining the conditions that lead to specific failures, in order to avoid future errors.

B-10 bearing life and basic gear service factor calculations may be the most well-known methods to predict gearbox failure, but upon forensic analysis, failures result from any number of contributing factors: design, fabrication, lubrication and the handling of gears and bearings. "If a failure is not properly addressed, the problem will continue to occur and will result in greater downtime and cost to the gearbox manufacturer," says Raymond Drago, who presents the course along with colleague Joseph Lenski.

Joe Lenski.

"The seminar will present methodologies that can be used to solve fatigue, scoring and wear problems," Drago says. "Each failure discussed is also accompanied by a discussion of the actions taken to avoid a recurrence of the failure and to avoid trading the problem solved for one created by the solution."

Primarily aimed towards gearbox designers, many other gear professionals can benefit from the seminar, including purchasers, specifiers and users who can learn how to get the most out of their gearboxes and help solve and do away with service problems. Gearbox maintenance people, overhaulers and operators who are often responsible for purchasing, maintaining and replacing gear systems will benefit by understanding the underlying causes of failure, which "is a crucial factor in the economical and long-lasting solution of problems that may occur in these very expensive [gearbox] systems," Drago says. "Further, by understanding the specific nature of the failures and their causative agents, these operators are also better prepared to specify exactly the gearbox systems that will operate well in their specific environment."

Drago and Lenski have worked with each other for over 40 years in the gear industry spanning an array of concentrations that include aerospace, consumer products, medical devices and various industrial applications. They have performed extensive R&D developing new gear and bearing designs. Drago currently serves as the chief engineer and founder of Drive Systems Technology, Inc., a mechanical power transmission consulting organization, and Lenski holds the title of chief bearing specialist for the firm.

This seminar "addresses a broad range of topics that include design,

continued

Ray Drago.

fabrication, lubrication and handling of gears and bearings that will have a great impact on the operation and service life of gearboxes," Drago says. "Interactions between the gears and bearings will be addressed and what should be done to optimize the gearbox for maximum life of both the gears and bearings."

The seminar's outline topics cover analysis of both gear and bearing design and failure, material and manufacturing related issues such as inclusions, hydrogen embrittlement, residual tensile stresses, porosity, heat treating and grinding burns. Preventive measures will also be covered in the seminar as well as appropriate cures—simple and complex ones—to eliminate root causes of failure.

"For gears, a simple change might be specifying an improved surface finish on the gear teeth while a more complex solution may be a tooth redesign to include appropriate lead and profile modifications to accommodate system and gear tooth deflections," Drago explains. "In many cases, simple cures can result in significant improvement in gearbox performance if incorporated early in the design. Complex cures are not recommended for all gearboxes but should be known to the designer in case these are the only way to improve or correct the problem."

Drago examines a SAG mill gear, which is about 35 feet in diameter, at a copper mine in Chile in order to determine the cause of tooth surface fatigue damage.

The instructors choose to limit the course size to allow ample opportunities for questions, either during class time or during the breaks and lunches, at which times Drago and Lenski remain with the group for this explicit purpose. They invite questions after each topic and hold a Q&A exchange to encourage a group experience and foster a class discussion.

Lenski and Drago also use photographic data presentations coupled with case studies they are intimately familiar with. "Every situation discussed is one in which we have been personally involved and thus we can speak with first party knowledge and insight. The photos presented are from actual case studies done for various problems ranging from wind mills, boats, satellites and space applications to gearboxes used in mines, steel mills and many other commercial applications," Drago says.

The next Gearbox CSI is being held at the Sheraton Sand Key Resort in Clearwater, Florida from November 12–14. Registrations costs start at $1,695 for AGMA members and range up to $2,395 for non-members. Fees encompass the educational materials, scheduled meals, an opening evening network reception and an AGMA certificate once the seminar is completed.

This year's course has sold out and has a waiting list, but AGMA is looking into expanding the seminar size. The 2009 course is scheduled for October 13–15. In consideration of the Gearbox CSI's popularity, AGMA is exploring options as far as offering it more, according to Jan Potter, AGMA vice president, membership. For more information contact Potter at (703) 684-0211.

November 4–6—CNC Machining Clinic. Doubletree Hotel Chicago-Oak Brook, Oak Brook, IL. Today, CNC machines are found almost everywhere, from small job shops in rural communities to Fortune 500 companies in large urban areas. There is hardly a facet of manufacturing that is not in some way touched by what these innovative machine tools can do. Examine purchasing decisions, capital equipment, fundamentals, total productive maintenance (TPM), optimization and more. For more information contact: Society of Manufacturing Engineers, (313) 425-3000 or visit *www.sme.org/cnc*.

November 11—Configuration Technical Seminar. Lancaster Host Conference Center, Lancaster, PA. This seminar, conducted by Misumi USA, is designed to assist engineers to overcome time and performance pressures that occur in custom machine building. The seminar's title is "Designing a Better Machine Faster with the Configurable Component." Misumi's manager of product development, Mike Melone, an engineer with experience in configurable components, is leading the tutorial. After his presentation, attendees will be allowed to pose questions for him. The seminar includes complimentary lunch and some give-away items. For more information and to register, visit *www.misumiusa.com/techseminars.aspx*.

November 12–14—Machine Shop Workshop. Doubletree Hotel Chicago–Oak Brook, Oak Brook, IL. Sponsored by *American Machinist* magazine, the 2008 Machine Shop Workshop is a two day event with peer presentations, best-practices and an emphasis on solving real-world problems. Presentations will be made by industry experts such as Fred Young, CEO of Forest City Gear and Matt Guse, VP and owner of M.R.S Machining. Roundtable discussion will be held for the opportunity to solve participants' business issues, get advice and learn troubleshooting tactics. Some of the sessions include preventive maintenance, improving quality through lean techniques, coping with material uncertainties and shop strategies for addressing the challenges from Southeast Asia. For more information, visit *www.machineshopworkshop.com*.

November 18–19—AmCon, Greensboro Coliseum, Greensboro, NC. AmCon is a contract manufacturing expo for all job shops and contract manufacturers that provide custom metal, plastic, rubber or electronic parts and related manufacturing services to OEMs. Attendees include top level purchasing, engineering and production managers who are directly involved in buying custom contract manufacturing services. Representatives from companies of all sizes attend from a range of industries, often with blueprints in hand. The AmCon shows occur regionally throughout the year. For more information, visit *www.amconshows.com*.

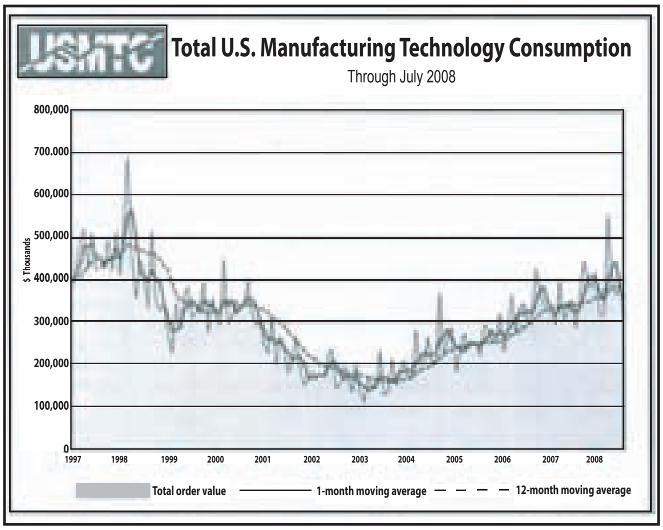
November 18–19—Aerospace Measurement, Inspection & Analysis. Fort Worth Convention Center, Forth Worth, TX. This two-day event is held alongside Aerotest America, and its focus is the unique quality challenges facing those in all levels of aerospace manufacturing. Manufacturing professionals coalesce to share insights, the latest technologies and process updates directed towards anyone responsible for quality in aerospace, including manufacturing directors and managers, new product and process development engineers, design engineers, quality engineers and manufacturing engineers. Measurement and inspection are the topics for the first day, covering technology updates, optimizing use of existing technology and case studies. The second day is devoted to data collection and analysis including technology/software updates, how to better use existing tools, process updates and improvements and case studies. For more information, contact the Society of Manufacturing Engineers at (800) 733-4763.

November 19–21 AWEA Wind Energy Fall Symposium. Desert Springs, a JW Marriott Resort and Spa, Palm Desert, CA. The fourth annual AWEA Wind Energy Fall Symposium expects to attract over 600 wind energy professionals in all areas of the industry. Attendees learn more about wind business in addition to policy and technical issues. Two pre-conference seminars will take place. At the Fundamentals of Wind Energy seminar a tutorial will cover the basics of utility-scale wind energy for industry newcomers. The Utilities and Wind Power seminar addresses issues with electric utilities as they increase integration with wind power. For more information, visit *www.awea.org/events/syposium08*.

December 8–10—Gear Manufacturing Technology Course. RP Machine, Statesville, NC. The Gear Consulting Group offers this three-day AGMA course to teach theory and practical aspects of gear manufacturing. Participants will learn about everyday problems and appropriate responses to troubleshooting. Instructors Geoff Ashcroft and Ron Green of the Gear Consulting Group will cover material including gear theory, inspection, manufacturing, hobbing, shaping, tools, production estimating, hard finishing and gear shaving. Course tuition includes all necessary materials, an AGMA reference manual and a certificate of completion. For more information, contact the Gear Consulting Group at (269) 623-4993.

Manufacturing Technology Consumption

STUMBLES IN JULY



U.S manufacturing technology consumption dropped 21.5 percent from June to July according to the American Machine Tool Distributors' Association (AMTDA) and the Association for Manufacturing Technology (AMT). Totaling $303.44 million, manufacturing technology consumption is up 5.7 percent from the figure for July 2007, and the year-to-date total is approximately $2.66 billion, which is up 15.5 percent from the same figure in 2007.

"Despite the consecutive 1.3 percent increases in durable goods orders for June and July, manufacturing technology orders turned down sharply in July, with much of the downturn in fabricating and forming," says Peter Borden, AMTDA president. "However, 2008 remains stronger than both 2006 and 2007. The fall trade show season will provide a clearer indication of the industry's health following IMTS and Fabtech."

From a regional basis, the northeast region's manufacturing technology consumption was down 11.8 percent from June to July, at $46.43 million. This value is 13.9 percent higher than the respective figure for July 2007, but the year-to-date total for 2008 is 1.8 percent less that it was in 2007.

Manufacturing technology consumption in July for the Southern region stood at $41.51 million, which is 41.0 percent less than it was in the previous month and 7.8 percent less than the total was in July 2007. However, the year-to-date total for the South is up 39.0 percent in comparison to 2007.

The Midwest region's manufacturing technology consumption was down 37.3 percent in comparison to June of this year and 14.4 percent less than it was in July 2007. The year-to-date total is 33.4 percent higher than it was at the same time in 2007.

At $100.17 million, July manufacturing technology consumption in the Central region was up 14.8 percent from June and 34.3 percent higher in comparison to July 2007. The year-to-date figure is 5.3 percent more than it was the previous year.

July manufacturing technology consumption for the Western region was down 27.1 percent with a total of 35.15 million. It was 6.5 percent higher than July 2007, and the year-to-date total is 2.7 percent less than that of 2007.

North America's Largest Gear Measuring Machine Purchased

British Columbia-based Vancouver Gear Works Ltd. has placed an order for what is believed to be the largest gear measuring machine in North America from Wenzel GearTech Division of Wenzel/Xspect Solutions, Inc. of Wixom, MI. The machine is specifically designed for inspecting windmill and

continued

gearboxes, and it is scheduled for delivery in March 2009.

"In the last several years we have focused more resources on the energy-related industries where gears and gear-related products are prominent," says Jim Mantei, general manager of Vancouver Gear Works. "We have invested upwards of $10 million during that time for state-of-the-art gear manufacturing equipment and ultimately found the need for better methods of inspecting and verifying those precision tolerances. We have recently secured contracts for the manufacture of large gears and the corresponding parts that are used in the production of windmills. This required purchasing new state-of-the-art inspection equipment to provide the necessary guarantee of our levels of quality."

The Model LHF GMM 30.60.20 is capable of measuring gears up to 3 meters in diameter in addition to large housings and components. It is a traveling bridge-type machine with a measuring range of 6 meter x 3 meter x 2 meter. The GMM uses two three-meter measuring zones. One has a 1,000 mm hydrostatic rotary table, which can accommodate gears up to 30,000 lbs. The other zone has a cast iron surface plate to inspect housings and other components using the traditional prismatic method.

"It took a number of concurrent actions that brought Wenzel and Vancouver Gear Works, Ltd. together for this unique capital equipment purchase," says Keith Mills, president of Wenzel/Xspect Solutions. "Wenzel/Xspect Solutions recently entered into a sales partnership with Great Lakes Gear Technologies, located here in Canton, Michigan, who over the years, has worked closely with Vancouver Gear."

Ray Mackowsky, president of Great Lakes Gear Technology, says, "Because of our company's sales know-how in gear manufacturing, Vancouver Gear looked to us to provide some recommendations for their specific needs. It was a fairly simple task, since there is only one company in the world that has the size and production-proven accuracy of gear measuring equipment that Vancouver needed."

Northstar Aerospace Expands,

CREATES 70 U.S. JOBS

Increasing its production of gears, components and spare parts for the CH-47 Chinook helicopter and Rolls-Royce programs, Northstar Aerospace Inc. is bringing 70 new jobs to three facilities in North America. About 10 machinists will be added to the facility in Anderson, Indiana; the Windsor, Ontario site will grow by about 20 machinists and support staff; and 40 new employees will join the Chicago site for CNC machining, ID/OD grinding, shaping, hobbing and inspection in addition to support staff in the heat treat, plating and assembly areas.

Orders for the CH-47 gears and components this year have reached $114 million, $60 million of which was announced in June alone. The helicopter orders in 2007 stood at $167 million.

"We are focused on transforming our growing backlog into production to meet the needs of our customers," says Glenn Hess, president and CEO. "The hiring campaign and investment in state-of-the-art, automated machining centers at our Chicago and Phoenix sites will help us increase production to needed levels."

Hanover Gear

BOUGHT BY ABUNDANT

The acquisition of Hanover Gear Manufacturing of Hanover, PA by RP Machine/Abundant Manufacturing Inc. includes machinery, equipment, inventory trade name and existing business; however, the Hanover property was excluded from the transaction. The acquisition should bolster the Abundant revenue by 30 percent and add 12 percent to the RP Machine/Abundant family, according to Greg Watson, CFO. Watson also estimates 12–14 jobs that will be created in response.

The purchase allows the company to continue growth patterns that started in 2005, according to Richard Piselli, CEO of RP Machine/Abundant Manufacturing Enterprises. In 2005 the company expanded its inventory in off-highway gears and began manufacturing and retrofitting larger gear machinery while entering the wind energy market, Piselli says.

Hanover Gear Manufacturing specializes in contract gear manufacturing, especially spur and helical gears, splined parts and other power transmission components.

"The acquisition of Hanover gear creates an incredible synergy between the product lines offered by Hanover and Abundant," says Kevin Smith, president of Abundant Manufacturing. "Hanover's equipment complements our base of equipment very well. Bill Fuss [president of Hanover Gear] has developed an industry presence over the last 34 years that we hope to capitalize on. Mr. Fuss will join the Abundant/Hanover team as a sales executive in not only maintaining but also growing the business."

New President

APPOINTED AT NIXON GEAR

Dean Burrows

Dean Burrows was named president of Nixon Gear, a Gear Motions, Inc. subsidiary based in Syracuse, NY. The company manufactures and supplies gears with emphasis on precision ground spur and helical gearing services. Burrows brings 20 years of operations experience in industries that include automotive, medical device and consumer products.

He previously worked as vice president of operations for the Marietta Corporation. He also served as director of supply chain for the residential light commercial international division of the Carrier Corporation—a division of United Technology—where he helped lead an acquisition integration while living in France.

Burrows received a master's degree in engineering management and a bachelor's degree in industrial engineering from the Rochester Institute of Technology. Nixon Gear expects Burrows to bring a fresh perspective and help the company develop as a global custom gear manufacturer of spur, helical, gear cutting and gear grinding services, according to a press release.

GearTec

JOINS UNITED STARS

Beloit, Wisconsin company United Stars Inc. announced the acquisition of GearTec Inc., of Willoughby, Ohio.

continued

GearTec produces short-run gears up to 100 inches in diameter. The company hobs, shapes, mills and grinds helical, double-helical, herringbone, worm and internal gear types to AGMA class 8 or better standards. GearTec gears are used in industries such as wind power, offshore drilling, tunnel boring, OEM, steel, crane and others.

United Stars already offers a full range of gear manufacturing services to several markets with United Gear and Assembly, which manufactures medium- to high-volume, smaller-sized gears that are found in construction equipment, agriculture, transportation, engines, transmissions and hydraulic pump and motor applications. United Stars also owns The Electric Materials Company, which produces copper components, as well as United Industries and United Stainless, which produce stainless steel tubing.

"GearTec is a highly respected company with a strong reputation for excellence in very large and precise custom gears for a unique group of end users," says Rich VanLanen, president of United Stars. "Having both GearTec and United Gear and Assembly on the same team gives us a powerful one-two punch in the gear-making arena. It is a very exciting time for the United Stars group of companies."

John Grazia, president of GearTec, comments that "Additional capital and operating support will enable us to take advantage of growth opportunities now and in the coming years. At the same time, our products complement the quality product lines offered by United Gear and Assembly, which enables us to jointly satisfy a total range of gearing needs."

Export Achievement Award

PRESENTED TO SUNNEN


Matt Kreider, president of Sunnen, accepts the Export Achievement Award from Mary Joyce, Midwest network director for the U.S. Commercial Service in Chicago.

The U.S. Department of Commerce's Commercial Service honored Sunnen Products Company with an Export Achievement Award in a ceremony at IMTS in Chicago. The award is given to companies that have succeeded in making a first export sale or expanded into new markets by Commercial Service assistance.

"U.S. manufacturers like Sunnen Products represent some of the best and most innovative companies whose products are working to keep America at the forefront of global trade and competitiveness," says Mary Joyce, the Midwest network director for the U.S. Commercial Service who presented the award to Sunnen's management.

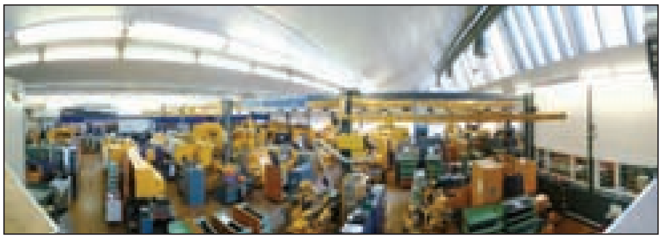
Sunnen benefited from the Commercial Service's export counseling with new sales in Brazil, India and Italy. Sunnen also has global facilities in France, China and Switzerland.

"Our founder, Joe Sunnen, strongly believed in serving all customers, regardless of their location around the world, and initiated a variety of export programs 70 years ago that formed the foundation for our success today," says Mike Haughey, COO of Sunnen. "Today, we continue to follow his vision by expanding our international presence and, as a result, grow our sales revenue. A high quality product has value, no matter where in the world a need rises."

Kaiser

CELEBRATES 60 YEARS OF PRECISION TOOLING

Heinz Kaiser AG rang in its 60th anniversary with a reception of employees, customers, salespeople and licensed partners at the Rümlang, Switzerland facility. Founded in 1948



by Heinz Kaiser, the company develops and manufactures high-precision modular tooling systems. The celebratory event included small-group tours of the facility in addition to refreshments and entertainment. Kaiser employs 115 people, and its brands include BIG Daishowa, Speroni, Unilock and Sphinx.

Schafer Gear

ADDS MANAGEMENT

Kevin Piefer has been appointed quality manager at Schafer Gear Works, and Carl Wedig was named manufacturing engineering manager. Both positions will be served at the company's Rockford, IL production facility.

Piefer's main responsibilities will be to develop and execute quality procedures in accordance with ISO 9001:2000 standards. He received a bachelor's degree in business management from North Central College in Naperville, IL. He previously served as quality manager for Orchid Monroe in Monroe, WI.

Wedig's position will focus on planning, developing and

continued

applying maintenance of manufacturing methods, processes and operations for both new and existing products. Prior to this appointment, Wedig served as plant manager for Raycar Gear and Machine, also in Rockford.

"Our company is well positioned for a new wave of growth," says Stan Blenke, executive vice president of Schafer. "With these appointments, we now have two leaders who can help us improve our production efficiency and quality of our products as well as help our company grow."

Moventas

PLANS NEW INVESTMENTS, MAKES EXECUTIVE APPOINTMENTS

Intending to triple wind turbine gear production capacity by the start of 2011, Moventas announced plans to invest 100 million euros with new manufacturing sites in the Midwest United States and Central Finland. The company has also made changes to its executive staff.

In order to fortify its position in the growing U.S. wind energy market, Moventas is establishing an assembly plant in Faribault, MN, about 45 miles south of Minneapolis. The company expects this plant to create about 100 new jobs in assembly and testing.

A new production plant is planned for Jyväskylä, Finland, which will focus on producing ring gears. The addition of this site expects to create 200 new jobs. Moventas is also building a factory in the Eteläportti area in Jyväskylä, Finland.

Moventas plans to begin breaking ground at these new facilities at the end of 2008 for production to begin at the end of 2009, and full capacity is anticipated for the start of 2010 in Jyväskylä and 2011 for the other locations. Once full production begins, the company expects capacity to increase to 7,000 MW.

"The decision to invest is a strong signal that Moventas wants to capture the growth of the business and to meet customer demand," says Ilkka Hakala, CEO and president of Moventas. "Expansion of production in North America is a significant step closer to our customers, and it strengthens our ability to meet the after-sales service demands in the market. On the other hand, the decision to build a third production site in Central Finland indicates that we want to develop our expertise and technology in Finland also in the future."

Moventas also recently appointed Ahti Ahonen as executive vice president, industrial gears. He will also fulfill the position of development director for international operations, a position responsible for developing international production and strategic investments.

Ahonen's former position, executive vice president, wind gears, is being filled by Jukka Jäämaa, who is also deputy to the CEO and a member of the management team.

Solar Atmospheres

EXPANDING IN CALIFORNIA

A West Coast expansion has been in the cards for Solar Atmospheres for some time, and the recent hire of Derek Dennis as president of the new California heat treating company lays the initial foundation. Dennis brings lifelong ties to California, managerial experience and knowledge of the metals business to the position.

Roger Jones, the corporate president of Solar Atmospheres, comments "Derek comes to Solar with a wealth of heat treating and management experience, primarily from the aircraft and petrochemical business as former director of operations for Valley Metals and Manufacturing Manager for Jet Products, both of San Diego. He also served in the U.S. Marine Corps as an aerial navigator and crew chief."

Derek is immediately charged with the task of locating a site for the new plant. The primary focus in the search is the area just east of Los Angeles, which is known to have an experienced labor pool and several potential plant locations.

"Our aim is to install large, high-tech furnaces and provide high quality operations, not currently available on the West Coast," says Bob Hill, Solar's president of Western Pennsylvania operations who was part of an advanced marketing research effort for this project.

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16. Publication of Statement of Ownership for a Requester Publication is required and will be printed in the November/December 2008 issue of this publication.

17. Signature and Title of Editor, Publisher, Business Manager, or Owner: Michael Goldstein / Publisher — Date: 9/26/08

I certify that all information furnished on this form is true and complete. I understand that anyone who furnishes false or misleading information on this form or who omits material or information requested on the form may be subject to criminal sanctions (including fines and imprisonment) and/or civil sanctions (including civil penalties).

PS Form **3526-R,** September 2007 *(Page 2 of 3)*

Overton Gear

ACQUIRES CHICAGO GEAR

One of the largest full-service gear and gearbox suppliers in the world emerges from the purchase of Chicago Gear-D.O. James Corporation by Overton Gear Corporation. The combination of these companies should expand their product lines while increasing capacity and efficiencies.

Chicago Gear-D.O. James, founded in 1888, has produced more than 350,000 large, specially designed gearboxes found in places like the Panama Canal, the Mississippi River locks and dams as well as the Mission Space ride at Epcot Center. Chicago Gear serves as the open gearing component of the company, and their largest gears surpass 100 inches in diameter.

Overton Gear is an employee-owned company that has been in business more than 50 years. Headquartered in Addison, IL, Overton manufactures custom gears for marine, off-shore, locomotive, mining, wind energy, transportation and construction industries. The company previously acquired the Illinois Gear Corporation and has a strategic alliance with Klingelnberg GmbH to supply large spiral bevel gears worldwide.

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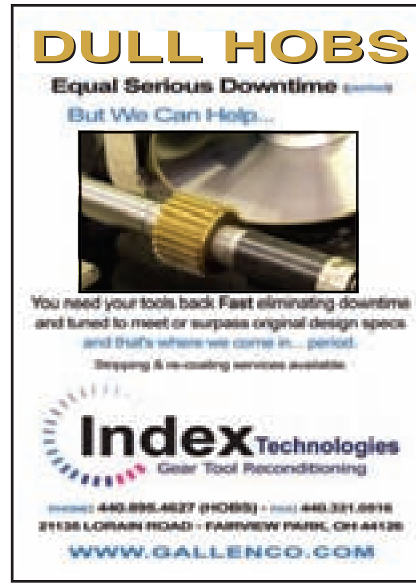
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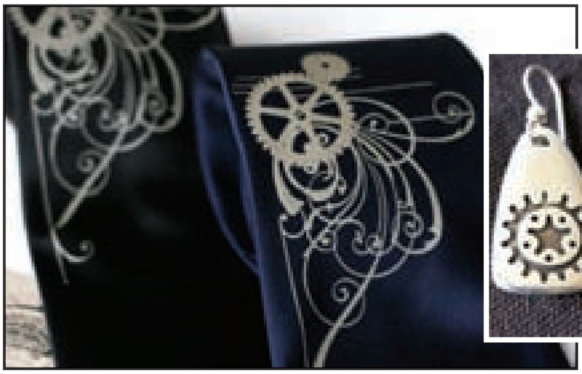
2008 Holiday Buyer's Guide

Just Say "No" to Gift Cards or Fruit Baskets this Time Around



There aren't many superstores sandwiched between Best Buy and Barnes & Noble that cater directly to gear enthusiasts. In order to find a gift for the gear head in the family, it's going to take a little time and creativity. This year, we've researched a variety of websites with the sole purpose of finding gears. Here are just a few of your options in 2008:

www.giftsforengineers.com

This multipurpose website offers a plethora of gear-related goodies like gear-themed clocks, necklaces, books and even a box of mechanical engineering chocolates featuring chocolate gears, nuts, bolts and T-squares. There are also great novelty items including an engineering rubber ducky that sports a hard hat, an orange safety vest and a roll of blueprints under his wing.

www.cyberoptix.com

Nothing says "I have no idea what to get you" like the gift of a neck tie. But it doesn't have to be like this. The Cyberoptix Tie Lab offers up the slogan, "TIES THAT DON'T SUCK!" The Victorian gear design boasts a variety of colors in both silk as well as a narrow microfiber, a wonderful accessory for dress-up Mondays at the office.

www.geeky-gadgets.com

This website specifically caters to the inner geek in all of us by highlighting the latest and greatest gadgets, gizmos and consumer electronics on the market. From gear-related watches and desk clocks to information on how to make your own gadgets, this is the place for the wanna-be engineer in the family that takes it all apart, but has no clue how to put it back together.

www.steepplanet.com

Founded in Oregon, Steep Planet features selected products with the active, outdoor lifestyle in mind. Mountain sports, cycling, fitness and meditation are just a few of the activities highlighted on the website. The sterling silver gear burst earrings are a perfect fit for readers that fondly appreciate fashion as well as the great outdoors.

www.fatbraintoys.com

The future manufacturer in the family can start learning the craft with a variety of games, puzzles, magnets and play sets that feature gears. Of special interest are the Make-a-Clock kits that allow kids to see working gears in motion and the Techno Gears Marble Mania Extreme, a motorized obstacle course they can construct using plastic gears.

www.amazon.com

If knowledge is power, then the right book is exactly what the gear enthusiast needs. Amazon offers many titles such as *Pulleys and Gears (Simple Machines), Handbook of Machining with Grinding Wheels* and *Precision Manufacturing*. There are also plenty of titles available on lean manufacturing, supply chain management and the global economy.

www.learningresources.com

Another great option for kids is this educational toy company that offers gear workstations, construction sets and activity books. Most of the sets are motorized, allowing children to create a variety of unique mechanical designs that focus on science and technology.

www.sme.org

Fresh off our Best Supporting Gears Addendum, this list of short films feature gears in a more instructional setting. Those new to the industry would benefit from a copy of several titles including *Gears and Gear Manufacturing, Basics of Grinding*, *Milling and Machining Center Basics, Threading Basics* and *Turning and Lathe Basics*, all available on DVD through the Society of Manufacturing Engineers.

www.winzelergear.com

Interested in being a little more creative this holiday season? Perhaps browsing the Winzeler Gear Art Gallery online will inspire your own personal projects featuring gears. Whether your interests lie in fashion, art, photography or architecture, a gift that you take some time and effort to create is better than anything you can find in a store.

www.geartechnology.com

At the end of the day, the best gift might just be the magazine in your hand. Wrap it up, tie a bow around it and give the gift of *Gear Technology*. Sure, it's a shameless plug, but we left just enough room at the bottom of the page for some gratuitous self-promotion.

In conclusion, we'd very much like to put an end to the holiday parties where family and friends receive gift cards, fruit baskets or "your-name-here" key chains. Please feel free to join us in our struggle. If you have any gear-related gift ideas, send them to ***publisher@geartechnology.com***, and once again, happy holidays.

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