July 2008

GEAR TECHNOLOGY

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The Journal of Gear Manufacturing

Features

- New-Generation Machine Tools Raise Multi-Tasking to New Levels
- Lean—A Never-Ending Journey with Ample Rewards along the Way

Technical Articles

- Optimizing Tooth Microtopographies for Noise/Stress Reduction Over Broad Torque Range
- Kinematics of Conical Involute Gear Hobbing

Plus

- Voices: Thinking Big—and Surviving—in Today's Global Market
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THE GEAR INDUSTRY'S INFORMATION SOURCE

GEAR TECHNOLOGY

July 2008

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PUBLISHER'S PAGE

$ Buy Now $

Uh, oh. Oil is marching toward $150 a barrel. In May, unemployment made its largest monthly jump in 22 years. The stock market is increasingly volatile. The dollar keeps falling. For many in the United States, economic bad news abounds.

So what should gear manufacturers do? I say go shopping! Buy machine tools. Invest in your company's technology.

What do you mean, it doesn't sound like a great time to go shopping? For a couple of reasons, it's exactly the right time.

Despite the gloomy economic outlook, much of American manufacturing is thriving—especially old-world industries like mining, steel making and even agriculture. Also, thanks to the weakened dollar, companies that export manufactured goods are finding increased demand, especially those that serve the exploding markets in India and China. Caterpillar, for example, has posted record profits four years in a row, and 2008 looks like it will be another good year.

If your company is benefiting from this manufacturing boom, it's probably because you've already made machinery investments.

But even those of you whose outlook is more challenging—those whose fortunes are tied more closely to the American automobile industry, for example—should consider making machinery investments today instead of waiting for better economic times.

No matter how the economy is affecting your company today, there may never be a better time to invest in capital equipment, thanks to Uncle Sam and the Economic Stimulus Act of 2008. While most of us are familiar with the personal income tax rebates authorized by the act, the business incentives may be even more important, especially for manufacturing companies.

For example, the act provides for a 50 percent bonus depreciation on new capital expenditures—including machine tools—purchased and put into service in 2008. That's 50 percent in addition to the 14 percent already allowed by law.

The 2008 act also increases the amount that small businesses can write off for new and used equipment purchases. The expensable amount almost doubled from $128,000 to $250,000 for qualifying equipment purchases up to $800,000—the price at which the allowable expenses begin to phase out. The phase out amount increased from $510,000.

The incentives for capital investments seem to be working. According to the latest figures from the AMT/AMTDA's Machine Tool Consumption Report, machine tool purchases are up almost 20 percent compared with last year. In March, monthly machine tool consumption spiked above $500 million for the first time since 1998.

Ordinarily at this time of year, I'm urging you to make plans to go to IMTS so you can see the latest technology and learn how to improve your company's productivity and capabilities—and I will continue to do so.

Those of you whose business is good probably already know the value of continually investing in the latest technology. It's probably what's been keeping you busy. But if you're suffering now because the industries you serve are suffering, making those investments may provide you with the technology to expand your offering, to branch out to other industries and protect yourself from this type of situation in the future.

But you can't afford to wait until IMTS if you want to take advantage of the incentives our government is currently providing. By September, there will be a mad rush, with everybody trying to make last-minute machine tool purchases at the same time. Your competitors will be throwing money at the machine tool suppliers, begging to get their orders to the front of the line and delivered before the end of the year.

Even if you're not in a position to make decisions about machine tool purchases, you're probably in a position to make recommendations, or you're the one who might benefit most from improved technology. So if you've been asking for new equipment—or if you've been waiting for approval on a requisition already submitted—the current incentives give you an excuse to go knock on the boss's door.

If you are the decision-maker, you need to call your suppliers now and see what they can do for you. Consult your accountant to see how much you could save by buying now instead of in 2009. You'll probably find that you can't afford to wait. These incentives are a gift that could save your company tens or hundreds of thousands of dollars in taxes.

You're in the manufacturing business. Your competitive edge, the quality of your products—your livelihood—depend on continual investment in technology. Why not make that investment when it's most advantageous?

Michael Goldstein,
Publisher & Editor-in-Chief

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Entrepreneurialism and Survival in the Global Market

By Joseph L. Arvin, President, Arrow Gear Company

For those who have manufactured products in the United States over the past 30 years, I'm sure the words "global competition" cause a very uneasy feeling in the pit of their stomachs—and with good reason. Since the late 1970s, American manufacturers have been pummeled with the low prices of one foreign source after another.

At Arrow Gear, where I have worked since 1972, we have seen this seemingly endless parade of foreign competition. I well remember losing our first customer to a Japanese supplier in 1979. After Japan, it was the general Pacific Rim, then Eastern Europe. Now with India and China just getting under way, I don't anticipate an end to the formidable forces of foreign manufacturers in the near future.

It's certainly no secret that numerous American companies have gone out of business in the face of foreign competition in the past two to three decades. On the other hand, many manufacturers have survived. In the case of Arrow Gear, we have been fortunate to not only survive, but to thrive. So, what is the difference between those organizations that made it and those that did not?

In 2006, I spoke at the Illinois Institute of Technology (IIT) in Chicago. As a member of IIT's Manufacturing Education Advisory Board, I was more than happy to share my thoughts and experiences with these students. The subject was entrepreneurialism in manufacturing and how to survive in the global market.

Entrepreneurialism Defined

In preparing for my presentation, I first asked myself the question, "What is an entrepreneur?" I've always assumed that this was someone like James J. Cervinka, one of the founders of Arrow Gear. He and his business partner Frank E. Pielsticker had the vision of starting a gear company. They took the risk of investing their money—and in the end, saw profits from their risk. Certainly, this must be the definition of an entrepreneur?

I turned to Webster's dictionary for the official definition—and was somewhat surprised by what I saw. According to Webster's, an entrepreneur is a person who "organizes, manages and assumes responsibility for a business or other enterprise."

By this definition, the term entrepreneur is expanded to roles other than that of a company's founder. In fact, as an employee, I had been charged with the role of entrepreneur through my assignment of organizing, managing and assuming responsibly for the company's growth. Taking this even further, I can cite many others at my company—including supervisors and managers—that fit this definition. After all, the success of a company translates not only to benefits for the owners, but to each individual that earns a living there.

> "The field of American manufacturing has seen the toppling of many industry giants through a mix of changes of competition and the inability to respond by returning successfully to the vision phase."

Entrepreneurship within a company includes those activities required to identify and introduce major innovations. Innovations can involve product, production, business systems or social systems.

I found this connection between entrepreneurialism and the general employees of a company to be very interesting. But this realization led to other aspects in my analysis of the subject.

Life Cycles of a Company

In understanding the relationship between entrepreneurialism and the success of a company, it is essential to understand the life cycles of a company. Many authors have described these cycles, so I will paraphrase from several sources.

First is the vision phase. This is when the founders conceive the concept of the business. Next is the startup phase. Here the enterprise begins and risk is assumed, typically in the form of time and financial investment. Following this is the success phase. This is when systems and people are in place and delegation becomes more prevalent.

However, the next phase—bureaucracy—is critical. In this phase, systems are becoming inefficient, and the company runs the risk of becoming a lumbering giant—unable to react effectively to changing markets and trends. Since a company cannot remain stagnant for long, the bureaucracy phase is followed either by further decay and decline or by the return

continued

to the vision phase as a way to continue building on the successes that have been accomplished thus far.

The field of American manufacturing has seen the toppling of many industry giants through a mix of changes of competition and the inability to respond by returning successfully to the vision phase. One of these giants was the Ohio-based machine tool manufacturer Warner and Swasey.

I visited Warner & Swasey in the late 70s and found that they were building new CNC machine tools with 50-year-old equipment. In 1981, I visited two major machine tool builders in Japan—Mazak and Toyoda. By sharp contrast, they were producing CNC machine tools using the latest state-of-the-art CNC equipment. Notably, Warner & Swasey, which was a clear leader in machine tool production, closed its doors in the late 1980s.

On the other hand, there are many examples of manufacturing giants that began to see their market share slip to foreign competition, yet they were able to return to the vision phase and effectively reinvent their operation and products in order to survive.

When I came to Arrow Gear, the company was about to enter the bureaucracy phase. The company owners recruited me as plant manager to assist with the company's next phase of growth. They understood that there was a need to return to the vision phase. As James J. Cervinka has said on many occasions, "Dormancy is actually the first step toward decline."

Returning to the Vision Phase

Through the early years of Arrow's history, the company was well known for expertise in spiral bevel gears. While this expertise remains today, Arrow has continually added new capabilities and product focus through a long-standing tradition of reinventing itself.

In the mid 1970s, Arrow began investing heavily in new equipment and the latest manufacturing technologies. This philosophy has continued through the present day.

As for product offering, the next phase of change also came in the mid-1970s with the addition of spur and helical products. In the early 1980s came expansion of the aerospace customer base, and by the mid-1980s, Arrow began pursuing the European market. By the mid-1990s, one third of Arrow's products were being shipped to Europe.

And by the late 1980s, Arrow saw the opportunity to move into complete gearboxes for the aerospace market and began

 VOICES

producing prototype and production units.

A key aspect of Arrow's success has been the high degree of technology in our production facility. In the early 1990s, we introduced the world's first fully integrated closed-loop system for design, manufacturing and inspection. This involved the direct linking of design computers with machine tools and inspection systems. The increase in efficiency and productivity was dramatic.

Another benefit of this proactive, state-of-the-art approach to manufacturing was that Arrow could now offer advanced capabilities for design and development—a capability that even now is offered by only a select handful of gear companies. Our decision to implement this technology was based on our observations of the industry. In the late 1980s, we saw that aerospace OEMs were getting away from doing their own gear design. As this trend became more widespread, Arrow was positioned to provide this valuable service to our customers, and this capability continues to be one of Arrow's major competitive strengths.

"How long can this parade of foreign competition last? Will there come a time when there is no longer a region of the world with significantly lower labor costs? Unfortunately, this dynamic will probably remain as the status quo for quite some time."

Finally, in the 2000s, we started going after overhaul and repair work and spare and replacement parts for military and commercial aircraft.

In all, Arrow Gear today is very different than it was in the early 1970s. I am convinced that had Arrow not made these changes, we too would have closed our doors many years ago. This ongoing reinventing of ourselves has been instrumental in our ability to survive the onslaught of foreign competition. The success of this approach is clearly demonstrated by Arrow's survival and growth during a period when the overall U.S. gear industry has shrunk by two thirds.

Understanding the Competition

In today's gear industry, there are primarily two basic types of work—high-volume/low-quality and low-volume/high-quality. Early on, we saw that the work going offshore was high-volume/low-quality. After all, this type of work is not as dependent on skilled labor and was more easily sent offshore. This is exactly what happened—first with Japan, then the Pacific Rim, then Eastern Europe, and more recently China and India.

Arrow saw this trend early on only because of our return to the vision phase, when the current market was assessed and the question was posed, "How do we change in order to compete within a changing global market?"

How long can this parade of foreign competition last? Will there come a time when there is no longer a region of the world with significantly lower labor costs? Unfortunately, this dynamic will probably remain as the status quo for quite some time.

To explain this, we need to look at the first serious foreign competitor, which was Japan in the 1970s. Japan had a much lower labor rate than the United States. In fact, on a tour of Japanese manufacturing facilities in 1981, I was told that the average labor rate at that time was $3.25 per hour (in U.S. dollars), while Arrow was paying $12.75. But eventually, the standard of living rose in Japan, and increased labor rates followed. By 1997, the labor rates in these Japanese facilities had climbed to $18.00, which was the same as Arrow Gear. This process took approximately 16 years—and it was a very difficult playing field during that time.

But as competition with Japan began to level, the cycle started all over again with manufacturers throughout the general Pacific Rim. As with Japan, the standard of living and labor rates in these areas are increasing to a more level playing field with U.S. labor.

Now, we face fierce competition with China, which obviously has a low labor rate. I was in China in the fall of 2006. At that time, the average wage for a factory worker was $1.25 per hour (in U.S. dollars); the engineers and supervisors were paid $2.25 per hour, about $5,000 a year. But how long will it take for China to catch up with U.S. labor rates, leading to a more level playing field? Unfortunately, the answer most likely lies in the volume of the population.

Japan's population during the period of 1970 through 2000 was an average of around 115 million. As stated before, it took 16 years for the increased manufacturing revenue to impact the standard of living for 115 million citizens.

By contrast, with its 1.3 billion citizens, China has more than 10 times the population of Japan. The question is, if it took Japan 16 years, how long it will take China's labor rates to rise to levels comparable with the United States? I believe it will take much longer than 16 years. This means that American manufactures will have serious competition well into the future.

The Life Cycle and Leadership

As it is most certain that foreign competition will not fade in the near future, it is essential that American manufacturers take carefully planned steps to remain competitive.

Foremost among these measures is the avoidance of succumbing to the bureaucracy phase. Instead, it is essential to continually return to the vision phase. This is easier said than done, however, and a key part of this process is leadership. To

continued

avoid the gap between senior management and the frontline workers, leadership requires effective communication skills in guiding the organization's return to the vision phase. It requires successful promotion of the entrepreneurial spirit throughout the company.

The old "Theory X" management style was to dictate your directions to subordinates and make sure they did what they were told. While this might work to some degree, more often, it will merely create a company of drones that only do what they're told. If all managers were omnipotent geniuses, this approach would be effective. But since managers don't always possess these qualities, it is essential to rely on the resources of people in the organization.

A company that can successfully return to the vision phase will have a roster of employees who are active resources with the ability to organize, manage and be responsible. The key to this is communication, empowerment and accountability.

When a manager is faced with a challenge that requires directing people, it is important for that manager to first provide the people with a background on the issue and specify the objective. Then the manager can proceed with presenting the plan—being mindful to allow for discussion on how the objectives can be met. This environment will promote good ideas and buy-in.

There are many tools for this type of communication, including meetings, presentations and printed documents. The company includes many people that all want the enterprise to work. After all, if the company is not successful, the paychecks will eventually stop for managers and employees alike.

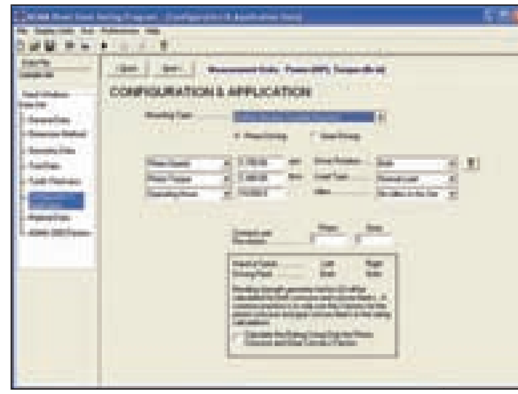
I've seen many times where the front-line worker has ideas and information that the top level managers don't have. Always remember that a successful manager cannot ignore this valuable resource.

Conclusion

The bottom line is this: Foreign competition will be with us for many years to come. To survive in this environment requires that all members of an organization embrace the concepts of entrepreneurialism, understand the lifecycles of a company and continually be aware of how the organization can offer services that the competition cannot. The low labor rate of foreign competition is a significant variable of the formula. But it's not the only variable, and it can be overcome.

Software Suite

SERVES FULL RANGE OF GEAR ANALYSIS



For seven years, AGMA's Computer Programming Committee has been working on bevel gear software that includes all of the AGMA standards. *The Bevel Gear Rating Software Suite* calculates geometry and ratings for straight, spiral, skew and Zerol bevel gears, according to ANSI/AGMA standards. These standards are followed without imposing design rules, providing a full range of gear analysis.

"There are many factors involved in calculating and rating bevel gears," says George Lian, vice chairman of the Bevel Gearing Committee and senior project engineer at Amarillo Gear Co. "With this software, you not only have calculations, you have the ability to customize the program and input the data however you see fit."

Lian says the flexibility of the software allows experienced and inexperienced users to easily navigate the screens and get accustomed to the program. Users can accurately compare their own designs and practices with these standards or simply understand their competitor's ratings.

"The software has a preference screen so the user can run a series of calculations that are common and adjust the items that aren't. It can convert both metric as well as U.S. units. These can be entered in the manner in which you prefer," says Robert Wasilewski, chairman of the Bevel Gearing Committee and design engineering manager at Arrow Gear.

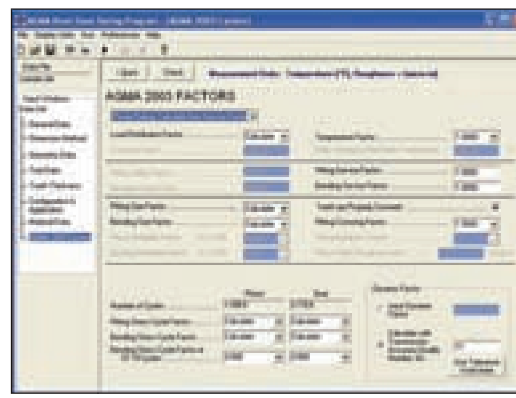
"The software can calculate the geometry factors or accept the factors that have been calculated outside the program," Lian says. "Tooth thickness can be measured, calculated by geometry factors or calculated from AGMA or ISO thickness factors."

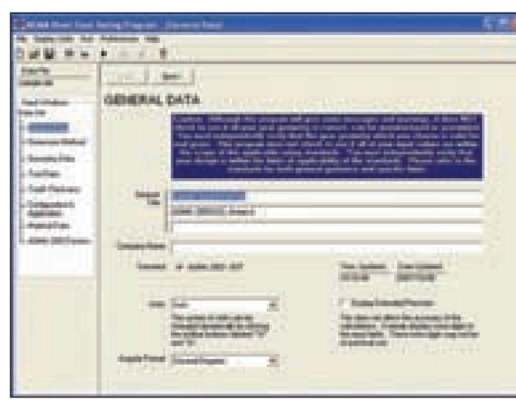
Customization might be the software's greatest selling point, as the flexible data entry concept offers users a variety of possible formats, including entering tooth thickness data as normal or transverse circular thickness, or as chordal measurements.

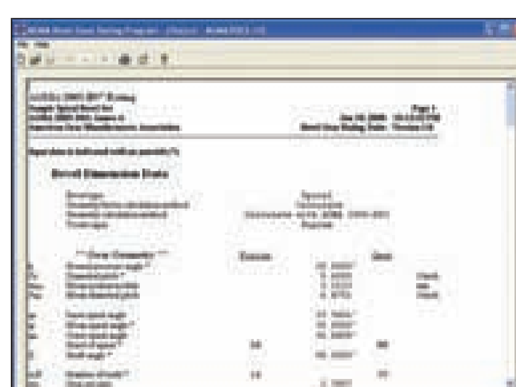
According to Lian and Wasilewski, if a user needs to start a new job while working on a project, the computer will memorize the necessary data and hold the pertinent information until the user returns to that specific job.

Other features include an intuitive user interface that allows drop-down boxes, lookup tables, and graphical guides; dynamic unit conversion that allows users to go back and forth between SI and inch units; a hardness conversion routine with eight different scales; tolerance worksheets; error and warning messages; and an online help feature that serves as the software's user manual.

The program output is displayed and may be printed directly by the program or stored in a rich text format. A printout can include symbols from the hardcopy of the AGMA standards







where applicable.

"If you're going through hundreds and hundreds of equations, you're bound to make a few mistakes," Wasilewski says. "With the bevel software suite, the user can be confident knowing it is written with the AGMA standards in mind."

The program covers material from the following standards:

- AGMA 208.03–System for Straight Bevel Gears (1978)
- AGMA 209.03–System for Spiral Bevel Gears (1964)
- AGMA 209.04–System for Spiral Bevel Gears (1982)
- AGMA 202.03–System for Zerol Bevel Gears (1965)
- AGMA390.03 a Gear Handbook Gear Classification, Materials and Measuring Methods for Bevel, Hypoid, Fine Pitch Wormgearing and Racks Only as Unassembled Gears (1988)

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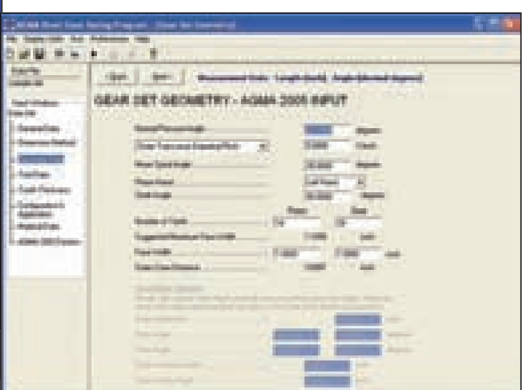

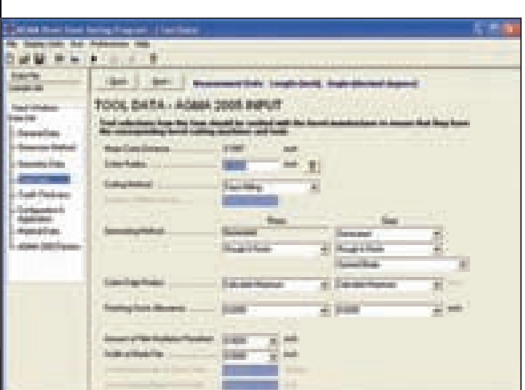

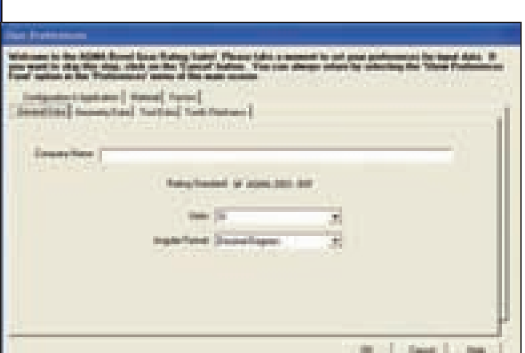

- AGMA 929–Calculation of Bevel Gear Top Land and Guidance on Cutter Edge Radius
- ANSI/AGMA 2003–Rating the Pitting Resistance and Bending Strength of Generated Straight Bevel, Zerol Bevel and Spiral Bevel Gear Teeth
- ANSI/AGMA 2005–Design Manual for Bevel Gears
- ANSI/AGMA 2009–Bevel Gear Classification, Tolerances, and Measuring Methods
- ISO 23509–Bevel and Hypoid Gear Geometry

The software suite is available for purchase with the built-in help file and electronic (read only) copy of all standards referenced by the program or with the help file only. AGMA members receive a discounted rate on their purchase. A demo of the program with a fixed number of teeth is available for download at *www.agma.org*. For more information, call (703) 684-0211.

Measuring, Inspection System Measures

GEARS UP TO 1,500 MM

The TTi line of CNC Gear Measuring and Inspection Systems, distributed in North America by KGK International, is designed to be modular, easy to use and highly accurate for gear configurations from 40 mm to 1,500 mm. "The TTi gear measuring and inspection systems offer a number of advantages to North American manufacturers. Smaller sized units in the 250 mm range are portable, and thanks to the modular reinforced construction, they can be moved from station to station without affecting accuracy," says Tom Donnowitz, KGK sales manager. "We're also exceptionally enthusiastic about the capability of measuring large-sized gears up to 1,500 mm."

Wear resistance of the shaft component is increased with a 0.1

micron mirror finish, and it is built symmetrically with respect to the center of the machine base, so measurement is stable. Each model is built in agreement with Class I of the VDI/VDE Independent Standard. The original BB-2 NC system developed by Tokyo Technical Instruments Inc. (TTI) is included along with a Windows-compatible PC system. The software is self-diagnostic and capable of assessing various standards including ISO, AGMA, DIN, GB/T, JIS and new JIS. The system displays results in numerical tables, graphs and profiles, and the data can be used for later comparisons—such as before and after heat treating.

"Although the TTi product line is new to this market, the company has been developing processes since it was founded in 1972, and the entire range of models has been proven in installations throughout the world," Donnowitz comments. "We believe that this will be an excellent addition to KGK's expanding selection of machinery and products for the industrial market."

For more information:
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(847) 465-0181
sales@kgki.com
www.kgki.com

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Dukane Lean Manufacturing Cell

PRIORITIZES CUSTOMER, OPERATOR NEEDS

The customers and operators were top priority for the Dukane Corporation in the design of its new Lean Manufacturing Cell, which uses ultrasonic welding, spin welding or heat staking applications to manually assemble plastic gears and other polymer components. The Lean Manufacturing Cell is compact in size, small enough to fit through a standard-sized door, robust and designed to be as compact as possible in an ergonomically-friendly manner. The product development of this lean work cell was driven by customer requests, and each one is custom-made per order. They are designed individually to meet "whatever the customer requirements are. There are no restrictions," says Mike Johnston, national sales and marketing manager for Dukane.

The three applications the cell accommodates enable it to assemble two halves and attach or insert some component to a gear. Ultrasonic welding, a process used to bond plastic, and heat staking are highly useful for plastic gear manufacturing because they are capable of driving brass inserts into gears, like for threading. "Ultrasonic can be used to make gears come off the plastic runner with a clean departure," Johnston says. "The gear will fall off the runner without burr or debris."

The cell's safety features are extensive. A key is typically the first line of safety defense, but this work cell has an added backup defense of electronically interlocked panels and OSHA-approved anti-tie-down activation switches. The pneumatics have an OSHA lock out device, and a filter regulator for added protection during maintenance and setup. Although each machine varies per customer order, "We've incorporated these standards in all the machines," Johnston says.

From an ergonomic feature standpoint, Dukane took operators and their comfort and safety into consideration with the Lean Manufacturing Cell. The table height is determined by operator control, and every control is within 18" of the operator's reach. The activation switch position is adjustable, as is the placement of the part bins. The enclosures can be made with sound-deadening material to minimize operator fatigue. The user interface is in plain view of the operator, and the part load area is open on three sides, making it obstruction free.

Dukane's Lean Manufacturing Cell's robust construction is produced from larger-sized, solid-aluminum extrusion, and it includes a 1"-thick MIC-6 anodized tool plate tabletop. Optional features include part presence, color detection, part clamping and bar code readers.

For more information:

Dukane Corporation
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St. Charles, IL 60174
Phone: (630) 797-4900
Fax: (630) 797-4949
usmktg@dukcorp.com
www.dukcorp.com

Extra-Large Atling Steady Rests

HOLD UP TO 38 INCH DIAMETER WORKPIECES

The Atling self-centering hydraulic steady rests are the largest LMC Workholding has built to date. Featuring a minimum clamp size of 340 mm (13.4 in.) and a maximum clamp size of 965 mm (38 in.), the steady rest holds workpieces with diameters as large as 38 in. A glass scale provides sizing feedback, and part diameters within +/- 5 microns are measured by a linear encoder.

The steady rests uphold the reliability of shaft or bar workpieces that may be subject to bending or deflecting under unstable cutting loads. Where limited space conditions are a factor, the Atling steady rests have a compact housing, and they are designed to fit most all CNC or conventional lathes, according to the company's press release.

For more information:
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Fax: (574) 722-6559
info@logan-mmk.com
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Swiss-Type Broach Holder

SERVES LARGER CNC MACHINES

Slater Tools Inc. expands the Swiss-Type, Adjustment-Free, Rotary Broaching Tool Holders product line with the 0700 series, which features a 0.500 inch tool bore. The 0700 is a modification of the previously released 0600 series designed for small Swiss-Type machines.

The tool holders are designed to be used with standard rotary broaches measuring 1.75" long with a 0.500" shank. The broaching tool holder does not require center indicating, and the design eliminates interference. Altering the tool holder shank length is unnecessary, as are secondary operations, and cycle times are swift. Most of the tool holders have in-stock availability, and they are primarily used on Swiss-type CNC machines and gang style lathes, according to the company's press release. They are designed for medical or aerospace industry applications. View the 0700 series in action at *http://www.youtube.com/watch?v=lcTJKySDDx8*.

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Production Cell

COMBINES ROBOTIC

MACHINING WITH SIX-AXIS ROBOT LOADER

The JobShop Cell from Methods Machine Tools integrates the RoboDrill

continued

Vertical Machining Center and FANUC six-axis robot to load and unload the cell automatically. The production cell is designed for job shops with small-to-medium lot sizes, and is capable of controlling any part that fits into the 6-inch vise or chuck, according to the company's press release.

With inbound and outbound conveyors and guarding, the work-handling interface can accommodate various hydraulic or pneumatic workholding choices. A fourth axis and workholding fixture are optional features designed to provide more uses for the cell. The automatic loading and

unloading capability resolves issues that correspond to those points in the production process, and it minimizes the skilled labor required while being available for use 24 hours a day, seven days a week.

The FANUC RoboDrill VMC component of the cell has a 14-tool changer—or optional 21-tool changer. Its features include torque to 56 ft-lbs, tapping to 5,000 rpm, x, y and z axes accelerations to 1.5 G, traverses to 2,125 ipm, feed rates to 1,181 ipm, thread milling and 54 work offsets. There are other options available within these specifications. The six-axis FANUC LR Mate 200iC robot is the other main component of the production cell, and it comes equipped with grippers, a teaching pendant and several customizable programs.

For more information:
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Expanded Vacuum Carburizing Applications

PRODUCE SUN PINION

Solar Atmospheres has begun vacuum carburizing a sun pinion made from AISI 4320 material. By connecting to a generator as a planetary speed increaser, the 225 lb. sun pinion is being used by a hydroelectric plant in South Carolina.

The sun pinion requires selective carburizing, so stop-off paint was used on the shaft to prevent carburizing and enable more machining. The gear's requirements include an effective case depth of .090"–.110" and a surface hardness of 58-62 HRC. The root-to-pitch ratio improves from 60 percent to 90 percent by employing the vacuum carburizing process, according to the company's press release. The gear's finished properties demonstrated improved distortion and overall quality compared to traditional carburizing methods.

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Lubricant

DESIGNED FOR HEAVILY LOADED INDUSTRIAL GEARS, BEARINGS

A high-performance, synthetic industrial EP gear lubricant has been developed by Dylon Lubricant Technologies Co. The Dylon LB-447 Synepoil family of ISO-grade lubricants is designed for industrial gears and bearings using anti-wear and R and O additives along with synthetic base stock.

With a low coefficient of friction at 0.12, the Synepoil lubricants reduce frictional losses that are responsible for heavy power consumption. Cleanliness is improved at several levels with increased thermal stability, low temperature fluidity and high sludge formation resistance. As a result, service intervals should be longer and seasonal oil changes will not always be necessary. Packaged in pails, kegs, drums and intermediate bulk containers, Dylon is offering a promotional price to introduce the lubricant to U.S. companies; interested customers should call for more details.

For more information:
Dylon Lubricant Technologies Co.
7700 Clinton Road
Cleveland, OH 44144
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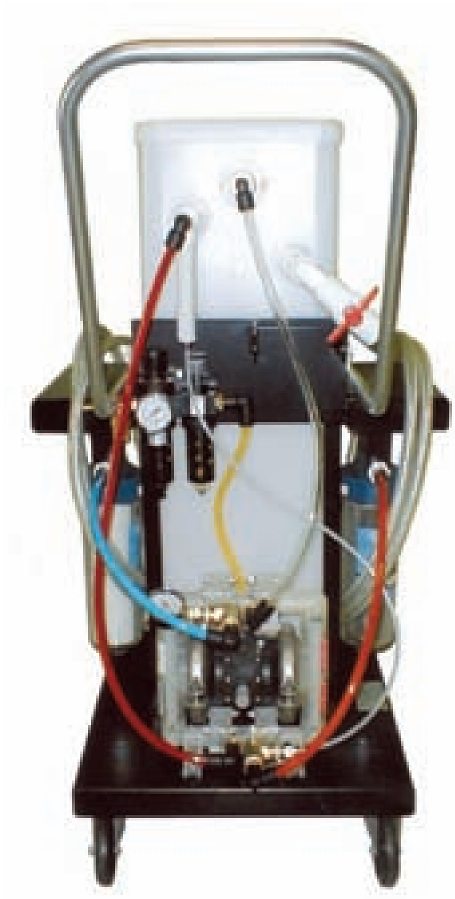
Fax: (216) 651-1777
www.dylon.com

Portable Coalescer

REMOVES WATER-BASED FLUIDS, PARTICLES FROM STRAIGHT OILS

The Snapdragon Sierra Portable Coalescer from Zebra Skimmers increases the life and reuse of expensive oil by eliminating water-based fluids and fine particles from straight oils.

Meg Grant, sales and service manager of Zebra Skimmers Corporation, says "Our Snapdragon Portable Coalescer series has been successfully serving the metalworking industry for many years, and this new addition to the fleet will help to lower costs for straight oil



use among metalworkers and other industries."

For more information:

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Ultrasonic System

CLEANS HOBS IN MINUTES

The Ultrasonic Cleaning system from Colonial Saw Company is capable of cleaning saw blades, router bits, shapers, knives, hoggers, CNC Spirals, hobs and more in minimal time. An average stack of 20 cabinet shop blades can be cleaned in two minutes, while an average stack of 20 lumbermill blades are cleaned in five minutes, according to the company's press release. Soaking, polishing and wire brushing are not necessary. Tanks come in four sizes, suitable for blades with diameters up to 40".

The Ultrasonic Cleaning System runs on twin generators that sweep 40 kHz frequency for distributing the cleaning force. Ultrasonic transducers

continued

are bonded right onto the interior walls of the system. The system tank is cleaned and refilled by inlet/outlet plumbing. The generator unit is detached and can be mounted as far as 15 feet from mist and dirt. Some optional accessories include a 49-gallon heated rinsing tank, digital readout timer, power control for adjusting wattage, tank cover, tool basket and a saw-blade rack.

For more information:

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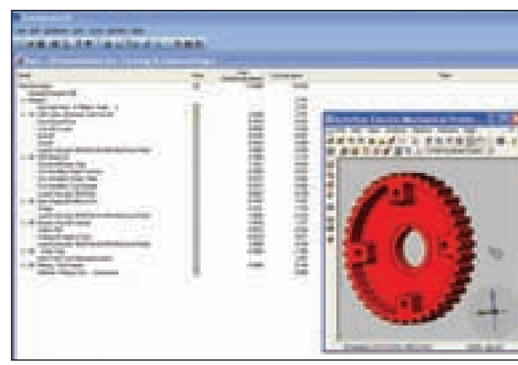
OEM Software

DEVELOPS COST MODELS



The Costimator OEM V9.0 is the latest cost-estimating solution from MTI Systems. "Version 9.0 delivers a number of new enhancements that will further enable manufacturing companies to assess, identify and reduce product costs throughout the product development lifecycle," according to David LaJoie, vice president of sales for MTI Systems.

The feature-based cost models have been expanded in version 9.0, and the feature library incorporates cost models for prismatic and cylindrical machined parts, sheet metal fabrication and secondary non-machining operations. "The newest Costimator OEM library further enhances the design engineer's ability to understand the impact that varying product design alternatives have on cost," LaJoie says. "Having this type of accurate, dependable cost knowledge has proven to save MTI's customers millions of dollars."

A new tool, called the cost modeler, allows users to create their own unique process or cost models. "Many of our existing and new customers have cost models they previously developed in Excel that they want to bring into Costimator OEM. Cost modeler gives them a flexible, easy-to-use tool to create cost models on their own, saving them time and money," LaJoie says.

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Machine Tool Evolution

INNOVATION/INGENUITY REMAIN KEY TO LONG TERM SUCCESS

Matthew Jaster, Associate Editor

The CNC machine helped change the face of manufacturing technology in the 1970s. By reducing the number of machine steps, manufacturers had the ability to improve the quality and consistency of their products. Errors were reduced and operators had extra time to work on additional projects. The invention of the multi-tasking machine was another key innovation in machine tool efficiency. Engineers now had a cost-saving solution in terms of labor, uptime and equipment by combining multiple fixtures and setups into a single unit.

Haas lathes, with automatic parts loaders, can run unattended to reduce idle time.

A Motoman 6-axis robot works between three Haas machines.

In today's machine tool market, productivity, process development and software innovation are essential as manufacturers look to reduce costs, shorten lead times and improve quality systems. Those involved in R&D continue to come up with fundamental machine tool advancements in their pursuit of lean principles; their goal simply to maximize efficiency on the manufacturing floor.

Just ask your customers. The general consensus is that most technological advancements in machine tools aren't developed in the basement of some raving, mad scientist—they come from frequent discussions with end-users.

"Not involving the customer upfront with product development and R&D is not an option in today's environment. The customer is directly or indirectly the source and/or inspiration for most innovation," says Alan Finegan, director of marketing at Gleason Corporation.

In order to meet these various demands, companies typically look toward customers for instant feedback. At Gleason, Voice of Customer (VOC) feedback is critical to new product development.

"All customers want the best quality for the lowest cost, but depending on the characteristics of their production and their cost structures, how they achieve these things can vary widely," Finegan says. "These customer requests may relate to a broad range of needs, from floor space to energy consumption, from ease of maintenance to software functionality and so on. Some are technological demands and some are not. All drive innovation in one way or another."

By establishing long-term customer relationships, machine tool manufacturers are able to see how their machine tools operate under various working conditions and what can be done to improve them over time.

"This means interpreting a customer's experience over years of operation," says Bill Miller, vice president of sales at Kapp Technologies. "Digesting, articulating and conveying this information throughout our organization leads to creating revolutionary (as well as evolutionary) solutions that make a difference with very low risk."

Bill Tandrow, director of mechanical engineering at Haas Automation, states that the continual improvement philosophy at Haas includes regular customer interaction and feedback directly to the product designers.

Product questions surrounding the complex machining of small diameter bar parts, for example, brought the creation of the GT-20, a flexible alternative to "Swiss style" bar processing in a familiar turret lathe platform. "Whether through sales, service or directly during factory tours, customer comments have always sparked product innovation," Tandrow says.

Produce more, spend less. Tandrow believes there's pressure on machine shops of all sizes to produce more varieties of parts with the same or fewer staff members and with highly stratified skill sets.

"The investment in the production process is often a cost amortized over repeat customer orders. That means that creation, capture and reuse of CNC machine setup information is equal in importance with fixture and tool maintenance for spindle utilization," Tandrow says. "The information connectivity of CNC machine tools is a reality today. A Haas CNC machine can export program, tool setup and work offset data for cataloging electronically. This data can be brought back to the machine remotely via network connections, and the time to reestablish a job setup can be minimized."

Haas has been working on improvements to automatic machine tool tending applications. Combining CNC machines with automatic part handling, according to Tandrow, can become complex due to the variation in handling needs and the variety of machine designs.

"Machine tool tending has been made simpler with the introduction of the Robot Ready interface, a standardized interconnect between the CNC machine tool and tending robots using network style communications," Tandrow says. "Each installation need not be a unique science experiment; rather, the integrator's package can be replicated and software can be reused as the use of material tending increases."

Kapp Technologies is working on many enhancements to shorten machine setup times via software features. R&D is concentrated on reduced machine setups, reduced non-grinding times (such as loading) and achieving high utilization of the equipment as well as fundamental advancements in grinding technology.

"Periodic visits by our application engineering specialists enable us to optimize machine performance and productivity as well as improve proficiency," Miller says.

Kapp's KX500 FLEX Gear Center was created to extend the range, speed and flexibility for gear grinding. It now offers profile and generating grinding and integrated tooth quality measurement, and is prepared for the integration of an automatic loading system.

"Value for the long run is what matters, regardless of the economic headlines," Miller says. "By manufacturing machines we know will be in operation in 10 to 20 years and by offering our complete process package, which dictates that we will serve the customer for the life of their Kapp or Niles machine, we are ensuring the customer is getting the very best cost-savings possible."

Higher feeds and speeds, elimination of "dead time" in the cycle and faster acceleration and deceleration rates of machine axes are just a few of the things Gleason strives for in regards to machine tool productivity. The company is developing and patenting software innovations such as adaptive feed rates and automatic workpiece centering on large cylindrical gear profile grinders that save customers time and money.

On the tooling side, Gleason continues to offer cutting tool designs, materials and coatings such as their Opti-Cut line, used in the production of large cylindrical gears, that all take advantage of machine innovations.

"Customers want value, which for them is part quality, productivity, flexibility, tool life, cost per piece, reliability, machine uptime and more," Finegan says. "Opti-Cut is an innovation in and of itself that brings the cutting tool on par with the advanced capabilities of our gear hobbers and shapers."

Currently, Gleason is making record levels of investments in R&D for bevel

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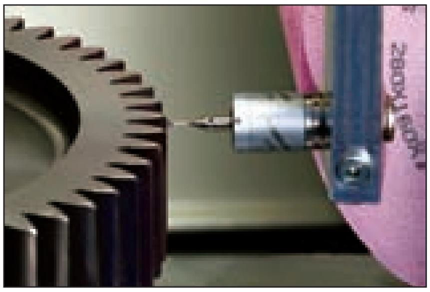
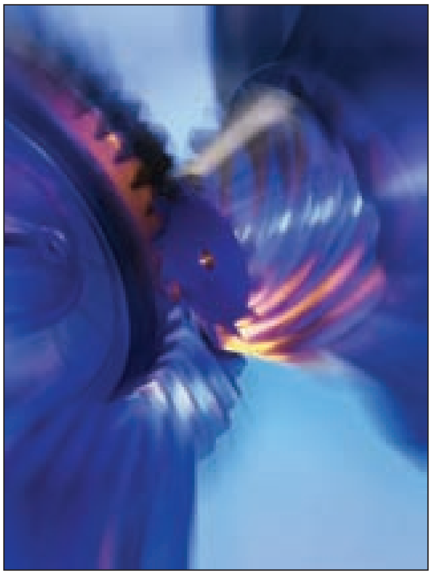


On-board inspection is one of many updated features of the Kapp KX500 Flex Gear Center.



The Gleason 360AT will be used for the MTConnect demonstration at IMTS in Chicago.

gear, cylindrical gear and gear metrology products and processes, Finegan says.

Focus on functions. Liebherr Gear Technology, Inc. has spent the last year upgrading its machine tools with functions that better serve its customers. Innovations on its LCS 150 machine include the universal application of either CBN or corundum grinding tools. Another advantage is the use of tools in optimal combination so that the benefits of generating grinding and profile grinding can be used individually as well as together.

"The main focus of this machine has been to increase productivity by reducing idle times of loading and meshing, which are now 50 percent lower than previous machines," says Scott Yoder, regional sales manager at Liebherr. "The machine bed of the LCS 150 was specifically designed for grinding, with a reduction of floor space by approximately 35 percent."

Floor space reduction, in fact, is a frequent customer request, one that Liebherr has addressed on many of its new machine tool designs.

In addition to the LCS 150, Liebherr has created the high-speed LC130 hobbing machine that features a fully-enclosed work area, an optimized chip-flow for dry cutting and a rigid deburring system. The rotational speed of the tool—up to 7,000 rpm—allows cutting speeds of 1,700 m/min maximum. Loading times have been reduced so that chip-to-chip time is only 4.4 seconds.

"Practical experience with our current machine generation has significantly influenced the development of these machines," Yoder says.

With experience in automation and material-flow, Liebherr has also created the PHS 1500, a pallet-handling system for machining centers that can be loaded in either a manual or fully automatic mode. This combination allows flexibility in production which has been unavailable until now, according to Yoder.

"In consideration for the various requirements for the storage places, the system can be adapted to a variety of applications and manufacturing situations. A modular design allows these units to be combined as the requirements increase," Yoder says. "Liebherr chose to partner with Soflex (a software supplier), which ensures a user-friendly system with excellent functionality for the entire cell."

The Liebherr PHS 1500 pallet handling system for machining centers will be at IMTS.

The PHS 1500 as well as the LCS 150 will make the trip to IMTS in Chicago this September.

The technology show of shows. When IMTS opens its doors, it will cater to 91,000 buyers from 119 countries. Exhibitors will have an opportunity to show off their latest tricks while keeping a close eye on the competition.

"Major shows like IMTS are an excellent way to showcase our newest products and technologies," Finegan says. "Shows like this are only one of a variety of promotional tools. It's highly dependent upon the specific region and culture as well as the product or service being promoted."

Miller cites shows like IMTS and EMO Hannover as beneficial in terms of the new opportunities available when Kapp participates.

"We've had a good experience searching for applications beyond parallel gears. IMTS generates many opportunities outside the traditional gear market. This has led directly to business in appliances, actuators, steering, non-gear aerospace and energy. The customer is the true innovator in these cases." Miller says.

Although the company keeps in contact with customers via a newsletter or by mailings, Miller says IMTS remains a unique venue.

"IMTS invigorates our business and leads to innovation. We always meet people who are searching enthusiastically for manufacturing methods for their new products. This drives us to find unique solutions and develop specialty products."

Haas Automation will be demonstrating improvements in the use and ownership of CNC machine tools in addition to new mill and lathe product offerings. Tanderow says the new Haas CNC control pendant has the USB interface and option controls located directly on the pendant as well as increased user storage compartments on many models.

IMTS 2008 will address machine tool innovations at the Emerging Technology Center with the first public demonstration of MTConnect, an open communications standard for manufacturing equipment interoperability.

It is being developed by a partnership between the AMT and several manufacturing technology companies. The University of California, Berkeley and Georgia Tech University are also directly involved in the project.

"We believe that once manufacturers take a look at the open standard and turn their engineers loose to develop products using the standard, we'll see productivity gains similar to those achieved with CNC in the 1970s," says John B. Byrd III, president of the AMT.

For 2008, the IMTS has rededicated itself to technology as its core purpose. The theme of the 2008 show is "Connecting Global Technology."

"We are producing six theme days at the IMTS Innovation Center, each with a specific industry partner that will provide opportunities to showcase new advancements and technologies as they apply to each of those industries," Byrd says. "IMTS is also returning to its roots as a science fair—we're producing the first Manufacturing Business and Technology Forum, which is a series of educational opportunities that cross the spectrum of new and developing technology, as well as manufacturing and management techniques.

"All of these events and new machine tool advancements can be previewed—and will be archived afterwards—at *www.imts.com*," Byrd says.

The wait and see approach. The list of customer demands can be daunting. They want value, reliability, quality and high-operating capacity. They want machine tools that cost less, produce more and can keep up with the day-to-day demands on the shop floor. In short, they want an investment they can rely on for many years to come.

"Developing products and technologies with the hope of opening opportunities is probably not a high-percentage play. I think the process works the other way around in our industry," Finegan says.

Gleason sees the range of applications for gearing expand as it makes machine tools more affordable to its customers. Finegan notes that the end markets Gleason serves today are more diverse than at any time in the history of the company.

"Understanding customers and markets through VOC activities, and using that information to guide technology and product development virtually insures that we can capitalize on opportunities as they are presented to us," Finegan says. "Like all investments, development activities must be actively managed and are subject to the same lean concepts and principles that drive our manufacturing.

"It's an investment in future business."

John Shook, senior advisor at the nonprofit Lean Enterprise Institute, discusses how to implement the elements of lean as a business system.

Journey to the Land of Manufacturing Milk and Honey

WHAT ARE YOU WAITING FOR?

Lindsey Snyder, Assistant Editor

You won't transform your gear business into a lean manufacturing enterprise with this article. Too bad it's not that simple. But you will read some compelling information that might leave you eager to embark on the life-long journey. The tools available to those interested in implementing lean manufacturing policies are abundant, including books, articles, web resources, consulting firms, non-profit research organizations and workshops, to name a few. But the tools alone won't motivate you to explore lean. The tremendous potential benefits to implementing lean practices are exemplified by the experiences of companies spanning many industries. Individuals who have witnessed and participated in the improvements champion lean manufacturing. Not itching to catch the lean bug yet? Read on.

The Logic for Lean

In general, companies that properly implement lean practices can expect to see 30–40 percent productivity increases, according to Rick Harris, president of Harris Lean Systems, who has seen productivity improve as much as 70–80 percent in some cases.

Harris advises that lean success does not occur overnight, and it is not a quick fix to a company's problems. Ralph Keller, president of the Association for Manufacturing Excellence (AME), agrees that initial growth takes time, and the move to become lean is not a quarter-focused project. After the first year, he says, there should be noticeable improvement in lead times and other areas, setting the stage for large-scale growth and progress. "Lean transformations don't produce small incremental improvements," Keller says. "They produce quantum leaps to new levels."

When done right, a lean enterprise could benefit substantially in a number of areas, including lead times; quality levels; cash flow; margin improvement; swifter introductions of new products; customer satisfaction; customer service; total enterprise value and reductions in cost and idle time. Lean manufacturing can also provide competitive advantage. Jorge L. Larco, owner of J.L. Larco & Associates, Inc., a lean consultancy

continued

AME's National Manufacturing Workshop included a tour of Batesville Casket's Manchester, TN plant and highlighted the company's nine-year use of manufacturing guiding principles to achieve consistently outstanding operational results.

business, cites improvements in gained outcome per unit of time and per person as being much more than 100 percent. He says that in reference to a single market, "Lean is a competitive advantage, but if you do not have it, you are cooked."

At Altra Industrial Motion, the voice of the customer is the starting point of the Altra Business System (ABS). "By understanding what the customer truly values in the goods and services we provide, we are able to focus our improvement efforts on the critical processes that provide value to the customer," says Al Mason, a director of ABS. "As we make significant improvements in the quality, delivery and cost of these processes, we satisfy our customers more thoroughly and consistently. Customer satisfaction then translates to long-term, profitable growth for the corporation."

Since beginning to implement lean practices at Boston Gear, an Altra Industrial Motion company, the on-time delivery to customer request date has improved from the low 80s to a mid- to high-90s percent range, according to Mason. Quality, measured by internal parts per million, improved by a factor of 10. Lead times have been reduced by 40 percent, and working capital turns improved by 50 percent. Sales per employee doubled at Boston Gear, and enough floor space was freed up to integrate a new product line.

The momentum started to build for a transition to lean practices around 1998 and 1999 at Boston Gear, Mason says. "The transformation has been a gradual and continuing one," he notes. "When we started, some areas of the shop were more traditional (batch and queue) than others, and as such, have required more of a transformation effort. But generally speaking, the business has steadily worked over the past 10 years to drive lean principles."

Mason previously served as corporate director of continuous improvement for Goodrich Corporation, an aerospace industry supplier, where he was involved in the initial efforts to deploy lean there. He started working for Altra in 2005, meaning he was not there during the initial lean implementation; "But in the past two and a half years, I have seen a relentless pursuit of perfection," he says. "Value streams that have been significantly improved are continually being revisited. Also, the business is more aggressively applying lean principles to logistics and engineering processes."

At Delta Research Corporation, lean manufacturing has helped the company with competitiveness in its production of prototype gears, gear boxes, casings and splines. The company also does milling, turning and grinding, producing all the solid parts of a transmission system. Delta Research's lean productivity began in 1995 when owner Bob Sakuta and his brother Dennis were discussing the future of the company. They made a list of the machines they would need, the layout of the shop floor and the amount of people that they would need in order to achieve their productivity goals. Today, nearly every one of those machines is on the shop floor.

Once Delta Research acquired sister company Tifco Gage and Gear, it was able to offset a problem with partner vendors that had slow and unreliable lead times. Tifco produces master gears, gages, aircraft pumps and automotive gears, so the acquisition enabled Delta to increase its gear production with shorter lead times. A few years ago Delta Research introduced a hands-free, automated gear cell that turns, inspects, hobs, grinds, cleans and deburs with an operator simply loading the material and unloading the finished product. "We're in a state of expansion right now, especially at Tifco," says Tony Werschky, a sales and marketing representative for both companies, "while many others in the business are down to half shifts or have shut down completely."

Delta Research experienced a 39 percent improvement in overall productivity since implementing lean. "[Lean] has helped us to stay competitive in this environment, which is very competitive, as you know, these days."

The Toyota Production System (TPS) is one of various models for implementing lean, but each company's experiences and improvements are unique.

Jim Sonderman, a consultant and coach for the Lean Learning Center, has a client that sells component parts to automotive companies that experienced a 17 percent improvement in productivity, rising from 36 pieces per labor hour to 42. Uptime went from

48 percent to 75 percent for this client, and floor space utilization improved by 14 percent. He estimates that the client reduced inventory by 71 percent. Another of his clients just began its lean journey in January of this year, and it has already improved labor-cost-per-unit by 27 percent within the initial implementation area.

Delphi Corporation's Rochester Operations, one of the company's core automotive manufacturing facilities in the U.S., experienced significant improvements under Harris' guidance. Annualized labor savings reached $2.7 million for the company while productivity improved by 190 percent, according to the HLS website. Harris says that HLS has helped about 50 companies and 180 plants worldwide.

Danaher and Toyota are the two most prominent "poster children" of lean manufacturing, according to Keller. Toyota is the most profitable automotive company in the world, and so far in 2008, the corporation has passed General Motors in first-quarter global sales, he says. Danaher Corporation began implementing lean in 1984 and "The initiative succeeded beyond anyone's expectations," the company's website states. "Since this modest beginning, DBS (Danaher Business System) has evolved from a collection of manufacturing-improvement tools into a philosophy, set of values and series of management processes that collectively define who we are and how we do what we do.

"Fueled by Danaher's core values, the DBS engine drives the company through a never-ending cycle of change and improvement," the website states.

Some other "poster children" Keller refers to include Wiremold, which sold for 23 times its earnings before income and tax (EBIT) value; this year, Ventanna Medical sold for 12 times what it was worth in sales and 68 times its income after only six years of implementing lean; and Actuant Corporation's stock price grew by a factor of 5 between 2000 and 2004.

AME's Rapid Continuous Improvement Workshop hosted by Hearth and Home Technologies included a tour of the plant. The event celebrated the company's 1.5 million hours without a lost-time incident and on-time delivery of more than 98 percent in June 2007.

Lean Sins

There are various reasons why a manufacturing enterprise may not succeed in lean implementation, but none of the reasons relate to lean principles. Horror stories do exist, which paint a negative picture of it; however, there are common elements of failure that come up in analysis. Serious roadblocks can be averted simply by being aware of some frequent mistakes to watch out for.

One such horror story Keller knows of involved a company that abandoned its lean practices after only eight months. In this situation the command and control manager was left in charge of the implementation, and the manager knew nothing about the crucial people development factor, which Keller believes is the most important piece of implementing lean. All levels of employees must understand and be committed to lean in order to succeed, and it starts from the top down. The company suffered as a result of this failure in several ways, including the loss of some of its senior most knowledgeable workers, who left out of frustration. Keller stresses that lean principles cannot be doled out as one person's job. Every single member of the organization must actively participate and be informed on lean methods. He says, "Most horror stories are related to trying to delegate the lean transformation."

There is no one correct way to implement lean, and consultants disagree among each other as to which are the most important points, but they agree that if change does not come from the very top of the enterprise, without a strong leader driving the lean movement, actively engaged and participating with commitment, a lean production system is doomed. In Sonderman's opinion, "Leadership is often the main reason why a company fails in its lean transformation."

Lean is often employed as a "program of the month," meaning that it is featured as a short-term goal or exercise for employees to practice. This strategy typically features lean methods for a short period without maintaining them. Since lean focuses on the idea of continuous improvement, this is an ineffective means of achieving that goal. A similar source of failure is based on the conception that the company will eventually meet its goal of being lean. This couldn't be further from the truth. Lean is a never ending business system manufacturers must commit to. It is not just a toolbox; practitioners need to know when and how to use the tools of lean in different circumstances.

continued

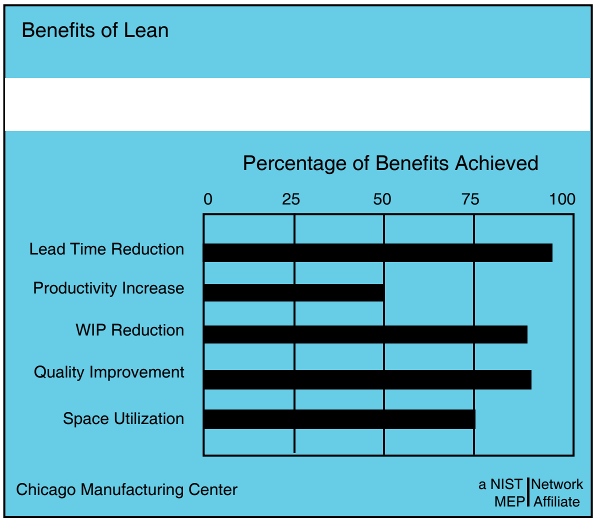
By nature, lean progress will eventually create idle capacity where employees become freed up due to improved production efficiency. This creates a hurdle along the path to lean. The natural reaction is that this is a good thing, and the company can afford to reduce its workforce in response and save more money. This is a major misconception that is likely to produce disastrous results. Lean principles are about the growth of an enterprise, not cost reduction. A main goal of lean is to reduce waste, but this is very different from reducing costs. Many companies believe laying off employees is appropriate once lean improvements take place, but Sonderman says that many consultants won't even consider working with a company infected with the layoff virus. He says it "is a good way to kill lean. Stop it dead in its tracks."

Downsizing the workforce in response to lean success is bad for employee morale as well. People see others being fired as a result of lean, and they are disillusioned and lose the commitment that is imperative for success. Reducing direct labor is a huge turn-off to the people implementing lean, but there are various ways to deal with extra employees effectively. One positive way to handle idle labor is to educate freed-up workers more about lean with continuous improvement programs. A kaizen workshop where teams can identify and practice significant improvements in process or create continuous flow cells is another way to occupy an idle workforce. They can help other employees expand their working knowledge of lean practices and discover how to apply them to gear manufacturing. Where continuous growth is the ultimate goal, new projects are bound to sprout up sooner than later.

The reward system is another factor that drives wrong behavior. Many companies, especially larger ones, institute programs where employees are rewarded for reaching goals or other achievements. People are often rewarded for performance by possibly receiving a promotion. The error in this thinking is that just because someone meets the right goals or executes their job well, does not mean that they have the necessary understanding of lean to practice it at a higher level. One possible result of this rationale is an employee being placed in a position of lean management without the proper know-how.

Success Factors

While there are common elements of failure in implementing lean manufacturing, there are also several critical factors to pinpoint success. The overall picture is far from gloomy, and the potential improvements far outweigh the challenges that exist.



Leadership is at the top of everyone's success factors list. It is imperative for the president and/or CEO to drive the lean movement and be visibly supportive. Keller believes that in order for employees to actively practice lean, they need to see the CEO embracing the new ideology and participating in shop floor training. There can be difficulties engaging mid-level managers, but with executive management truly leading the transition, other levels of participation should fall into place. Keller notes that the management pyramid is turned upside down in a lean production system, so management's role is to ensure the proper training, material and machinery are available to the people that are creating value for customers. Mason has never seen a significant success story that wasn't led directly from the top. He says, "One of the real upsides in the Boston Gear story has been the high level of management support and commitment, both at the local and corporate level."

It is also important to work with an individual who has experience implementing lean practices. Lean specialists work extensively to bring hands-on techniques to the enterprise. An outside view with experience is instrumental to guiding continuous improvement, and many organizations are committed to providing lean guidance.

Policy deployment is another element for success. This involves analyzing the methods for improvement to determine whether they will produce the long-term goals for success. This is especially challenging for a large company because there are longer communication lines, and there are essentially many people pulling in different directions. It also involves having a clear conception of goals and objectives while aligning them with strategies and developing a specific blueprint for achieving them. Value-stream mapping is one popular tool where each stride in material and information flows is carefully charted from the point when a product is ordered to its delivery. Value-stream mapping also involves thinking at the system level with a broader view of production. Unfortunately this approach doesn't come naturally to most humans. The overall culture of the work environment needs to change and reflect this more expansive outlook.

A dramatic change in the work culture is essential to the lean transformation. Individuals are essential, and they need to embrace the drastic culture change. It starts with hiring the most qualified and talented people; proper training is the next step. Larco says that at the lower level, employees must participate, and they will do so once they are convinced that the transformation is real. At the intermediate employee level, a fundamental transformation must take place to move from directing flow to coaching, training and truly enticing and involving people, he says. The people need to transform their thinking from an individualistic perspective to that of a team member. Larco says, "That transformation from I to we, where your personal ego is rewarded by saying we rather than I, that kind of reward is incredible, but it's a dramatic transformation."

Lean ideas seem to be spreading rapidly, although the level of effective implementation is arguable. Harris says he has started to see a push towards more interest in lean, but a problem remains in that companies still want too much too fast. He sees the same level of commitment to implementation that he did five years ago, although lean ideas have managed to grab more attention. Larco has witnessed lean practices worldwide, and he sees many people embracing lean in Europe, Asia and Latin America. The biggest challenge in store may be the time and effort required converting a business to lean—but the success is incredible.

More companies all over the world are reaping the huge benefits that lean has to offer, and lean manufacturing might seem to be merely a popular trend at the moment. But with all the buzz and improvements it has generated, lean is not likely to sink into the shadows any time soon. Resources are growing and models like TPS continue to improve. With the explosive benefits successful lean implementers are experiencing, lean manufacturing is thriving as an ideology and is being promoted extensively. With more than 15 years of lean practice under his belt, Larco views it as much more than a trend.

"I think this is an evolution of the industrialized world that is not going back."

Get Your Gears on the Lean Bandwagon

Lean manufacturing is known by many names: lean production, lean enterprise, Just-in-Time manufacturing, continuous improvement and the Toyota Production System (TPS). In simple terms, lean refers to producing goods by removing waste and using flow, as opposed to the traditional batch and queue method. This is achieved by using various tools, but the tools alone won't produce the desired results. Having started to revolutionize its production system as early as the post-World War II years, Toyota is the model most lean principles are based on. The company aimed to reduce inventory and produce goods upon customer orders, while in turn embarking on a voyage towards efficiency and quality. The biggest misconception that exists is that lean is a goal one can look towards achieving one day. Despite Toyota's position as a prototype for lean manufacturing, over a half century later the company is the first to insist that they still have a long way to go. There is no say-all end-all point in a lean enterprise. "Lean is a journey, not a destination," says Ralph Keller, president of the Association for Manufacturing Excellence (AME).

A plethora of lean manufacturing success stories exist, but some sources insist that lean principles are not well accepted. Jim Sonderman, a consultant and coach at the Lean Learning Center, believes lean is not poorly accepted; it is applied poorly. The Lean Learning Center in Novi, Michigan focuses on educational training and coaching through seminars and other services, including a five-day intensive course about the principles and rules of lean manufacturing. The attendees are polled about their experiences in lean. Sonderman says approximately 50-80 percent of the participants claim they are engaged in some lean practices. Out of this group, 75 percent say they have seen some degree of benefits; however, the numbers plummet from 75 percent to 2 percent with people who claim to have been "wildly successful." Despite the low success rate, he says "It is very difficult to find companies that haven't engaged in lean."

Here's a list of some resources to aid lean transformation that were mentioned in this article:

The Association for Manufacturing Excellence (AME)
3115 No. Wilke Rd., Suite G
Arlington Heights, IL 60004
Phone: (224) 232-5980
Fax: (224) 232-5981
info@ame.org
www.ame.org

Harris Lean Systems
5072 Spanish Oaks Court
Murrells Inlet, SC 29576
Phone: (843) 651-0794
Fax: (843) 651-0852
www.harrisleansystems.com

J. L. Larco & Associates, Inc.
11 Alder Way
Bedford, MA 01730
Phone: (781) 275-3925
Fax: (781) 275-3959

Lean Enterprise Institute, Inc.
One Cambridge Center
Cambridge, MA 02142
Phone: (617) 871-2900
Fax: (617) 871-2999
info@lean.org
www.lean.org

Lean Learning Center
40028 Grand River - Suite 300
Novi, Michigan 48375
Phone: (248) 478-1480
Fax: (248) 478-1589
info@leanlearningcenter.com
www.leanlearningcenter.com


HARD BROACHING
MACHINE
MQL POWER CELL
NACHi
ROBOTS
ROUND TOOLS
BEARINGS

International Manufacturing Technology Show
IMTS08
CONNECTING GLOBAL TECHNOLOGY
September 8 - 13, 2008 · Chicago, IL · McCormick Place
Booth Number : B7021 and F2974
HYDRAULIC EQUIPMENT
PRECISION CUTTING TOOLS
Wonders from NACHI, Art in NACHI
Contributing to progress in the world of Manufacturing.
SPECIAL STEELS
HOBBING MACHINE
GEAR SHAVING MACHINE
GEAR MEASURING INSTRUMENT
Gear Production Alliance
GPA
Kashifuji
KANZAKI
OSAKA SEIMITSU KIKAI
NACHi

A Methodology for Obtaining Optimum Gear Tooth Microtopographies for Noise and Stress Minimization over a Broad Operating Torque Range

Jonny Harianto and Dr. Donald R. Houser

(This article first appeared in the proceedings of the ASME 2007 International Design Engineering Technical Conference & Computers and Information in Engineering Conference, IDETC/CIE 2007, September 4–7, 2007, Las Vegas. It is reprinted here with permission.)

Management Summary

This paper presents a method for evaluating the effect of microgeometric or microtopographic variation on various gear performance parameters, i.e., noise excitations, gear contact and root stresses, film thickness and surface temperature under loaded conditions. Micro geometries that are considered are profile crown, profile slope, lead crown, lead slope and bias modifications variations. Various combinations of these micro geometries are considered in analytical simulations in which respective gear design metrics are evaluated based on the calculated load distributions. This paper will provide a walk-through analysis for a helical gear design in order to describe the procedure.

Introduction

Classical gear designers use bending and contact stress formulas along with a series of correction factors for their design metrics (Ref. 1). The designer establishes the number of teeth, center distance, module, etc., so that the design will achieve durability performance goals. However, prior to completing the design, the designer must make some decisions regarding the profile and lead modifications that must be specified.

These modifications, which are usually specified as tolerance bands on profile and lead charts, are used for several purposes, including:

- Compensation for misalignment, in that the peak stresses do not occur on the tooth edges
- Reduction of noise excitations
- Minimization of scoring potential by minimizing loads at peak sliding regions of the contact zone

The type of modifications needed to minimize one of the above factors is often in conflict with the modification needed to minimize the others. Also, one often finds that noise minimization is desired across a broad range of loads that are much lower than the durability design load. Thus, designers require a tool that allows them to create topographical modifications that provide a reasonable compromise between many design metrics (noise, stress, tribology, etc.) over a broad range of torques.

This paper presents a graphical procedure for selecting lead and profile modifications that provide a good compromise of results for each of the above mentioned design factors. The procedure allows one to observe the impact of these modifications over a broad torque range on a large number of gear design metrics.

Background

Perfect involute profiles of both spur and helical gears only exhibit conjugate motion at no load conditions. Once load is applied to a gear pair, deflections occur and the motion transfer is no longer conjugate. In order to get the motion back to some semblance of conjugacy, the tooth profile is modified, usually by the removal of material from portions of the tooth surface. Profile modifications in the form of tip or root relief compensate for tooth bending deflections, and lead modifications in the form of either lead crown or end relief compensate for manufactured lead errors, shaft misalignments and shaft deflections.

These concepts have been applied to gears for many years; some of the more classic treatises that refer to tooth modifications are presented in References 2–8. Since the contact regimes of spur and helical gears have some differences, the approaches to their modification also have differences. In this paper, the concentration is on helical gears, but a brief discussion of the approaches that have been applied to both spur and helical gearing will be discussed in this section.

Spur gearing. Extensive research has been conducted on the profile modifications that are appropriate for spur gears. However, one will find that the following generalizations exist:

1. Apply tip relief on both the gear and pinion or tip and root relief on one of the members (Refs. 9–10).

2. For tip relief, start the modification at the highest point of single-tooth contact, and for root relief start at the lowest point of single-tooth contact (use highest and lowest points of double-tooth contact for spur gears with contact ratios greater than 2.0).

3. The amplitude of the relief should be at least as great as the peak mesh deflection at the load in which smooth motion is desired. If one wishes to compensate for spacing errors, one should add the peak tooth-to-tooth spacing error of each gear to the mesh deflection value (Ref. 11).

4. Most investigators have used either a linear or parabolic shape for the relief (Refs. 12–13).

5. Some gears required some combination of linear and parabolic modification in order to obtain more perfect compensation for the nonlinear tooth pair deflection (Ref. 14).

The above procedure yields a modification that works well at the "design" load, but as the load either increases or decreases, the motion error will increase (Ref. 15). In order to reduce the torque sensitivity, a scheme that combines profile modifications with lead crowning has been applied (Ref. 16).

Another approach to the reduction in noise excitations at lower loads is to use what is called "short relief," where the start of the modification is moved closer to the tooth extremes (Refs. 9 and 17).

In this scheme, it is possible to have zero transmission error at no load and still have a reduction of excitation at higher loads. However, the reduction at the design load will be much less than for the "long relief" method described above.

For narrow face width spur gears, correction in the face width direction is usually not used, but for medium-to-wide face widths, lead crowning may be needed in order to compensate for lead errors and misalignment. When lead crowning is used, one must reassess the scheme for determining the best modification.

Helical gearing. In order to get adequate benefits of the axial load sharing of helical gears, they usually have medium-to-wide face widths that likely require some lead crowning in order to compensate for misalignment. Many gear researchers (Refs. 18–27) have shown the effects of crowning shape and amplitude on load distribution. In each instance it was shown that for a given level of misalignment, there is a range of crowning that will provide a

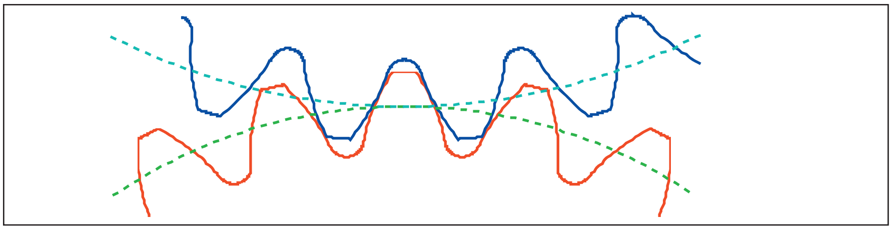


Figure 1—Schematic of Gears in Mesh.

Table 1: Helical Gear Geometry

	GEAR1		GEAR2
Number of Teeth	25		31
Module (mm)		2.7796	
Pressure angle (deg)		22.21	
Helix angle (deg)		28.9	
Operating center distance (mm)		88.9	
Outside diameter (mm)	85.293		104.343
Root diameter (mm)	71.399		90.449
Face width (mm)	31.750		31.750
Standard pitch diameter, SPD (mm)	79.375		98.425
Transverse tooth thickness at SPD (mm)	4.888		4.888
Profile, face, total contact ratio		1.37 / 1.76 / 3.13	

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Professor Emeritus Donald Houser *received his BS, MS and PhD degrees from the University of Wisconsin in Madison. He served on the faculty of the Department of Mechanical Engineering at Ohio State for 35 years. Dr. Houser is founder of the Gear Dynamics and Gear Noise Research Laboratory (GearLab), an industrial research consortium that currently has over 45 participating companies. He has supervised over 115 graduate student theses and has consulted on gearing problems with over 50 companies. Professor Houser has authored over 130 publications, most of them in the gearing area.*

reduction in the peak root and contact stresses, and will shift the peak stresses from the edge of the tooth to the center.

However, it was also shown that while excessive crowning can certainly shift the peak contact stress closer to the center of the tooth, it will also result in an increase in the peak contact stress due to the concentration of load in the center of the crown region. These investigators, however, seldom mentioned the effects of crowning on gear noise excitation.

Numerous investigators have developed schemes for creating optimum modifications for helical gears, most of them based on the minimization of transmission error. Several researchers (Refs. 28–32) have developed methods for simultaneously studying the profile and lead modifications of helical gears. Umezawa's (Ref. 30) performance diagram that allows the study of many variables is similar to what is done in this study, but his emphasis was much more related to the best gear geometries to minimize dynamics.

Finally, it has been shown that it is possible to obtain a zero-transmission-error surface topography for any gear pair (Ref. 33). Constraints are that the method will work at only one load, and it requires that the load distribution be defined beforehand. In addition, the modifications that result have peculiar shapes that would be difficult to manufacture.

Load Distribution Analysis Procedure

When considering topographical modifications and misalignment effects on the performance of gearing, it is necessary to have a computation procedure that determines the load distribution along the lines of action of the gears. In this paper, the basis of the load distribution analysis comes from the research of Conry and Seireg (Ref. 34). However, similar approaches have been used by a number of investigators (Refs. 35–37). The work of Conry and Seireg has led to the creation of a computer program called the *Load Distribution Program* (Ref. 38) at the GearLab at The Ohio State University. Unlike general purpose, finite element approaches that are often used for this type of analysis, the approach used here is—computationally—extremely fast, allowing the performance of numerous simulations in a short time. Microgeometry modifications can be applied to the load distribution solver by treating them as the initial separation of the gears in mesh. A brief discussion of the bases of this program is provided in an annex at the back of this paper.

Outline of Procedure of Topographical Modifications

The following is an outline of the graphical procedure that is used to come up with optimal modifications. This procedure essentially allows the user to observe the effects of any of a number of response variables, such as root and contact stresses; noise excitations such as transmission error and sum-of-forces; and tribology properties such as surface temperature and film thickness. Sum-of-forces is the sum of the first harmonic of transmission error, shuttling force and friction force (Ref. 39). The effect of any of these variables may be viewed as a function of torque in a number of ways. The designer now has the opportunity to select the modification that can provide adequate noise response over a broad range of loads while at the same time satisfying load distribution, contact stress and bending stresses requirements at the higher loads. The final process of the evaluation is to perform a manufacturing sensitivity analysis to check the design's sensitivity to random errors in manufacturing and assembly, such as housing misalignment, lead and profile errors.

1. Select two design variables and their ranges. Possibilities include profile crown, profile slope, tip relief, lead crown, misalignment, shaft deflection and bias modification.

2. Select evaluation torques and run the load distribution program.

3. Select design evaluation metrics (transmission error, sum-of-forces, contact stress, bending stresses, surface temperature and film thickness).

4. Create a 2D parameter map, and move the cursor around on the appropriate evaluation metrics map at the selected torque until the user achieves a desirable torque response for that pair of variables. One may repeat this selection to compare the performance of various combinations of these variables. At each selection, a graph versus torque will be made for that variable and each selection is superimposed on the same graph. One may look at other design metrics and then select the composite "best" design or designs.

5. With the chosen values of the design variables, select two additional design variables and repeat the above procedure. One may need to iterate on the original variables upon completion of initial evaluations, but generally, iteration is not required.

6. Perform robustness analysis of the selected designs.

Procedure Example

The procedure is quite flexible in the order in which variables are analyzed and in the number of variables that are considered. Therefore, an example of only one procedural possibility is presented. Usually, it was found that the most important variables should be dealt with in the early steps. One approach to rank ordering the importance of the design variables is with a Taguchi-type, factorial design analysis (Ref. 40).

In the example that is shown below, a helical gear pair that was designed and tested by NASA (Ref. 41) will be optimized to minimize noise excitations at a range of pinion torques between 100 and 250 Nm (mean value is 175 Nm). The rated design torque for these gears is roughly 400 Nm. Figure 1 shows a transverse plane schematic of the gears in mesh. Table 1 presents a summary of the geometry of this gear pair.

Selection of lead crown to compensate for misalignment. There are essentially three types of decisions that are made regarding the selection of lead crown, including:

1. Use none; this may be done for narrow face widths or when extremely accurate manufacturing methods are used.

2. Use company-specified standards that are based on company experience.

3. Establish the amount of peak misalignment and perform a load distribution analysis to establish a level of crown that shifts the peak load from the tooth edge. Usually this evaluation is done at the design load since the goal is to control the root and contact stresses.

Here, the latter situation will be demonstrated. One method is to use an AGMA-quality number as a reference. For example, AGMA A8 quality will be assumed to establish the manufacturing misalignment. This number will be doubled in order to account for the misalignment due to the housing, giving a peak misalignment of 15 μm across the face width of the gear set. Because of the relatively narrow face width of the gear pair, circular lead crowning will be applied so that the contact stresses and root stresses do not peak at the edge of the gear. Figure 2 shows the typical contact stress without any misalignment (peak at 1,225 MPa). Figure 3 shows the contact stress of the misaligned part (peak at 1,600 MPa). Figure 4 shows the contact stress of the misaligned part with 5 μm of lead crown (peak at 1,350 MPa). One can see that for this 5 μm lead crown, the contact stresses at the edge are avoided. In each case, a sharp profile tip modification is applied so that the high stress at the corner of the tip and root can be reduced or ignored. Table 2 shows contact stress and bending stresses for different misalignment and lead crown.

Another method is to use the 2D parameter maps of contact stress and bending stress for lead crown versus misalignment, as shown in Figure 5.

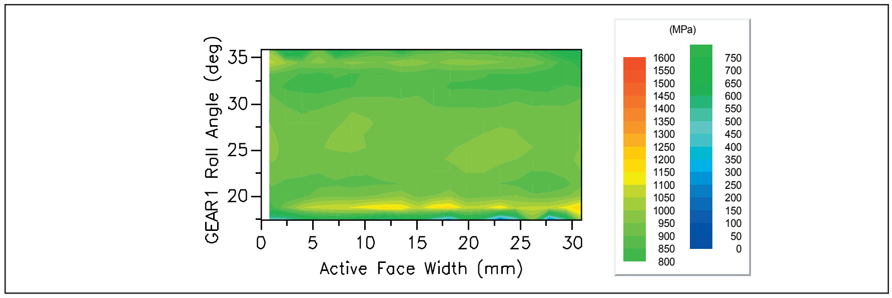


Figure 2—Contact Stress at 400 Nm (No Misalignment).

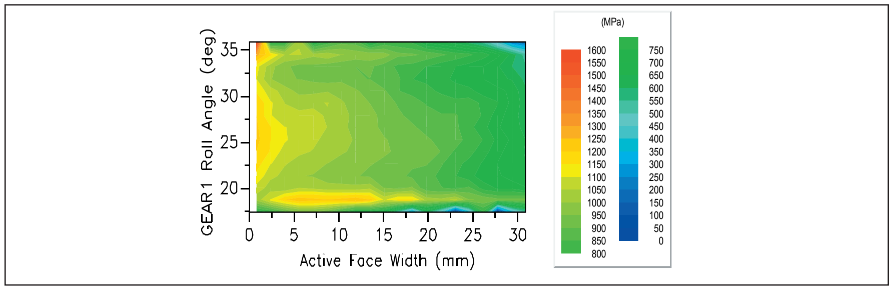


Figure 3—Contact Stress at 400 Nm (With Misalignment).

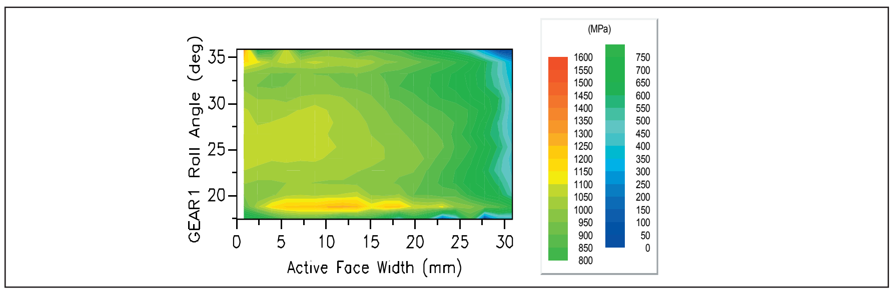


Figure 4—Contact Stress at 400 Nm (5.07 μm Lead Crown and 15.23 μm Misalignment).

Table 2: Summary of Contact and Bending Stress at the Design Torque for Several Misalignment and Lead Crown				
Misalignment	Lead Crown	Stresses (MPa)		
(μm)	(μm)	Contact	GEAR1	GEAR2
0	0	1225	230	234
15	0	1600	348	306
15	5	1350	310	267
0	5	1250	247	252

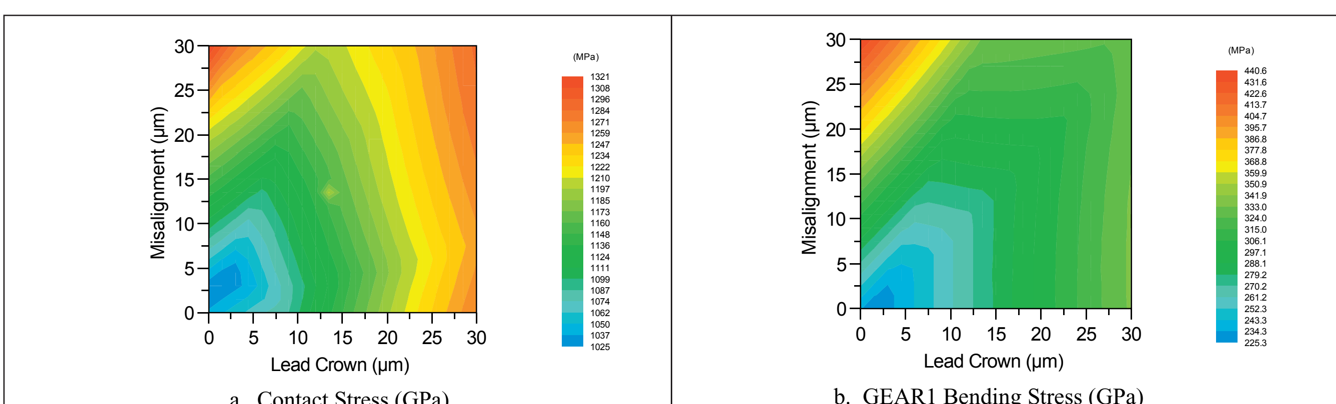


Figure 5—Interaction of Misalignment and Lead Crown at Design Torque (400 N-m).

This method could be used to establish the appropriate lead crown needed at known misalignment values. Using this method, it would seem like the best lead crown would be about 7 μm for contact stress and about 10 μm for bending stress.

In the next stage of the analysis, the interactions between lead crown and profile crown will be performed for perfectly aligned teeth.

Profile crown and lead crown interactions. Experience has shown that circular crown provides nearly optimal modification for the minimization of transmission error of helical gears. Beside that, profile crown and lead crown are the two most important variables in minimizing transmission error. Thus, it is logical that they are the first variables to be considered when minimizing transmission error.

In this case, standard parameter maps were generated for lead crown and profile crown amplitudes from 0–50 μm.

Maps of peak-to-peak transmission error for six torque levels are shown in Figure 6. One may traverse any of the parameter maps with a mouse cursor and by clicking it, one may select any combination of profile and lead crowns. Then one can plot any of a large number of gear design metrics (transmission error, sum-of-forces, contact stress, bending stress, etc.) versus torque for that combination of crowns. Each time a new combination is clicked, a new set of torque plots will be superimposed on the plots of design metrics.

One example—Figure 7—shows such a set of plots with the letters A, B, C, D and E being the selected pairs. From the peak-to-peak transmission error versus torque plots, we see that each of the crowning pairs gives a minimum transmission error at a different torque. The second noise metric—sum-of-forces—is less clear with regard to the minimum values; but if the procedure is repeated using the sum-of-forces parameter map, one might have found better minimum values for this variable.

The remaining plots show how the other design metrics respond to the increasing profile and lead crowning. Usually, as we increase crowning, stresses go up since loads are being concentrated more in the center region of the tooth-meshing zone.

Since the center of the noise torque region is at 175 Nm, we will try to find the profile crown and lead crown that would minimize the peak-to-peak transmission error at this torque and its surrounding torques. Figure 8 shows the profile crown and lead crown (AB, AC, AD and AE) that are used for the next runs that are used to select the second-order tooth modifications.

Profile slope and lead slope interactions. The next step is to use the selected cases from profile crown and lead crown (AB, AC, AD and AE) and evaluate the interaction of profile slope and lead slope from –25 to 25 μm.

Figure 9 shows the peak-to-peak transmission error for case AE. At one torque, changing the profile slope and lead slope interaction may improve the design matrix, while the reverse may be true at other torque. The behavior of one torque might be different at another torque. For example, at 50 Nm, one could see that for –10 to 10 μm lead slope, one could select a large range of profile slope. However, at 400 Nm torque value, increasing lead slope and increasing profile slope are required to

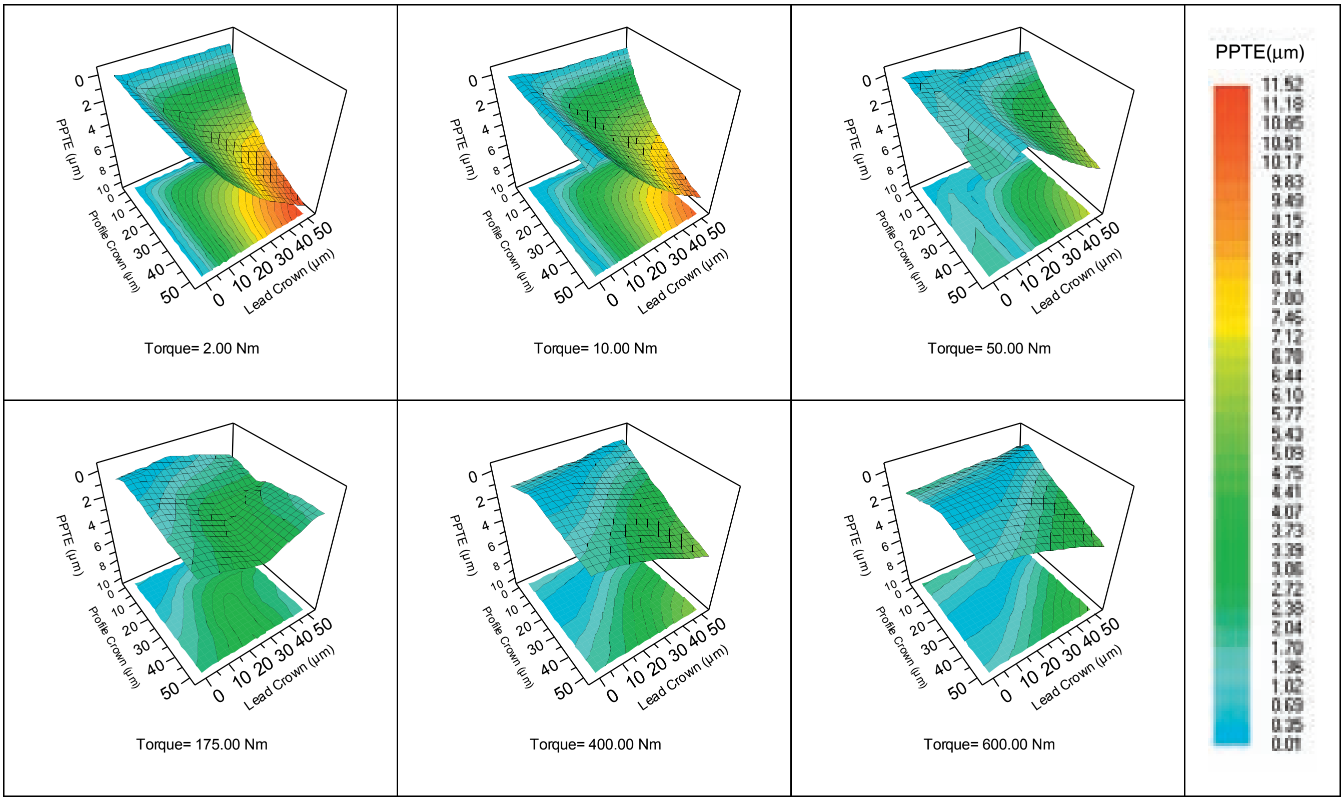


Figure 6—Peak-to-Peak Transmission Error for Profile Crown and Lead Crown Modification.

have low peak-to-peak transmission error. The user could move around the mouse cursor to find the relationship of profile slope and lead slope. Here cases AE0, AE1, AE2, AE3 and AE4 will be chosen.

Although not shown here, a similar procedure is used for the other cases (AB, AC and AD). Thus, the selected designs would have their own profile crown, lead crown, profile slope and lead slope. By and large, the profile slope and lead slope modifications do not have much effect on the transmission error results, but in a couple of instances, some improvements were observed. It is interesting to note that the maximum transmission error across the entire torque region is about 1.25 μm. To reduce the next interaction study (bias modifications), one or two of each main case are used, which is shown in Figure 10.

There are a total of seven selected designs—AB0, AC0, AC1, AD0, AD2, AE0 and AE2.

Bias modification interactions. Bias

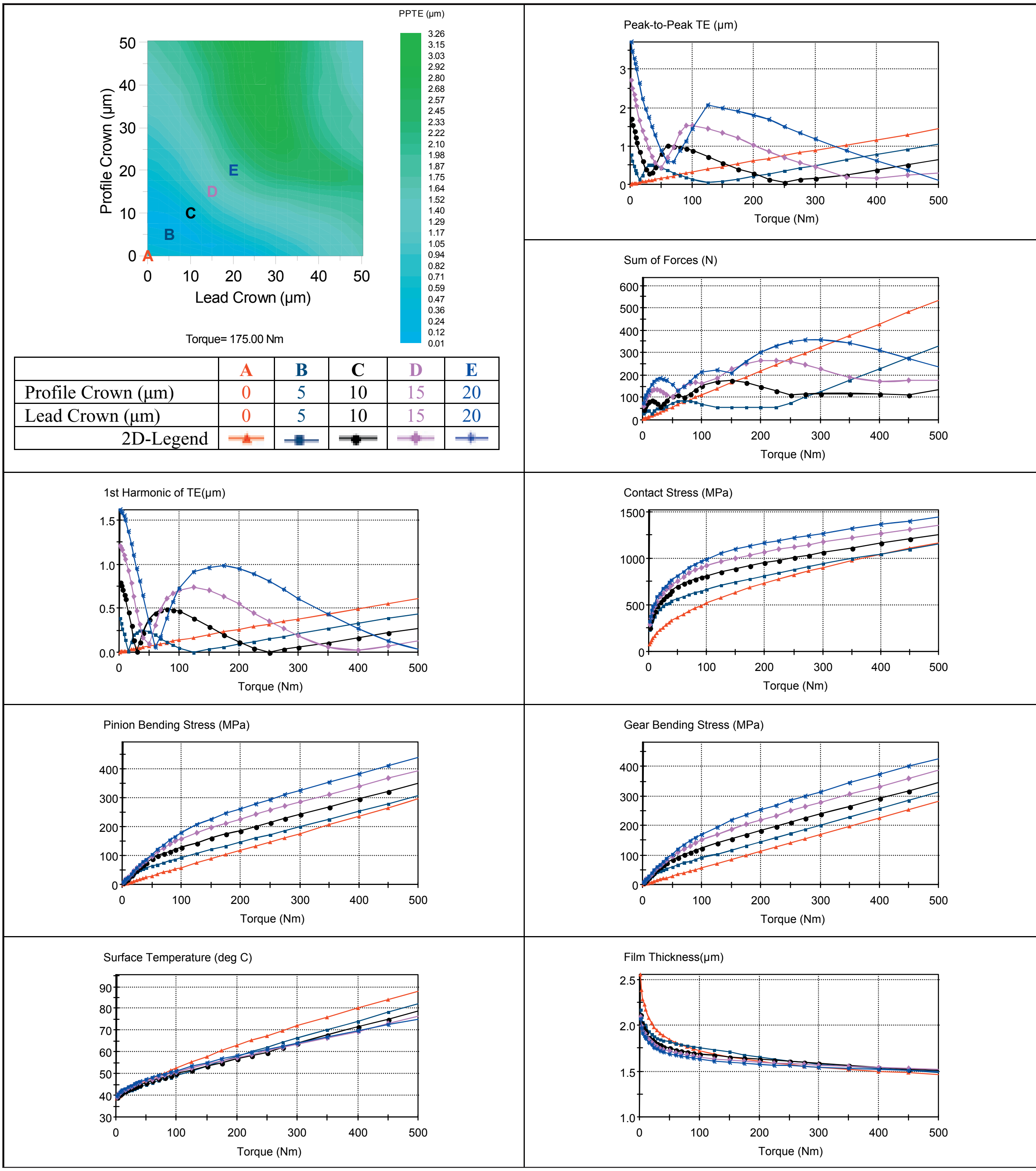
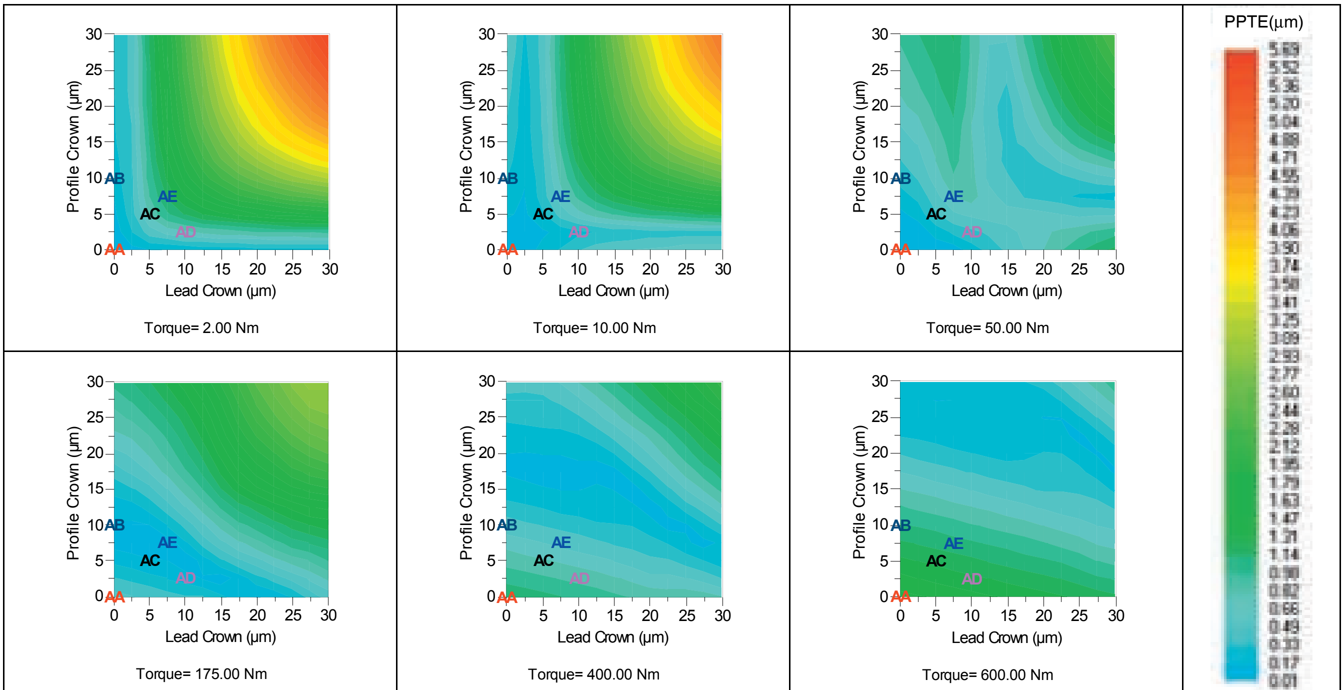


Figure 7—Interaction of Selected Profile Crown and Lead Crown to Several Gear Design Metrics.

modifications (twist in the tooth shape) from –7.5 to 7.5 μm are used for the next procedure. "Bias out" modifications remove more material from the entering and leaving regions of contact, and "bias-in" modifications remove material from the opposite corners of the teeth. Table 3 shows the bias that was added to the earlier modifications. Figure 11 shows the schematic on applying the bias modification in the load distribution program.

Figure 12 shows the effect of bias modification of several design metrics for case AE0. Here, adding bias-in modification seems to improve the peak-to-peak transmission error and sum-of-forces, and at the same time it does not increase the contact stress from the no-bias case. For this particular case, –5 μm (bias-in, case AE02), seems to be the best selection. The same procedure is done (not shown here) for all other cases and the bias modifications are selected that optimize the design metrics.

Figure 13 shows the comparison of these selected modifications. Cases AC03 and AD02 seem to be the better modification, when compared to other cases from the peak-to-peak



	AA	AB	AC	AD	AE
Profile Crown (μm)	0.0	10.0	5.0	2.5	7.5
Lead Crown (μm)	0.0	0.0	5.0	10.0	7.5
2D-Legend					

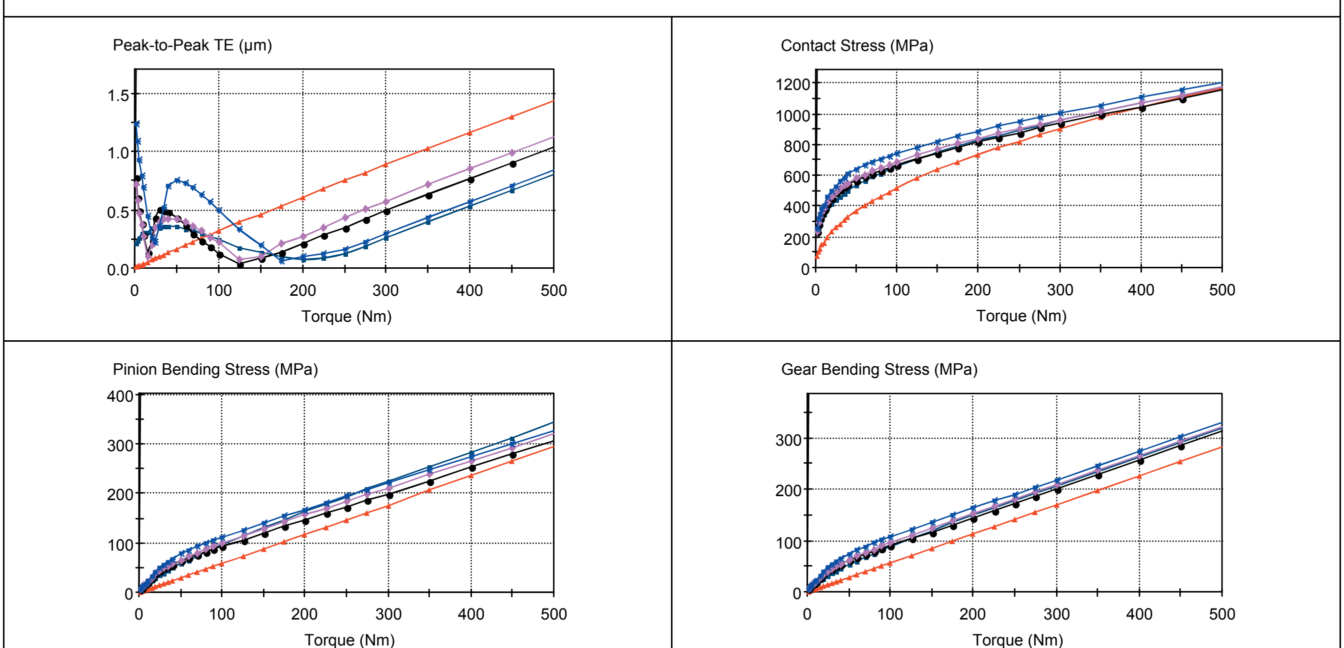
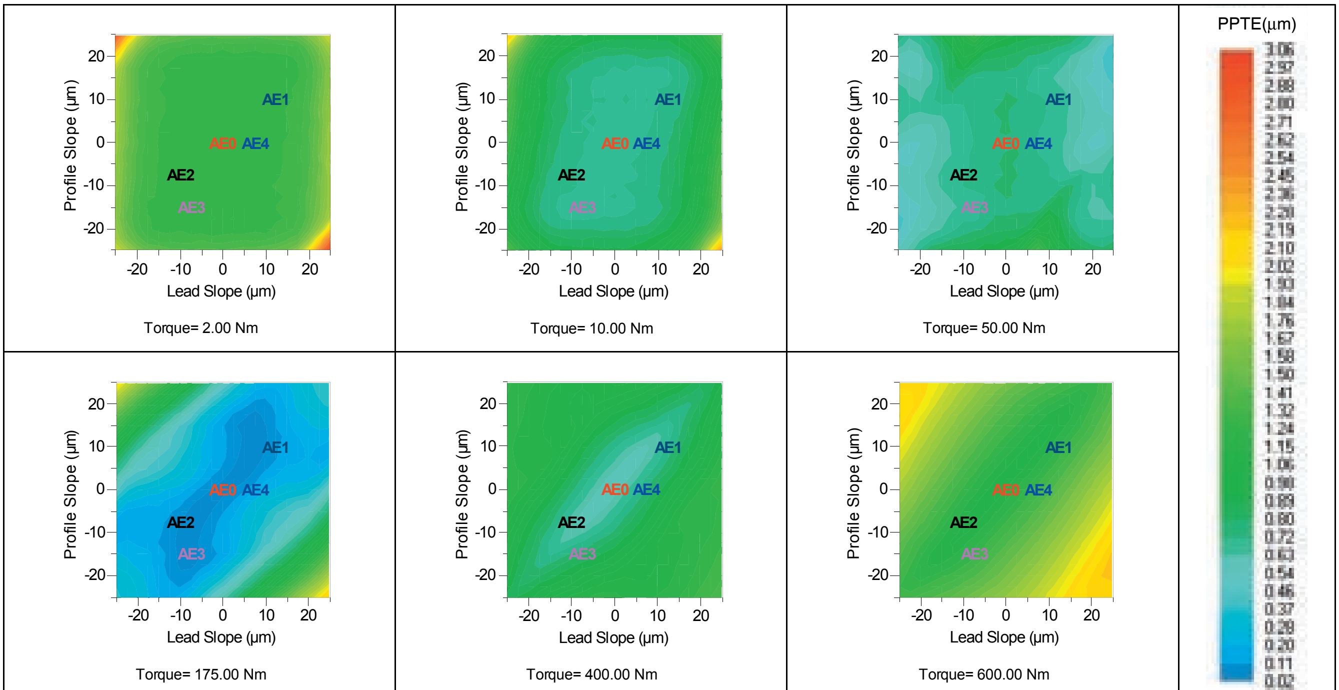


Figure 8—Several Gear Design Metrics Results using Selected Profile Crown and Lead Crown Modifications.

transmission error point of view at noise torque ranges. However, cases AB02 and AE02 seem to be the better modification compared to other cases from the sum-of-forces point of view. Case AC03 seems to be the better modification to the other cases from a contact stress point of view. Bending stresses show similar results as the contact stress (not shown here). Figure 14 shows the baseline modification for each selected modification. Thus, the gear designers would need to compromise between the noise and stresses for the modification to be used as a baseline. Next step would be to analyze which of these modifications are less sensitive to manufacturing error (robustness analysis).

Robustness analysis. Robustness analysis is a Monte Carlo-type simulation (Ref. 42) that applies randomly selected errors to profile slope, profile curvature, lead slope (includes misalignment), lead curvature and bias modifications. In this analysis, the standard deviation of each variable is created from the AGMA A8 accuracy specifications by which 100 manufacturing errors are selected randomly from the normal distributions for the gear. The distributions of errors



	AE0	AE1	AE2	AE3	AE4
Profile Slope (µm)	0.0	10.0	-7.5	-15.0	0.0
Lead Slope (µm)	0.0	12.5	-10.0	-7.5	7.5
2D-Legend					

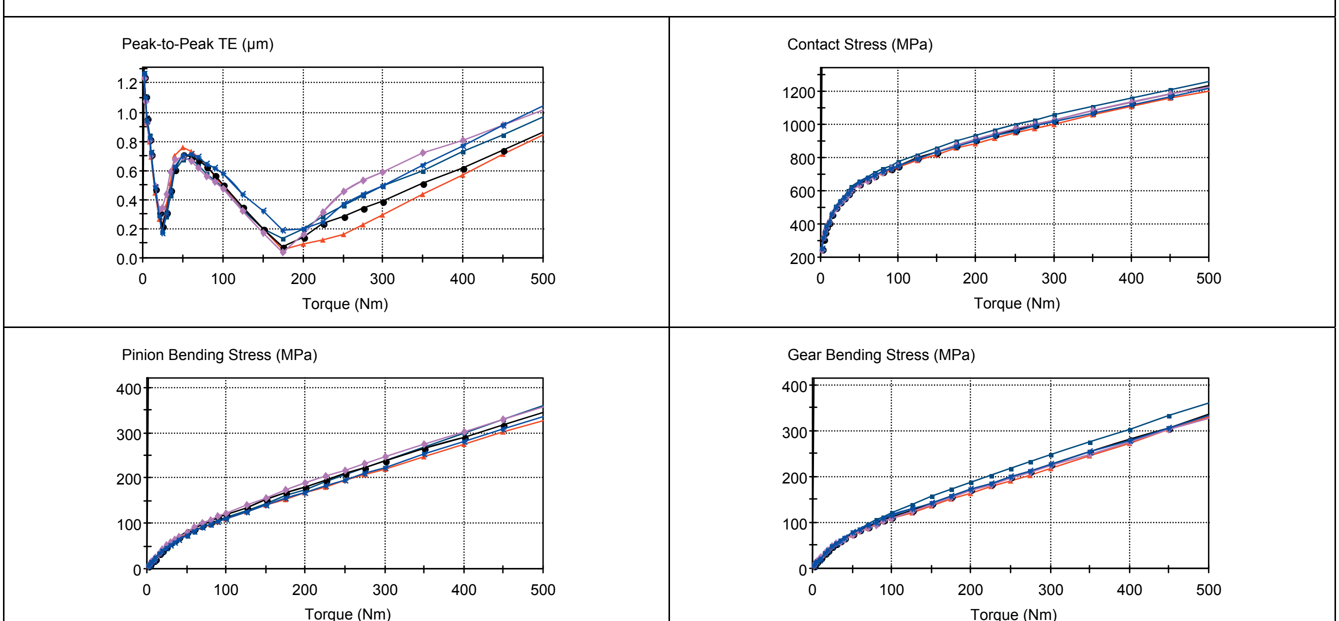


Figure 9—Peak-to-Peak Transmission Error for Varying Profile Slope and Lead Slope with a Fixed 7.5 µm Profile Crown and 7.5 µm Lead Crown (Case AE).

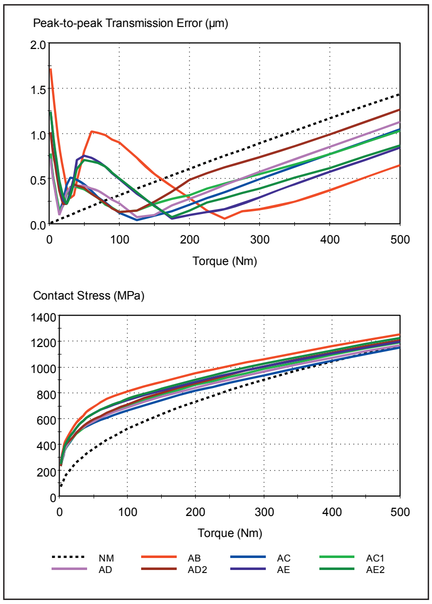
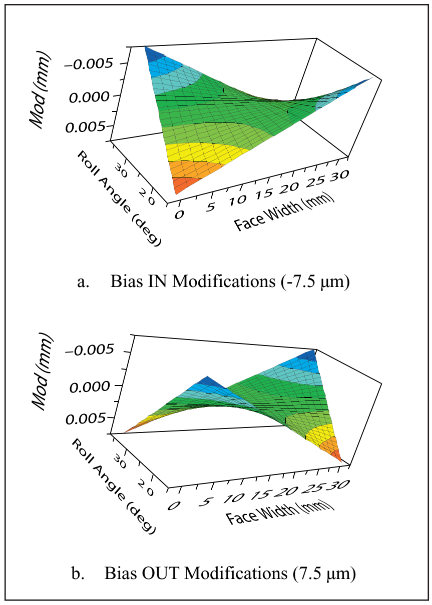


Figure 10—Stacking Selected Cases.



a. Bias IN Modifications (-7.5 µm)

b. Bias OUT Modifications (7.5 µm)

Figure 11—Bias Modification on GEAR1.

Table 3: Bias Modification Information

Case	Bias Modification (µm)	Type
file1	-7.5	Bias-In
file2	-5.0	Bias-In
file3	-2.5	Bias-In
file4	0.0	No Bias
file5	.5	Bias-In
file6	5.0	Bias-In
file7	7.5	Bias-In

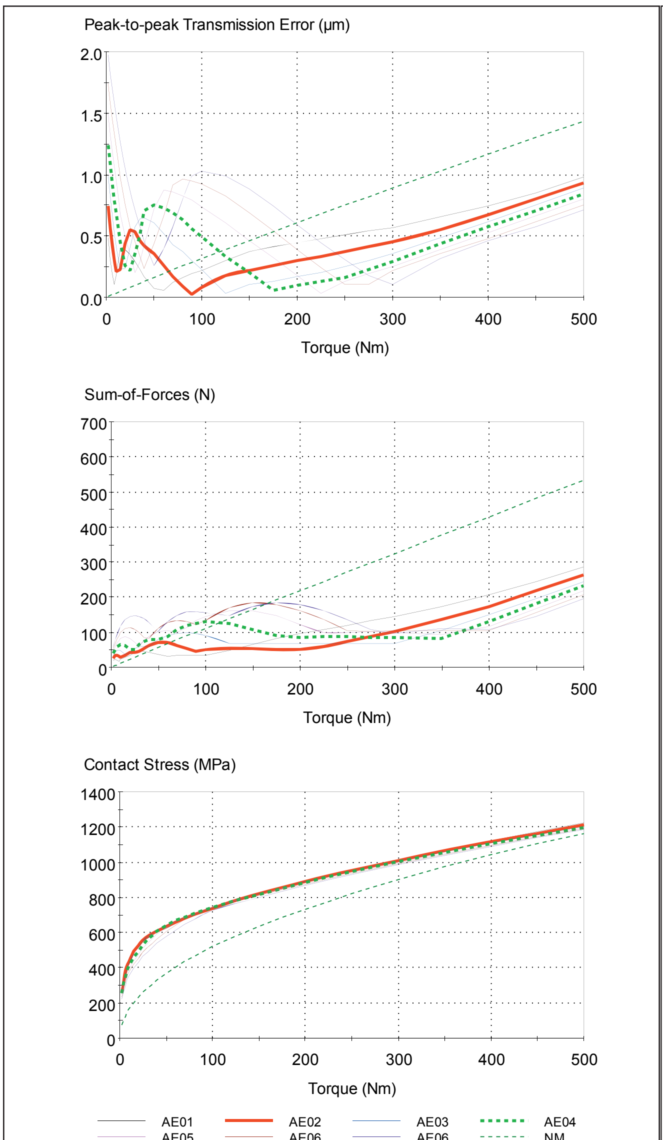


Figure 12—Effect of Bias Modifications (-7.5 to 7.5 µm) to Case AE (7.5 µm Profile Crown, 7.5 µm Lead Crown, 0.0 µm Profile Slope, and 0 µm Lead Slope.)

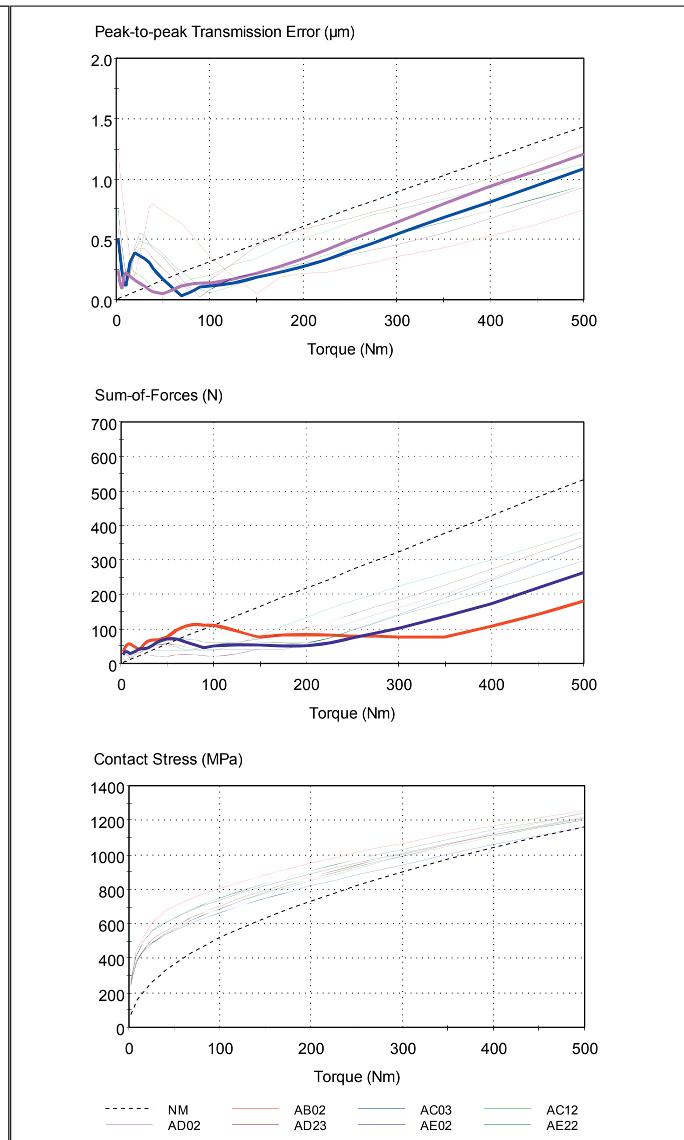


Figure 13—Summary Results of Selected Design.

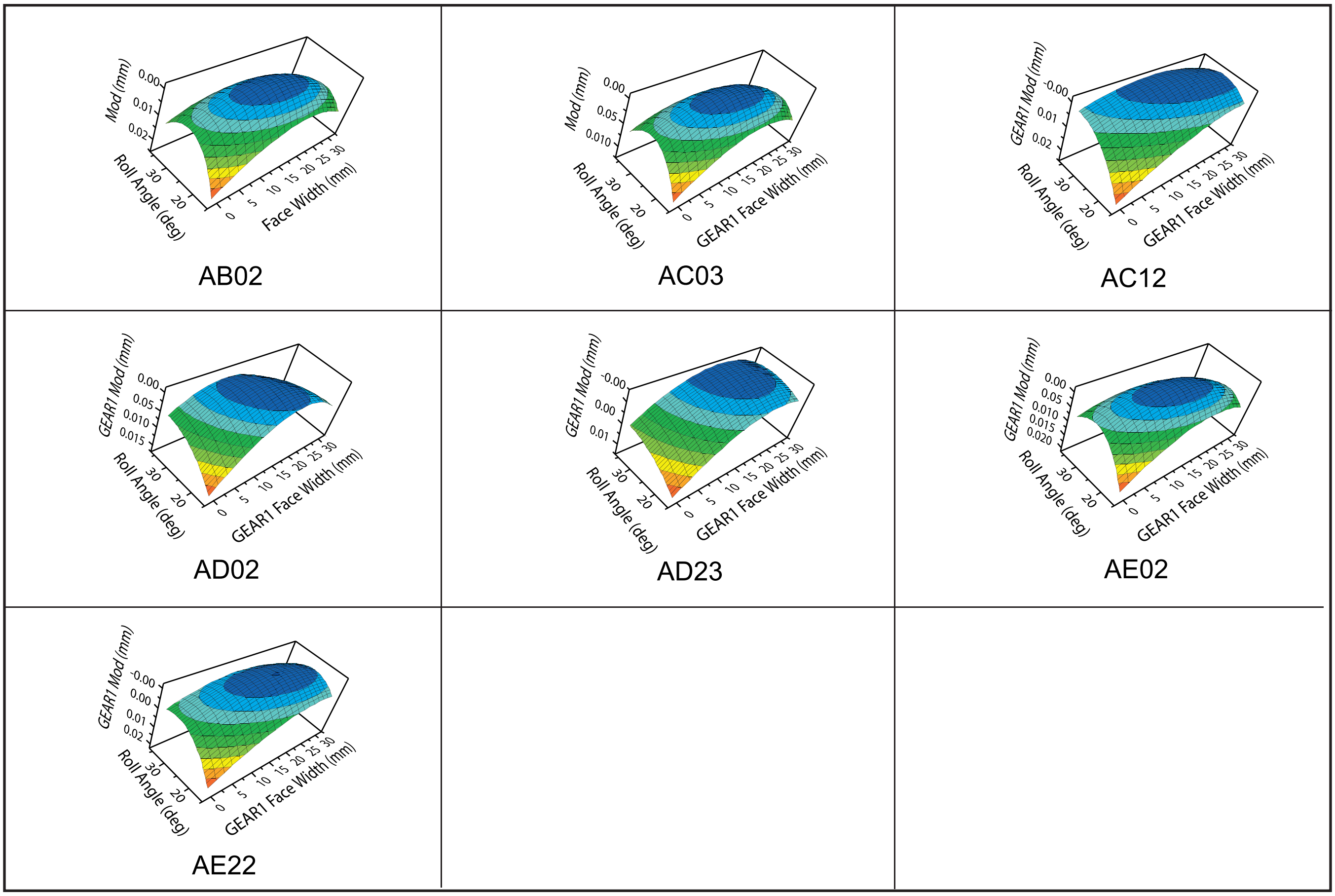


Figure 14—Total Modification For Selected Case.

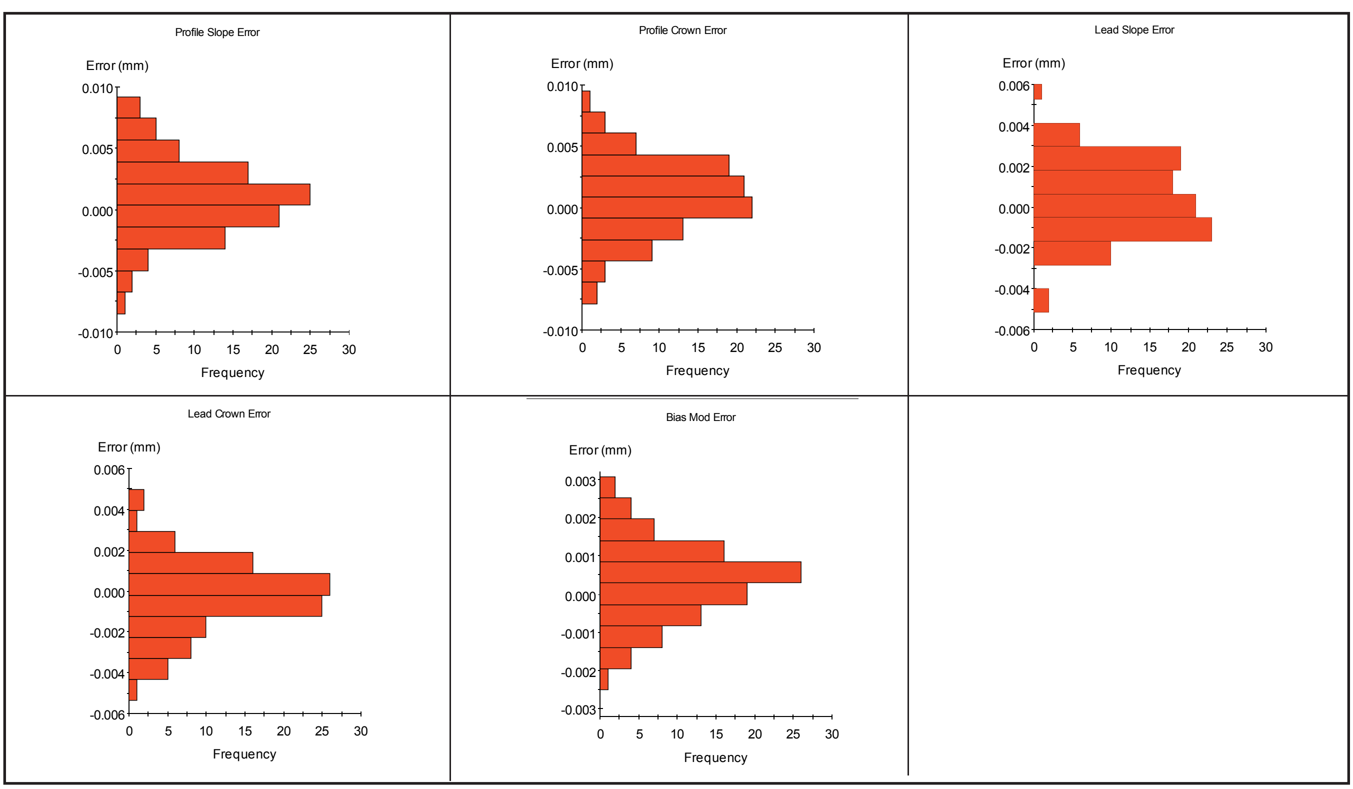


Figure 15—100 Random Samples of Manufacturing Error.

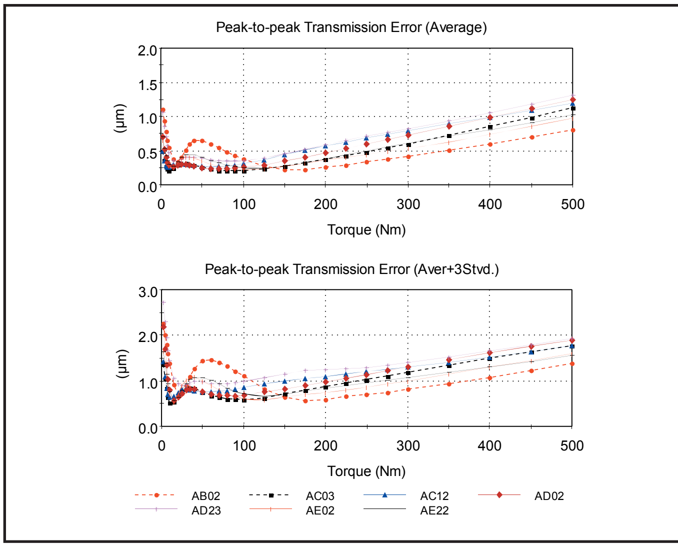


Figure 16—Robustness Analysis (Average and Aver+3 Stvd) for Peak-to-Peak Transmission Error.

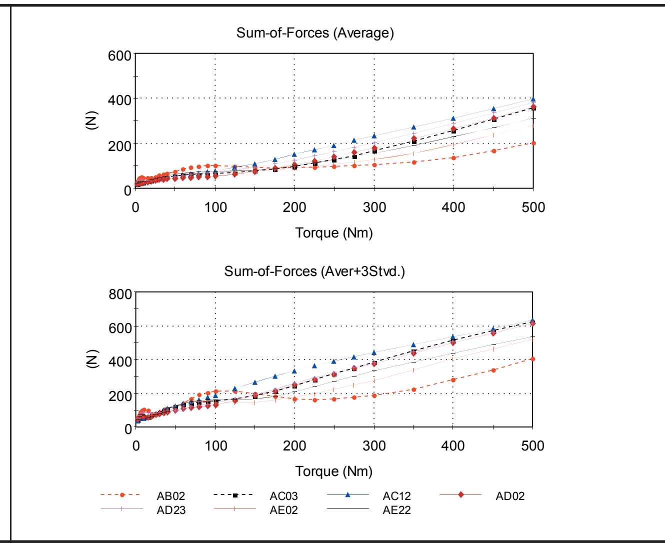


Figure 17—Robustness Analysis (Average and Aver+3 Stvd) for Sum-of-Forces.

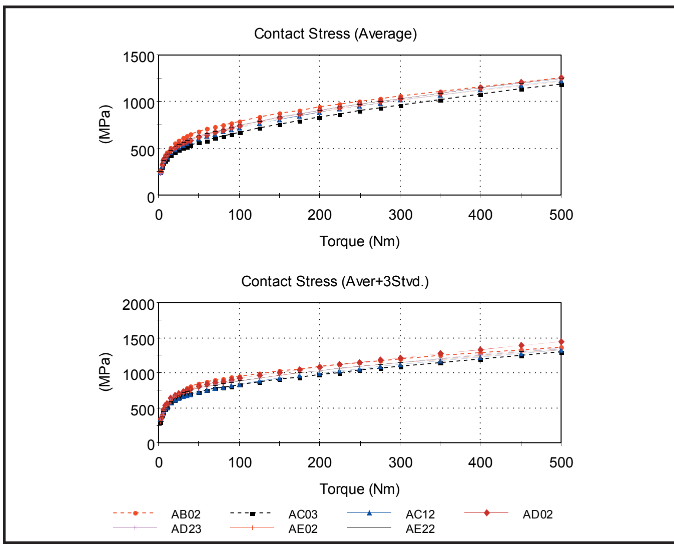


Figure 18—Robustness Analysis (Average and Aver+3 Stvd) for Contact Stress.

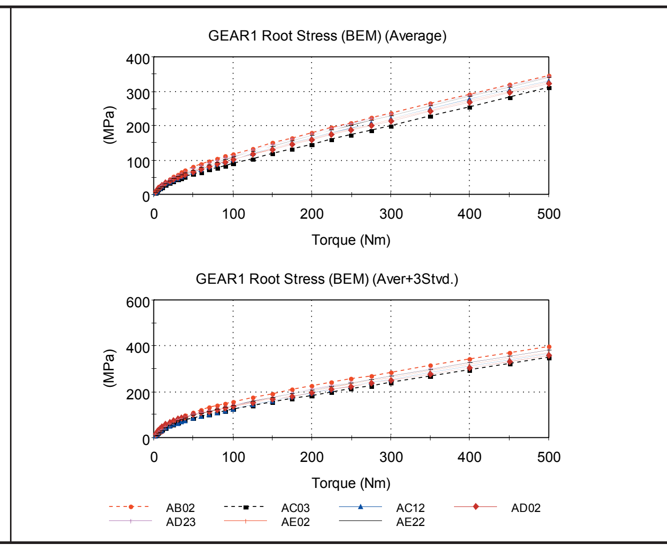


Figure 19—Robustness Analysis (Average and Aver+3 Stvd) for GEAR1 Bending Stress.

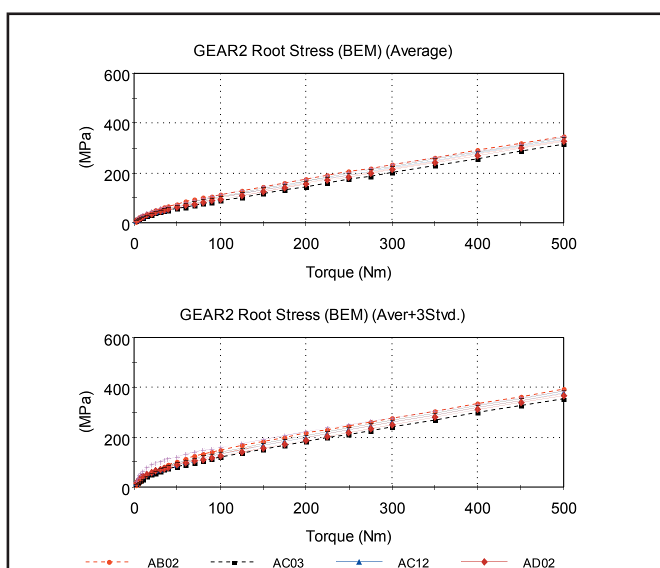


Figure 20—Robustness Analysis (Average and Aver+3 Stvd) for GEAR2 Bending Stress.

of each of the variables are shown in Figure 15. Then, load distribution simulations are performed from low to high torque values.

Results are shown for each design metric in Figures 16–20. Results are shown in terms of the average value of each 100 simulations and the mean-plus-three standard deviations (Aver+3Stvd), which gives an indication of the worst-case situation. Designs AB02, AE02 and AE22 each have transmission error average values that do not exceed 0.5 μm over the noise design torque range, and the same design's mean-plus-three values stay under 1 micron for the same torque range.

From a sum-of-forces viewpoint, the design AB02 data seems to be the best. For these designs, the AC03 design has the lowest root and contact stresses. The topography for these designs is provided in Figure 14.

Summary

This paper has presented an interactive, graphical procedure for determining gear tooth topography designs that minimize the noise and stresses of gears. The method is also a valuable educational tool for understanding the effects of numerous topographical changes of the tooth surface on gear performance.

For the example helical gears, the following conclusions may be drawn:

1. Lead crown to adequately compensate for the misalignment is from 1/2 to 1/3 of the amount of misalignment across the face width.

2. Minimizing root stress requires a greater lead crown than does minimizing contact stress.

3. Profile and lead crown are the most important modifications for minimizing transmission error.

4. Profile slope and lead slope corrections have little effect on transmission error (shaft deflections are not considered here).

5. Bias modifications provide some improvements in transmission error over the noise design torque range, but the type of bias that is used depends upon the initial values of profile and lead crown.

6. Designs that are best for the perfect gear set are usually also the most robust gear designs.

In the future, there is a desire to apply these techniques to a broader range of gear geometries to see if the above conclusion can be extended to a broader range of gear geometries.

Acknowledgments

The authors would like to thank the sponsors of the Gear Dynamics and Gear Noise Research Laboratory (GearLab) at The Ohio State University for their support of the development of the modeling procedures used for the gear analyses presented in this paper.

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Annex
Formulation of the Load Distribution Calculation Procedure

The formulation of the solution of the load distribution (Ref. 1) in gears is equivalent to the formulation of the solution of the generalized, elastic contact problem. The discussion that follows is a condensation of the work of Conry and Seireg (Ref. 2) pertaining to elastic bodies in contact. Given the compliance of each point in the contact zone—the initial separations (or approaches) under zero-load and the applied load—the load distribution and the overall system rigid body rotation may be obtained using a modified, Simplex-type algorithm.

All elastic deformations and forces are assumed to be acting along the line of action in the transverse plane. For the gear teeth to be in contact at any point, the sum of the elastic deformations and initial separations must be equal to the rigid body displacement of the point with respect to the reference line. To determine the position of contact, the gear teeth are taken to be of perfect, involute form. Any tooth surface errors are interpreted as initial separations or approaches.

Two criteria are proposed for the mathematical formulation of the solution of the contact problem. The condition of compatibility of deformation outlines the condition for which points may come into contact. The condition of equilibrium assures that the sum of torques acting on the system are zero.

For any point, k, in the contact zone, the total sum of elastic deformations and initial separations must be greater than, or equal to, the rigid body approach along the line of action. This condition may be written as:

$$W_{1k} + W_{2k} + e_k \geq R_b\theta \qquad (1)$$

The sum of all torques acting on a gear body must be zero. The sum of moments about the line of action must be equal, but opposite in sign, to the applied torque. This condition is represented as:

$$\sum_k (F_k \times R_b) + T = 0 \qquad (2)$$

The compatibility conditions as defined by the inequality equation (1) may be transformed into an equation of equality through the introduction of a slack variable, Y_k. Equation 1 may now be written as

$$W_{1k} + W_{2k} + e_k - R_b\theta - Y_k = 0 \qquad (3)$$

Consequently, if $Y_k > 0$, then the two bodies are not in contact at point k and $F_k = 0$. If $Y_k = 0$, then contact exists and $F_k \geq 0$. Thus, the solution to the load distribution problem may be stated as follows

$$-[S][F] + R_b\theta\,[i] + [I\,][Y] - [e] = 0 \qquad (4)$$

$$[I^{\,t}] + [F]R_b + T = 0 \qquad (5)$$

A modified Simplex type algorithm is then used to solve for the load distribution.

The major assumptions that are used in the load distribution calculation are:

1. All contact is along the line of action. This assumption does not allow so called "corner" contact that occurs when the modifications are not sufficient to compensate for tooth deflections as teeth enter and leave contact. There is a corner contact option that does allow this to be included for both spur and helical gearing (Ref. 3).
2. The edges of helical gear teeth are modeled as being perpendicular to the normal plane.
3. Rims and webs are assumed to be solid.
4. Tooth bending and shear deflections are computed using a Rayleigh-Ritz solution of a tapered plate model (Ref. 4).
5. Additional tooth deflection components include Hertzian deflections (Ref. 5) and deflection of the tooth base (Ref. 6).

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Nomenclature

W_{1k} W_{2k}	total elastic deformation of point k on body 1 and body 2
e_k	initial separation at point "k" between body 1 and body 2
R_b	base radius
F_k	the discrete force acting at point "k"
T	the applied torque
Y_k	slack variable
$[S]$	matrix of the influence coefficient
$[F]$	vector of forces
$[I]$ $[I_t]$	vector of ones; the transpose of $[I]$
$[Y]$	vector of slack variable
$[e]$	vector of initial separation

The Kinematics of Conical Involute Gear Hobbing

Carlo Innocenti

(This article first appeared in the proceedings of IMECE2007, the 2007 ASME International Mechanical Engineering Congress and Exposition, November 11–15, 2007, Seattle, Washington, USA. It is reprinted here with permission.)

Management Summary

It is the intent of this presentation to determine all rigid-body positions of two conical involutes that mesh together, with no backlash. That information then serves to provide a simple, general approach in arriving at two key setting parameters for a hobbing machine when cutting a conical (beveloid) gear. A numerical example will show application of the presented results in a case study scenario. Conical involute gears are commonly seen in gearboxes for medium-size marine applications—onboard engines with horizontal crankshafts and slightly sloped propeller axes—and in automatic packaging applications to connect shafts with concurrent axes whenever the angle between these axes is very small (a few degrees).

Introduction

Conical involute gears, also known as beveloid gears, are generalized involute gears that have the two flanks of the same tooth characterized by different base cylinder radii and different base helix angles (Refs. 1– 4). Beveloid gears can be mounted on parallel-, intersecting- or skew-axis shafts. They can be cheaply manufactured by resorting to the same cutting machines and tools employed to generate conventional, involute helical gears. The only critical aspect of a beveloid gear pair, i.e., the theoretical punctiform (single-point) contact between the flanks of meshing gears, can be offset by a careful choice of the geometric parameters of a gear pair (Refs. 5–7). On the other hand, the localized contact between beveloid gear teeth comes in handy should the shaft axes be subject to a moderate, relative position change in assembly or operation.

The technical literature contains plenty of information regarding the tooth flank geometry (Refs. 8–9) and the setting of a hobbing machine in order to generate a beveloid gear (Refs. 10–15). Unfortunately, the formalism usually adopted makes determination of the hobbing parameters a rather involved process, mainly because the geometry of a beveloid gear is customarily—though inefficiently—specified by resorting to the relative placement of the gear with respect to the standard rack cutter that would generate the gear, even if the gear is to be generated by a different cutting tool. To make things worse, some of the cited papers on beveloid gear hobbing are hard reading due to printing errors in formulae and figures.

This paper presents an original method to compute the parameters that define the relative movement of a hob with respect to the beveloid gear being generated. Pivotal to the proposed method, together with a straightforward description of a beveloid gear in terms of its basic geometric features, is the determination of the set or relative rigid-body positions of two tightly meshing beveloid gears. Based on this set of relative positions, the paper shows how to assess the rate of change of the hob-work shaft axis distance as the hob is fed across the work, as well as the rate of the additional rotation of the hob relative to the gear. These parameters have to be entered into the controller of a CNC hobbing machine in order to generate a given beveloid gear.

The proposed method also can be applied to the grinding of beveloid gears by the continuous generating grinding process. Furthermore, it can be extended to encompass the cases of swivel angle modification and hob shifting during hobbing.

Lines of Contact

The hob that generates a beveloid gear in a hobbing machining process can be considered an involute gear. Most commonly, such a gear is of the cylindrical type, although adoption of a conical involute hob would be possible too, in principle. For this reason, the kinematics of hobbing will be presented in this paper by referring to a beveloid hob, and subsequently specialized to the case of a cylindrical hob.

This section introduces the nomenclature adopted in the paper and summarizes known results pertaining to the loci of contact of a conical involute gear set. The reader is referred to References 16 and 17 for a detailed explanation of the reported concepts and formulas.

The tooth flanks of a conical involute gear are portions of involute helicoids. As opposed to classical helical involute gears, the two helicoids of the same tooth of a conical involute gear do not generally stem from the same base cylinder, nor have the same lead.

In a beveloid gear set composed of gears G_1 and G_2, the axis of gear G_1 is here directed in either way by unit vector $\boldsymbol{n}_1$. The orientation of the axis of gear G_2 by unit vector $\boldsymbol{n}_2$ is then so chosen as to make the right-hand flanks of gear G_1 come into contact with the right-hand flanks of gear G_2. With reference to Figure 1, the distinction between right-hand and left-hand tooth flanks is possible based on the sign of the ensuing quantity:

$$\boldsymbol{q}_i \times (F_i - O_i) \cdot \boldsymbol{n}_i \tag{1}$$

where F_i is a point on a tooth flank of gear G_i (i=1, 2), O_i is a point on the axis of gear G_i, and $\boldsymbol{q}_i$ is the outward-pointing unit vector perpendicular to the tooth flank at F_i. A tooth flank is a right-hand or left-hand flank, depending on whether Equation 1 results in a positive or, respectively, negative quantity. (In Figure 1, point F_i lies on a right-hand flank of gear G_i). In the sequel, index j will be systematically used to refer to right-hand (j = + 1) or left-hand (j = −1) tooth flanks.

The basic geometry of gear G_i (i=1, 2) is defined by its number of teeth N_i, the radii $\rho_{i,j}$ (j = ± 1) of its base cylinders, the base helix angles $\beta_{i,j}$ of its involute helicoids ($|\beta_{i,j}| < \pi/2$; $\beta_{i,j} > 0$ for right-handed helicoids), and the base angular thickness φ'_i of its teeth at a specified cross section. All these parameters—with the exception of N_i—are reported in Figure 1, which also shows the involute helicoids as stretching inwards up to their base cylinders, irrespective of their actual radial extent.

The normal base pitch is the distance between homologous involute helicoids of adjacent teeth of the same gear. A beveloid gear has two normal base pitches—$p_{i,1}$ and $p_{i,-1}$—one for right-hand flanks and one for left-hand flanks. Their expression is:

$$p_{i,j} = \frac{2\pi\rho_{i,j} u_{i,j}}{N_i} \qquad (i = 1,2;\ j = \pm 1) \tag{2}$$

where

$$u_{i,j} = \cos\beta_{i,j} \qquad (i = 1,2;\ j = \pm 1) \tag{3}$$

Two beveloid gears can mesh only if they have the same normal base pitches, i.e., only if the ensuing conditions are satisfied:

$$\begin{cases} p_{1,1} = p_{2,1} \\ p_{1,-1} = p_{2,-1} \end{cases} \tag{4}$$

Owing to Equation 4, the following equations and ensuing text and figures will refer to the normal base pitch of right-hand and left-hand flanks of both gears as p_1 and p_{-1} respectively.

The common perpendicular to the axes of a pair of meshing beveloid gears intersects the axes themselves at points A_1 and A_2 (Fig. 2). The relative position of these axes

continued

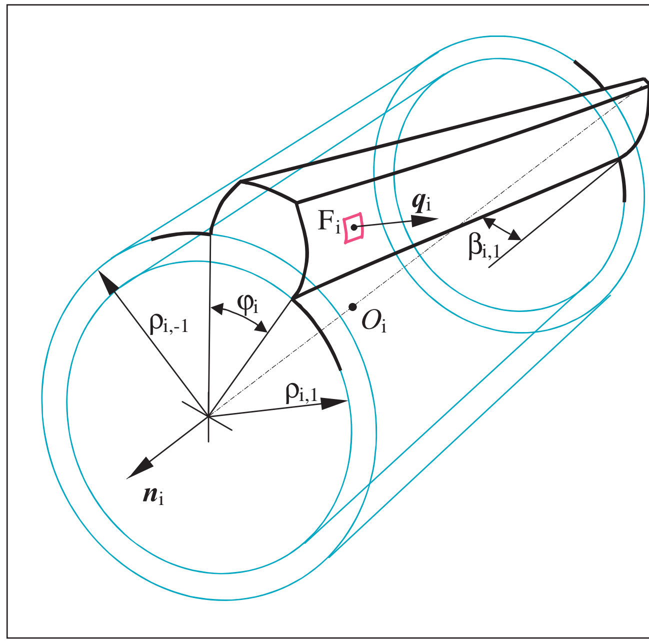

Figure 1—Basic dimensions of beveloid gear G_i.

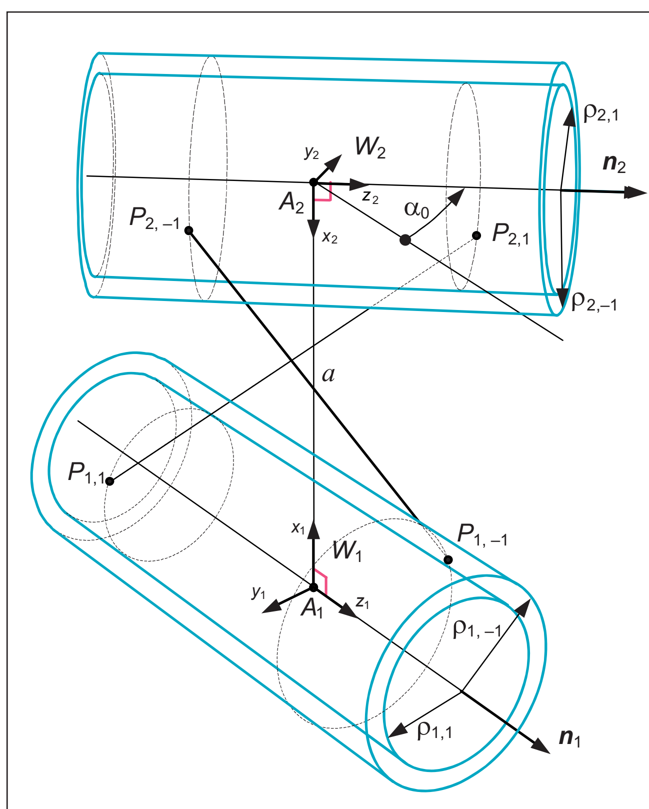

Figure 2—The lines of contact.

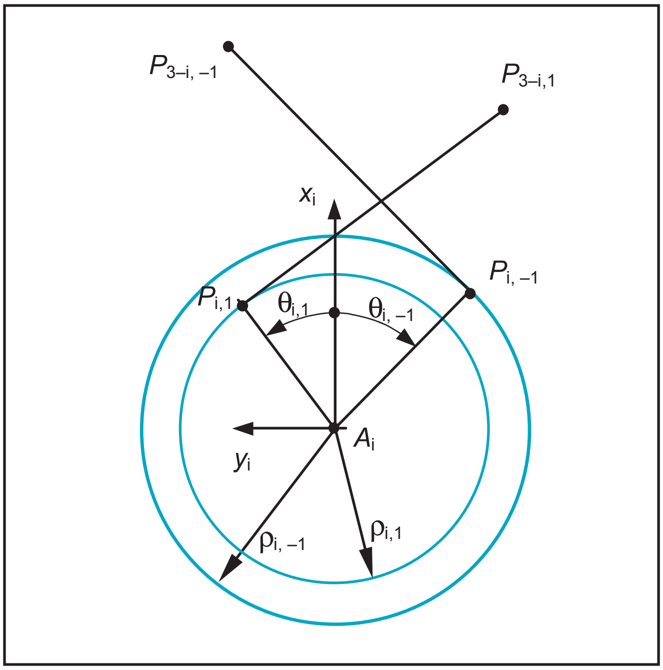

Figure 3—Coordinates of the extremities of the lines of contact.

is defined by their mutual distance a_0, together with their relative inclination α_0. Specifically, angle α_0 is the amplitude of the virtual rotation—positive if counterclockwise—about vector $(A_2–A_1)$ that would make unit vector $\boldsymbol{n}_1$ align with unit vector $\boldsymbol{n}_2$. Two fixed Cartesian reference frames W_i (i =1, 2) are then introduced with origins at points A_i, x_i axis pointing towards A_{3-i}, and z_i axis directed as unit vector $\boldsymbol{n}_i$.

For a conventional beveloid gear set composed of two meshing conical involute gears that revolve about their fixed and non-parallel axes, the locus of possible points of contact between the involute helicoids of right-hand (left-hand) flanks is a line segment that has a definite position with respect to either of reference frames W_i (i=1, 2). More specifically, the line of contact is tangent at its ending points $P_{1,1}$ and $P_{2,1}$ ($P_{1,-1}$ and $P_{2,-1}$) to the base cylinders of the right-hand (left-hand) flanks of the two gears. This is true even if the actual tooth flanks—being limited portions of the above-mentioned involute helicoids—touch each other along line segments that are shorter than the above-mentioned lines of contact and superimposed on them.

The cosine and sine of the angle $\theta_{i,j}$ that the projection of vector $(P_{i,j} –A_i)$ on the xy-plane of reference frame W_i forms with the x-axis of W_i (see Fig. 3) are indirectly provided by:

$$c_{i,j} = -\frac{v_{h,j} + u_0 v_{i,j}}{v_0 u_{i,j}} \qquad (i = 1,2;\ h = 3 - i;\ j = \pm 1) \tag{5}$$

$$s_{i,j} = \lambda_j \frac{\sqrt{Q_j}}{v_0 u_{i,j}} \qquad (i = 1,2;\ j = \pm 1) \tag{6}$$

where

$$c_{i,j} = \cos\theta_{i,j}; \qquad s_{i,j} = \sin\theta_{i,j} \qquad (i = 1,2;\ j = \pm 1) \tag{7}$$

$$v_{i,j} = \sin\beta_{i,j} \qquad (i = 1,2;\ j = \pm 1) \tag{8}$$

$$u_0 = \cos\alpha_0; \qquad v_0 = \sin\alpha_0 \tag{9}$$

$$\lambda_j = j\,\mathrm{sign}\left[v_0\left(a_0 - \rho_{1,j}c_{1,j} - \rho_{2,j}c_{2,j}\right)\right] \qquad (j = \pm 1) \tag{10}$$

$$Q_j = v_0^2 - v_{1,j}^2 - v_{2,j}^2 - 2u_0 v_{1,j} v_{2,j} \qquad (j = \pm 1) \tag{11}$$

The function sign(.) in Equation 10 returns the value + 1, 0, or –1 depending on whether its argument is positive, zero or negative. In Figure 3, angle $\theta_{i,1}$ is positive, whereas angle $\theta_{i,-1}$ is negative.

Due to the square root in Equation 6, two beveloid gears can properly mesh only if condition

$$Q_j \geq 0 \qquad (j = \pm 1) \tag{12}$$

is satisfied for both values of j. In the following equations, text and figures Equation 12 will be supposed as holding.

Thanks to Equations 5 and 6, angle $\theta_{i,j}$ can be expressed as:

$$\theta_{i,j} = 2\lambda_j \arctan\frac{v_0 u_{i,j} + u_0 v_{i,j} + v_{h,j}}{\sqrt{Q_j}} \tag{13}$$

(i=1,2; h= 3 – i; j= ± 1).

The z-coordinate $b_{i,j}$ of point $P_{i,j}$ in reference frame W_i (i=1,2; j= ± 1) is given by:

$b_{1,j}$=

$$-\lambda_j \frac{a_0 u_{1,j}\left(v_{1,j} + u_0 v_{2,j}\right) + \rho_{1,j} v_0\left(u_0 + v_{1,j} v_{2,j}\right) + \rho_{2,j} v_0 u_{1,j} u_{2,j}}{v_0 u_{1,j}\sqrt{Q_j}} \tag{14}$$

Based on the relative positions of reference frames W_1 and W_2, together with the cylindrical coordinates $\rho_{i,j}$, $\theta_{i,j}$, and $b_{i,j}$ of points $P_{i,j}$ with respect to W_i (i=1,2; j= ± 1), the length σ_j of the line of contact $P_{1,j}P_{2,j}$ can be determined by (Refs. 16–17):

$$\sigma_j = j\lambda_j v_0 \frac{a_0 - \rho_{1,j}c_{1,j} - \rho_{2,j}c_{2,j}}{\sqrt{Q_j}} \qquad (j = \pm 1) \tag{15}$$

The results summarized so far are directly applicable to a conventional beveloid gear set, i.e., to a pair of meshing beveloid gears that revolve about their rigidly connected axes. They also represent a convenient starting point for determining all possible relative positions of a beveloid hob relative to the beveloid gear being machined.

A Backlash-Free Beveloid Gear Set

Due to the single-point contact between tooth flanks of a beveloid gear set, the assortment of rigid-body positions of a beveloid hob relative to the beveloid gear being machined cannot be confined to the simple infinity of relative positions of two gears in a conventional beveloid gear set. Otherwise, a gear machined by a beveloid hob would not have involute helicoidal flanks; rather, only one curve on these flanks would belong to the desired involute helicoids.

Therefore the double infinity of points on a tooth flank of a beveloid gear machined by a beveloid hob has to be obtained at least as a two-parameter envelope of the positions of the hob tooth flanks. Each of these positions must correspond to a meshing configuration—with no backlash—of the beveloid hob with the finished beveloid gear.

There is a quadruple infinity of configurations of a beveloid gear tightly meshing with a beveloid hob (the contacts between right-hand flanks, as well as the contacts between left-hand flanks, each diminish by one the original six degrees of freedom that possess, in principle, a freely-movable hob that does not touch the gear). The double infinity of rigid-body positions of the hob relative to the generated gear is only a subset of the quadruple-infinity possible meshing configurations. This latter set of configurations can be found by first determining the simple infinity of all relative rigid-body positions of two beveloid gears tightly meshing in a conventional gear set, which is just the scope of the present section.

With regards to the same set of beveloid gears considered in the previous section, let φ_{0i} be the common-normal angular base thickness of a tooth of gear G_i (i=1, 2), i.e., the angular base thickness at the cross-section of gear G_i through point A_i (Figs. 1 and 2). On a generic cross-section identified by the axial coordinate z_i, the tooth angular base thickness φ_i' of a tooth of gear G_i is given by

$$\varphi_i' = \varphi_{0i} + \left(k_{i,-1} - k_{i,1}\right) z_i \qquad (i = 1,2) \tag{16}$$

In this equation, quantity $k_{i,j}$ is defined as

$$k_{i,j} = \frac{\tan\beta_{i,j}}{\rho_{i,j}} \qquad (i = 1,2;\ j = \pm 1) \tag{17}$$

Incidentally, $k_{i,j}$ can be given a geometric meaning: If the base helix of an involute helicoid j of gear G_i is projected on the unitary-radius cylinder coaxial with the gear, then the resulting projection is a helix whose inclination angle with respect to the gear axis is $\tan^{-1}k_{i,j}$.

If the surface of the unitary-radius cylinder of gear G_i is now cut along the generator that intersects the negative x-axis of reference frame W_i (Fig. 2), and subsequently flattened (Fig. 4), the former projections of the base helices of a tooth of gear G_i appear as straight lines. In Figure 4, coordinate δ_i—measured from the generator that intersects the positive x-axis of W_i—parameterizes the generators of the unitary-radius cylinder associated with gear G_i. The common-normal angular base tooth thickness φ_{0i} is also shown in Figure 4, together with the point H_i of intersection of the common normal A_1A_2 with the unitary-radius cylinder of gear G_i.

The orientations of gears G_1 and G_2 about their respective axes are defined here by considering an arbitrarily selected reference right-hand flank (involute helicoid) $\Sigma_{1,1}$ on gear G_1, together with the right-hand flank (involute helicoid) $\Sigma_{2,1}$ of the tooth of gear G_2 in contact with $\Sigma_{1,1}$. The reference angular position of gear G_i is chosen here as characterized by helicoid

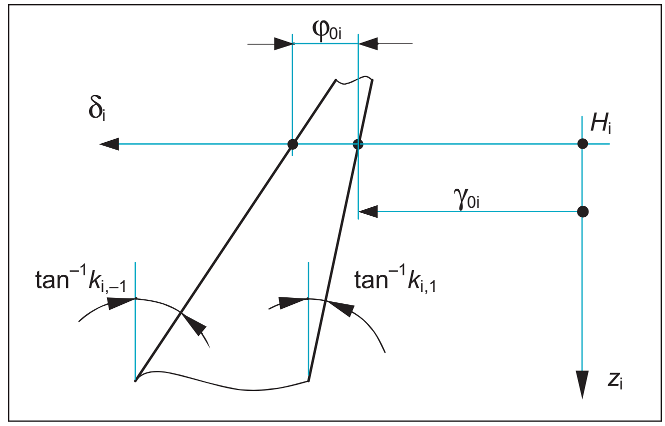

Figure 4—The developed unitary-radius cylinder of gear G_i.

$\Sigma_{i,1}$ intersecting the minimum distance segment A_1A_2 (Fig. 2) at a point of the base cylinder of gear G_i (i=1,2). (Equivalently, at the reference angular position of gear G_i the projection of the base helix of $\Sigma_{i,1}$ on the unitary-radius cylinder of the gear goes through point H_i.) A generic angular position of gear G_i is then identified by the angle γ_{0i} of the rotation that carries the gear from its reference angular position to the considered position. Angle γ_{0i}—positive for a counterclockwise rotation with respect to unit vector $\boldsymbol{n}_i$ (Fig. 2), can also be highlighted on the developed unitary-radius cylinder (Fig. 4).

The angular positions γ_{01} and γ_{02} of two beveloid gears that mesh together with zero backlash are clearly interrelated. An obvious mutual constraint is the differential condition that stems from expressing the gear ratio in terms of the number of teeth N_i (i=1,2) of the two gears

$$\frac{d\gamma_{02}}{d\gamma_{01}} = -\frac{N_1}{N_2} \tag{18}$$

In order to find a finite relation between γ_{01} and γ_{02}, two maneuvers are envisaged. The first maneuver starts with the first gear at position γ_{01}= 0 and—by exploiting the tooth contact between right-hand flanks only—carries the second gear at position γ_{02}= 0. The second maneuver is similar to the first one, but for the reliance on the contact between left-hand flanks.

The first step of the first maneuver consists of making the reference involute helicoid $\Sigma_{1,1}$ of gear G_1 go through the extremity $P_{1,1}$ of the line of contact between helicoids of right-hand flanks (Fig. 2). The corresponding rotation $\Delta\gamma_{01a}$ of gear G_1 is given by

$$\Delta\gamma_{01a} = \theta_{1,1} - k_{1,1} b_{1,1} \tag{19}$$

where $\theta_{1,1}$ and $b_{1,1}$ are provided by Equations 13 and 14, respectively. Equation 19 can be justified by elementary geometric reasoning on the unitary-radius cylinder of gear G_1. (The reader is referred to Reference 16 for further details.)

The second step makes helicoid $\Sigma_{1,1}$ go through point $P_{2,1}$. The necessary rotation of gear G_1 is

continued

$$\Delta\gamma_{01b} = -\frac{2\pi\sigma_1}{N_1\, p_1} \quad (20)$$

Now helicoids $\Sigma_{1,1}$ and $\Sigma_{2,1}$ touch each other at $P_{2,1}$. By the third—and last—step, gear G_2 is so turned about its axis as to make the base helix of helicoid $\Sigma_{2,1}$ intersect the common normal A_1A_2. The corresponding rotation of gear G_2 is given by

$$\Delta\gamma_{02c} = -\left(\theta_{2,1} - k_{2,1}b_{2,1}\right) \quad (21)$$

At the end of the considered three-step maneuver, gear G_2 is at its reference position ($\gamma_{02} = 0$), whereas gear G_1 is at a position identified by

$$\Delta\gamma'_{01} = \Delta\gamma_{01a} + \Delta\gamma_{01b} - \frac{N_2}{N_1}\Delta\gamma_{02c} \quad (22)$$

Owing to Equation 18, and to the existence of a meshing configuration characterized by $(\gamma_{01}, \gamma_{02}) = (\Delta\gamma'_{01}, 0)$, the ensuing relation between γ_{01} and γ_{02} must be satisfied

$$N_1\left(\gamma_{01} - \Delta\gamma'_{01}\right) + N_2\gamma_{02} = 0 \quad (23)$$

Now the second maneuver is taken into account. Its first step consists of bringing gear G_1 from the reference angular position to the position where helicoid $\Sigma_{1,-1}$ goes through point $P_{1,-1}$. Here $\Sigma_{1,-1}$ is the helicoid that, together with the reference helicoid $\Sigma_{1,1}$ defined above, bounds the same tooth of gear G_1. The corresponding rotation of gear G_1 is (see Figs. 2–4):

$$\Delta\gamma_{01d} = -\varphi_{01} + \theta_{1,-1} - k_{1,-1}b_{1,-1} \quad (24)$$

By the second step, gear G_1 is revolved until helicoid $\Sigma_{1,-1}$ goes through point $P_{2,-1}$. The incremental rotation of gear G_1 is provided by

$$\Delta\gamma_{01e} = \frac{2\pi\sigma_{-1}}{N_1\, p_{-1}} \quad (25)$$

After this step, the helicoid $\Sigma_{1,-1}$ of gear G_1 touches the helicoid $\Sigma_{2,-1}$ of gear G_2 at point $P_{2,-1}$. It is worth observing that, while $\Sigma_{1,1}$ and $\Sigma_{1,-1}$ bound the same tooth of gear G_1, $\Sigma_{2,1}$ and $\Sigma_{2,-1}$ delimit the same tooth space of gear G_2.

The third step of the current maneuver consists in making helicoid $\Sigma_{2,-1}$ intersect the common normal $A_1 A_2$ at a point of its base helix. The additional rotation of gear G_2 is provided by

$$\Delta\gamma_{02f} = -\left(\theta_{2,-1} - k_{2,-1}b_{2,-1}\right) \quad (26)$$

The fourth—and last—step brings gear G_2 at the reference position $\gamma_{02} = 0$, i.e., makes helicoid $\Sigma_{2,1}$ intersect the common normal A_1A_2 at a point of the base helix. The corresponding further rotation of gear G_2 is

$$\Delta\gamma_{02g} = -\frac{2\pi}{N_2} + \varphi_{02} \quad (27)$$

The angular position of gear G_1 at the end of the whole maneuver is provided by

$$\Delta\gamma''_{01} = \Delta\gamma_{01d} + \Delta\gamma_{01e} - \frac{N_2}{N_1}\left(\Delta\gamma_{02f} + \Delta\gamma_{02g}\right) \quad (28)$$

Therefore in addition to Equation 23, another constraint between γ_{01} and γ_{02} can be found

$$N_1\left(\gamma_{01} - \Delta\gamma''_{01}\right) + N_2\gamma_{02} = 0 \quad (29)$$

By considering the expressions of $\Delta\gamma'_{01}$and $\Delta\gamma''_{01}$ provided by Equations 22 and 28, Equations 23 and 29 can be rewritten as

$$N_1\gamma_{01} + N_2\gamma_{02} + B' = 0 \quad (30)$$

$$N_1\gamma_{01} + N_2\gamma_{02} + B'' = 0 \quad (31)$$

Quantities B′ and B" that appear in these equations are given by

$$B' = N_1\left(k_{1,1}b_{1,1} - \theta_{1,1}\right) + N_2\left(k_{2,1}b_{2,1} - \theta_{2,1}\right) + \frac{2\pi\sigma_1}{p_1} \quad (32)$$

$$\begin{aligned} B'' = {} & N_1\left(k_{1,-1}b_{1,-1} - \theta_{1,-1} + \varphi_{01}\right) + \\ & N_2\left(k_{2,-1}b_{2,-1} - \theta_{2,-1} + \varphi_{02}\right) - 2\pi\left(\frac{\sigma_{-1}}{p_{-1}} + 1\right) \end{aligned} \quad (33)$$

For a given backlash-free beveloid gear set, Equations 30 and 31 must be satisfied simultaneously. Since the considered gear set is a mechanism with one degree of freedom, it should be possible to arbitrarily select γ_{01} (or γ_{02}), and then determine γ_{02} (or γ_{01}). Therefore Equations 30 and 31, when considered as linear equations in γ_{01} and γ_{02}, should be linearly dependent. This requirement translates into the ensuing condition:

$$B' = B'' \quad (34)$$

By taking into account Equations 32 and 33, Equation 34 can be rewritten as follows (Ref. 16):

$$S_1 + S_2 + T = 0 \quad (35)$$

where

$$S_i = N_i\left(k_{i,1}b_{i,1} - k_{i,-1}b_{i,-1} - \theta_{i,1} + \theta_{i,-1} - \varphi_{0i}\right) \qquad (i = 1,2) \quad (36)$$

and

$$T = 2\pi\left(\frac{\sigma_1}{p_1} + \frac{\sigma_{-1}}{p_{-1}} + 1\right) \quad (37)$$

Equivalent to the equation set composed of Equations 30 and 31, the equation set formed by Equations 30 and 35 encapsulates the meshing condition—with no backlash—of a pair of conical involute gears, each bound to revolve about its own axis. More specifically, Equation 35 involves the geometry of the two beveloid gears, together with the relative placement of the two gear axes and the axial placement of each gear on its own axis. It had already been presented, though in a slightly different form, in Reference 16. On the other hand, by also encompassing the angular position of the two gears, Equation 30 provides information about their phasing. To this author's knowledge, no such equation has ever been published before.

Unconstrained Beveloid Gears in Tight Mesh

As anticipated at the beginning of the previous section, the whole collection of relative rigid-body positions of two tightly meshing beveloid gears can be determined by generalizing the results just found for a conventional beveloid gear set.

Let us consider again a beveloid gear set, composed of two meshing beveloid gears connected to a rigid frame through revolute pairs. If the distance a_0 and angle α_0 between the revolute pair axes, together with the geometry of the two gears—notably their common-normal base angular thicknesses φ_{01} and φ_{02}—comply with Equation 35, then Equation 30 is satisfied by a simple infinity of values for the ordered pair (γ_{01}, γ_{02}), i.e., the mechanism has a simple infinity of configurations.

Now the two mentioned revolute pairs are replaced by cylindrical pairs, which implies that both gears can be displaced along their axes, in addition to being revolved about them. Consequently, the common-normal base angular thickness φ_{0i} of gear G_i, i.e., the base angular thickness on a cross section of gear G_i going through point A_i (Fig. 2), becomes linearly dependent on the axial placement of the gear. With the aid of Figure 5, the ensuing condition can be laid down (See also Eq. 16)

$$\varphi_{0i} = \varphi_i + \left(k_{i,-1} - k_{i,1}\right)\zeta_i \qquad (i = 1,2) \tag{38}$$

In this equation, φ_i is the reference base angular thickness of gear G_i, i.e., the base angular thickness of a tooth of gear G_i at a reference cross section fixed to the gear. Moreover, ζ_i is the displacement of the common-normal cross section, measured from the reference cross section, positive if concordant with the direction of the z-axis of reference frame W_i (a positive ζ_i is shown in Figure 5).

Quantity γ_{0i} is no longer suited to parameterize the angular position of gear G_i. For instance, if $\gamma_{0i} = 0$ and gear G_i is axially displaced, then the base helix of the reference helicoid $\Sigma_{i,1}$ keeps intersecting the minimum distance segment A_1A_2, which means that the gear undergoes a screw motion with respect to the rigid gear-set frame, thus changing its orientation.

Henceforth the angular position of gear G_i will be parameterized by angle γ_i, which is an angle measured on the reference cross section of the gear. Precisely, γ_i is the angle between two lines belonging to the reference cross section of gear G_i—the projection of the minimum distance segment

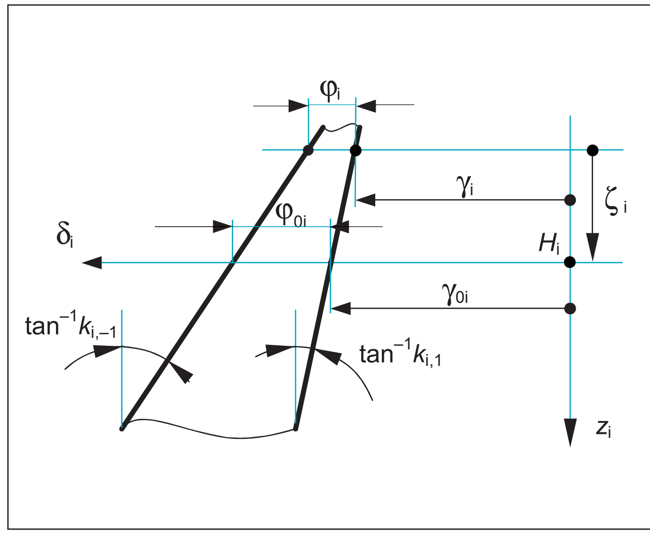

Figure 5—Reference and common-normal parameters.

A_1A_2, and the radial line through the point on the base helix of reference flank $\Sigma_{i,1}$. As shown in Figure 5, the following relation exists between γ_{0i} and γ_i

$$\gamma_{0i} = \gamma_i + k_{i,1}\zeta_i \qquad (i = 1,2) \tag{39}$$

By taking into account Equations 38 and 39, Equations 35 and 30 are transformed into a set of two equations in the four unknowns ζ_1, ζ_2, γ_1 and γ_2. It is generally possible to arbitrarily choose either of γ_i (i=1,2) and either of ζ_i (i=1,2), and then determine the remaining two unknowns. Therefore a set of two beveloid gears connected to the frame by cylindrical pairs has two degrees of freedom, with no need to satisfy any prerequisite similar to Equation 35.

At this point, the frame of the gear set is suppressed altogether, so that parameters a_0 and α_0 are no longer bound to be constant. We are now left with two beveloid gears that can be freely moved in space, provided that they keep meshing with no backlash. All relative rigid-body positions of the two gears are those satisfying Equations 35 and 30, which can be rewritten in the ensuing concise form

$$\begin{cases} U\left(a_0, \alpha_0, \zeta_1, \zeta_2\right) = 0 \\ V\left(a_0, \alpha_0, \zeta_1, \zeta_2, \gamma_1, \gamma_2\right) = 0 \end{cases} \tag{40}$$

In Equation 40, U and V are, respectively, what the left-hand sides of Equations 35 and 30 turn into, once φ_{0i} and γ_{0i} (i=1,2) have been replaced with the expressions provided by Equations 38 and 39.

Equation 40 is a set of two conditions in six unknowns, namely, a_0, α_0, ζ_1, ζ_2, γ_1 and γ_2. Therefore, Equation 40 can be solved in a quadruple infinity of ways, which means that there is a quadruple infinity of possible relative placements for the two beveloid gears.

As explained hereafter, each of these relative placements can be determined by relying on the values of a_0, α_0, ζ_1, ζ_2, γ_1

continued

and γ_2. At first, a skeleton is built based on a_0 and α_0. Such a skeleton is formed by the axes of the two gears, together with the common normal segment A_1A_2 (Fig. 2). Subsequently gear G_i (i= 1,2) is axially placed on its axis by relying on the value of ζ_1. Finally, the orientation of gear G_i about its axis—and with respect to the skeleton segment A_1A_2—is provided by angle γ_i.

Equation 40 will prove pivotal in computing the hobbing parameters of a beveloid gear.

Gear-Hob Relative Movements

As already mentioned at the beginning of the section addressing a backlash-free beveloid gear set, the tooth flanks of a beveloid gear result from a two-parameter envelope by the hob thread flanks. The double-infinite subset of the quadruple infinity of possible gear-hob relative placements is chosen here on the analogy of the hobbing operation of a cylindrical gear by a cylindrical hob. Throughout this section, the generated gear and the enveloping hob will be referred to as gear G_1 and G_2 respectively.

As is known, a cylindrical gear can be hobbed by keeping constant the work-hob axis angle α_0, by revolving the hob about its axis, and by simultaneously moving such an axis across the gear width. In case no hob shift takes place during hobbing—as usually happens—parameter ζ_2 is kept constant too. If the gear and the hob are both cylindrical, it is easy to prove that function U in Equation 40 is deprived of arguments ζ_1 and ζ_2. Therefore, the constancy of the shaft angle α_0 implies the constancy of the axis distance a_0, too; parameters γ_1 and ζ_1 can be thought of as the two parameters of the enveloping process. And for any choice of their values, the second condition in Equation 40 yields quantity γ_2.

Now the hobbing of a beveloid gear by a beveloid hob is analyzed. Similar to the previous case, the shaft angle α_0 is supposed as constant. Its value might be chosen, for instance, with the aim of minimizing the shaft axis distance a_0 at a given cross-section of the gear (see, for instance, Reference 17 for application of this criterion to the hobbing of cylindrical gears). The independent parameters of the envelope are again γ_1 and ζ_1, whereas parameter ζ_2 is kept constant. For any choice of γ_1 and ζ_1, the first and second conditions in Equation 40 yield, respectively, the values of a_0 and γ_2.

The instantaneous movement of the hob relative to the gear being machined can be thought of as the superimposition of two movements:

1. The relative movement of hob and gear as they revolve about their axes (only the independent envelope parameter γ_1 varies)
2. The relative movement of hob and gear when the gear is shifted along its axis without turning with respect to the hobbing machine (only the independent envelope parameter ζ_1 varies)

Since the former movement can be found straightforwardly via Equation 18, only determination of the latter will be pursued here. More specifically, the ensuing ratios of differentials are of interest

$$f_a = \frac{da_0}{d\zeta_1}; \qquad f_\gamma = \frac{d\gamma_2}{d\zeta_1} \tag{41}$$

The differentials on the right-hand sides of Equation 41 are computed with these assumptions

$$\begin{cases} d\alpha_0 = 0 \\ d\zeta_2 = 0 \\ d\gamma_1 = 0 \end{cases} \tag{42}$$

Ratio f_a is the rate of change of the gear-hob axis distance as the hob is moved along the gear width; it is zero for cylindrical gears, but not so for beveloid gears. Ratio f_γ, on the other hand, provides information about the hob rotation as the hob is fed across the work. It would be zero for a spur gear, but is different from zero for helical gears and for most beveloid gears.

To compute ratios f_a and f_γ, Equation 40 is now differentiated by taking into account Equation 42

$$\begin{bmatrix} \frac{\partial U}{\partial a_0} & \frac{\partial U}{\partial \zeta_1} & 0 \\ \frac{\partial V}{\partial a_0} & \frac{\partial V}{\partial \zeta_1} & \frac{\partial V}{\partial \gamma_2} \end{bmatrix} \begin{bmatrix} da_0 \\ d\zeta_1 \\ d\gamma_2 \end{bmatrix} = 0 \tag{43}$$

Based on Equation 43, the ensuing expression for quantities f_a and f_γ can be obtained

$$f_a = -\frac{\partial U}{\partial \zeta_1}\left(\frac{\partial U}{\partial a_0}\right)^{-1} \tag{44}$$

$$f_\gamma = \left(\frac{\partial U}{\partial \zeta_1}\frac{\partial V}{\partial a_0} - \frac{\partial U}{\partial a_0}\frac{\partial V}{\partial \zeta_1}\right)\left(\frac{\partial U}{\partial a_0}\frac{\partial V}{\partial \gamma_2}\right)^{-1} \tag{45}$$

The partial derivatives in Equations 44 and 45 can be easily computed as shown hereafter. The partial derivative of U with respect to a_0 is given by the ensuing concatenation of relations

$$\frac{\partial U}{\partial a_0} = \frac{\partial S_1}{\partial a_0} + \frac{\partial S_2}{\partial a_0} + \frac{\partial T}{\partial a_0} \tag{46}$$

$$\frac{\partial S_i}{\partial a_0} = N_i\left(k_{i,1}\frac{\partial b_{i,1}}{\partial a_0} - k_{i,-1}\frac{\partial b_{i,-1}}{\partial a_0}\right) \qquad (i = 1,2) \tag{47}$$

$$\frac{\partial T}{\partial a_0} = 2\pi\left(\frac{1}{p_1}\frac{\partial \sigma_1}{\partial a_0} + \frac{1}{p_{-1}}\frac{\partial \sigma_{-1}}{\partial a_0}\right) \tag{48}$$

$$\frac{\partial b_{i,j}}{\partial a_0} = -\lambda_j \frac{v_{i,j} + u_0 v_{h,j}}{v_0\sqrt{Q_j}} \qquad (i = 1,2;\ h = 3-i;\ j = \pm 1) \tag{49}$$

$$\frac{\partial \sigma_j}{\partial a_0} = \lambda_j \frac{v_0}{\sqrt{Q_j}} \qquad (j = \pm 1) \tag{50}$$

As for the partial derivative of U with respect to ζ_1, it is provided by

$$\frac{\partial U}{\partial \zeta_{s1}} = \frac{\partial S_1}{\partial \zeta_{s1}} + \frac{\partial S_2}{\partial \zeta_{s1}} + \frac{\partial T}{\partial \zeta_1} \tag{51}$$

$$\frac{\partial S_1}{\partial \zeta_{s1}} = -N_1 \frac{d\varphi_{01}}{d\zeta_1} = N_1 \left(k_{1,1} - k_{1,-1}\right) \tag{52}$$

$$\frac{\partial S_2}{\partial \zeta_{s1}} = \frac{\partial T}{\partial \zeta_1} = 0 \tag{53}$$

The partial derivative of V with respect to a_0 is given by

$$\frac{\partial V}{\partial a_0} = N_1 k_{1,1} \frac{\partial b_{1,1}}{\partial a_0} + N_2 k_{2,1} \frac{\partial b_{2,1}}{\partial a_0} + \frac{2\pi}{p_1} \frac{\partial \sigma_1}{\partial a_0} \tag{54}$$

The derivatives on the right-hand side of this equation are in turn provided by Equations 49 and 50.

Finally, the partial derivative of V with respect to ζ_1 and γ_2 can be easily determined based on Equations 30 and 39:

$$\frac{\partial V}{\partial \zeta_1} = N_1 k_{1,1} \tag{55}$$

$$\frac{\partial V}{\partial \gamma_2} = N_2 \tag{56}$$

Thanks to Equations 46–55, the ratios f_a and f_γ can be rewritten in the ensuing explicit form:

$$f_a = N_1 v_0 \frac{k_{1,-1} - k_{1,1}}{D_1 - D_{-1}} \tag{57}$$

$$f_\gamma = \frac{N_1}{N_2} \frac{k_{1,1} D_{-1} - k_{1,-1} D_1}{D_1 - D_{-1}} \tag{58}$$

Quantities D_j ($j = \pm 1$) in Equations 57 and 58 are defined by:

$$D_j = \frac{\lambda_j \sqrt{Q_j}}{p_j} \qquad (j = \pm 1) \tag{59}$$

Equations 57–59 show that f_a and f_γ, do not depend on ζ_1, but only on the geometry of gear and hob, together with the swivel angle α_0 (which has been supposed as constant). The constancy of f_a, in particular, explains why the root surface of a beveloid gear is conical—at least if the gear is cut by keeping constant the angle α_0.

Equation 59 provides the expression of D_j ($j = \pm 1$) for any pair of beveloid gears. When gear G_2 is a hob, the number of teeth N_2 is small and the absolute values of the base helix angles $\beta_{2,1}$ and $\beta_{2,-1}$ are relatively large. Therefore the ensuing inequality is generally satisfied

$$a_0 - \rho_{1,j} c_{1,j} - \rho_{2,j} c_{2,j} > 0 \qquad (j = \pm 1) \tag{60}$$

Consequently, Equation 10 reduces to

$$\lambda_j = j\,\mathrm{sign}(v_0) \qquad (j = \pm 1) \tag{61}$$

In addition, since the most commonly used hobs can be considered as involute cylindrical gears rather than beveloid gears, the ensuing additional conditions come into play

$$\begin{cases} p_1 = p_{-1} \\ k_{2,1} = k_{2,-1} \\ v_{2,1} = v_{2,-1} \end{cases} \tag{62}$$

The values of ratios f_a and f_γ, are always needed when programming a CNC hobbing machine that has to cut a beveloid gear. Thanks to Equations 57 and 58, these ratios can be straightforwardly assessed. Therefore, from a utilitarian standpoint, Equations 57 and 58 are the main result of this paper.

Should angle α_0 change while ζ_1 varies (for instance, to constantly minimize the shaft distance a_0), and/or hob shifting occur during machining (for instance, to reduce the scalloping of the tooth flanks of the gear), then the mentioned equations would no longer be applicable. In this occurrence, Equation 40—the true theoretical contribution of this paper—should be resorted to again, and applied afresh to the case at hand.

Numerical Example

The formulae derived in the previous section are here applied to determine quantities f_a and f_γ for the hobbing of a beveloid gear (gear G_1) by a cylindrical hob (gear G_2).

Throughout this section, non-integer quantities are expressed by a high number of digits—all meaningful—in order to allow the reader to accurately check the reported result.

The hob has one thread ($N_2 = 1$) and is characterized by a module of 2 mm and a pressure angle of 20°. Based on this data, the ensuing dimensions can be easily computed

$$\rho_{2,1} = \rho_{2,-1} = 2.735229431127 \text{ mm}$$

$$\beta_{2,1} = \beta_{2,-1} = 69.90659103379 \text{ deg}$$

$$\varphi_2 = 1220.353746100 \text{ deg}$$

The normal base pitches of hob and gear can be derived from the geometry of the hob

$$p_1 = p_{-1} = 5.904262868187 \text{ mm}$$

The gear is characterized by

$$N_1 = 14$$

$$\rho_{1,1} = 13.16838183156 \text{ mm}$$

$$\rho_{1,-1} = 13.15931278723 \text{ mm}$$

$$\beta_{1,1} = -2.515092347823 \text{ deg}$$

$$\beta_{1,-1} = -1.343231785965 \text{ deg}$$

On a given (reference) cross section of the gear, the angular base thickness of the gear teeth is:

$$\varphi_1 = 16.94800000000 \text{ deg}$$

The angle α_0 between the axes of gear and hob is chosen in such a way as to minimize the shaft distance a_0 when point A_1 (Fig. 2) lies on the mentioned reference cross section of the gear. The corresponding values of α_0 and a_0 are:

$$\alpha_0 = -86.03648170877 \text{ deg}$$

$$a_0 = 43.96806431329 \text{ mm}$$

The ratio f_a between the change of axis distance a_0 and the displacement ζ_1 of point A_1 (Figs. 2 and 5) along the gear axis is provided by Equation 57

$$f_a = 0.02988732132842 \text{ mm/mm}$$

On the other hand, the ratio f_γ between the rotation angle of the hob and the displacement ζ_1 of point A_1 along the gear axis—when the gear does not rotate with respect to the hobbing machine—is provided by Equation 58

$$f_\gamma = 2.050765894724 \text{ deg/mm}$$

(To obtain this value, a conversion of unit of measurement has been necessary, since the ratio f_γ yielded by Equation 58 is expressed in radians per unit of length.)

Conclusions

With reference to the generation of a beveloid gear by a hobbing machine, the paper has presented a simple and general method to determine the rate of change of the hob-work axis distance and the differential rotation of the hob as the hob itself is fed across the work. Because it relies on a very few intrinsic dimensions of a beveloid gear, the method is conducive to concise expressions for the desired quantities.

The results presented here refer primarily to the hobbing of a beveloid gear by a beveloid hob, provided that the swivel angle remains constant. On the one hand, they can be readily specialized to the case of a cylindrical hob cutting a beveloid gear. On the other hand, it is easy to extend them to make provision for hob shifting and swivel angle change during hobbing.

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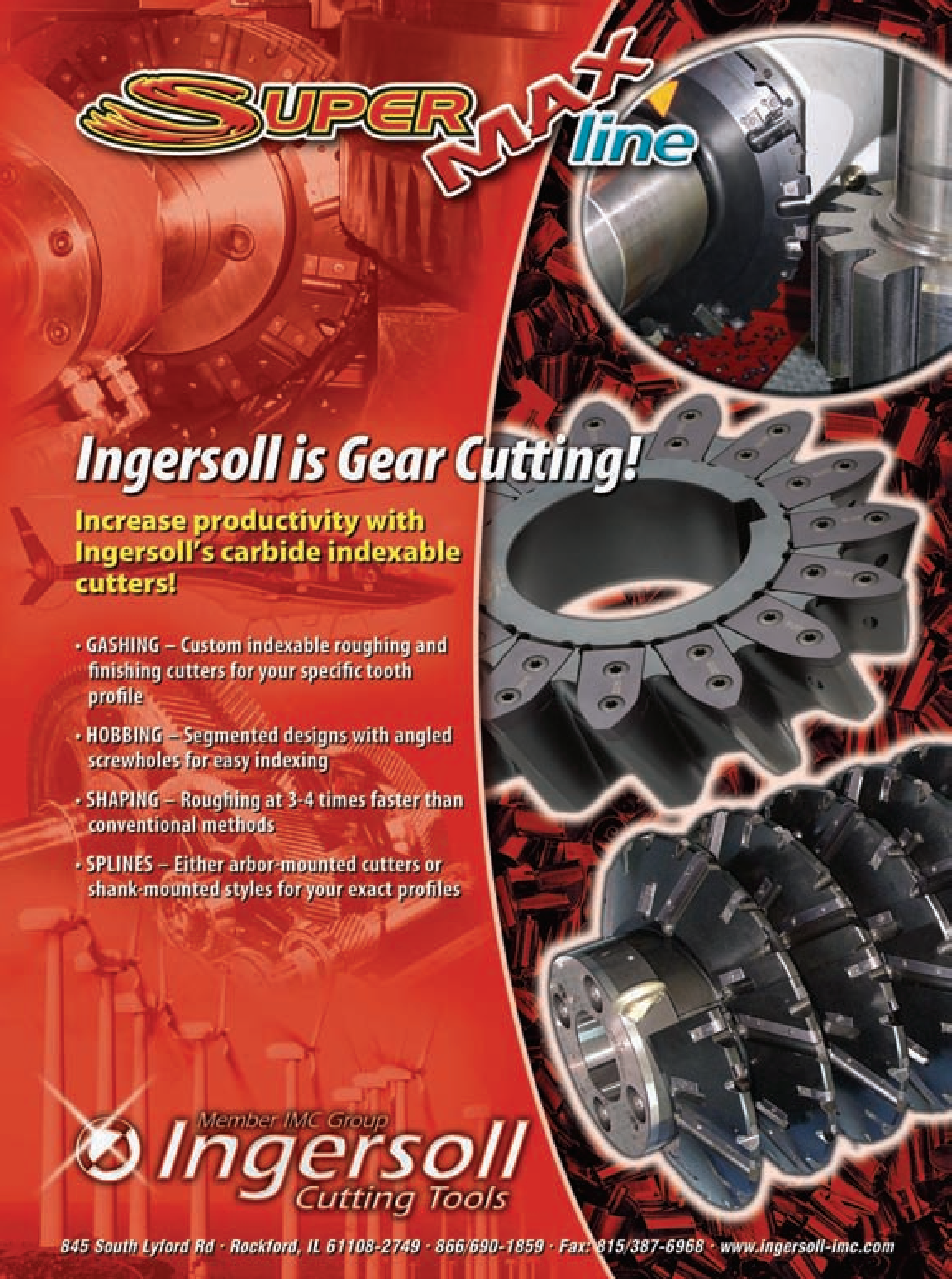
Carlo Innocenti *is a professor of Mechanics of Machines at the University of Modena and Reggio Emilia, Italy. After obtaining a degree in mechanical engineering in 1984, he worked for two different companies—as a computation engineer and as a mechanical designer, respectively. He then joined academia in 1990. The papers that he has authored or co-authored are mainly focused on the kinematic analysis and synthesis of planar and spatial mechanisms; rotordynamics; vehicle dynamics; robot calibration; gears and gearing systems.*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Vacuum Carburizing

TAKES CENTER STAGE AT MICHIGAN SYMPOSIUM

ALD-Holcroft Vacuum Technologies Co. will host a two-day technical symposium at the Henry Ford Museum in Dearborn, Michigan September 23–24, 2008. Expected attendees include business executives, design engineers, manufacturing engineers, drivetrain engineers, metallurgists and current and prospective users of vacuum carburizing processes and equipment.

"Although the symposium is a learning experience, we've found that equally important are the socializing opportunities in which far more detailed experiences are shared and informational networking opportunities are developed," says William Gornicki, vice president of sales and marketing at ALD-Holcroft. "It's always interesting to see how lessons learned in one particular industry can help those in another and vice versa."

The symposium will bring together individuals from a variety of organizations to speak on the trends and recent advancements in low pressure carburizing (LPC). Topics covered in the symposium include the economics of converting from atmosphere processing to vacuum processing, case studies providing actual process experiences and expectations, new standards in workload fixtures and the advantages of LPC in the design process.

"Over this past decade, we've seen the development of several different methods for applying this technology," Gornicki says. "It's only now that the weaker methods are falling to the wayside and the more viable methods are rising to the top."

Technical papers will be presented at the symposium by industry experts from Bodycote, Dana, General Motors, Praxair, Timken, the Herring Group, Vac-u-Heat, SGL Carbon and ALD-Holcroft Vacuum Technologies. All papers are related to the practice of LPC, and the speakers will be available for question-and-answer sessions as well as open discussions.

While most technical conferences are relegated to hotel banquet centers, the vacuum symposium will take place in Lovett Hall—a renowned 1930s ballroom within the Henry Ford Museum complex.

"We felt it was time to raise the bar and make the event a little more fun," Gornicki says. "Given our proximity to the Henry Ford Museum, and the irony

of Ford's mainstay (the automotive industry) and our leading industry (gear carburizing), we felt the match was a natural."

The complimentary dinner provided on the first evening will take place on the grounds of Greenfield Village, a 240-acre facility that features nearly 100 historic buildings from colonial America. The supper will be served at the Eagle Tavern as guests are treated to true 1851 cuisine and hospitality by the innkeeper and his family.

"This two-day symposium is shaping up to be the technical exchange of the year regarding LPC," Gornicki says. "We've lined up a dynamic list of topics and an all-star roster of presenters, and we're excited to bring this learning and networking opportunity to the market."

The cost for the symposium is $295 and includes lunch each day as well as the 1851-style dinner. Pre-enrollment is necessary by September 8, 2008. For more information regarding the symposium, visit *www.ald-holcroft.com* or call (248) 668-4004.

July 21–23—Powder Metallurgy Basic Short Course. Penn Stater Conference Center Hotel, State College, Pennsylvania. The three-day course sponsored by the Metal Powder Industries Federation (MPIF) is designed for engineers, tool and product designers, metallurgists, supervisors, technicians, QC personnel and managers, although a technical background is not necessary. The short course is appropriate for PM parts users or aspiring ones, specialists hoping to broaden their background and newcomers looking for an introduction to the field. Attendees will learn about the history of PM, the current state of the industry and what's in store for the future. They will also learn why PM parts are popular, how they are produced, different designs, the role of sintering in developing functional properties, injection molding, special tests and MPIF standards. The program includes a tour of The Pennsylvania State University's Center for Innovative Sintered Products. Course fees, based on MPIF/APMI membership, include various literature, a certificate of completion and breakfast and lunch each day. For more information, visit *www.mpif.org*.

September 9–13—AMB 2008. Trade Centre, Stuttgart, Germany. This international exhibition for the metalworking industry is supported by the German Engineering Federation and the Association of German Machine Tool Manufacturers. The exhibition is located in the heart of Baden-Württemberg, which is known as the leading high-tech region in Europe. This is the first year the event is being held at this newly built trade center, which is much larger and located next to the Stuttgart airport. Various industry users, exhibitors and visitors are native to this area, including 50 percent of the largest European machine tool manufacturers and over 40 percent of the European metalworking industry. They will be attending along with specialists from both small- and medium-sized ventures and industry newcomers using AMB as a marketing platform. Eight months before the exhibition 1,000 companies have registered already. This year the event is organized according to exhibitors' themes, so the fairgrounds may be twice as large, but visitors should be able to find what they're looking for. For more information, visit *www.messe-stuttgart.de/amb*.

September 15–16—Gear Failure Analysis Seminar. Big Sky Resort, Big Sky, Montana. The American Gear Manufacturers Association presents this 17th annual Gear Failure Analysis Seminar designed to help manufacturers avoid gear failure by knowing what causes to be aware of and how to fix problems. This technical education seminar looks at specific types of gear failure like macropitting, micropitting, scuffing, tooth wear and breakage, and methods to avoid these issues. Expert Robert Errichello, of the GEARTECH consulting firm, will be leading the seminar using lectures, slide presentations, hands-on workshops and Q&A sessions to educate attendees about the reasons for gear failure. Participants will also receive Errichello's *Failure Analysis Textbook and Gear Failure Analysis Atlas*. For more information, visit *www.agma.org*.

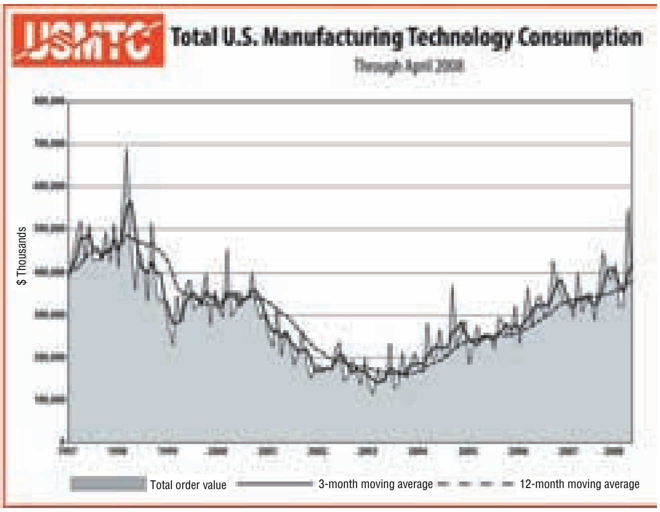
September 16–18—Basic Gear Noise Short Course. Gear Dynamics and Gear Noise Research Laboratory, The Ohio State University, Columbus, Ohio. For more than 29 years the Basic Gear Noise Short Course has been offered by The Ohio State University's GearLab as a tool for gear designers and noise specialists challenged by gear noise and transmission design problems. The course shows participants how to design gears so that major excitations of gear noise, such as transmission error, dynamic friction forces and shuttling forces, are reduced. The course teaches the basics of gear noise generation and measurement, in addition to gear rattle, transmission dynamics and housing acoustics. Instructors will demonstrate specialized gear analysis software and several Ohio State gear test rigs. Attendees are invited to share specific gear and transmission noise concerns in an interactive workshop session. For more information, contact Jonny Harianto, OSU Department of Mechanical Engineering, 201 West 19th Avenue, Columbus, OH 43210; phone (614) 688-3952, fax (614) 292-3163, *harianto.1@osu.edu*, or visit *www.gearlab.org*.

September 17–19—Gear Manufacturing Technology Course. Gear Manufacturing Inc., Anaheim, CA. 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.

September 22–23—Fastener Fair. SYMA Event Centre, Budapest, Hungary. The Fastener Fair represents every aspect of the fastener and fixing market by bringing together manufacturers, machine suppliers, wholesalers, distributors, importers, exporters and tool suppliers. The fair aims to be an environment for new ideas and partnerships, so professionals can come away with new business. The event also intends to highlight Europe's near East as an appealing market for manufacturing and assembly because of low-cost labor and economies and new members of the EU. The primary objective of the exhibition is to be cost- and time-efficient, meaning all stands will be closely monitored in size. The Fastener Fair is marketed to large end users in order to gauge both ends of the market, from end user to distributor. For more information, visit *http://www.fastenerfair.com/page2153/budapest-2008.aspx*.

Manufacturing Technology Consumption

UP NEARLY 20 PERCENT



U.S. manufacturing technology consumption (USMTC) amounted to $396.47 million in April, according to the Association for Manufacturing Technology (AMT) and the American Machine Tool Distributors' Association (AMTDA). This total is down 27.6 percent from March, but this April's total is up 29.2 percent from the previous year—the total for April 2007 was only $306.86 million. So far to date, the 2008 USMTC totals $1,587.27 million, which is an increase of 19.9 percent in comparison to 2007.

"The demand for manufacturing equipment continues to be healthy, with year-to-year growth of nearly 30 percent since April 2007," says John B. Byrd III, AMT President. "Export demand for U.S. manufactured products and the global boom in infrastructure development continues to fuel the surprising growth in capital equipment investment. The decline from March to April is not surprising, considering the extraordinary results posted in March."

The AMT and AMTDA are two trade associations that characterize the production and distribution of manufacturing technology, and they supply regional and national data for U.S. consumption of domestic and imported machine tools and related equipment. The USMTC report also breaks down the data into five U.S. geographic regions.

The Northeast region's manufacturing technology consumption in April was down 35.2 percent from March's $71.70 million, at $46.47 million. This amount is 10.1 percent less than April's total in 2007. The year-to-date total is up 4.3 percent from 2007.

The Southern region's April manufacturing technology consumption totaled $49.37 million, which is a decrease of 59.8 percent from the previous month but 26.8 percent higher than the 2007 total. The year-to-date total is $271.61 million, 69.6 percent higher than it was in 2007.

With a 10.4 percent decrease from March, the Midwestern region's manufacturing technology consumption stood at $154.14 million, but this is a 81.5 percent increase from April 2007. The year-to-date amount, $534.69 million, is 46.0 percent more than the 2007 figure.

In the Central region, April manufacturing technology consumption stood at $79.88 million, a decline from the March total by 31.7 percent and 6.1 percent from April 2007. The year-to-date total of $351.00 million is down 5.8 percent from 2007.

The Western region experienced a rise in manufacturing technology consumption by 3.4 percent from March, at $66.61 million. The data is 44.1 percent higher than the amount was in April 2007. The year-to-date total is $208.26 million, down 1.9 percent from 2007.

USMTC — U.S. Manufacturing Technology Consumption — April 2008

A joint statistical program of AMT and AMTDA

	Apr 08 (P)	Previous Month	% Change	Year Ago Month	% Change	YTD 08 (P)	YTD 07 (R)	% Change YTD
National								
Metal Cutting	381.41	521.73	-26.9%	284.72	34.0%	1,510.31	1,250.65	20.8%
Metal Forming	15.06	26.08	-42.3%	22.14	-32.0%	76.97	73.17	5.0%
Total	396.47	547.81	-27.6%	306.86	29.2%	1,587.27	1,323.81	19.9%
Regional								
Northeast								
Metal Cutting	44.70	67.35	-33.6%	49.73	-10.2%	208.26	203.84	2.2%
Metal Forming	1.77	4.34	-59.1%	1.93	-7.8%	13.45	8.74	53.9%
Total	46.47	71.70	-35.2%	51.67	-10.1%	221.71	212.58	4.3%
South								
Metal Cutting	45.62	117.33	-61.1%	34.65	31.7%	257.20	146.20	75.9%
Metal Forming	3.75	5.40	-30.5%	4.29	-12.6%	14.41	13.96	3.2%
Total	49.37	122.73	-59.8%	38.94	26.8%	271.61	160.16	69.6%
Midwest								
Metal Cutting	149.09	160.30	-7.0%	76.23	95.6%	505.44	339.03	49.1%
Metal Forming	5.05	11.72	-56.9%	8.71	-42.0%	29.25	27.22	7.5%
Total	154.14	172.02	-10.4%	84.95	81.5%	534.69	366.25	46.0%
Central								
Metal Cutting	76.26	113.00	-32.5%	79.51	-4.1%	334.72	357.63	-6.4%
Metal Forming	3.63	3.92	-7.3%	5.56	-34.6%	16.28	15.16	7.3%
Total	79.88	116.92	-31.7%	85.07	-6.1%	351.00	372.59	-5.8%
West								
Metal Cutting	65.76	63.75	3.2%	44.58	47.5%	204.68	208.11	0.3%
Metal Forming	0.85	0.70	21.0%	1.65	-48.5%	3.58	4.23	-15.5%
Total	66.61	64.45	3.4%	46.23	44.1%	208.26	212.31	-1.9%

$ = millions of dollars
P - preliminary
R - revised
Totals may not match due to rounding

Haas

OPENS TWO FACTORY OUTLETS IN INDIA

In order to accommodate India's growing manufacturing economy, Haas Automation is opening two Haas Factory Outlets (HFOs) in the country. The ventures are located in Bihwadi, near Delhi, and Ahmedabad, raising the total HFOs in India to four, with a fifth scheduled to open in Chennai around June.

Last year Haas sold 549 machine tools in India, and it expects to sell more than 700 in 2008. By 2010, Haas anticipates selling more than 2,000 machines in India, according to the company's press release. "India has about 8 to 9 percent GDP each year, and this year should be no different. The HFO network grows with India's industrial revolution," says Justin Quan, Haas business manager for Asia.

HFOs distribute machine tools in the Haas product line, and they are locally owned and operated. They feature spare-parts inventories, Haas sales, service and applications engineers, service vehicles and customer training facilities. All Haas products are built at the Southern California headquarters and are distributed by the HFOs.

"Our HFOs are independently owned and operated to provide the local attention our customers deserve. They offer a professional atmosphere where end users can test-drive Haas products and see the latest technology," says Bob Murray, general manager of Haas Automation. "Customers can expect the same high level of service from each and every outlet, regardless of location."

Romax

JOINTLY DEVELOPS 3MW WIND TURBINE GEARBOXES

Romax continues to develop relations in the Chinese wind power industry with a contract to develop 3MW wind turbine gearboxes jointly with Dalian Huarui, a gearbox manufacturer. The project is already underway, according to Chen Lixin, the director of general gear reducer works at Dalian Huarui.

The companies will use Romax's in-house software, *RomaxDesigner*, for the design process. The prototypes' test and performance inspections will use Dalian Huarui's 3MW + gearbox test rig. The gearboxes will feature the product specifications of Sinovel Wind and follow wind turbine gearbox design guidelines. They intend to comply with Germanischer Lloyd certification.

"We are very pleased to be working with Romax, an international gearbox specialist," Chen says. "This collaboration will bring a global perspective to our business, and we are very confident that Dalian Huarui will become a significant player in the production of gearboxes worldwide."

Romax has been actively engaged in the Chinese wind energy market in various ways. The software was purchased by the China Classification Society, and it will be used to

continued

develop the wind turbine certification business. Baoding Tianwei, a transformer and electric transmission producer, recently completed a joint project with Romax in which they designed the main rotor bearing housing for a 1.5MW wind turbine. Romax participated in Windpower Shanghai and presented two technical papers at the European Wind Energy Conference. Romax CEO and founder, Dr. S.Y. Poon says, "We are proud not only to be in a position to provide the expertise and innovation that the Chinese wind energy industry needs to develop, but also to play some part in efforts to reduce the impact of the world's fastest growing economy on the environment."

DTR

OPENS CHICAGO-AREA SALES OFFICE

Formerly Dragon Precision Tools, DTR Corporation has a new regional office in Des Plaines, IL to help accommodate its expanding hob client-base in the United States. Alex Roh, in his role of sales manager, will be providing support to DTR's sales representatives and customers.

"As part of our ongoing expansion, DTR is pleased to open its U.S. office in the Chicago area," says Jongyoon Chun, CEO of DTR Corporation. "The strategic location of the U.S. office, as well as the addition of a U.S. sales manager, allows us to better serve our growing U.S. client base. Our local office will offer fast response to customers in need of technical assistance, quotations and delivery status."

Roh will be responsible for DTR's strategic direction in the U.S., and he has over 10 years of export sales and marketing experience. He has a B.S. in international trade and an MBA in marketing management. The office is located at 2400 E. Devon Ave., Suite 210, Des Plaines, IL 60018. Roh can be reached at (847) 375-8892 or e-mail *alex@dragon.co.kr*.

AMT

NAMES ROBERT K. SIMPSON PRESIDENT

Robert K. Simpson

The Association for Manufacturing Technology (AMT) Board of Directors announced that Robert K. Simpson will be its new president beginning October 7, 2008. He joined the association on June 2 to work in transition with current president John B. Byrd III, who is retiring after 5 years. Byrd says that Simpson "will be an excellent spokesman and advocate for the producers of American manufacturing technology. He has the skills, energy and experience to lead AMT and its members to continued great success."

Simpson, 48, has 25 years of international manufacturing operations experience in 17 countries. His most recent position was at Milacron Inc. of Cincinnati, where he served as corporate vice president and president of global plastics machinery. Other positions he has held include president of Siegel-Robert Automotive Inc. and executive positions within Textron and TRW.

"There's more to Bob Simpson than his experience," says Doug Currie, AMT chairman. "He has a passion for manufacturing, and he is committed to fulfilling AMT's mission, which is to help our members be the global leaders in advanced manufacturing technology solutions. Bob's been a leader his whole career, and we look forward to him taking AMT and our membership to the next level of success."

Simpson is a native of the Cincinnati area. He earned a bachelor's degree in manufacturing engineering from Miami University in Oxford, Ohio, and he has an MBA from Western International University in Phoenix, Arizona. Along with a search committee, the eight-month search for a new president was led by former AMT chairman Bradley L. Lawton, who expressed excitement over the appointment and says, "Bob Simpson impressed each of us, and we're confident that AMT will be in great hands for years to come."

SAE Shot Peening Sub-Committee

HOLDS FIRST MEETING

The Aerospace Metals Engineering Committee (AMEC) of the Aerospace Materials Division of the Society of Automotive Engineers has formed a shot peening sub-committee, which held its first meeting in Pacific Grove, California in January. The first order of business for the group is to update current shot peening specifications and generate new ones for the shot peening industries of today.

"Shot peening specs written in the 1940s by OEMs just don't make sense for today's MROs, job shops, product and media manufacturers," says Jack Champaigne, chairman of the shot peening sub-committee. "This group will be responsible for bringing shot peening specs into the 21st century."

The meeting was attended by 27 representatives from 22 companies in the United States, Europe, Asia and the U.S. Army. In the two-day work session, shot peening specs were reviewed. They discussed creating new specs for flapper peening, needle peening, ultrasonic peening, eddy current non-destructive testing, low-sodium glass bead for peening and the new industry standard—SAE J-2597—for use of computer-generated curves in shot peening intensity tests. Champaigne submitted 14 concepts to AMEC for consideration. Membership in the AMEC shot peening sub-committee is open to qualified individuals. Contact Jack Champaigne for more information at *jack.champaigne@electronics-inc.com.*

Richard Alan Group

LAUNCHES PUMPS & GEARBOXES LTD.

Pumps & Gearboxes Ltd., the new business unit from the Richard Alan Group, increases pump and gearbox availability to OEMS while fulfilling maintenance requirements in the U.K. market. "This industry has systematically reduced spare and redundant capacity, increased efficiency and ramped up production to the point where extended lead times on the supply of new and repaired pumps and gearboxes can have a massive impact on production," says Andy Batey, general manager of the new business unit. "Pumps & Gearboxes represents a combination of expertise and engineering capacity, brought together to address this issue by providing gearbox and pump solutions quickly and at a very competitive rate."

The company already sealed a major gearbox distribution deal offering Yilmaz Reduktor gearboxes and geared motors. The gearbox line includes worm, helical and bevel helical gearboxes with up to 40,000 Nm capacities, and they all come with a two-year warranty. Delivery service is quick due to the range of sizes stocked in modular format at the Dewsbury, U.K.-based engineering center, so they can be built-to-order the same day, according to the company's press release. The company's engineers can directly access Yimlaz production to order any shaft size and thread in any combination: imperial, metric or both.

Pumps & Gearboxes grew out of Panda, the pumps and ancillaries supply division for Richard Alan Engineering. A range of pump suppliers is meant to suit any pumping requirement. Suppliers include Wilden Pumps, Sandpiper Pumps, Versamatic Pumps, Aro Pumps, Ebara Pumps,

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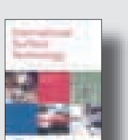


NEWS

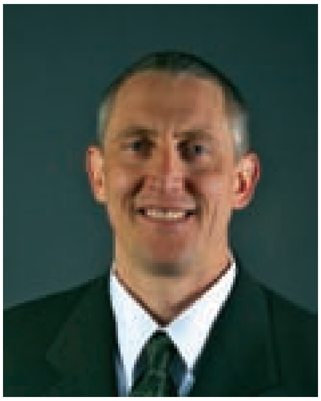
Boyser Peristaltic Pumps and Cemo Pumps' mono, orbit and interchangeable units. Mounts and base plates can be sourced quickly by accessing Ramco-CNC, the parent company's precision machining specialist division. The business unit provides repairing services on many pumps and gearboxes with short lead times.

"Customers can rely upon our experience and resources to come up with the right solution, a new replacement or a repair—even a temporary unit to cover emergencies," Batey says. "We pride ourselves on a problem solving attitude and great attention to detail. The business is already gaining new customers and quickly expanding into a market where time and cost is of the essence, and we are supplying that demand."

Bison Gear VP

RECOGNIZED BY NORTHERN ILLINOIS UNIVERSITY COLLEGE OF ENGINEERING

The Mechanical Engineering Outstanding Alumni award was presented to James Parejko, vice president, continuous improvement and engineering for Bison Gear and Engineering Corp. at the Northern Illinois University Engineering and Technology Alumni Society's second annual Alumni Recognition Dinner.



James Parejko

Parejko received both a Bachelor of Science Mechanical Engineering and a Master of Science Mechanical Engineering from NIU. Currently, he serves on the NIU Mechanical Engineering Industrial Advisory Board. He was promoted to his present position at Bison Gear earlier this year after joining the company in 2006 as engineering manager. He is responsible for directing the company's five areas of continuous improvement and engineering. Parejko has 15-plus years of experience in the power transmission industry at such companies as Emerson Power Transmission and International Truck and Engine Corp.

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Gear Garden Germinates from County's Industrial History

A new breed of blossoms sprouted this spring in York, PA cultivated from gears, sprockets, railroad spikes and other recycled metal items. So far, seven 10-foot-tall flowers, four 6-foot-tall flowers and a toadstool have burgeoned out of 12 sprockets or gears. Other metal parts used include a rock drill bit and electric motor housing as the toadstool's stem and cap, respectively. Railroad spikes were used to decorate the flowers, and other items were taken from a closed pottery plant including plate molds, propeller-shaped clay mixers and gears from the drying ovens, according to city council member Genevieve Ray. "Flowers are made to order, with each flower sponsored by a donor. Sponsors' contributions cover design, fabrication, installation, lighting and long-term maintenance."

And who would pass up the opportunity to buy a giant gear-flower? Ray, in charge of the sales effort, didn't expect the cash to roll right in considering the $5,000 price tag. She was pleasantly surprised to receive six orders right off the bat. There is space in the garden for at least a dozen more flowers or two dozen smaller three-foot flowers, which will fill the garden in and are selling for $750 each. Ray hopes more people will be drawn towards the more economical buds, especially now that the project has been such a success, which she believes is due to the fact that "It's locally grown. The gears and other artifacts are all from local industries, the artists are York County residents, and other elements of the park reflect York County industrial heritage."

York County is known as the factory tour capital of the world with doors open for visitors at more than 15 factories, including the Harley-Davidson York Vehicle Operations, Snyder's of Hanover and Hope Acres—one of the first robotic milking dairy farms in the U.S. The county was home to the York Motor Car Company, which created the Pullman automobile, a six-wheeled vehicle produced between 1905 and 1917. Today in York, you can find the North American headquarters for Voith Siemens Hydro, the only turbine manufacturer in the United States with a hydraulic laboratory, according to its website.

Photos courtesy of Tim Diener.

Blacksmith Tom Moore and sculptor Bob Machovec are responsible for sowing the gear garden. Machovec is known for sculpting creatures from metal objects he has found, like a blue heron made of shovel blades with a pickaxe head, horseshoes and rebar. Moore began blacksmithing in 1976 after working as a high school metal shop teacher. He invited Machovec to tour an abandoned quarry site Ray had learned of where the owner was willing to give away discarded industrial relics. The idea was born while the three of them were surveying the deteriorating quarry grounds. Machovec looked down at a sprocket and had a vision, "You could make a flower out of that."

"A Gear Garden!" Ray exclaimed.

The city of York was looking to re-design the downtown Foundry Park, which is on a main road but has been obscured by a parking lot. Ray and urban designer Robert F. Brown were searching for an entrance to entice passers-by and make the park more visible. "The Gear Garden was designed to provide a highly visible entry to the reconfigured park. It sits right on the street, and its giant flowers invite pedestrians into the park," Ray says.

A ribbon-cutting event took place June 12 while a series of day and evening events was organized to introduce the new park to the public. Activities featured musicians ranging from symphonic to hip-hop genres, and a kayak slalom race was sponsored by the local canoe club.

Urban renewal design, fine arts sculpture, fashion, whatever the medium, gears continue to evoke inspiration and find new avenues for expression. Those interested in sponsoring a gear flower should contact Genevieve Ray at (717) 848-3320.