November/December 2007

GEAR TECHNOLOGY

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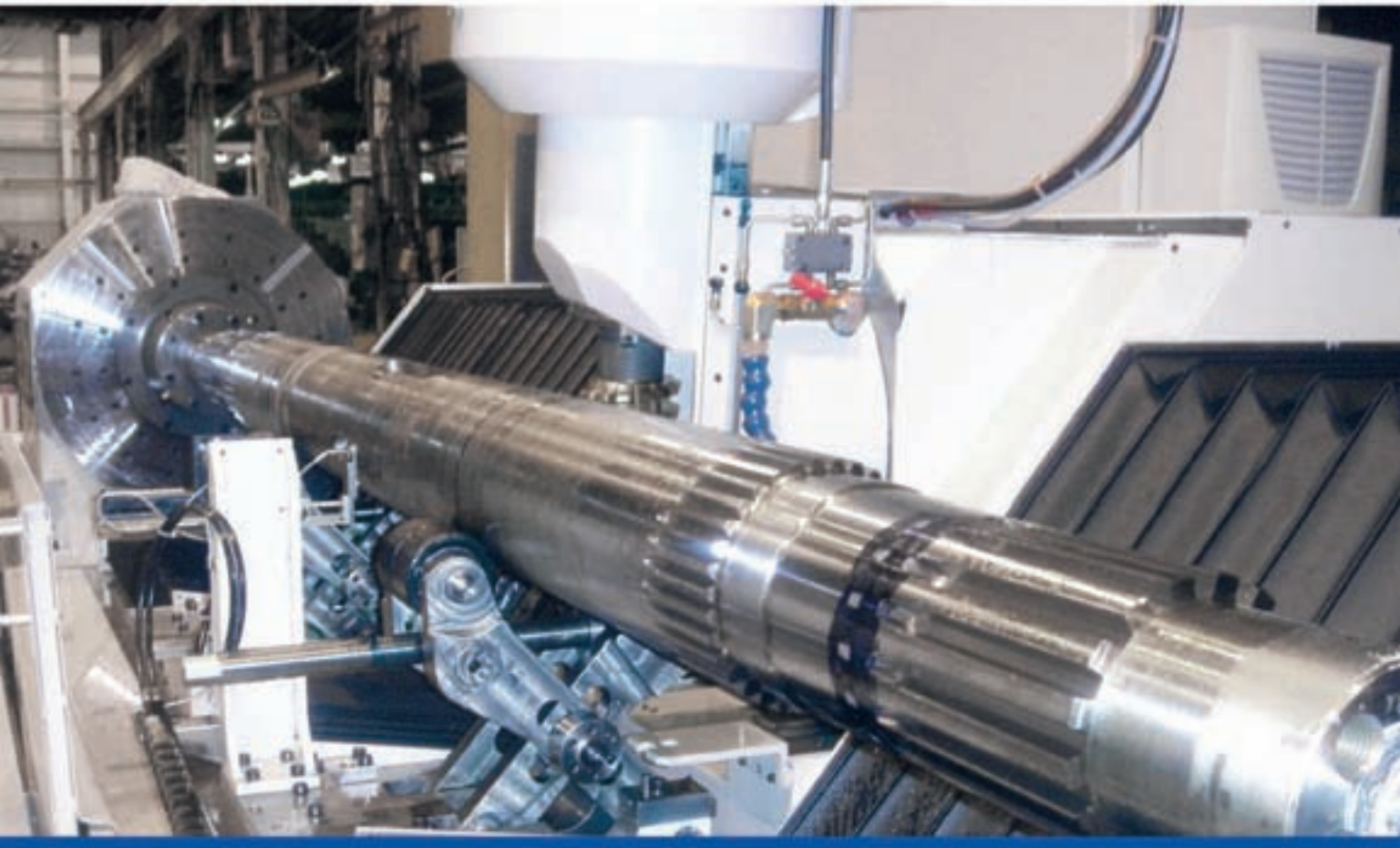

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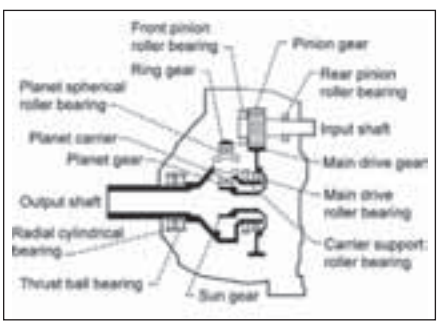

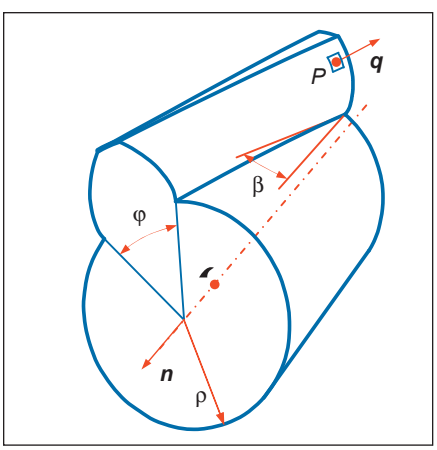

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Randall Publishing, Inc.
1425 Lunt Avenue
P.O. Box 1426
Elk Grove Village, IL 60007
Phone: 847-437-6604
Fax: 847-437-6618

EDITORIAL

Publisher & Editor-in-Chief	Michael Goldstein
	publisher@geartechnology.com
Managing Editor	William R. Stott
	wrs@geartechnology.com
Senior Editor	Jack McGuinn
	jmcguinn@geartechnology.com
Associate Editor	Matthew Jaster
	mjaster@geartechnology.com
Editorial Consultant	Paul R. Goldstein
Technical Editors	Robert Errichello, Don McVittie,
	Robert E. Smith, Dan Thurman

ART

Art Director	Kathleen O'Hara
	kathyohara@geartechnology.com

ADVERTISING

Advertising	RK Media, Inc.
	Ryan King
	ryanking@geartechnology.com

CIRCULATION

Circulation Manager	Carol Tratar
	subscribe@geartechnology.com

RANDALL PUBLISHING STAFF

President	Michael Goldstein
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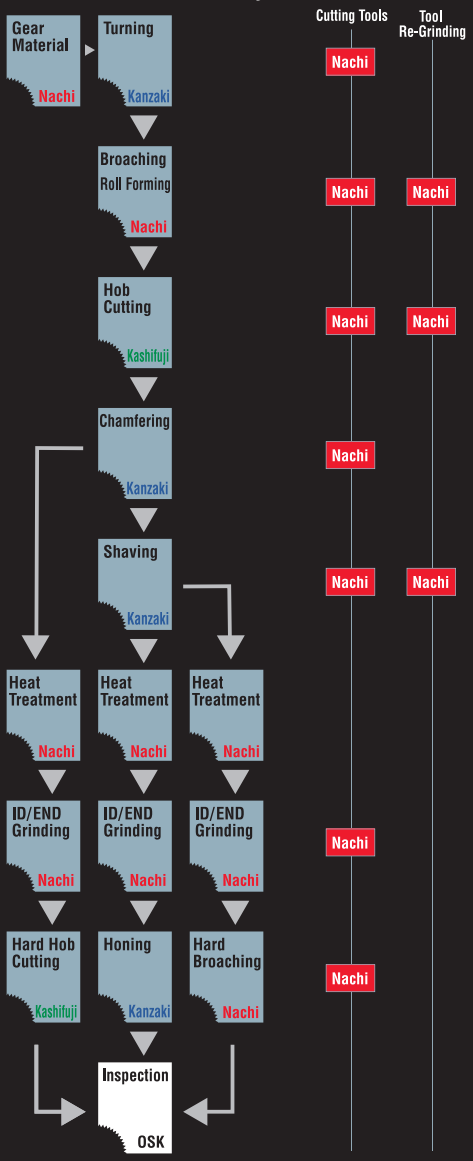

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Musings On Turning 65

Recently, I reached a milestone. In October, I turned 65. I've been thinking a lot about it lately, and certain things are bothering me about this time of my life.

It's not about growing older. My dad used to tell me, "You only get old if you get lucky." And I don't feel old. No, what bothers me are many of the issues surrounding retirement age and our country's Social Security system.

As the baby boomers begin to dip their straws into the U.S. Treasury, the Social Security system is going to look like a riverbed out West in the middle of the winter—dried out and breaking apart. The idea of registering for Social Security and Medicaid has caused me to think about this terrible problem that few dare to address, let alone actively attempt to solve—and it's only going to get worse.

The thing I can't get out of my mind is the enormous burden being placed on our younger generations. When FDR signed the Social Security Act in 1935, there were more than 30 workers for every retiree. The whole program seemed like a great idea. In 1950, there were 16 workers for every retiree—still not too bad. Today, there are just over three workers per retiree. Some estimates say that by 2020, there will be only two workers per retiree.

Over much of the recent history of the program, there has been plenty of money to pay benefits, because there have been significantly more workers than retirees. Even today, a surplus of money is collected. Unfortunately, though, that surplus just goes to offset a portion of the huge deficits the government runs outside of Social Security.

Within the next 10 years, the Social Security system will have a negative cash flow. The retiring baby boomers will only exacerbate an already untenable financial situation.

Because of the bureaucracy—a lot of people work for the Social Security Administration—the government has to take in more money than it is able to pay out, even if it's just going to break even. Hypothetically speaking, let's say a retiree is getting $15,000 per year in Social Security benefits. In order to support that retiree, the government might have to take in $20,000. If we have only two workers per retiree, then each worker will have to contribute $10,000 a year.

When I was a young man, there was never enough money to provide everything for my family, and I don't imagine that's changed. In fact, I'm pretty sure families today have a harder time making ends meet. Having to give up $10,000 while trying to accumulate money for a young family's needs—like a home, a car, saving for college, investing for the future, etc.—is a daunting or impossible scenario for all but the super-rich.

Many people who are getting or are about to get Social Security benefits look at it as an entitlement. You've been paying into the system your whole career. Now it's time to cash out.

Unfortunately, though, our government hasn't set this system up to work like an annuity, where the money is set aside, invested, grown and then made available for spending upon retirement. Instead, it's a "pay as you go" system; today's workers are paying the benefits for today's retirees.

Basically, this is an income transfer from the younger generation to the older generation, not an investment or savings scheme, as many people view it.

The government doesn't have an inexhaustible supply of money. We have to remember that we are the government—and whatever the government spends, it gets from us, one way or another. So in order to keep Social Security afloat, taxes will have to be raised, spending in other areas will have to be cut, or Social Security benefits will have to be reduced. Possibly, all three might happen. The money has to come from somewhere.

I have no solution to these problems, only concerns. Even if I did have a solution, who would listen? Politicians seem to want to only promise to give us more, never to discuss the consequences or where the money will come from. That's a problem for future generations and future politicians. I'm only here to lament and muse about something that's bothering me as I make this transition in my life.

If my dad was right, and you only get old if you get lucky, I expect that the next generation will have to be even luckier than those in the past to have anywhere near the security and lifestyle the current generation of retirees will experience.

Michael Goldstein,
Publisher & Editor-in-Chief

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Letters to Editor

A Few Minor Tweaks

Chicago's Movable Bridges—

A Perfect Balancing Act of Gear-Driven Splendor

Jack McGuinn, Senior Editor

Every spring and fall, a flotilla of pleasure craft rely on Chicago's string of downtown movable bridges for access to and from Lake Michigan. (Photo by David M. Solomon)

Chicago has been known as many things over the years—"Hog Butcher to the World," "The City That Works," "The Windy City" and "The City of Big Shoulders" among them.

Although perhaps lesser known, add "City of Bridges" to the list.

But we're not talking about traditional, stationary bridges. Rather, we're talking trunnion bascule bridges—or drawbridges—that span the Chicago River along many of the Loop's downtown intersections.

Although Chicago's first trunnion bascule bridge is the Cortland Street Bridge, built in 1902, the crown jewel of the city's movable bridge system is the Michigan Avenue Bridge at the intersection of Wacker Drive and Michigan Avenue. Begun in 1917 and completed in 1920, the bridge was conceived by renowned city planner and architect Daniel Burnham—he of "make no little plans" fame—and designed by transplanted Englishman and Chicago Plan colleague Edward H. Bennett. Burnham envisioned the Michigan Avenue Bridge in part as a means of encouraging more development of Michigan Avenue north of the river, which had been neglected due to a lack of speedy access from its more populated area to the south. Reminiscent of the meeting of the eastern and western railroads in the previous century, the bridge dedication was celebrated by cannon fire, marching bands, blaring fog horns from the ships below and thousands of Chicagoans.

continued

Dear Editor:

As one who has been intimately involved with movable operating machinery for the past 40 years, I was delighted to read your article about Chicago's many movable bridges. Over the years, I've worked closely on bridges with Dan Burke and Tom Powers, as well as many of their predecessors going back to the days of Henry Ecale, Fred Olivi and Lou Konza.

While your description of the bridges is most interesting, I noticed several errors that should be corrected. When describing the Michigan Ave. double-leaf, double-deck bascule, you conclude, "… most other bridges in the system have only one leaf." This is not true. Fact is, most Chicago trunnion bascules have two leaves, but only a few are double-deck bridges—Michigan Ave., Lake Shore Drive, etc.

A couple paragraphs later, in describing the drive system, it is noted, "… those four motors drive the trunnion—or axle—which in turn activates a gear set of increasing size to power the bridge. The last in the series—the bull gear—has the largest teeth of the set and actually rotates the rack and in turn powers the raising and lowering of the bridge sections." This description is a bit confusing as well as somewhat inaccurate since the motors do not drive the trunnion.

Here's another way to describe the drive system: Each of the four motors drives a series of reduction gears, which reduce the speed and increase the torque to a final member called the rack pinion, which meshes with a large gear segment called the rack that is mounted on the movable leaf; thus causing the leaf to rotate about the centerline of its axles, known as trunnions, to the open or closed position. One might visualize the leaf being a segment of the last gear in the power train, rotating only about 90 degrees.

Please accept these as constructive observations and not nit-picking; overall you did a good job in publicizing movable bridges.

Sincerely,
Bob Cragg

Call To Action

Dear Editor:

I am writing to express my appreciation for the superior coverage you and your staff have given this year's Gear Expo. Your clever cicada editorial in July was a good call to action, and you put the burden right on the bull's eye. This show belongs to the industry. Due to your good words and the actions of a number of others—exhibitors have mailed over twice the number of guest passes as in any prior year—indications are that this will be a good year for attendance.

We have had to ask the hotels to increase the room blocks for Gear Expo multiple times; advanced registrations are well ahead of the history, and registrations for the educational programs appear strong at this point.

But you outdid yourself with the August issue! Beyond the informative show preview, your action to make sure the Gear Expo logo was on every exhibitor's ad makes the event truly stand out. Again and again it is one of the first images one sees on each page. Thanks. Please extend my appreciation to your staff. I know it takes many hands and brains to do such good work.

Sincerely,
Joe T. Franklin, Jr.
American Gear Manufacturers Assoc.
Alexandria, VA

Editor's note: *According to AGMA numbers, Gear Expo 2007 attracted more than 3,000 total attendees (including exhibitors), making it the largest show since 2001. See our post-show coverage on p. 64.*

PUBLISHER'S PAGE

What's All the Buzz About?

What's that you said? I'm sorry, could you repeat that? Would you speak up, please? I can't hear a thing you just said!

Sorry about that. It's just hard to hear around here lately with all the cicadas. Many of you have probably heard—or maybe you've been infested as well—but the Midwest is home to a very populous brood of 17-year cicadas, and 2007 was their year.

For 17 years, these cicadas live underground as nymphs, quietly sucking the sap from tree roots. Most of us don't even know they're down there all that time. But boy, do we know now. We're told that, in some areas, there are 1–1.5 million cicadas per acre (400,000–600,000 per hectare, or 250–370 cicadas per square meter).

Around the beginning of June, the nymphs began crawling out of the ground and making their way up tree trunks, where they attached themselves, shed their skins and transformed into winged adults. And then the chorus began.

In some areas, the noise is almost deafening. It seems more or less constant, but if you stand and listen, you can hear the cicadas react to each other en masse, their chirps rising in great waves of crescendos.

And then there are the dive-bombers. Cicadas aren't particularly good navigators, and they'll latch onto pretty much anything they run into, even—say—an editor out for an evening stroll.

By the time you read this, this batch of the 17-year cicadas will all be gone. They'll have mated, the females will have laid their eggs, and the adults will have died and their songs ended. Of course, the eggs will hatch and new nymphs will drop to the ground and burrow underground, where they'll stay for 17 more years.

Even though they are above ground for only about six weeks, the cicadas have given us plenty to talk about around Chicago. The whole town is abuzz with them, if you'll pardon my pun.

But for me, all the talk about cicadas reminds me of another cyclical event, one which is about to arrive in the gear industry. Of course, I'm talking about Gear Expo, which will take place October 7–10 in Detroit's Cobo Center.

Like the cicadas, Gear Expo comes on a regular cycle.

So I wonder: Will Gear Expo 2007 generate the same kind of buzz in the gear industry that the cicadas have generated in the Chicago area? Will gear industry professionals swarm to Gear Expo like the cicadas have swarmed some of our neighborhoods? Will we be talking about Gear Expo 2007 for months and years to come, or will we forget about it as soon as it's over?

It's hard to say. Mostly, the answers depend on you. AGMA has been running ads in every issue urging you to "Grab Your Gear and Go" to Gear Expo. (See page 55 for AGMA's show ad.) But how many of you have taken the time to log on and register? How many have penciled it in to your schedules? How many have booked reservations?

In theory, people attend trade shows when they have money to spend, when they need extra manufacturing capacity, when there are new technologies being introduced, or just to meet friends and develop relationships. I know that most of you are pretty busy. Hopefully, that means you're also making money. It probably means you're in need of extra capacity, but it also means you might not find the time to go to the show. But like the song of the cicadas, Gear Expo is a limited engagement. If you miss it, you'll have to wait another two years.

The onus is on you. If you value the show, if you value seeing new technology, if you value the opportunity to meet with the designs, sales and service personnel of your suppliers, then support Gear Expo by going to Detroit.

Let's reverse the recent trend of dwindling attendance and make this a year to remember.

After all, a strong show demonstrates a strong industry. We'll be there, as always, making as much noise as we possibly can. We hope you'll join us. Let's give the gear industry something to buzz about.

Michael Goldstein,
Publisher & Editor-in-Chief

Outside the Gear World

PUBLISHER'S PAGE

THE WAR

Coming September 2007

I recently had the opportunity to attend a presentation given by Ken Burns and Lynn Novick about their upcoming 7-part documentary *The War*, which is scheduled to air on American public television (PBS) beginning in late September. Included in the presentation was a 1½-hour preview of the film.

When the presentation was over, I was physically and emotionally stunned, as were most of the people in the audience.

This is not just another war story, but a story about *The War*, WWII—the one that engulfed most of the world, the one in which 50 million people died. At the presentation, Burns and Novick described their approach and reasons for making this film.

I realize as I write this that many of you readers are German, Japanese or Italian and might think about this film as a story told from a victor's standpoint, but it is universal in its scope and treatment. The film's war-era footage comes from the British, German, Japanese and American archives. It includes interviews with veteran German soldiers, U-boat captains, sailors and civilians. This is not a story about famous historical battles or decorated heroes. It's not a story about who won or lost the war. Rather, it's a compelling story about the people who lived during that time and how the war affected them.

Its impact on older people, like some of the veterans in the audience, was significant. Those people lived through this time. Those of my generation, who were born during the war, remember our fathers and uncles and cousins who went off to fight, some returning to tell their experiences. But even for those born later, who may not have any direct connection with WWII, the universal themes and lessons allow everyone to relate.

Saipan, 1944. Credit: U.S. National Archives and Records Administration. Courtesy of PBS.

Guadalcanal, Solomon Islands, February 1, 1943. Credit: AP Images. Courtesy of PBS.

When you see footage of thousands of men on a beach, or on ships at sea, and then cut to interviews with veterans, wives, children, sisters and brothers, you get a sense of the enormous loss and destruction caused by that war—or any war, for that matter. Each of those thousands was an individual, and they had wives, children, sisters and brothers who thought of them and feared for them while they were gone. Seeing this film—and I only saw a fraction of it—imparts a better understanding of the impact on everyone involved.

Bringing such a huge and overwhelming world event down to the personal level was extremely important to Burns and Novick. In a recent interview, the filmmakers explained why they didn't use professional historians. "We wanted to make sure everyone in our film was either in the war or waiting anxiously for someone to come back from the war," Burns said. "You can make a film with historians 20 or 30 years from now," Novick added.

At the presentation, they said they needed to make this film now because WWII veterans won't be around to interview in another 10 years. According to the filmmakers, WWII veterans are dying at the rate of 1,000 per day.

For their interviews, Burns and Novick picked four towns—Mobile, AL; Sacramento, CA; Waterbury, CT; and Luverne, MN—that they thought would give them a good cross-section of America geographically and economically, with a good mix of farming, manufacturing and urban communities. The filmmakers went to those towns and advertised their project, wanting to meet, talk to and film everyone and anyone that had some experience related to that time.

One of the most powerful effects this film had on me was the realization of how young the soldiers were. Do you know any 18-year-olds? Sons, daughters, nieces, nephews? Can you imagine any of them going off to fight in a war? How about the kid who mows your lawn or bags your groceries? Can you picture any of them fighting for their lives, fighting for their countries? One of the things the film expresses very

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Dear Editor:

I just wanted to let the Editor-in-Chief know how much I enjoyed his editorial and suggestion that I watch *The War* series on PBS. As a 40-something daughter of a WWII vet, I knew little. My father didn't talk too much about his experiences, but always showed such pride and dedication to our country that it filtered into our family values and moral obligations. Thanks for writing a piece outside of the gear world. Although my father and many WWII vets have now passed, the special helped me understand some of what he (they) must have felt, and how the country at home dealt with this victory/loss, etc.

I am a better American and just wanted to say thanks.

Maria Scherer
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Gleason built a factory in Harbin, China to be closer to its customers.

For many of those in the gear and gear products business, these may seem like the best of times. The global economy is very robust, led to a large extent by emerging economies like China, India and Eastern Europe, and growth in many industry sectors. In addition to demand from mature end-markets like automotive, truck and aerospace, developing countries and geopolitical issues have spurred extraordinary demand for energy, construction, mining and various other elements of infrastructure. Consequently, worldwide demand for metalworking equipment is running at record levels, and the markets for gears and gear products are beneficiaries of this global boom.

However, living in times like these and capitalizing on them may be two entirely different matters. The first step is recognition and acceptance of the fact that it's not business as usual out there; it hasn't been for some time, and it never will be again. Our markets and customers are continually changing and evolving, driven by conditions and factors far greater than our relatively small industry. Consider some of the trends that drive our customers and markets today, and reflect on how you are changing to accommodate them. I am looking from the perspective of a tooling and equipment provider, but most of these trends cut across most manufacturers, including gear producers.

Globalization of the Customer Base. Gone are the days when a customer could be clearly identified as a U.S. customer, a European customer, a Japanese customer and so forth. Mergers, acquisitions and alliances among our customers have proliferated. Many have expanded their production into emerging markets, forming a truly global customer base, with significant implications for all of us. Global purchasing practices have become commonplace. Communications technology has overcome the distance factor. Markets are more transparent, as customers worldwide have far greater and timelier access to information on available technologies, suppliers and support capabilities. Manufacturing technology in one successful installation on one continent is often replicated in another.

Lean Enterprise. To one degree or another, many of our customers have implemented lean practices and expect suppliers to support them with the same. Lead time reductions, reduced inventory levels, elimination of "waste" and higher equipment reliability and uptime are expected. Manufacturers continually assess and focus on core competencies, resulting in the shedding of non-core processes and activities. Design, parts-making and whole assemblies may be candidates for outsourcing—often to low-labor-cost countries.

Loss of Gear Expertise. Many

OEMs have lost much of the technical gear knowledge they once had. They are looking for more automated solutions both in terms of design as well as manufacturing processes. "Intelligent" systems are evolving to meet the increased expectations for higher precision, productivity and repeatability. This trend creates incremental demand for gear-related services and training, as well as opportunities for such things as improved machine and process capabilities, including on-board inspection, adaptive controls, networking, closed loop inspection and corrective systems, part marking and tracking systems, and a whole host of other technologies.

Global Competition. Competition within the markets in which our customers compete as well as within the gear and gear equipment industries has never been greater. One might think that with global market conditions being relatively strong, the competitive pressures would not be as great. Not the case. Gear quality standards have never been higher, and tolerances have never been tighter. Gear accuracy, strength and noise quality characteristics are higher than ever. As a result, grinding and other hard finishing processes are seeing tremendous growth for many applications. The capabilities of equipment are being extended, tool life and performance are constantly being challenged, and more manufacturing processes are required to be "dry" and environmentally friendly. Because of globalization and advances in technology, competition is escalating like never before.

Sounds pretty daunting? What is a gear producer or a tooling and equipment supplier to do? Market and customer change is the environment we are in; accept that environment and embrace the opportunities it presents.

Provide Customers with Complete Solutions. Today's market demands not just discrete products and services, but "system" solutions to customer needs. Gleason has responded by providing a broad array of gear-related products and services, including solutions for virtually all types and sizes of both cylindrical and bevel gears. We produce equipment supporting all the major gear processing methods including hobbing, shaping, shaving, lapping, finish hobbing, grinding and honing, along with a full range of test and inspection equipment. Importantly, we can complement this with a full line of cutting tools, dressing tools and workholding solutions. All of these capabilities are necessary to be a total solutions provider for gear processing. In what way can you provide more complete solutions to your customers?

Be Near your Customer. In spite of the trend toward globalization, today's customer base demands even greater service and support at the local level. Gleason has 10 manufacturing operations on three continents, in addition to six technical support centers located in key markets, augmenting customers' gear expertise and bringing local support in the form of application engineering,

continued

service, spare parts, tool sharpening, maintenance and management.

In emerging markets, rapid growth and development is providing significant opportunities that will only be realized by those suppliers who can demonstrate support at the local level. Demand is strong and will be filled by someone—either by established suppliers or by a new breed of domestic competitors for gears and gear equipment. Gleason has been in China since the early 1970s and now has two manufacturing plants, one for cutting tools and one for machine assembly, in that fast growing market. We also have cutting tool manufacturing and machine rebuild capabilities in India, which in recent years has been growing at a similar pace to China. Being in markets like China and India is not as much about low cost as it is about being near your customers.

A passion for improvement. As with many of our customers, Gleason has embraced lean principles that have helped us to streamline operations, reduce lead times and reduce costs. We visit many factories to learn best practices and have had hundreds of customers visit our operations around the world to understand how we are building a lean culture. We hold many kaizen events with our customers aimed at eliminating waste in the system and reducing process times. It is rewarding to be part of the improvement process with your customers. After all, our success is tied to making them more successful.

For our company, I wish it was so easy to say that Gleason is the largest producer of gear equipment and tools in the world. What else is there to do? Well, bigger is not necessarily better. "Better" starts with listening to your customers and being passionate about serving them—creating a culture in your company that encourages change and is then committed to implementing those ideas.

The economy will cycle up and down; demands for lower costs, faster production times and higher quality will increase. I can make these statements with total certainty. I can also say with certainty that those companies that are customer-centric and have a culture of continual improvement will not just say these are the "best of times" now, but that the best is yet to come.

John J. Perrotti,
president and CEO, Gleason Corp.

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Richardon proudly offers the probably best ever offered Gear Hobbing Machine for gears up to 40" in diameter, .8 DP, 50"face.

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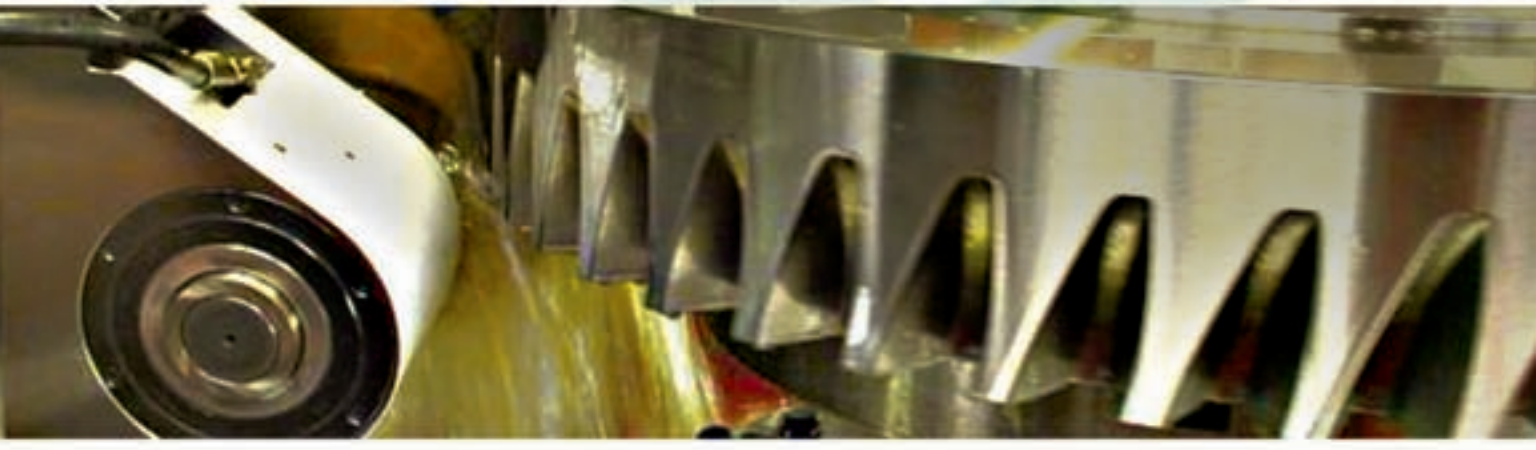

Cut your expensive grinding times with an already accurate gear before grinding.

Richardon offers turn key installations and operator training included. 2 years of warranty for the machine (one year for electrical parts) and 5 years of warranty on the table drive and bearing and all mechanical Richardon components*, Siemens service contract available upon request.

Richardon GmbH is a first generation, privately owned corporation with a passion to built first class hobbing machines. For special requests: talk to the owner.

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Reinvesting in New Equipment Pays Dividends

Frederic M. Young, president, Forest City Gear Corporation

Spending money (on new machinery) to make money has worked well for Forest City Gear.

Recently, I was approached by a colleague who is a manufacturer outside the gear industry. He was impressed by Forest City Gear's ability to continually acquire new manufacturing equipment, and he wanted to know more about our capital investment strategies. I responded to my colleague by e-mail, but I thought the readers of *Gear Technology* would be equally interested in some of my comments.

Our strategy over the years has been to buy new equipment to gain the experience of higher productivity and quality. Generally speaking, we end up expanding our size range and technology with each new machine, allowing us to pursue jobs for which we may not have been competitive previously.

A further benefit is the marketing gain you realize when customers see all the new toys on your floor. Your employees will take more pride in their work, and with each new acquisition, there is an opportunity for additional employee training on state-of-the-art equipment, allowing you to produce faster.

We receive cross-pollination when training with the setup guys who come in to install new equipment, as they have been exposed to benchmark shops all over the world and have been challenged to conquer very difficult work at their individual customers' locations.

You should have more uptime on newer equipment and should not have any significant maintenance expenses. Because we have been purchasing so much new equipment for so long, our depreciation allows us to continue to buy lots of new stuff courtesy of the depreciation tax advantage.

Finally, over time, we have developed quite a list of potential buyers for our surplus equipment, selling it to customers, competitors or trading it in against newer equipment. Usually you are better off if you can sell it yourself versus trade-in, as your vendor has to buy low enough to allow him to warranty and bury sales expenses.

We also get a fair amount of work steered in our direction by the folks from whom we purchase our equipment.

As I am fond of saying, your customers want you to have the

equipment on your floor with people already trained to use it, rather than your potential offer to go out and purchase new if they give you an order.

By buying new, we've been able to expand our productivity without adding a proportionate amount of people. For over 30 years, I have been re-investing 25–40% of our gross revenues in new equipment, much to the chagrin of our accountant, lawyer and bankers. Needless to say, I believe I've finally convinced them this is a safe and viable option which will ensure the survival of our company and allow us to grow.

A cancer in our country is the need for short-term return, which seems to be driven by accountants and stockholders. The Europeans and Asians seem to take a longer-term view—say over a 10-year period versus six months or a year in the USA.

Without reinvestment, one day you'll wake up to discover you are sitting on a hollow shell—a dinosaur which is worth nothing. In the interim, you have lost your ability to compete in a global marketplace. I hope to bequeath a healthy, world-class operation to my successors with a reputation that will continue to abet their success.

If I were to change the modus operandi, it would be to purchase the equipment myself, setting up a separate corporation to take advantage of tax deductions for me. This, of course, depends on whether you are a C or S corporation and your personal tax situation. Also, because we have bought so much foreign equipment, I have suffered through some currency fluctuation cycles when the cost of equipment is artificially higher due to the exchange rate. With lots of crystal ball gazing, I would be sure to purchase lots of equipment when the dollar is strong. Usually, that occurs at a low point of the business cycle for us, when it is hard to justify buying new stuff for lack of work. However, I have discovered that you can offset or attract a bigger market share by having newer equipment, which negates that argument.

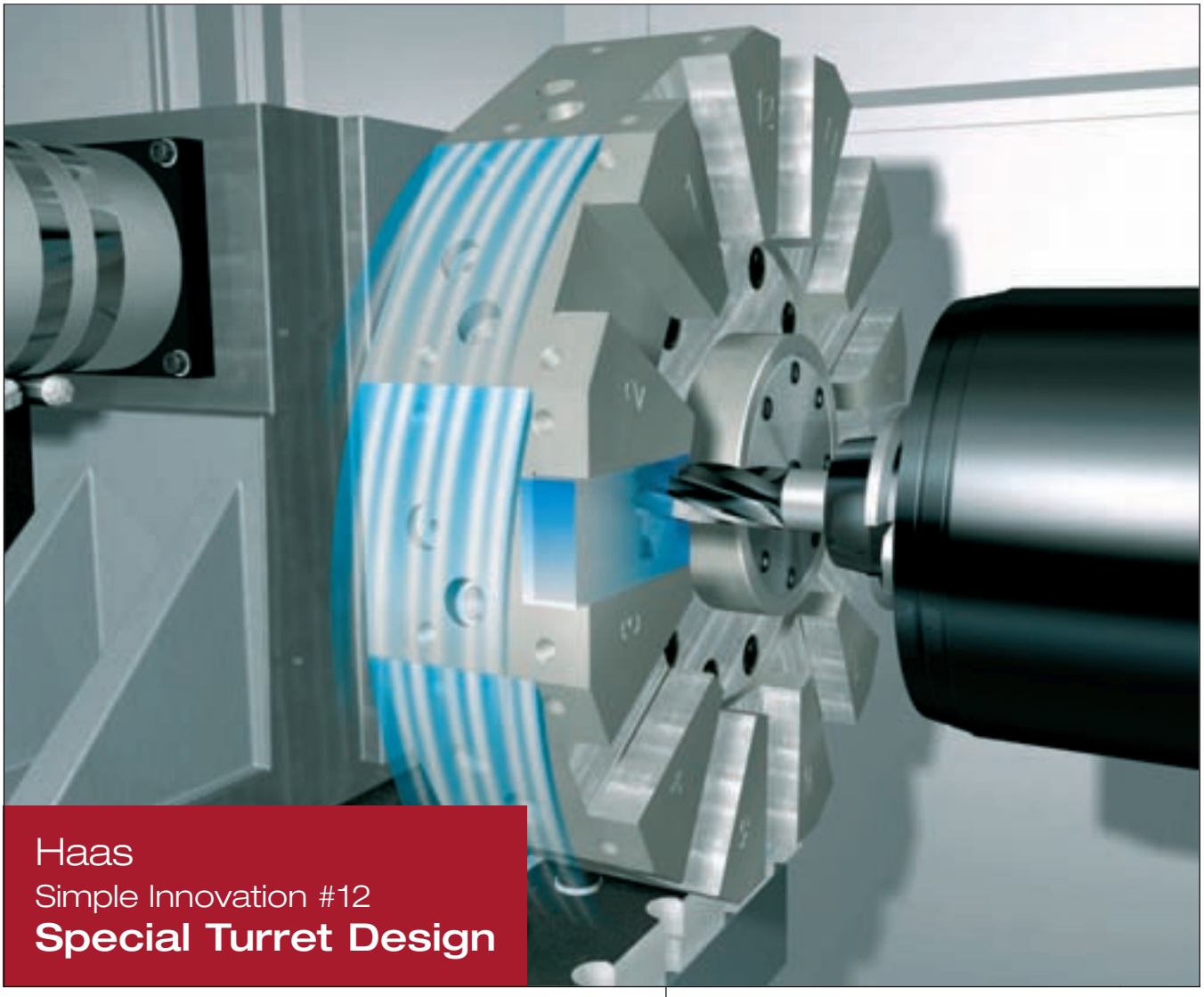
Once you get the snowball rolling downhill, it acquires its own momentum.

Our bankers seem to applaud these efforts because of the significant cash flow from our acquisitions and the higher profit margins we obtain by being able to produce gears upon which few can compete.

To encourage my employees to take advantage of the potential gains from the new equipment, I have often sent them for training directly to where they are made and at the same time tried to get them in other shops who are benchmark manufacturers. Naturally, sending an employee to Europe is expensive, but I am paid back by their loyalty and enthusiasm for the new equipment.

Hopefully, this conveys my strategy and philosophy on reinvesting in new equipment. I have been lucky enough to convince a few of my peers in the gear industry of the wisdom of this strategy, and in each case, they report to me the benefits of this approach.

Frederic M. Young, president
Forest City Gear Co.



Haas

Simple Innovation #12

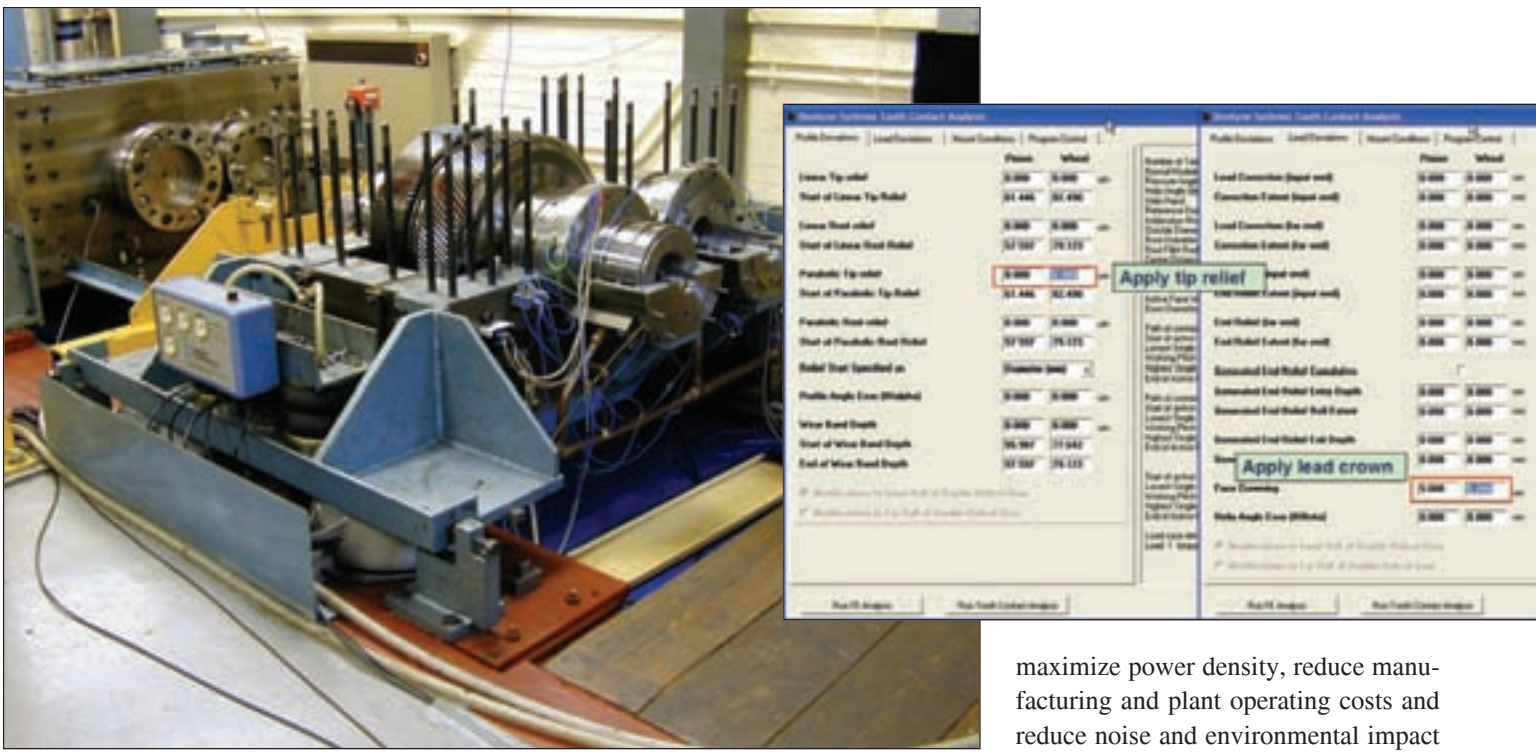
Special Turret Design

Clever Machining Yields High-Accuracy Positioning

The turrets for Haas SL Series turning centers are finish-machined as a unit – completely assembled with the coupling, gearbox and drive system. Keying off the turret's centerline, all critical features of each station are machined using single-axis moves, then the turret is indexed to the next station. This ensures the absolute concentricity of each station with the turret's true center. Compared to machining the turret before assembly, this reduces positioning variation from tool to tool by more than 90%. The result is easier job setup, higher accuracy and better repeatability.

Dontyne

DEBUTS GATES SOFTWARE FOR ITS GEAR PRODUCTION SUITE



Research efforts between Dontyne and the Design Unit include noise and vibration studies seen here.

Dontyne Systems, a U.K. company founded by Michael Fish and David Palmer, recently unveiled a new software program for its Gear Production Suite. *Gear Analysis for Transmission Error and Stress* (*GATES*) is a program for the calculation of stress and transmission error conditions in loaded gear systems.

Based on the *DU-GATES* 3D FE package from the Design Unit at Newcastle University, U.K., the program allows users to achieve reductions in gear noise. Dontyne Systems has compiled more than 10 years of research and noise and vibration studies on single- and double-helical gears to compile its data.

"The *GATES* analysis addresses an important need in the industry," says Fish, managing director, Dontyne Systems Ltd. "Specific instances are marine applications and wind turbines where such a punishing environment could benefit greatly from an application to minimize potential vibration."

According to Fish, recent revisions to ISO 6336 (the calculation of load capacity of spur and helical gears) have significantly improved the validity of this gear rating standard. However, limitations do still exist. It does not optimize tip and root relief or helix corrections, the so-called 'micro-geometry' of the gear. Correctly designed gears can reduce noise and vibration, minimize local contact stresses, increase scuffing resistance and help minimize mesh losses.

The goal of Dontyne Systems is to maximize power density, reduce manufacturing and plant operating costs and reduce noise and environmental impact of their products.

The *GATES* program itself examines a gear set during operation. The graphic and reporting features offer a quick interpretation of the results for customer reports. Links to inspection equipment permit data to be adopted, thus improving the accuracy of the model. This function also enables existing components to be optimized. Basic operating features include surface definition and thin-strip analysis. Advanced features include 3D analysis and definition of shaft and bearing conditions.

While Fish states the program does not in fact design a quiet gearbox, application of the gear-optimizing software can ensure a quieter running of the gearbox across the design range compared to using non-optimized gears. Proven application of *GATES*

continued

has seen up to 16 dB noise reduction.

Recently, *GATES* has been implemented by several U.K. gear design and manufacturing institutions, and used in an extensive range of marine, industrial and automotive applications. "The demonstration version of the program has been received by the current users who have commented on the ease of use," says Fish. Tooth contact analysis as well as complex results can be easily visualized thanks in part to a user-friendly interface that interprets the data. Fish believes the calculation speed will significantly improve in future releases. He also believes *GATES* is an excellent teaching tool for both the experienced gear designer and the novice.

Fish believes the collaboration between Dontyne Systems and the Design Unit at Newcastle University, U.K. has been beneficial to the entire process, though there's still work to be done. "There is the possibility to extend the analysis capabilities to greater range of gear forms in the coming years with appropriate test programs out to validate the calculations," says Fish. "We would welcome collaboration with the industry on the subject."

For more information:

Dontyne Systems
1 Simonside
Prudhoe, Northumberland NE42 6LJ
England
Phone: + (44) 1661 833 828
Internet: *www.dontynesystems.com*
E-mail: *uk@dontynesystems.co.uk*

Gleason

DEBUTS OPTI-CUT TOOLS FOR GASHING, SHAPING AND HOBBING

A new family of cutting tools was recently introduced by the Gleason Corporation for the gashing, shaping and hobbing of gears. This new line is targeted for larger gears, particularly for the wind energy, truck, tractor and power generation markets.

"Our new line of Opti-Cut cutting tools perfectly complements our machine products by extending our capabilities to provide a complete solution to our customers' gear processing needs," says Robert Phillips, senior vice president of Gleason's Tooling Products Group.

The cutting tools were recently exhibited at Gear Expo in Detroit as

well as at EMO in Germany. "There was a lot of interest at both shows," says Dave Melton, communications manager at Gleason. "Our customers were pleased they can get this technology at Gleason."

Opti-Cut is being offered through a cooperative venture with Ingersoll Cutting Tools Company, a supplier of metal removal tooling and a developer of indexable carbide cutting tools.

The entire line of Opti-Cut tools is designed to operate on the customer's existing equipment. According to the company's press release, replaceable inserts offer more efficient cutting action with higher feeds and speeds, lower cost per workpiece and optimized machine power consumption.

According to Melton, wind energy has been the biggest driver of this technology, though potentially anyone cutting large gears can benefit from this type of tool. The mining, construction and energy fields also use this type of tooling.

For more information:
Gleason Corporation
1000 University Avenue
P.O. Box 22970
Rochester, NY 14692-2970
Phone: (585) 473-1000
Fax: (585) 461-4348
Internet: *www.gleason.com*

ADCOLE

OFFERS HIGH-SPEED CRANKSHAFT GAGE FOR BETTER

A new high-speed, in-line, automatic crankshaft measurement gage for production environments that provides sub-micron accuracy has been introduced by Adcole Corp. The Adcole

continued

Model 1300 High-Speed Crankshaft Gage features a powered tail-stock and part driver to permit fully automatic loading. It can accommodate crankshafts up to 1.0 meters.

According to the company's press release, the gage is capable of measuring diameters, roundness, cylindricity, timing angle and stroke with sub-micron accuracy. The in-line gage utilizes individual followers for the rod and main journals and can achieve less than one-minute cycle times.

Designed for operating two or three shifts per day, the crankshaft has a granite surface and incorporates the latest technology in controls, air bearings and optical linear scales for end-of-line accuracy. Utilizing a touch-screen and Windows XP user interface, specific programs are offered to provide routines that can measure, calculate and output printed as well as plotted results. The Adcole Model 1300 High-Speed Crankshaft Gage is priced from $495,000, depending upon configuration. Literature is available upon request.

For more information:

ADCOLE Corporation
J. Brooks Reece, vice president
669 Forest St.
Marlborough, MA 01752-3067
Phone: (508) 485-9100
Fax: (508) 481-6142
E-mail: *breece@adcole.com*
Internet: *www.adcole.com*

KAPP KX 500 FLEX

ADDS MULTI-STATION TURNTABLE AND TWIN-SPINDLE DRESSER

The KAPP Group has introduced the KX 500 FLEX Gear Center. Like other KX models, the KX 500 FLEX offers continuous generating grinding, discontinuous profile grinding, or a combination of both methods.

But the KX 500 FLEX features a rotating circular table that incorporates the tailstock support. The profile dressing unit is rotated into dressing position at the work spindle by the circular table and is flexible enough to use either conventional dressable worms or

continued

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profile grinding wheels. The unit also accommodates a single- or twin-spindle dresser. Depending on the applied dresser, different technologies can be used for the dressing process.

The KX 500 FLEX Gear Center grinds external spur and helical gears of modules up to 10 mm (0.4 in) with an outside diameter of maximum 500 mm (19.7 in) and a gear width of up to 520 mm (20.5 in). The machine offers the diverse tooling concepts of dressable ceramic tools for prototype machining and grinding of medium- to high-volume series, and non-dressable tools for manufacturing medium- and high-volume series. It also grinds more challenging gear geometry by using smaller tools.

The position of the workpiece spindle on the rotary table makes it possible for the KX 500 FLEX to be conveniently loaded by hand or by an automatic unit. For hand loading, the table swivels the part to the operator's door. For automatic loading, it is shifted laterally 90 degrees. A detailed configuration is created for each specific application. Automation options include a standardized combination of pallet conveyer and gantry loader, or a robot system.

The KX 500 FLEX uses a Siemens Sinumerik 840 D control system. A total of eight NC-axes perform the linear and rotary motions necessary for the machining process. The circuit table and tailstock are used as NC-axes, too. The part and tool are directly driven.

There is an optional measuring unit for measuring gear profile plus gear or tooth size over span. Additionally, a sample inspection of the gear quality can be sequenced to occur periodically during normal production runs.

For more information:

Pattea Carpenter
Kapp Technologies
2870 Wilderness Place
Boulder, CO 80301
Phone: (303) 447-1130
Fax: (303) 447-1131
E-mail: *info@kapp-niles.com*
Web: *www.kapp-niles.com*

ITW ROCOL LAUNCHES FOOD-GRADE LUBRICANT

ITW ROCOL recently announced

the launch of a food-grade lubricant called Accu-Lube FG-2000. According to the company's press release, the Accu-Lube FG-2000 is a natural/vegetable-based, environmentally safe metalworking oil recommended for use in food and medical applications requiring machining, drilling or tapping. It can also be utilized for the sawing of solids up to 6" in diameter as well as tubing almost any size. Since it's consumed in the cutting process, the need for clean-up and disposal is eliminated. Certified by the NSF, Accu-Lube is available in one- or fifty-five gallon containers and is safe on all metals. ITW ROCOL North America designs and manufactures a line of coolants and lubricants for the metalworking industry.

For more information:

Catherine Fuhr
Marketing Manager
ITW Rocol North America
3624 West Lake Avenue
Glenview, IL 60026
Phone: (847) 657-6185
E-mail:
cfuhr@rocolnorthamerica.com
Internet: *www.rocolnorthamerica.com*

NILES LAUNCHES ZP B HYBRID MACHINES

Built to meet the demands of large gear grinding, Niles recently launched the ZP B hybrid machine. The machine is designed for the complete finish of external spur gears—including gear teeth, bores and end faces—merging two technology machines together.

By combining internal cylindrical

continued

grinding and face grinding, the ZP B hybrid machine can perform all tasks in a single clamping. According to the company's press release, the machines come equipped with two dressing devices—a CNC dressing unit for profiling of corundum and sintered corundum grinding wheels and a second dresser for bore and face grinding.

The two-column design of the ZP B hybrid machine produces high rigidity, yet the modular design is flexible. The second column may be fitted with additional equipment and options.

For more information:
Pattea Carpenter
Kapp Technologies
2870 Wilderness Place
Boulder, CO 80301
Phone: (303) 447-1130
Fax: (303) 447-1131
E-mail: *info@kapp-niles.com*
Web: *www.kapp-niles.com*

Faro

UNVEILS ADVANCED PORTABLE COMPUTER-AIDED MEASUREMENT ARM

Faro Technologies Inc. recently announced the release of the Quantum FaroArm—the first eight-foot arm proven to measure accurately to within .0007", a fifth of the width of a human hair. According to company president and chief executive officer Jay Freeland, the FaroArm provides support for companies constrained by the limitations of fixed coordinate measuring machines and less portable models. The FaroArm has the capability to inspect and perform CAD-to-part analysis in any environment.

For more information:
Faro Technologies Inc.
125 Technology Park
Lake Mary, FL 32746
Phone: (800) 736-0234
Internet: *www.faro.com*

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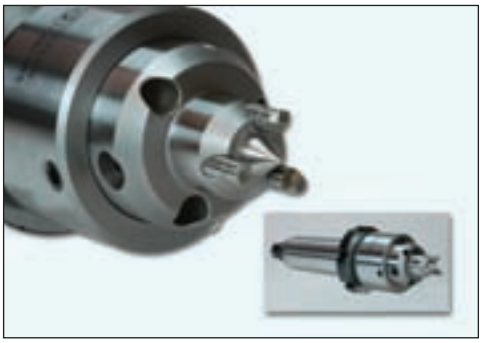
LMC Workholding

INTRODUCES NEIDLEIN FACE DRIVER

LMC Workholding recently unveiled its Neidlein FBS MK5 model face driver, designed specifically for hard tuning or grinding operations. These face drivers allow the entire O.D. of a workpiece to be machined in a single operation, providing increased productivity, quality and cost efficiency.



The FBS MK5 turns a workpiece end over end in a single operation, and eliminates drive dogs.

The mechanical face drivers are maintenance free and can allow for heavy and interrupted cuts, turn small and large parts and increase flexibility in faster cycle times.

For more information:

LMC Workholding
P.O. Box 7006
Logansport, IN 46947-7006
Phone: (574) 735-0225
Fax: (574) 722-6559
E-mail: *info@logan-mmk.com*
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Management Summary

In the post-World War II era, the major technology drivers for improving the life, reliability and performance of rolling-element bearings and gears have been the jet engine and the helicopter. By the late 1950s, most of the materials used for bearings and gears in the aerospace industry had been introduced into use. With improved manufacturing and processing, the potential improvement in bearing and gear life can be as much as 80 times that attainable in the early 1950s. This article summarizes the use of laboratory fatigue data for bearings and gears coupled with probabilistic life prediction and EHD theories to predict the life and reliability of a commercial turboprop gearbox. The resulting predictions are compared with field data.

Introduction

By the close of the 19th century, the bearing industry began to focus on sizing bearings for specific applications and determining bearing life and reliability. In 1896, R. Stribeck (Ref. 1) in Germany began fatigue testing full-scale bearings. J. Goodman (Ref. 2) in 1912 in Great Britain published formulas based on fatigue data that would compute safe loads on ball and cylindrical roller bearings. In 1914, the *American Machinists Handbook and Dictionary of Shop Terms* (Ref. 3) devoted six pages to rolling-element bearings, discussing bearing sizes and dimensions and recommending (maximum) loading and specified speeds. However, this publication did not address the issue of bearing life. During this time, it would appear that rolling-element bearing fatigue testing was the only way to determine or predict the minimum or average life of ball and roller bearings.

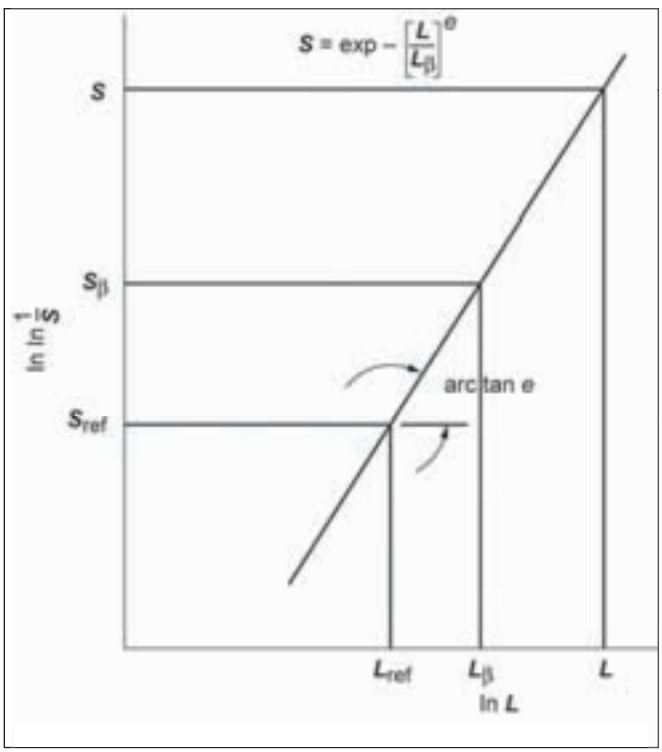

Figure 1—Weibull plot where (Weibull) slope of tangent or line is e. Probability of survival S_β of 36.8 percent, at which $L = L_\beta$ or $L/L_\beta = 1$.

In 1924, A. Palmgren (Ref. 4) in Sweden published a paper in German outlining his approach to bearing life prediction and presented an empirical formula based upon the concept of an L_{10} life, or life at which 90 percent of a population survives. During the next 20 years, he empirically refined his approach to bearing life prediction and matched his predictions to test data (Ref. 5). However, his formula lacked a theoretical basis or an analytical proof.

In 1939, W. Weibull (Refs. 6 and 7) in Sweden published his theory of failure. He was a contemporary of Palmgren and shared the results of his work with him. In 1947, Palmgren, in concert with G. Lundberg, also of Sweden, incorporated his previous work along with that of Weibull and what appears to be the work of H. Thomas and V. Hoersch (Ref. 8) in a probabilistic analysis to calculate rolling-element (ball and roller) life. This has become known as the Lundberg-Palmgren theory (Refs. 9 and 10). (In 1930, H. Thomas and V. Hoersch (Ref. 8) at the University of Illinois, Urbana, developed an analysis for determining subsurface principal stresses under Hertzian contact (Ref. 11). Lundberg and Palmgren (Refs. 9 and 10) do not reference the work of Thomas and Hoersch (Ref. 8) in their papers.)

The Lundberg and Palmgren life equations have been incorporated in both the International Organization for Standardization (ISO) and the American National Standards Institute (ANSI)/American Bearing Manufacturers Association (ABMA) standards for the load ratings and life of rolling-element bearings (Refs. 12–14), as well as in current bearing codes to predict life.

As mentioned, in the post-World War II era, the major technology drivers for improving the life, reliability and performance of rolling-element bearings and gears have been the jet engine and the helicopter. By the late 1950s, most of

the materials used for bearings and gears in the aerospace industry were introduced into use. By the early 1960s, the life of most steels was increased over that experienced in the early 1940s, primarily by the introduction of vacuum degassing and vacuum melting processes in the late 1950s (Ref. 15).

The development of elastohydrodynamic (EHD) lubrication theory in 1939 by A. Ertel (Ref. 16), and later by A. Grubin (Ref. 17) in 1949, showed that most rolling bearings and gears have a thin EHD film separating the contacting components. The life of these bearings and gears is a function of the thickness of the EHD film (Ref. 15).

Computer programs modeling bearing and gear dynamics that incorporate probabilistic life prediction methods and EHD theory enable the optimization of rotating machinery based on life and reliability. With improved manufacturing and material processing, the potential improvement in bearing and gear life can be as much as 80 times that attainable in the early 1950s (Ref. 15).

Between 1975 and 1981, Coy, Townsend and Zaretsky (Refs. 18–21) published a series of papers developing a methodology for calculating the life of spur and helical gears based upon the Lundberg-Palmgren theory and methodology for rolling-element bearings.

A probabilistic life model for planetary gear trains has been developed (Refs. 22–27). This model is based on the individual reliabilities of the gearbox bearings and gears based on classical rolling-element fatigue. The reliability of the gearbox system is treated as a strict series probability combination of the reliabilities of the gearbox components based on the Lundberg-Palmgren theory (Refs. 9 and 10). Each bearing and gear life was calculated, and the results were statistically combined to produce a system life for the total gearbox. The method was applied to a turboprop gearbox by Lewicki, et al. (Ref. 28).

The work presented in this report summarizes the use of laboratory fatigue data for bearings and gears coupled with probabilistic life prediction and EHD theories to: (1) predict the life and reliability of a commercial turboprop gearbox, and (2) compare the resulting prediction with field data.

Enabling Equations and Analysis

Weibull Analysis. In 1939, Weibull (Refs. 6 and 7) developed a method and equation for statistically evaluating the fracture strength of materials. He also applied the method and equation to fatigue data based upon small sample (population) sizes, where the two-parameter expression relating life and probability of survival is

$$\ln\ln\left(\frac{1}{S}\right) = e\ln\left(\frac{L}{L_0}\right) \quad \text{where } 0<L<\infty, 0<S<1 \tag{1}$$

When plotting the ln ln [1/S] as the ordinate against the ln L as the abscissa, fatigue data are assumed to plot as a straight line (Fig. 1). The ordinate ln ln [1/S] is graduated in statistical percent of components failed or removed for cause

Nomenclature

a	major semiaxis of contact ellipse, m (in.)
a_1	life adjustment factor for reliability
a_2	life adjustment factor for materials and processing
a_3	life adjustment factor for operating conditions including lubrication
B	gear material constant, N/m1.979 (lbf/in.1.979)
C_D	basic dynamic capacity of a ball or roller bearing, N (lbf)
C_t	basic dynamic capacity of gear tooth, N (lbf)
c	stress-life exponent
d	diameter of rolling element, m (in.)
e	Weibull slope; exponent
F_t	normal tooth load, N (lbf)
f	tooth face width, m (in.)
f_{cm}	bearing geometry and material coefficient
h	elastohydrodynamic (EHD) lubricant film thickness, m (in.); exponent
i	number of rows of rolling elements
k	gear tooth stress cycles per input shaft revolutions
L	life, hr, stress cycles, or revolutions
$L_{ß}$	characteristic life or life at which 63.2 percent of population fails, hr, stress cycles, or revolutions
L_{10}	10-percent life or life at which 90 percent of a population survives, hr, stress cycles, or revolutions
ℓ	length of stressed track, m (in.)
L	roller length, m (in.)
N	number of gear teeth
n	life, stress cycles
P_{eq}	equivalent bearing load, m (in.)
p	load-life exponent
r	pitch circle radius of gear, m (in.)
S	probability of survival, fractional percent
V	stressed volume, m^3
X_n	fraction of time spent at load-speed condition n
Z	number of rolling elements per row
z	depth beneath the surface of maximum orthogonal or maximum shear stress, m (in.)
α	contact angle, deg
η_{10t}	L_{10} life of single gear tooth, stress cycles
Λ	lubrication film parameter $h/$ (Eq. 14)
ρ	curvature sum, m^{-1} ($in.^{-1}$)
σ	composite surface roughness, rms, m (in.)
σ_1, σ_2	surface roughness of bodies 1 and 2, rms, m (in.)
τ	maximum orthogonal or maximum shear stress, Pa (psi)
φ	gear pressure angle, deg

Subscripts

1, 2	bodies 1 or 2; load-life condition 1, 2, etc.
B	bearing
G	gear
n	body n or load-life condition n
ref	define
s, *sys*	system
t	tooth

as a function of ln L, the log of the time or cycles to failure. The tangent of the line is designated the Weibull slope e, which is indicative of the shape of the cumulative distribution or the amount of scatter of the data.

The method of using the Weibull distribution function for data analysis for determining component life and reliability was later developed and refined by Johnson (Ref. 29).

Bearing Life Analysis. Lundberg and Palmgren (Refs. 9 and 10) extended the theoretical work of Weibull (Refs. 6 and 7) and showed that the probability of survival S could be expressed as a power function of shear stress τ, life n, depth of maximum shear stress z, and stressed volume V:

$$\ln\frac{1}{S}\sim\frac{\tau^c n^e}{z^h}V \tag{2}$$

$$\ln\frac{1}{S}\sim\frac{\tau^c n^e al}{z^{h-1}} \tag{3}$$

By substituting the bearing geometry and the Hertzian contact stresses for a given load into Equation 3, the bearing basic dynamic load capacity C_D can be calculated (Ref. 9). The basic dynamic load capacity C_D is defined as the load that a bearing can carry for a life of one-million inner-race revolutions with a 90-percent probability of survival (L_{10} life). Lundberg and Palmgren (Ref. 9) obtained the following additional relation:

$$L_{10}=\left(\frac{C_D}{P_{eq}}\right)^p \tag{4}$$

where P_{eq} is the equivalent bearing load and p is the load-life exponent.

Formulas for the basic dynamic load ratings derived by Lundberg and Palmgren (Refs. 9 and 10) and incorporated in the ANSI/ABMA and ISO standards (Refs. 12–14) are as follows:

Radial ball bearings with $d \leq 25$ mm:

$$C_D=f_{cm}(i\cos\alpha)^{0.7}Z^{2/3}d^{1.8} \tag{5}$$

Radial ball bearings >25 mm:

$$C_D=f_{cm}(i\cos\alpha)^{0.7}Z^{2/3}d^{1.4} \tag{6}$$

Radial roller bearings:

$$C_D=f_{cm}(il\cos\alpha)^{7/9}Z^{3/4}d^{29/27} \tag{7}$$

Equation 4 can be modified using life factors based on reliability a_1, materials and processing a_2, and operating conditions such as lubrication a_3 (Refs. 15 and 30) where

$$L=a_1a_2a_3L_{10} \tag{8}$$

Gear Life Analysis. Between 1975 and 1981, Coy, Townsend and Zaretsky (Refs. 18–21) published a series of papers developing a methodology for calculating the life of spur and helical gears based upon the Lundberg-Palmgren theory and methodology for rolling-element bearings. Townsend, Coy and Zaretsky (Ref. 31) reported that for AISI 9310 spur gears, the Weibull slope e is 2.5. Based on Equation 2, for all gears except a planet gear, the gear life can be written as

$$L_{10_G}=\frac{N^{-1/e_G}\eta_{10_t}}{k} \tag{9}$$

For a planet gear, the life is

$$L_{10_G}=\frac{N^{-1/e_G}\left(\eta_{10_{t1}}^{-e_G}+\eta_{10_{t2}}^{-e_G}\right)^{-1/e_G}}{k} \tag{10}$$

The L_{10} life of a single gear tooth can be written as

$$\eta_{10_t}=a_2a_3\left(\frac{C_t}{F_t}\right)^{p_G} \tag{11}$$

where

$$C_t=Bf^{0.907}\rho^{-1.165}J^{-0.093} \tag{12}$$

and

$$\rho=\left(\frac{1}{r_1}+\frac{1}{r_2}\right)\frac{1}{\sin\varphi} \tag{13}$$

and η_{10t} is the L_{10} life in millions of stress cycles for one particular gear tooth. This number can be determined by using Equation 11, where C_t is the basic load capacity of the gear tooth; P_t is the normal tooth load; p_G is the load-life exponent usually taken as 4.3 for gears based on experimental data for AISI 9310 steel; and a_2 and a_3 are life adjustment factors similar to that for rolling-element bearings (Table 1). The value for C_t can be determined by using Equation 12, where B is a material constant that is based on experimental data and is approximately equal to 1.39×10^8 when calculating C_t in SI units (newtons and meters), and 21,800 in English units (pounds and inches) for AISI 9310 steel spur gears; f is the tooth width; and ρ is the curvature sum at the start of single-tooth contact.

Life factors a_2 for materials and processing are determined experimentally. Table 2 shows representative life factors obtained from surface fatigue testing of spur gears by NASA (Refs. 15 and 30).

The L_{10G} life of the gear (all teeth) in millions of input shaft revolutions at which 90 percent will survive can be determined from Equation 9 or Equation 10 where N is the total number of teeth on the gear; e_G is the Weibull slope for the gear and is assumed to be 2.5 (from Ref. 31); and k is the number of load (stress) cycles on a gear tooth per input shaft

revolution.

For all gears except the planet gears, each tooth will see load on only one side of its face for a given direction of input shaft rotation. However, each tooth on a planet gear will see contact on both sides of its face for a given direction of input shaft rotation. One side of its face will contact a tooth on the sun gear, and the other side of its face will contact a tooth on the ring gear. Equation 10 takes this into account, where η_{10r1} is the L_{10} life in millions of stress cycles of a planet tooth meshing with the sun gear, and η_{10r2} is the L_{10} life in millions of stress cycles of a planet tooth meshing with the ring gear.

Elastohydrodynamic Lubrication. An important parameter to consider when designing and operating rolling bearings and gears is the elastohydrodynamic (EHD) lubricant film thickness that forms between heavily loaded contacting bodies. Ertel (Ref. 16) and Grubin (Ref. 17) are credited with the first useful solution. A summary of EHD film thickness calculations can be found in Reference 15.

The life of a rolling bearing or gear is a function of a lubrication film parameter Λ where

$$\Lambda = \frac{h}{\sigma} \tag{14}$$

and

$$\sigma = \left(\sigma_1^2 + \sigma_2^2\right)^{1/2} \tag{15}$$

The lubricant film parameter Λ can be used as an indicator of bearing and gear performance and life. For $\Lambda < 1$, surface smearing or deformation accompanied by wear will occur on the rolling surfaces. For $1 < \Lambda < 1.5$, surface distress may be accompanied by superficial surface pitting. For $1.5 < \Lambda < 3$, some surface glazing can occur with eventual failure caused by classical subsurface-origin, rolling-element fatigue. At $\Lambda \geq 3$, minimal wear can be expected with extremely long life, and failure will eventually be by classical subsurface-origin, rolling-element fatigue.

The most expedient way of attaining a higher Λ ratio is to reduce the bearing or gear operating temperature and thus increase the lubricant viscosity. Another way is to select a lubricant with a higher viscosity at operating temperature, a larger pressure-viscosity coefficient, or both. The most expensive way of attaining a higher Λ ratio is to select a high-quality surface finish on bearings and gears (Refs. 15 and 30). The effect of film thickness on bearing life is shown in Figure 2. The life factor (LF) obtained from this figure is used to modify or adjust the calculated lives of bearings and gears (Refs. 15 and 30). This constitutes the life factor a_3 in Equations 8 and 11.

System Life Prediction. The L_{10} lives of the individual bearings and gears that make up a rotating machine are calculated for each condition of their operating profiles. For each component, the resulting lives from each of the operating conditions are combined using the linear damage

TABLE 1. RERESENTATIVE VARIABLES AFFECTING BEARING LIFE AND RELIABILITY [From ref. 30]

Life adjustment factor	Variable
Reliability, a_1	Probability of failure
Materials and processing, a_2	Bearing steel Material hardness Residual stress Melting process Metal working
Operating conditions, a_3	Load Misalignment Housing clearance Axially loaded cylindrical bearings Rotordynamics Hoop stresses Speed Temperature Steel Lubrication Lubricant film thickness Surface finish Water Oil Filtration

TABLE 2. RELATIVE SURFACE PITTING FATIGUE LIFE FOR VAR AISI 9310 STEEL AND AIRCRAFT-QUALITY GEAR STEEL (ROCKWELL C 59 to 62) [From refs. 15 and 30]

Steel [a]	10-percent relative life, L_{10}
VAR AISI 9310	1.0
VAR AISI 9310 (shot peened)	1.6
VIM-VAR AISI 9310	2.5
VAR Carpenter EX–53	2.1
CEVM CBS 600	1.4
VAR CBS 1000	2.1
CEVM Vasco X–2	2.0
CEVM Super Nitra	1.3
VIM–VAR AISI M–50 (forged)	3.2
VIM–VAR AISI M–50 (ausformed)	2.4
VIM–VAR M50 NiLlloy (5Ni-2A1)	11.5

[a] VAR, vacuum arc remelting; CEVM, consumable-electrode vacuum remelting; VIM–VAR, vacuum induction melting – vacuum arc remelting.

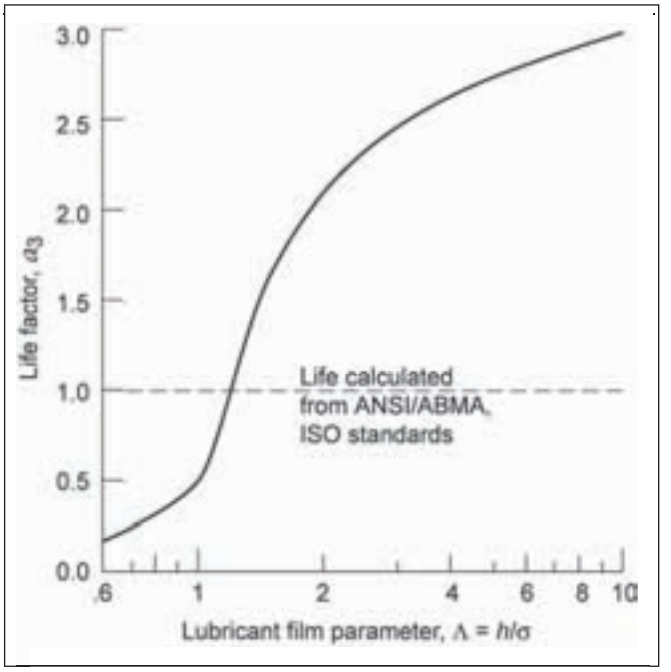

Figure 2—Life factor a_3 as function of lubricant film parameter Λ (Ref. 15).

(Palmgren-Langer-Miner) rule (Refs. 4, 32 and 33) where

$$\frac{1}{L} = \frac{X_1}{L_1} + \frac{X_2}{L_2} + \cdots \frac{X_n}{L_n} \quad (16)$$

The cumulative lives of each of the machine components are combined to determine the calculated machine system L_{10} life using the Lundberg-Palmgren formula (Ref. 9):

$$\frac{1}{L_{10}^{e}} = \left(\frac{1}{L_{B_1}^{eB}} + \frac{1}{L_{B_2}^{eB}} + \cdots \frac{1}{L_{B_n}^{eB}} \right) + \left(\frac{1}{L_{G_1}^{eG}} + \frac{1}{L_{G_2}^{eG}} + \cdots \frac{1}{L_{G_n}^{eG}} \right) \quad (17)$$

The calculated system life is dependent on the resultant value of the system Weibull slope *e*. This value is normally not known with absolute certainty and is usually assumed to be the same as that of the shortest-lived component in the system.

Results and Discussion

Predicted Life of a Turboprop Gearbox. A commercial turboprop gearbox used for this analysis is shown in Figure 3. It consists of two stages with a single-mesh spur reduction followed by a 5-planet planetary gearbox comprising 11 rolling-element bearings and 9 spur gears (Ref. 28). The first stage consists of the input pinion gear meshing with the main drive gear. The second stage is provided by the fixed-ring planetary driven by a floating sun gear as input with a five-planet carrier as output. The input pinion speed is constant at 13,820 rpm, producing a carrier output speed of 1,021 rpm.

The operational profile includes loads for takeoff, climb, cruise and descent. The cruise segment of the profile consumes 68 percent of the flight time with a little less than half of the power required for the takeoff, which lasts for less than 3 percent of the flight time.

The cause for removal can be assumed to be that one or more bearings or gears had fatigue or damage resulting in wear and/or vibration detected by magnetic chip detectors and/or vibration pickups. The gearboxes are removed from service before secondary damage occurs. The removed gearbox is inspected and the failed part or parts are replaced. The gearbox is then put back into service. Individual occurrences are not predictable but are probabilistic. No two gearboxes run under the same conditions fail necessarily from the same cause and/or at the same time. At a given probability of survival, the life of the gearbox will always be less than the shortest-lived element in the gearbox.

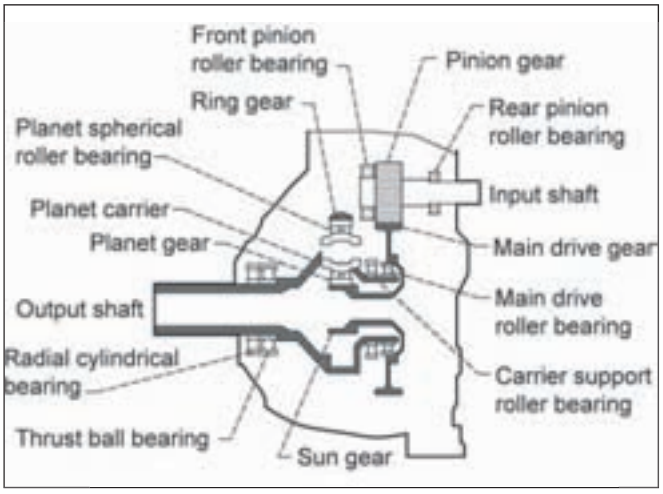

Figure 3—Commercial turboprop gearbox.

Using Equations 4–8 for bearings and Equations 9–13 for gears and appropriate computer programs incorporating these equations, the lives of each of the bearings and gears making up the gearbox were calculated for each of their operating conditions. These lives are shown as the Weibull plots in Figure 4.

The L_{10} life of a single double-row, spherical bearing is 3,529 hr. From Equation 14, the system L_{10} life for the five-bearing planetary set is 774 hr. For all the bearings in the gearbox, the bearing system L_{10} life is also predicted to be 774 hr.

Using Equation 17 for the individual gears, the gear system predicted L_{10} life is 16,680 hr. Combining the bearing and gear lives to obtain a gearbox L_{10} life, again using Equation 17, the predicated L_{10} life for the gearbox is 774 hr. The lives of the individual bearings, and more specifically, those of the planet double-row spherical bearings, determine the life of the gearbox in this example. The system lives of the bearings, gears and gearbox are summarized in Figure 4c.

Gearbox Field Data. The application of the Lundberg-Palmgren theory (Ref. 9) to predict gearbox life and reliability needs to be benchmarked and verified under a varied load and operating profile. The cost and time to laboratory test a statistically significant number of gearboxes to determine their life and reliability is prohibitive. A practical solution to this problem is to benchmark the analysis to field data. Fortunately, these data were available for the commercial turboprop gearbox used in this study.

No two gearboxes are expected to operate in exactly the same manner. Flight variables include operating temperature and load. Small variations in operational load can result in significant changes in life. Hence, the accuracy of our calculations is dependent on how close the defined mission profile is to actual flight operation.

The gearboxes are condition-monitored and are removed from service on the detection of a perceived component failure. At the time of removal, the gearboxes are functional. The removal precludes secondary damage. That is, the damage is limited to the failed component.

Field data were collected for 64 new commercial turboprop gearboxes. From these field data, the resultant time to removal of each gearbox is plotted in the Weibull plot of Figure 5. For these data, there was not a breakdown of the cause for removal or the percent of each component that had failed. The resultant L_{10} life from the field data was 5,627 hr., and the Weibull slope *e* was 2.189. Using the Lundberg-Palmgren method (above), the predicted L_{10} life was 774 hr and the Weibull slope *e* was 1.125. The field data suggest

that the life of the gearbox was underpredicted by a factor of 7.56.

Although errors in the assumed operating profile of the gearbox may account for the difference between actual and predicted life, it is suggested that using the Lundberg-Palmgren equations results in a life prediction that is too low for the bearings.

Referring to Equation 4, in their 1952 publication (Ref. 10), Lundberg and Palmgren calculate a load-life exponent p equal to 10/3 for roller bearings, where one raceway has point contact and the other raceway has line contact. The 10/3 load-life exponent has been incorporated in the ANSI/ABMA/ISO standards first published in 1953 (Refs. 12–14). Their assumption of point and line contact may have been correct for many types of roller bearings then in use. However, it is no longer the case for most roller bearings manufactured today and, most certainly, for cylindrical roller bearings. Experience and the analysis suggest that the 10/3 load-life exponent p for roller bearings is incorrect and underpredicts roller bearing life (Ref. 34).

The work of Poplawski, Peters and Zaretsky (Ref. 34) suggests that p for roller bearings is equal to or greater than 4 but is less than 5. This premise can be easily tested based on the data for the turboprop gearbox.

From Equation 17, assuming that the bearing system has the same Weibull slope as that of the gearbox (e = 2.189),

$$\frac{1}{L_{GS}^{e_b}} = \frac{1}{L_B^{e_B}} + \frac{1}{L_G^{e_G}} \tag{18a}$$

$$\frac{1}{(5627)^{2.189}} = \frac{1}{L_B^{2.189}} + \frac{1}{(16\,680)^{2.5}} \tag{18b}$$

From equation (18b), the actual bearing system life is

$$L_B = 5627 \text{ hr} \tag{18c}$$

From Lundberg-Palmgren (Ref. 9), the predicted bearing system life is

$$L_B \sim \left(\frac{C_D}{P_{eq}}\right)^4 \sim 774 \text{ hr} \tag{19a}$$

Then,

$$\left(\frac{C_D}{P_{eq}}\right) \sim 5.27 \tag{19b}$$

Calculating a revised value for the load-life exponent p for the gearbox bearings based on the actual bearing system life of 5,627 hr. (Eq. 18c),

$$\left(\frac{C_D}{P_{eq}}\right)^p \sim (5.27)^p \sim 5627 \text{ hr} \tag{20a}$$

Solving for load-life exponent p,

$$p = 5.2 \tag{20b}$$

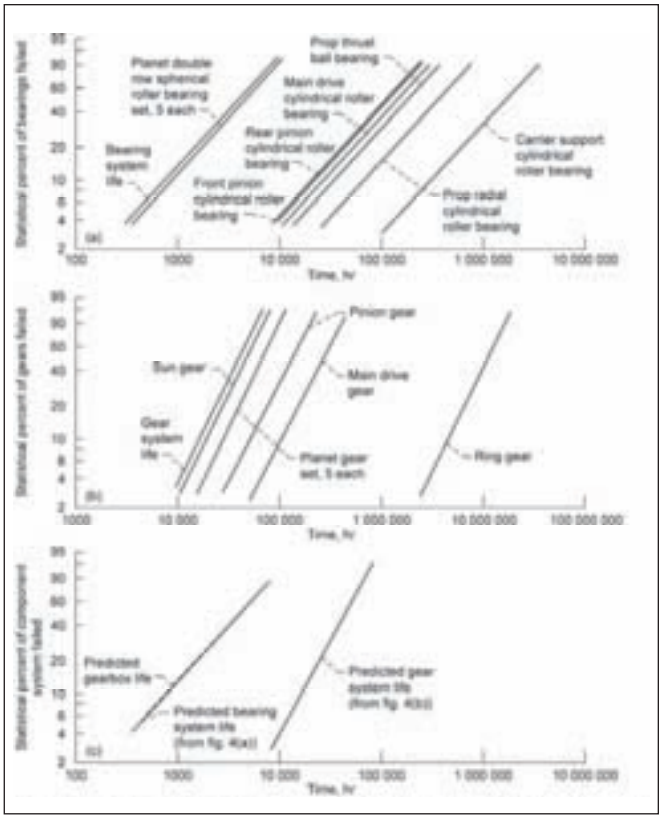

Figure 4—Predicted lives for commercial turboprop gearbox and its respective bearing and gear components using Lundberg-Palmgren life theory: (a) bearing component lives; (b) gear component lives; (c) gearbox life and component lives.

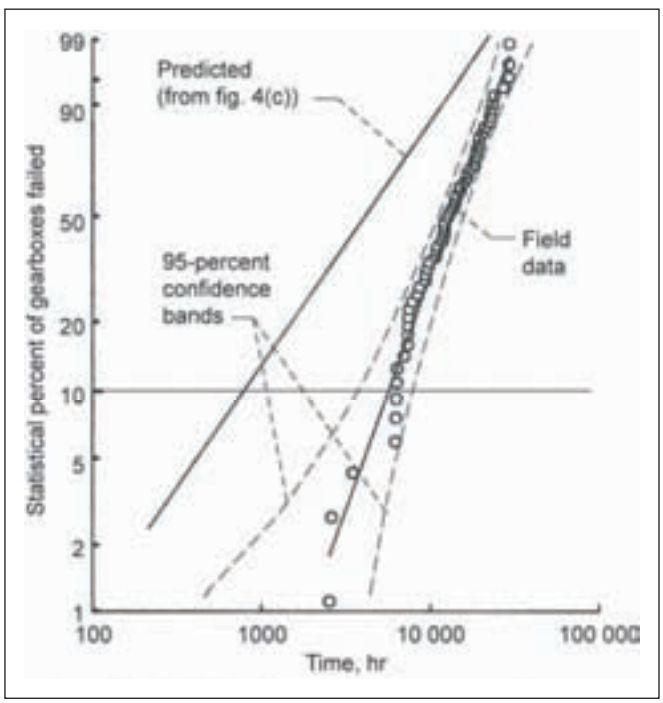

Figure 5—Field data for lives of turboprop gearboxes compared with predicted lives using Lundberg-Palmgren life theory. Failure index: 59 out of 64.

where according to Poplawski, Peters, and Zaretsky (Ref. 34),

$$4 \leq p \geq 5 \qquad (20c)$$

Were the bearing lives to be recalculated with a load-life exponent p equal to 5.2, the predicted L_{10} life of the gearbox would be equal to the actual life obtained in the field.

Summary of Results

Laboratory fatigue data for rolling-element bearings and gears coupled with probabilistic life prediction and elastohydrodynamic analysis were used to predict the life and reliability of a commercial turboprop gearbox. These data were compared with field data. The following results were obtained:

1. Using Lundberg-Palmgren theory, the predicted gearbox L_{10} life was less than that obtained in the field. The predicted gearbox L_{10} life was 774 hr. whereas the actual L_{10} life was 5,627 hr.
2. The gearbox life was dependent on the system life of the rolling-element bearings in the gearbox. Changing the load-life exponent p for the roller bearings from 4 to 5.2 would give a predicted gearbox L_{10} life of 5,627 hr.
3. The ANSI/ABMA/ISO standards using the unmodified Lundberg-Palmgren theory underpredict rolling-ele ment bearing life, resulting in conservative values for the gearbox life prediction.

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Optimal Choice of the Shaft Angle for Involute Gear Hobbing

Carlo Innocenti

Management Summary

With reference to the machining of an involute spur or helical gear by the hobbing process, this paper suggests a new criterion for selecting the position of the hob axis relative to the gear axis. By adhering to the proposed criterion, the hob axis is set at the minimum distance from the gear axis, thus maximizing the depth of the tooth spaces of the gear. The new criterion is operatively implemented by solving a univariate equation, which stems from a new, synthetic analysis of the meshing of crossed-axis, involute gears. A numerical example shows application of the suggested procedure to a case study and compares the optimal hob setting to the customary one.

Introduction

Hobbing of both spur and helical gears is generally done by setting the axes of the gear and the hob at an angle that is the algebraic sum of the pitch helix angles of gear and hob (Refs.1–2). Such a standard way of determining the shaft angle—although conducive to satisfactory results—does not rely on a convincing rationale. Suffice it to say that any referral to pitch helix angles is questionable because the meshing of a hob with the gear being machined does not involve any pure rolling of a pitch cylinder on another pitch cylinder (as would be the case, instead, for the meshing of two gears mounted on parallel-axis shafts).

The possibility of choosing the setting angle of the hob cutter in a non-standard way is mentioned in Reference 3, together with the related implications on the tooth thickness of the hobbed gear for a given gear hob cutting distance. Nevertheless, the technical literature does not seem to have explored this hint, and even more recent contributions on the hobbing process—see, for instance, References 4 and 5—do not question the standard choice of the hob setting angle as the sum of the gear pitch helix angle and of an angle that characterizes the hob.

This paper first revises the kinematics of meshing two crossed-axis, involute helical gears (Refs. 3 and 6), and presents an original, concise relationship for determining the meshing backlash in terms of the gear dimensions, shaft axis distance and shaft axis angle.

Subsequently, the paper narrows the analysis down to the meshing of a gear with a hob. By considering a zero-meshing backlash, the optimal shaft angle for hobbing is determined as the value of the shaft angle that minimizes the shaft axis distance. By adopting this criterion, the depth of the tooth spaces of the gear is maximized, which could be favorable for the contact ratio of a gear pair (undercutting issues are beyond the scope of this work). The paper shows that the value for the optimal shaft angle stems directly from numerically solving a univariate equation.

Embracing the presented method gen-

erally leads to shaft angles for hobbing that are very close to those determined by the standard procedure. Even so, the paper highlights the arbitrariness and limitations of the standard procedure for the selection of the hobbing shaft angle. Moreover, it makes available a consistent procedure that is easily employable, despite being more involved than the standard one.

Due to the similarities between the kinematics of the two manufacturing processes, the results reported in the paper for gear hobbing are also applicable to gear grinding when carried out by a threaded grinding wheel.

A numerical example shows application of the presented procedure in a case study and compares the new results to those obtainable by the standard, albeit less-than-optimal procedure.

Contact Between Involute Helicoids

This section and the next one reformulate the basic equations that are instrumental in the analysis of the meshing of a pair of involute helical gears mounted on crossed-axis shafts. Because they involve only the elemental geometric parameters of the gears in mesh, the presented formulas are simpler than those reported in the technical literature, and thus more suited to be algebraically manipulated in the pursuit of this paper's scope. Some of the equations reported in this section stem from specialization of formulas traceable in Reference 7.

The fundamental geometric parameters of an involute helical gear are the radius ρ of the base cylinder, the base helix angle β ($-\pi/2 < \beta < \pi/2$ radians), the number of teeth N, and the angular base thickness φ of a tooth. Aside from N, all of these parameters are shown in Figure 1 with reference to a tooth of a helical gear. (In Figure 1, the involute helicoids are shown as emerging from the base cylinder, irrespective of the actual extent of the tooth flanks. Furthermore, angle β in Figure 1 has to be considered as positive because the base helix angle is right-handed.) The normal base pitch ρ of the gear is the distance between involute helicoids of homologous flanks of adjacent teeth. It is provided by (Ref. 3):

$$p = \frac{2\pi\rho\cos\beta}{N} \qquad (1)$$

As soon as the axis of the gear is directed in either way by a unit vector **n**, a tooth flank is a left-hand flank or a right-hand flank according to whether the following quantity is negative or, respectively, positive, as in:

$$\boldsymbol{q}\times(P-O)\cdot\boldsymbol{n} \qquad (2)$$

In Equation 2, (P–O) is the vector from a point O on the gear axis to a point P on the tooth flank, whereas **q** is the outward pointing unit vector orthogonal to the tooth flank at point P.

Two meshing helical involute gears—from here on known as Gear 1 and Gear 2—are now considered. As is known, in order for the gears to mesh, they must have the same normal base pitch. This condition translates into the following equation (see Eq. 1):

$$\frac{N_1}{N_2} = \frac{\rho_1 u_1}{\rho_2 u_2} \qquad (3)$$

where quantities u_i (i = 1,2) are defined by:

$$u_i = \cos\beta_i \qquad (i = 1,2) \qquad (4)$$

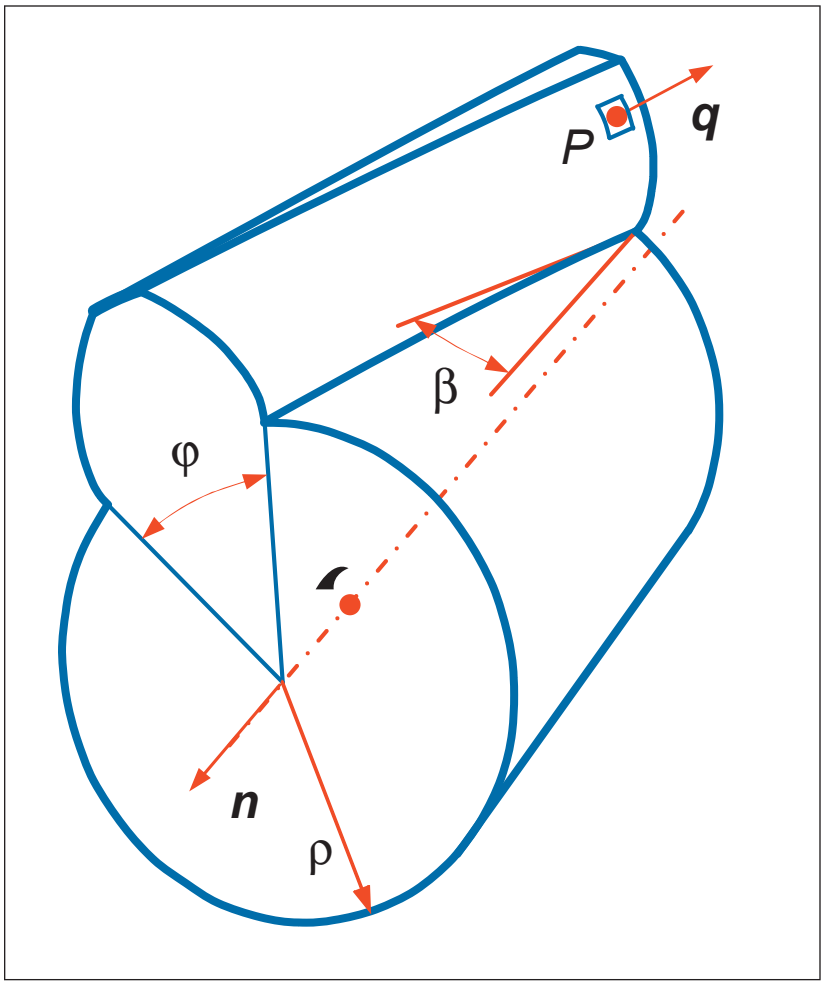

Figure 1—Basic geometric parameters of a helical involute gear.

With reference to Figure 2, the distance between the skew gear axes is denoted by a_0. As soon as the axis of Gear 1 is directed in either way by unit vector $\mathbf{n}_1$, unit vector $\mathbf{n}_2$ is so directed as to make a left-hand flank of a tooth of Gear 1 contact a left-hand flank of a tooth of Gear 2. This also implies that if the angular velocity vectors of Gear 1 are positive with respect to $\mathbf{n}_1$, the angular velocity of Gear 2 is negative with respect to $\mathbf{n}_2$).

The common perpendicular to the gear axes intersects the axes themselves at points A_1 and A_2. A fixed reference frame W_1 is now introduced with origin at A_1, x-axis oriented towards A_2, and z- axis parallel to unit vector $\mathbf{n}_1$, with the same direction as $\mathbf{n}_1$. Similarly, another fixed reference frame, W_2, is introduced with origin at A_2, x-axis oriented towards A_1, and z-axis parallel to unit vector $\mathbf{n}_2$, with the same direction. The angle α_0 between the gear axes is defined as the rotation about the x-axis of reference frame W_1 that would make $\mathbf{n}_1$ parallel to $\mathbf{n}_2$. The 4×4 matrix M_0 for transformation of coordinates from W_2 to W_1 is:

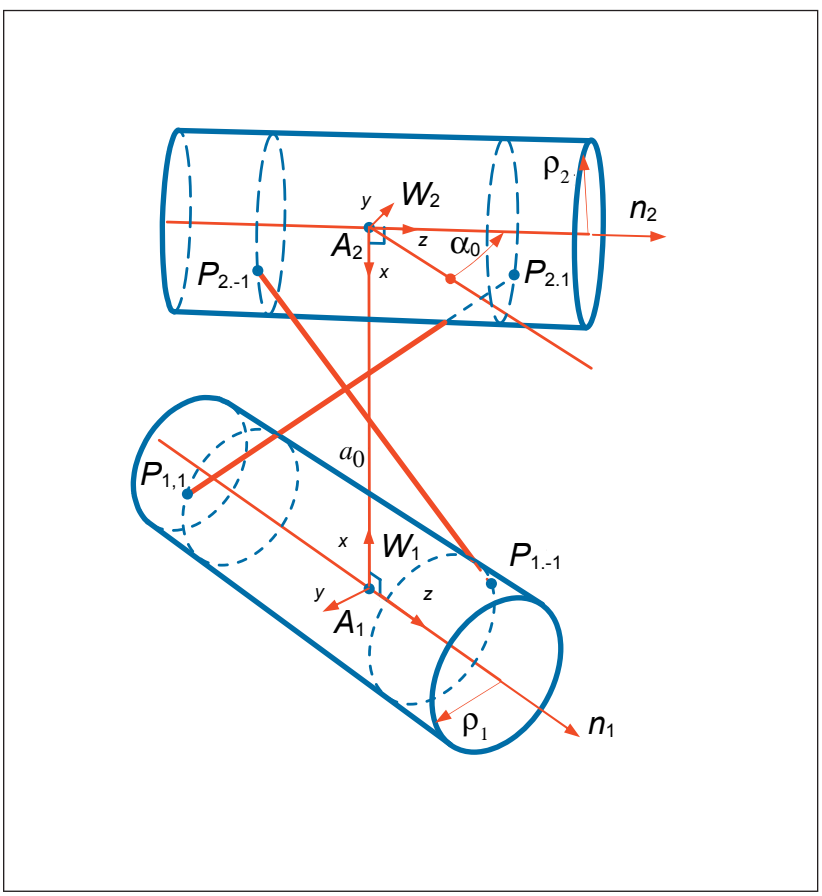

Figure 2—Paths of contact of a crossed-axis helical gearing.

$$M_{i,1} = \begin{bmatrix} c_{i,1} & -s_{i,1} & 0 & 0 \\ s_{i,1} & c_{i,1} & 0 & 0 \\ 0 & 0 & 1 & b_{i,1} \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (5)$$

where:

$$u_i = \cos\beta_i \qquad (i = 1,2) \quad (6)$$

The point of contact between a right-hand flank of Gear 1 and a right-hand flank of Gear 2 is bound to lie on a straight line that is tangent to the base cylinders of the two gears at points $P_{1,1}$ and $P_{2,1}$. (In this two-index notation, the first index refers to the gear and the second index to the tooth flank—1 for a right-hand flank and –1 for a left-hand flank). The line segment $P_{1,1}$ and $P_{2,1}$ is the path of contact for right-hand flanks.

To determine points $P_{1,1}$ and $P_{2,1}$, together with their mutual distance σ_1, two auxiliary reference frames—$V_{1,1}$ and $V_{2,1}$—are introduced. The origin $B_{i,1}$ of $V_{i,1}$ (i =1, 2) is on the axis of gear i, at the transverse section for gear i that contains point $P_{i,1}$. The z-axis of $V_{i,1}$ has the same orientation and direction as the z-axis of W_i, whereas the x-axis of $V_{i,1}$ is oriented from $B_{i,1}$ to $P_{i,1}$ (Fig. 3). If $\theta_{i,1}$ is the angle of the rotation about the z-axis of W_i, that would make the axes of W_i parallel to the axes of $V_{i,1}$, the 4×4 matrix $M_{i,1}$ for transformation of coordinates from $V_{i,1}$ to W_i is given bv:

$$M_{i,1} = \begin{bmatrix} c_{i,1} & -s_{i,1} & 0 & 0 \\ s_{i,1} & c_{i,1} & 0 & 0 \\ 0 & 0 & 1 & b_{i,1} \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (7)$$

where:

$$c_{i,1} = \cos\theta_{i,1}, \qquad s_{i,1} = \sin\theta_{i,1} \quad (8)$$

and $b_{i,1}$ is the z-coordinate of point $B_{i,1}$ in reference frame W_i.

The homogeneous components in $V_{i,1}$ of the unit vector $\mathbf{e}_{i,1}$ of the contact path $P_{1,1}$ $P_{2,1}$, directed from $P_{i,1}$ to the other extremity of the contact path, is provided by:

$$\mathbf{e}_{i,1}\big|_{V_{i,1}} = [0 \quad -u_i \quad v_i \quad 0]^T \quad (9)$$

where u_i is defined by Equation 4, while v_i is given by:

$$v_i = \sin\beta_i \qquad (i = 1,2) \tag{10}$$

Similarly, the homogeneous coordinates of point $P_{i,1}$ with respect to $V_{i,1}$ are:

$$P_{i,1}\big|_{V_{i,1}} = [\rho_i \quad 0 \quad 0 \quad 1]^T \tag{11}$$

The ensuing vector conditions:

$$\mathbf{e}_{1,1} + \mathbf{e}_{2,1} = 0 \tag{12}$$

$$(P_{2,1} - P_{1,1}) = \sigma_1 \mathbf{e}_{1,1} \tag{13}$$

are conducive to determination of unknowns $\theta_{1,1}$, $\theta_{2,1}$, $b_{1,1}$, $b_{2,1}$, and σ_1. Specifically, Equation 12 imposes the parallelism of the unit vectors $\mathbf{e}_{1,1}$ and $\mathbf{e}_{2,1}$ normal to the right-hand tooth flanks of Gears 1 and 2 at points $P_{1,1}$ and $P_{2,1}$ respectively, whereas Equation 13 calls for unit vector $\mathbf{e}_{1,1}$ to be parallel to contact path $P_{1,1}P_{2,1}$.

To solve Equations 12 and 13, all vectors can be expressed through their components in reference frame W_1. This implies left-multiplying $\mathbf{e}_{1,1}|_{V1,1}$ and $P_{1,1}\,|\,v_{1,1}$ by matrix $M_{1,1}$, and $\mathbf{e}_{2,1}|_{V2,1}$ and $P_{2,1}|\,v^{2,1}$ by matrix $M_0M_{2,1}$. If the gear axes are not parallel—i.e., $v_0 \neq 0$—the last two components of Equation 12 linearly provide the ensuing expressions for $c_{1,1}$ and $c_{2,1}$ as:

$$c_{1,1} = -\frac{v_2 + u_0 v_1}{v_0 u_1}; \qquad c_{2,1} = -\frac{v_1 + u_0 v_2}{v_0 u_2} \tag{14}$$

For a given value of the axis angle α_0—and regardless of the axis distance a_0—the two considered gears can mesh together only if Equation 14 yields cosines of real angles; i.e., only if the following inequality is satisfied:

$$Q \geq 0 \tag{15}$$

where:

$$Q = v_0^2 - v_1^2 - v_2^2 - 2u_0 v_1 v_2 \tag{16}$$

By supposing that Equation 15 holds, the first component of Equation 12 provides information on unknowns $s_{1,1}$ and $s_{2,1}$ as:

$$u_1 s_{1,1} = u_2 s_{2,1} \tag{17}$$

As quantities u_1 and u_2 are both positive—they are the cosines of angles lower in magnitude than $\pi/2$—Equation 17 ensures that $s_{1,1}$ and $s_{2,1}$ have the same sign. By also considering that $s_{1,1} = \pm\sqrt{1 - c_{1,1}^2}$, $(i=1,2)$, and taking advantage of Equation 14 as well, the ensuing expressions for $s_{1,1}$ and $s_{2,1}$ can be easily found as:

$$s_{1,1} = \lambda\frac{\sqrt{Q}}{v_0 u_1}; \qquad s_{2,1} = \lambda\frac{\sqrt{Q}}{v_0 u_2} \tag{18}$$

In Equation 18, quantity Q is provided by Equation 16, whereas λ is a yet-to-be determined integer whose value is +1 or –1.

An explicit expression of $\theta_{1,1}$ can be obtai ned through the following trigonometric identity:

$$\tan\frac{\theta_{1,1}}{2} = \frac{1 - c_{1,1}}{s_{1,1}} \tag{19}$$

With the aid of Equations 14 and 18, Equation 19 yields:

$$\theta_{1,1} = 2\lambda \arctan\frac{v_2 + u_0 v_1 + v_0 u_1}{\sqrt{Q}} \tag{20}$$

The expression of $\theta_{2,1}$ is likewise given by:

$$\theta_{2,1} = 2\lambda \arctan\frac{v_1 + u_0 v_2 + v_0 u_2}{\sqrt{Q}} \tag{21}$$

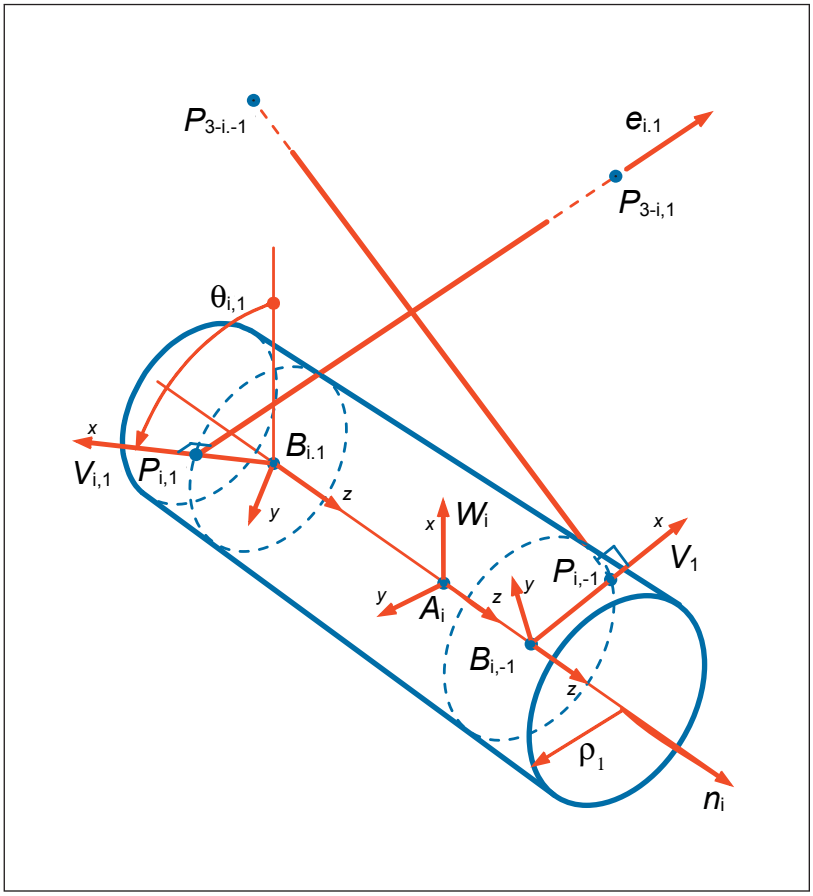

Figure 3—Auxiliary reference frames from gear i (i=1.2).

Equation 13 can now be linearly solved for unknowns $b_{1,1}$, $b_{2,1}$, and σ_1. Specifically, the expression for the length σ_1 of contact path $P_{1,1}$ and $P_{2,1}$ is

$$\sigma_1 = \lambda v_0 \frac{a_0 - \rho_1 c_{1,1} - \rho_2 c_{2,1}}{\sqrt{Q}} \quad (22)$$

Since σ_1 has to be positive, quantity λ is selected as follows:

$$\lambda = \begin{cases} -1 & \text{if} \quad v_0 \left(a_0 - \rho_1 c_{1,1} - \rho_2 c_{2,1} \right) < 0 \\ 1 & \text{if} \quad v_0 \left(a_0 - \rho_1 c_{1,1} - \rho_2 c_{2,1} \right) \geq 0 \end{cases} \quad (23)$$

Thanks to Equation 23, $s_{1,1}$ and $s_{2,1}$ can be determined by Equation 18. By also taking into account Equation 14, angles $\theta_{1,1}$ and $\theta_{2,1}$ can be unambiguously evaluated in the range $[-\pi, \pi]$ radians through Equations 20 and 21. The expressions for $b_{1,1}$ and $b_{2,1}$ stemming from Equation 13 are:

$$b_{1,1} = -\lambda \frac{a_0 u_1 \left(v_1 + u_1 v_2 \right) + \rho_1 v_0 \left(u_0 + v_1 v_2 \right) + \rho_2 v_0 u_1 u_2}{v_0 u_1 \sqrt{Q}} \quad (24)$$

$$b_{2,1} = -\lambda \frac{a_0 u_2 \left(v_2 + u_2 v_1 \right) + \rho_1 v_0 u_2 u_1 + \rho_2 v_0 \left(u_0 + v_1 v_2 \right)}{v_0 u_2 \sqrt{Q}} \quad (25)$$

The results obtained for the contact between right-hand tooth flanks can also be exploited to infer information about the contact between left-hand tooth flanks, as in the path of contact $P_{1,-1}$ $P_{2,-1}$. Indeed, the left-hand flanks of both gears turn into right-hand flanks if the directions of the gear axes—as defined by unit vectors $\mathbf{n}_1$ and $\mathbf{n}_2$—are reversed. Thanks to this observation, the following relationships can be straightforwardly derived as:

$$\begin{aligned} \theta_{1,-1} &= -\theta_{1,1} & \theta_{2,-1} &= -\theta_{2,1} \\ b_{1,-1} &= -b_{1,1} & b_{2,-1} &= -b_{2,1} \\ & \sigma_{-1} = \sigma_1 \end{aligned} \quad (26)$$

This concludes determination of the loci of points where the involute helicoids of the two considered gears can come into contact.

Crossed Involute Helical Gears in Mesh

By relying on the results reported in the previous section, this section is devoted to determining the gearing backlash through a procedure similar to the one explained in Equation 7.

The gearing backlash H is here defined by:

$$H = N_1 \Delta\gamma_1 = N_2 \Delta\gamma_2 \quad (27)$$

where N_i and $\Delta\gamma i$ are, respectively, the number of teeth and the angular backlash of gear i (i=1,2).

In order to determine the meshing backlash for Gears 1 and 2 revolving about two given axes, the meshing with zero backlash of Gear 1 with a fictitious gear, referred to as Gear 2′ in the sequel, will be considered. The angular base thickness of Gear 2′ is greater than that of Gear 2 by an amount equal to $\Delta\gamma_2$, in turn related to the meshing backlash H of Gears 1 and 2 by Equation 27.

The involute helicoids L_2 and R_2 defining the left-hand and right-hand flanks of a tooth of Gear 2′ are now considered. Together with any other involute, helicoid gears mentioned in this section, L_2 and R_2 are supposed to extend indefinitely, starting from their base cylinders. L_2 is bound to come into contact with the left-hand flank L_1 of a tooth of Gear 1, while R_2 will touch the right-hand flank R_1 of another tooth of Gear 1, adjacent to the previous one. To find the relationship among the dimensions of the two gears and the relative positions of their axes, the following four-step man euver is imagined:

a) the contact point between involute helicoids L_1 and L_2, initially supposed at $P_{1,1}$, is moved to $P_{2,1}$ by suitably rotating both gears about their axes;

b) by further rotating Gears1 and 2′, helicoids R_1 and R_2 are made to go through point $P_{2,1}$;

c) the contact point between R_1 and R_2 is moved from $P_{2,1}$ to $P_{1,1}$;

d) helicoids L_1 and L_2 are made to go through point $P_{1,-1}$, which brings Gears 1 and 2′ to the position they had at the beginning of the first maneuver.

In the first maneuver, Gear 1 is rotated by an angle γ_{1a} given by (see also Ref. 7):

$$\gamma_{1a} = \frac{\sigma_1}{\rho_1 u_1} \quad (28)$$

In the second maneuver, Gear 2′ revolves about its axis by the following angle:

$$\gamma_{2b} = 2\theta_{2,1} + \varphi_2 + \Delta\gamma_2 - 2\frac{b_{2,1} v_2}{\rho_2 u_2} \quad (29)$$

The corresponding rotation angle for Gear 1 is:

$$\gamma_{1b} = -\frac{N_2}{N_1}\gamma_{2b} \quad (30)$$

To execute the third maneuver, Gear 1 has to revolve by the following angle:

$$\gamma_{1b} = -\frac{N_2}{N_1}\gamma_{2b} \quad (31)$$

Finally, the fourth maneuver requires Gear 1 to be rotated by:

$$\gamma_{1d} = -2\theta_{1,1} - \varphi_1 + 2\frac{b_{1,1}v_1}{\rho_1 u_1} + \frac{2\pi}{N_1} \qquad (32)$$

The series of the above-considered four maneuvers does not alter the angular position of Gear 1; hence the following relationship holds:

$$\gamma_{1a} + \gamma_{1b} + \gamma_{1c} + \gamma_{1d} = 0 \qquad (33)$$

By taking into account Equations 1, and 27 through 32, Equation 33 translates into the following condition:

See this page for equation (34)

Replacement into Equation 34 of the expressions for σ_1, $b_{1,1}$, and $b_{2,1}$—provided by Equations 22, 24 and 25—leads to:

See this page for equation (35)

This is the key condition for determining the meshing backlash H of a pair of involute helical gears mounted on skew axis shafts. Through Equation 35, the meshing backlash is expressed as a function of the gear geometry p, N_1, N_2, φ_1, φ_2; the relative placement of the gear axes, determined by a_0 and α_0, and quantities that are simple and known functions of these parameters: v_0, λ, Q, $\theta_{1,1}$, $\theta_{2,1}$. See also Equations 6, 23, 16, 20 and 21.

As far as the author is aware, this is the first time that the meshing backlash of two crossed-axis, involute gears is expressed in so concise a form.

Optimal Hob Setting

While a cylindrical involute gear with spur or helical teeth is being hobbed, the meshing of the gear with the hob can be considered as the meshing of two cross-axis, involute helical gears with zero backlash. Therefore the equations drawn in the previous two sections can be employed to analyze the kinematics of the hobbing process, provided that quantity H is set to zero.

The gear being cut and the hob—labeled in the sequel as Gears 1 and 2, and in no specific order—have known kinematically relevant dimensions. The parameters of the relative position of the gear axes—the axis distance a_0 and the axis angle α_0—are subject

$$\frac{4\pi}{p}\left(\sigma_1 + b_{1,1}v_1 + b_{2,1}v_2\right) - N_1\left(\varphi_1 + 2\theta_{1,1}\right) - N_2\left(\varphi_2 + 2\theta_{2,1}\right) + 2\pi = H$$

Equation 34.

$$\frac{4\pi\lambda\sqrt{Q}a_0}{pv_0} - N_1\left(\varphi_1 + 2\theta_{1,1}\right) - N_2\left(\varphi_2 + 2\theta_{2,1}\right) + 2\pi = H$$

Equation 35.

$$F(a_0,\alpha_0) = \frac{4\pi\lambda\sqrt{Q}a_0}{pv_0} - N_1\left(\varphi_1 + 2\theta_{1,1}\right) - N_2\left(\varphi_2 + 2\theta_{2,1}\right) + 2\pi$$

Equation 37.

to the ensuing condition (Equation 35):

$$F(a_0,\alpha_0) = 0 \qquad (36)$$

where:

See this page for equation (37)

Since Equation 36 is the only condition that parameters a_0 and α_0 have to comply with, there exists a simple infinity of possible relative settings of the hob axis with respect to the gear axis. More precisely, any axis setting that satisfies Equation 36 cuts out the flank of the gear teeth from the same set of involute helicoids (here considered as surfaces with indefinite extent). Simply, different choices of a_0 and α_0 that comply with Equation 36 select different patches from the same set of involute helicoids.

The criterion suggested in this paper for choosing the relative position of the axes of gear and hob is the minimization of the axis distance a_0. The rationale of this choice lies in the consequent maximization of the radial extension of the tooth flanks, for a given hob and a prescribed tip diameter of the gear.

The gear hob axis distance a_0 reaches an extreme value when the ensuing condition is satisfied as:

$$\frac{\partial F}{\partial \alpha_0} = 0 \qquad (38)$$

The minimum possible value of a_0, together with the corresponding value for α_0, derive from simultaneously solving Equations 36 and 38.

To take advantage of Equation 38, some partial derivatives have to be computed. On the right- hand side of Equation 37, quantities

$$\frac{2\pi a_0}{p v_0}\left[v_1 v_2 + u_0\left(v_1^2 + v_2^2 + u_0 v_1 v_2\right)\right] + N_1\left(v_1 + u_0 v_2\right) + N_2\left(v_2 + u_0 v_1\right) = 0$$

Equation 42.

$$a_0 = -\frac{p v_0}{2\pi}\,\frac{N_1\left(v_1 + u_0 v_2\right) + N_2\left(v_2 + u_0 v_1\right)}{v_1 v_2 + u_0\left(v_1^2 + v_2^2 + u_0 v_1 v_2\right)}$$

Equation 43.

$$2\lambda\sqrt{Q}\,\frac{N_1\left(v_1 + u_0 v_2\right) + N_2\left(v_2 + u_0 v_1\right)}{v_1 v_2 + u_0\left(v_1^2 + v_2^2 + u_0 v_1 v_2\right)} + N_1\left(\varphi_1 + 2\theta_{1,1}\right) + N_2\left(\varphi_2 + 2\theta_{2,1}\right) - 2\pi = 0$$

Equation 44.

Q, v_0, $\theta_{1,1}$, and $\theta_{2,1}$ are functions of α_0. Finding the derivatives of Q and v_0 with respect to α_0 poses no hurdles whatsoever. To determine the derivative of $\theta_{1,1}$ with respect to α_0, both sides of the first of Equations 14 are derived with respect to α_0. Following elementary algebraic manipulation, the ensuing condition is obtained:

$$s_{1,1}\frac{d\theta_{1,1}}{d\alpha_0} = -\frac{v_1 + u_0 v_2}{v_0^2 u_1} \tag{39}$$

Insertion of the expression of Equation 18 for $s_{1,1}$ yields:

$$\frac{d\theta_{1,1}}{d\alpha_0} = -\lambda\frac{v_1 + u_0 v_2}{v_0\sqrt{Q}} \tag{40}$$

The derivative of $\theta_{2,1}$ with respect to α_0 is obtained in a similar way:

$$\frac{d\theta_{2,1}}{d\alpha_0} = -\lambda\frac{v_2 + u_0 v_1}{v_0\sqrt{Q}} \tag{41}$$

With the aid of Equations 40 and 41, Equation 38 can be rewritten as:

See this page for equation (42)

In order to simultaneously solve Equations 36 and 42, the expression of a_0 as a function of α_0 is first linearly obtained from Equation 42:

See this page for equation (43)

and then inserted into Equation 36. The resulting equation contains α_0 as the only unknown:

See this page for equation (44)

In Equation 44, quantities u_0, v_0, Q, $\theta_{1,1}$, and $\theta_{2,1}$ are functions of α_0. Their expressions in terms of α_0 are given by Equations 6, 16, 20 and 21.

A further comment pertains to quantity λ, which appears in Equation 44, both explicitly and implicitly (see Eqs. 20 and 21). Although the value of λ should be obtained by Equation 23, for the case of a hob cutting a gear, the sum of the base cylinder radii ρ_1 and ρ_2 is generally smaller than the axis distance a_0, hence the following inequality is satisfied as:

$$a_0 - \rho_1 c_{1,1} - \rho_2 c_{2,1} > 0 \tag{45}$$

Consequently, in this case, λ can be given the ensuing simplified expression:

$$\lambda = \begin{cases} -1 & \text{if} \quad v_0 < 0 \\ 1 & \text{if} \quad v_0 \geq 0 \end{cases} \tag{46}$$

The hob shaft angle that allows a given hob to cut a given gear at the minimum axis distance is the value of α_0 that satisfies Equation 44. Once this value has been numerically determined, its insertion into Equation 43 straightforwardly yields the corresponding, or minimum, axis distance a_0.

More generally, Equations 44 and 43 can be resorted to whenever it is of interest to find the relative position of the axes of two cylindrical helical gears with involute teeth that have to mesh together—with no backlash—at the minimum axis distance.

Numerical Example

The results presented in the previous section are here applied to determine the setting parameters for the finish-hobbing operation of a given helical involute gear. Two cases will be considered—cutting by a given right-handed hob and cutting by a left-handed hob that is the mirror image of the former. The results obtained in these two cases will be compared to those stemming from the corresponding customary choice of the hob shaft angle.

Let us suppose that the gear to be hob-machined is characterized by the number of teeth $N_1 = 17$; normal pressure angle $\xi_n = 20°$; normal module $m_n = 5$ mm; helix angle at the standard pitch cylinder $\beta_{p1} = 29.5°$; profile shift $x_1 = 3$ mm.

The hobs are double-threaded, i.e., $N_2 = 2$. They have the same normal pressure angle and normal module as the gear. Moreover, their addendum is 1.25 times the normal module, and the radius of their tip cylinder is $R_{e2} = 65$

mm. The cylinder coaxial with each hob that intersects the left-hand and right-hand flanks of the hob threads at equally spaced helices has radius $R_{p2} = R_{e2} - 1.25 m_n = 58.75$ mm.

As shown hereafter, standard computations lead to the normal base pitch (p), the angular base thickness (φ_1 and φ_2) and base helix angle (β_1 and β_2) of gear (1) and hobs (2), via determination of the transverse module (m_1 and m_2) and the transverse pressure angle (ξ_1 and ξ_2):

$$p = \pi m_n \cos\xi_n = 14.76065717 \text{mm} \quad (47)$$

$$m_1 = \frac{m_n}{\cos\beta_{p1}} = 5.744777708 \text{ mm} \quad (48)$$

$$\xi_1 = \tan^{-1}\left(\frac{\tan\xi_n}{\cos\beta_{p_1}}\right) = 22.69398023 \text{ deg} \quad (49)$$

See this page for equation (50)

See this page for equation (51)

$$m_2 = \frac{2R_{p_2}}{N_2} = 58.75 \text{ mm} \quad (52)$$

$$\beta_{p_2} = \pm\cos^{-1}\left(\frac{m_n}{m_2}\right) = \pm 85.11785767 \text{ deg} \quad (53)$$

$$\xi_2 = \tan^{-1}\left(\frac{\tan\xi_n}{\cos\beta_{p2}}\right) = 76.83910904 \text{ deg} \quad (54)$$

See this page for equation (55)

See this page for equation (56)

The foregoing computations also yield the hob helix angle β_{p2} at the standard pitch cylinder (β_{p2} is the helix angle measured at distance R_{p2} from the hob axis; the positive value of β_{p2} refers to the right-threaded hob).

All parameters relevant to the analysis at hand (i.e., normal base pitch p, numbers of teeth N, angular base thickness φ, and base helix angle β) are listed in Table 1. The substantial number of decimal digits used in reporting both data and results have the only purpose of allowing the reader to accurately trace the computations here summarily described.

By following the procedure explained in the previous section, the optimal hob settings reported in Table 2 have been obtained. It is worth noting that the shaft angle α_0 referred to in Table 2 has the meaning explained in Section 2, which does not always coincide with the meaning assigned to this term by other authors (for instance, the shaft angle Σ defined in Reference 3 is the opposite of the shaft angle α_0 adopted in this paper).

In order to compare the gain attainable by the proposed procedure for determining the hob setting, the shaft angle $\alpha 0$ has been

$$\varphi_1 = \left(\frac{\pi m_1}{2} 2x_1 \tan\xi_1\right)\frac{2}{m_1 N_1} + 2\,\text{inv}\,\xi_1 = 0.2803854852 \text{ rad} = 16.06490494 \text{ deg}$$

Equation 50

$$\beta_1 = \tan^{-1}(\tan\beta_{p1}\cos\xi_1) = 27.56320246 \text{ deg}$$

Equation 51

Table 1—Key geometric parameters of gear and hobs.		
	Gear	Hobs
π [mm]	14.76065717	14.76065717
N	17	2
α[deg]	16.06490494	426.38980178
σ [deg]	27.56320246	±69.43646886

Table 2—Optimal hob settings.		
	Left-threaded Hob	Right-threaded Hob
α_0 [deg]	55.84099326	–114.86308717
σ_0 [mm]	110.57666813	110.57666813

Table 3—Customary hob settings.		
	Left-threaded Hob	Right-threaded Hob
α_0 [deg]	55.61785767	–114.61785767
σ_0 [mm]	110.58061052	110.58061052

$$\varphi_2 = \frac{\pi}{N_2} + 2\,\text{inv}\,\xi_2 = 7.441905938 \text{ rad} = 426.38980178 \text{ deg}$$

Equation 55

$$\beta_2 = \tan^{-1}(\tan\beta_{p2}\cos\xi_2) = \pm 69.43646886 \text{ deg}$$

Equation 56

set at its customary value, i.e. $-(\beta_{p1}+\beta_{p2})$, and the corresponding shaft distance a_0 has been computed by solving Equation 36 (now a linear equation in a_0). The results are reported in Table 3.

A comparison of Tables 2 and 3 reveals that the gain obtained by the proposed procedure is marginal to say the least. With reference to the standard hob setting, the optimum hob setting would require a variation of the shaft angle by a fourth of a degree, the result being a decrease of the shaft axis distance by only 0.004 mm.

Similar observations could be made when comparing the optimal and standard hob settings for an involute spur gear.

Conclusions

The paper has suggested a criterion for selecting the hob setting for cutting spur and helical involute gears. Implementing the proposed criterion requires a transcendental equation in only one unknown to be numerically solved.

Although computationally not very demanding, the presented procedure is more complex than the standard one, and conducive to marginally better results. Therefore the effectiveness of the standard procedure practically rests confirmed.

On the other hand, addressing the considered hob setting problem has led to devising a new formulation of the equations governing the meshing of crossed-axis involute gears. These equations, more lean and compact than those published thus far in the technical literature, could find application to other contexts as well.

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A shorter version of this paper was presented at the 2006 ASME International Mechanical Engineering Congress and Exposition, November 5–10, 2006, Chicago, Illinois, USA.

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State of the Gear Industry

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

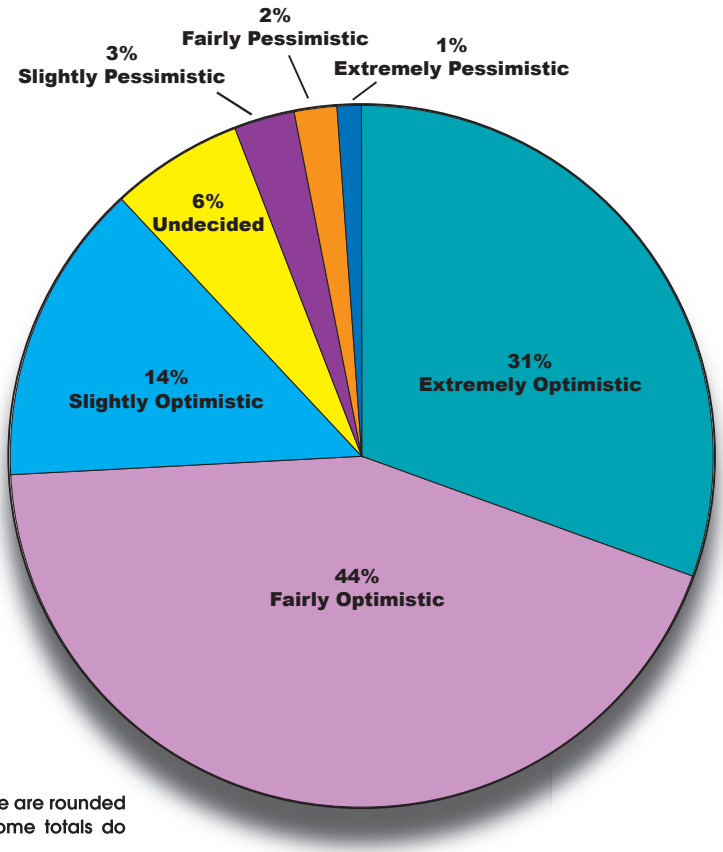
In October, Gear Technology conducted an anonymous survey of gear manufacturers. Invitations were sent by e-mail to thousands of individuals around the world. More than 400 individuals at gear manufacturing locations responded to the online survey, answering questions about their manufacturing operations and current challenges facing their businesses.

The respondents considered here all work at locations where gears, splines, sprockets, worms and similar products are manufactured. They work for gear manufacturing job shops (42%), captive shops at OEMs (48%) and shops manufacturing gears for maintenance, spares and their own use (10%).

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

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

91% of Gear Industry Respondents are Optimistic About their Ability to Compete over the next 5 Years



Note: All percentages in this article are rounded to the nearest whole number. Some totals do not equal 100% as a result.

Most in the gear industry would tell you it's been another good year. Based on our research, gear manufacturers around the world clearly remain optimistic about their future. In fact, the number who are slightly optimistic or better about their ability to compete over the next five years is 88% (essentially equivalent with last year's 91%).

Employment increased at 55% of gear industry operations, roughly the same as last year. Most expect that employment at their operations will increase again next year—if they can find and keep qualified employees—an issue which continues to be one of the most significant facing the industry.

"Developing and retaining a skilled workforce to support the increased complexities in current and future processes" is one of the biggest challenges facing a manufacturing engineer at a U.S. material handling equipment manufacturer. Many others had similar comments.

Most respondents at gear manufacturing locations also saw large increases in their production volumes in 2007. 69% reported at least some increase, with more than 20% of respondents indicating their production volumes increased by more than 20% over 2006.

And it doesn't appear to be slowing down in 2008, either. In fact, more respondents predicted production increases for 2008 (79%) than experienced production increases in 2007.

"Can't make stuff fast enough," said a design engineer for a U.S. manufacturer of wind turbine gears.

From reading through the responses, our editors got the feeling that growth would be even faster in 2008 if the materials, machines and workers were available to make more gears.

"Finding the capital to support growth" was one of the major challenges facing a corporate executive at a company that manufactures timing gears in Mexico.

"Getting new machines fast enough" was the significant challenge cited by a corporate executive at a gear manufacturing job shop in Denmark.

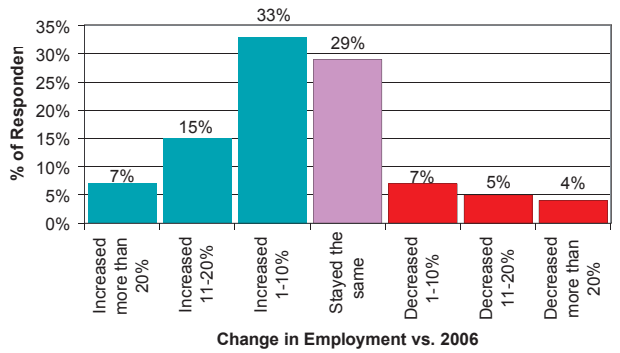
"Having difficulty in finding skilled help is slowing the growth of the company," said a manufacturing production worker for an aerospace gear manufacturer in California.

Sales rose for most in 2007, and most expect them to continue to rise in 2008. 71% saw sales volumes rise in 2007 (the same percentage as last year). 75% expect further increases next year (slightly higher than last year).

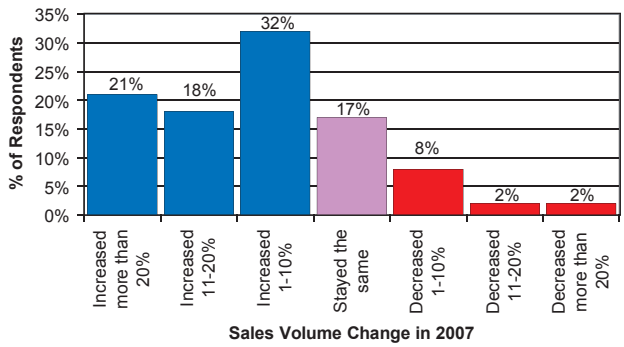
Thanks to the weak dollar, most U.S. respondents are seeing increases in exports, and their concerns aren't so much about foreign competition as they are about increased competition from their fellow U.S. manufacturers. Corporate executives at more than one gear manufacturing job shop cited domestic competition as the most significant challenge facing their businesses.

A number of Canadian respondents indicated their difficulty in selling to America because the weak U.S. dollar makes their products more expensive.

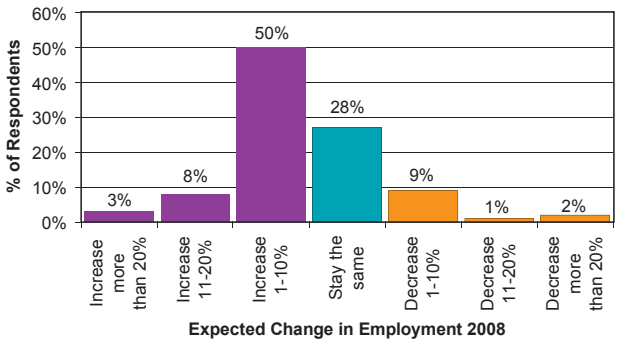
55% of Gear Industry Respondents Work at Locations Where Employment Increased in 2007



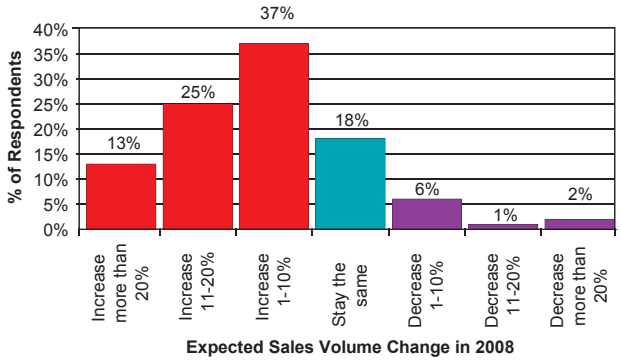
71% Saw Sales Volume Increase in 2007



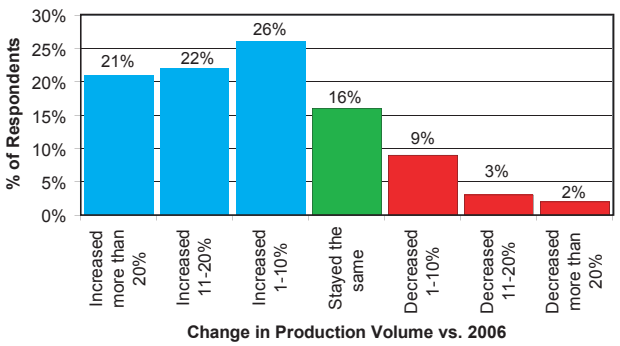
61% of Gear Industry Respondents Expect Employment at their Location to Increase in 2008



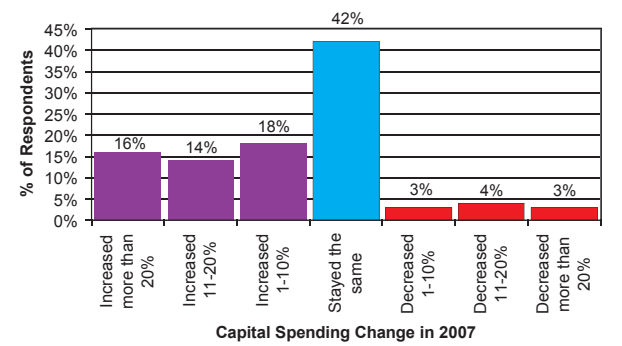
75% Expect Sales Volume Increase in 2008



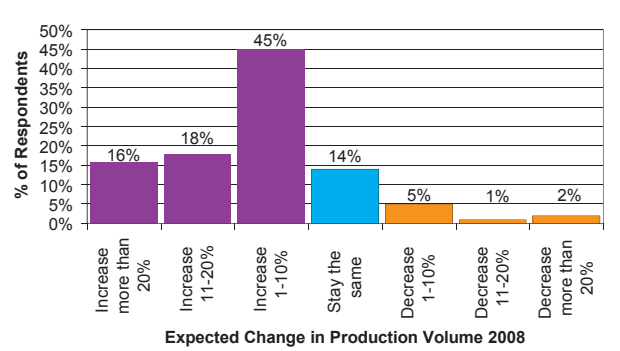
71% Saw Production Volumes Increase in 2007



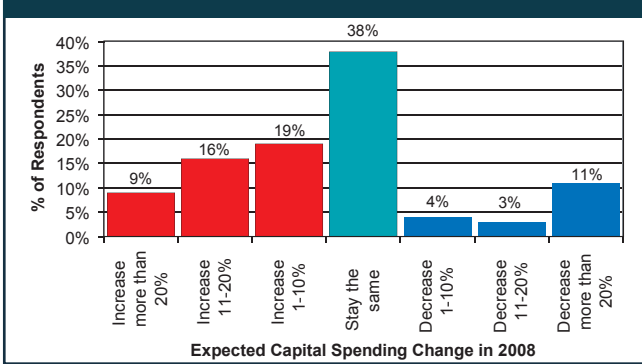
48% Work at Locations Where Capital Spending Increased in 2007



79% Expect Production Volume to Increase in 2007



44% Expect Capital Spending at Their Locations to Increase in 2008



Not a single U.S. respondent mentioned any worries about China. But more than a few outside the U.S. did.

"The biggest challenge beyond finding good skilled labor is cheap competition who have low overhead cost and worthless machinery but are ready to give you the challenge," said a manufacturing engineer at an Indian gearbox manufacturer.

"The Chinese currency is not free floating," said a purchasing agent at a transmission manufacturer in India, "which makes Chinese imports cheaper."

A significant number of respondents (43%) reported that capital spending increased at their locations in 2007, and about the same number (44%) expect that the increases will continue in 2008.

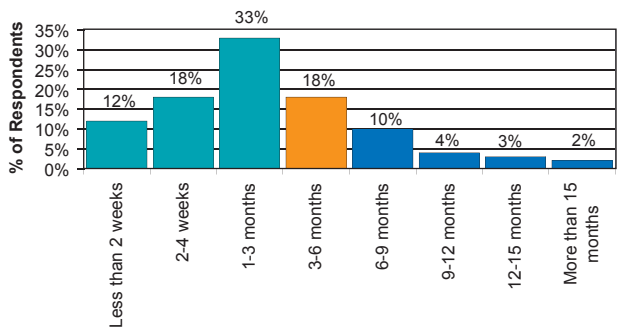
We also asked respondents specifically about their biggest manufacturing and engineering challenges. Overwhelmingly, the responses came back: reduce costs, shortening lead time and improve quality.

When respondents mentioned quality, they weren't so much concerned about the quality of individual parts or specific quality levels. Rather, they are becoming acutely aware of the need for improved quality systems. Lean manufacturing seemed to be of the most interest.

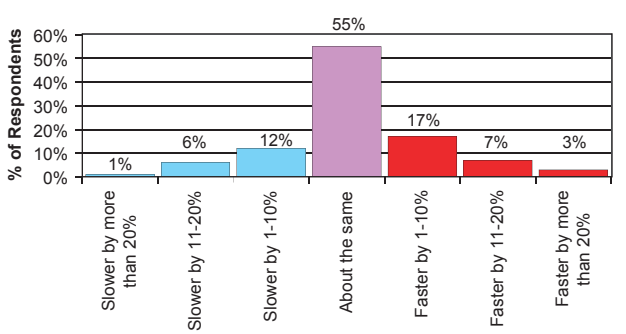
While sales and production are up, and most gear manufacturing operations are growing, it's clear from the responses of this survey that most gear manufacturers are under enormous pressure. They need to reduce lead times, they need to improve throughput. They're looking at automation and quality management programs. They're desperate for qualified employees.

But still, they're hopeful.

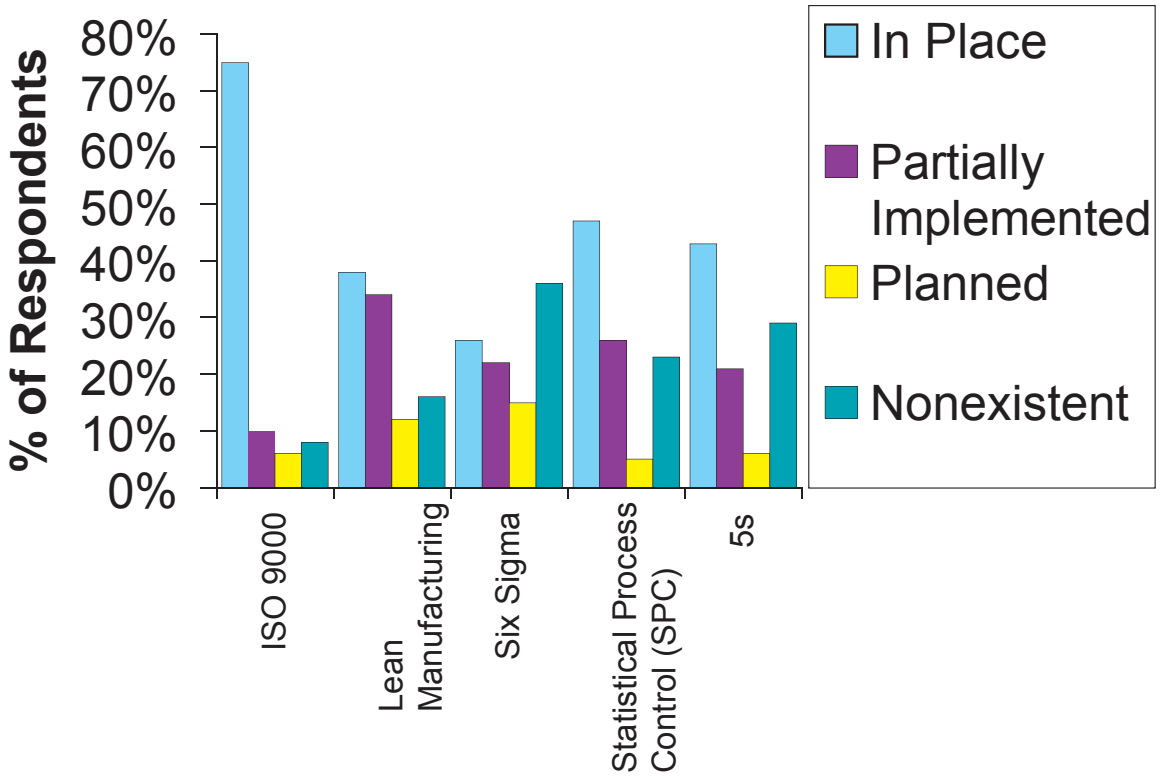
Current Delivery Time for New Orders



Delivery Time vs. 2006



Extent of Quality Measures



What Factors Are Presenting Significant Challenges to Your Business?

"Awareness of career potential in metal cutting being presented by the current educational systems."
—A manufacturing engineer for a major off-road equipment OEM

"Can't make stuff fast enough."
—Design engineer for a U.S. manufacturer of wind turbine gears

"Domestic competition."
—Corporate executive for a NY-based gear manufacturing job shop

"Keeping up with demand and freeing up sufficient machining capacity to reduce leadtimes. Reduce the amount of sub-contracted parts currently produced."
—Design engineer for a U.K. manufacturer of measuring devices

"Lack of upper management with leadership and change agent abilities."
—Employee at a U.S. gearbox manufacturer

"Poor management and scheduling of production, also poor choice in equipment purchasing."
—Employee at a CA-based aerospace gear manufacturer

"Shortened lead time requirements."
—Corporate executive at an OH-based gear manufacturing job shop

"So many continuous improvement challenges of various names."
—Corporate executive at a gear manufacturing job shop in MI

"The cost of training new employees with no experience and the increase in scrap and rework due to new employees."
—Corporate executive at a gear manufacturing job shop in IL

"Exceptionally poor management."
—Design engineer for a U.S. Tier 1 automotive component supplier

"Finding skilled engineers."
—Marketing manager at a U.S. manufacturer of aerospace gears

"Gear manufacturing quality level."
—Design engineer for an automotive OEM with global operations

What Factors Are Presenting Significant Challenges to Your Business?

"Good quality on time for the best price."
—Manufacturing engineer at a gear manufacturing job shop in Belgium

"Having difficulty in finding skilled help is slowing the growth of the company."
—Manufacturing production worker for a CA-based aerospace gear manfacturer

"Heavy turnover of good engineers."
—Corporate executive at a large gear manufacturing job shop in India

"Implementation of labor savings strategies."
—Marketing manager for a Midwest manufacturer of gearmotors

"Lack of upper management with leadership and change agent abilities."
—Employee at a U.S. gearbox manufacturer

"Management approving manpower request."
—Manufacturing engineer for a U.S. bearings manufacturer

"Material and tooling supply times."
—Manufacturing engineer for a U.S. OEM of automotive transmissions

"Outsourcing."
—Manufacturing engineer at a U.S. OEM of actuators

"Poor management and scheduling of production, also poor choice in equipment purchasing."
—Employee at a CA-based aerospace gear manufacturer

"Staffing."
—Manufacturing engineer at a gear manufacturing job shop in South Africa

"Talent retention."
—Corporate executive at a gear manufacturing job shop in India

"The biggest challenge beyond finding of good skilled labor is cheap competition who have low overhead cost and worthless machinery but ready to give you the challenge."
—Manufacturing engineer at a gearbox manufacturer in India

What are your company's greatest manufacturing/ engineering challenges for 2008?

"Automation and Increased inspection."
—Quality control manager at a U.S. plastic gear manufacturer

"Cost reduction & quality improvement."
—Design engineer at a NY-based gear manufacturing job shop

"Cost reduction without affecting the quality of end product."
—Manufacturing production worker at an OEM of gearmotors in India

"Cross training lean implementation."
—Corporate executive at an IL-based gear manufacturing job shop

"Face gear manufacturing."
—Design engineer at a MI-based manufacturer of geared systems

"Finding skilled labor and designers."
—Marketing manager at an OEM of transmissions in Canada

"Increased throughput."
—Corporate executive for an IL-based gearbox manufacturer

"Increasing machining capacity by looking at „lights out" technology. To continue to develop new products for a more diverse market."
—Design engineer at a manufacturer of measuring devices in the U.K.

"To continue to develop new products for a more diverse market."
—Design engineer at a manufacturer of measuring devices in the U.K.

"Improving lead time."
—Design engineer at a WI-based gear manufacturing job shop

"International site start-up."
—Manufacturing engineer for a major U.S. OEM of transmissions

"Involving staff in modern philosophies."
—Manufacturing engineer at a gear manufacturing job shop in the U.K.

"Just trying to keep the doors open and the lights on."
—Design engineer at a major U.S. Tier 1 automotive supplier

"Keeping our overheads low and hopes high to stay in business."
—Marketing manager for a gear manufacturing job shop in Malaysia

"Knowing where new and existing customers need us to be in terms of services offered."
—Corporate executive for a NC-based gear manufacturing job shop

"Lack of time and capacity. Engineering throughput."
—Marketing manager at a MI-based gear manufacturing job shop

"Machine uptime."
—Manufacturing engineer at a U.S.-based OEM of automobile transmissions

"Maintaining quality."
—Design engineer for a U.S.-based manufacturer of locomotives

"Material availability."
—Manufacturing production worker for a gear manufacturing job shop in New Zealand

"Materials."
—Corporate executive at a U.S.-based gear drives OEM

"Meeting customer requirements at current or reduced costs."
—Manufacturing engineer for a U.S.-based aerospace OEM

"Mfg: Reducing labor costs and shortening lead time. Eng: Developing lower cost gearmotors in shorter timeframes."
—Marketing manager for an IL-based OEM of gearmotors

"New customer development."
—Manufacturing engineer for a NC-based gear manufacturing job shop

"New product designing."
—Manufacturing engineer for a gear manufacturing job shop in India

"New product evolution and revolution, new market introduction, elimination of waste in manufacturing processes."
—Consultant to a major U.S. manufacturer of power transmission components

"Planetary gearboxes, windmill gearboxes of 1MW and above."
—Corporate executive at a gear manufacturing job shop in India

"Process controls."
—Prchasing agent at a major U.S. OEM of jet engines

"Process improvement."
—Quality control manager at an OEM of gear drives in Taiwan

"Raising skill level of inexperienced employees, and then being able to keep them."
—Design engineer for an OH-based OEM of gear drives

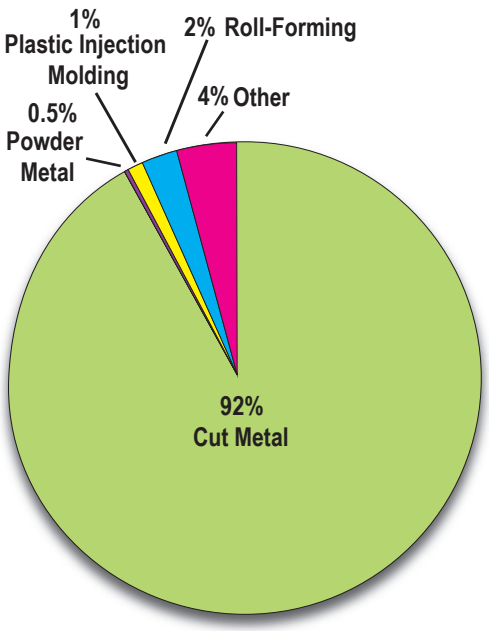
"Reducing lead times."
—Manufacturing engineer at a WI-based OEM of heavy equipment

"Reducing lead times."
—Corporate executive for a gear manufacturing job shop in Brazil

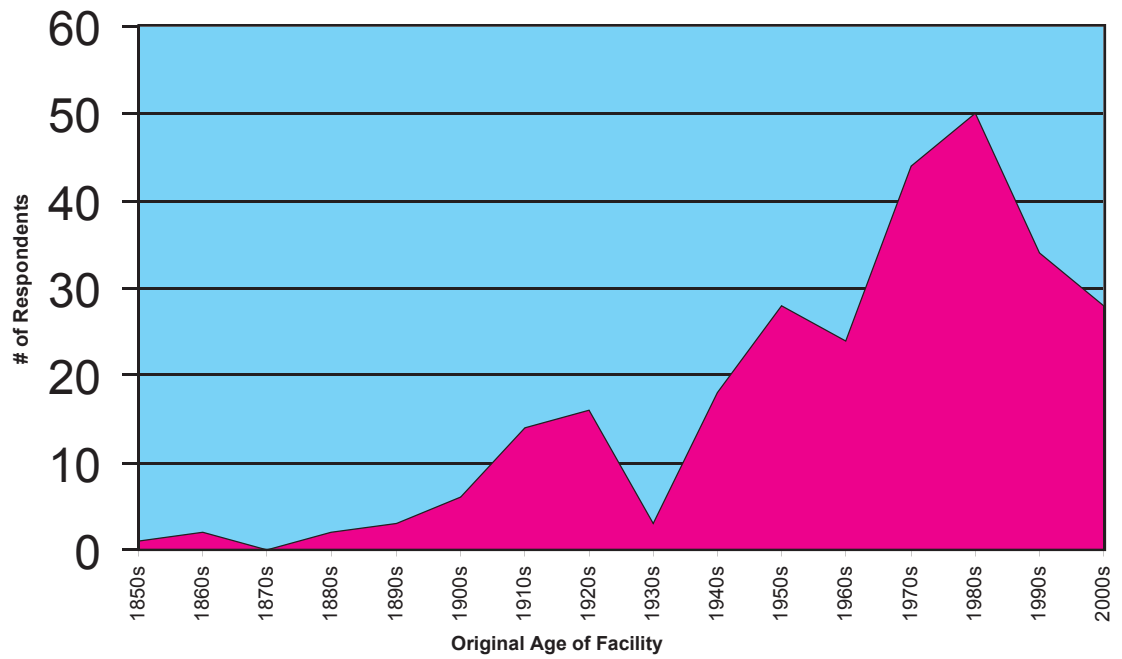
"Reducing overhead cost and this comes with new modern machinery which give better productivity but the competition does not allow you to do this."
—Manufacturing engineer for an OEM of gearboxes in India

"Securing new orders to replace existing orders which are coming to an end."
—Manufacturing engineer for a gear manufacturing job shop in the U.K.

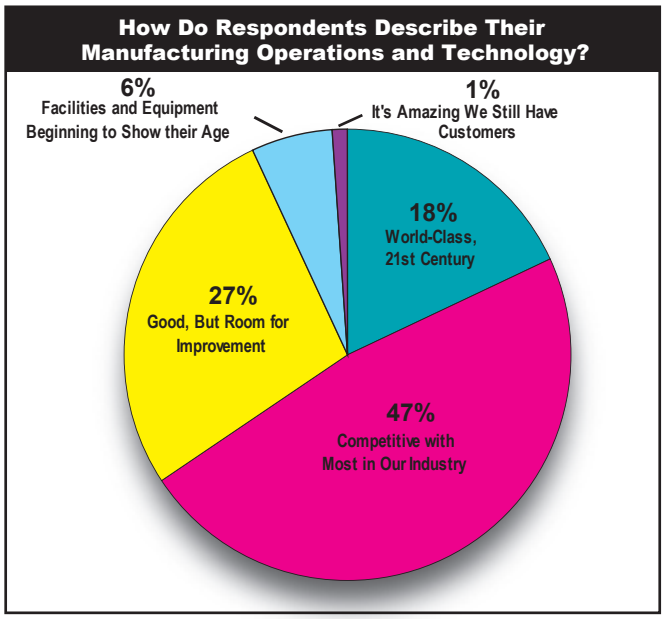
Primary Method of Manufacture

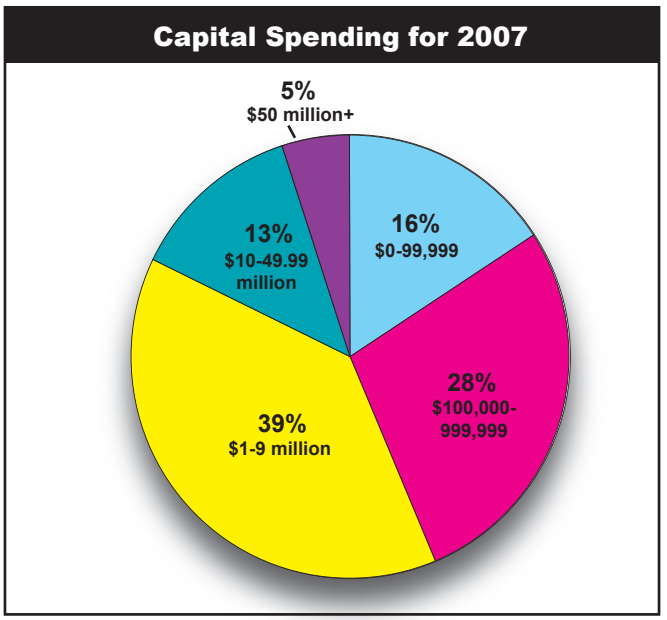


Age of Manufacturing Location

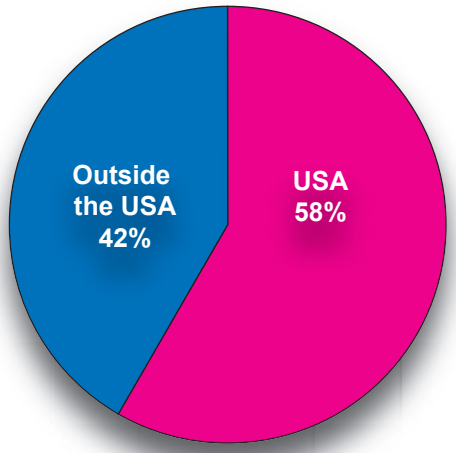


Half of gear industry respondents work in facilities originally built before 1967

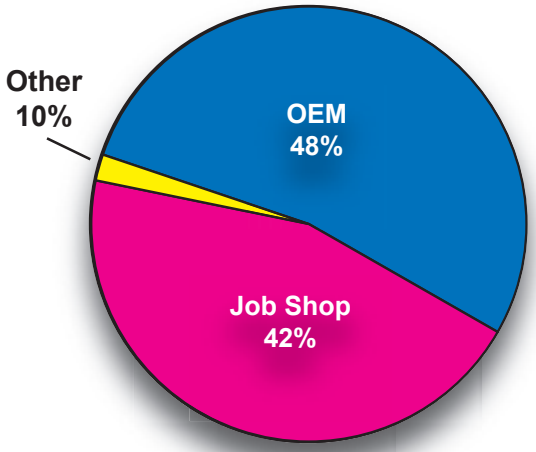




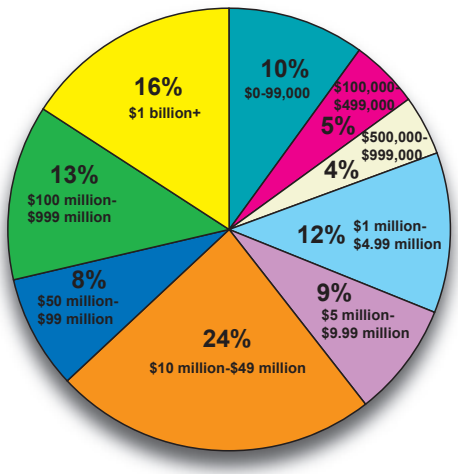
State of the Gear Industry: Who Responded



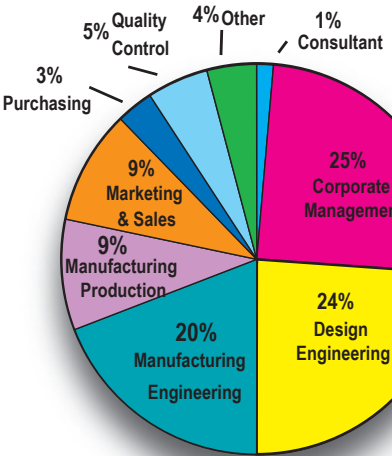
Type of Operation



Sales Volume of Company



Job Title/Function of Respondent



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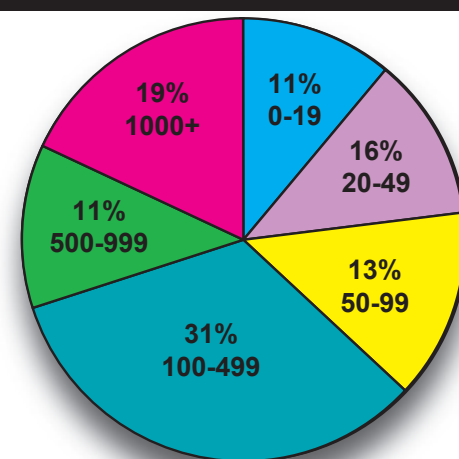
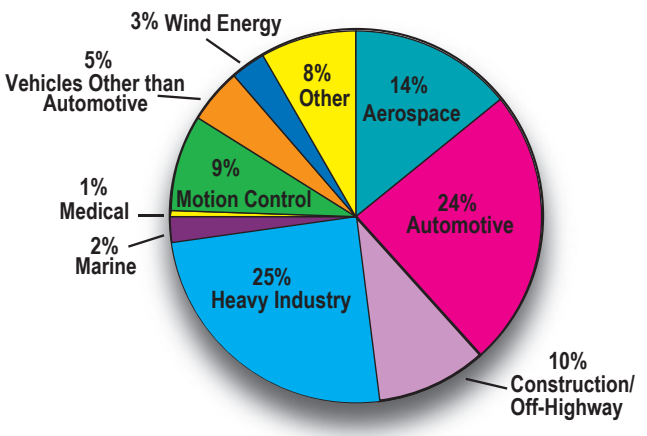


Prime Industry



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Gear Expo 2007—
AN UP-TICK IN ATTENDANCE AND HIGH HOPES FOR 2009

The general impression—whether encouraged by AGMA or developed anecdotally—is that Gear Expo 2007 was a reasonable—though certainly relative—success for many of the exhibitors and attendees. A final count supplied by AGMA shows that 3,035 (1,212 exhibitors and 1,823 attendees) were on hand—an appreciable 20% increase over 2005. A breakdown of visitor demographics reveals that 43 states and 28 foreign countries, spanning six continents, were represented.

And not to bury the lead, but the biggest news to come out this year's show was the AGMA announcement that, in addition to Expo 2009 moving to Indianapolis, the show will open on a MONDAY. (Please see adjoining sidebar.)

As for this year's expo, Kurt Medert, AGMA vice president and Gear Expo show manager, suggested that it appeared—excepting Sunday and Wednesday—to generate more legitimate interest among attendees in exhibitor offerings.

"What was most striking to me was the number of exhibitors who commented that the higher attendance and the fact that attendees spent more days at the show generated visibly more foot traffic, and that the attendees were mostly high-quality visitors," he says.

Another success for the show was the Solutions Center, inaugurated in 2005. In fact, the presentations on a variety of gear industry issues were well attended and received, evidenced by the fact that this year's numbers far outstripped those of two years ago.

"Again, the vast majority of responses were positive," says Medert. "Eight hundred and sixty six attended the presentations—an average of (approximately) 30 per session.

The obvious show-stoppers—standing room only—were Mike

Bradley's crisp snapshot of the industry's economic forecast, and presentations on the rapidly growing American wind turbine industry by the American Wind Energy Association (AWEA) and the auto industry by leading automotive consultant Casey Selecman.

Beyond AGMA's satisfaction with the show, we also talked to a number of exhibitors to gain their honest impressions of Gear Expo 2007. On whole, most respondents' impressions mirrored those of AGMA. Others, not so much. Following is a sampling covering the show's benefits, suggestions for improvements, and whether they plan on exhibiting in 2009.

"ABA-PGT was very pleased with the booth activity," says Rick Wheeler, company president. "We were pleasantly surprised at the interest in plastic gears; not so much as metal-to-plastic replacements, but for new projects demanding lower weights in smaller torque load requirements. (I would) start the activities on Monday and run a Monday-Friday show. The FTM paper presentations on Sunday are a family hardship for presenters. ABA-PGT plans to exhibit at the 2009 Gear Expo."

"Detroit is a bad venue," says Brian Cluff, vice president of sales and application engineering for Star SU. "And again the expo was poorly attended, although not as badly as the last show. The cost to go to the show exceeds the return we get from it." As for improvements, "Move Gear Expo in with IMTS as a separate pavilion." And for 2009? "We will exhibit with a very small, minimal presence, but only because we are already committed," says Cluff. "After 2009, we will not attend."

Heat treater Solar Atmosphere's sales manager Mike Johnson says, "For 17-4, 440C and 430 grades for thru-hardening, day two was very productive. I commented in the (AGMA post-show) survey that I thought the show should only be two, maybe three days long. A

half-day on Sunday was a waste, and the fourth day was dead. Good show, though. It yielded more PR than I thought it would."

"Visitors seemed to be much more focused this year—more specific projects/needs to be discussed—less than 'just visiting,'"says Jim Vosmik, president of Drake Manufacturing. "Yes, we plan on exhibiting in 2009."

Lori Rugh, AWEA (American Wind Energy Association) marketing manager for exhibits and sponsorships, says, "(We) found the show to be greatly beneficial, as we raised the visibility and awareness of gear manufacturers about the wind industry and the need of their products in our industry. The show is great and the only improvement might be to have more folks staffing the registration desk, as there seemed to be a bottleneck there." She adds that "hopefully," AWEA will be there in 2009. "We really appreciate all that AGMA did for AWEA. It was all great."

Lastly, Carl Eckberg, vice president of gear products for Bourn & Koch, Inc., says, "The show offered an excellent opportunity to introduce new products to the attendees. The machine (we) exhibited was not for sale and was intended to be used in our own gear department. But of the three companies that were interested in purchasing it, one is writing the order this week. We will build a duplicate for our own use." Eckberg adds he is "OK" with the show as is, and, "Yes," they will be back in 2009.

On balance, even the show's harshest

critics would have to admit that this year's Gear Expo was at least a positive step forward. After all, it beats losing attendance. One might reasonably expect—as AGMA most assuredly does—that the improvement in this year's numbers bodes well for 2009 when the show setting will be in—with sincere apologies to the Motor City—a thriving urban setting.

Gentlemen and Ladies, Start Your Engines. It's Indy in '09. And ASM, Too.

It is now fact that Gear Expo 2009 will return to a city with a long and legendary motor sport pedigree—Indianapolis, IN. AGMA made it official in announcing last month that Gear Expo 2009 will be held at the Indiana Convention Center. Show dates (tentative as we go to press) are September 14–17, and—to the delight of many—note that the show will NOT open on a Sunday. The Monday opening is due to a scheduling conflict at the Center. AGMA's Medert also says that "minor modifications" to the schedule are also under consideration, but further details were not available.

"Gear Expo '95 was held (in Indianapolis) and everything went well," he says.

Of equal significance, Gear Expo 2009 will for the first time be co-located with another show—the ASM International Heat Treating Society Conference and Exposition. This new partnership will add upwards of 4,000 attendees from 25 countries to the mix, a move that many longtime exhibitors and attendees will applaud. With—until this year—a continued slip in attendance for the last several shows, those with influence in the gear industry—and with AGMA—have been calling for just such a new strategy. Some have suggested a pavilion at IMTS; others have looked to join a show like the ASM conference and exhibition. By most accounts, the show needed tweaking, and thus the change.

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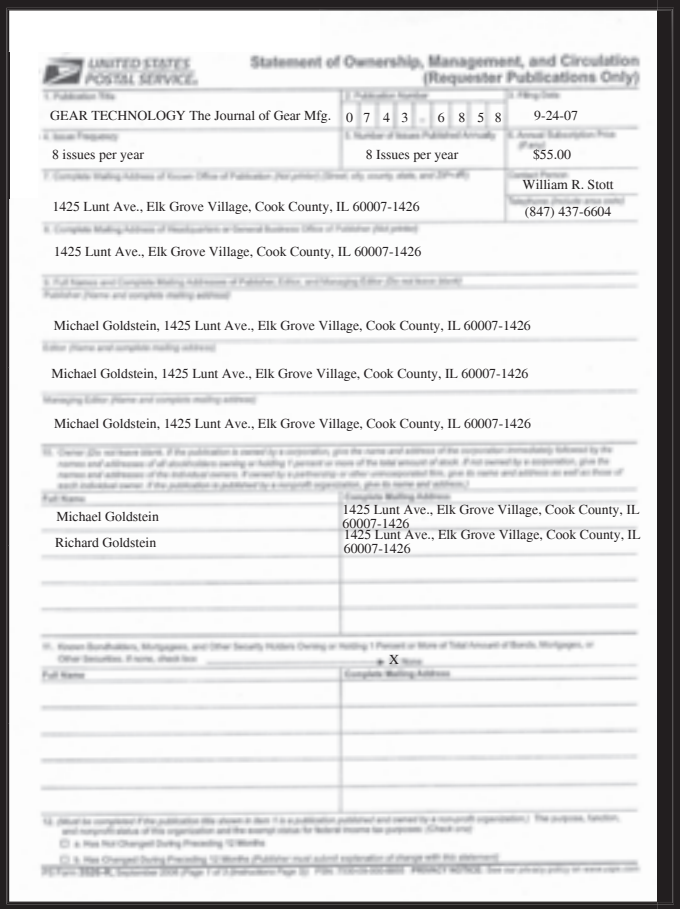

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Michael Goldstein, 1425 Lunt Ave., Elk Grove Village, Cook County, IL 60007-1426

Full Name	Complete Mailing Address
Michael Goldstein	1425 Lunt Ave., Elk Grove Village, Cook County, IL 60007-1426
Richard Goldstein	1425 Lunt Ave., Elk Grove Village, Cook County, IL 60007-1426

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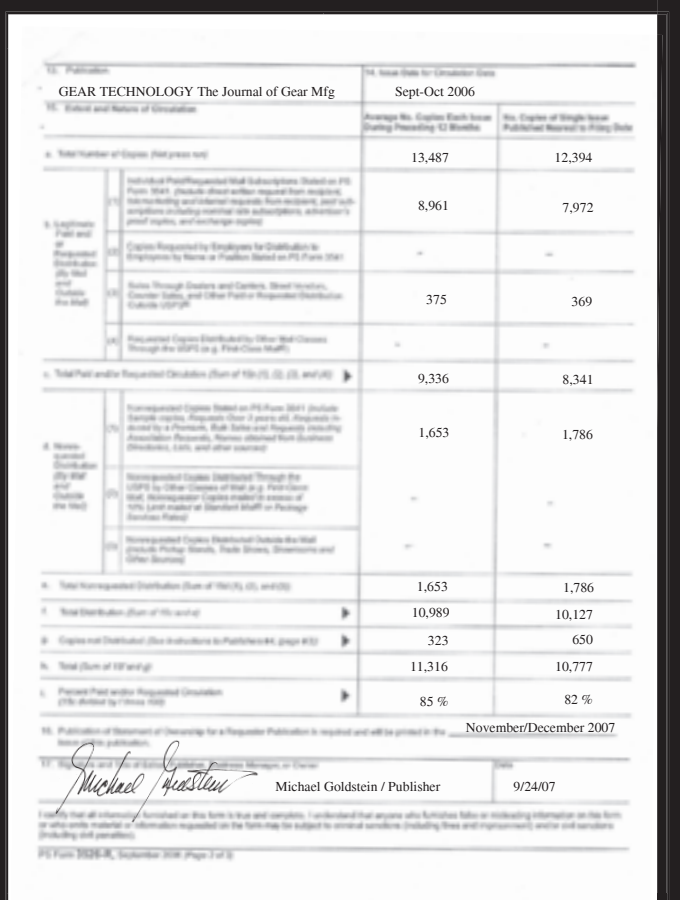

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16. Publication of Statement of Ownership for a Requester Publication is required and will be printed in the November/December 2007 issue of this publication.

17. Signature and Title of Editor, Publisher, Business Manager, or Owner — Michael Goldstein / Publisher — Date 9/24/07

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PS Form 3526-R, September 2006 (Page 2 of 3)

December 3–5—Gear Manufacturing Technology Course. R.P. Machine Enterprises, Statesville, NC. Perry Technology Co., New Hartford, CT. Instructors Geoff Ashcroft and Ron Greene aim to teach participants gear theory as well as practical aspects of manufacturing and troubleshooting techniques. The course will cover gear inspection, gear manufacturing, gear hobbing, gear shaping, hobs, gear shaping tools, production estimating, hard finishing and gear shaving. $750 includes tuition, materials, an AGMA reference manual and certificate of completion. For more information, contact the Gear Consulting Group at (269) 623-4993.

December 3–7—International CTI-Symposium Innovative Automotive Transmissions. Maritim Hotel, Berlin, Germany. This five-day event focuses on panel discussions and solution forums involving automotive transmissions. Planned programs include transmission requirements, double-clutch transmissions, transmission development and all-wheel-drive. For registration information, contact Gudrun Meixner, customer service, at *gudrun.meixner@informa.com* or visit *www.transmission-symposium.com.*

December 4–5—Human Error Prevention Seminar. Dallas/Fort Worth, TX. This high-technology seminar focuses on implementing techniques to recognize and avoid hazards. Learning objectives include: terminology; the seven human error causal factors; the four levels of human error; the three levels of barriers to human error; techniques by which to strengthen barriers; error-inducing conditions and error-likely situations; behavioral techniques to counteract these conditions and situations; Thought processes and behaviors that lead to non-conservative decisions; thought processes and behaviors that lead to conservative decisions; human error measurement. Registration costs $895. For more information, contact Ben Marguglio at (845) 265-0123 or visit *www.hightechnologyseminars.com.*

December 4–7—2007 International Gear Technology Exhibition China. Chinese Export Commodities Fair, Pazhou complex Guangzhou, China. Exhibits include standard and special gears, gear production and testing equipment, cutting tool, machine accessories and manufacturing services. For more information, visit Gear China 2007 at *www.macomponents.com*.

December 6–7—Problem Reporting, Root Cause Analysis & Corrective Action Seminar. Dallas/Fort Worth, TX. This high-technology seminar focuses on implementing techniques to identify and report problems. Learning objectives include: understand problem reporting, root cause analysis and corrective action terminology; design and implement a management system, or assess the management system for problem reporting, root cause analysis and corrective action; perform or facilitate the performance of root cause analysis; change Analysis; failure mode & effects analysis; hazard-barrier-effects analysis; timeline analysis; cause & effects analysis/fishbone analysis (both process-based and affinity-based); probabilistic risk/safety Analysis, using event and fault trees; and management oversight & risk tree analysis; Understand various models of defense in depth. Registration costs $895. For more information, contact Ben Marguglio at (845) 265-0123 or visit *www.hightechnologyseminars.com.*

January 30–February 1—AGMA Gear Materials Seminar. Location to be determined. The AGMA is offering a class entitled, "Gear Materials: Selection, Metallurgy, Heat Treatment and Quality Control." The class will be taught by Raymond J. Drago, P.E., chief engineer and Roy J. Cunningham, senior metallurgist, Drive Systems Technology Inc. Content will be valuable to a broad range of interests including: gear design engineers and management involved with the design and manufacture of gearing type components; metallurgists and materials engineers; laboratory technicians; quality assurance engineers; furnace design engineers; and equipment suppliers. For more information, visit *www.agma.org*.

May 5–7—Intertech 2008
Contemporary Resort, Walt Disney World, Orlando, FL. Designed to provide a blend of technical and commercial topics, Intertech 2008 hosts global superabrasive suppliers, toolmakers, research organizations and end users to reach professionals involved in all aspects of machining, grinding, drilling, sawing, texturing, polishing, wear parts, diamond windows and wire dies. The 2008 conference primarily focuses on increased productivity, new technological developments and ways to reduce business costs. Please send your abstracts online by January 1, 2008 to Terry M. Kane at *tkane-ida@insight.rr.com*.

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August Manufacturing Technology Consumption UP 26.6%

According to the Association for Manufacturing Technology and the American Machine Tool Distributor's Association, manufacturing technology consumption totaled $347.25 million in August. This total was up 26.6% from July, and up 9.7% from the total of $316.44 million reported for August 2006. With a year-to-date total of $2.6 billion, 2007 was up 6.2% compared with 2006.

These numbers and all data in this report are based on data reported by companies participating in the USMTC program. "Machine tool orders have outpaced last year's results due in part to the decline in the dollar's strength, which is making manufacturing in the U.S. more competitive," says John B. Byrd III, AMT president. "However, it may be difficult to sustain the current growth rate through year end due to the very strong activity during and shortly following IMTS 2006."

According to the company's press release, the United States Manufacturing Technology Consumption (USMTC) report, jointly compiled by the two trade associations representing the production and distribution of manufacturing technology, provides regional and national U.S. consumption data of domestic and imported machine tools and related equipment.

U.S. manufacturing technology consumption is also reported on a regional basis for five geographic breakdowns of the United States.

Northeast Region: August manufacturing technology consumption in the Northeast Region totaled $50.19 million, 29.1% higher than July's $38.88 million and up 7.5% when compared with August 2006. At $421.03 million, the 2007 year-to-date total was 15.4% higher than the comparable figure for 2006.

Southern Region: Manufacturing technology consumption in the Southern Region in August rose to $46.52 million, up 7.4% from July's $43.33 million and 3.8% more than the total for August last year. At $349.36 million, the 2007 year-to-date total was 4.1% higher than the comparable figure for 2006.

Midwestern Region: At $105.45 million, Midwestern Region manufacturing technology consumption in August was up 18.2% when compared with July's $89.23 million, and 30.6% higher than the August 2006 total. The $754.53 million year-to-date total was down 0.2% when compared with the 2006 total at the same time.

Central Region: Central Region manufacturing technology consumption in August totaled $99.97 million, up 42.2% from July's $70.31 million and up 14.6% when compared with last August. The year-to-date total of $708.63 million was 13.4% more than the comparable figure in 2006.

Western Region: At $45.11 million, Western Region manufacturing technology consumption in August was up 38.2% when compared with July's $32.63 million, but 20.9% less than the tally for August a year ago. Compared with 2006 at the same time, the year-to-date total of $375.86 million was up 0.2%.

Wenzel

OPENS NORTH AMERICAN FACILITY IN WIXOM, MI

Wenzel GmbH of Germany, a large manufacturer of coordinate measuring machines (CMMs), opened a new 24,000-square-foot headquarters and manufacturing facility in Wixom, MI on October 9, 2007. The two-story building houses all of Wenzel's North American administrative and engineering offices.

"The North American operation, known as Wenzel/Xspect Solutions, becomes our third worldwide manufacturing center, complementing what we currently have in Europe and Asia," says Frank Wenzel, managing director. "In 2006, Wenzel produced and sold over 500 machines worldwide and currently has over 5,000 CMMs in operation around the world, as well as over 10,000 software licenses."

The dedication ceremony preceded a two-day open house for current and prospective Wenzel/Xspect Solutions customers. The proceedings included a demonstration and display of the Wenzel X-Checker shop floor CMM, the X-Cite CMM and X-Measure, a new, specially developed user interface for OpenDMIS software.

Powder Metallurgy Market

EXPECTED TO APPROACH 30 BILLION BY 2012

According to a recent study by Materials Technology Publications, the powder metallurgy market should approach $30 billion by 2012, growing from $23 billion in 2007 at an average growth rate near 5%. The market study entitled, "The Powder Metallurgy Industry Worldwide 2007–2012," features 450 pages with more than 100 tables of statistical data.

In the report, statistical data finds that the North American and European markets will lose shares to China, India and other Asian countries over this period. The North American powder metallurgy parts business, worth $5 billion in 2007, will grow

continued

to only $5.5 billion in 2012. The $9.5 billion European market is expected to reach $11.6 billion in 2012. In contrast, the Asian market, estimated at $7.62 billion in 2007, is expected to increase to $12.6 billion in 2012.

As the PM markets mature in North America, Western Europe and Japan, the industry will look to expand into other geographic regions in China, India and Eastern Europe.

The Powder Metallurgy Industry Worldwide 2007–2012 is available as a printed version or an electronic version on CD. The cost of the report is $2,600. For more information contact Ted Giese at *info@powdermetallurgymarket.com.*

Powder Industry

OPENS INTERNATIONAL DESIGN COMPETITION

The Metal Powder Industries Federation (MPIF) has opened the 2008 International PM Design Excellence Awards Competition in an effort to recognize outstanding achievements in the commercial production of powder metallurgy components.

Categories for the competition include automotive, aerospace, lawn and garden, industrial motors, hardware, medical/dental and electronic/electrical. Members of the MPIF and their customers may submit entries for the competition. Judging guidelines are based on design configuration, engineering properties and promotional value to the PM industry.

Winners will be announced at an awards luncheon during the 2008 World Congress on Powder Metallurgy & Particulate Materials, June 8–12 in Washington D.C. Entries must be received no later than January 31, 2008. For entry forms, contact MPIF at 105 College Road East, Princeton, New Jersey 08540-6692 or by phone at (609) 452-7700.

Manufacturing Economist

DISAGREES WITH FEDERAL RESERVE REPORT

A recent report from the Federal Reserve that stated manufacturing production showed no change in September of 2007 is not accurate, according to industry insiders. David

Huether, chief economist of the National Association of Manufacturers, believes the report does not reflect what's currently going on in the industry.

"While overall manufacturing activity was flat, there's an awful lot of activity under the surface," says Huether. "The ongoing downturn in housing continues to impact some segments. Also, the disruptive strikes in the auto industry were a major cause of the 3.3 percent drop in motor vehicle production. But outside motor vehicles, manufacturing production rose a solid 0.3 percent in September."

Huether believes export growth could offset the current housing recession. "Encouraging exports by lowering overseas trade barriers would be a no-brainer," says Huether. "Currently, there are four free trade agreements waiting to be approved by Congress. We hope that Congress will act in the interest of American's manufacturing base and pass them."

Like Father, Like Son

AT DERITEND

Mark Brady

Deritend Industries recently announced the appointment of Mark Brady as operations director at RMB Engineering. Brady will be taking over for his father, Ray, who is stepping down to take a position on the board of directors. Mark Brady has learned the business from the grass roots and has many years' experience meeting the 24/7 supply demand in industries like quarrying, mining, power generation and water and steel.

"It is a really exciting time to be taking over," says Mark Brady. "My task will be to drive operational improvements at all levels to ensure we maintain the lead we have over our competitors. Many companies advertise themselves as providing 24/7 service, but few actually provide it."

RMB Engineering is an industrial gearbox supply and repair organization in the U.K. They recently integrated the company into Deritend Industries, where it has more facility space for future expansion.

October Report

LOOKS POSITIVE FOR METALFORMING INDUSTRY

According to the October 2007 Precision Metalforming Association (PMA) Business Conditions Report, business conditions will dip slightly during the next three months. The monthly report is an economic evaluation that samples over 159 metalforming companies in the United States and Canada.

"PMA members are considerably more positive today than they were one year ago, as reported in our October 2006 Business Conditions Report," says William E. Gaskin, PMA president.

Though metalforming companies expect incoming orders to decrease slightly during the next three months, the number of companies with a portion of their workforce on short time or layoff fell to eight percent in October, down from 11% in September. It's currently at the lowest level since April of 2006.

"While expectations for current shipments are only modestly improved over one year ago, expectations for new orders over the next few months and the overall assessment of the general economy are substantially more positive than last year," says Gaskin. For a complete copy of the monthly conditions report, visit *www.pma.org/about/stats/BCreport*.

American Axle

FORMS JOINT VENTURE WITH SONO KOYO

American Axle and Manufacturing Holdings Inc. recently announced the formation of a joint venture with Sona Koyo Steering Systems Limited of India. The new company will manufacture and sell light truck, passenger car and SUV axle assemblies as AAM Sona Axle Private Limited. The corporate headquarters, expected to open in 2008, will be located in Pune, India.

"AAM Sona Axle gives AAM a manufacturing and operations presence in India with the ability to provide current and future customers with driveline systems with the highest quality, technology and delivery that are the hallmark of AAM and Sona Koyo," says Richard E. Dauch, AAM co-founder, chairman and chief executive officer.

Currently AAM manufactures driveline and drivetrain systems in the United States as well as overseas. Sona Koyo supplies axle assemblies for Suzuki Maruti in India.

Chicagoan

RECOGNIZED FOR WORK ADVANCING TECHNICAL EDUCATION

Dan Swinney, executive director of the Chicago Manufacturing Renaissance Council (CMRC), was chosen as one of three individuals nationwide to receive the National Career Academy Coalition's (NCAC) award for creating and fostering strategic business partnerships in support of enhanced learning choices. Swinney received the award at the NCAC's 11th annual convention, held November 5 at the Hyatt Regency Hotel in St. Louis.

Daniel Swinney

Swinney was a leading force in the creation of Austin Polytechnical Institute, a Chicago public school located on the city's West Side with a core curriculum that promotes and nurtures potential careers for young people in engineering and high-skilled manufacturing. The school, now in its first year, was made possible through the enthusiastic support and participation of 26 small- to medium-size local manufacturers who provide mentoring, internships and career opportunities for the school's students and graduates.

"As the project manager for Austin Polytech, Dan has assembled a stellar coalition of business, labor and government partners to make this school a success," says Sue Klonsky, an Austin Polytech Design Team member. "Reversing 'the race to the bottom,' this new school represents an educational pathway to the top."

Swinney also founded in 1982 the Center for Labor and Community Research (CLCR) in Chicago. The center's 2001 study—in conjunction with the Chicago Federation of Labor—Creating a Manufacturing Career Path System in Cook County (IL), was at the time considered a groundbreaking achievement. The study in fact served as the impetus for the CMRC's creation in 2005.

The NCAC is the most influential association for public ca-

reer academies in the country, and provides assistance to public schools with programs specifically devoted to promoting careers in every sector of the economy.

Moventas

APPOINTS GENERAL MANAGER FOR CHINA


Yang Xilin

Yang Xilin has been appointed general manger of Moventas' Industrial Gears division in China effective November, 1, 2007. In addition to his role as general manager, Xilin will be responsible for business operations and act as a chief representative officer of Moventas in Shanghai. He has extensive experience in mechanical power transmission due to previous positions at Santasalo and SEW Eurodrive.

He will report to Jari Ainali, executive vice president at Moventas Industrial Gears.

Tower Tech

ANNOUNCES ADDITIONS TO MANAGEMENT TEAM

Tower Tech Holdings Inc., a Wisconsin-based manufacturer of components for energy and infrastructure-related industries, recently announced the addition of two employees to their management team. Lars Moller has joined as executive vice president of business development and Matthew Gadow as executive vice president of strategic planning.

Moller has more than 20 years experience in the wind industry with companies like DMI Industries, Vestas Americas, Difko and BONUS Energy. Gadow has worked for DMI Industries and is a certified public accountant.

"We believe Lars' and Matt's experiences are unparalleled in component manufacturing for the wind industry," says J. Cameron Drecoll, chief executive officer of Tower Tech. "We look forward to their leadership in developing relationships with our current customers and attracting new wind industry customers."

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Seriously, folks, Gear Technology's

2007 Holiday Buyer's Guide

We love gears. We love talking about gears, writing about gears and examining gears. If you're reading this cover to cover, it's a safe bet you feel the same way. We also love collecting information for *Gear Technology*'s holiday buyer's guide. Call us sentimental.

Will the 2007 holiday season be another year of scented candles, coffee-table books, sweaters and tube socks? Not if we can help it. Those in the know can turn the holiday season upside down with these gear-related T-shirts, toys and other goodies sure to please every member of the family:

"From stocking stuffers to last minute gift ideas, the Addendum Staff has you covered."

Playskool Busy Basic Gears: The wee ones can enjoy a hands-on exploration with this play set featuring electronic gears, flashing lights and songs. If it makes too much noise, the Addendum staff isn't opposed to wrapping it up and giving it to your brother's or sister's kids come Christmas morning. $14.99 and available at *www.hasbrotoys.com.*

Wooden Clock Kits: Want a project at home that doesn't involve shoveling snow this winter? The Serpentine Clock Kit allows you to construct your very own wooden gear clock with just a screwdriver, hammer, wood glue and sandpaper. Order online at *www.wooden-gear-clocks.com* for $149 plus shipping and handling (free time it takes to complete project not included).

Lego Gears Set: This basic introduction to gearing, for ages 8-12, includes 67 elements like gear wheels, axles and beams as well as two full-color building instruction guides. Have your children pause their video games for a day and try building something from scratch with the greatest building blocks on the market. $17.36 and available at *www.homeschoolingsupply.com.*

507 Mechanical Movements: The official title of this 2006 publication is *507 Mechanical Movements in Dynamics, Hydraulics, Hydrostatics, Pneumatics, Steam Engines, Mill and other Gearing*. Now say it five times fast. A great read for anyone interested in engineering, mechanics and the way things work. A better read if you spend your free time actually contemplating such topics. $13.95 and available at *www.amazon.com.*

Gear-Related Merchandise: Ever wish you had a gear-themed T-shirt? How about a fancy gear messenger bag? The fine people at cafepress.com allow shoppers to choose any gear-related logo and put it on boxer shorts, T-shirts, bags, bibs—even a BBQ apron. Choose from several different logos by visiting *www.cafepress.com* (keywords "engineering" and "gears").

Gearrings: According to the website, Martha Stewart has finally met her match. These hand-crafted earrings are made from colorful glass beads and FAA-approved aircraft hardware. Choose from 11 styles and colors starting at $16. Visit *www.womenfly.com* for more information.

Gears Gears Gears Lights & Action Building Set: New from Learning Resources is this 121-piece set, for ages 5-10, that includes a power motor, flashing lights and glow-in-the-dark stickers. Pieces snap together to let children easily build anything they can imagine. $49.95 by calling (800) 333-8281 or at *learningresources.com.*

123 Robotics Experiments for the Evil Genius: If you've ever wanted a solid foundation in robotics, electronics and programming, let this be your guide. Published in 2004, this instructional reference book includes a printed circuit board to give a hands-on understanding of the robotics industry. Prepare for world domination at the low price of $24.95 by visiting *www.giftsforengineers.com.*

Refrigerator Gear Magnets: This engaging kit for budding engineers offers 20 plastic gears in assorted colors motorized by a picture frame that runs on AA batteries. The gears interlock in various patterns and turn all by themselves. Toss the magnet up on the fridge and be the envy of everyone else in your kitchen. $24.95 at the American Science & Surplus store, visit *www.sciplus.com* for more information.

So there you have it. From stocking stuffers to last minute gift ideas, the Addendum staff has you covered. They said Mom was impossible to shop for and Dad had everything he needed. We hope we proved them wrong. Until next year, **Happy Holidays**!

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