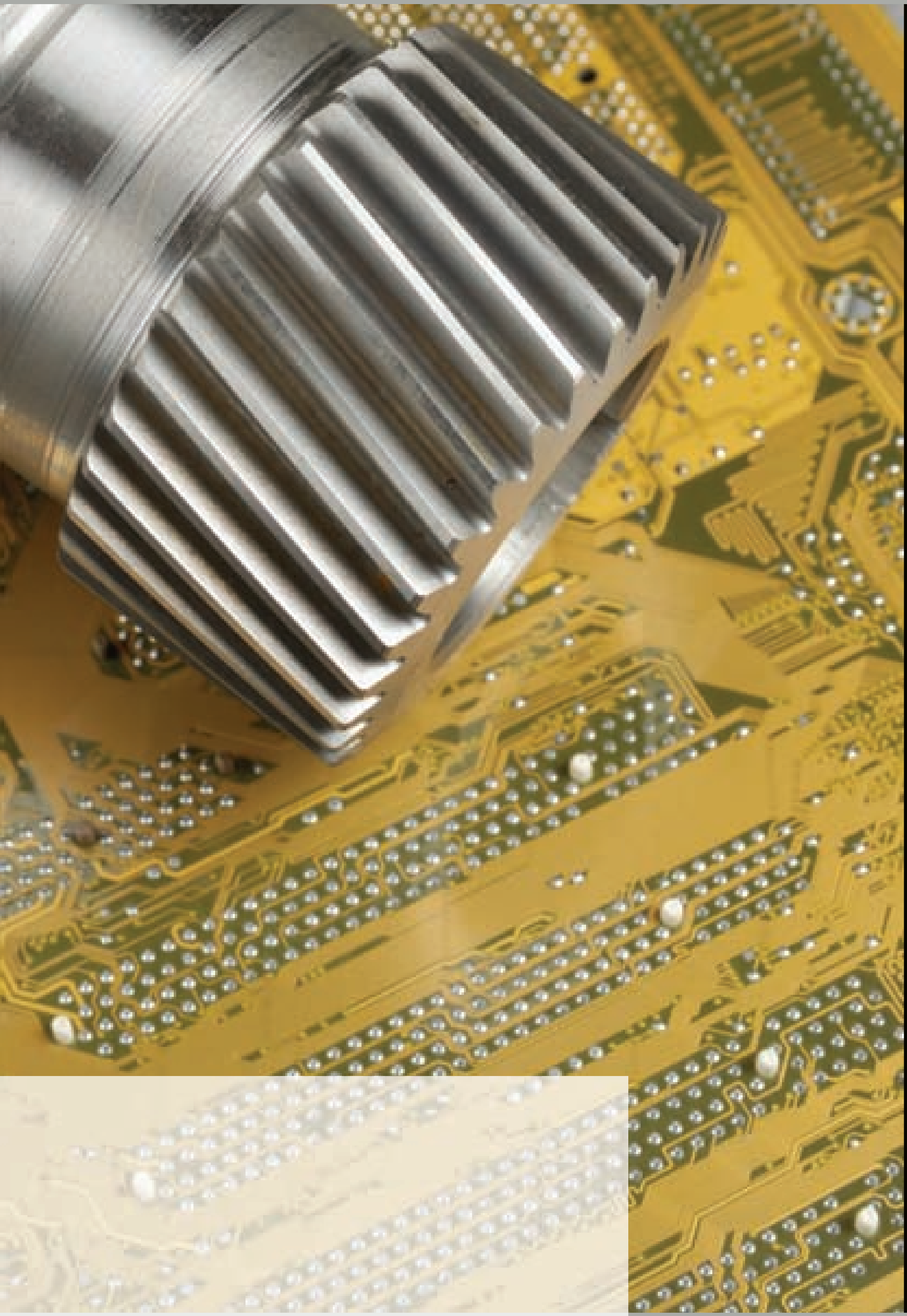
GEAR TECHNOLOGY

March/April 2008

www.geartechnology.com

The Journal of Gear Manufacturing



Features

- The U.S. Needs More Engineers
- Gear Software
- Repairing the Space Station

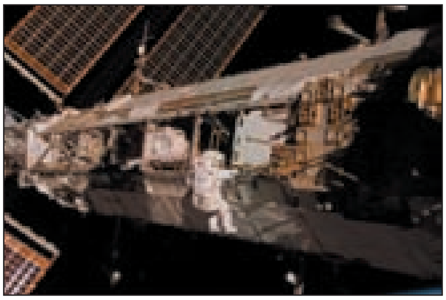
Technical Articles

- Translating Single-Tooth Bending Fatigue
- Pitting and Bending Fatigue Evaluations
- Superfinishing Effect on High-Speed Helical Gearing

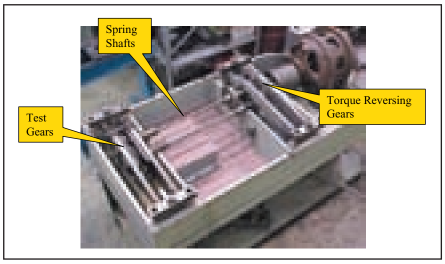
THE GEAR INDUSTRY'S INFORMATION SOURCE

CONTENTS

FEATURES



TECHNICAL ARTICLES



Vol. 25, No. 2. GEAR TECHNOLOGY, The Journal of Gear Manufacturing (ISSN 0743-6858) is published monthly, except in February, April, October and December by Randall Publishing, Inc., 1425 Lunt Avenue, P.O. Box 1426, Elk Grove Village, IL 60007, (847) 437-6604. Cover price $7.00 U.S. Periodical postage paid at Arlington Heights, IL, and at additional mailing office (USPS No. 749-290). Randall Publishing makes every effort to ensure that the processes described in GEAR TECHNOLOGY conform to sound engineering practice. Neither the authors nor the publisher can be held responsible for injuries sustained while following the procedures described. Postmaster: Send address changes to GEAR TECHNOLOGY, The Journal of Gear Manufacturing, 1425 Lunt Avenue, P.O. Box 1426, Elk Grove Village, IL, 60007.

CONTENTS

DEPARTMENTS

ON LINE

GEAR TECHNOLOGY.com
www.geartechnology.com

- **BUYERS GUIDE**
  Search for gear industry products and services and communicate with the industry's leading suppliers.
- **SUBSCRIPTIONS**
  Sign up for subscriptions to *Gear Technology* and the *geartechnology.com* e-mail newsletter.
- **E-GT**
  Subscribers get access to back issues of *Gear Technology.*

powertransmission.com
www.powertransmission.com

- ***Power Transmission Engineering***
  Read the February 2008 issue online. Sign up for a free subscription to our new magazine.
- **NEWS**
  The latest products, industry news and calendar items.
- **BUYERS GUIDE**
  Search for power transmission products and services and communicate with the industry's leading suppliers.


Flexible Technology!
SIGMA POOL

GEAR TECHNOLOGY

VOL. 25, NO. 2

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

Assistant Editor Lindsey Snyder
lsnyder@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

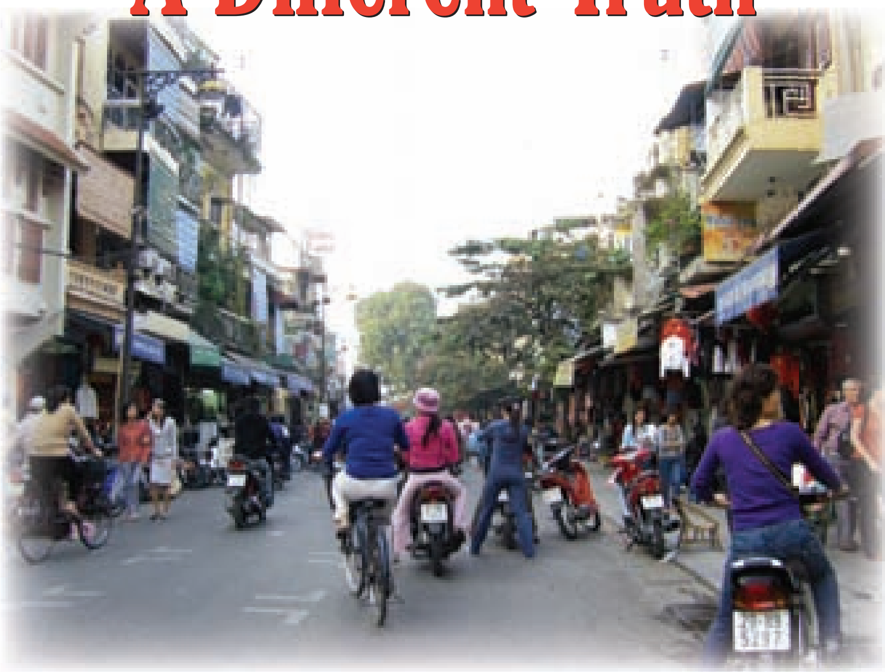
CIRCULATION

Circulation Manager Carol Tratar
subscribe@geartechnology.com

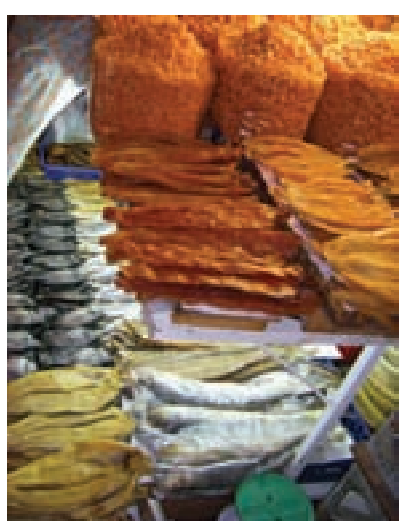
RANDALL PUBLISHING STAFF

President Michael Goldstein
Vice President Richard Goldstein
Accounting Luann Harrold

A Different Truth



Transportation in Vietnam consists mainly of two-wheeled vehicles. This relatively uncrowded photo doesn't do justice to the sheer chaos of motorbikes normally found on the streets.



The colors, flavors, sights and sounds of the marketplaces in Vietnam overwhelm the senses with varieties of fruits, nuts, meats and more that you didn't even know existed.

Vietnam is the war Americans don't like to talk about. Even today, many of us struggle to understand the what and the why of that war. We remember the protests, the atrocities, the way our soldiers were treated. Even as wars go, it was an ugly war. It's also the one we lost.

Many of you are probably about my age. That means you either had to fight in that war, or you knew people who did—or, like me, you were old enough, but lucky enough not to have to go. Even the younger generations, who have learned about Vietnam from history class or movies like *Apocalypse Now*, carry some of the same emotional angst.

Having lived through that era, I thought I had a pretty good understanding—at least in general—about what went on there, based on what our government was telling us. I watched the evening news. I read the newspapers. I saw the protests.

Recently, though, I actually visited Vietnam for the first time. Before the trip, I felt a lot of unresolved emotions, anxiety relating to my feelings about a war that ended more than 30 years ago.

How would the Vietnamese people react to an American visiting today? How would I feel standing on that soil where so many kids came to die?

I took a lot of cultural baggage with me on the trip, but I'm glad to say, after visiting, I left at least some of it behind.

On the whole, I found Vietnam to be a beautiful country. Open-air markets inundated my senses with the colors, textures, smells and noises of people buying and selling everything from fish to fruit to flowers to fabric. Hanoi provided a wonderful mixture of cultures, with a strong French influence. And everywhere I went, jungle-covered mountains dominated the landscape.

Despite all the beauty, though, I have to admit that in many places, especially outside the cities, I found it hard not to hear the thumping of helicopters in my head. At times, I swear I could hear Jim Morrison singing "The End" in the background. Some associations are hard to break.

But the more time I spent there and the more Vietnamese people I met, the more my anxiety dissipated. To the Vietnamese, the Americans were just the latest in a long line of occupying forces that had eventually been

Continued

kicked out or otherwise convinced to leave. To them, the past is in the past, and I never felt as though any of the people there held a grudge. That in itself made me feel easier about being there.

The Vietnamese attitude also made it much easier for me to see and understand things from their perspective.

A good example was my visit to the Cu Chi Tunnels outside Saigon. The tunnels were originally dug to fight the French. During the American war, the Viet Cong used them as a base for ambushing American and South Vietnamese troops. More than 150 miles of very narrow, heavily boobytrapped tunnels wind through the countryside.

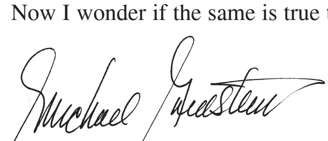
Prior to my trip, I knew that the tunnels were one of the devices the Viet Cong used to frustrate our military efforts there. But what struck me about the tunnels wasn't their tactical significance in the war. It was the fact that people actually lived in these things. Babies were born and families survived down there, with very poor air quality, among the dirt and death traps.

Many of the other places I visited had some relevance to the war, but everywhere I went, I learned a different perspective. For example, the Vietnamese have their own views about what really happened at the Gulf of Tonkin and why Ho Chi Minh sought the support of the communists in the first place.

According to our tour guides, Ho Chi Minh and the Vietnamese just wanted help overthrowing their French imperial overlords. The communists provided that help. From their perspective, communism was a means to freedom, which makes America's efforts there seem all the more ironic. Even today, under a communist government, Vietnam is one of the most entrepeneurial, fastest growing countries in Southeast Asia, with enormous amounts of private enterprise. The ideology we once feared may never have been as important in Vietnam as we imagined.

I'm not sure my trip helped me resolve all my feelings about the Vietnam War. Even by the end of the trip, I was still hearing helicopters. But going there and hearing the perspectives of the Vietnamese people gave me insight I didn't have before.

Many people have drawn parallels between the Vietnam War and America's current involvement in Iraq. I went to Vietnam thinking I knew some things about the war. I came back with the understanding that the truth isn't as easy to understand as I thought. Now I wonder if the same is true today.



Michael Goldstein,
Publisher & Editor-in-Chief

Despite the peaceful beauty of much of the country, it was hard not to imagine the thumping of helicopters when viewing scenes like this.

The Cu Chi Tunnels outside Saigon were too small for most to enter. The narrow entrances and air holes disguised as termite mounds give an idea about the difficulty of the conditions the Viet Cong endured. The sharp spikes of the traps inside and outside the tunnels give an idea why the tunnels proved so difficult for enemy forces to overcome.

Approximately 1,600 small islands and islets make Ha Long Bay in the Gulf of Tonkin one of the most beautiful areas in Vietnam. The area is a UNESCO World Heritage site.

PG-Series Bore Gages

MOUNT DIRECTLY TO MACHINES

With no need for electrical or air connections, Sunnen's PG-Series bore gages can be mounted directly to machining stations, making them immune to fluctuations in line voltage or air pressure. The unique fixtures enable a two-minute setup, eliminating the need for plug gages, mikes, Jo blocks, master rings or micrometers.

According to the company's press release, the PG-Series bore gages remove complexity, skill and "feel" from the measurement process, allowing machine operators to confirm part size with a visually intuitive analog scale. The speedometer-type scale provides readouts down to 0.000050" for inspection of machining operations or fast sampling of large vendor lots.

The gages are available for shops of all sizes and can be purchased outright or leased on a monthly basis directly from Sunnen. The lease program includes maintenance and system updates. Models are available for gaging IDs from 0.090" to 4.310". Metric models cover IDs from 2.0–109.47 mm.

The PG-Series bore gages can examine the entire bore for diameter, taper, barrel, bell mouth, out-of-round and lobbing. A sliding faceplate with adjustable stops permits end-to-end examination of bores. Stops can be set to examine particular sections of the bore.

Accuracy is not affected by temperature, moisture, air pressure or vibration. The floating reed system helps to eliminate friction and wear to maintain repeatability and linearity, no matter the gage's age or amount of use. A PC-Series bore gage system comes standard with everything needed to measure any bore size in its range. Accessories are available for special requirements.

For more information:

Sunnen Products Company
7910 Manchester Ave.
St. Louis, MO 63143
Phone: (314) 781-2110
Fax: (314) 781-2268
bdavis@sunnen.com
www.sunnen.com

VMC Infimatic Control

COMPATIBLE WITH STANDARD G CODES AND PART PROGRAMS

MAG Fadal recently announced its FX series will be available April 2008 with the Infimatic Freedom NC2000, a digital control for metal-cutting machine tool applications. The control is compatible with standard G codes as well as part programs created for Fadal CNC controls.

According to the company's press release, the operator pendant offers a full keyboard, integrated mouse and LED-ringed pushbuttons. VeriCode, the system's color-coded editor, boasts real-time syntax and semantic verification. It also has the ability to facilitate background editing during the machining process.

Special editions of both the VMC 3016 FX and the VMC 4020 FX are being offered on a limited basis prior to the April 2008 release. These machines include a 10,000 rpm air/oil 40-taper spindle, coolant thru-spindle feature, dual-arm tool changers with 24-tool capacity on the 4020 and 20-tool capacity on the 3016.

Both versions feature box-way construction with integral flame-hardened ways complemented by Steinmeyer ETA+ dual-mounted ball screws. Rapid traverse is 1,000 inches per minute with a cutting feed rate up to 800 inches per minute.

For more information:
Fadal Machining Centers, LLC
20701 Plummer Street
Chatsworth, CA 91311
Phone: (818) 407-1400
Fax: (818) 407-0020
www.fadal.com

FA Advance Wire EDM Series

EXTRACTS DATA DIRECTLY WITHIN CONTROL

MC Machinery Systems, Inc. introduces its new FA Advance Wire EDM series featuring an M700 series control. The Windows-based system with a 15" LCD screen provides a menu for easy navigation, according to the company's press release.

The 3D CAD feature can now import Parasolid files, extract 3D model contours and create NC data directly within the control. 2D CAM helps import DXF and IGES files, revise geometry or define additional paths. The series has the world's first 3D-adaptive EDM control and onboard support is now standard on the machine. A cost/save mode reduces consumable and electrical costs, and a self-cleaning device feature helps reduce maintenance times.

For more information:

MC Machinery Systems, Inc./
Mitsubishi EDM
1500 Michael Drive
Wood Dale, IL 60191
Phone: (630) 616-5920
www.mitsubishi-world.com

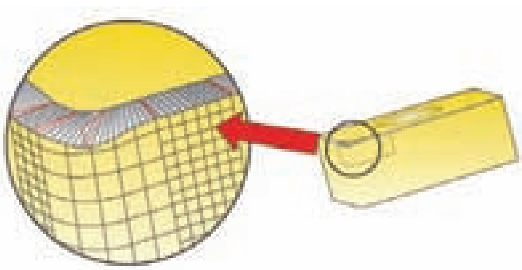
Helix Wiper Geometries

REDUCE VIBRATIONS AND INCREASE TOOL LIFE

Seco Tools Inc. recently unveiled Helix wiper geometries for operations in high-feed machining of case hardened steels where standard wipers cannot be used. According to the company's press release, the Helix concept is designed to reduce vibrations, increase tool life and increase the number of parts produced per cutting edge. It has wipers on both sides of the corner radii that are standard, but the protection chamfer is twisted from negative to positive or positive to negative.

Seco's wiper geometries enable the operator to obtain a smooth surface by lessening the effect of the workpiece feed pattern created during conventional turning. Helix wipers are available in brazed format CBN050C or a coated low-CBN grade.

"Reduction in production time along with the near doubling of tool life equates directly to significant savings in total production costs," says Henrik Sandqvist, Seco Tools Inc. product manager of advanced materials. "The result is better than that of conventional case-hardened steel finishing where slower speeds and feeds are typical and additional grinding processes are often required."



For more information:

Seco Tools
11177 East Eight Mile Rd.
Warren, MI 48089
Phone: (586) 497-5000
Fax: (586) 497-5627
secotools.us@secotools.com
www.secotools.com

Digimar Height Gage

PROVIDES SMOOTH MOVEMENTS FOR POSITIONING

The Digimar 817 CLM height measuring instrument from Mahr Federal will be featured at WESTEC 2008 from March 31–April 3. The gage offers three distinct ways to initiate measurement. Along with normal keypad initiation, the quick mode feature allows the cycle to be initiated by simply pushing the carriage in the direction of the object. Speed keys allow the operator to move the carriage to the desired position to start a measurement. Combined with the teach-in mode, these features can reduce inspection time for a miniseries consisting of only a few test pieces.

According to the company's press release, the gage offers an air bearing system facilitating smooth movement for positioning. It boasts an optical incremental measuring system with a double reader head that's insensitive to dirt. The probing system provides high repeatability, and a stainless steel guide column with precision ball bearings tracks the measuring head.

The Digimar 817 CLM can function in both 1D and optional 2D, including dynamic measurement functions with analog display such as max, min and max/min. Data can be printed via a USB-connected printer and saved on integrated USB memory or output using a RS232-interface. Mahr Federal's MarCator 1081 digital indicator and MarCator 1088 digital indicator will also be on display at WESTEC 2008.

For more information:
Mahr Federal Inc.
1144 Eddy Street
Providence, RI 02905
Phone: (401) 784-3100
Fax: (401) 784-3246
information@mahr.com
www.mahr.com

SafeTap Plus

PROVIDES SAFE AND CLEAN WORKING ENVIRONMENT

ITW Rocol recently unveiled a new water-based tapping fluid with a custom blend of synthetic additives that provide cooling properties with the viscosity of a straight oil. According to the company's press release, SafeTap Plus gives higher viscosity and lubricity than many oil-based products while also providing rust protection. This, in turn, creates a potentially safer and cleaner working environment for employees. The fluid contains no mineral oils or solvents and leaves no oily residue on work pieces. SafeTap Plus is available

continued

in 4- or 16-ounce bottles as well as 1-, 5- and 55-gallon containers.

For more information:

ITW Rocol North America
3624 West Lake Ave.
Glenview, IL 60026
Phone: (800) 452-5823
Fax: (800) 952-5823
egibbons@rocolnorthamerica.com
www.rocolnorthamerica.com

LED Light Ring

ILLUMINATES WITHOUT HEAT OR FIBER OPTIC CABLES

NovaTech, Inc. has introduced an LED light ring for laser and micro-welding microscopes that provides illumination without heat or fiber optic cables. According to the company's press release, the universal ring light features 60 LEDs and includes a 110 VAC power supply with variable brightness. The lamp's color temperature is 6400K with a rated lamp life of 17,000 hours. An adaptor that fits most Olympus SZ and Nikon SMZ fixed microscope heads is supplied.

For more information:
NovaTech, Inc.
345 Farnum Pike
Smithfield, RI 02917-1205
Phone: (800) 442-6682
Fax: (401) 233-0098
Ralph@novatechinc.com
www.novatechinc.com

alpha & Wittenstein

INCREASE MEASUREMENT ACCURACY

New sensor technology from alpha & Wittenstein integrates torque and force sensors into the housing of servo gear reducers, enabling measurements to be more precise. The alphaIQ loads sensors to the gearbox, which allows all mechanical loads to be processed; this includes torques, shear forces, speed, thermal loads, wear and backlash. Computer software comes with the product, so the variables can be displayed on a PC.

For more information:
alpha gear drives
1249 Humbracht Circle
Bartlett, IL 60103
info@alphagear.com
www.alphagear.com

VAC-U-MAX Vacuum Cleaning System

EXTRACTS FLUID AND SOLIDS

Powered by compressed air, the VAC-U-MAX 55MW filters liquids and removes metal chips from coolant in order to increase coolant life and extend machine tool filtration systems. Operators determine where and when fluid comes out; as a result, according to the company's press release, there is less exposure to bacteria and dermatitis that can result from emptying machine tool sumps, machining beds, oil/water separators, parts washers, rinse tanks and cleaning up liquid spills. The 55MW is 26" wide, allowing the unit to be up close to machine tools, minimizing vacuum hose lengths. The hoses are designed to prevent accumulation of liquids and chips, and the cart base has slotted holes to avoid pooling of fluid.

For more information:
VAC-U-MAX
37 Rutgers Street
Belleville, NJ 07109
Phone: (800) 822-8629
info@vac-u-max.com
www.vac-u-max.com

Bodine Motors

COMBINE CONTROL, MOTOR AND GEARHEAD

Bodine Electric Company expands the INTEGRAmotor product line to integrate electronic and mechanical motors into the same unit. Combining a speed control, brushless DC motor and gearhead into one product lowers engineering time and assembly costs while simplifying the wiring complexity and reducing panel space and the risks of electromagnetic interference, according to the company's press release.

The new INTEGRAmotor gearmotors have a larger frame size that can be operated from a regulated 24V DC power supply, as well as an on-board PWM (pulse width modulation) control accepting inputs from an external motion controller or programmable logic controller (PLC). The control also has amplifier enable, direction input and dynamic braking. Bodine developed the new line for office equipment, packaging machines, conveyor systems, medical equipment, printing machinery and factory automation applications.

The company is responding to a growing industry demand for manufacturers to consolidate product designs and minimize the component parts. "Our experience with custom engineered control and gearmotor solutions helped us a great deal when we looked at integrating motor and control elements into one enclosure. We already knew how to work within tight limits of power consumption and thermal performance," says Mike Marhoefer, manager of brushless DC technology and control development.

For more information:
Bodine Electric Company
2500 West Bradley Place
Chicago, IL 60618
Phone: (773) 478-3515
Fax: (773) 478-3232
info@Bodine-Electric.com
www.bodine-electric.com

N.A. Woodworth's Gear Chucks

CHANGE QUICKLY

The Quick-Pitch Diaphragm PLD chucks from N.A. Woodworth include quick-change, precision top jaw tooling sets. The new product is designed for gear manufacturing, grinding and hard-part processing where low- to medium-volume runs are typical.

continued

The chucks offer 60-second changeouts for part-to-part changeovers and can be used in applications with sun, ring, pinion and output gears with spur and helical gear tooth forms, according to the company's press release. Four chuck units are available among three-jaw versions with pitch diameter capacities of 25–100 mm, 100-170 mm, 170–245 mm, or six-jaw units with 245–350 mm gear pitch diameter range.

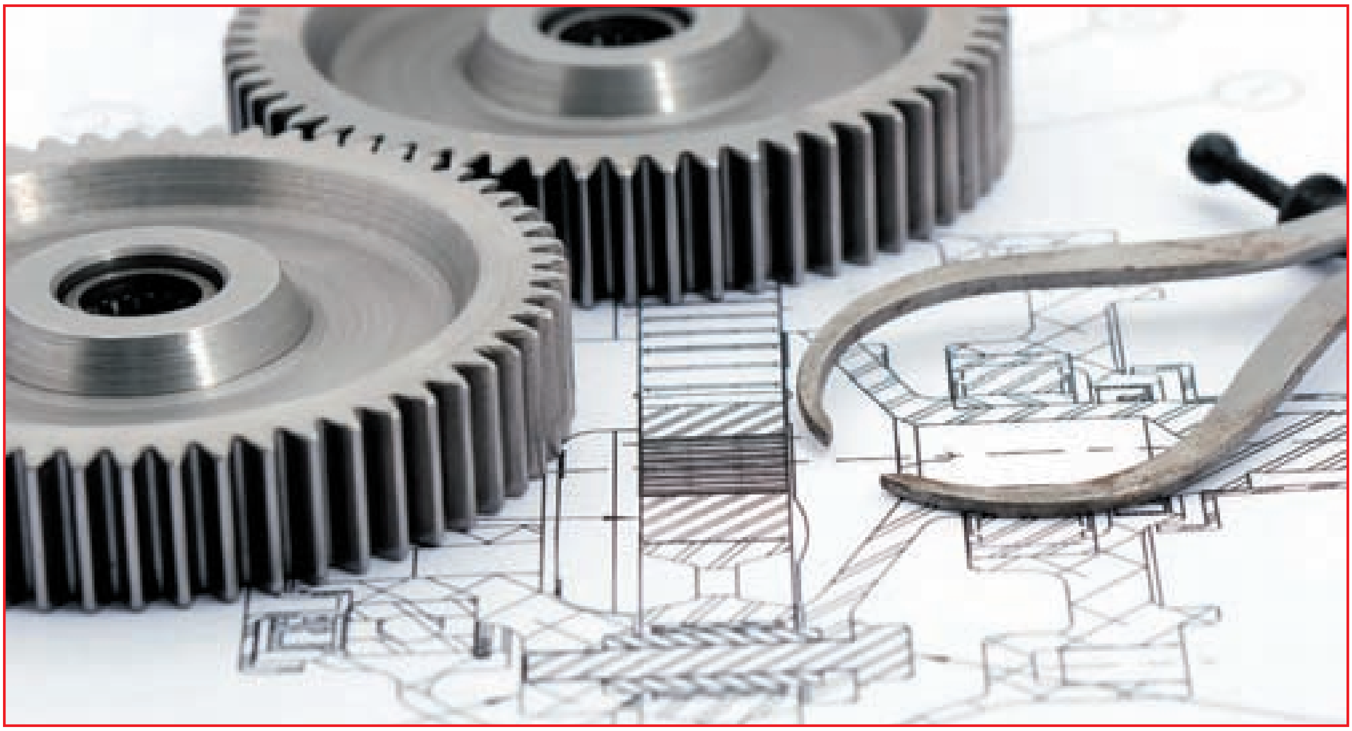
The Quick-Pitch Diaphragm chuck uses a locating wedge pin design, so every pin set in a jaw snaps into place, locating the individual pins' locations relative to the master jaw set. Two

clamp screws hold the jaw in place, allowing the chucks to be changed over without any adjustments. The operating system has a pull-back action to fixed stops, so workpiece position is more accurate and repeatable, and a centrifugal force compensation feature sustains gripping force for safety and accuracy and negates the effects of running at higher machining speeds. The chucks are designed with weight-saving engineering ideas to reduce any potential damage to machine tool and grinding system spindles and bearings.

A sealed design is used in the Quick-Pitch Diaphragm chucks to prevent contamination from harsh work environments where grinding sludge and coolants can be harmful. With the diaphragm component, the need for sliding parts that are easily polluted is eliminated. Both these features result in less maintenance for the chucks. Autoload options are available using synchronizing pins available from N.A. Woodworth.

For more information:

N.A. Woodworth
1391 Wheaton, Ste. 700
Troy, MI 48083
Phone: (800) 544-3823
Fax: (248) 743-4401
sales@itwworkholding.com



The U.S. Needs More Engineers

STATE SCHOOLS LACK FUNDING. WHO LOSES? *WE ALL DO.*

Jack McGuinn, Senior Editor

From the boardroom to the employee lunch room, discussion continues unabated in the world of manufacturing and elsewhere regarding the outsourcing of engineering capabilities and a host of skilled positions that go unfilled. All of which typically lead to the questions, why don't we make things here anymore? Or perhaps more accurately, who cares?

But perhaps something even more sinister is at work here.

Suspend your disbelief and consider for a moment the fall of the Roman Empire. You recall from your history classes that the proud Romans, had become collectively fat and lazy. By the fourth century, they were relying more and more on slave labor and mercenaries to do the work they no longer cared—or forgot how—to do. Very long story short, we all remember how that worked out.

Sound familiar? If you think it's a stretch, think again about the owner of a gear company or job shop, for instance, in need of hiring young gear designers or skilled machinists. Much has been reported—here and elsewhere—on the lost generations of potential mechanical engineers coming out of school over the last 30 years or so who have instead chosen to find work designing video games or iPods. Understand, it is not that "kids today" don't want to make things; it is that they simply don't seem to have much of an interest in designing and seeing built the space shuttles, bridges, locomotives and other essential needs of today and tomorrow.

And so given the state of U.S. manufacturing today, it may come as a shock to learn that many state-funded universities across the country are now charging student engineering majors higher tuition and fees—in some cases as much as $2,000 more per year. It's what is known in academic circles as differential tuition, the practice of charging higher tuition and fees for various majors such as engineering, business and science, for example. (*Ed. Note: The schools that agreed to talk to us for this article impose differential tuition, but the upcharge is minimal in comparison to others*.) Granted, the cost to the university is higher for these programs as opposed to, say, the social sciences or English literature. But it begs—or screams—the question: What is wrong with this picture?

A telling omen of things to come? Comparisons to Rome's demise aside, the United Sates' future role in the global economy might be dictated by a seeming intent to opt out of its once-held role as the world's manufacturing leader.

At a time when NASA in particular, or the aerospace and defense industries in general, are desperate for an infusion of eager, well-educated young people to come and continue the work of their retiring predecessors, should we be making it even more difficult for them to do so?

Likewise, while a little appreciated but absolutely essential industry such as gear design and manufacture goes begging for new talent, we as a nation continue to ignore the fact that our children in grade and high schools across the nation are not receiving the encouragement needed from teachers to inspire them to want to design and build things for the common good.

So what happened?

Remember the social contract? "We're moving from a sense that higher education is supported as a public good to something that you're purchasing as a service or investment," says Robert Gustafson, associate dean of the college of engineering for undergraduate education and student services at Ohio State University. "And if you look at what's happened in higher education in the shift from state support to tuition-and-fees support, the clear message you read from that is that education is not as strongly considered as being in the public interest as it once was.

"As a society, it's a real issue. How do we get the message out to value science and technology careers? Because right now, the message is—don't go there."

Mark Kushner, dean, college of engineering at Iowa State University (which imposes a modest hike in engineering-related tuition and fees), is of the same mind.

"I think it's a philosophical issue at work, which is difficult to quantify," he says. "There was a time in public education that acknowledged that the public education of a single individual is in the public good. (Graduates) go out to be school teachers and engineers, doctors and lawyers, journalists and engineers. And they sort of make society work and that's a good thing."

According to Kushner, "what happened" was that "During the difficult budget times between the mid- to late-'90s, that philosophy changed to the public education of an individual is good because they go out and get a job and make a comfortable living. So if they're getting that direct benefit, they should pay for it."

Indeed, just about everyone interviewed for this article pointed to cuts—or a deceleration—in school funding for state universities. That in itself is a major indicator as to where our priorities as a country reside. After all, if there's a shortfall in a state's higher education funding, you can always raise tuition and fees. Or put another way, you can borrow from Peter to pay Paul. But who pays for the long-term consequences? We do. In some cases, says Kushner, it is a matter of world view and experience.

"The (members of) legislatures tend to be less from engineering and more from the service side (lawyers, entrepreneurs, etc.) of the economy," says Kushner. "So they may not have an appreciation for what it takes to develop the technologies that make their lives what they are."

Kushner adds, rhetorically, "It's terribly difficult for states to decide where you have to balance your budget every year. (If) you have overflowing prisons, you have absolutely no choice but to build more and bigger prisons. You have to get the money from someplace; well, you can always raise tuition at a state university. I'm not implying it's prisons versus universities, but universities are a little bit more unique in publicly funded projects."

Just to provide a snapshot of our country's priorities regarding education, consider this. According to Kushner's "back-of-the-envelope calculation," the yearly undergrad tuition for every engineering student in America totals roughly $2.5 billion; the war in Iraq is costing us $1 billion per day.

Colorado State University is another school with differential pricing for certain majors, including engineering. The upcharge is modest, about $200 over an entire year, but needed, according to Sandra Woods, dean of engineering.

"A lot of state universities have employed differential tuition for programs that are very expensive to deliver," she says. "Engineering is probably one of the highest-cost programs because of our faculty salaries, and also the large number of laboratories that we deliver for technical programs. So that is when we made the decision to implement differential tuition, just to reflect the cost of the program."

The good news, according to Woods, is that every dime is invested back into the engineering department to cover costs and, most importantly, to hire more teachers. Woods adds that the school was able to hire three more faculty members for the department—thus reducing student-per-instructor ratios—a distinct learning advantage. Woods also points out that professors of engineering command higher pay because it is a given that they can make much more money in the private sector.

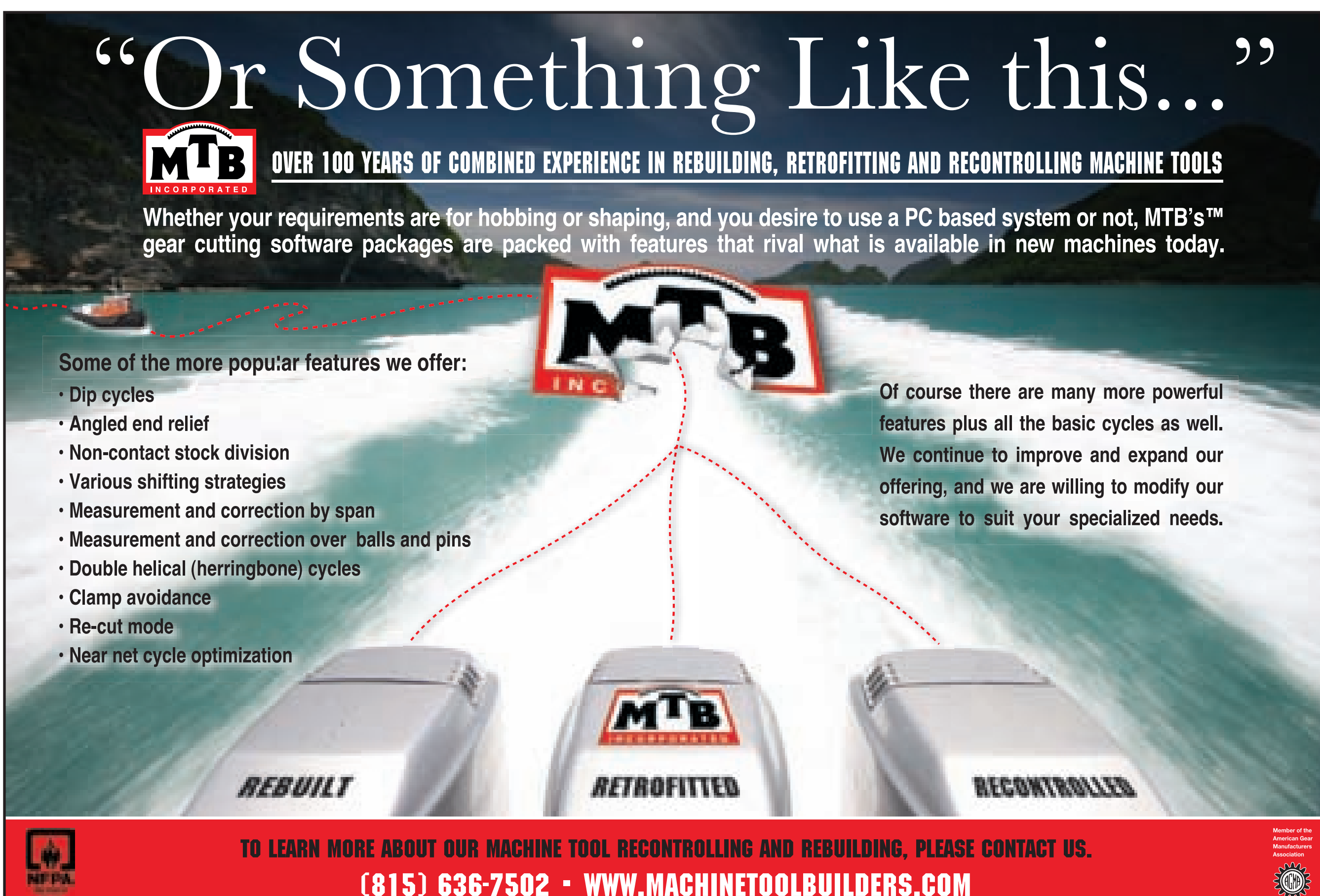
"It's the market," Woods says. "If you compare hiring a mechanical engineering faculty member and an English or social sciences professor, it may be a 50 percent increase in salary that you need to pay in order to recruit the best faculty."

At Pennsylvania State University (home of the Gear Research Institute) there is no differential tuition; merely computer and lab fees, according to Suren Rao, institute director. He also states that while there has not been a cut in financing in his state, "The rate of growth of state funding has declined." Some would interpret that as a cut, but it apparently is not affecting undergraduate enrollment.

In the final reckoning, differential tuition and the schools that impose it are not, ultimately, the real issue. That is simply nibbling around the edges. And it is not as if one can point a finger at any one sector of our society in identifying why there is a brain drain and a lack of will to regain our nation's manufacturing and technology preeminence.

But one place to start—as has been pointed out in this publication before—is the primary and secondary schools. It is while young people are of that certain age that a seed can be planted and nurtured in encouraging a career in engineering and the sciences. And, according to most of the people interviewed for

continued



this article, that nurturing and inspiration have, over the past 30 years, not been forthcoming. The end result is a general ignorance and lack of regard for the engineering profession itself.

Respect in short supply. "There is a respect issue," says Iowa State's Kushner. "In places like Korea, China, India, it is the parents' purpose or goal to work with their children to get them into the best science and engineering schools there are, because it is considered such a crowning achievement and is so beneficial to the country.

"Here, for whatever reason, academics and the technically educated are not held in particularly high esteem. If a person has a medical degree, you call that person doctor. If you have a Ph. D. in physics, you're called mister. In the scheme of things, maybe that's not so important, but it is delivering a message to young people that says, 'Where am I going to go to get respect?'

"I give a lot of rotary-type presentations, and when I ask the audience, 'Have any of you benefitted from the work of an engineer since you got up this morning?' Nobody raises their hand. We would all be standing naked in open fields if there were no engineers."

Angel Otero, chief of space operations at the NASA Glenn Research Center in Cleveland, has one son in engineering at Ohio State and another on the way. His concern is the absence of new blood that will be needed to help the agency in its goal to get back to the moon in the next 10 years.

"The aging (of existing engineers) is a big problem with NASA because a lot of our folks retire at a fairly young age. With civil service, they can retire at 55 and then go to work for a (NASA) contractor for a few years and retire again. And we are not getting the influx of young people to come in behind them to bring the energy, the new ideas, things like that."

Asked where he sees things 10 years out, "It could be an interesting situation," he cautions, with a discernible note of dread in his voice. "We will have less and less of a properly skilled workforce to handle getting ready to go back to the moon, for example. We need to be hiring right now."

Speaking of aerospace, there is another parallel to be drawn; this one dating back to 1957, when the Russians successfully launched Sputnik. That served as a national wake-up call to the nation that our space program was lagging and that we needed to get back to sharpened pencils and slide rules.

"We need to make this (lack of young engineers, etc.) our generation's Sputnik," says Niel Tebbano, vice president of operations for Project Lead the Way (PLTW), a highly regarded nationwide foundation that exists to enlist young people in tomorrow's engineering challenges. "We're shortsighted if we don't do something about it."

The good news is that some people are, mostly at the grassroots level. In Montana, Democratic Senator Max Baucus has proposed free—let me repeat that—free tuition for math and science majors as part of his Education Competitiveness Act initiative in the state. His intent is to better prepare students for college and to help the U.S. stay globally competitive with countries such as China and India. In return, engineering, math, and science and technology graduates agree to work or teach in a related field for four years.

And in addition to groups like PLTW and the Austin Polytechnical Academy in Chicago (a technical-based high school created in 2007 to help students across all demographics along the path of high technology and learning), there are numerous companies in the private sector with programs and competitions created to encourage participation in the technical sciences and engineering.

"It is my opinion that major corporations do provide support for undergraduates in a variety of ways," says the Gear Institute's Rao. "The co-op programs are the most valuable. Where they provide minimal support is in the graduate programs for research. In the past, faculty could cheaply hire foreign graduate students who went on to stay in the U.S.

"However, today many of these students from Korea, China and India go back home and the U.S. is not producing enough graduates with advanced degrees in engineering to keep us on the cutting edge. More support from corporations for research would help in recruiting native-born students into (graduate programs)."

But in the final analysis, the consensus seems to be that more—much more—is needed from lower education and government if we are to have any hope of creating a groundswell of interest in the hard sciences among young students. That's because right now, it's not happening. Consider Otero's telling example of a Dad's Day experience he once had.

"When my son was in grade school, I took a NASA spacesuit to his class so the kids could touch a real spacesuit and ask questions, and you could see the excitement on their faces. But then when he got to junior high, I never got a call. The teachers didn't have the same enthusiasm at that level."

"What your country can do for you"—if willing. As for governmental involvement, Tebbano at PLTW believes the states need to lead the charge, especially given the lack of direction from Washington.

"I think the states need to understand that there's a direct relationship between their ability to produce a qualified future workforce and their own goals related to economic development and so on. What they're really doing (by not adequately funding schools) is exacerbating an existing problem. There already is a dearth of young people entering these fields, as we know. They're really not doing anything to contribute to solving the overriding issue, which is where are we going to get the quality, quantity and diversity for our future engineering and technical workforce?"

Revisiting the earlier Sputnik reference, Tebbano reminds those of us old enough to remember—not a problem, unfortunately, in the gear industry—how the Sputnik success spurred government to action.

"When Sputnik went up, the federal government responded with the National Defense Loan System, and they waived certain percentages of student loans for young people entering science and education. That had a huge effect on choices that young people made at that time, and I can speak to that because I took advantage of that program as a college student. So do incentives have an impact on some of that?—absolutely.

"Using that logic, doing the opposite (reduced funding; differential tuition) would in turn have the opposite effect. If a young person considering college has a choice between paying $5,000 to go into liberal arts, and $7,500 to go into engineering or science, if they're on the fence you know which way they're going to lean."

Productivity on Demand

ADAPTATION KEY TO SUCCESS FOR GEAR SOFTWARE DEVELOPERS

Matthew Jaster, Associate Editor

In 1995, *Gear Technology* urged readers to jump aboard the computer manufacturing bandwagon. For most engineers, trading their pencils and paper for computer technology was a daunting task. Learning new software meant starting from square one with each upgrade. By the time employees were properly trained on software programs, they could be outdated or obsolete.

The concept of full automation was originally associated with the aerospace and automotive industries. Large manufacturers utilized CAD programs, CNC machines and PCs in their design departments. Companies like Boeing and Chrysler began creating models entirely on the CAD system. The computer had become an intricate part of the day-to-day routine for manufacturers.

RomaxNVH: Exaggerated 3D view of a normal mode shape in a transaxle gearbox.

The gear industry had no choice but to follow suit.

Today, the computer has its hard drive in everything from human resources to the shop floor in an effort to help gear manufacturers increase production and minimize costs. Companies like KISSsoft AG, Romax Technology and Dontyne Systems Limited began designing practical software applications with these goals in mind. Goals that have led developers to collaborate with research institutions, consultants and universities all in the name of technology.

Gear Software, Then and Now. According to Dr. Mike Fish of Dontyne Systems and retired Prof. R.G. Munro from the University of Huddersfield in England, the earliest benefit of gear software appeared before many current engineers were ever involved in gearing.

"This arose from the fact that gear geometry often involves calculating a small difference between two large numbers using 7-figure logarithm tables or involute trigonometry tables. It was a big step forward to be able to solve the transcendental involute function equation with just a simple iterative program," Fish says. "The development of computers in the 1970s and 1980s allowed designs to be calculated quicker than on a drawing board. Today, the calculation speed has significantly improved to the point that more comprehensive data analysis can be carried out beyond linear methods of design."

Fish and Munro add that gear metrology has made great use of software, in conjunction with the advent of precise instrumentation for large displacement angular and linear movements such as optical gratings and laser devices. Software has also improved presentation formats, simply by replacing "the old rolls of curly chart paper" with data that can be read and easily filed.

"The rapid expansion of computing power in the early 1990s made calculations practical in the sense that a comprehensive analysis of 'what if' possibilities could be played out in the model in a practical time scale," Fish says. "The interpretation of the result is equally as important as the quality of the analysis itself."

L. Kissling & Co. was an early developer of software for the calculation of machine components. In order to ensure the upkeep and maintenance of their software, KISSsoft AG was founded as an independent company in 1998.

"*KISSsoft* was first written on a Commodore PET in *BASIC*," says Dr. Stefan Beermann, vice president at KISSsoft. "It has come a long way since then. Fortunately, the programming needed is comparably simple. It's the technical background in machine design that's most important."

Beermann says the *KISSsoft* product is 25 years old and is constantly being modified and maintained. The company's strategy is simply evolution over revolution. They take pride in the fact they've managed a large project like *KISSsoft* over the years with maximum effort and reliability.

"Our software is under permanent maintenance. Once in awhile a complete module is rewritten because of the problem of maintaining very old code," Beermann says, "but we're hesitative with this because 25 years of debugging is part of the main capital our company has, so we're always looking for a method to keep the calculation code as is."

Romax Technology has been developing software tools and performing technical consulting to major automotive OEMs and suppliers for over 20 years. Their software package, *Romax Designer*, was first released in 1993 but can trace its philosophical roots back to an NIST funded research program from the late 1980s.

"Romax tends to perform one major release per year with additional interim updates," says Andy Poon, director of software & strategy at Romax. "We believe incremental updates enable our customers to have access to new features as soon as possible rather than waiting for major releases in order to benefit from improvements to the software."

The software development team at Romax has grown enormously over the last 10 years. Poon says the most significant changes have occurred with its internal development processes and the way the company interfaces with its customers.

"In the very beginning, we pored over classical reference papers, documents from the standards committees, published technical papers and other technical documentation to figure out the requirements and how to program the algorithms," Poon says. "Today, we work very closely with our customers on joint software developments and spend an increasing amount of time on research and development projects and university-based research programs."

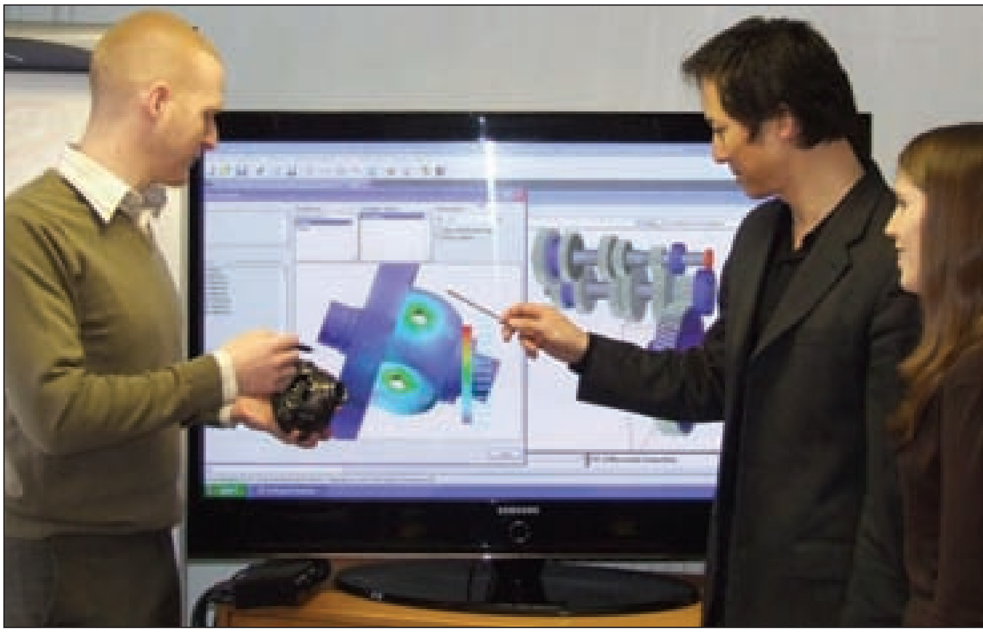
The Growing Needs of the Customer. The satisfaction of a job well done comes when the developers see customers solving real-world problems with their software. Poon says engineers at Romax believe the software is built with their customers in mind, noting its ready-to-use features and accessibility as being vital to the market.

"Though we had access to a number of similar software packages, we continued using Romax for its ease of use and accuracy," says Jon Adler, engineer at McLaren Performance Technologies, Inc. "Romax gives us a quick and complete way to evaluate the effects of design parameters on components within a gearbox. By evaluating loads in three dimensions, we can avoid over- or under-designing bearings and shafts."

Fish explains the relationship with Dontyne's customers is vital to enhance the "off-the-shelf" product. "In order to do justice to our products, we work closely with our customers to ensure that the software is properly implemented," Fish says. "This has a dual benefit in that it allows the customer to become quickly adept, and allows us to realize where future product development will be most effective."

With relatively few software developers in the gear industry, companies are forced to maintain a strong customer base globally. Dontyne, Romax and KISSsoft maintain offices around the world to keep up with customer demands. Although KISSsoft is headquartered in Switzerland, the company works regularly with clients such as Brad Foote Gear Works in Illinois. (Now Tower Tech Holdings)

"Our interaction with KISSsoft has been excellent," says Chuck Schultz, vice president of engineering at Brad Foote Gear. "My senior engineer went to a training session and came back fully confident he could use the software efficiently. When questions have come up



Collaborative gearbox development with Romax Software.

Software Bits 2008

Dontyne Systems Ltd.

Dontyne's *Gear Production Suite* is a package of software products to design, machine and inspect gears during production. Each module has been developed with three levels of operation including basic, standard and advanced. The functionality of one or more of the tools can be embedded in existing metrology or machine tool equipment to form part of an expert system. Highlights for the software package include *Gear Design Pro* to define gear pair geometry and the calculation of their rating according to ISO standards and *GATES* software, a program for the calculation of stress and transmission error conditions in loaded gear systems. Dontyne's website notes that a concept design tool for calculation speeds and torques in a planetary system is currently being developed. The company recently confirmed an exclusive deal with partner Gaudlitz GmbH in the plastics field until 2010 for the implementation of a module for the optimization of tool forms.

KISSsoft AG

There are currently two major new developments in the works at KISSsoft AG. The first is a completely new interface (GUI) for *KISSsoft*, replacing a 10-year-old concept. The second is that shaft calculation is now based on a Finite-element core, which gives the company the ability to model the loads applied to the shaft more realistically. Furthermore, the company has some new developments with *KISSsys*, a set of models for typi cal wind turbine gearbox kinematics and a gearbox model that implements a thermal analysis of the gearbox. *GEARCALC*, a software package for sizing and rating cylindrical gears includes three parts for engineers including *GEARCALC, RATE2001 and LUBE925*. The software package was recently launched in the United States.

Romax Technology

Romax is constantly working to increase both its breadth and depth of coverage for gear applications. Underlying many new developments are two key ideals the company is striving for: 1) Analysis should not be done using idealized (nominal) parameters. For simulation to accurately reflect the real world, it must embrace the fact that the real world is full of imperfections, and the variations of manufacturing, assembly alignment and loading should be considered in the design stage. 2) The different analysis disciplines should not be spread amongst different tools. You should not have to duplicate the building of analysis models to look at different phenomena such as fatigue life, modal response, transient dynamic events, efficiency, etc. The company provides an integrated approach to gearbox design, analysis and virtual testing with a software suite that includes *RomaxDesigner, RomaxDurability, Romax Dynamics, RomaxNVH* and special industry packages including *RomaxWind* and *RomaxBearing*.

concerning the software, our local rep has been able to get the answers within a day or so; as fast as you'd expect given the time difference. We went with KISSsoft originally on the recommendation of our own customers," Schultz says.

Beermann says language barriers pose additional challenges when servicing a global community.

"Our software is currently available in five languages," Beermann says. "We not only have to build a system and manage it, we have to know the technical terms of each language. We have an employee on staff strictly for language. It's not just translating, it's localization. It's separating each technical term. Challenging work, but well worth it to get the product out to our global customers."

Success in this industry, however, is achieved first and foremost in one's own backyard. Fish says companies need to improve their own products with good housekeeping and effective data transfer between departments.

"Many companies have developed in-house calculation procedures based around international standards suited to their own manufacturing capability," Fish says. "A major consequence of leaving this to a single person in the company is not often felt until that person leaves or retires."

Is There a Qualified Engineer in the House? Effective software tools begin and end with a well-trained engineer. Beermann is concerned the most educated crop of experts might be on their way out. The company is preparing itself for a significant decline in gear industry expertise. He believes the typical "hard-core" gear expert in the U.S. is close to retirement.

"Our tools do not replace an engineer," Beermann says. "You need someone in place with some know-how concerning the implemented methods." (*Editor's Note: Please see our feature on engineering tuition/education on page 25 for additional information.*)

Fish says there's currently a major move to re-train engineers in the industry. The focus will be to ensure that any knowledge contained in software can be formatted and applied by engineers.

"Much of what was learned by sound and analytical techniques in the 60s and 70s has had to be relearned by a generation of engineers implementing a software routine."

Dontyne has been working within the industry to help define the design procedures as well as the interpretation of equations and graphs in common standards. Along with a group that includes the ISO, BGA and AGMA, they're currently looking at the influence of these decision-applying standards. Romax has also been involved in these discussions.

With the limited pool of experts in the field, it's no surprise how important collaboration is to the success of software development in the gear industry.

Beermann believes most gear conferences often have the air of a family gathering.

"In my experience, most of the gear engineers are open to exchanging know-how and ideas with each other. The same is valid for companies providing gear design software."

While the designers tend to split between independent software vendors (ISVs) and academic institutes, Poon doesn't think there are any hard and fast rules to collaboration.

"Some ISVs are quite secretive. Likewise, the academic institutes or consortiums have to serve their members, although they do collaborate with industry partners. However, even the academics do specific work which is often subject to non-disclosure agreements."

Dontyne's *GATES* software was developed in collaboration with the Design Unit at Newcastle University, U.K.

"Our industrial development partners and beta testers test the analysis tools before they are released and form an important bridge from theoretical analysis to the engineering tool," Fish says. "The most rewarding part of the job is seeing the change in attitude when a company realizes the benefits of our products to their operation. We're delighted to promote our collaboration as much as possible. It publicizes the fact our capabilities extend beyond the software itself."

Moving Forward. For Dontyne, gear software development in the coming years rests on bridging the gaps that sometimes exist between design, machining and inspection.

"Digital data from CNC equipment can be directly accessed to enhance the models. Machines can have value-added by incorporating further analysis features. A machine tool with on-board correction capability can also have on-board modeling of the component. Advanced modeling ensures that variables in the production processes can be calculated and analyzed, and the corrective action defined before the machining process even begins.

KISSsoft AG has also been adapting to the learning curves associated with software development. "CAD programs change, hardware platforms change, operating systems change and gearing standards are under continuous modifications," Beermann says. "These are not problems, but a necessary part of our customer service to keep our software ahead of the industry. Due to the higher performance of computers, we can now implement features today that would have been out of discussion 20 years ago."

Emerging manufacturing markets in China, India and Korea appear to be the focus for many developers in the months ahead. Beermann is quick to note the market is large enough for all the software developers to coexist.

"There are only a few companies left that develop this technology. It's important to maintain connections and have good relationships with everyone involved."

At Romax, the future involves ensuring their software remains at the cutting edge. They're currently working on some long-term solutions to enable a larger proportion of mechanical engineers to have easy and affordable access to the tools they need.

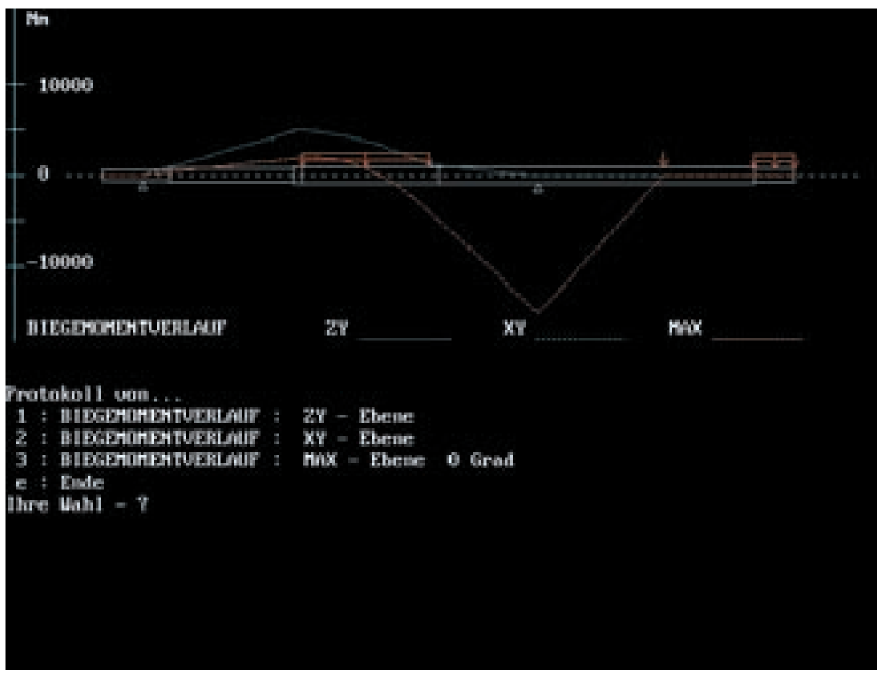
"When Romax first entered the market with *RomaxDesigner* in 1993, it was a unique approach. Our challenge is to continue to innovate whilst delivering solutions that give our customers the competitive edge," Poon says.

Still, the biggest concern for most developers comes back to the education and training needs of the engineers.

"A common trap in a great deal of engineering software use is thinking it will do the work of an engineer. With very few exceptions, software is still only a labor-saving device," Fish says.

The problem isn't just replacing the engineer; it's getting each one on the same page. Fish says there's still extreme confusion in comparative software programs. He states that even experienced gear engineers that make one different choice from a well-recognized standard can result in two completely different sets of analysis for the same set of problem parameters.

"It's essential to understand this concept to improve software training on a



KISSsoft shaft calculation 20 years ago.

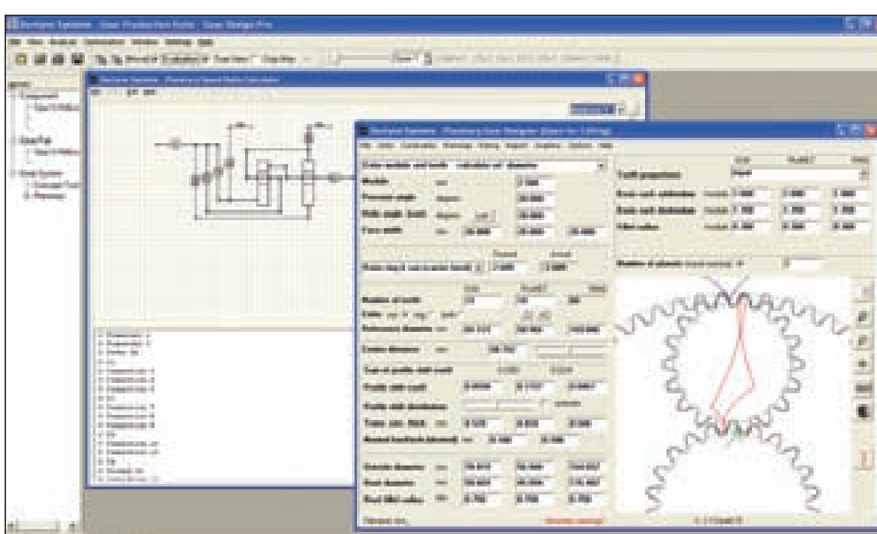


Image concept and planetary design tools by Dontyne Systems.

global scale, especially when entering new and emerging markets," says Fish.

As technology evolves, companies must be prepared to adapt to the ever-changing face of software development. While the programs of the early 1990s look prosaic by the standards in place today, one can only speculate what short-cuts and tricks will be available five years from now.

"The speed and scope of analysis and data in gear software is incredible," Fish says, "but it will only ever be as good as the engineer using it."

For more information:

Brad Foote Gear Works
1309 S. Cicero Ave.
Cicero, IL 60650
Phone: (708) 298-1100
Fax: (708) 652-4140
chucks@bradfoote.com
www.bradfoote.com

Dontyne Systems
1 Simonside
Prudhoe, Northumberland NE426JL
United Kingdom
Phone: +(44) 1661 833 828
Fax: +(44) 1661 523 614
mike@dontynesystems.co.uk
www.dontynesystems.com

KISSsoft AG
Uetzikon 4
CH-8634 Hombrechtikon
Switzerland
Phone: +(41) 55 254 20 50
Fax: +(41) 55 254 20 51
Stefan.beermann@KISSsoft
www.kisssoft.ch

McLaren Performance Technologies Inc.
26555 Evergreen Rd., Ste. 900
Southfield, MI 48076
Phone: (248) 358-6039
Jon.Adler@linamar.com
www.mclarenperformance.com

Romax Technology
Rutherford House
Nottingham Science and Technology Park
Nottingham, Nottinghamshire NG7 2PZ
United Kingdom
Phone: +(44) 115 951 8800
Fax: +(44) 115 951 8801
bekia@romaxtech.com
www.romaxtech.com

Space Station's Solar Power Compromised by Balky SARJ Unit

UNDUE VIBRATIONS, POWER SPIKES AND GRIT GIVE NASA PAUSE

Jack McGuinn, Senior Editor

Shuttle Discovery flight engineer Daniel Tani during a more than 6-hour spacewalk dedicated in part to an inspection of the International Space Station's malfunctioning Solar Alpha Rotary Joint (SARJ). Tani returned to the station with metal "shavings," samples of an as yet unknown origin gleaned from beneath the joint's multi-layered insulation covers.

Since last October, NASA engineers on the ground, as well as International Space Station (ISS) and shuttle Discovery astronauts, have been working to indentify and rectify an ongoing problem with one of the two solar array panels that power the station's energy system. The area of concern is one panel's two 10-foot-wide bull gears, fitted with two redundant-drive motors, or drivelock assemblies (DLA). Together, they comprise one SARJ—or solar alpha rotary joint—in the power truss that serves to direct the massive starboard solar panels towards the sun. Only one DLA is needed at any time to power the outboard

continued

bull gear. The other panel is problem-free. While at this writing a root cause remains elusive, the problem does not present any imminent danger to the crew or station, according to NASA. That's because despite what NASA SARJ recovery team leader Kevin Window characterizes as a "major anomaly," both solar panels can still be maneuvered, if necessary. In other words, the motor and/or bearings that power the panel in question have not seized and are therefore operational, if only manually for the time being.

After ruling out possible meteor damage, the investigation continues both in space and on the ground at the Johnson Space Center (JSC) facility in Houston. ISS station commander Peggy Whitson and flight engineer Daniel Tani, along with lead and fellow spacewalkers Scott Parazynski and Doug Wheelock, respectively, have led the space-borne investigation while Window coordinates earthbound testing.

Bad vibes a concern. Flight control first became aware there was a problem when ground personnel noticed unwarranted vibration during the affected panel's rotation. A surge in the panel's power mechanism was also detected. As a result, space walks and other activities planned for the ISS and Discovery crews were put on hold until it was determined that the problem was not life-threatening. Subsequent inspections by crew members ultimately revealed the presence of an unidentified grit—believed to be metal shavings—in the 12-set, trundle bearings gear race. Using swatches of tape, samples were collected from the surface by astronaut Tani. According to NASA, the debris could be causing the power surge which, if not controlled, can cause major damage to the SARJ unit. The samples were delivered to JSC courtesy of the Discovery STS-120 crew upon their recent return to Earth on November 7.

Window is faced with the challenge of trial-and-error simulation and other life-testing protocols. He has generated what is known as a fault tree that hopefully serves to facilitate fault and potential root cause analysis.

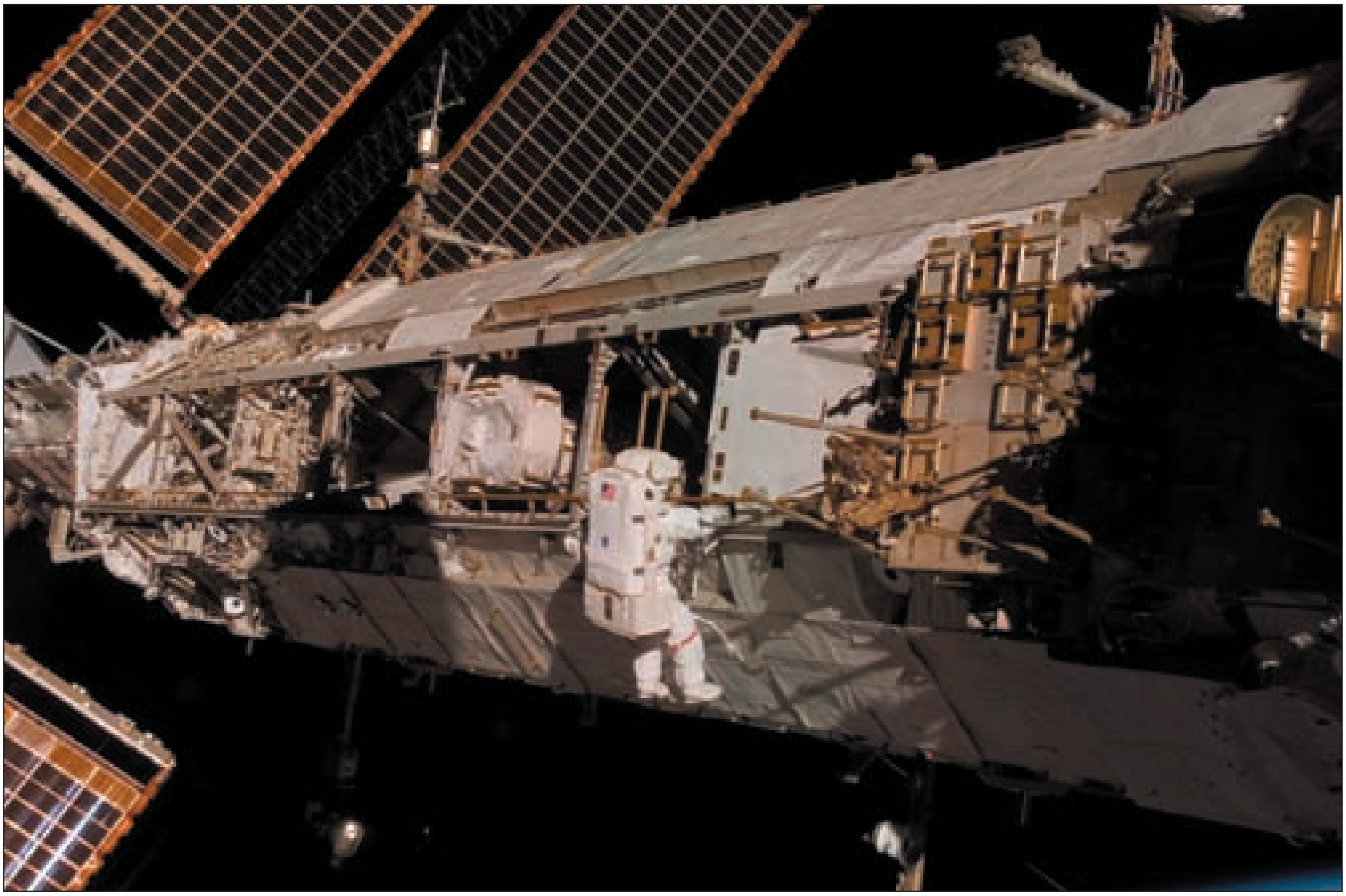
"We have a couple things to work with," Window says. "We have the structural test article, which is the SARJ with its internal bearings and DLA mounted to it that was utilized for the life testing of the (component). So we have that on the ground, and the other thing we have is the contractors (Lockheed-Boeing) that designed and developed the SARJ." Window adds that Lockheed is spearheading the development of a test rig that will best simulate on the ground what might be going on with the unit up at the space station. Window says it is a "wild guess at this point" as to why exactly the problem exists.

To date, however, Window says the inspected samples collected in space show debris on the nitrided layer of the race ring, which is where the flaking is occurring, perhaps ruling out bearing and or gear degradation. He cautions, though, that seized bearings have not been ruled out.

As for the gears, "From inspections we have been able to perform at this point, we have not seen any gear teeth degradation or that anything looking like the drivelock assembly bull gear is causing any type of damage to the gear teeth or the raceway."

Knowing what they don't know. As for what has in fact been observed, Window explains.

"We have seen in our analysis of the witness coupon



Expedition 16 shuttle Discovery commander Peggy A. Whitson (above) was also involved in the spacewalk investigation of the ISS solar array's (top and next page) faulty SARJ unit. She is seen here examining the damaged starboard component.

(sample) that there is what we call a subsurface networking and cracking in that nitride layer. And what that means is that when you go through the heat treating process of the nitride, what it appears at least in the coupons we have analyzed is that there are areas in that layer in which the molecules didn't completely bond, and that's causing the networking and cracking." Given that, the recovery team is at this point looking closely at whether the heat treat process was compromised in some way.

Additionally, Window says, "The other thing we believe we're seeing is a potential castoring effect of the bearings in which the bearings are sliding and then trying to guide back (into position). And that of course increases surface friction; that could be a contributor as well."

Also under consideration as potential root causes are lubrication and defective steel. And although there is no "wet" lubrication on the bearings surface in question—the roller bearings are gold plated—engineers and tribologists at the NASA Glenn Research Center in Cleveland have been brought into the mix to determine whether the addition of a wet lube might enhance performance, even for just the short term.

As for the steel, Window says, "For now, it is believed that all of the race rings developed went through the exact same process. When you have the port side (where the second SARJ and panel are located) that hasn't seen this problem, it makes one wonder."

For now, testing continues as Window and his team attempt to configure testing and analysis in as close an apples-to-apples scenario as possible in order to isolate a root cause and corrective action.

Stay tuned.

Methodology for Translating Single-Tooth Bending Fatigue Data to be Comparable to Running Gear Data

D.R. McPherson and S.B. Rao

Introduction

The gear industry is under continual pressure to increase power density and reliability of geared transmissions while at the same time reducing costs. To meet these demands, new materials and manufacturing processes are being evaluated on a continuing basis. The first step in this evaluation is to conduct screening tests to compare the performance of gears fabricated using the new materials and processes with that of gears manufactured using the incumbent materials and processes. However, once promising new materials and processes have been identified, the issue rapidly becomes one of developing accurate design data to permit effective utilization of these new materials and processes. The accepted practice in the gear industry is that accurate design data be derived from running gear tests.

Running gears can fail via a number of modes, many of which are shown generically in Figure 1. Screening tests are conducted in a manner that allows evaluation of performance relative to one of these modes while avoiding damage via the others. A common approach is to evaluate bending strength using the single-tooth bending fatigue test (STF) and surface durability using the rolling/sliding contact fatigue test (RCF). From the bending strength point of view, this ensures that tests intended to evaluate bending strength will not have to be terminated due to surface durability (pitting, wear or scoring) failures. The design of running gear test specimens to evaluate bending strength (or surface durability) requires a good initial estimate of bending strength so that the specimens can be designed to fail by the target failure mode at about the desired life without

__D.R. McPherson__ is a research engineer at the Drivetrain Technology Center/Applied Research Laboratory, at The Pennsylvania State University. His areas of research include: performance characterization of material systems for gearing; contact fatigue; surface topography characterization; bending fatigue; scuffing; and gear manufacturing. McPherson holds a B.S. in naval architecture and marine engineering from the Webb Institute, and an M.S. in mechanical engineering, from Vanderbilt University.

__Suren B. Rao__ is a senior scientist at the Applied Research Laboratory of The Pennsylvania State University, and managing director of the Gear Research Institute. He holds a Ph.D. (University of Wisconsin-Madison), a M.Eng. (McMaster University-Canada) and a B.Eng. (Bangalore University-India), all in mechanical engineering. He has over 38 years of experience in manufacturing research in academia, industry and government, of which about 25 years have been focused on mechanical power transmission components and systems. He has authored many papers in refereed journals, conference proceedings and several book chapters. He also holds several patents in the field of gear manufacturing and is a member of the ASME, the AGMA and NAMRI/SME.

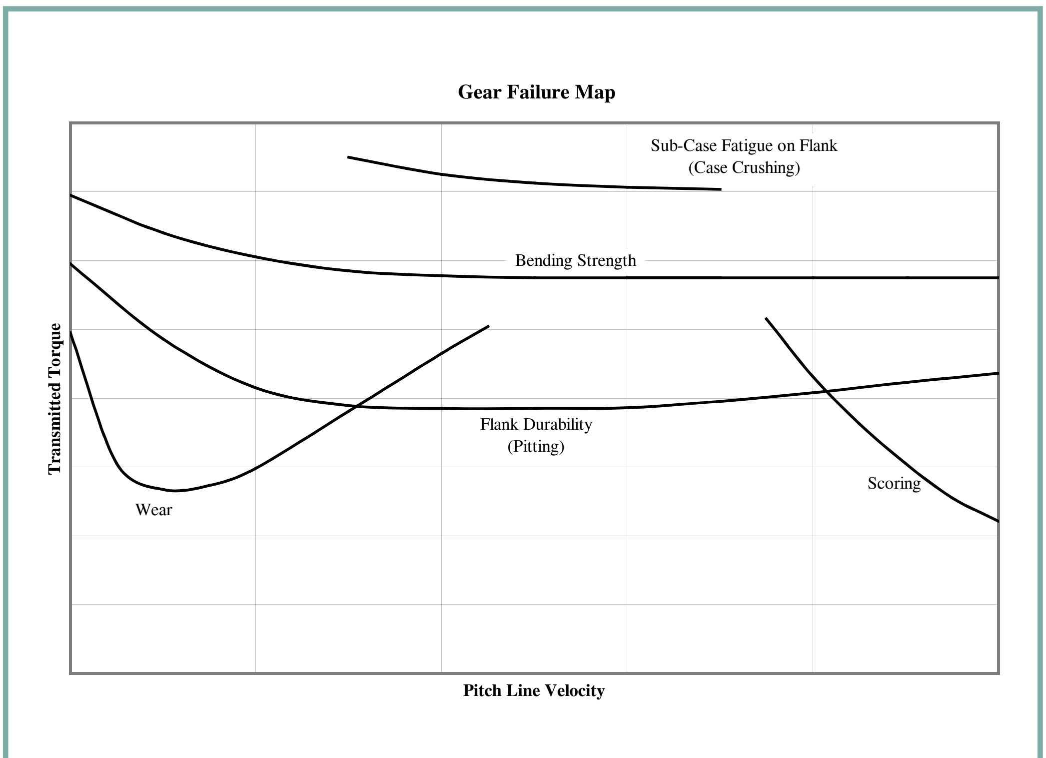


Figure 1—Gear failure map.

undue risk of failure by unwanted modes. This paper describes a method developed by the Gear Research Institute to extrapolate running gear bending strength data from STF results that is invaluable in comparing bending performance of different materials and processes. This methodology has also proven useful in the design of running gear bending strength test specimens. It is strongly recommended that data extrapolated by this or other similar methodologies should not be used as the basis for the design of gears for applications.

STF Tests Compared to Running Gears

In the STF test, a specimen gear is held in position and one tooth at a time is tested by applying a cyclic load normal to a fixed point on the flank. Care is taken in the selection of the load point and in the design of the loading appliance to ensure that the surface of the test tooth is not locally overloaded at the point of contact. Thus, the influences of all the surface durability aspects of testing running gears are eliminated, and tests may be continued as long as needed to achieve failure via bending. The cyclic load is varied between the selected maximum and some fixed percentage of that maximum (10% or 5%, depending on the compliance of the system) to maintain preload on the system. This limits the distance the hydraulic loading cylinder must travel and permits testing at comparatively high frequencies. A typical STF fixture, with specimen installed, is shown in Figure 2.

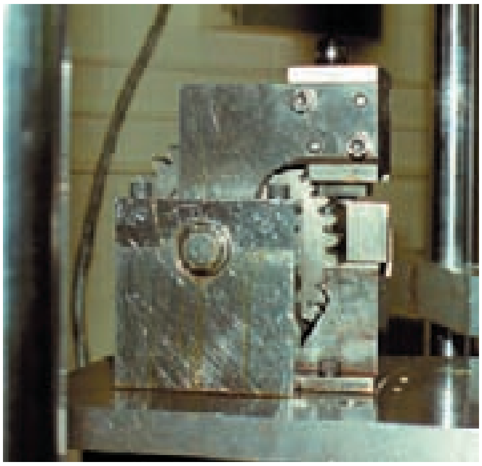


Figure 2—Gear Research Institute's standard STF specimen mounted in fixture.

These test conditions yield three categories of correlation issues that must be taken into account when translating STF test results to running gear test results. The easiest to explain is the correlation issue of the stress range experienced by running gear teeth compared to STF test teeth. As noted above, in the STF test, stress varies from 10% (or 5%) of the maximum up to the maximum. In running gears, the load is completely released as the tooth passes out of mesh. In some cases, depending on geometry and operating speed, the critical area in the root fillet is subjected to a small amount of compression as the next tooth is loaded. Thus stress varies from zero (or a small negative percentage of the maximum) to the maximum.

The next correlation issue is due to the fact that the teeth that will break on running gears represent only the weakest part of the statistical population defined by the teeth tested in the STF test. In the STF test, several (normally at least four, often eight, sometimes up to sixteen) teeth are tested on each gear, one at a time, and each represents a separate data point. In running gears all of the teeth are tested on each gear in the same time, and the failure of the first tooth represents failure of the gear. Much of this paper is dedicated to presenting a method to account for this difference.

The last correlation issue, and the most difficult to explain and quantify, is related to confidence. It is not economically feasible to conduct enough screening tests with new materials and manufacturing processes to be able to draw many statistically reliable inferences from the result. The object of most single-tooth fatigue testing is to determine the mean load resulting in failure at the run-out number of cycles. To make the best possible estimate of this load with a reasonable number of tests, recent test programs have been conducted using a two-load approach. Enough tests are conducted in an up-and-down sequence to find two loads that result in a non-zero and non-unity failure rate. In other words, at each load some of the tested teeth break by the run-out limit while others do not. Further tests are conducted until six have been completed at each of these loads. Ideally, one of the failure rates at these two loads will be above 0.5 and the other below 0.5. Analysis of the result will allow the load resulting in 50% failure at the run-out limit to be determined with some statistical reliability. The range within which the load to result in 50% failures could vary is illustrated by the confidence bands in Figures 5 and 9. In the test programs these figures are drawn from, tests were conducted following an up-and-down sequence and less than six tests

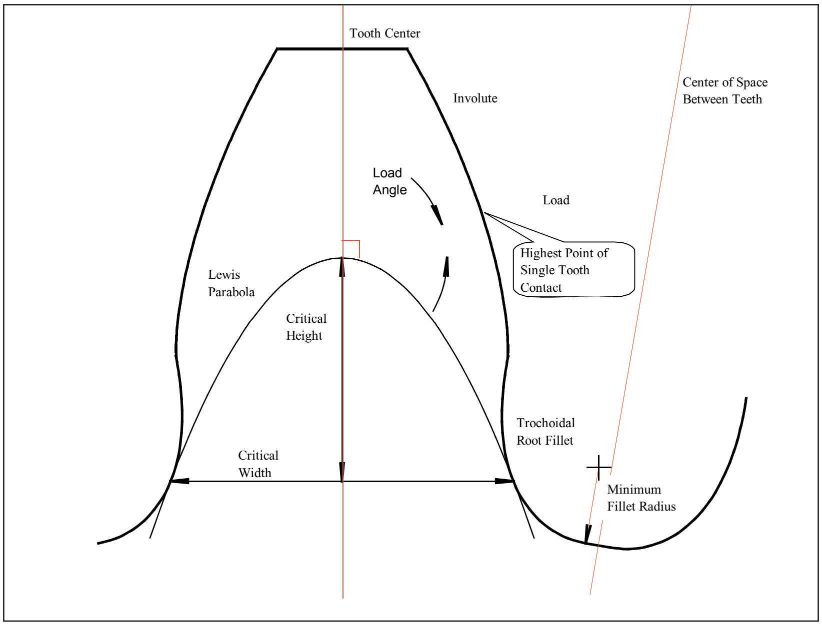


Figure 3—Layout of spur gear showing loading and Lewis parabola.

$$\text{Bending Stress} = \frac{\text{Load} \times \text{Cos}\left(\text{Load Angle}\right)}{\text{Face Width}}\left[\frac{6 \times h}{s^2} - \frac{\text{Tan}\left(\text{Load Angle}\right)}{s}\right]K_f$$

Equation 1.

$$K_f = \text{Stress Concentration Factor} = H + \left(\frac{s}{r}\right)^L \left(\frac{s}{h}\right)^M$$

Equation 2.

were conducted at most of the loads. If there were two loads at which six tests had been conducted, the range would correspond to an approximate 90% confidence interval.

What the result of this limited number of tests does not provide with any reliability is the standard deviation of the load that results in failure at the run-out limit. Methods are presented in the literature to determine standard deviation (Refs. 1, 2); however, to accomplish this with statistical reliability requires that two, four or more times as many tests be conducted. A good knowledge of the standard deviation is needed to adjust loads to account for different failure rates, such as are encountered with running gears where all of the teeth are subjected to loading compared to one tooth at a time with single-tooth fatigue testing. The empirical method presented here avoids this need for an accurate value of standard deviation.

Conversion of Load to Stress

Before comparisons can be made between tests with specimens having differing geometries—such as those used in STF and running gear tests—the applied test loads must be converted to bending stress. Figure 3 shows a spur gear tooth with a point load applied at the highest point of single-tooth contact. This point of loading corresponds to the highest bending stress when there is effective load sharing between gear teeth. Specimen gears used in rig tests should have effective load sharing, so this is the appropriate point of loading for determining bending stress for running gear specimens used in rig tests. For gears tested in single-tooth bending fatigue, the actual point of loading established by the test fixture is used in calculating bending stresses.

The Lewis parabola is drawn from the point the load line intersects the center of the gear tooth and is tangent to the root fillet. The methods used to lay out this parabola vary, depending on how the root form is generated, and the full particulars are lengthy and presented in detail elsewhere (Ref. 3). The critical height and width are determined from the Lewis parabola as shown in Figure 3. The angle between the load line and a normal to the tooth center is termed the load angle (it differs from the pressure angle at the point of loading because of the thickness of the tooth). The bending stress is thus:

See pg 44 for Equation 1

Where
s = Critical Width from Lewis Parabola
h = Critical Height from Lewis Parabola

See pg 44 for Equation 2

r = Minimum Fillet Radius
H = 0.331–0.436 x (Nominal Pressure Angle –Radians)
L = 0.334–0.492 x (Nominal Pressure Angle –Radians)
M = 0.261–0.545 x (Nominal Pressure Angle –Radians)

This equation for bending stress can be derived from first principles or from AGMA standards (Refs. 3, 4) by taking the forms of relevant formulas pertinent to spur gears and setting all design factors at unity. A similar formula could be developed for helical gears.

For a given gear design and loading condition, such as the Gear Research Institute's Standard STF specimen, loaded in its standard fixture, Equation 1 can be simplified to the following form:

$$\text{Bending Stress} = \text{Load x Stress Factor} \tag{2}$$

Where the stress factor comprises all the items on the right side of Equation 1 except load. For the Gear Research Institute's standard STF test, this becomes:

Face Width	1.000 inch
Load Angle	24.8 degree
h	0.286 inch
s	0.335 inch
K_f	1.53
Stress Factor	19.3 psi bending stress per pound load

Correlation Issues

The three correlation issues discussed earlier are now treated in detail.

Correction for Allowable Stress Range. In the Gear Research Institute's standard STF test, the load is varied from 10% to 100% of the maximum load. These R = 0.1 stresses are converted to the required R ratio stresses via ASR diagrams. The ASR diagrams are constructed to be representative of brittle materials following the method described in Reference 5. The pertinent equations are as follows:

See pg 45 for Equation 3

See pg 45 for Equation 4

(Ultimate stress is taken as the bending stress corresponding to the linear deviation point load from the fast bend single overload test.)

$$Y = \frac{1 - \dfrac{\sigma_M}{\sigma_U}}{1 + \dfrac{\sigma_M}{\sigma_U}} \tag{5}$$

$$\sigma_R = \text{Fully Reversed Stress} = \frac{\sigma_A}{Y} \tag{6}$$

These equations can be algebraically manipulated to yield an expression for any desired R ratio stress. Strain gage calibration with the Gear Research Institute's standard running gear bending test specimens show that the stress varies from negative 20% to positive 100% when the gears are tested at standard operating speed (Ref. 6). Thus, R loading for the examples shown here is equal to negative 0.2. By way of example, an allowable

$$\sigma_A = \text{Alternating Stress} = \frac{\text{Maximum Stress} - \text{Minimum Stress}}{2}$$

Equation 3.

$$\sigma_M = \text{Alternating Stress} = \frac{\text{Maximum Stress} - \text{Minimum Stress}}{2}$$

$$\sigma_U = \text{Ultimate Stress}$$

Equation 4.

stress range diagram constructed in the manner described above is shown for the running gear G50 stress for the first example set of data.

Statistical Analysis—Accounting for Differing Populations. The statistical step from failures of individual teeth to failures of gears is made using a probability diagram comparing maximum applied test load (abscissa) and failure rate in terms of the variate of a probability distribution (ordinate). The exact nature of the scales to be used on this diagram is not intuitively obvious. It is customary to make the scale on the life axis of stress-life diagrams logarithmic; however, the scale on the stress axis may be either linear or logarithmic. Based on this precedent, the scale on the load axis for the diagram to be constructed here could be either linear or logarithmic. The Weibull distribution is frequently used to characterize fatigue; however, it is customary to use the normal probability distribution to analyze failure rates at the fatigue endurance limit. Thus, there are four reasonable sets of scales that could be used on this diagram.

ANSI/AGMA 2001-C95 gives a table of reliability factors to relate allowable stress and various failure rates (Ref. 4). The values in this table represent experience with gears and show the magnitude of difference in applied stress to result in progressively lower failure rates. The reliability factor appears in the denominator of the rating equations. Thus the reciprocal of the table values is proportional to the stress difference associated with the difference in failure rate. These are failure rates for gears each having a definite number of teeth. The STF results are for tests of individual teeth. To be helpful in determining the scales to be used on the diagrams constructed here, this information must be converted to failures rates of teeth. The Gear Research Institute's standard specimen that has been used in running gear tests has 18 teeth, making this a convenient number of teeth-per-gear for the calculations presented here. In this case, one failure in two gears tested corresponds to one failure in 36 teeth tested

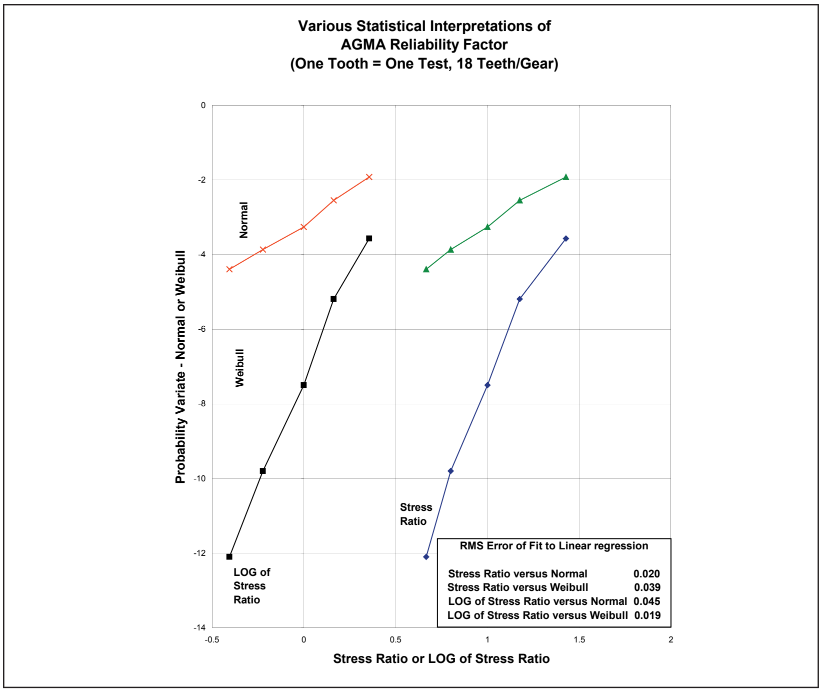


Figure 4—Various statistical interpretations of AGMA reliability factor (1 tooth = 1 test, 18 teeth/gear).

and the corresponding normal probability variate is negative 1.916 (contrasted to considering failures of gears where one failure in two gears tested results in a failure rate of 0.5 and a normal probability variate of 0.000).

Figure 4 shows the reciprocals of the values from Table 11 in Reference 4 scaled four ways. The abscissa shows these reciprocals plotted on a linear scale and a logarithmic scale. The ordinate shows the failure rate values divided by 18 to represent failures of individual teeth, plotted against normal probability variate and Weibull probability variate. Normal probability variate is x from Equation 7, and is shown as "x" in normal probability tables.

$$\text{Failure Rate} = \frac{1}{\sqrt{2\pi}} \int_{-\infty}^{x} e^{-\frac{1}{2}t^2}\, dt \qquad (7)$$

Weibull probability variate is given by Equation 8.

$$w = Ln\left(Ln\left(\frac{1}{1 - \text{Failure Rate}}\right)\right) \qquad (8)$$

The relationships in Figure 4 that are closest to linear are the ones with the logarithmic stress scale versus Weibull probability and the linear stress scale versus normal probability. Both have roughly equally good fit to a linear regression. Since normal probability has been used frequently to characterize failure rates at the fatigue endurance limit, the charts presented here were based on normal probability. The best fit with reliability factors and normal probability was with the linear stress (or load, since it is proportional to stress) scale; thus, the diagrams presented here use these scales.

Maximum test load is used in constructing the diagrams presented here. This is an artifact of the manner in which the method was developed; load could just as easily be converted to stress before the statistical analysis as after (as is done here). Test results are sorted by load and the failure rate is determined at each load. These results are plotted in terms of normal probability variate value corresponding to the failure rate at each load tested. These points are plotted as hollow diamonds in the sample diagrams presented later in this paper. The normal probability variate corresponding to 100% failures is positive infinity, that corresponding to 0% failures is negative infinity. In order to keep the scale of the diagram reasonable, these values are plotted as positive three and negative three, respectively, and not used in fitting a line to the data representing a mixture of failures and no failures. Because of this, it is necessary to have data from at least two loads that resulted in a mixture of failures and no failures.

The step from failures of individual teeth to failures of gears is based on fitting a line through these data points. When a line is fit to data using a least squares fitting technique, it can be shown that one point on the line will always be the average point of the data. The average point of the data is noted mean failure point on the charts presented here, and is plotted as a solid diamond. The abscissa for this point is defined as the average maximum load in tests at loads that resulted in a mixture of failures and no-failures. The ordinate is defined as the NPV value corresponding to the average failure rate in these tests. The tests are summed individually to account for different numbers of tests at each load. A line is fit through the mean failure point and is extrapolated down to normal probability variate equal to negative 1.916, which corresponds to one failure in 36 teeth or one failure in two 18-tooth gears to find the 50% failure load corresponding to running gears. The exact slope of this line cannot reasonably be determined with the limited data typically available; the method used to fit it is discussed in the following section.

Statistical Analysis—Accounting for Confidence. In most instances, three to six teeth were tested at each load. Thus the confidence in the ordinate values is limited. An approximate confidence interval is constructed based on the limiting normal probability variate values if one additional test were to be conducted at each load. The upper bound represents what the failure rate would have been if the additional test failed, the lower bound that if it did not fail. When all teeth tested at a given load fail, the corresponding normal probability variate value would be positive infinity. In most cases, no more than six tests are conducted at loads that result in 100% failure. Thus, if one more tooth were tested that did not break by the designated run-out limit, the failure rate would be six of seven and the corresponding limiting (minimum) normal probability variate value would be on the order of positive one. As noted previously, infinite values of normal probability variate are not useful for fitting lines to the data; however, the limiting values at the bottom of the range (or top of the range if several teeth are tested at a common load and all do not fail) are useful, as described below.

For each set of STF test results, two statistical diagrams are constructed. The first (labeled Step One) is focused on the load range used in STF tests. A line is fit by eye judgment through the mean failure point with a slope that seems to fit the data, and the 50% failure load is determined from this line. Standard deviation of the mean test load for the STF condition (sigma) is the reciprocal of the slope of this line. Given the size of the confidence intervals at each point (see Figures 5 and 9), it is clear that standard deviation cannot be estimated within a factor of two with any statistical reliability, given the number of tests conducted. Rather than attempt to extract standard deviation from too little data, it is assumed that sigma (for the STF test condition) is a fixed percentage of the 50% failure load. Based on examination of as many STF data sets as possible, with tests conducted over a span in excess of 10 years, this value for sigma is taken as 10% of the 50% failure load.

A second statistical diagram (labeled Step Two) is then constructed for each set of STF test results. A wider range of maximum loads is included in this diagram, and it is used to find the 1% failure and/or minus three-sigma load. A line is drawn through the mean failure point at the slope determined in Step One. This line is labeled Mean Fit—Load versus Failure Rate. A second line is drawn parallel to the first located to encompass all (or most) of the confidence intervals for each data point. (In some cases, the confidence intervals diverge further below the mean fit line than they do above. In these cases, the second line is drawn as far above the first as one would have to be drawn below it to encompass the confidence intervals.) This second line is labeled Conservative Fit—Load versus Failure Rate. The idea behind using two lines is to attempt to untangle scatter inherent in fatigue test results from change in failure rate with changing load, and to ultimately make a consistently conservative estimate of 1% failure and/or minus three-sigma bending strength.

The 50% failure load for running gears is selected from the Mean Fit—Load versus Failure Rate line. As noted previously, with the Gear Research Institute's eighteen-tooth specimen, 50% failure corresponds to one failure in 36 teeth tested, and the NPV is negative 1.916. Thus, the 50% failure load for running gears is the point on the Mean Fit—Load versus Failure Rate line at NPV equal to negative 1.916. The 10% failure load for running gears is selected from the Conservative Fit–Load versus Failure Rate line. With the 18-tooth specimen, 10% failure corresponds to one failure in 180 teeth tested, and the NPV is negative 2.54. Thus, the 10% failure load for running gears is the point on the Conservative Fit—Load versus Failure Rate line at NPV equal to negative 2.54.

Many industries consider the design condition to be 1% failure. With 18-tooth specimens, this is one failure in 1,800 teeth tested, and the corresponding NPV value is negative 3.26. The load corresponding to 1% failure is found by drawing a line through the loads selected for 10% and 50% failure with running gears and picking off the value at x equal negative 3.26. The aerospace industry considers the design condition to be minus three-sigma (i.e., one failure in 740 odd parts tested). With 18-tooth gears this is one failure in 13,333 teeth tested, and the corresponding NPV value is negative 3.79. The load corresponding to minus three sigma is found by drawing a line through the loads selected for 10% and 50% failure, with PC gears and picking off the value at x equal to negative 3.79.

Analysis Method—Step by Step

The first task in the analysis is to sort the data by load and find the failure rate at each

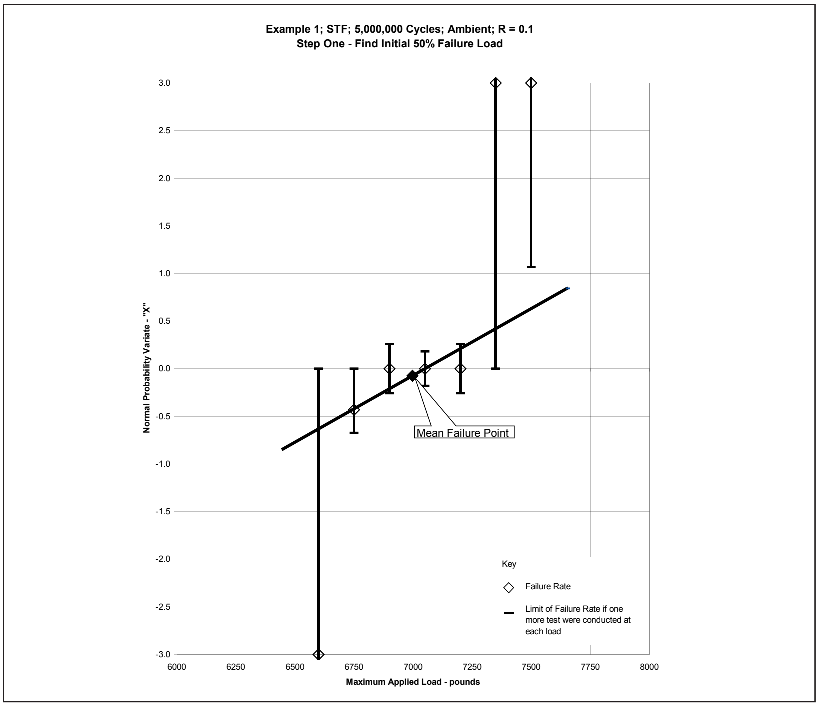


Figure 5—Example 1: STF; 5,000,000 cycles; ambient; R = 0.1. Step 1—Find initial 50% failure

Table 1 — Example 1 (Carburized) Data

Maximum Load (Pounds)	Number of Tests	Number of Failures	Failure Rate
7,500	6	6	1.000
7,350	1	1	1.000
7,200	4	2	0.500
7,050	6	3	0.500
6,900	4	2	0.500
6,750	3	1	0.333
6,600	2	0	0.000

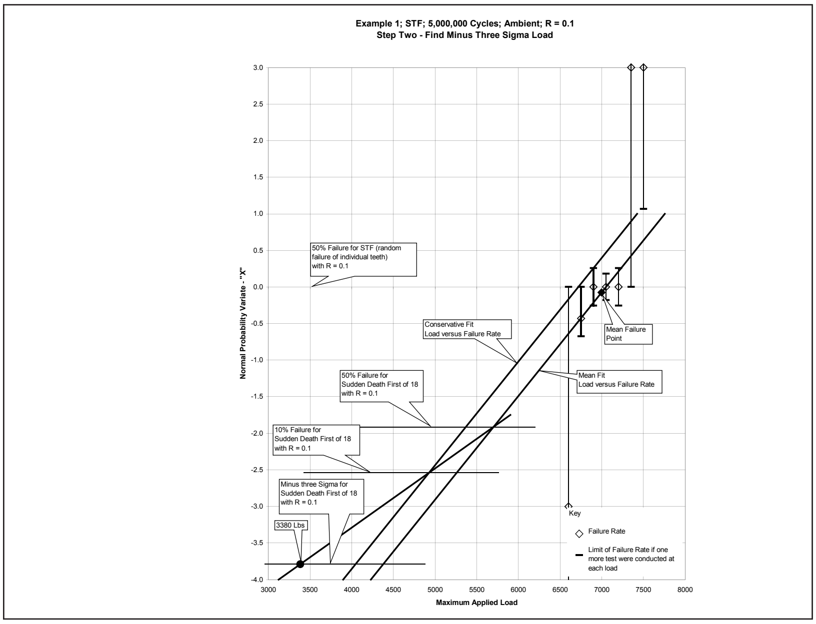


Figure 6—Example 1: STF; 5,000,000 cycles; ambient; R = 0.1. Step 2—Find minus 3 sigma load.

load. These failure rates are plotted in terms of normal probability variate versus load as hollow diamonds on the chart previously described. The mean failure point (average of all the tests at loads that resulted in a mixture of failures and no failures) is plotted as a solid diamond. The confidence ranges are calculated for each load and plotted as lines for each load. A line is fit through the mean failure point to suit the data and stay within the limits for each load. This line is used to determine the load corresponding to 50% failures. The slope of the lines on the second probability diagram is the reciprocal of 10% of this load.

A second probability diagram is drawn with the data points along with the mean failure point, and confidence ranges for each data point. The Mean Fit—Load versus Failure Rate is drawn through the mean Failure Point at the slope determined above. The Conservative Fit—Load versus Failure Rate line is drawn at the same slope to encompass all/most of the confidence ranges as previously described. The load corresponding to 50% failures is picked from the mean fit line, the load corresponding to 10% failures is picked from the conservative fit line, and the load corresponding to the desired design condition is selected by fitting a line through these two points and extrapolating to the required normal probability variate, all as described previously.

The loads are converted to stresses. The R equal 0.1 "running gear" stresses corresponding to 50% failures, 10% failures and design condition are adjusted to account for the stress range anticipated with running gears using allowable stress range diagrams as previously described. A stress-cycles diagram is constructed. A STF 50% failures (G50) curve is fit through the stress corresponding to 50% failure at the run-out limit determined in the statistical analysis and the rest of the data points using the best method available. For the limited data in the following examples, this is by eye. If more data were available, Weibull analyses could be conducted at several loads and used to define the finite life portion of the curve more precisely. This curve is moved linearly downward to the adjusted stresses at the run-out limit to represent 50% failures, 10% failures, and design condition for running gears. The full procedure is illustrated with the following two examples.

Example 1—Case Carburized Gears. The STF specimen gears used in the program the first set of sample data was taken from are Gear Research's standard pattern; hence the factor for converting load to stress is 19.3 psi bending stress per pound of load. The test results are summarized in Table 1.

Figure 5 shows the first statistical diagram constructed from this data. This diagram is used to select the load that corresponds to 50% failures; in this case the value is 7,050 pounds, which appears intuitively obvious from an examination of Table 1. The reciprocal of the slope of the fit lines shown in Figure 6 is taken as 10% of this value. Figure 6 shows the statistical step from STF to running gear data. In this case, the selected slope of the fit lines appears to fit the data very well. The translated R = 0.1 running gear G50 maximum load is 5,700 pounds, which corresponds to 110 ksi maximum bending stress. The allowable stress range diagram shown in Figure 7 shows the adjustment of this stress to R = negative 0.2 stress, which is 94.9 ksi.

Figure 8 shows a stress cycles diagram with STF and running gear data. An approximate STF G50 curve was fit by eye judgment to the data, starting with 136 ksi (corresponding to 7,050 pounds load) at 5 million cycles. The other curves were located by calculating the stress at five million cycles, as described above, and moving the entire curve linearly down to that point. The exact shape of the translated running gear curves below five million cycles cannot be accurately determined with the limited data available, so this method was adopted as the simplest expedient. Bending results from running gear tests are shown as hollow squares. These tests were conducted at extremely high overload to ensure that bending failures occurred rather than surface durability failures. STF tests were not conducted at high enough loads to directly compare with these results.

Given the paucity of data, it appears at first blush that the translated stresses are too low. In a later test program, three surface durability tests were conducted with the same grade of carburized steel gears at a load corresponding to the translated running gear G50 shown in Figure 8; one of these tests resulted in a bending failure and is plotted in Figure 8. Also, the result of one of the original running gear bending tests was an unexplained low-side-outlier; this result appears to lie in the region that would be extrapolated from the translated bending strength curves. This additional data tends to confirm the large step predicted between STF results and running gear result. Figure 8 also shows curves adapted from ANSI/AGMA 2001-C95 allowing for 10% and 50% maximum failures rates (KR = 0.85 and 0.70 respectively; all other rating factors set to unity), which fit the running gear data and the translated curves reasonably well.

Example 2 — Induction Hardened Gears. The single-tooth fatigue specimen gears used in the program for the second set of sample data was taken from the same general design as those used to develop the first set. The hob

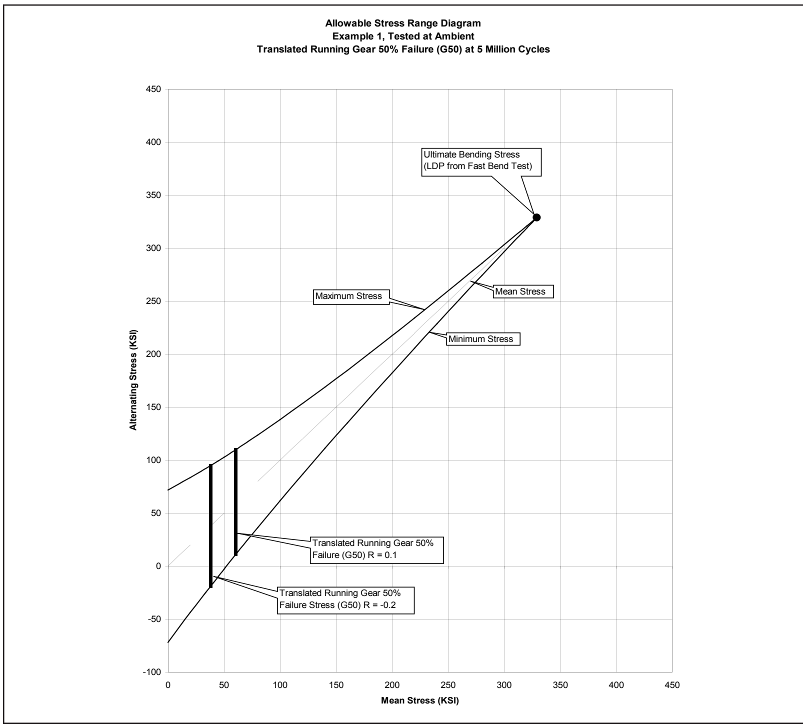


Figure 7—Allowable stress range diagram; Example 1: Tested at ambient; translated running gear 50% failure (G50) at 5 million cycles.

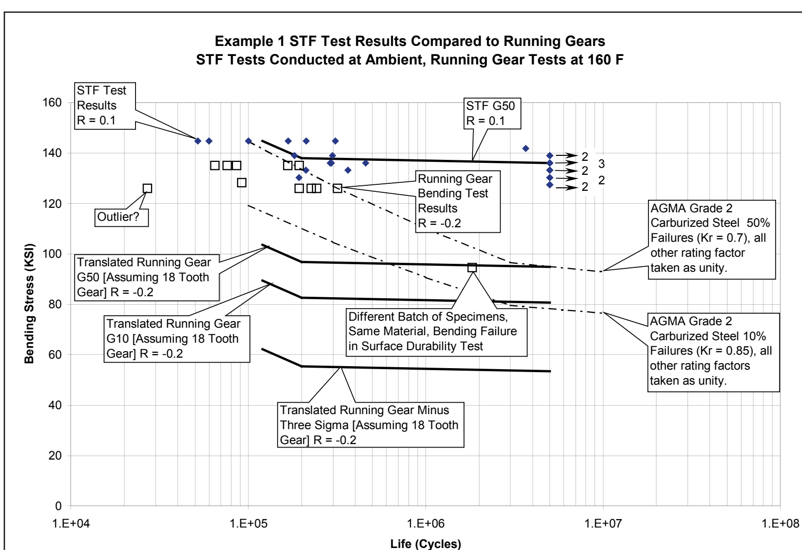


Figure 8—Example 1 STF test results compared to running gears; STF tests conducted at ambient, running gear tests at 160° F.

Table 2 — Example 2 (Induction Hardened) Data			
Maximum Load (Pounds)	Number of Tests	Number of Failures	Failure Rate
9,500	6	6	1.000
9,000	6	4	0.667
8,500	6	3	0.500
8,000	3	1	0.333
7,500	1	0	0.000

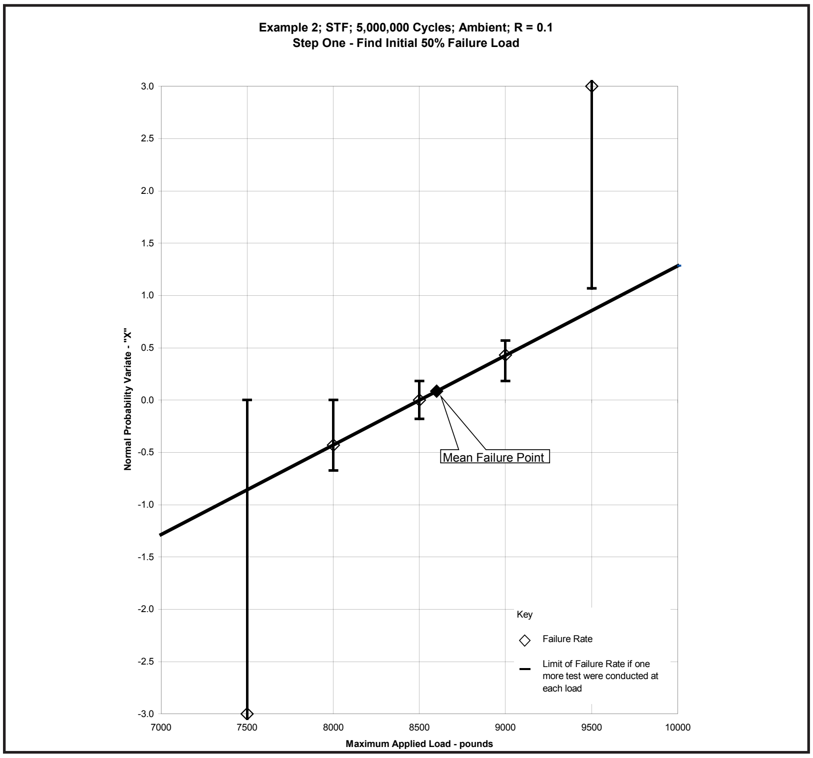


Figure 9—Example 2: STF; 5,000,000 cycles; R = 0.1; Step 1—Find initial 50% failure load.

used to cut these specimens was a short-lead hob that had been sharpened too many times, resulting in a different form in the root fillet. The bending stress was calculated in the manner previously described, with the result that bending stress was 20.3 psi per-pound-load (contrasted to 19.3 psi per-pound- load for the standard root fillet). The running gears used in this program again had 18 teeth. STF test data is summarized in Table 2.

Figures 9 and 10 show the application of Steps One and Two of the analysis method to this data. The slope of the fit lines in Figure 10 appears to be non-conservative (predicting high values of translated running gear bending strength) when compared to the data. Figure 11 is a stress-cycles diagram showing STF results, running gear test results and curves for STF G50, running gear G50, running gear G10 and running gear minus three-sigma. As was the case with the first example set of data, it would have been desirable to conduct more tests and better define the stress-cycles relationship. The STF G50 line is laid in by eye and is a compromise between the four failures below 200,000 cycles and two run-outs at the second highest load, and the six failures below 160,000 cycles at the highest load. The running gear bending results fall very close to the translated running gear's G50 curve. (This particular data set was selected because it comprises the longest cycle running gear bending failure data obtained with the Gear Research Institute's standard specimen gears, giving a better comparison to the portion of the stress-cycles relationship best defined by the STF test.)

All of the running gear bending data points fall above the translated running gear G10 curve, except one outlier run at a lower load in what was intended as a surface durability test. The specimen gears used in these tests were induction hardened. The origin of this outlying failure was at a large inclusion at the case core juncture some 0.050 inches below the root surface, further down the root fillet than the point maximum stress was expected. The material was commercial quality (air melt) cleanliness; however, this inclusion was larger than to be expected in commercial quality material. Thus, this outlying point represents an extreme condition, and it still falls above the translated running gear minus three-sigma curve.

Conclusion and Discussion

The method presented here, while being empirical, makes a reasonable approximation of running gear bending strength based on limited STF bending results. Prior work done in this area by the Gear Research Institute was based on running gear data obtained at very high overloads (as in Example 1), and predicted a smaller difference between STF and running gears. Results such as the bending failure in a surface durability test shown in Figure 8 were considered to be unexplained, low-side-outliers. Using the method presented here, this result fits the predicted trend.

Factors such as residual stress and dynamic loading have not been directly considered here. The STF and running gear specimens used to obtain the data shown in Example 1 were processed in the same manner, which should result in very similar residual stresses. The same was the case for the specimens used in Example 2. Running gear tests were conducted with low-mass gears at low speed in a machine with long shafts (providing torsional springiness, see Figure 12) to recirculate the applied load, resulting in low dynamic stresses. Strain gage measurements with this set-up at standard speed and double standard speed confirm that dynamic loading was minimal.

This area has been examined in the past by other investigators. Seabrook and Dudley (Ref. 7) found that the results of STF tests predicted 30% more strength than was the case with running gear tests using the same materials. This was attributed to a dynamic effect, even though the running gear tests were conducted on a rig designed to minimize dynamic loading. It is interesting to note that this is almost exactly the same difference found here (with gears reflecting four-decades advances in materials and manufacturing processes) that seems to be related to the statistical differences between STF and running gears.

References

1. Dixon, W.D., and A. M. Mood. "A Method for Obtaining and Analyzing Sensitivity Data," *Journal American Statistical Association*, Vol. 43, 1948, pp 109–126.
2. Little, R.E. *Manual on Statistical Planning and Analysis for Fatigue Experiments*, STP 588, ASTM, Philadelphia, 1975.
3. "Information Sheet: Geometry Factors for Bending Strength of Spur, Helical, and Herringbone Gear Teeth," AGMA 908-B89 (R1995), American Gear Manufacturers Association.
4. "American National Standard: Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth," ANSI/AGMA 2001-C95, American Gear Manufacturers Association.
5. Dolan, T.J. "Stress Range," in ASME Handbook, *Metals Engineering—Design*, O.J. Horger, Editor, McGraw-Hill, 1953, page 87.
6. Schwanger, V.C. "Experimental Measurement of Dynamic Bending Stresses at the Root Fillet of Spur Gear Teeth;" A Thesis in Industrial Engineering, The Pennsylvania State University, The Graduate School, Industrial and Manufacturing Department, State College, PA, August, 2003.
7. Seabrook, J.B., and D. W. Dudley. "Results of a Fifteen-Year Program of Flexural Fatigue Testing of Gear Teeth," *Transactions of the American Society of Mechanical Engineers, Series B, Journal of Engineering for Industry*, Volume 86, American Society of Mechanical Engineers, New York, NY, August, 1964, pp. 221–239.

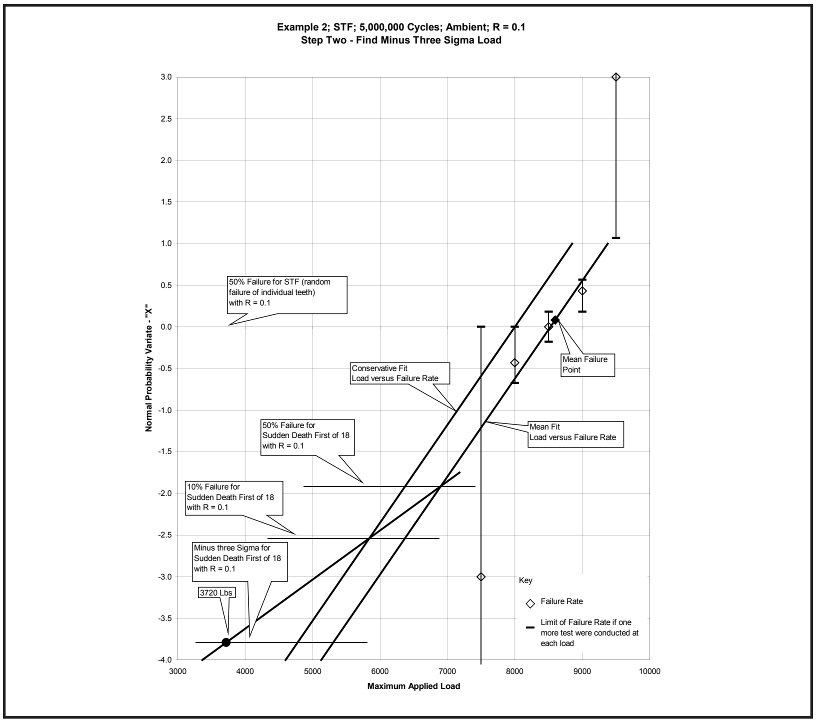


Figure 10—Example 2: STF; 500,000 cycles; ambient; R = 0.1; Step 2—Find minus 3 sigma load.

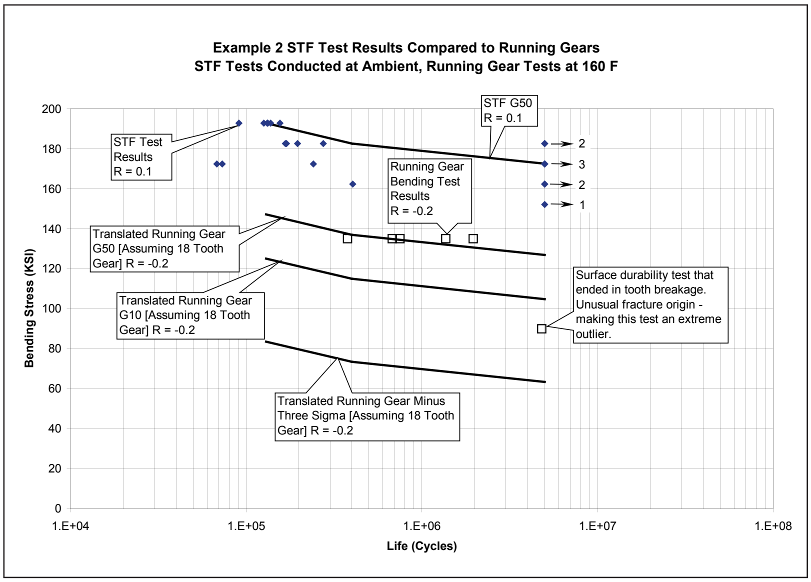


Figure 11—Example 2 STF test results compared to running gears; STF tests conducted at ambient, running gear tests at 160 °F.

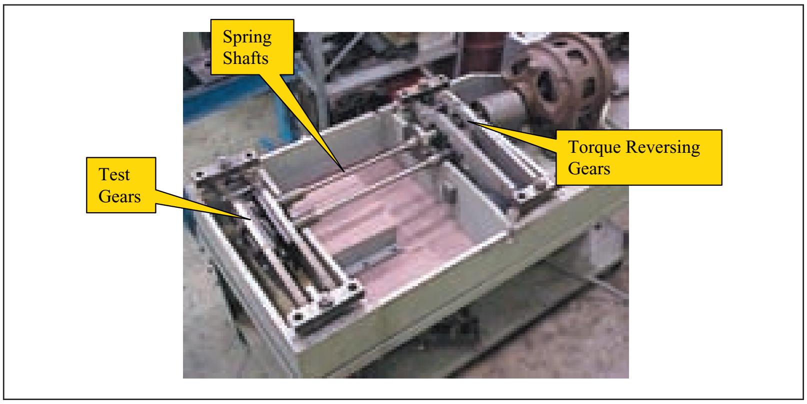


Figure 12—Power re-circulating gear test rig.

Pitting and Bending Fatigue Evaluations of a New Case-Carburized Gear Steel

Timothy Krantz and Brian Tufts

Management Summary

The power density of a gearbox is an important consideration for many applications and is especially important for gearboxes used on aircraft. One approach to improving power density of gearing is to improve the steel properties by design of the alloy. The alloy tested in this work was designed to be case-carburized with surface hardness of Rockwell C66 after hardening. Test gear performance was evaluated using surface fatigue tests and single-tooth bending fatigue tests. The performance of gears made from the new alloy was compared to the performance of gears made from two alloys currently used for aviation gearing. The new alloy exhibited significantly better performance in surface fatigue testing, demonstrating the value of the improved properties in the case layer. However, the alloy exhibited lesser performance in single-tooth bending fatigue testing. The fracture toughness of the tested gears was insufficient for use in aircraft applications as judged by the behavior exhibited during the single-tooth bending tests. This study quantified the performance of the new alloy and has provided guidance for the design and development of next-generation gear steels.

Figure 1—Gears for surface fatigue testing on the NASA Glenn Research Center gear fatigue test apparatus.

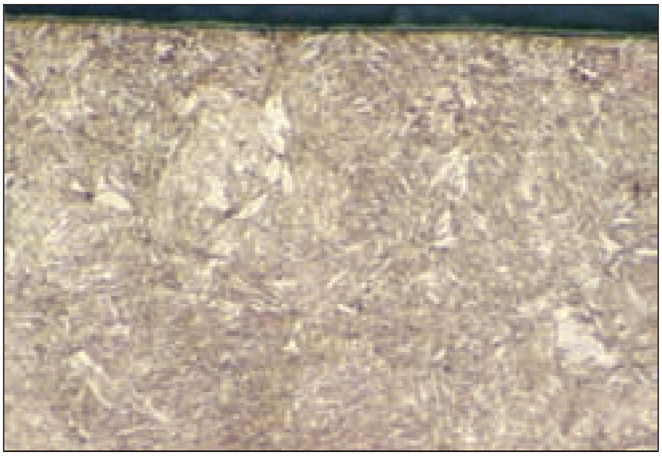


Figure 2—Microstructure of a Ferrium C69 test gear, microphotograph prepared with 2% nital solution.

Introduction

Gearbox power density is limited by ability of the gear teeth to transmit power for the required number of cycles without pitting, spalling or fatigue in the root-fillet region (bending fatigue). Methods for improving fatigue life capabilities of gears are highly desirable. Significant research has been conducted to develop new alloys and new steel processing techniques to improve the performance of mechanical components (Refs. 1, 2).

The subject of this article is the fatigue performance of gears made from a relatively new steel alloy, Ferrium C69. Helicopters make extensive use of two gear steels, AISI 9310 (AMS 6265) and Pyrowear 53 (AMS 6308B). Fatigue test data for these alloys were available and selected as baselines for purposes of comparing the performance of the gears of the present work. The alloy was selected and processed to achieve a higher surface hardness compared to production gears made from the just mentioned alloys currently in use for helicopters. For through-hardened steels, it has been demonstrated experimentally that a higher surface hardness provides for longer surface fatigue lives (Ref. 3). One might anticipate a similar benefit for case-carburized surfaces. Rakhit (Ref. 4) discusses gear steels, definitions of effective case depth and general trends of bending fatigue strength as a function of core hardness. As pointed out by Rakhit (Ref. 4), the relationship of the case and core properties of gear teeth to the fatigue strength performance is an ongoing subject of study and understanding. For any given gear application using case-carburized alloys, the heat treatment processing can be used to balance the fatigue resistance of the carburized surface with needed toughness of the subsurface core. Fatigue testing is needed to quantify performance. This article documents gear testing done to evaluate the fatigue properties of gears made from a new steel alloy having a high surface hardness.

Test Specimens, Apparatus and Procedure

Test Specimens. The dimensions of the gears are given in Table I. The gears (Fig. 1) are 3.175 mm module (8 diametral pitch) and have a standard 20-degree involute pressure angle with tip relief of 0.013 mm (0.0005 in.), starting at the highest point of single-tooth contact. The nominal face width is 6.35 mm (0.250 in.), and the gears have a nominal 0.13 mm (0.005 in.) radius edge break to avoid edge loading.

The gears of the present study were made from Ferrium C69 alloy. After completion of all heat treatment and grinding operations, a gear was cut and a tooth was removed for microstructural characterization. The microstructure of the near-surface material is provided in Figure 2. The surface was etched in 2% nital. The gear tooth has a uniform lath, martensitic structure. Micro-hardness was measured in both the root and flank locations. Vickers hardness measurements were made, and the results were converted to Rockwell C scale. Figure 3 provides the measured hardness as a function of depth from the surface for the flank location.

Gear Test Apparatus for Surface Fatigue. The gear surface fatigue tests were performed in the NASA Glenn Research Center's gear test apparatus. The test rig is shown in Figure 4a and is described in Reference 5. The rig uses the four-square principle of applying test loads so that the input drive only needs to overcome the frictional losses in the system. The test rig is belt-driven and the variable-speed motor was operated at a fixed speed for the subject testing.

A schematic of the testing apparatus is shown in Figure 4b. Oil pressure and leakage replacement flow are supplied to the load vanes through a shaft seal. As the oil pressure is increased on the load vanes, located inside one of the slave gears, torque is applied to its shaft. This torque is transmitted through the test gears and back to the slave gears. In this way, power is re-circulated, and the desired load and corresponding stress level on the test gear teeth may be obtained by adjusting the hydraulic pressure. The two identical test gears may be started under no load, and the load can then be applied gradually. This arrangement has the feature that changes in load do not affect the width or position of the running track on the gear teeth. The gears are tested with the faces offset as shown in Figure 4. By making use of the offset arrangement, the desired contact stress can be achieved within the torque capacity of the testing machine. Because of the offset testing arrangement, four tests can be completed for each pair of gears.

Separate lubrication systems are provided for the test and slave gears. The two lubrication systems are separated at the gearbox shafts by lip seals. The two lubrication systems use the same type of oil. The test gear lubricant is filtered through a 5-µm (200-µin.) nominal fiberglass filter. A vibration transducer mounted on the gearbox is used to automatically stop the test rig when gear surface fatigue damage occurs. The gearbox is also automatically stopped if there is a loss of lubricant flow to either the slave gearbox or the test gears, or if the lubricant overheats.

TABLE I—Spur Gear Data [Gear Tolerance per AGMA 2000-A88 Class 12]	
Number of teeth	28
Module, mm	3.175
Diametral pitch	8
Circular pitch, mm (in.)	9.975 (0.3927)
Whole depth, mm (in.)	7.62 (0.300)
Addendum, mm (in.)	3.18 (.125)
Chordal tooth thickness reference, mm (in.)	4.85 (0.191)
Tooth width, mm (in.)	6.35 (0.25)
Pressure angle, deg.	20
Pitch diameter, mm (in.)	88.90 (3.500)
Outside diameter, mm (in.)	95.25 (3.750)
Root fillet, mm (in.)	1.02 to 1.52 (0.04 to 0.06)
Measurement over pins, mm (in.)	96.03 to 96.30 (3.7807 to 3.7915)
Pin diameter, mm (in.)	5.49 (0.216)
Backlash reference, mm (in.)	0.254 (0.010)
Tip relief, mm (in.)	0.010 to 0.015 (0.0004 to 0.0006)

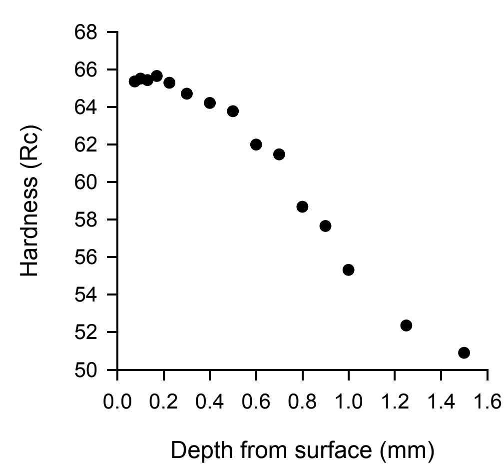


Figure 3—Measured hardness of a Ferrium C69 test gear tooth. The data are an average value of three measurements made at each reported depth.

Gear Testing Apparatus for Bending Fatigue. The gear bending fatigue tests were performed in the NASA Glenn Research Center Fatigue Lab using a commercially available fatigue test machine with a special fixture specifically designed to load the test gear tooth at the highest point of single-tooth contact. The fatigue test machine used for this investigation was a servo-hydraulic test system with 20,000 lb. load capacity. With the exception of the gear test assembly, the test system was in standard configuration for fatigue testing smooth- shank, cylindrical specimens. The load mechanism features an alignment fixture which can be used for closely controlled positional and angular adjustments. A load cell was used for control purposes and to monitor the test loads. The gear test assembly is mounted in the load frame using a support arm, which is attached to the two-post load frame. The bottom grip is attached to the actuator, which serves to power the system using 3,000 psi oil supplied by a central hydraulic system. An AC-type linear variable differential transformer (LVDT) is attached to the base of the load frame to measure actuator displacement or stroke. The stroke was monitored throughout the test for quality control purposes.

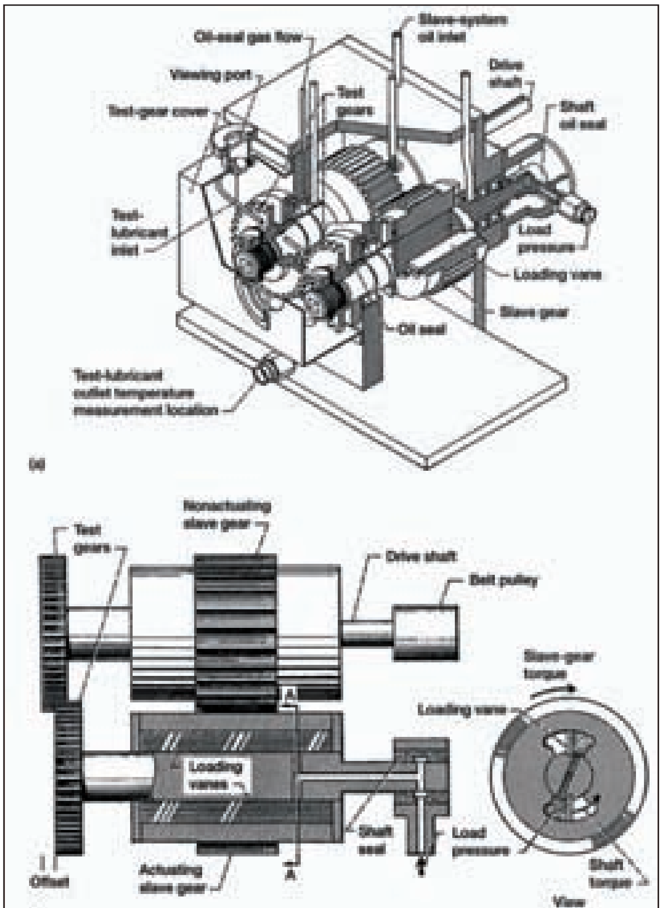


Figure 4—NASA Glenn Research Center gear fatigue test apparatus. (a) Cutaway view. (b) Schematic view.

TABLE II—LUBRICANT PROPERTIES	
Additive	[a] Lubrizol 5002
Kinematic viscosity, cSt	
311 K (100 °F)	31.6
372 K (210 °F)	5.7
Specific gravity	0.83
Flash point, K (°F)	544 (520)
Pour point, K (°F)	211 (–80)
[a]Partial content of additive: phosphorus, 0.6 wt%; sulfur,18.5 wt%	

Test Procedure for Surface Fatigue Testing. The gears were lubricated with a single batch of synthetic paraffinic oil. Physical properties of this lubricant are summarized in Table II. Five percent of extreme pressure additive, with partial contents including phosphorus and sulfur, were added to the lubricant. This lubricant has been used extensively for gear fatigue testing in the NASA Glenn spur gear fatigue rigs. For example, Krantz (Ref. 6) reported results of 146 tests using this same oil (termed "NASA standard" in the referenced article) to evaluate the surface fatigue lives of AISI 9310 steel gears. The oil and additive mixture used in this work is similar to 5-centistoke oils used for helicopter main-rotor gearboxes.

The test gears were run with the tooth faces offset by a nominal 3.3 mm (0.130 in.) to give a surface load width on the gear face of 3.0 mm (0.118 in). The actual tooth face offset for each test is based on the measured face width of the test specimen, and the offset is verified upon installation using a depth gage. The nominal 0.13 mm (0.005 in.) radius edge break is allowed for to calculate load intensity. All tests were run-in at a load (normal to the pitch circle) per unit width of 123 N/mm (700 lb./in.) for 1 hour. The load was then increased to 587 N/mm (3,350 lb./in.), which resulted in a 1.72-GPa (250-ksi) pitch-line maximum Hertz stress. At the pitch-line load, the tooth bending stress was 0.21 GPa (30 ksi) if plain bending was assumed. However, because there was an offset load, there was an additional stress imposed on the tooth bending stress. The combined effects of the bending and torsional moments yield a maximum stress of 0.26 GPa (37 ksi). The effects of tip relief and dynamic load were not considered for the calculation of stresses. The dynamic tooth forces on the gear teeth for test gears made to geometry specifications consistent with the present work have been previously reported (Ref. 6).

The gears were tested at 10,000 rpm, which gave a pitch-line velocity of 46.5 m/s (9,154 ft./min.). Inlet and outlet oil temperatures were continuously monitored. Lubricant was cooled and supplied to the inlet of the gear mesh. The lubricant outlet temperature was recorded and observed to have been maintained at 349±5 K (169±7 °F) once the rig was running and the temperatures were stabilized. The tests ran continuously (24 hr/day) until a vibration detection transducer automatically stopped the rig. The transducer is located on the gearbox adjacent to the test gears. Also, the surfaces of the teeth on the driven gear were inspected at regular intervals (normally not exceeding 50 million cycles between inspections) to verify the absence of surface fatigue. The lubricant was circulated through a 5 μm (200 μin.) nominal fiberglass filter to remove wear particles. For each test, 3.8 liters (1 gal) of lubricant were used. Six identical test rigs were used in this work. The pairing of gear samples and the run order was done randomly to minimize rig-to-rig and temporal differences.

The film thickness at the pitch point for the operating conditions of the surface fatigue testing was calculated using the computer program *Extern*. The computing tool is an implementation of the methods of References 7 and 8. For the purposes of the calculation, the gear surface temperature was assumed to be equal to the average of the oil inlet and outlet temperatures. Using the stated assumptions, the calculated pitch-line film thickness is 0.54 μm (21 μin.)

Test Procedure for Single-Tooth Bending Fatigue Testing. The gear test assembly is depicted in Figure 5. The gear test specimen is press fit on a shaft in the fixture's casing. The test assembly was designed to conduct tests on gear teeth in sets of three. This approach was adopted in part to provide the necessary clearance for the two load rods. To permit access to the gear tooth to be tested, several teeth nearby needed to be removed. Teeth were removed using the Electrode Discharge Machining (EDM) process. The upper load rod contacts the reaction gear tooth near the root of the tooth. In contrast, the lower load rod contacts the test gear tooth at the highest point of single-tooth contact. Adopting this approach, highest bending stresses are introduced into the test gear tooth and the location of fatigue failure is predetermined with a high degree of confidence. The rotational orientation of the test gear is precisely established using setup tooling. The rod that loads the test tooth at the highest point of single-tooth contact is representative of a rack gear (flat profile or infinite radius of curvature) contacting the test tooth.

A check of the fixture alignment was made using machinists dye (bluing) applied to the gear tooth profile prior to testing. The dye removal created by the load rod contact areas gave a clear indication that uniform load distributions on both gear teeth had been achieved. The contact pattern procedure was followed in all tests with excellent results.

The single-tooth bending tests of this study were conducted using unidirectional loading. Testing was done in load control. The gear was positioned to provide load on the test tooth at the theoretical highest point of single-tooth contact for the case of the test gear mating with an identical gear at the standard center distance (radius of 44.30 mm, or 1.774 in.). The load is cycled from a small, minimum load to the maximum load desired for the given fatigue test. The load range was maintained at a constant value throughout the test. Loading was cycled at 0.5 Hz using a sinusoidal waveform. Crack initiation was defined to occur when the loading rod stroke increased approximately 2% (~ 0.0002 in. change) relative to the stroke for the gear tooth at test initiation.

Results and Discussion

Surface Fatigue Test Results. Surface fatigue testing was completed on a set of gears manufactured from Ferrium C69 steel. The set of gears was case-carburized and ground. The test conditions were a load per unit width of 587 N/mm (3,350 lb./in.), which resulted in a 1.72-GPa (250-ksi) pitch-line maximum Hertz stress. For purposes of this work, failure was defined as one or more visible spalls or pits that can also be detected by tracing over the pit with a sharp stylus. The visual appearance of the surface fatigue (Figure 6) was typical of other alloys tested in the same conditions. The gears were qualitatively judged to have good wear resistance. Results of the surface fatigue testing are summarized in Table III. Ten gear pair surfaces were tested. Five of the tests resulted in surface fatigue failures. The other five tests resulted in no surface fatigue and no impending indication of fatigue, and therefore can be considered as suspended tests.

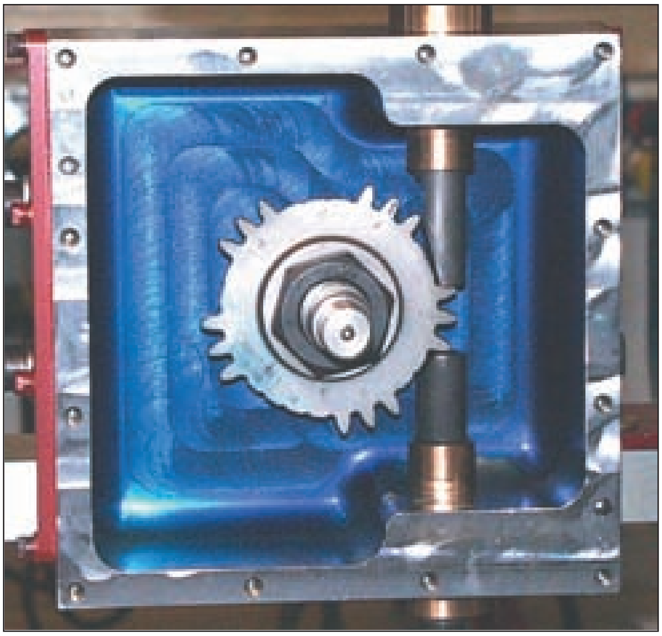


Figure 5—Test gear installed in test fixture, cover removed for photograph.

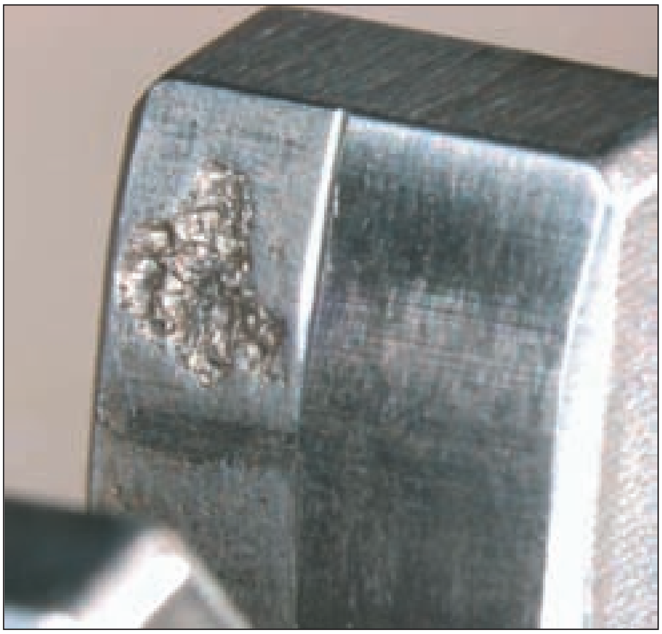


Figure 6—Typical surface fatigue failure of the Ferrium C69 alloy.

The experiments conducted were accelerated life tests. That is, for a helicopter application, the gears would operate at stress levels less than the stresses used for testing. Such an approach is used to economically produce surface fatigue data. To evaluate the significance of the surface fatigue test results, one can compare the results to past work done using different gear alloys but identical gear geometry specifications and test procedures. Often, fatigue test results are compared on the basis of the predicted ten-percent lives (Refs. 1, 5, 10). The present work comprised ten tests with five of the ten tests resulting in surface fatigue. An estimate of the ten-percent life from such a limited dataset provides for estimates with large statistical confidence intervals (Ref. 6). Therefore, for the present work, comparisons are made using estimates of the median (50 percent) lives. The distribution of fatigue life was modeled as a two-parameter Weibull distribution. The median life estimate was calculated by the method of Johnson (Ref. 9).

Krantz (Ref. 6) provided a summary of 18 sets of experiments conducted using test methods consistent with the present work, gears made from AISI 9310 steel, and various lubricants and surface treatments. Results of tests conducted using test methods consistent with the present work and using gears made from Pyrowear 53 steel have also been reported (Refs. 10, 11). Some of the referenced data are for gears having shot peening or superfinishing treatments to improve the surface fatigue lives. To provide the most direct comparison with the present study, the data for "standard-ground finished" gears (no shot peening or superfinishing of the contacting surfaces) were selected as the most appropriate baseline. Table IV provides a comparison of the results of the present study with the results of the previous works (Refs. 10, 13). The data of Table IV are the results for the best-performing sets of "standard-ground finished" gears made from the noted alloys and tested to date on the same gear test apparatus using the same test method. The estimated surface fatigue lives (median values) of Pyrowear 53, AISI 9310 and Ferrium C69 are 134, 200, and 361 million cycles, respectively. For contact stress of 1.7 GPa, as used in the present testing, 40 of the 54 tests of the baseline alloys exhibited surface fatigue before 300 million cycles. For the Ferrium C69 alloy, five of the 10 tests exceeded 300 million cycles without surface fatigue.

TABLE III—Surface Fatigue Test Results of Ferrium C69 Alloy

Test Number	Running Time (hours)		Running Time (million of cycles)	Test Result
1	64		38.3	surface fatigue
2	94		56.2	surface fatigue
3	296		177.3	surface fatigue
4	307		184.2	no failure — suspended
5	325		195.1	surface fatigue
6	566		339.3	no failure — suspended
7	572		343.2	surface fatigue
8	1,224		734.4	no failure — suspended
9	1,235		741.0	no failure — suspendedd
10	1,405		879.0	no failure — suspendedd
	Hours	Million cycles		
Sum of Run Times	6,147	3,688		

Bending Fatigue Test Results. Single-tooth bending fatigue testing was completed on a set of gears manufactured from Ferrium C69 steel. The set of gears was case-carburized and ground. Testing was done using unidirectional loading and load control. Testing was done until crack initiation occurred or, in some cases at the lowest levels of test load, tests were suspending with no cracks initiated. Crack initiation was defined to occur when the loading rod stroke increased approximately 2% (~ 0.0002 in. change) relative to the stroke for the gear tooth at test initiation.

Prior to conducting single-tooth bending tests on the test gears of the present study, tests had been conducted on gears made from AISI 9310 alloy (Ref. 12). In the previous work, crack initiation was defined to occur when the loading rod stroke increased approximately 2% (~ 0.0002 in. change) relative to the stroke for the gear tooth at test initiation. At this point a crack would be initiated with a size on the order of the case depth. Additional test cycles could then be applied to the AISI 9310 gears and stable crack growth would occur until the test was terminated with a crack of length 30~50% of the distance across the base of the tooth. The tooth with the fatigue crack could then be removed using a single impact load in order to observe the fracture surface. The present testing was initiated with the same definition of crack initiation in mind. However, it was observed that complete failure (removal of the tooth from the rim) occurred before a crack could grow in a stable manner to even the small size that would increase the loading stoke by 2%. Figure 7 provides examples of failed gear teeth. The differing crack trajectories illustrate the relatively brittle nature of the gear teeth made from the new alloy as compared to the baseline AISI 9310 gear teeth.

To express the intensity of the test loads, the maximum principal tensile stress in the fillet region was calculated using linear elastic finite element analysis (Ref. 12). The finite element model geometries were adjusted to closely match the actual manufactured tooth geometries as recorded using a digital microscope, including actual tooth thickness and root-fillet geometry details. The resulting stress owing to the applied load is reported herein as a stress index. It is appropriate to report test condition as a stress index since this test stress does not take into account the residual stress that exists in the case-carburized hardened gears. Furthermore, the reported test stress owing to the applied load was calculated assuming linear elasticity, and so plasticity and nonlinear effects were not considered. The relationships of crack initiation life as a function of the stress index for the present work and previous work (Ref. 12) are depicted in Figure 8 as a semi-logarithmic plot. For loads resulting in crack initiation within the cycle range 100–100,000 cycles, it is clear that AISI 9310 has superior performance relative to the new alloy. Also, the AISI 9310 exhibited excellent toughness, having stable crack growth, but the new alloy exhibited brittle behavior with fast fracture occurring over essentially the full base of the tooth in all cases. For consideration as an alloy for safety-critical gear applications such as main rotor transmissions of helicopters, such brittle behavior is unacceptable. It should be noted that S-N curves for case-carburized gear teeth are often depicted as two lines on semi-logarithmic axes, one line in the region of (100 cycles~5 million cycles) and another line for the very high cycle regime (> ~500 million cycles). The behavior of the new alloy in the very high cycle regime could not be investigated in a reasonable amount of time using the test method of this project (0.5 Hz loading frequency). Note that four tests of the Ferrium C69 steel were suspended with no failure after cycle counts approaching 1 million cycles. The trends of all the data from Figure 8 suggest that the Ferrium C69 gear might exhibit good performance against crack initiation in the very-high cycle regime.

(*Editors note: This work provides some data on the behavior of the newly developed alloy. The selection of an alloy for a particular application of course requires the consideration of many properties, and the present work was not a full evaluation. For example, the scoring resistance of the new alloy was not investigated. The fracture toughness of the gear tooth is very important for safety-critical aviation gearing, and the fracture toughness was not evaluated in this study. This study quantified some performance characteristics of the new alloy and has provided guidance for the design and development of next-generation gear steels.*)

Dr. Timothy Krantz *has been employed by the Army Research Laboratory Vehicle Technology Directorate since 1987. During this time, he's worked in cooperation with the NASA Glenn Research Center's Mechanical Components Branch. He works jointly with Army and NASA staff to advance mechanical component and system technology. His research interests include all aspects of drive systems including gear and bearing strength; fatigue and durability; gear manufacturing processes; gear noise and vibration; tribology; and condition-based maintenance. He is a member of the ASME Power Transmission and Gearing Committee.*

Brian Tufts *serves as manager-business development at QuesTek Innovations and is an expert in the design and use of high-strength steels. Tufts holds both an MS in materials science engineering from Northwestern University and an MBA in marketing/strategy from the Kellogg School of Management. His current work involves emerging high-strength alloys for use in aerospace applications, specifically for powertrain gearing and in landing gear. QuesTek Innovations LLC is a materials solutions company that designs and licenses proprietary alloys. Tufts has served as principal investigator on multiple design contracts and owns specific expertise in case-carburized gear and bearing steels.*

TABLE IV— SUMMARY AND COMPARISON OF GEAR SURFACE FATIGUE PERFORMANCE

Gear Material	Number of Failures	Number of Test Completed	Median Life (million cycles)
Pyrowear™ 53 (Ref. 10)	15	21	134
AISI 9319 (Ref. 13)	25	33	200
Ferrium™ C69 (present study)	5	10	361

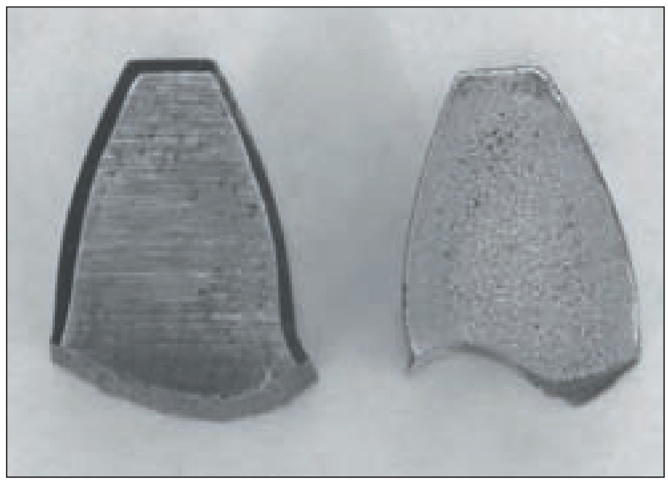


Figure 7—Typical failed teeth after single-tooth bending fatigue testing for Ferrium C69 (left) and AISI 9310 (right) steel alloys.

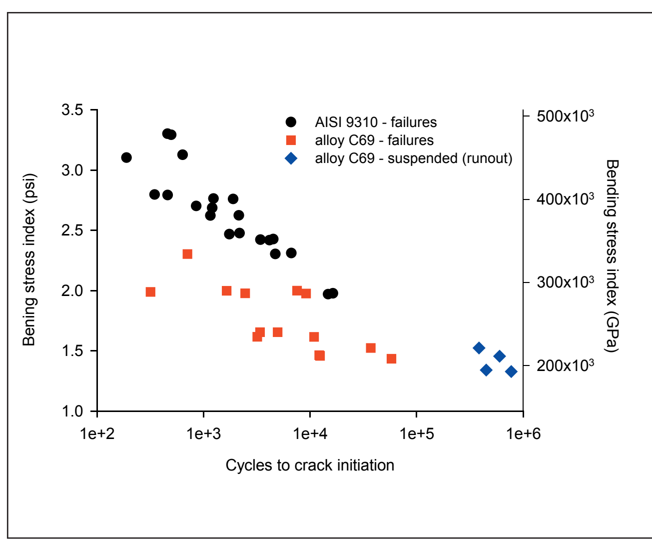


Figure 8—Results of single-tooth bending fatigue testing of Ferrium C69 and AISI 9310 steel gears (data for AISI 9310 are from Ref. 12).

This article was originally published by the American Society of Mechanical Engineers (ASME) for the ASME 2007 International Design Engineering Technical Conferences, and Computers and Information in Engineering Conference. It is republished here with the permission of the ASME.

Operational Condition and Superfinishing Effect on High-Speed Helical Gearing System Performance

R. Handschuh, C. Kilmain and R. Ehinger

Management Summary

An experimental effort has been conducted on an aerospace-quality helical gear train to investigate the thermal behavior of the gear system. Test results from the parametric studies and the superfinishing process are presented. The tests indicated that superfinishing offered no improvement in performance due to the high lubricant film thickness generated by the extremely high pitch line velocity at which the majority of the tests were conducted. Increasing lubricant inlet temperature had the most dramatic effect on performance improvement.

Introduction

High-speed, heavily loaded and lightweight gearing components are found in propulsion systems for rotorcraft. The high pitch-line velocity that is part of these systems makes the thermal aspects of the gear system design very important. Also, transmission systems used in certain applications have gear trains that provide the proper spacing between the parallel engine and rotor shafts (Ref. 1). These gear trains can cause additional thermal problems as the idler gears in this system receive two meshing cycles (on opposite sides of the gear teeth) per revolution. Therefore, weight-optimized aerospace drive system components can have difficulty when operation includes primary lubrication system failure.

In prior studies using this gear train system, testing has focused on basic operational characteristics (Refs. 2 and 3), as well as on the comparison of analytical predictions to experimental results (Ref. 4). The results from these studies have shown and quantified the effect of operational conditions on the power required to rotate the gear system at high speed and load. Also, gear windage was shown to be one of the primary gear mesh losses and is the least understood of the gearing loss mechanisms (Refs. 5 and 6). At the present time, gear windage reduction techniques have been trial-and-error. However, there is a performance benefit that can be realized if the design engineers can come up with an effective system to reduce these losses.

The objective of this study was to investigate the effect of lubricant input temperature and additional gear surface treatment beyond the baseline finish ground surface on performance parameters measured in the test facility. The baseline gears used in prior test programs were removed from the test facility and had the superfinishing process applied. The inlet temperature of the lubricant was varied from 160 to 250°F. Tests were conducted at these various operating conditions (including the inlet temperature) in the as-ground (including run-in) and with superfinishing applied. The superfinishing process has been shown to be very useful in contact fatigue life improvement (Refs. 7 and 8). This process has also been shown in other studies to reduce power loss and scoring loads when applied to the gear flanks.

Test Facility, Test Instrumentation and Test Hardware

Test Facility. The test facility used for this study is shown in Figure 1. The facility is a closed-loop, torque-regenerative testing system. There is a test gearbox and slave gearbox that are basically mirror images. Each gearbox has an input gear, three idlers and one bull gear. The gearboxes are joined together through the input gears and bull gears via shafting.

The facility is powered by a 500 hp DC drive motor, and its output speed is increased using a speed-increasing gearbox. The output of the speed-increasing gearbox then passes through a torque and speed sensor before connecting to the slave gearbox. The entire test stand configuration is shown in Figure 2.

Each gearbox has separate oil supply, and scavenge pumps and reservoirs. All flow rates have been calibrated at various temperatures and pressures prior to installation for accurate flow rate measurement. Lubrication system flow rate is controlled using the supply pressure. Temperature is controlled via immersion heaters in the reservoir and heat exchangers that cool the lubricant returned from the gearboxes. Each lubrication system has very fine 3-micron filtration. Nominal flow rate into the test or slave gearboxes at 80 psi is approximately 15 gpm.

The lubricant used in the tests to be described was a synthetic turbine engine lubricant (DoD–PRF–85734). This lubricant is used in gas turbine engines as well as the drive systems for rotorcraft.

Test Instrumentation. The test instrumentation used in this study included thermocouple rakes for locations across the facewidth and thermocouple arrays at the exit region of the helical gear axial pumping location (Ref. 4). The test instrumentation measured the fling-off lubricant from the gears in the radial and axial directions. Locations of the two different measurements are shown in Figure 3 (locations of rake and array probes, respectively). Shown in Figures 4 and 5 are photographs of the instrumentation rakes and arrays, respectively. The rake probes had five thermocouples across the face width, and the array sensors had nine thermocouples distributed as shown in Figure 3b. The thermocouple rakes were located at three positions, and as close as possible to the location of oil being flung radially out of mesh. The thermocouple arrays were centered at the axial point where the pitch diameter of the meshing gears meet.

Test Hardware. The test hardware used in the tests to be described is aerospace-quality hardware. The basic gear design information is contained in Table 1. The input and bull gear shafts have a combination of roller bearings with ball bearings to contain the resultant thrust loads, whereas the idler gears only have roller bearings. The partially disassembled test gearbox is shown in Figure 6. The bearing inner race is integral to the shafts on the idler gears and at other radially loaded bearings on the input and bull gear shafts. Shrouds for the gears were used to minimize the windage losses. Figure 7 shows the shrouds installed in the test gearbox.

The gears shown in Figure 6 were the as-ground com-

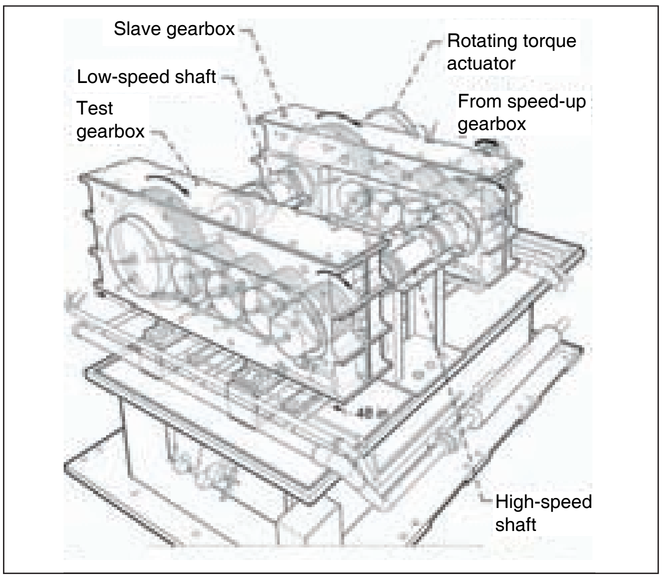


Figure 1—NASA High-Speed Helical Gear Train Test Facility.

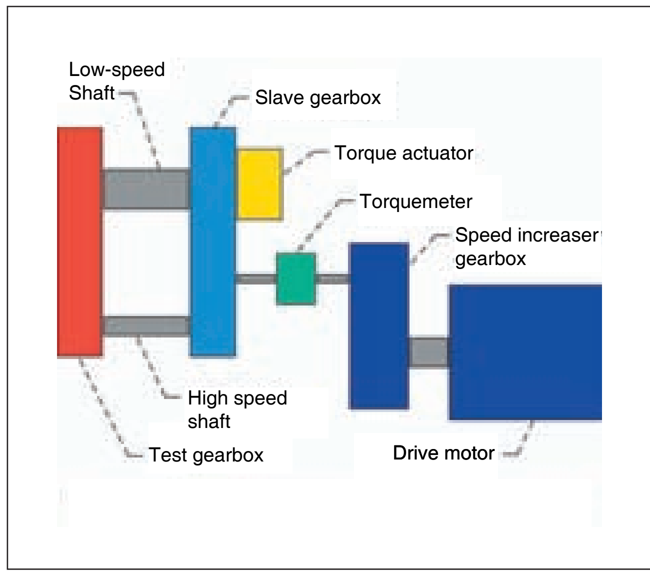


Figure 2—Layout of NASA High-Speed Helical Gear Train Test Facility.

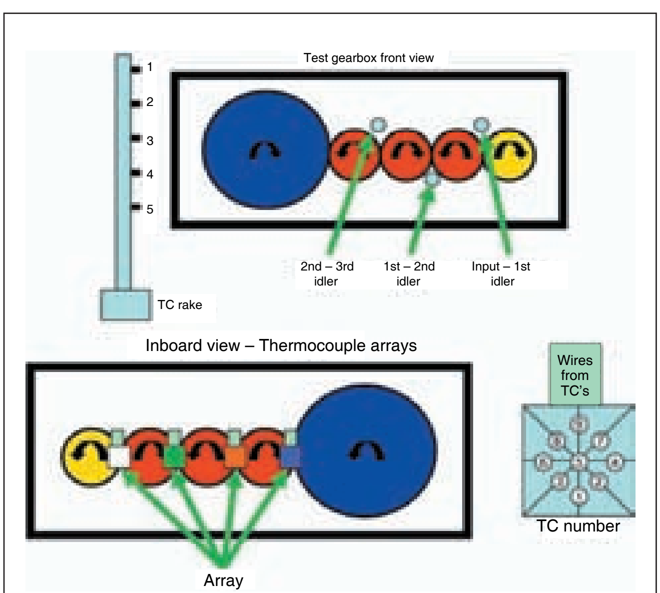


Figure 3—(a) Location and orientation of the thermocouple rakes in the test gearbox. (b) Location and orientation of thermocouple arrays in the test gearbox.

Figure 4—Photograph of thermocouple rake used in these tests.

Figure 5—Photograph of thermocouple array used in these tests.

TABLE 1 - BASIC GEAR DESIGN DATA	
Number of teeth input and 2nd idler	50
Number of teeth 1st and 3rd idler	51
Number of teeth Bull gear	139
Nodule, mm (diametral pitch (1/in.))	3.033 (8.375)
Face width, mm (in.)	67.2 (2.625)
Helix angle, degre	12
Gear material	Pyrowear 53


Figure 6—Baseline gear train installed in half of the test gearbox.

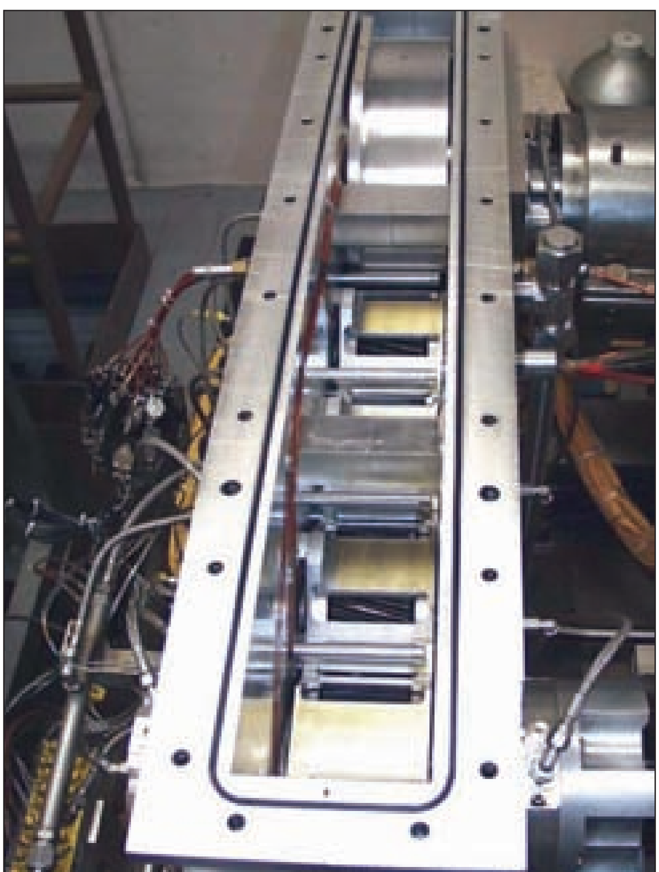


Figure 7—Test gearbox cover removed showing gear shrouding.

ponents. The gears were subsequently superfinished and reinstalled in the test gearbox. These gears are shown for the entire gear train in Figure 8 and a close-up in Figure 9.

The as-ground gears that were run extensively before superfinishing had an average surface roughness of 3.91 μin. Ra and 28.3 μin. Rz. The gear surfaces after superfinishing had a surface roughness of 1.71 μin. Ra and 13.2 μin. Rz, where Ra is the average surface roughness and Rz is the maximum surface roughness. These measurements represent an average of three measurements taken radially across one tooth on each gear.

Data Acquisition. The test facility data system monitors three important facility parameters during operation. Speed, torque (supplied torque and loop torque) and temperature measurements were made during all the testing conducted. The test system loop torque is measured on the shaft connecting the bull gears from the test and slave gearboxes. A telemetry system was utilized in this location.

The data recording system used in this study is capable of taking data from all parameters at a rate of one sample per second. Tests in this study recorded data every two seconds. The data is displayed to the test operator in real time. Data is stored in a spreadsheet format, and each sensor can be viewed at any time during a test and when post processing the results.

Test Operation. The test procedure for collecting the data to be presented was the following: For a given set of conditions (speed, torque, lubricant pressure and lubricant oil inlet temperature), the facility was operated for at least five minutes, or until the temperatures of interest had stabilized (or reached steady state).

Experimental Results. Tests were conducted and operational conditions maintained at steady-state conditions. As mentioned, the primary data from the testing include temperatures, speed and torque. The following data will be presented as the subject of this paper: temperature measurement location versus typical operational conditions; lubricant jet pressure (or flow); lubricant inlet temperature effects; and a comparison of the baseline to superfinished gears.

The first set of results is presented as an example of the large amount of data taken throughout this study. The test shown in Figure 10 was attained for the superfinished gears, keeping the lubricant inlet temperature and pressure constant. In Figure 10, the array (thermocouple #5) and rake (thermocouple #3) mid-temperature locations are shown for seven conditions of this particular test. The seven conditions included a warm-up; then three conditions of torque at 12,500 rpm; and finally the speed was increased to 15,000 rpm and the three torque conditions repeated. The test operational conditions can be found in Table 2 with a summary of some of the other data of interest. In this figure, typical for this gear system, the array temperatures were always much higher than the corresponding rake temperatures at the same location. As shown in this figure, the maximum temperature over the inlet lubricant temperature was on the order of 180°F.

Next, in Figure 11, a single meshing location is shown for all rake positions and for the vertical thermocouples from the array (thermocouples 1, 5 and 9) for the 2nd–3rd idler location. Once again the same data set as in Figure 10 and Table 2 was used. In Figure 11, the highest array temperatures exceeded any temperature on the rake. Also, the rake temperatures are not uniform across the face width. From the data shown, the difference in rake temperatures was as high as ~50°F.

Another test that was performed was to vary the lubricant jet pressure from the nominal 80 psig to two lower settings. This data is shown in Figure 12. In this figure, the as-ground and superfinished data are plotted for 33% of the maximum load and at two input rpm speeds. The results from either case were nearly the same, with a very slight reduction in power loss due to a reduction in the lubricant pressure from 80 to 60 psig.

In Figures 13 and 14, the effects of lubricant inlet temperature are shown for drive motor power required (for the entire test stand) and temperature increase across the gearbox, respectively. The gears in this test were superfinished; the load on the system was 100% and the lubricant jet pressure was 80 psig. In Figure 13, by varying the inlet temperature from 160 to 250°F, the power required decreased approximately 10 hp. In Figure 14, the temperature differential between inlet and exit of the test gearbox resulted in a much lower temperature differential across the gearbox as the inlet temperature was increased.

The last comparison to be made between the as-ground and superfinished gears will be made in Figure 15. The data provided in this figure show the drive motor power to rotate

Figure 8—Entire gear train after superfinishing.

Figure 9—Close-up of a superfinished gear.

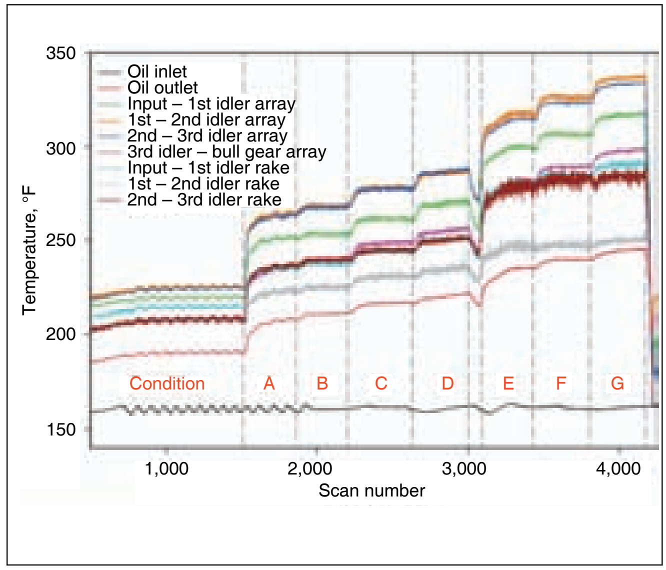


Figure 10—Superfinished temperature data at mid-face rake and at array center for all sensor locations (conditions shown in Table 2, one scan = 2 s, 160°F oil inlet temperature).

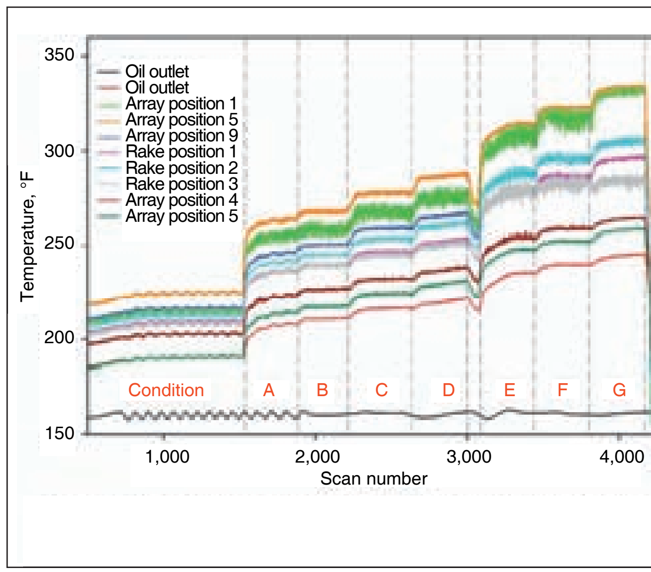


Figure 11—Superfinished gear test data comparing 2nd–3rd idler location for rake and array data (condition shown in Table 2, 160°F oil inlet temperature).

TABLE 2 - CONDITIONS FOR FIGURES 10 AND 11. TESTS WERE CONDUCTED AT 160 f OIL INLET TEMPERATURE ON SUPERFINISHED GEARS.

Condition	Input shaft speed	Loop power	Tem-perature increase across	Drive motor power
	(krpm)	(hp)	(F)	(hp)
A	Warm up			
B	12.5	1379	50.6	138.0
C	12.5	2801	55.1	149.2
D	12.5	4170	59.7	160.1
E	15.0	1657	73.8	201.9
F	15.0	3366	79.1	213.2
G	15.0	4986	83.0	225.1

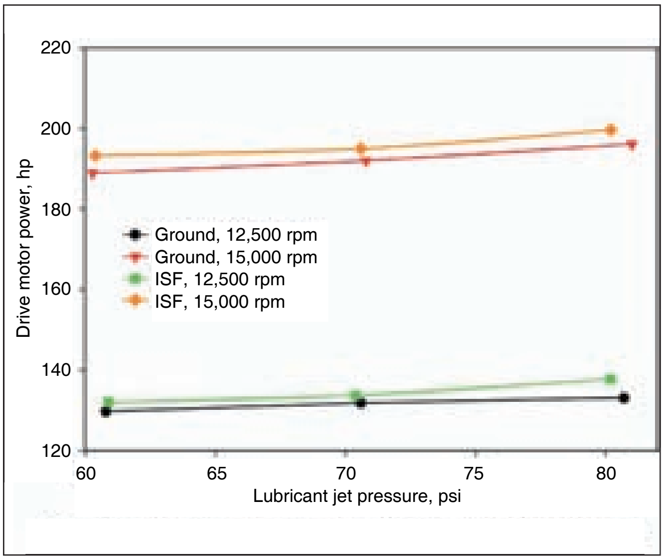


Figure 12—Effect of lubricant jet pressure on drive motor power required to rotate the entire test facility (~200°F oil inlet temperature, 33% of maximum torque applied).

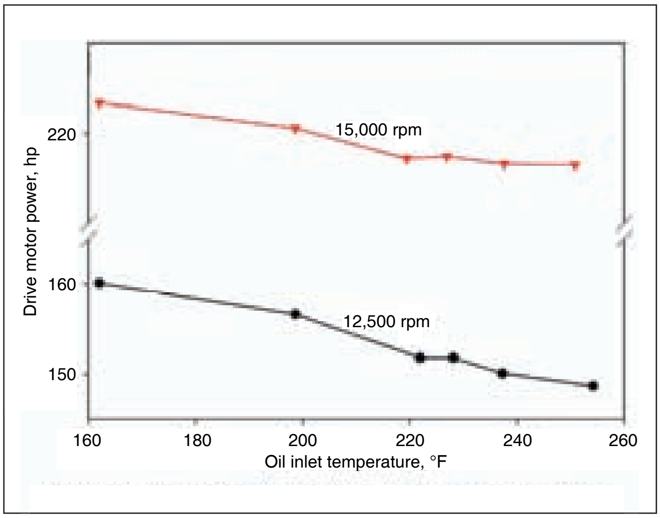


Figure 13—Effect of oil inlet temperature on drive motor power to rotate the entire test facility (superfinished gears, 80 psi lube jet pressure, 100% torque).

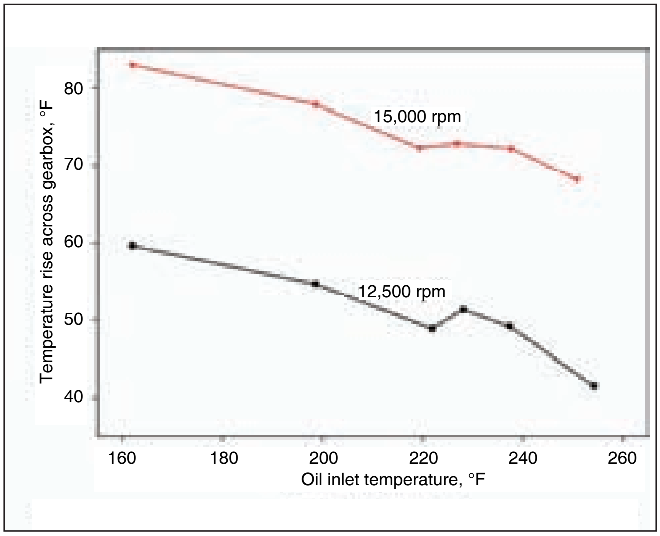


Figure 14—Effect of oil inlet temperature on the temperature rise across the test gearbox (superfinished gears, 80 psi lube jet pressure, 100% torque).

the test rig at two different speeds (12,500 and 15,000 rpm) and three different bull gear torques (~33, 67 and 100%). A linear behavior is shown with increasing load at constant speed. From this figure it can be concluded that improving the surface finish of these components had no efficiency benefits, as the data were nearly identical.

Discussion of Results. The question raised when reviewing the results of this study is "Why didn't superfinishing improve the performance of this high-speed gear mesh?" To determine why no benefit was found, an analysis of the input –1st idler gear mesh was made to determine the lambda ratio (lubricant film thickness/composite surface roughness of the meshing gears). For the analysis, the bulk temperature (gear and lubricant) was assumed to be 230°F. In Figures 16 and 17, the results of the analysis are shown. In Figure 16, the input shaft speed is held constant at 15,000 rpm and the surface roughness and input torque are varied. Only for the case where the surface roughness is <12 μin. would there be a lambda ratio >1. In Figure 17, the torque was held constant at 100%, and the surface roughness and input shaft speed were varied. In this case, the lambda ratio could be reduced below one for all conditions if the shaft speed was reduced to a low enough level at the full power condition. Therefore, superfinishing of the parts did not take a system that was in the mixed elastohydrodynamic condition and move it to the fully flooded condition where the lambda ratio is <1. In most of the tests in this study, for both as-ground and superfinished tooth surface conditions, the lambda ratio was two or greater.

Conclusions

For the results attained, the following conclusions can be made:

1. Superfinishing provided no measurable performance benefit to the high-speed gearing system under study. The film thickness to composite surface roughness was two or greater for most of the tests conducted.
2. Increasing lubricant inlet temperature provided the most beneficial effect to the performance of the drive system.
3. Thermocouple rakes and arrays installed in the test gearbox provided data that the fling-off temperatures vary with location across the face width of the gears, as well as the location within the gearbox where the temperatures were measured. The idler-idler gear meshes typically produced the highest rake and array temperatures measured in all tests.
4. The change in flow rate (due to lowering the lubricating jet pressure from 80 to 60 psig) had only a very minor effect on power loss.

References:

1. Kilmain, C., R. Murray and C. Huffman. "V–22 Drive System Description and Design Technologies," American Helicopter Society 51st Annual Forum, May, 1995.
2. Handschuh, R. and C. Kilmain. "Preliminary Investigation of the Thermal Behavior of High-Speed Helical Gear Trains," NASA/TM—2002-211336, ARL–TR–2661, March, 2002.
3. Handschuh, R. and C. Kilmain. "Efficiency of High-Speed Helical Gear Trains," NASA/TM—2003-212222, ARL– TR–2968, April, 2003.

4. Handschuh, R. and C. Kilmain. "Preliminary Comparison of Experimental and Analytical Efficiency Results of High-Speed Helical Gear Trains," ASME 2003 Design Engineering Technical Conference, September, 2003, Chicago, IL.
5. Handschuh, R. and C. Kilmain. "Experimental Study of the Influence of Speed and Load on Thermal Behavior of High-Speed Helical Gear Trains," NASA/TM—2005213632, ARL–TR–3488, July, 2005.
6. Handschuh, R. and C. Kilmain. "Operational Influence on Thermal Behavior of High-Speed Helical Gear Trains," NASA/TM—2006-214344, ARL–TR–3969, November, 2006.
7. Krantz, T., H. Alanou, H. Evans and R. Snidle. "Surface Fatigue Lives of Case-Carburized Gears With an Improved Surface Finish," NASA/TM—2000-210044, ARL–TR–2170, DETC2000/PTG–14373, April, 2000.
8. Krantz, T. "The Influence of Roughness on Gear Surface Fatigue," NASA/TM—2005-213958, ARL–TR–3134, October, 2005.

Dr. Robert F. Handschuh *has since 1982 been an aerospace engineer in the U.S. Army Research Laboratory, vehicle technology directorate, at the NASA Glenn Research Center in Cleveland. From 1984 to date, he has applied his expertise to power transmission, with a focus on experimental and analytical studies of planetary, spiral-bevel, face and high-speed gearing. The author of well over 100 technical papers, he holds a patent on a gas turbine engine shroud seal, and two patents on gas turbine bearing coking and minimization removal. An ASME fellow, he also serves on its power transmission and gearing committee.*

R. Ehinger *is a senior engineer in the drive system design group, and has been with Bell Helicopter-Textron since 2004. He received his bachelor's degree in mechanical engineering from Texas A&M University in 2003 while working summers as a draftsman and machinist for the oil industry. In addition, Mr. Ehinger is working on a master's degree in engineering management at Southern Methodist University. As project area lead for drive system research, he is responsible for the strategic planning and development of technology in areas such as materials, lubrication, configuration, efficiency and manufacturing. He is author of the AHS paper "Evaluation of Isotropic Superfinishing on a Bell Helicopter Model 427 Main Rotor Gearbox," and is co-author of "Application and Configuration Issues of Resin Transfer Molded Composite Transmission Housings—A Program Overview."*

Charles Kilmain *is the chief of tiltrotor drive system design and research with Bell Helicopter-Textron, where he has served since 1985. Kilmain received a BSME from the University of Maryland in 1985. His experience includes layout, detailed design and development testing for the V-22 proprotor gearbox (FSD); lead engineer on the V-22 proprotor gearbox (EMD); and lead engineer on the BA609 commercial tiltrotor midwing gearbox and interconnect shafting system. Kilmain is also the lead engineer for Bell's heavy lift rotorcraft drive system configuration and has served as drive system research project engineer for drive system research programs. Research programs include Advanced Rotorcraft Transmission II; H1 Composite Top Case; Advanced Gear Technology and High-Speed Helical Gear Loss-Of-Lube Evaluation at NASA Glenn. Publications include "V-22 Drive System Description and Design," and "Composite Applications for Rotorcraft Drive System Housings." He is also co-author of several works on thermal efficiency of high-speed helical gear trains and presentations on Advanced Rotorcraft Transmission II.*

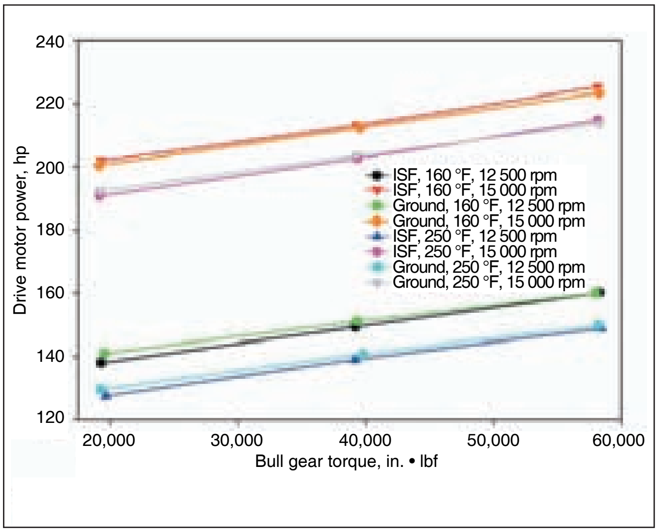


Figure 15—Comparison of the baseline (ground) to superfinished gears at two temperatures, two input shaft speeds and three load levels.

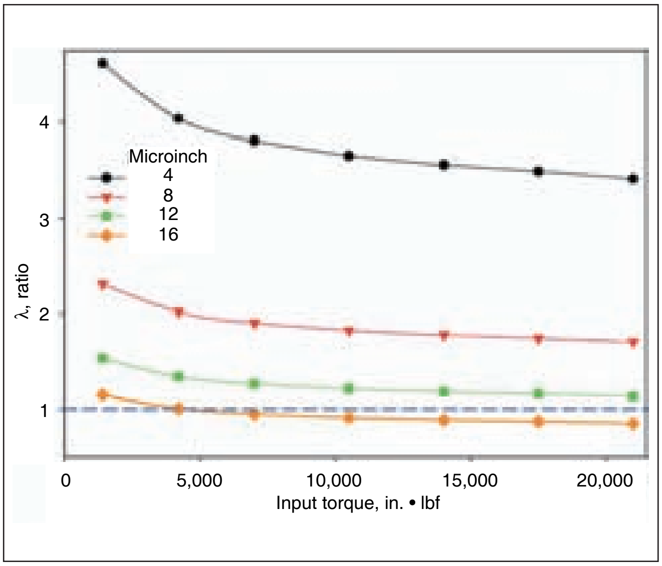


Figure 16—Effect of input torque on λ ratio for four different levels of surface finish at constant input shaft speed.

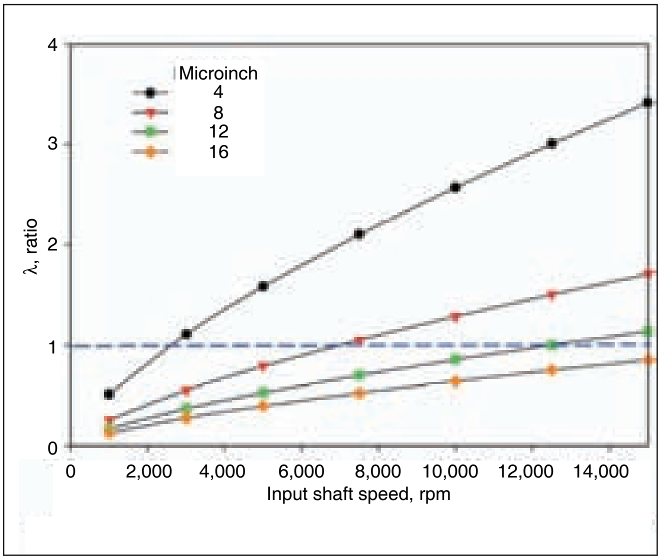


Figure 17—Effect of input shaft speed on λ ratio for four different levels of surface finish for constant torque.

A participant in the RoboCup German Open 2007 preps for the competition. (Image provided by Deutsche Messe).

Tapping Into Technology

AT HANNOVER 2008

It's safe to say Hannover Fair 2008 is big, and we're not just talking square feet or the number of exhibitors/attendees. Even the promotional brochure is a thick 35 pages. Each year professionals from 57 countries converge in Germany for technical presentations, networking possibilities and a chance to catch up on the latest market trends, innovations and business developments in the field of industrial technology.

"The energy and automation exhibits have increased in both size and scope in 2008, with a focus on climate protection, power stations and mobile robotics," says Katja Havemeister, press officer at Hannover.

One of the highlights will be the Partner Country presentation in which representatives from Japan will discuss innovations in robotics, micro- and nanotechnologies, environmental and energy technology, biotechnology and life science. A German/Japanese summit will bring senior politicians and business leaders together to discuss their specific roles in these various fields.

A presentation entitled "TectoYou," returns this year specifically targeted at students. During last year's fair, 23,500 school children participated in the event, which encourages kids to consider a career in technology.

Accompanying the energy exhibit this year is the World Energy Dialogue, a forum for energy experts to discuss the demand for energy in growth markets such as China and India.

"The World Energy Dialogue provides an opportunity for knowledge and opinion sharing on the subject of climate-friendly, energy efficient power plant technologies," says Sepp D. Heckmann, chairman of the managing board at Hannover.

Power Plant Technology is a new flagship exhibition that gives the power plant industry a platform for the latest developments in engineering. Themes include large-scale power station construction, decentralized industrial power plants and all the peripheral technologies and services.

"By choosing to focus on key themes of energy and automation this year," says Dietmar Harting, chairman of the exhibitor's advisory committee, "Hannover proves once again it has its finger on the pulse of technology. We're fully behind this strategy and plan to give the global public answers to questions on climate protection and energy efficiency."

A new presentation entitled "Mobile Robots & Autonomous Systems" showcases the rising demand for robots in three target groups in the industrial, public and private sector. The mobile robots presentation focuses on the use of robotics in everything from transportation systems and inspection to lawn mowing and cleaning. The international RoboCup German Open, a fun sporting competition on robotic technology, will once again take place in the mobile robotics hall.

The "TechtoYou" program in 2007 featured a host of events for budding engineers.

Power Plant technology is now a flagship exhibition at Hannover.

For the 5th time in Hannover history, a company will win the Hermes Award, an international technology prize that highlights groundbreaking innovations and solutions tested by industry professionals. The award is limited to products that are on display for the very first time at Hannover in 2008. In 2007, the award was jointly won by Bayer Technology Services in Germany and Ingenia Technology Ltd. in London.

Hannover Fair takes place from April 21–25 at the fairgrounds in Hannover, Germany. More than 200,000 industry professionals will attend the 10 flagship international trade shows that make up the event. Exhibitors have the opportunity to launch new products, meet face to face with customers, establish distribution channels and generate brand awareness. The 10 flagship trade fairs for 2008 include factory automation, process automation, industrial building automation, digital factory, subcontracting, energy, power plant technology, pipeline technology, micro technology and research and technology. For registration and contact information, call (609) 987-1202 or visit *www.hf-usa.com*.

Images provided by Deutsche Messse.

March 31–April 3—WESTEC 2008 Exposition and Conference. Los Angeles Convention Center, Los Angeles. With a focus on technology and professional advancement, WESTEC offers free educational programs including sessions on comparative technologies, lean principles and business management strategies in manufacturing. This year's conference features a back-to-basics program offering pathways to manufacturing careers. The Society of Manufacturing Engineers sponsors the event. For more information, contact the SME resource center at *service@sme.org*.

March 31–April 4—METAV 2008. Düsseldorf, Germany. This trade fair highlighting manufacturing and automation covers the entire scope of machine tools, precision tools, automation components and systems. Exhibitors and visitors can share information, discuss industry trends and network for job opportunities. The international symposium this year will focus on energy efficiency in production. For more information, visit *www.metav.messe-duesseldorf.de*.

April 2–4—AGMA's Gear Manufacturing Technology Course. Hoffman Estates, IL. Offered by the Gear Consulting Group, this course will run participants through practical aspects of gear manufacturing. The course will be instructed by Geoff Ashcroft and Ron Green. Topics include gear theory, inspection, manufacturing, hobbing, shaping, estimations, hard finishing and gear shaving. Tuition for the three-day course is $750 per person and includes all necessary materials, AGMA reference manual and a certificate of completion. For more information, contact Gear Consulting Group at (269) 623-4993.

April 4–13—SIMTOS 2008 Korea International Exhibition Center (KINTEX), Seoul. The 13th Annual Seoul International Machine Tool Show 2008 will include an estimated 500 companies with 3,500 exhibition booths. Companies will discuss the latest trends and display new brand designs for the global machine tool market. More than 70,000 visitors from 30 countries participated in the 2007 exhibition. The event is organized by the Korea Machine Tool Manufacturers Association. For more information, visit *www.simtos.org*.

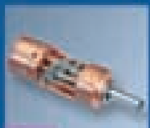
April 14–17—SAE 2008 World Congress. Cobo Center, Detroit. Geared toward sharpening the skills of budding automotive engineers and training managers, the SAE 2008 World Congress includes 1,500 technical papers on topics such as propulsion and powertrain, electronics, materials, emissions and environment, management and marketplace, and safety and testing. The congress offers 40 SAE seminars, a global exhibition show floor, a technology pavilion and an international booth. Attendees represent fields in engineering, management, purchasing, manufacturing, military, academia and research in automobile technology. Sessions new to the 2008 program include "Designing On-Board Diagnostics for Light- and Medium-Duty Emissions Control Systems," "High-Performance Engine Design and Development," "Automotive Product Lifecycle Management" and many others. For registration and contact information, call (877) 606-7323 or visit *www.sea.org/seminarinfo*.

April 22–25—Control: International Trade Fair for Quality Assurance Stuttgart, Germany. Control is the world's only trade fair focusing strictly on quality assurance. Companies exhibit new products, systems and solutions in measuring technology, material testing, analysis and optoelectronics. The trade fair presents updated hardware and software tools for the manufacturing sector. It assists companies in maintaining higher manufacturing and quality requirements at all production facilities. Advances in automotive, communications and microsystems technologies will be discussed. For more information, visit *www.control-messe.com*.

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, and wire dies. The 2008 conference primarily focuses on increased productivity, new technological developments and ways to reduce business costs. For more information, contact Intertech 2008 at *www.intertechconference.com*.

May 13–15—Porous Materials, Inc. Short Course Ithaca, NY. "Advanced Techniques for Pore Structure Characterization; Theory and Practice" will cover the basic principles, current practices and critical assessments of pore structure characterization. The course includes three days of informative lectures as well as hands-on demonstrations in their testing laboratory. Interested participants are encouraged to bring samples with them for testing. Attendance is flexible, from one to three days. For registration information, contact Dr. Krishna Gupta at (607) 257-5544 or *info@pmiapp.com*.

May 20–22—EASTEC 2008 Advanced Productivity Exposition. West Springfield, MA. 14,000 manufacturers, plant managers, and shop owners visit EASTEC to evaluate advanced technologies, production methods and management concepts. From the latest machine tools to lean strategies, the exposition keeps the East Coast manufacturing community up-to-date on the competition. The Society of Manufacturing Engineers sponsors the event. For more information, call (800) 733-4763 or contact the SME resource center at *service@sme.org*.



Donald R. McVittie

1930–2008

Donald R. McVittie

It is with regret we report that Donald R. McVittie (1930–2008) passed away January 20, 2008 at his Seattle home. He was 77. Mr. McVittie was diagnosed several years ago with lung cancer.

At the request of his family, AGMA has created a memorial scholarship fund—the Donald R. McVittie Memorial Education Fund—to help fund the education of gear engineers. Contributions should be sent to the AGMA Foundation, 500 Montgomery Street, Suite 350, Alexandria, VA 22314 and designated for the Donald R. McVittie Memorial Education Fund.

McVittie is survived by his wife, Red, three children and four grandchildren.

Born in western New York, McVittie attended schools in Niagara Falls, NY and Portland, OR. He graduated from the University of Michigan in 1952 with a BSE in Mechanical Engineering.

While attending Michigan, McVittie worked for a local engineering firm which specialized in paper mill processes and power generation. After graduation, he moved to Seattle and worked in marine engineering, designing, selling and manufacturing a variety of innovative auxiliary machines, mostly for the commercial fishing industry.

In 1962 he moved to Chile, where he supervised the design, procurement, construction and startup of an integrated fish production and processing operation in Iquique, on the north coast of the country.

McVittie joined The Gear Works—Seattle, Inc. in 1969 as executive vice president and was responsible for operations until his retirement in December, 1986.

In 1973 he helped form Gear Engineers, Inc., an engineering corporation specializing in gear design, analysis of problem gears and computerized gear capacity studies. He was the principal engineer of the company since its organization, and in 1983 became its president.

McVittie was an active participant in the American Gear Manufacturers Association since 1972. His main interest was the Technical Division, where he served on many committees, including the Gear Rating, Wind Turbine Gearing, Inspection & Handbook, Epicyclic Enclosed Drives and Computer Programming committees.

He was President of AGMA in 1984-1985 and served as Vice President of AGMA's Technical Division from 1986 to 1991. In 1991, he became one of the original technical editors for *Gear Technology* magazine, and continued in that capacity until his illness.

McVittie received every possible award and recognition given by AGMA—the Technical Division Executive Committee Award, the Product Division Executive Committee Award, the Board of Directors' Award, the E. P. Connell Award and the Lifetime Achievement Award for service to AGMA. He was elected an honorary life member of AGMA in 1998. Additionally, he received ANSI's Meritorious Service Award in 1995 for his work in international standardization.

McVittie was a licensed professional engineer in Washington, and a member of the American Society of Mechanical Engineers, the American Society for Metals and the Society of Automotive Engineers.

A memorial service was held March 2nd at the Corinthian Yacht Club in Seattle.

Sicmat

CELEBRATES TWO MILESTONES

Italian machine tool builder Sicmat S.p.A. celebrated its 75th year of operation with the announcement that it reached a two-year goal by shipping its 50th gear shaving machine to North America.

Sicmat, established in 1932 and based in Pianezza near Turin, Italy, originally produced a variety of universal machine tools, including radial drilling machines, shaping machines and hydraulic presses. In 1950, the company began manufacturing gear shaving and chamfering machines as its main products.

"A middle/small enterprise is not able to produce everything, and the competitive pressure forced us to become specialists in one type of technology," says Sicmat managing director Ettore Miletto. According to Miletto, Sicmat began producing gear shaving machines locally in Italy because traditional methods for gear finishing were too slow, and importing the technology from the United States, where it was first developed by National Broach, was too expensive.

Today, Sicmat is one of only about four manufacturers of CNC shaving machines in the world. The company supplies gear shaving machines to the world's largest automatic transmission manufacturer, as well as many other companies.

"We have made big investments in shaving technology and created a wide range of products, but our main focus is to analyze customer requirements, to share our know-how and to engineer the most technical and economical solutions," Miletto says.

Sicmat offers two main gear shaving products, the RASO 200 and the RASO 400.

The RASO 200 has a slant-bed structure that gives it the stiffness of larger machines. Designed for both large manufacturers as well as smaller operations, it has options for loading systems, layout and number of axes. Its flexibility and high cycle speed performance enable its use by manufacturers of automatic gearbox gears, motorcycle gears and pump gears.

The RASO 400 employs a large, open “C” structure, allowing it to shave shafts and gears in varying sizes. When combined with internal and external automation, it can also handle auxiliary operations, including chamfering, deburring and centrifugation (spin off) marking. The RASO 400 is designed with the power and stiffness required for producing gears with large modules and face widths, such as those for agricultural equipment, earthmovers and industrial vehicles.

According to a written statement, Sicmat has an ongoing series of research and development projects, including integrating shaving machines with deburring and chamfering machines in order to eliminate multiple fixtures in production lines. Sicmat has also patented a new deburring and chamfering process that uses linear tools rather than rotational cutters.

For more information:

Mark Ritchie
Star SU LLC
5200 Prairie Stone Parkway
Ste. 100
Hoffman Estates, IL 60192
Phone: (847) 649-1450
Fax : (847) 649-0112
sales@star-su.com
www.star-su.com/RASO

PMA

RELEASES JANUARY BUSINESS REPORT

According to the January 2008 Precision Metalforming Association (PMA) Business Conditions Report, metalforming companies expect business conditions to remain steady over the next three months despite low shipping levels. The monthly report samples economic information from 159 metalforming companies in the United States and Canada.

When asked what they expect the trend in general economic activity to be over the next three months, 20 percent of participants reported that conditions will improve, 50 percent anticipate activity will remain the same and 30 percent expect a decline in business conditions. These same percentages were reported in December 2007.

Continued

Metalforming companies also expect little change in their incoming orders over the next three months. Thirty-one percent of companies predict an increase in orders (down from 33 percent in December), 44 percent anticipate no change (compared to 37 percent last month) and 25 percent forecast a decrease in orders (down from 30 percent in December).

Current average daily shipping levels, however, plummeted in January to their lowest levels since January 2002. Only 13 percent of companies reported that current shipping levels are above levels of three months ago (compared to 24 percent in December), 43 percent reported no change (the same percentage reported last month) and 44 percent reported that January shipping levels are below levels of three months ago (up from 33 percent in December).

The number of metalforming companies with a portion of their workforce on short time or layoff fell to 14 percent in January, down from 18 percent in December.

"PMA member companies are reflecting overall economic uncertainty in their projections for the first quarter of 2008, especially compared to their outlook one year ago, in January 2007, which was decidedly more positive than it is today," says William E. Gaskin, PMA president. "While daily shipping levels at year-end fell sharply, expectations for incoming new orders over the next three months remain positive, with 75 percent expecting higher or at least similar levels of customer orders as received in the fourth quarter of 2007. Profits in 2008 will be challenged by constant price pressure from customers and rising steel and energy costs."

Full report results are available at *www.pma.org/about/stats/BCreport.*

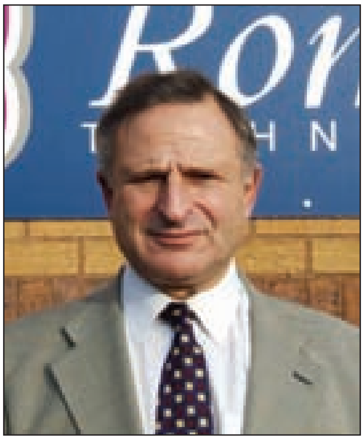
Bernstein

JOINS ROMAX ENGINEERING STAFF

Romax Technology recently announced that Charles Bernstein has joined its engineering department. Bernstein has over 40 years of experience in design and development in the transmission and automotive industries. According to the company's press release, Bernstein recently took the technical lead in the development of transmissions at FAW and Tsingshan in China. Before joining Romax, Bernstein was a senior manager in transmission engineering at BMW, Rover Group and Land Rover.

"I'm looking forward to being part of this fast-growing and dynamic company and bringing with me another dimen-

Continued



Charles Bernstein

sion to their engineering capabilities," Bernstein says. "There are many business opportunities and great challenges ahead."

Barry James, engineering manager at Romax, adds, "Charles' addition ensures practical implementation of our advanced technology, allowing us to expand the scope of our engineering services and enhance our existing capabilities, providing a complete package to our customers from design to development through to manufacturing."

Fairfield

UNVEILS NEW RESEARCH AND DEVELOPMENT CENTER IN INDIANA

Fairfield Manufacturing Company, Inc. officially opened the James R. Dammon Center with a ribbon-cutting ceremony in December of 2007. According to the company's press release, the 8,800-square-foot engineering center in Lafayette, IN will be equipped with a broad range of testing equipment for mechanical-hydraulic-and electric-powered applications.

The ribbon-cutting and dedication ceremony was hosted by Fairfield President and CEO Gary J. Lehman.

Joining in the festivities were Lafayette, IN Mayor Tom Roswarski, State Senator Ron Alting, State Representative Sheila Klinker as well as key representatives from Purdue University.

The research and development lab was dedicated to James R. Dammon, vice president of engineering at Fairfield, who joined the company in 1966 after graduating from Purdue University's School of Engineering.

Dammon's list of accomplishments at Fairfield includes pioneering the development of electric drive gear assemblies for an industrial forklift application as well as a fully integrated drive assembly for a mobile scissors lift.

"Jim Dammon's innovative spirit, dedication and commitment to Fairfield speak for itself," Lehman says. "Dedicating this facility to Jim is just a small recognition of our appreciation for Jim's 41 years with this company and all that he's done to put Fairfield out in the front as a technology leader in the industry."

Dedicating the new research complex to Dammon was kept a secret at Fairfield until the ribbon-cutting ceremony last December. According to attendees of the event, Dammon was shocked, surprised and honored for the recognition of his accomplishments. For more information, visit *www.fairfieldmfg.com.*

Ipsen Inc.

ADDS 12 TO ROSTER IN 4TH QUARTER

Ipsen, Inc., a designer and manufacturer of industrial vacuum and atmosphere furnaces, recently expanded its North

Continued

American roster with 12 new hires in the 4th quarter of 2007. The positions have been filled in engineering, production, quality control, purchasing and aftermarket services. A strong showing in 2007 allowed the company to expand its staff.

"Adding a balanced mix of experienced and entry-level positions at the Rockford (IL) and Souderton (PA) sites assures that we can deliver high performance for years to come," says John Menne, human resource manager at Ipsen. "We are taking initiatives now to re-launch a revitalized Ipsen internship program in the first half of 2008 to ensure that we also train the experts of tomorrow."

Hypertherm

ADDS TWO MEMBERS TO NORTH AMERICAN TEAM

John Brennan

Hypertherm, a designer and manufacturer of plasma cutting technology, recently announced the appointment of John Brennan as North American distribution director, and Randy McMurtry as national distribution development manager.

Brennan brings 27 years of experience to the job, including work as a vice president of marketing for Merriam Graves and 19 years of welding and cutting industry experience at Hypertherm. Brennan most recently served as the company's national distribution manager.

McMurtry recently served as a divisional manager for Hypertherm. In his new position, he'll support the national distribution network, working to execute sales and marketing programs. He will also ensure distributors have the knowledge to sell plasma cutting systems and consumables.

"The appointment of John and Randy to these two new positions brings added strength to Hypertherm's North American team," says Jeff Deckrow, Hypertherm's North American regional director.

Randy McMurtry

ADVERTISERS INDEX

Use this index to contact advertisers

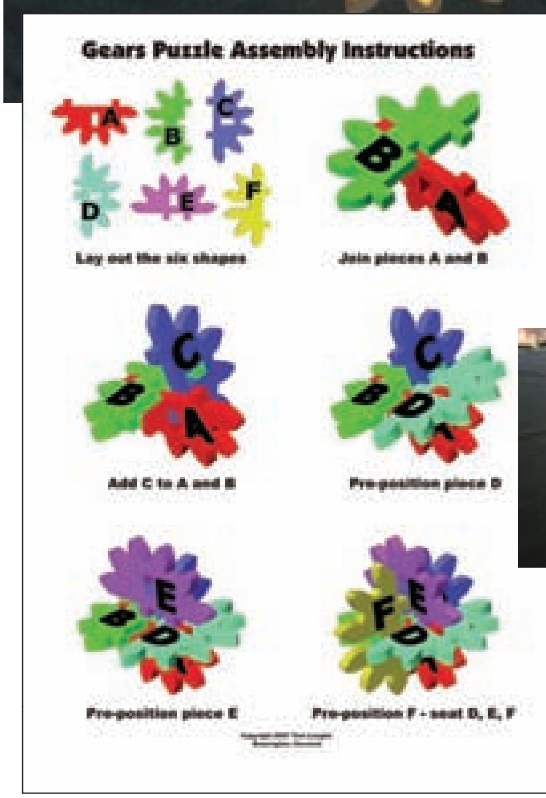
ADVERTISER	PAGE	PHONE	E-MAIL	INTERNET
American Metal Treating Co.	79	(216) 431-4492		www.americanmetaltreating.com
American Wera	64	(734) 973-7800		www.american-wera.com
Arrow Gear Co.	35	(630) 969-7640		www.arrowgear.com
B&R Machine & Gear Corp.	IBC	(731) 456-2636, (800) 238-0651	inquiry@brgear.com	www.brgear.com
Bourn & Koch Machine Tool Co.	71	(847) 649-1450	sales@star-su.com	www.star-su.com
The Broach Masters/Universal Gear Co.	16	(530) 885-1939	info@broachmasters.com	www.broachmasters.com
Butler Gear	79	(262) 781-3270	gears@butlergear.com	www.butlergear.com
Circle Gear & Machine Co.	79	(708) 652-1000		www.circlegear.com
Clifford-Jacobs Forging Co.	13	(217) 352-5172	sales@clifford-jacobs.com	www.clifford-jacobs.com
Cole Manufacturing Systems Inc.	78	(248) 601-8145	dsmith@colemfgsystems.com	www.colemfgsystems.com
Colonial Tool Group Inc.	15	(734) 946-2733	info@colonialtool.com	www.colonialtool.com
Comtorgage Corp.	18	(401) 765-0900	kgradolf@comtorgage.com	www.comtorgage.com
Donner + Pfister	70,74	(585) 586-5746	gearmet@frontiernet.net	www.dpag.ch
Dragon Cutting Tools	35	+(82) 32-814-1540	dragon96@kotis.net	www.dragon.co.kr
Engineered Tools Corp.	8	(248) 619-1616	rdeneau@engineeredtools.com	www.engineeredtools.com
Fässler Corp.	6	(414) 769-0072	sales@faessler-ag.ch	www.faessler-ag.ch
Forest City Gear Co.	40–41	(815) 623-2168	fyoung@fcgear.com	www.fcgear.com
Gear Consulting Group	78	(269) 623-4993	gearconsulting@aol.com	
Gear Manufacturing Inc.	79	(800) 773-GEAR		www.gearmanufacturing.com
Gear Motions Inc.	79	(315) 488-0100	sales@nixongear.com	www.gearmotions.com
The Gear Works—Seattle Inc.	79	(206) 762-3333	sales@thegearworks.com	www.thegearworks.com
Haas Automation	36	(800) 331-6746		www.haascnc.com
Havlik Gear	24	(519) 624-6240	havlik@sympatico.ca	www.havlikgear.com
Höfler	OBC	(734) 416-9300	inquiries@greatlakesgeartech.com	www.greatlakesgeartech.com
Index Technologies	78	(440) 995-4627		www.gallenco.com
Ingersoll Cutting Tools	72	(815) 387-6600	ictc@ingersoll.com	www.ingersoll.com
Insco Corp.	70	(978) 448-6368	sales@inscocorp.com	www.inscocorp.com
Kapp Technologies	3	(303) 447-1130	info@kapp-usa.com	www.kapp-usa.com
LMC Workholding	33	(574) 735-0225	info@logan-mmk.com	www.logan-mmk.com
Machine Tool Builders Inc.	27	(815) 636-7502	kflowers@machinetoolbuilders.com	www.machinetoolbuilders.com
McInnes Rolled Rings	34	(800) 569-1420	sales@mcrings.com	www.mcinnesrolledrings.com
Metal Finishing News magazine	76	(212) 633-3100	metalfinishing@elsevier.com	www.metalfinishing.com
Micro Surface Corp.	78	(408) 723-0700	info@ws2coating.com	www.ws2coating.com
Midwest Gear & Tool Inc.	73	(586) 779-1300	midwestgear@sbcglobal.net	
Mitsubishi Gear Technology	10	(248) 669-6136	info@mitsubishigearcenter.com	www.mitsubishigearcenter.com
Mutschler Edge Technologies	78	(440) 748-2520	info@mutschleredgetech.com	www.mutschleredgetech.com
Nachi Machining Technology Co.	7	(586) 263-0100	sales@nachimtc.com	www.nachimtc.com
Nelson Engineering	79	(714) 893-7999	sales@nelson-eng.com	www.nelson-eng.com
Precipart	68	(631) 694-3100	sales@precipart.com	www.precipart.com
Precision Gage Co. Inc.	74	(630) 655-2121	sales@precisiongageco.com	www.precisiongageco.com
Presrite Corp.	29	(216) 441-5990		www.presrite.com
Process Equipment	22	(800) 998-4191	msdsales@processeq.com	www.gearinspection.com
Proto Manufacturing	73	(519) 737-6330		www.protoxrd.com
The Purdy Corp.	20	(860) 649-0000	finance@purdytransmissions.com	www.purdytransmissions.com
Reishauer Corp.	78	(847) 888-3828	reishauer-us@reishauer.com	www.reishauer.com
Riverside Spline & Gear	19	(810) 765-8302	valerief@splineandgear.com	www.splineandgear.com
Schafer Gear Works Inc.	21	(574) 234-4116		www.schafergear.com
Schnyder S.A.	71	(630) 595-7333	hanikcorp@aol.com	www.hanikcorp.com
The Shot Peener magazine	75	(800) 832-5653, (574) 256-5001		www.shotpeener.com
Sigma Pool	5	(734) 429-7225	info.lgt@liebherr.com	www.sigma-pool.com
Solar Atmospheres	79	(800) 347-3236		www.solaratm.com
Star SU LLC	IFC–1,22,71,78	(847) 649-1450	sales@star-su.com	www.star-su.com
Stock Drive Products/Sterling Instrument	17	(516) 328-3300		www.sdp-si.com/e-store
Ticona	23	(800) 833-4882	info@ticona.com	www.ticona.com
TPS	78	(800) 423-4031		www.tpsintl.com
TSA America	14	(440) 614-0170		www.tsageartools.com

SERVICE

GEAR MANUFACTURING

HEAT TREATING

For The Love of Gears!



Digital sculptor extraordinaire Tom Longtin has always had an interest in gears. The aesthetics of their motion and interaction have been appealing to the sculptor/graphic designer based in Bennington, Vermont. So much so, he's created several award-winning animations on the subject.

As a computer programmer for Cranston/Csuri Productions in Columbus, Ohio, Longtin created several surreal gear designs for a variety of computer graphics applications. An ad agency for an aerospace/automotive manufacturer asked Longtin and his co-workers to incorporate gears into a 30-second television animation entitled, "The Best Idea in the World."

He created a full-screen field of 50-odd metallic synchronized, rotating gears using the book "Spur Gears," by Earle Buckingham as reference. The commercial spot won Best Corporate TV Commercial by *Computer Pictures* magazine in 1985. In addition, Longtin's work has been featured on magazine covers and his animations were used on Miramar's *The Mind's Eye* video.

Longtin enjoys creating 3D sculptures in wood and plastics. As a designer, he appreciates the role gears played in the industrial revolution and their use in nearly every mechanical device known to man.

"Most people readily identify with the iconic status of gears," says Longtin. "You can see this in the graphic caricatures seen in many advertisements."

Longtin belongs to a local artist guild in Bennington that promotes his work. When he came across instructions for constructing puzzles on Pavel's blog at *www.keltis.us/puzzles*, he decided to create and sell a few basic puzzles of his own using laser-cut MDF board and acrylic. He then moved on to a six-piece 3D apple puzzle and a pumpkin puzzle by rounding out the perimeters of the three intersecting squares using *Rhino* software.

"Having a long history of producing gear-themed images and animations," Longtin says, "it followed that I should do a puzzle with gears."

And so he did.

Longtin sells the laser-cut yellow expanded polystyrene models for $10 and can fill order requests via e-mail. He'd also be more than happy to provide the puzzle layout for free so fellow gear aficionados can make their own.

Currently, he's hard at work on a gear-themed peace symbol car magnet and contemplating future gear-related projects.

For more information on Longtin's work, visit *http://design.osu.edu/carlson/history/ACCAD-overview/overview4.html*. To contact him for your own gear-themed puzzle e-mail Tom Longtin at *tom-longtin@fulcrum-design.com*