gear TECHNOLOGY®

JUL 2015

INSPECT THE UNEXPECTED

IMPROVING QUALITY ASSURANCE ON THE SHOP FLOOR

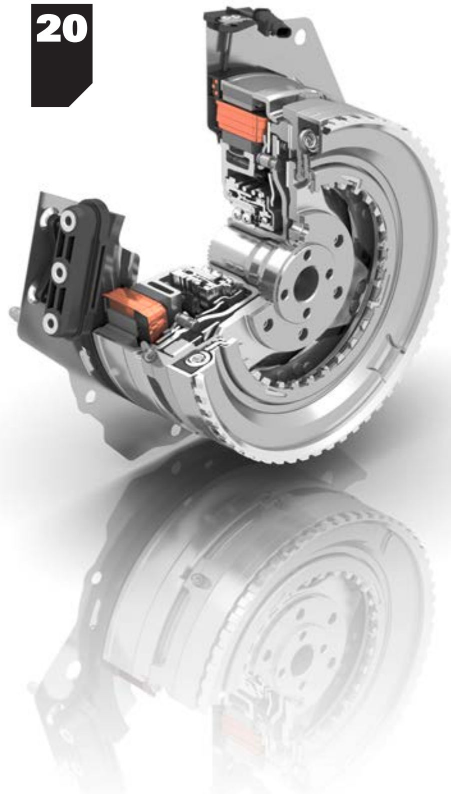
MAKING BIG GEAR MEASUREMENT A MUCH SMALLER TASK

PLUS:

HEAT TREATING EXPERTS HELP YOU STAY OUT OF HOT WATER

gear TECHNOLOGY

JUL 2015



20

48

32

features

technical

Vol. 32, No. 5 GEAR TECHNOLOGY, The Journal of Gear Manufacturing (ISSN 0743-6858) is published monthly, except in February, April, October and December by Randall Publications LLC, 1840 Jarvis Avenue, 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 Publications 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, 1840 Jarvis Avenue, Elk Grove Village, IL, 60007.

gear TECHNOLOGY
Vol. 32, No. 5

departments

Cover photo by David Ropinski.

THE GEAR INDUSTRY'S INFORMATION SOURCE

www.geartechnology.com

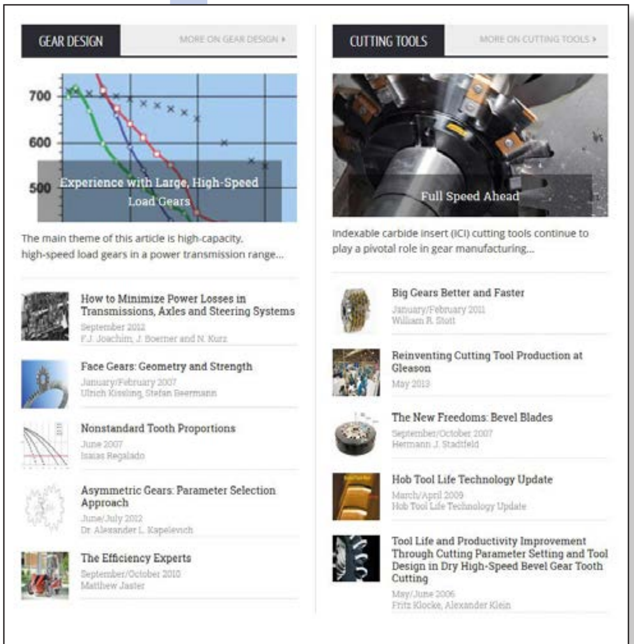
Buyers Guide: Recently Added

The following companies have recently upgraded to premium listings on *geartechnology.com*. Now you can find out more about their products, and you can contact them quickly and easily, through the *Gear Technology* Buyers Guide:

FOERSTER

pure perfection FRENCO

KOEPFER AMERICA, LLC



This Month's Highlighted Topics:

Every month we feature two topics from our extensive archive of 31 years of back issues. On the home page you can find a sampling of these key topics, along with links to the archive. Stop by *geartechnology.com* to see this month's featured topics:

Gear Design

Cutting Tools

But Wait, There's More!

You don't have to rely solely on the home page topics to find great articles from our archive. If you don't see what you're looking for on the home page, try our handy search feature. All the articles in our archive are tagged by keyword. So if you want to find articles on inspection, grinding, lubrication and so on, just type what you're looking for in the search box! Want some suggestions? Try our subject index: *www.geartchnology.com/subjects/*

Stay Connected

Follow us on Twitter
twitter.com/#!/Gear_Technology

Connect with us on LinkedIn
www.linkedin.com/groups/Gear-Technology-Magazine-3893880

Subscribe Online
www.geartechnology.com/subscribe.htm

gear TECHNOLOGY

RANDALL PUBLICATIONS LLC
1840 JARVIS AVENUE
ELK GROVE VILLAGE, IL 60007
(847) 437-6604
FAX: (847) 437-6618

EDITORIAL

Publisher & Editor-in-Chief
Michael Goldstein
publisher@geartechnology.com

Associate Publisher & Managing Editor
Randy Stott
wrs@geartechnology.com

Senior Editor
Jack McGuinn
jmcguinn@geartechnology.com

Assistant Editor
Erik Schmidt
erik@geartechnology.com

News Editor
Alex Cannella

Editorial Consultant
Paul R. Goldstein

Technical Editors
William (Bill) Bradley, Robert Errichello, Octave Labath, P.E., John Lange, Joseph Mihelick, Charles D. Schultz, P.E., Robert E. Smith, Mike Tennutti, Frank Uherek

DESIGN

Art Director
David Ropinski
dropinski@geartechnology.com

ADVERTISING

Associate Publisher & Advertising Sales Manager
Dave Friedman
dave@geartechnology.com

Materials Coordinator
Dorothy Fiandaca
dee@randallpublications.com

China Sales Agent
Eric Wu, Eastco Industry Co., Ltd.
Tel: (86)(21) 52305107
Fax: (86)(21) 52305106
Cell: (86) 13817160576
eric.wu@eastcotec.com

ON-LINE

Digital Content Manager
Kirk Sturgulewski
kirk@geartechnology.com

CIRCULATION

Circulation Manager
Carol Tratar
subscribe@geartechnology.com

Circulation Coordinator
Barbara Novak
bnovak@geartechnology.com

RANDALL STAFF

President
Michael Goldstein

Accounting
Luann Harrold

BPA WORLDWIDE BUSINESS



One More Reason

Publisher & Editor-in-Chief
Michael Goldstein

If you haven't already done so, you should make plans to attend Gear Expo in October. It's a unique and important show, and you should take advantage of it.

Over the years, I've often described Gear Expo as the greatest concentration of gear knowledge under one roof at one time. The exhibitors include all the leading companies of the gear industry—both the equipment suppliers and the gear manufacturers—most of whom bring their top technical people and make them available to answer your questions and solve your gear-related problems. So whether you make gears or buy them, a lot of knowledge is available to you.

The standard reasons for attending are still valid, but I'm excited to add just one more.

You see, this year, *Gear Technology* is doing something a little different at Gear Expo—something new and exciting that we believe will enhance your experience by making it easier to tap into all that expertise.

For the first time ever, we're offering a live, in-person version of our popular "Ask the Expert" column. We've already lined up a number of experts who will be available in our booth (#2030) to answer your questions. These are some of the most knowledgeable people in the industry, the type who routinely write for our magazine, who've spent decades solving gear-related problems, and who are actively involved with developing gear manufacturing technology. These are the gurus from the machine tool suppliers, leading universities and independent consultancies who are the most knowledgeable about gear manufacturing in the world.

I can't tell you who they are just yet, because we're still finalizing our roster and our schedule, but you'll recognize and be impressed by their names. Suffice it to say that if Gear Expo is the greatest concentration of gear knowledge under one roof at one time, ***Ask the Expert Live*** will be the greatest concentration at Gear Expo in one booth.

These experts have graciously agreed to give us some of their valuable time. Normally, these people are extremely busy at a show like Gear Expo. When they're working at their own booths, they are often tied up with customers, so unless you have an appointment, you might not get the chance to talk to them directly. We're making them available to you, no appointment necessary.

We'll be scheduling our experts by session, with a moderator, and each session will emphasize a broad topic and last about an hour. That way, if you have a question about gear grinding, for example, you can make sure to stop by the booth when we have grinding experts on the schedule. The topics are just guidelines, however. They shouldn't dissuade you from stopping by any time, because most of our experts have very broad experience on a wide variety of topics.

Also, Ask the Expert Live is for everybody. We welcome questions from the very basic to the very technical. So, if you're brand new to the gear industry and just want to know what an involute is, please stop by. Likewise, if you've been making gears for 30 years and you're looking to solve a very specific problem, please stop by. No matter who you are, and no matter your level of experience, our experts can help you.

Even if you don't have any specific questions, we hope you'll stop by, because you can learn a lot just by listening in.

In fact, learning the answers to other people's questions is what's made our printed Ask the Expert column so popular over the last several years. The column has allowed us to expand our mission of being "The Gear Industry's Information Source" in a very practical, hands-on way. We're confident that the live version at Gear Expo will take that fundamental premise one step further. The in-person format not only allows for instant gratification—answers to your questions right when you ask them—but it also will afford you the opportunity to take part in a back-and-forth dialogue between the experts and the audience.

So come to Gear Expo, and bring your questions.

Junker

BUILDS NEW PLATFORM FOR CYLINDRICAL AND NON-CYLINDRICAL GRINDING OF WORKPIECES

Junker recently built a new platform for cylindrical and non-cylindrical grinding of workpieces. The platform has a swing diameter of 470 mm and a part length capacity of up to 4,800 mm, and it will first be used in the Jucrank series for grinding large crankshafts.

As these weigh up to 1,000 kg, it is a challenge to set the parts up for the process. To adjust the table assemblies, Junker has developed a slide with an integrated length measuring system. As a result, the setup technician first brings the work heads into position, then the steadies.

To enable the processing of such unstable workpieces, Junker had to develop its own steady. The new patent-pending steadies are CNC-controlled and have only one axis each. This increases their stability and stiffness. Each of up to a maximum of 11 steadies can be controlled individually and applied to a section at any time — even during the process. This key feature allows for higher sequence flexibility of the grinding process. To make this possible, Junker applied its control concept to a larger, high-performance control system.

Large crankshafts are mainly produced in small batches, and in some cases as single pieces. Junker has added an integrated measuring system to overcome these challenges.

First, the two grinding wheels, each mounted on a wheelhead with its own X and Z-axis, pre-grind the main and pin bearings. The diameters are measured during the process. The grinding machine also measures the entire workpiece after pre-grinding.

Based on the measuring data, the Jucrank 8 finishes the grinding process while using the WK axis developed further by Junker: During grinding it swivels the grinding spindle, compensating for tapers in the process. With this technology the grinding machine can provide each main and pin bearing with its own profile shape, i.e. if necessary with specific crowning. With this functionality, the machine then also grinds the two shaft ends if required. These often feature a taper and not a flange or post end.

As a result, the forged crankshaft is completely ground and ready for installation after only one set-up. Another possibility of applying the new Jucrank technology is for the re-grinding of used crankshafts.

For more information:
Junker Group
Phone: +49 (0)7838 84-0
www.junker-group.com

New Tongati MT Series Lathes

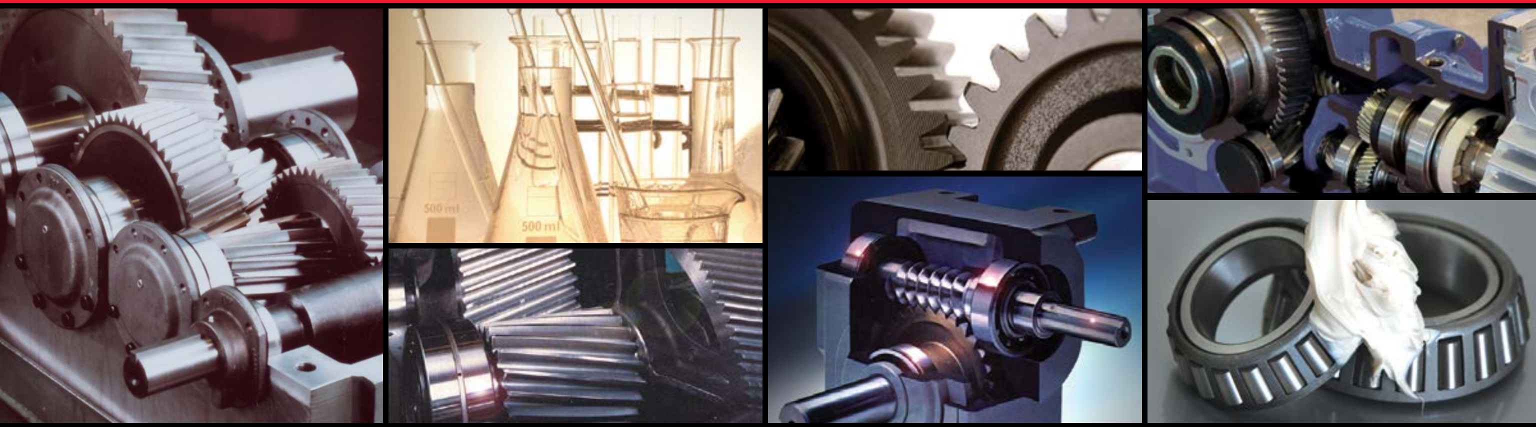
HAVE TWIN SPINDLES AND INDIVIDUAL MACHINING ZONES

Tongtai's newest MT Series Lathes were developed for improving cycle times and turning processes for the automotive industry, where small parts are typically made within 60 seconds. Tongtai developed the MT series for precision turning, high-production volume, automatic production, and insertion into mass-production lines. Tongtai machines are available from Absolute Machine Tools, Inc., headquartered in Lorain, OH.

The MT Series has twin spindles and two individual machining areas. The turrets and spindles are designed parallel to each other. This design makes parts on one machine that usually need two processes to be finished. The addition of a gantry-type robot saves floor space and improves labor flexibility. Depending on cycle time, a single robot arm/single stocker or twin robot arms/twin stockers for high production chucking work are both available on these lathes. The Tongtai MT lathes come equipped with either 6" or 8" chucks on spindles that rotate at 6,000 or 4,500 RPMs respectively. The control system offers a robot teaching function so that the operator can adjust positioning. The function coordinates the robot arm, on-screen positioning diagram, input coordinates, number and names of positions, three-axis settings and single-axis settings.

The work area on the main foundation of the lathes has two individual working areas with separate bed structures. This design decreases the transferring of harmonic vibration, contributing to improved machining accuracy and high-quality surface finishes. The compact design allows for a short cutting flow from start to finish, which enhances machining rigidity and heavy cutting ability. Maximum swing diameter is 210 mm with a machining diameter of 210 mm or 120 mm (with robotic arm), and machining length of 145 mm or 100 mm (with robotic arm).

The MT series lathes are equipped with workpiece positioning protection to ensure sealing between the workpiece surface and chuck. Pneumatic pressure leaks can be detected, and if this occurs, the robot arm will reload the workpiece. The gantry type robotic arm is able to process three axes of movement and is



THE BRAND OEMs CAN TRUST

WHEN SPECIFYING LUBRICANTS FOR THEIR MACHINERY

More and more original equipment manufacturers are specifying Lubriplate Lubricants for their machinery. Why? With 145 Years of Lubrication Experience, they know Lubriplate has the Quality Products and Services to meet their needs, the needs of their equipment and the needs of their customers.

A DEDICATED OEM DEPARTMENT

- OEM Technical Support - Dedicated to helping you select the best lubricant for a given application.
- Helps ensure equipment performance & service life.

A WIDE RANGE OF QUALITY PRODUCTS

- A Full Line of Advanced Technology Synthetic Lubricants.
- Traditional Petroleum-Based Oils and Greases.
- NSF H-1 Registered, NSF/ISO 21469 Certified Lubricants.

A WORLDWIDE DISTRIBUTION NETWORK

- A Strong Worldwide Distribution Network.
- Ensures availablity of correct service lubricants regardless of equipment location.

145 YEARS OF EXPERIENCE IN EVERY PRODUCT

"It's the Film" LUBRIPLATE — PREVENTS WEAR and CORROSION — SUPER-LUBRICATION

Lubriplate® Lubricants

Newark, NJ 07105 / Toledo, OH 43605 / www.lubriplate.com
Contact Ellen GIrard, Lubriplate's OEM Marketing Analyst
Tel: 908-489-7355 / E-mail: egirard@lubriplate.com

Backed By: Lubriplate® ESP

Extra Services Package

Plant Surveys / Tech Support / Training
Color Coded Lube Charts & Machine Tags
Lubrication Software / Follow-Up Oil Analysis

driven by a servo motor. The programmable arm allows the operator to adjust positioning points and moving routes. Rapid traverse speeds for the robotic arm are 160 m/minute in X, 120 m/minute in Y, and 35 m/minute in Z. The rotary axis moves at 180° in one second.

The pallet stacker comes in three types: three poles and a center, three poles, and a central pole type. The number of pallets can be 10, 14, or 16 with allowable part diameters from 30 mm through 150 mm.

The MT series comes standard with coolant through the spindle, A2-5 or A2-6 spindle nose, 0.001" indexing increments, 4,500 RPMs or 6,000 RPMs optional. With cutting feed rates of 0.001–5,000 mm/minute this horizontal turning center is designed for high-production, fast-paced manufacturing environments.

For more information:
Absolute Machine Tools, Inc.
Phone: (800) 852-7825
www.absolutemachine.com

Schunk EGN Safety Gripping System

CONTINUOUSLY POWERED EVEN IN A SAFETY STOP

The EGN Safety is the first certified safety gripping system from Schunk for use with human and robot interaction. It is certified in accordance with DIN EN ISO 13849 specifications.

If the production process is interrupted by an emergency shut-off, the Schunk EGN goes into either a safety speed mode or a safe stop mode, depending on the activated protection zone.

Unlike other options on the market, the Schunk safety gripping system is continuously powered even if in a safety stop, and will safely hold the gripped part. As soon as the protection zone is released, the gripper switches back to the regular operating mode without having to be restarted.

The EGN gripper is modularly designed and covers a wide range of applications. The gripping force, speed and finger position can be flexibly controlled during the handling process. All sizes have multi-toothed guidance and are controlled by the Schunk ECM controller equipped with standard connector technology with either Profibus or CAN bus.

To use the certified SLS, SOS, and STO functions, the Schunk safety module is added to the gripper and the controller. Because of its modular design, existing EGN applications can be retrofitted with the safety functionality at any time.

For more information:
Schunk Intec, Inc.
Phone: (905) 712-2200
www.schunk.com

Emuge EF Series Drills

INCORPORATE SPECIAL GEOMETRY FOR FASTER PENETRATION AND LONGER TOOL LIFE

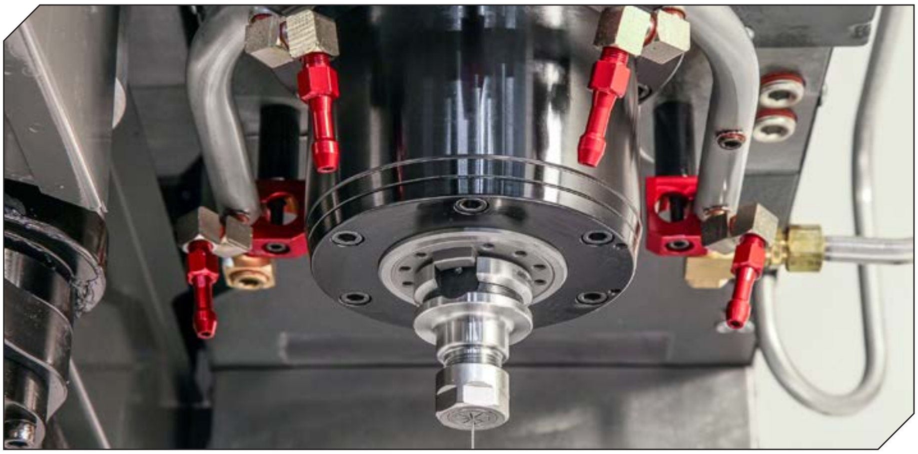
Emuge Corp. recently announced the North American debut of its line of high-performance solid carbide drills. The new Emuge EF series drills incorporate special geometry, proprietary carbide grades and a PVD coating design. The result is three to five times faster penetration rate than conventional carbide and cobalt drills, in addition to high-quality threads and longer tool life.

"We are excited to roll out this major introduction to the U.S. and Canadian marketplace," said Bob Hellinger, president of Emuge Corp. "It makes perfect sense for Emuge to have a drill product line to complement our leading line of taps and thread mills, and it provides our customers the best holemaking solutions for their tapping applications."

In addition to the drill line introduction, beginning in the fourth quarter of 2015, Emuge will be offering complete grinding/reconditioning services for all drill products at their West Boylston, MA facility.

The EF solid carbide drills, engineered and made by Emuge in Germany, feature a double-margin flute design for added stability and rounder/straighter holes. A unique flute construction enables improved chip evacuation, and a self-centering design allows drilling in one shot, eliminating peck cycles and pre-spot operations. Emuge drills are made of a sub-micron grain carbide grade for increased abrasion resistance and durability, and a multi-layer PVD coating resists chipping/cracking for longer tool life.

For more information:
Emuge Corp.
Phone: (800) 323-3013
www.emuge.com



Haas DT-1 Drill/Tap Center

ALLOWS RIGID TAPPING TO 5,000 RPM

The DT-1 Drill/Tap center from Haas Automation, Inc., is now available with a 20,000-rpm inline direct-drive spindle, giving customers the ability to run higher feed rates for small tools and high-speed machining operations.

The optional 20K spindle is for applications that require high spindle speeds, and powerful enough to mill hard-to-machine materials. It allows rigid tapping to 5,000 rpm, with up to four times retract speed to reduce cycle times. The spindle is powered by a 15 hp vector drive system that yields 16 ft-lb of cutting torque, and the motor is coupled directly to the spindle to reduce heat, increase power transmission, and provide improved surface finishes.

The DT-1 is a lean-style machining center with a compact footprint that allows multiple machines to be placed side-by-side, allowing for the most efficient use of shop floor space. It features a 20" × 16" × 15.5" work cube and 26" × 15" T-slot table, while maintaining a very small footprint. The machine provides cutting feed rates to 1,200 ipm for high-speed milling, and the 20+1 side-mount tool changer swaps tools quickly to reduce non-cutting time. High-speed 2,400 ipm rapids combine with high acceleration rates to shorten cycle times and increase throughput.

For efficient chip removal, the DT-1 features steeply sloped internal sheet metal. Optional twin chip augers transport chips to exit at the rear of the machine, allowing multiple machines to be placed close together. A 45-gallon flood coolant system is standard, with options for a programmable coolant nozzle and high-pressure through-spindle coolant systems.

For more information:
Haas Automation, Inc.
Phone: (805) 278-1800
www.haascnc.com

Sumitomo Turning Grades for Machining Stainless Steels

FEATURE ABSOTECH GRADE COATING

Sumitomo Electric Carbide Inc. recently released new AC6030M and AC6040M turning grades for machining stainless steels.

The AC6030M and AC6040M grades feature Sumitomo's new Absotech grade coating. Absotech coating improves wear resistance, adhesion resistance and fracture resistance, which extends tool life. The AC6030M possesses a new TiB_2

surface layer that creates adhesion and chipping resistance. The AC6040M has an optimized Ti and Al composition resulting in improved flank and notch wear resistance.

The AC6030M and AC6040M grades have an expanded chipbreaker lineup highlighted by the newly developed EEM breaker, which provides improved chip control when roughing.

For more information:
Sumitomo Electric Carbide Inc.
Phone: (800) 950-5202
www.sumicarbide.com

New Transmissions Make the Gas GREENER

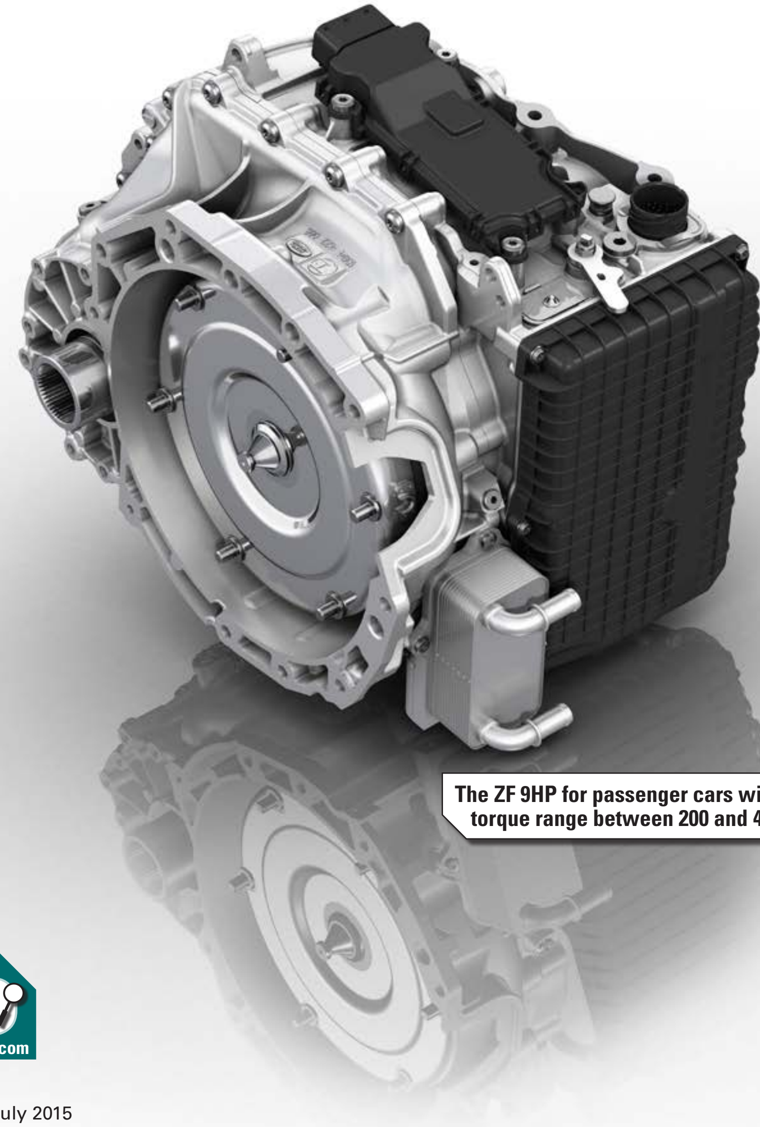
Jack Mc Guinn, Senior Editor

***"Highway vehicles release about 1.7 billion tons of greenhouse gases (GHGs) into the atmosphere each year—mostly in the form of carbon dioxide (CO_2)—contributing to global climate change. The CO_2 emissions of a car are directly proportional to the quantity of fuel consumed by an engine. In 2013, U.S. greenhouse gas emissions from transportation were second only to the electricity sector—an increase of about 16% since 1990."* (EPA.GOV).**

Want to know what is *truly* hot in the automobile industry these days? Not just here in North America, but like—*everywhere*? Want to know the industry's singular issue that can literally affect national elections, the world economy and whatever passes these days for stability in the Middle East? An issue that in fact affects the very quality of life on this Planet and we "driving enthusiasts" who motor around it? It is keeping engineers' hair on fire—with clattering slide rules juggling gear ratios high and low—in order to produce more power and torque for an ever-bigger-bang in an ever-shrinking gearbox.

Boil it to the bone and the Number One issue is revealed: How to extract more clean driving miles from a gallon of fossil fuel-based gasoline? (But, of course, without any sacrifice to speed and "drivability.")

It's about e-n-e-r-g-y. About fuel—and how your Ford or Toyota burns it.



The ZF 9HP for passenger cars with front-transverse drive covers a torque range between 200 and 480 Nm (Photo courtesy ZF).

More precisely, it's about the regular unleaded that powers our cars. And more importantly, it is about the CO_2 gas that is created as a result and then released into the ether via your car's exhaust system. And the hard truth for automakers and consumers alike is that one of the most targeted generators of greenhouse gas is — Ford Help Me — the internal combustion engine. For years it was thought the chemists could solve the problem via re-engineered fuels; e.g., ethanol — a pricey alternative if you happen to live in some parts of the Midwest.

It has taken decades for an industry, its customers, and the governments that regulate them to even begin facing the question: How to extract more CO_2-friendly driven miles from a gallon of gasoline?

We have less than a decade to find the answer. Consider: U.S. automakers face a federal mandate for an industry-wide fuel economy average of 54.5 mpg by 2025. (European carmakers have been dealing with CO_2 mandates for some time now.) To reach that 54-plus mpg target, engineers of all stripes are looking from bumper to bumper for potential efficiency gains and energy savings. (*By design, the scope of this article does not include hybrid or all-electric engine technology. That said — you will see later that it is impossible to discuss today's latest-technology combustion engine-based transmissions without some mention of electric motors.*)

But above all, engineers are zeroing in on transmissions. Why is that?

Before even mentioning gears and transmissions, Dr. Hermann Stadtfeld, Gleason Corporation vice president, bevel gear technology and R&D, explains the answer in part derives from the constant market-driven push for smaller engines to fit today's smaller cars. Smaller engines, smaller gearboxes, smaller gears.

"The development of down-sized engines has led to small-displacement engines — in most cases using turbo chargers, not only for diesel engines," he says. "The newer transmissions consider the changed power and torque distribution of these engines and support the fuel-efficient behavior by offering more transmission ratios (eight-speed and higher). This helps to lower the engine RPM (and) fuel consumption. In case of higher power output requirements, the new transmissions enable the engines — due to more gears — to always operate at or near their torque-optimal point."

Buttressing Stadtfeld's explanation, consider this from Sebastian Strunk, a Gleason applications engineer at their Ludwigsburg, Germany facility.

"In addition to lowering the overall RPM level, several techniques are used to lower the internal transmission losses. For example, by reducing friction and oil churning due to active, targeted lubrication, and by earlier engagement of the lock-up clutch in the case of automatic transmissions with hydraulic torque converters. Automatic transmissions are very complex. In most common automatic transmissions the required ratios are realized via the combination of planetary gearset stages. In the design phase, millions of combinations are possible and are pre-calculated on fast computers, still taking several weeks to come to reasonable results. From these results a preferred combination has to be chosen by a team of human experts, which is a very big responsibility. Because not only the ratios have to be taken into account, but also the future behavior and interaction of the lamellae-clutches and lamellae-brakes in the transmission have to be predicted to anticipate the shifting quality, efficiency and noise emissions.

"Also the manufacturability, strength and wear behavior are crucial, as well as cost-effectiveness. Taking into account that such automatic transmissions will be part of the manufacturer's vehicle lineup for a decade and longer, already during development it must be assured that there is potential for a Generation II and III of the newly developed design in order to adjust to future requirements, and to be able to present frequent product improvements to the customer."

Joining the conversation is Noel R. Mack, director and business unit leader for Riccardo Driveline & Transmission Systems. "(Reducing) CO_2 emissions goes hand-in-hand with using less fuel. Quite simply, the newer transmissions assist in improving the efficiency of the propulsion system. This is achieved by a) implementing more gear ratios, allowing the engine to spend more time operating in the zone of ideal or optimized efficiency; b) with a greater ratio spread, we can operate with smaller engines and still achieve the acceleration levels and top speeds that we are accustomed to; and c) the transmission itself operates more efficiently, wasting less energy and transferring more torque to the wheels."

Dr. Jürgen Greiner is the development head of passenger car transmissions for ZF. And while he agrees, "in principle," that "transmissions with higher transmission-ratio spreads and more gears with smaller gear steps… reduce fuel consumption," it is also his position that , "Generally, the number of gears is not a primary feature of a transmission for ZF. When it comes to developing transmissions, the relevant factor is not the number of gears but the spread of gear ratios and drivability that plays an important role; the number of gears is simply the result. Only with excellent shift quality is a high number of gears perceived as comfortable — with no tractive force interruption."

But of course the need for speed will always exist for drivers of all stripes. As Stadtfeld points out, "Although efficiency is very important today, the higher number of gears also manages an immediate adjustment of the ratio to a point where the engine is capable of the highest possible power delivery in order to

> "The tendency is that the market share of conventional automatic transmissions is going to decrease in certain segments, like medium and smaller vehicle classes, and the diversity increases. Other forms will appear — like DCTs, or more exotic transmissions like IVTs — and will deliver the torque and efficiency requirements while having lower costs and acceptable comfort."
>
> **— Hermann J. Stadtfeld**


Premiere in the BMW 5 Series: The 520d is the first production model to come off the production line with the new ZF 8HP transmission generation (Photo courtesy ZF).

realize fast acceleration."

By the way — just *what kind* of transmissions are we (or should be) talking about? For our general purposes, two — 1) the best-known-in-the-U.S. automatic transmission (AT); and 2) the continuously variable transmission (CVT), well known and used in Japan. Though others exist, the talk is mainly about ATs vs. CVTs, along with some "hybrid-type" transmission talk as well. ZF's Greiner begins some transmission anatomy for us:

"An *automatic transmission* (AT) comprises several (planetary) gearsets, in different quantities and combinations. For example, there are four planetary gearsets in the new ZF 9-speed automatic transmission. The various ratios — i.e., gears — of the planetary gearset are generated by fixing one gear and by connecting and disconnecting various shafts. This is done using so-called shift elements (friction shift elements such as multi-disk clutches or multi-disk brakes), and constant-mesh elements. In the new 9-speed automatic transmission, there are a total of 6 shift elements, two sets of multi-disk clutches and brakes, as well as two dog clutches — a first in passenger car automatic transmissions.

"Most types of transmissions are stepped transmissions. This means they have a specific number of gear pairs that offer a fixed number of gear ratios, depending on the design. The power range of the combustion engine cannot be fully harnessed, however, with this limited number of transmission stages. In contrast, a *continuously variable transmission* (CVT) is, as the name suggests, a transmission that can change seamlessly through an infinite number of effective gear ratios. This is particularly beneficial for the characteristics of a combustion engine. Along the entire engine performance graph, operation can be adjusted with a continuously variable transmission ratio to achieve optimum fuel consumption or fastest acceleration on request."

But, as Greiner then adds, "Unlimited speeds, like in the case of the CVT, reduce efficiency since you need energy for the transmission of power. The system-related lower efficiency; limitations when transmitting torques; the maximum transmission ratio range to be achieved; and the higher weight reveal the physical limits of (CVT) design. The CVT is the most demanding and most expensive (to manufacture)."

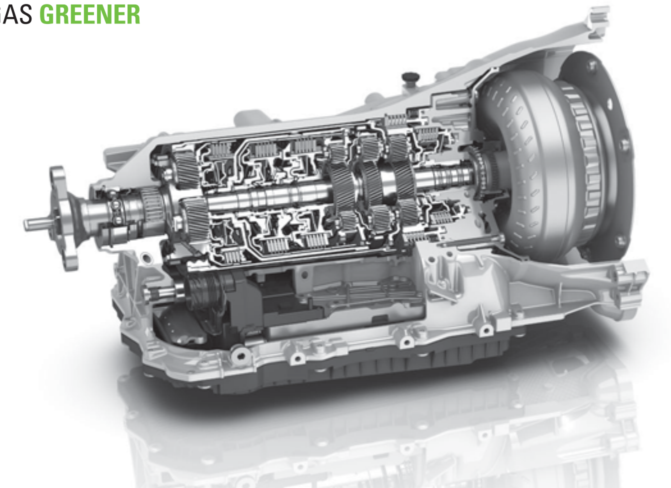
"A 'conventional' gear transmission is really region-specific," Mack points out. "In, say, North America, it (transmission) would be a torque converter, planetary automatic. In many other parts of the world (e.g., most of Europe) it would likely be a manual transmission.

"There are also many variants of 'hybrid transmissions' under consideration," says Mack. "Most current hybrids in passenger cars today are electric hybrids. With many possible variants — from an inline electric motor on the transmission input to the well-known input split transmission as used in the Ford Escape, which has two electric motors with an internal combustion engine, with one motor operating across a planetary gearset — creating an (IVT)."

But, "Maybe the name 'conventional' automatic transmission becomes outdated," suggests Stadtfeld. "In the U.S., from the 1950s until the early 2000s, these kinds of transmissions with three and later mainly four-speeds were most popular. Then even more speeds were introduced in conventional ATs. In addition, CVTs and double-clutch transmissions (DCTs) had been introduced. For example, the market share of CVTs in 2014 is around 19% of all automatic transmissions (*Source: EPA Technology Trends 2014*).

"The gears transmit the power via positive-form engagement, whereas the clutches transmit the power via friction contact. Nevertheless, due to the intelligent use of other positive-engagement units like dog couplings, lamellae brakes can be replaced, helping to reduce friction losses. An additional standard element to improve the efficiency is the lock-up clutch to bypass the torque converter in situations when it is not required and would only lower the efficiency due to slippage.

"CVTs require, next to their variable transmission unit, either a lamellae clutch or a torque converter as a start-up-element. The power is then transmitted to one of the variable cone-shaped pulleys. This pulley transmits the power via a metal push belt to the other pulley that transmits the power to the output shaft. The required ratio is generated by an adverse variation of the two pulley diameters. The power is transmitted via friction contact. For this reason the pressing forces between the pulleys and the belt need to be high enough to avoid slippage at the required torque. This leads in general to a high power consumption of the internal hydraulic system and the friction between the belt joints, lowering the efficiency of the system."



ZF's enhanced second-generation, eight-speed automatic transmission (8HP) is designed to provide effective support for automotive manufacturers in meeting the ever-stricter legal CO_2 standards — and in a cost-effective manner in combination with conventional or hybridized drives (Photo courtesy ZF).

Well what then of the ballyhooed (gas-and-electric-powered) hybrids?

Says Stadtfeld, "Today there is a large diversity, with increasing numbers of automatic transmissions based on different concepts. The tendency is that the market share of conventional automatic transmissions is going to decrease in certain segments, like medium and smaller vehicle classes, and the diversity increases. Other forms will appear — like DCTs, or more exotic transmissions like IVTs — and will deliver the torque and efficiency requirements while having lower costs and acceptable comfort."

Is it harder to design these new transmissions for Cadillacs than Fiats, for example?

As Mack explains it, "The role of the transmission is to most effectively integrate the requirements of the vehicle to the characteristics of the engine. It is really all about transferring the power from the internal combustion engine to the wheels. The transmission ratio essentially defines the speed at which the engine operates, and we need to get the engine into its most efficient zone of operation. If the engine torque is reduced and the vehicle requirements remain the same, the transmission will need to be "longer lever" — i.e. more ratio coverage. We also need smooth transitions from one discrete ratio to the next and sufficient refinement. The level of challenge in designing for a powerful-luxury vehicle or a smaller segment is very much the same."

Greiner believes that, "In principle, it makes sense to use transmissions with higher transmission-ratio spreads and more gears with smaller gear steps to reduce fuel consumption."

But with increasing ratios come, it would seem, greater complexity.

Greiner responds that, "If we could fit even more gear steps in the space currently available, it is initially technically possible for the number of gears to reach double-digit figures. Gradually, we are reaching the boundaries of what is sensible. For example, the transmission's spread of gear ratios is now so wide that engines are at the limit of being drivable. Electric motors and combustion engines need to be combined to work together in harmony. Integrated electronics as well as an intelligent drive management embedded in driver assistance functions are necessary to achieve this."

While it is hoped the above back-and-forth has brought some clarity to the discussion of newest-technology, CO_2-friendly automobile transmissions, the question arises: If there were no air pollution, would we even be having this discussion, or is this transmissions evolution simply a marketplace inevitability?

What a stupid question to ask engineers.

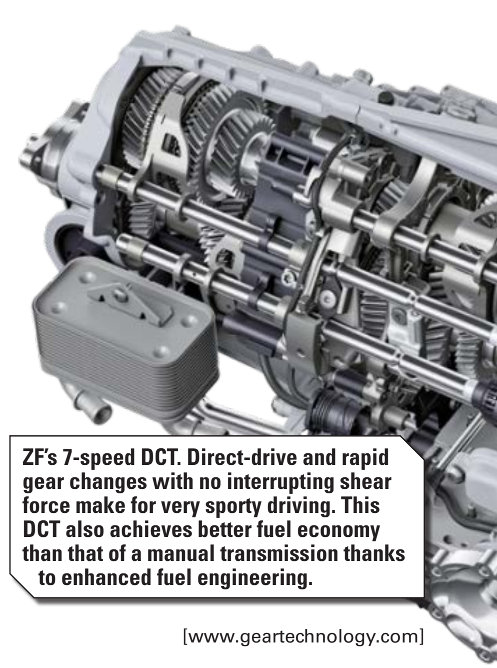
"In our nature as humans, essentially we always want to improve upon the status-quo, and strive for a competitive advantage," says Mack. "The competitive marketplace drives the need for more features, more performance, driving 'fun' and, of course, higher value. The government regulations are merely accelerating the rate of development with certain criteria in mind. The drive for lower CO_2 emissions or higher MPG can also be viewed as wasting less energy. Over the prior century we have constantly evolved the science of transmission engineering with a common goal — to increase efficiency, transfer more of the torque, and waste less."

"Pollution is only one aspect why we have this transmission evolution," says Stadtfeld. "Besides the cost of gasoline for the consumer, also the production of gasoline is connected to certain environmental implications. Vehicles with automatic transmissions newer than model year (MY) 2010 already consume, on average, the same or less amount of gasoline than their manual counterparts, while offering other advantages. For example, current and future automated driving systems would be impossible with manual transmissions. In addition, more gears deliver a smoother driving experience with better capability to use the engine's power. But more gears require the transmission to be automatic because only a very few people would accept shifting through 9 or more gears in a car while constantly watching the optimal shift timing to achieve smooth and efficient driving." Here's one last big question: Which of the new technologies has the best chance of replacing "conventional" transmissions? Is there a logical endgame?

It depends. Good luck defining "conventional," for instance, when describing transmissions.

"This is not an easy question to answer, and I wish I had a crystal ball," Mack admits. "The definition of a 'conventional transmission' is not standing still, and is regionally defined. There are two primary factors to consider. First, from a propulsion system viewpoint, we need to attain an optimum system efficiency, which includes using the transmission to "move" the operating point of the engine into its sweet-spot; and secondly, the transmission itself, regardless of architecture chosen, will evolve to constantly improve the state of art. Gears have been and will likely remain the most efficient method of transferring torque. Amongst the 'new transmissions' we would include DCTs and many hybrid electric variants. With recent developments in some CVT technologies, there may be a challenge on the horizon. There are multiple technologies with promise and the potential means to effectively achieve the goals. We continue to see an evolving mix of technologies — with no single winner. Consumer demand will also play a role in the proliferation of technologies on our roads. Pros and cons are unfortunately not based merely on technical merit, but also on many other market drivers; e.g. — legislation, consumer acceptance, cost, manufacturing investment, etc."

"We see new focal points" Greiner allows — if not so much for gears — "in the field of power electronics, the intelligent driving strategy, the development of ever improved electric motors, or the integration of the electric drive in the installation space which before was



ZF's 7-speed DCT. Direct-drive and rapid gear changes with no interrupting shear force make for very sporty driving. This DCT also achieves better fuel economy than that of a manual transmission thanks to enhanced fuel engineering.

reserved exclusively for the transmission. The further you go in the direction of hybrids or electric vehicles in the future, the less gears will be needed, dependent on to what extent combustion engines make up the drive. Electric motors and combustion engines need to be combined to work together in harmony."

We had to ask for predictions of what kind of transmission will be used in the first *truly popular* car with an *all-electric* engine? Will torque control still rule?

Gleason's Strunk: "Due to the significant difference in power output characteristics between ICEs and vehicles with electric motors that deliver instant torque over a wide RPM-band, theoretically no transmission is required to fulfill standard requirements regarding drivability and efficiency.

"Of course the development of transmissions for ICE engines showed soon that "standard" is not enough. From an objective point also electric motors have a peak point of efficiency which can only be utilized over a wide range of driving conditions with a multiple speed transmission. Based on the limited and expensive battery capacity it seems even more important to save precious energy onboard the electric vehicle by optimizing the drivetrain.

"Recent developments show that a two to four speed transmission in an electric vehicle will contribute to achieve higher efficiency levels, higher top speeds and even higher performance levels with better packaging. But apparently there is also a subjective component of being the first to offer a multi speed transmission in electric vehicles similar to the raise in 8, 9 and 10 speed automatics in fuel burning vehicles."

Says Riccardo's Mack. "With the nature of the torque-generating capability (the torque curve) of an electric motor, a single-speed transmission should be adequate in a small passenger car-type application. When applied to a larger or more luxurious vehicle, a single reduction ratio will either limit the lower-end torque or the higher-end speed capability. To overcome this, a two-speed transmission will be utilized to provide more low-end torque at the wheels while being able to operate at a higher top speed. A two-speed transmission will also allow the electric motor to spend more time operating at or near the motor's area of optimum efficiency. We would predict that we will still use multi-ratio transmissions with many of these electric vehicles."

Manufacturing Jobs Are There — Workers to Fill Them ARE NOT

And DMG MORI is doing something about that

Jack Mc Guinn, Senior Editor

An all too common — and disturbing — question these days: Are you having trouble finding skilled workers? Taking that a step further begs the next question — Are you having trouble finding *customers* with skilled workers to use your products?

This job skills dearth is quite the slippery slope, i.e. — while machinery/tooling manufacturers are beating the bushes for qualified machinists to *build their* machines for their gear-making customers, gear manufacturers are likewise constantly in search of qualified personnel to *operate the machinery*.

It is a two-part dilemma that continues to stump and bedevil manufacturers, with no overnight solution in the offing.

DMG MORI has been busily at work trying to make a difference in that regard. Realizing that there were no quick fixes to this problem, DMG MORI sucked it up and decided to do it themselves. Their work in that regard thus far was one of the presentations at the recent DMG MORI Innovation Days Open House event, held at their Hoffman Estates location just outside of Chicago.

But before talking about the good stuff, here comes the bad-tasting medicine to give you some perspective on just how dire this skills shortage dynamic has become. Now, open wide…

- In a survey on recent hiring results, 60% of manufacturing organizations are having difficulty recruiting specific jobs. (*Source: SHRM — Society for Human Resource Management*)
- It is clear that many jobs are not being filled due to a mismatch between job seekers and open positions. (*SHRM*)
- According to a BLS (Bureau Labor Statistics) labor market report from August 2014, the government stated that 9.6 million Americans were unemployed while employers posted 4.8 million jobs. The unemployment rate for July was 6.2; for August was 6.1; and Sept 5.9 Note that despite 4.8 million jobs posted, the unemployment rate remained essentially *unchanged.* (*BLS*)

Many feel that the reason the unemployment needle is not moving much in manufacturing can be attributed to bullet point two above; i.e. — the jobs are there, but qualified candidates to fill them are not. Following is a dismal object lesson illustrative of this statistic:

DMG MORI service apprentices in a training class working through a trouble shooting exercise (photo courtesy DMG MORI).

A CEO of a metalworking company in Northeast Indiana was in need of 100 skilled workers.

- The company received *130* applications
- Accepting those with *some* experience narrowed that list to *40*
- When those applicants were given industry tests, only *four* passed
- Those four were offered jobs; but *only one* accepted

The bottom line? A search for 100 skilled workers netted only one viable addition to its workforce. (SHRM)

No wonder we are finally witnessing a push for STEM (Science/Technology/Engineering/Math) courses in the schools as a belated but critically needed recognition of the fact that our elementary and high schools need to force-feed, if necessary, these disciplines to students — sooner than later. In today's brave new world of manufacturing, the sophistication level of worker skills that this coursework imparts is essential for any jobseeker.

And should you be a STEM-scoffer, check out these grisly facts comparing U.S. student skills with those of 21 other countries:

- Reading and Writing skills ranked *20th*
- Nearly 67% failed to meet the *minimum* standards for working with numbers which placed America last
- 56% failed to meet the *basic* proficiency in Problem Solving/Technically Rich Environments (PS/TRE) category — ranking America *last* among the 22 countries
- The average U.S. scores for these skill levels have *declined* over the last 10 years


Photo by David Ropinski

And yet the fact remains that if anyone seeks a career in manufacturing, they will have to be proficient in the areas targeted by manufacturers. Here's hoping our young people will step up, because it is a STEM future, and here are some impressive numbers backing that up:

- Job growth rate for STEM jobs is *dou-*

ble the rate for *all other* career fields

- In 2013, there were approximately two jobs available to each person entering these fields
- By 2018, the projection is that there will be *230,000 more jobs than people* entering these fields (*Source: Organization for Economic Co-operation and Development*)

Who will fill those 230,000 jobs?

Richard Templeton, chairman, president and CEO of Texas Instruments, would certainly like to know. "In Texas alone, the economy is poised to add nearly 760,000 STEM-related jobs within the next four years. Nationally, demand for scientists and engineers will increase four times faster than for all other positions over the next decade." (*Source: Dallas Morning News, July 2014*)

Societal issues are at play as well. But this one — the shrinking of middle-class America — seems to be one of those "can't see the forest for the trees" scenarios. For those paying attention, like the clichéd slow train wreck, we see the disappearing act right before our eyes, yet do nothing.

Put another way, "The erosion of the middle-class has made it hard to fill open jobs requiring more education and training than a high school diploma but less than a four-year college degree. (*Source: "Bridge the Gap: Rebuilding America's Middle Skills," SHRM Magazine Dec 2014.*)

What to do? One suggestion:

"Policymakers and other stakeholders will need to shift the conversation from one of educational attainment to one that acknowledges the importance of skills. (*SHRM Study: "U.S. Millennials' Skills Don't Match Education"*)

Infinitely easier said than done. Only 20% of those polled believe "the school system in my community encourages students to pursue careers in manufacturing. And, only 35% say they would "encourage their child to pursue a career in manufacturing." (*Deloitte* and the *Manufacturing Institute)*

All of which brings us back to DMG MORI's own problems with finding qualified help. They eventually determined there were only two options available to them in adding necessary skill sets to their workforce in the short term, i.e. — hire from the current workforce and/or "grow your own."

Finding themselves in the same leaky boat as manufacturers all across the country, they chose the latter — and in a meaningful way.

"The decision to move forward with our own service and apps engineering training/development programs occurred after the company conducted a series of nationwide recruitments, but could not identify candidates with the basic skill sets for those positions," says Bill Gaeth, DMG MORI lead, apprentice program & online training customer interface.

And make no mistake — the curriculum regimen and completion requirements for apprenticeship status at the

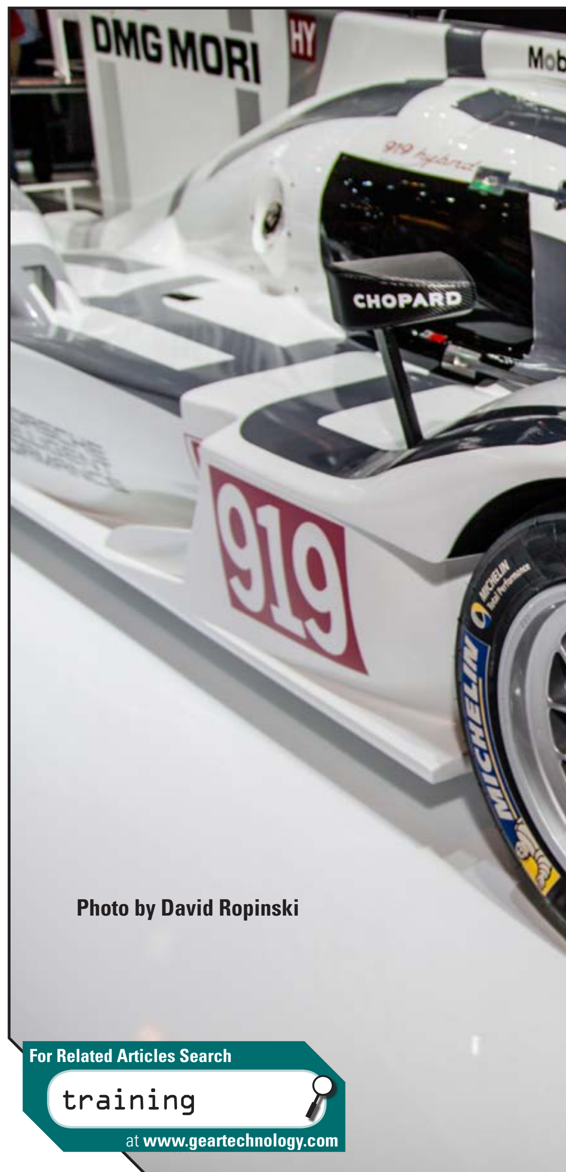


Photo by David Ropinski

For Related Articles Search
training
at www.geartechnology.com

DMG MORI Academy is no collection of ho-hum, jolly-numbers exercises (*dmgmsaondemand.com*). Areas covered include Machine Programming & Operation; Machine Maintenance & Repair; EOD Machine Operation (Online); EOD Machine Programming (Online); Manufacturing Skills; Quality, Measurement & Gauging; Fluid Power & Mechanical; Green Energy and numerous other skills-specific offerings. And after the Academy coursework is done, even more remains.

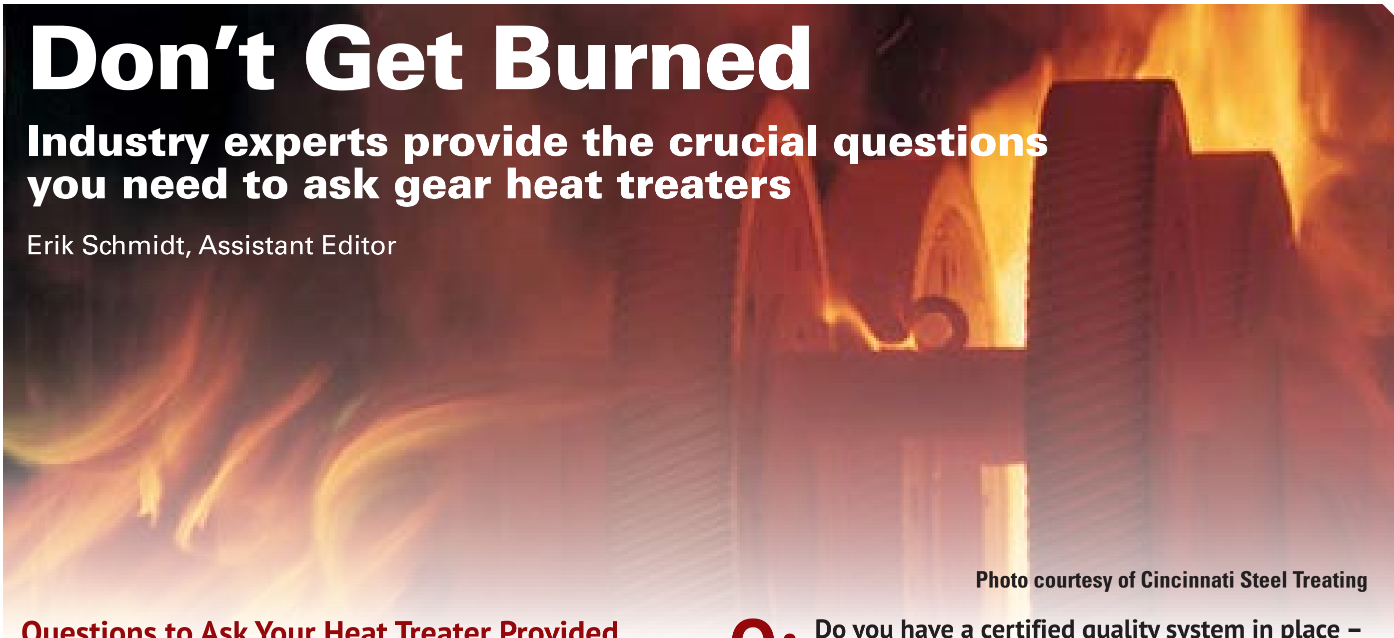
"An apprentice must also complete shadow training with an experienced employee and a regime of factory training. The programs are competency-based, but usually take an apprentice two to three years to reach the level of competency we look for to graduate from the program," Gaeth explained.

Seeing the DMG MORI programs in operation, one wonders why more companies are not doing this. Is it a matter of won't or can't with them?

"It is more 'can't' than 'won't,' stated John Roufis, Manager of Technical Instruction "because, for the most part, our customers don't have the resources to provide the machine-specific training they want their employees to have."

"And so," Roufis continued, "the DMG MORI Academy is set up to provide training that their customers require because we have the equipment, the instructor staff, and the facilities to train their employees."

For more information on DMG MORI Academy, please visit *www.us.dmgmori.com*.



Don't Get Burned

Industry experts provide the crucial questions you need to ask gear heat treaters

Erik Schmidt, Assistant Editor

Photo courtesy of Cincinnati Steel Treating

Questions to Ask Your Heat Treater Provided by Justin Lefevre (Joyworks LLC, Ann Arbor, MI), Kathy Hayrynen (Applied Process, Inc., Livonia, MI) and Vasko Popovski (Applied Process, Inc.)

Q: What heat treat processes do you specialize in?

Most heat treaters specialize in processes that their equipment is optimized for. They are great at some processes, but do not have the equipment (or knowledge) for other types of heat treatment. It is a risk to assume that the same shop that anneals parts is capable of producing a case-hardened gear.

Q: What materials do you routinely heat treat?

Heat treaters with experience in steel processing may not have familiarity with cast irons. There is a big difference between what someone "can" run and what they "actually" run.

Q: Who do routinely do work for?

Due to nondisclosure agreements, a heat treater may or may not be able to divulge specifics, but you can, at minimum, ask what industries they heat treat for. For example, if a heat treater exclusively processes nonferrous aerospace parts, they might not be prepared to work with ferrous components for construction equipment.

Q: What equipment will be used for heat treatment?

The equipment being used needs to be appropriate for the work to be processed. For example, a belt line is well-suited to running small-sized steel parts but is ill-suited to running large castings. Furthermore, there are important details to consider in addition to the physical type of furnace such as: protective atmospheres, quench media, quench severity, quench tank agitation, quench media to part weight ratio, etc., to name a few. Many of these factors will influence the required hardenability to properly heat treat a component. Different suppliers may require different alloys based on their equipment, which is a cost that should be considered.

Q: Do you have a certified quality system in place – what accreditations do you hold?

It is important to understand how your heat treater's quality system interfaces with yours. What data do they record and how is it used to control the quality of your product throughout the heat treat process?Are these controls appropriate for your application and do they satisfy the standards that your component must be designed to? Make sure you understand what quality system requirements your heat treater should meet. These will typically be dictated by the end user. Is CQI-9 compliance sufficient or is ISO or NADCAP certification required?

Q: How will my parts be loaded into the furnace for heat treatment? What does the load plan look like?

The load plan is critically important to cost and quality questions. A heat treater might want to squeeze as many parts into a load as physically possible, but such an arrangement can prevent uniform heat treatment of every part in the load. Every part should experience the same thermal cycle whether it is in the corner or center of

Quality audits are an important part of quality heat treating (courtesy of Applied Process).

the load. In addition, some components require special fixtures to achieve and maintain specific dimensional stability requirements.

Q: How will you certify that my parts have been run to specification?

Most heat treaters will provide a certification with hardness information. Keep in mind that hardness does not guarantee microstructure, and it is the microstructure that ultimately delivers the desired mechanical performance. This does not mean that you should not accept a certification with hardness readings. Rather, it means that you should make sure that your heat treater has the ability to design a capable heat treat process for your component that is carefully controlled. Hardness is a very powerful tool, but only when you have a capable process that is carefully controlled. If you desire additional tests like a microstructure or tensile tests, find out if your heat treater has the capabilities to do the testing in-house or if you need to arrange for an outside A2LA-accredited facility to do the work.

Q: Do you have any material and/or process recommendations to improve the functionality of my component or to reduce cost?

Your heat treater can be an invaluable resource. All you have to do is ask. An example of potential assistance that your heat treater might be able to provide includes:

a. Part design considerations that may save you money such as how to optimize the shape of the component for heat treatment;

b. Customized fixtures to minimize shape changes that will require post heat treatment machining or finishing steps;

c. Optimum lot sizes for heat treatment to minimize costs that could result from processing partial loads.

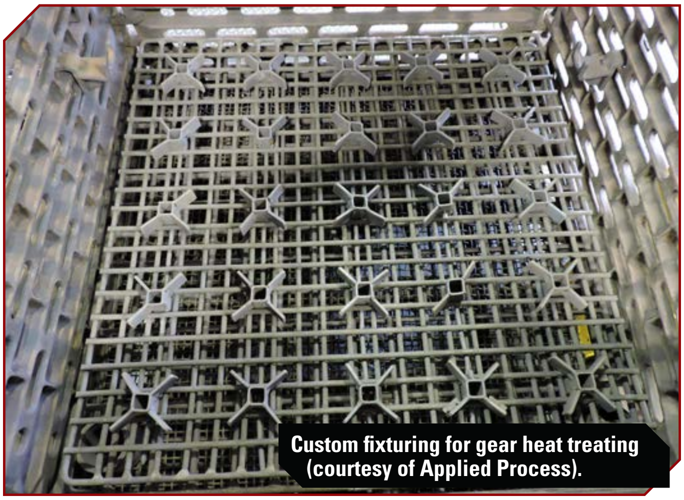
Batch Furnace (courtesy of Applied Process).

Q: Can I tour your heat treat facility?

Observations about cleanliness and organization of work flow will convey important information about the way a heat treater runs the shop, and ultimately how your work will be handled.

Q: What is the typical turnaround time for heat treatment? Will you provide a priority contact list in case I have a question or problem?

It is not unreasonable to ask your heat treater to produce data on turnaround time for the past year. Any business has seasonal peaks and valleys, which will affect turnaround time. Furthermore, you should know who to contact for shipping concerns, production scheduling, technical questions, etc. An email address, while useful, won't help if you have a matter that needs immediate attention. Make sure you receive phone numbers that will eventually be answered by a human being.



Custom fixturing for gear heat treating (courtesy of Applied Process).

Questions to Ask Your Heat Treater Provided by Mike Reichling and Ty Cooper (Cincinnati Steel Treating, Cincinnati, OH)

Q: Are you experienced in heat treating gearing?

Heat treating a forging or a piece of bar is one thing, heat treating gearing is another. Typically when the heat treater sees it, it's semi-finished, meaning there are very few finishing operations to go afterwards. The heat treater has to be somewhat familiar with how to handle gearing. You just don't throw it in a basket—you treat it with care and respect; you fixture it properly to try to minimize any distortion. So you want to find someone who has a certain level of experience.

Q: What processes do you offer?

There are different methods of hardening or heat treating parts. It depends on what's required. Does [the customer] need induction hardening; does [the customer] need flame hardening; does [the customer] need it nitrided or nitrocarburized? A heat treater experienced with gearing should be well-versed in all types of heat treating as it relates to gearing. He may not offer all the processes, but he should be knowledgeable about them and be able to recommend someone who can.

Q: Are you interested in being involved from design to finish?

This is really a pet-peeve of ours. Many times, we'll get a part in here and say, "Oh God, that's a nightmare to heat treat." We encourage customers to get us involved at the beginning. Many times there are things that are let go that could be stopped; there are things they can do that would help the part not distort as much during heat treat. Generally, once you heat treat it, at best you're going to grind and then the stock allowance is minimal — especially if you have a very flimsy, light gear. So the time to talk to your heat treater is at the beginning, before all that happens. Let your heat treater get involved in the manufacturing and design process, because he knows how it's going to react during heat treat.

Q: How big of a part can you handle as it relates to each process offered?

Each process is a little different. Can he induction harden a part that's 40" in diameter, or can he only carburize it? You have to decide what process you want and then find a heat treater who can offer that in that size of part.

Q: Do you have your own metallurgical lab or do you rely on an outside lab for testing?

It's very important that the heat treater have his own in-house lab where he can do microstructure analysis, micro-hardening analysis and carbon analysis. If you rely on an outside lab, then you're going to add time to the process. Most customers don't have that luxury of extra time. Once they make that delivery to the heat treater, they're usually very close to the delivery schedule or they're already

Photo courtesy of Cincinnati Steel Treating

late. You have to have your own lab, and customers need to make sure they're going to a heat treat with a lab and an experienced staff who knows how to use it. This could save several days.

Q: What specifications and/or certifications do you work to as it relates to gear manufacturing?

A lot of the requirements for heat treating gearing will reference a specification. The heat treater has to be certified to this specification or they have to be able to work to a given specification. For instance, in gearing, it's an AGMA standard. There are also ASM standards, where there are very specific requirements you have to meet: the furnace has to meet this certain uniformity; the carbon has to be within a certain range. I would say that not all heat treaters work to those standards. Probably, for gearing, only about 80% of heat treaters have those specifications.

Q: Do you offer copper plating as a maskant?

Oftentimes — and this is strictly for carburizing — there are areas on the gear that don't need to be hard. Many heat treaters will mask those areas off with a paint. Copper plating is 100% effective, while painting is probably 90% effective. Very few heat treaters offer plating, as well. Generally, they can recommend a good plater.

Q: What is the deepest case depth you feel comfortable achieving, and what method of testing is used to verify the case depth achieved?

The case depth is time- and temperature-dependent. The

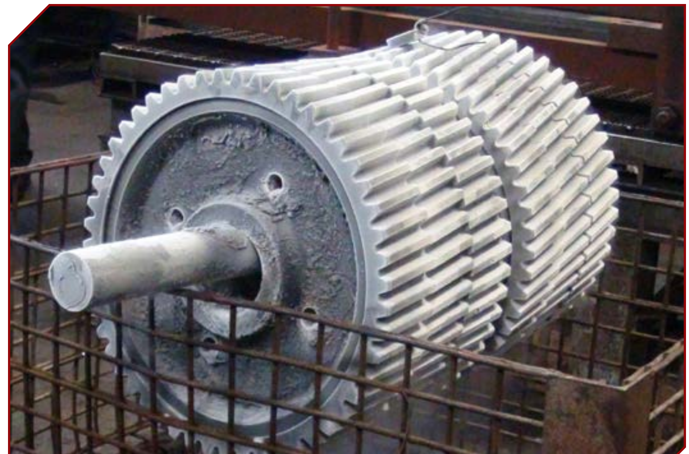


Photo courtesy of Cincinnati Steel Treating.

deeper the case, the longer it takes. You may get some heat treat companies that don't want to tie up their furnace for more than 20 hours at a time, so they will only carburize a part so deep. We run some parts here that will be in the furnace for five days, because they want a quarter-inch case depth. It really depends on the place.

Q: Do you have a metallurgist on staff?

This goes in line with a previous question. Today's [metallurgists] are just not yesterday's [metallurgists]. Today's people are more material scientists; they deal with ceramics, polymers and things like that. The old-time [metallurgists] just know steels and irons and aluminum. So does the heat treater have someone with experience in the kinds of materials that the customer wants processed?

Q: Do you have any experience with failure analysis as it relates to gearing?

Whether it was something they heat treated or something somebody else heat treated, if they have experience at failure analysis, then they're experienced in heat treating gearing. They can offer a lot in pre-design with that information that they have. It's not very common for heat treaters to be experienced in this.

For more information:

Allied Process, Inc.
(734) 464-8000
www.appliedprocess.com

Cincinnati Steel Treating
(513) 271-3173
www.steeltreating.com

Joyworks LLC
(734) 775-3437
www.joyworksstudio.com

For Related Articles Search
heat treating
at www.geartechnology.com

Ipsen

Determining Bull Gear Wear

Email your question—along with your name, job title and company name (if you wish to remain anonymous, no problem) to: *jmcguinn@geartechnology.com*; or submit your question by visiting *geartechnology.com*.

QUESTION

We are currently experiencing wear on the bull gear on our converter at the steel plant.

We want to be able to draw the original gear profile to compare this with the worn tooth before we decide on the next steps.

I have attempted this, but there is a correction factor given and I am unsure how to apply this. Could someone give advice on this?

Please find attached the PDF's for the bull gear and the pinion gear. They are old drawings! The wear is on the wheel.

Expert Response Provided by Charles Schultz, PE:

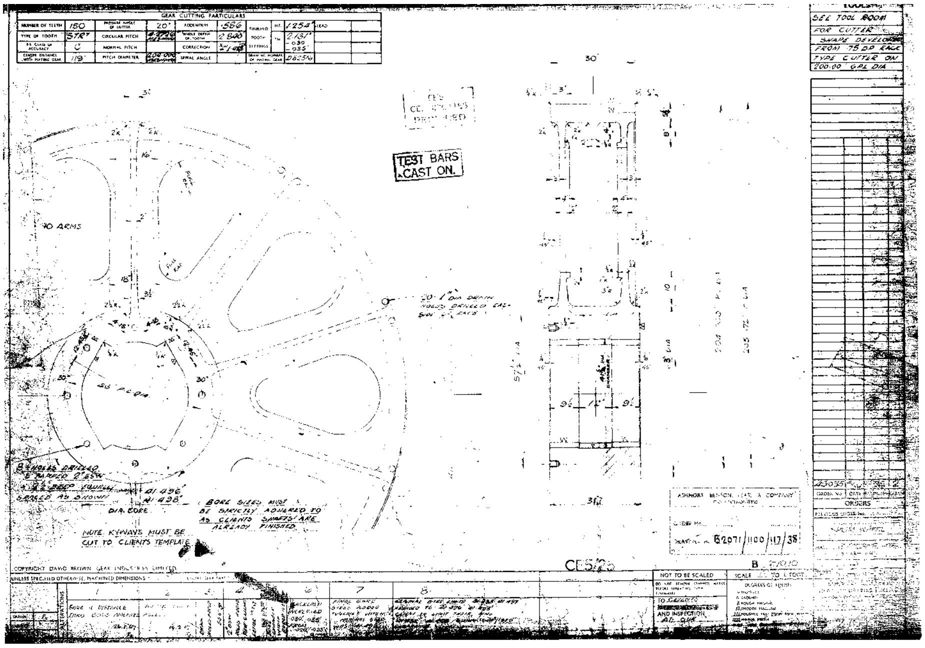
Thanks for your very interesting question and the opportunity to revisit the beginning of my life in gears. As a freshman in high school drafting class, one of my assignments was drawing a spoked gear complete with the tooth flanks. Back in 1967, around the time your gear set was designed, this was an arduous task that certainly tested my skills with a compass and French curve set. Technology has improved since then; you have a number of methods available to you.

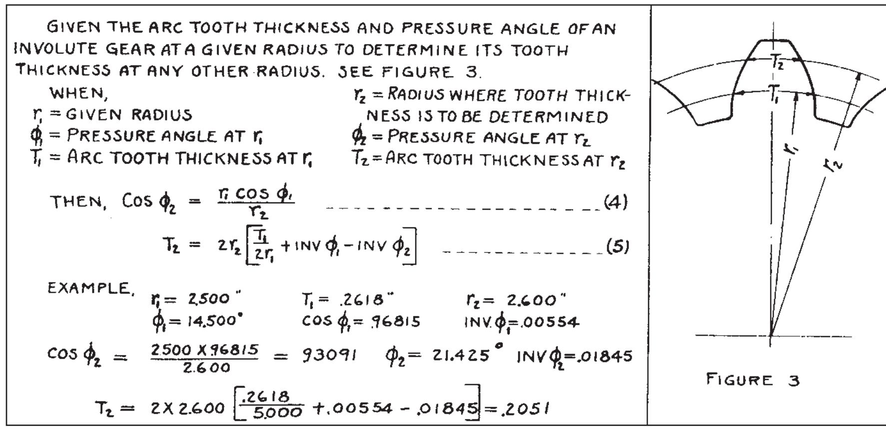
Method 1: tracing the existing tooth flank. You didn't mention if both gear flanks were equally worn. In most cases they are not, so take advantage of symmetry and make a tracing of the least-worn flank, smooth out the profile with a thin flexible scale, and use it to check the depth of damage on the tooth flank. I have used this method in the field and think it will be sufficient for your purposes.

Method 2: use a manual drafting method from the Internet. The biggest change in engineering since 1967 has been the development of the Internet. Now in a matter of seconds you can seek the advice of dozens of people on how to do almost anything. Some of the advice is about as good as I would have gotten from a high school classmate; at least one carefully demonstrated method I downloaded was so bad it wouldn't have passed that 1967 test. Others are very program-specific and may work better.

Method 3: Develop a "point map" from tooth thickness equations. As you mention in your letter, some methods of plotting tooth profiles do not handle "non-standard" geometry well. Your gearset is on "standard centers" but the pinion outside diameter has been enlarged while the gear outside diameter has been contracted. In U.S. gear nomenclature this is a "long and short addendum" situation; the rest of the world calls it a positive and negative rack offset. No matter what you want to call it, there is enough information on the drawing to calculate the tooth thickness at the pitch diameter and the operating pressure angle at the pitch diameter. From those values it is possible to calculate the tooth thickness at any other diameter on the gear. Once you have the tooth thickness at the outside diameter, pitch diameter, and a couple of other diameters of your choosing, it is a simple matter of plotting the profile curve and making a template.

Method 4: Use a commercial program to produce a computer file of the tooth flank. If all of the above sounds like way too much work, there are several gear rating programs and even a few CAD

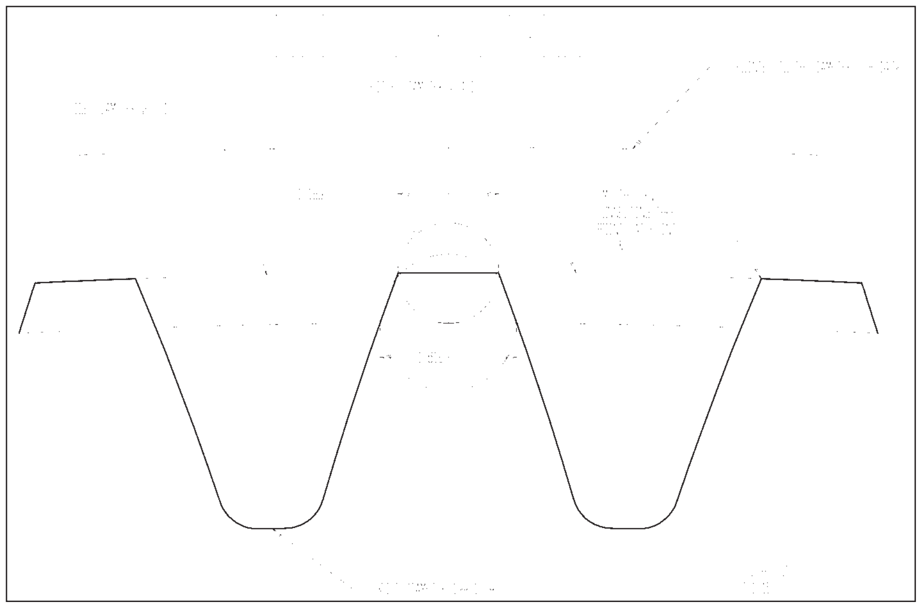




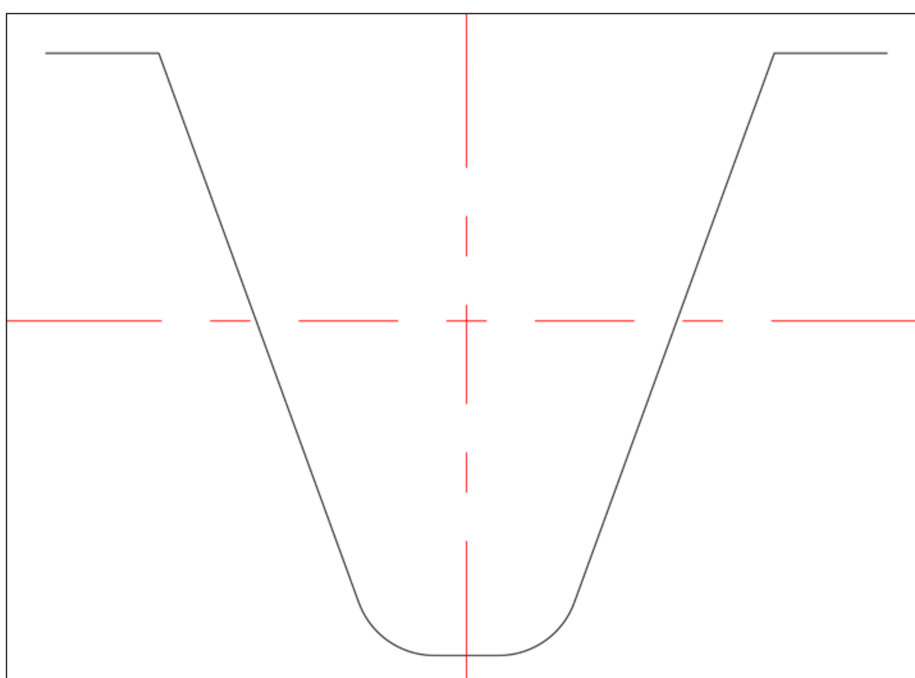
Calculation of gear radii based on tooth thickness, from *Manual of Gear Design,* by Eliot K. Buckingham and Earle Buckingham.

programs with the capability of quickly producing a .dxf file of your gear profile. I used *MITcalc's* spur gear routine and obtained the needed result within a few minutes. The most difficult part of the task was calculating the rack offset coefficient [X1]; several equally reliable methods are available online.

I am curious over what you hope to accomplish by measuring the wear step on a 43-year-old gearset. The specifications call for an induction-hardened pinion running against a 220 HB gear; with proper paste lubrication such gears can last 40 years or more, as your set demonstrates. Your gear could probably be re-cut to obtain a clean tooth flank and be returned to service with a new induction-hardened pinion (suitably adjusted on tooth thickness to maintain backlash) for another 40 years of work. If such a rework is not in the plans, you can certainly monitor the wear step to project the end of useful life or to evaluate the effectiveness of new lubricants. I have had excellent results with moly additives on moderately worn flanks. Your results may vary due to the depth of the existing wear step. Sometimes you can't make the parts last forever.



CAD file created based on customer drawings



Close-up of the final tooth space

Chuck Schultz, P.E., longtime gear man, AGMA member/contributor, and Gear Technology technical editor and twice-weekly blogger, can be reached at *gearmanx52@gmail.com*.

Gear Inspection in a Shop Floor Environment

Günter Mikoleizig

As in nearly all industries, more cost-effective solutions are currently called for in the gear manufacturing industry.

In many respects, quality assurance of gearing plays an important role in achieving such solutions. In the automotive industry in particular, there is now a demand for gearing with more complex designs and individual variants while meeting higher quality requirements, all at a lower cost.

In this regard, an organizational structure with the goal of carrying out quality assurance on the shop floor offers advantages in this context, just to name a few:

- Ready availability of measuring results, for instance, following a tool change on the machining equipment
- Immediate performance of random sample measurements in series production
- Avoidance of rejects due to shorter and faster response times

The following article will address the most important prerequisites for implementing gear measurement in a shop floor environment. In addition to requirements pertaining to the measuring devices, the prerequisites for safe operation of the device will be described, and measuring accuracy will also be examined more closely.

Introduction

New transmission concepts are needed in the automotive industry, with a view to modern drives providing greater energy efficiency. Increasing power density, greater gear ratios, lower noise emissions and improved efficiency result in higher quality requirements for the gearing used in the industry. For this reason, quality assurance for the relevant components must be more comprehensive, but without exceeding the budget.

Quality assurance performed on the shop floor bridges the gap between quality improvement and simultaneous cost limitation. Of course, shop floor quality assurance also involves risks. With careful planning, however, these can be largely minimized.

Installation Location Requirements for a Shop-Floor Measuring Device

The installation conditions of a measuring device are crucial for ensuring the accuracy and reliability of the measuring results (Figure 1). Major influences include the effect of temperature on the measuring device and on the workpiece undergoing inspection, as well as the effect of vibrations on the device and the cleanliness of the workpiece.

The influences shown here are largely absent in an air-conditioned measuring chamber. A measuring chamber has known disadvantages, however, such as long distances and wait times for measurement, air-conditioning costs, and additional staff.

With smart planning, the effects of negative influences can be minimized, even when the measuring device is used on the shop floor:

- Avoid drafty installation locations
- Avoid direct sunlight on the measuring device through windows or roof openings
- Keep blower air from processing machine cooling units away from measuring devices
- Place the workpiece undergoing inspection alongside the measuring device (for temperature regulation)
- Do not install the measuring device next to transport routes (risk of impact, vibrations)
- Keep the measuring device location free of significant sources of vibration (test stands, heavy-duty machining operations, etc.)
- Ensure that the installation location is generally clean

The negative influences outlined here can frequently be found in production lines set up according to outdated standards. Newly installed, modern production lines, by contrast, provide significantly improved prerequisites for shop-floor quality assurance. This is because modern machine tools are completely encapsulated and are frequently equipped with exhaust systems. Quite often, partial air conditioning systems are installed to maintain narrow production tolerances.

Figure 1 Gear inspection in a shop floor environment.

Measuring Devices on the Shop Floor

Measuring devices used on the shop floor must be capable of compensating the prevailing negative effects of this environment. This includes factors such as temperature effects, vibration loads, and dust and dirt at the installation location, among others.

Precision measuring centers for gear measurement, as shown in Figure 2, provide the appropriate prerequisites:

- The measuring centers are made from a single material throughout (cast iron or steel) to avoid material deformation (bi-metal effect) due to temperature influences.
- Moreover, metal (cast iron/steel), when used as a design material for a measuring device, provides the advantage of being identical to the material of the workpieces undergoing inspection. Thus under the influence of temperature, both components exhibit the same behavior, and a natural temperature compensation takes place due to physical effects.
- As the length measuring system, temperature-neutral gauges made of a special glass alloy (Zerodur) are used, with a constant dimension relative to the reference temperature of 68°F.
- Integrated temperature sensors measure the ambient temperature and the temperature of the measuring device. Combined with a software compensation model, the measuring accuracy remains nearly constant over a broad temperature range (64.4°F–86°F). A test conducted over 12 hours comparing measurement results with and without temperature compensation is shown in Figure 3.
- The current workpiece temperature can also be measured and compensated for with an additional sensor.
- High-precision guidance systems for the linear axes and workpiece rotary axis have an anti-friction guideway design and require no energy supply, e.g., via purified compressed air.
- All guidance and measurement systems are installed covered and are protected against soiling and external influences.
- To insulate the measuring device against vibration effects, the device is installed on three diaphragm air spring elements, as seen in Figure 2. Low-frequency vibrations and impacts are thus reliably absorbed.
- The measuring device features a highly accurate 3D tracer head for recording measured values on gearing. In addition to software monitoring of the measuring ranges, the system also includes mechanical protective devices to prevent damage in the event of improper operation or collision.

Figure 2 Precision measuring centers.

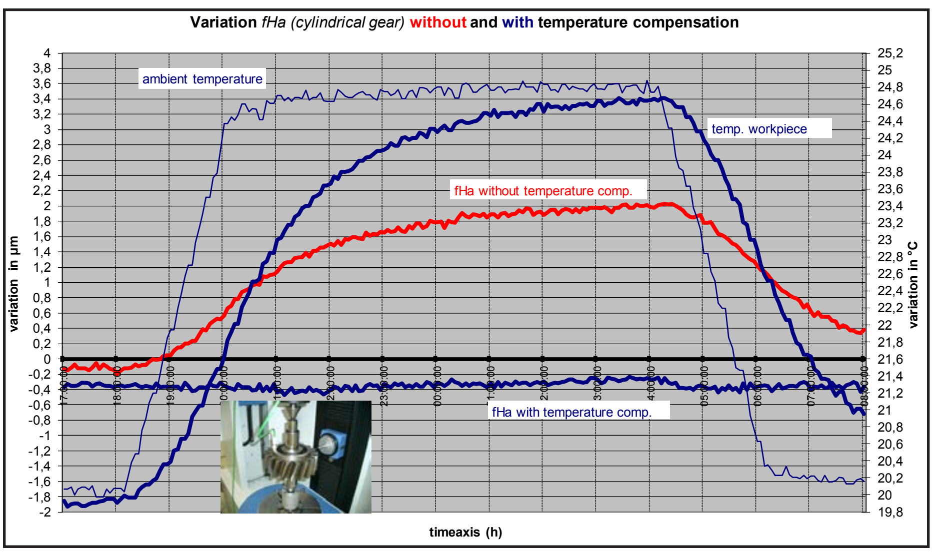


Figure 3 Temperature compensation test results.

Gearing measuring centers as shown in Figure 2 also have the advantage that measured value recording takes place for cylindrical gears with involute gearing according to the generation principle (Figure 4). The involute curve is generated automatically through the coupled rotational movement of the workpiece with the tangential measuring axis, resulting in stable measurement results, even in the case of positional deviations of the probing system due to environmental influences in production, for example.

By contrast, profile measurement based on polar coordinates, as is common in general CMM, is significantly more sensitive to positional variations.

Prerequisites for Reliable Performance of Shop-Floor Measurements

An important feature of shop-floor measurement is that the operating staff for the production equipment (hobbing machines, grinding machines, etc.) also carries out the gear measurements.

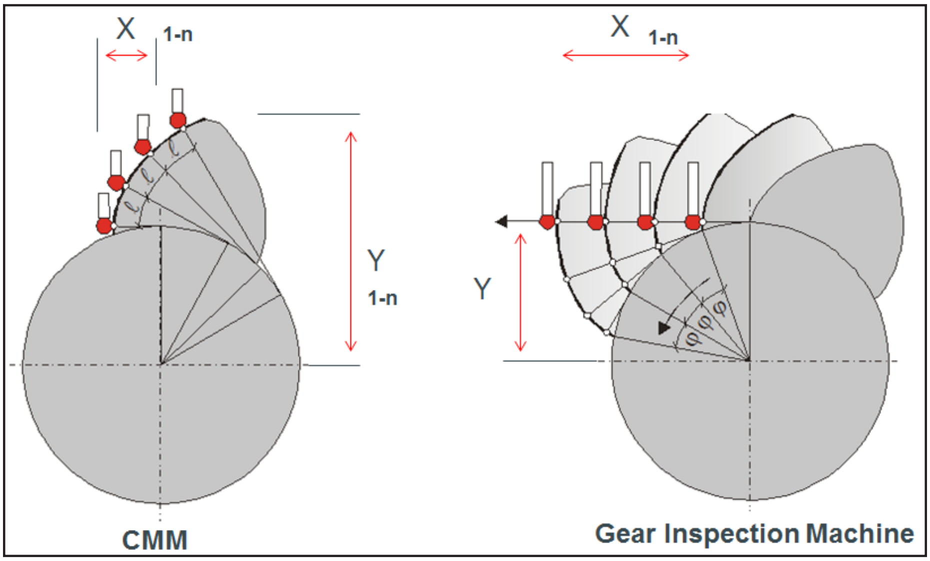


Figure 4 **Measuring principle.**

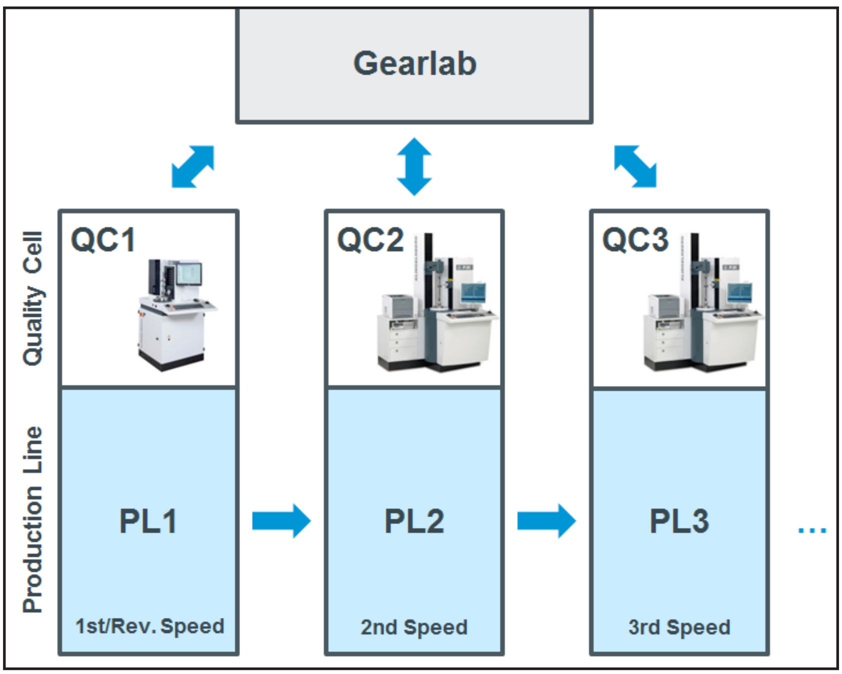


Figure 5 **Layout of a gear component factory.**

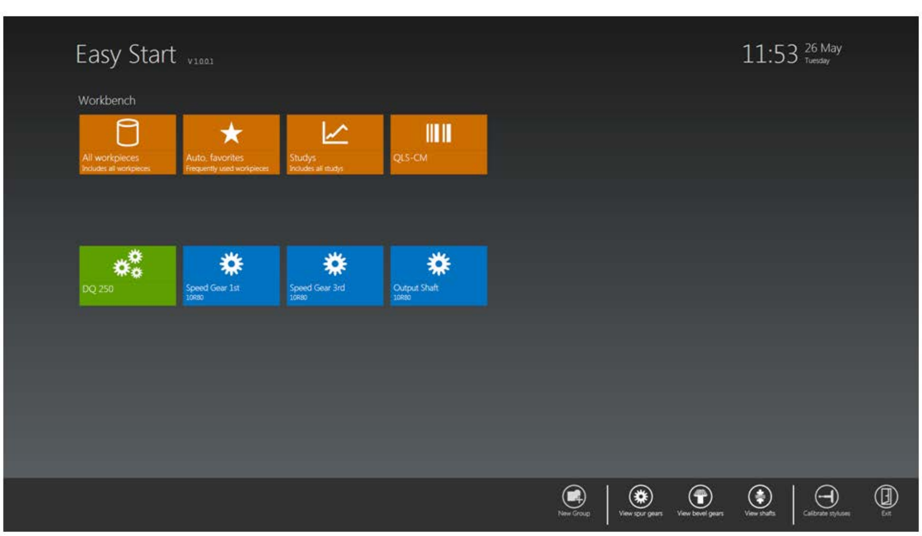


Figure 6 **Software HMI.**

The purpose of the measurements is to inspect gearing quality following a workpiece change, for instance, or during ongoing production.

This means that the measurements are not carried out by trained measurement technicians, but rather by machine operators. Additional measures are therefore required to perform reliable, accurate measurements.

Figure 5 shows the layout of a shop floor with an integrated measuring station. There are production lines (PL1–PLn) for the individual components of a transmission. Each production line has its own test station (QC1–QCn). Each test station is also networked with a central measuring chamber. In this central measuring chamber, prototype measurements are carried out, and the individual measurement programs that will subsequently be transferred to the test stations in production are created. The measuring results recorded on the shop floor are then transferred back to the measuring chamber, where approval will be granted or additional evaluations, such as statistical analyses, will be carried out.

The objective is to enable the operator to carry out the necessary measurements on the shop-floor measuring machine with as much ease and reliability as possible. This relies in part on a simple, precise workpiece fixture. A workpiece fixture between centers lends itself well for shaft-type workpieces. For disk-shaped workpieces and internal gearing, a chuck is the fixture of choice. The chuck should be designed so that few or no exchange parts are required for different workpieces.

An automatic probe change rack on the measuring devices is recommended when using different probe elements for gearing measurement. The probe change rack enables all necessary probe elements to be calibrated automatically at fixed intervals (once per day or once per shift).

The individual measurement programs are accessed by means of a simply designed graphical user interface, as shown in Figure 6. All measurement programs needed for a measuring station or specific transmission components can be programmed into this page as on a desktop and launched as necessary.

The system also includes measurement program retrieval via a barcode scanner.

As already mentioned at the out-set, workpiece cleanliness is important for obtaining accurate, reliable measuring results. The measuring station should therefore also have a workpiece washing station nearby. The advantage of a cold-water washing station is that the workpieces are not warmed, thus obviating temperature-induced measurement variations.

Workpiece temperature is generally a consideration for accurate measurements. In gear measurement, certain measurement parameters are insensitive to temperature, while others respond with greater sensitivity.

Due to the relatively small gearing dimensions in the automotive industry, temperature-induced changes in length are relevant here only for certain parameters. The relative test parameters for profile, tooth trace, pitch and concentricity measurements are rather insensitive. Large, temperature-induced variations can occur, however, when determining tooth thickness or dimension over balls, when the temperature of the workpieces deviates from the reference temperature of 68°F. In this case, a measuring device with workpiece temperature probes is used for compensation.

Maintenance and Calibration of Shop-Floor Measuring Devices

Use of a measuring device on the shop floor generally requires slightly more maintenance and calibration.

Fixed procedures should therefore be specified for individual measuring stations as follows, for example:

Daily maintenance

- Inspection of traversing paths of measuring device axes
- Monitoring of the device for unusual noises during the measuring procedure
- Calibration of the probe elements used (once per shift)

Weekly maintenance

- Inspection of the measuring device for measuring accuracy by means of comparative measurements against the reference standard
- Compressed air supply check
- Cleaning of workpiece fixture components

Quarterly maintenance

- Replacement of filter elements in the control cabinet
- Emptying of condensate from the compressed air service unit

Annual maintenance

- Inspection of the measuring device by an OEM service employee

As regards measuring accuracy, a comparative measurement against a reference standard across all measuring devices is also needed at regular intervals.

Of course, the manner in which the operating staff handles the measuring device is an essential factor in maintenance expenses and in ensuring consistent measuring accuracy. It is certainly an advantage if staff members receive basic training in the proper handling of high-precision measuring devices.

Summary

Quality monitoring of gearing workpieces carried out on the shop floor offers a number of benefits:

- Variations in measuring results are identified more quickly, thereby reducing the number of rejects.
- When changing tools on the machining equipment, the operator can immediately verify the settings by measuring the workpiece and can make any necessary corrections without lengthy production breakdowns.
- The machine operator is thus directly involved in the quality assurance process and consequently performs with a greater overall focus on quality.
- The number of complicated measuring chambers can be reduced.

To successfully introduce quality assurance on the shop floor, however, appropriate prerequisites must be in place. Careful planning is absolutely necessary.

Exchanges of experience with other companies that have successfully introduced this production structure can certainly be helpful.


Dipl.-Ing. Günter Mikoleizig currently heads the Product Management and Application Engineering-Department for gear inspection machines at the Klingelnberg GmbH, Germany. For more than 30 years he is working in the field of gear inspection technology and he is well experienced with the design and development of inspection machines and the product management. He developed a product line of inspection machines for all kind of gears and other related parts with small dimensions up to very big sizes. Mikoleizig presented papers about gear inspection worldwide and is also an active member of national and international standardization committees.

I Like Big Gears and I Cannot Lie!

The elements of a successful big-gear metrology system

Dwight Smith

Many years ago, when asked how the five-meter gear was checked, the quality manager responded, "When they're that big, they're never bad!" That may have been the attitude and practice in the past, but it no longer serves the manufacturer nor the customer. Requirements have been evolving steadily, requiring gears to perform better and last longer.

Industry has shifted towards lower weight components, higher efficiency and more compact gearboxes. The transition from mild steel gears, which would "run in," to hardened and ground gears has increased the need for accurate inspection of large gears.

Past practice was to check large gears (which, for the purpose of this article, are gears over three meters outside diameter) for what could be reasonably checked: span measurements, and perhaps root circle and tip circle measurements. The gear's quality was then inadvertently tested out in the actual application, perhaps on a large piece of mining equipment or a process critical gearbox.

If the manufacturing plant had appropriate size-over-pins measuring tools, multiple pin sizes could be used to estimate the involute profile, typically at three places. In some shops, this methodology may have been the only practical way to get even an approximate check of the involute profile. Helix evaluations, if they were ever done, were crude and imprecise. It is quite possible for a gear to be in tolerance for size over pins and be out of tolerance for many other parameters.

In the current industrial gear marketplace, even large gears are often finished with the grinding process. This allows designers the ability to optimize the micro-geometry with profile and helix modifications. These modifications need to be verified, which leads to the requirement for gear metrology systems for large gears.

Current gear grinders for large gears typically include on-board inspection. This provides very valuable setup and process information, but is often not accepted by the end customer as final inspection. Since the measurement uses the machine tool's axes for measurement, errors in alignment may not be detectable. If, for example, the grinder is out of mechanical alignment or adjustment, an improperly ground helix may be undetectable by the on-board system since the same incorrect motion is used for both grinding and measuring. Similar error masking is possible if the part is eccentrically mounted on the work table. Independent inspection will correctly measure the gear relative to the appropriate datum.

Market and competitive forces are also driving the need for large gear inspection. A manufacturer may enjoy a competitive edge by being able to supply accurate gear inspection information as proof of conformance. Many manufacturing people say, "The first question is: how will I make it? The second question is: how will I check it?" Customers are increasingly asking, "How will you prove that you made the gear to the specifications?" Independent gear inspection (not on the manufacturing machine) provides this proof to the customer or end user of conformance to specifications and quality.

Figure 1 Large gear metrology system.

Measuring Large Gears

Manufacturers of large gear inspection equipment have to accept and meet quite a number of challenges with respect to physical size, moving masses and thermal effects. Safety is also an important factor due to the size and weight of large gears and the potential for injury.

Even though the measuring principles on very large equipment are the same or similar to small ones, it would be wrong to expect identical assemblies — just with larger dimensions. Just taking a small machine and making it larger isn't the solution (Fig. 1).

Mechanical Motion Control

The large column and horizontal slides of a large gear inspection system are heavy and have significant inertia. Moving these masses with accuracy and precision is difficult. Conventional machine design with ball screws and roller ways often exhibits stick/slip and other mechanical characteristics that degrade the accuracy of motion. These nonlinear responses of an axis introduce hysteresis into the system and reduce repeatability of measurements.

The current state-of-the-art in large gear inspection machines incorporates granite guideways with air bearing slides (Fig. 2). This provides a very low friction and eliminates the stick/slip of roller ways. This in turn produces more linear responses, better repeatability, and makes error compensation more accurate.

Granite has been used for highly accurate CMMs and other measuring instruments for many years. It offers a number of benefits for accuracy-determining components of industrial measuring technology:

- Due to the age and origin, granite is free of internal tensions and thus externally stable in the long term
- The thermal expansion coefficient is only about half that of steel and a quarter of aluminum
- It is highly wear-proof, pressure-proof and deforms 25% less under load than an identical aluminum component – at a higher specific weight of only about 5%
- Granite is non-magnetic, non-corroding, and has vibration damping characteristics
- Proper processing produces nearly pore-free, flat and level surfaces that provide a solid basis for air bearing guides — a well-established technology that was a prerequisite for the ultimate accuracy of coordinate measuring equipment.

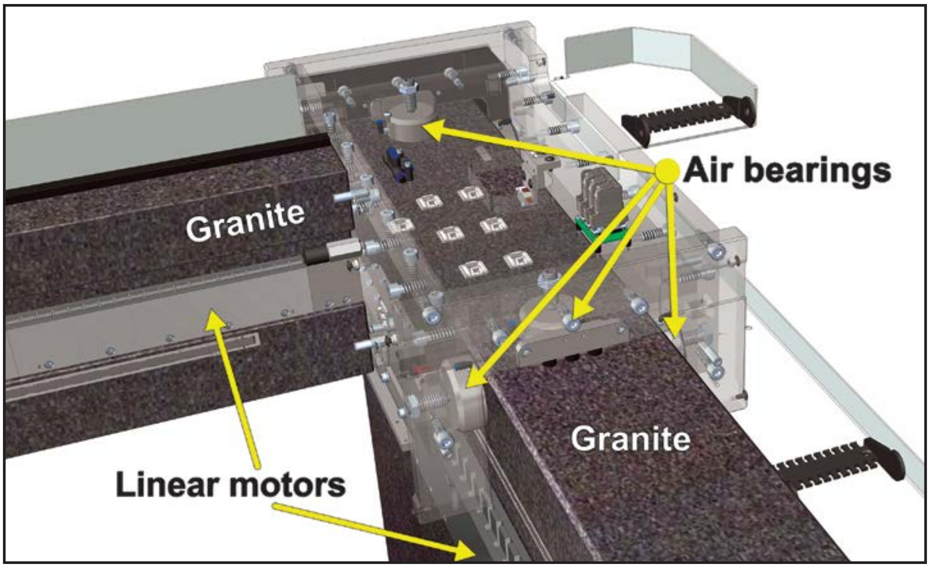


Figure 2 Granite guideways and air bearings.

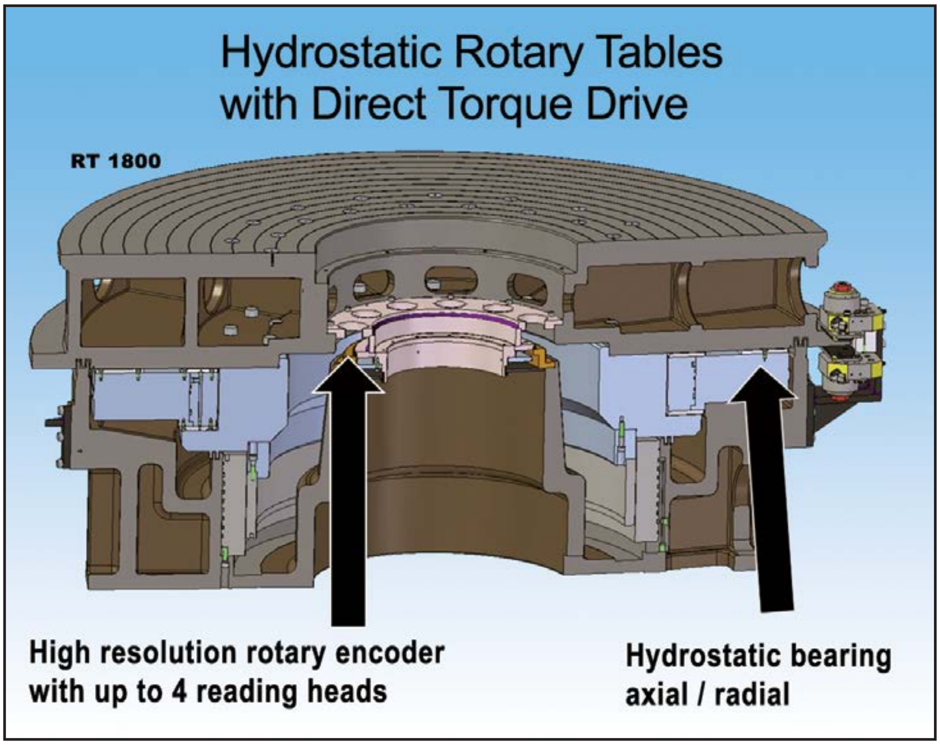


Figure 3 Hydrostatic rotary table.

Hydrostatic rotary tables (Fig. 3) with direct drive torque drive motors can provide the precision required for measurement of large diameters and heavy gears up to 40,909 kg (90,000 lbs.). Hydrostatic bearings have no metal-to-metal contact, extremely low friction and virtually limitless life. Conventional ball bearing tables and air bearing tables typically have much lower weight capabilities. These well-proven and technically advanced hydrostatic rotary tables are normally used in large gear grinding machines.

Special mechanical solutions for the rotary table and specific control measures are needed for measuring large gears in respect to polar inertia. The machine needs to be able to accurately position a part weighing 40,000 kg as well as one weighing as little as 28g (1 oz). In addition to the part's polar inertia, the inertia of the equipment's workpiece support has to be considered. In order to determine the actual conditions, the gear inspection system needs to be capable of automatically adjusting multiple drive parameters in order to gain accurate and repeatable measuring results. This is accomplished through the use of intelligent algorithms and intelligent drives.

A special design of rotary encoder is used for this large application. It has a large diameter and utilizes up to four reading heads and double-scale interpolation for enhanced accuracy.

Thermal Stability

The larger the component and the longer the inspection time, the larger the change in size as a result of temperature change. By using granite for the machine guideways, this change is minimized due the low coefficient of expansion and granite's relatively slow rate of change.

To compensate for deviations caused by temperature change, the equipment needs to provide several temperature sensors for the machine and workpiece, and corresponding algorithms to consider the temperature effects on the evaluation of measuring data.

For production efficiency, inspection equipment is typically installed close to the production facilities. Due to equipment size and location, workpiece transportation methods and paths often lead to a working environment with temperature swings. To meet this challenge, large gear inspection equipment must have long-term thermal stability, temperature compensation and way covers.

To achieve this stability, the base plate and at least the Y and Z linear axes (used for the needed motion for generative gear inspection) should be of granite construction to take advantage of the material's low coefficient of expansion and resistance to the effects of temperature. Each axis has temperature measurement equipment built in, and a probe is used on the part to measure the temperature. This data is used to compensate for the temperature of the part and the machine's axes.

The ways are covered and create a microclimate effect, which reduces temperature variation. This damps rapid transient fluctuations in temperature. On large systems, these way covers are also walkways for operator access to the machine and workpiece.

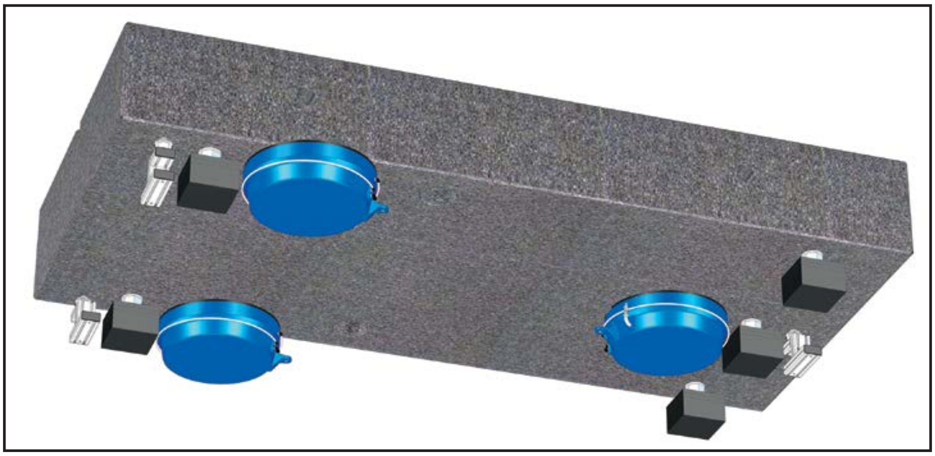


Figure 4 Air suspension vibration isolation.

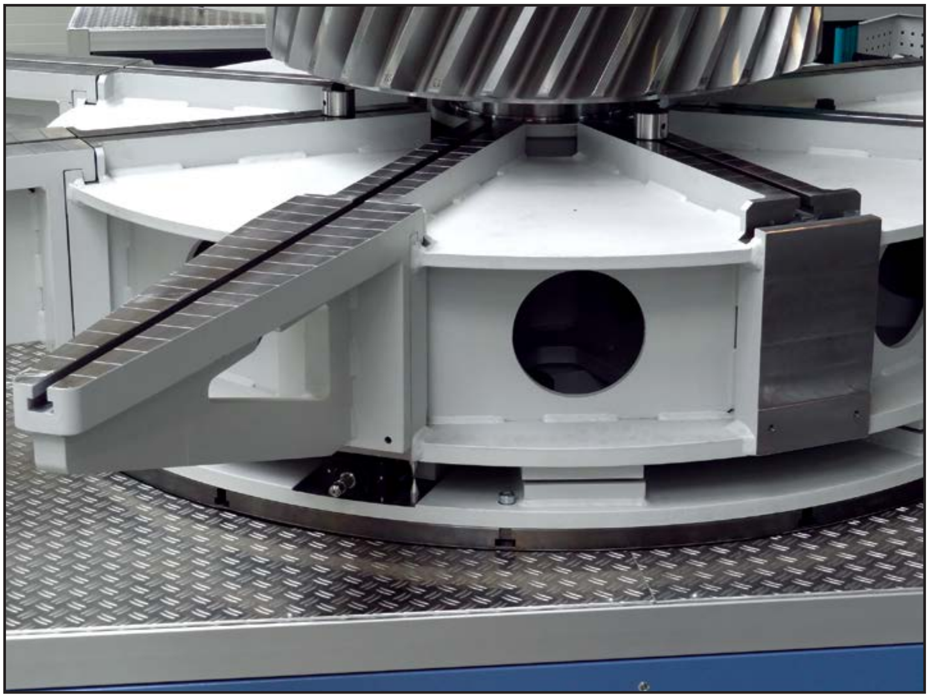


Figure 5 Mounting and centering.

Vibration Control

Rather than relying on an expensive special foundation, a modern large gear inspection system employs an active, computer controlled, air suspension system (Fig. 4). This isolates the machine from vibrations in the shop environment and keeps the machine constantly level. This technology has been long proven in high accuracy CMM applications for many years.

A concrete floor or foundation is subject to changes over time. Rather than relying on the foundation or floor for stability, a large gear measuring system requires intrinsic stability independent of the floor or foundation.

In addition to the stability it provides, the active air suspension, with its effective isolation, can substantially reduce installation cost and allow more freedom in locating the machine.

Mounting and Centering

In stark comparison to small gears, large workpieces (externals or internals) require intelligent, operator-friendly solutions for safely mounting and centering the parts. It is no longer necessary or acceptable to use a huge hammer or swinging ram to center heavy parts. Once mounted, the system completes the fully automatic procedure to determine the axis of the workpiece.

Figure 5 shows a leveling table with a cross-slide and the possibility to extend the mounting range by means of extension arms (spider).

The mounting and centering a heavy part can be accomplished in four steps:

1. Mount 3 "fix points" on the table in positions provided by the machine using built-in utilities
2. Load part on the leveling table against or near fix points for pre-positioning
3. The system will automatically probe the actual position of the part to determine the needed movements of the cross-slide
4. Move the cross-slide axis by means of monitoring the actual position on the screen, shown in real time by the operator software

In addition to the ease of use this provides, a substantial time savings and efficiency improvement is also realized. It is also safer for operators and can reduce the chance of damage to the part.

Safety

To protect operators from accidents during the measurement process, large gear inspection machines must provide additional safety features not needed on small- or medium-size machines. These safety systems must be standard equipment because safety is required and not optional.

A PC-based intelligent laser system (Fig. 6) allows customizable safety zones around the machine and exclusion zones for fixed areas like room columns. Signal lights indicate the status of the machine. If there is movement into the protected area, layers of increasing audible and visible warnings precede the last warning with horn, and finally, an emergency stop will occur.

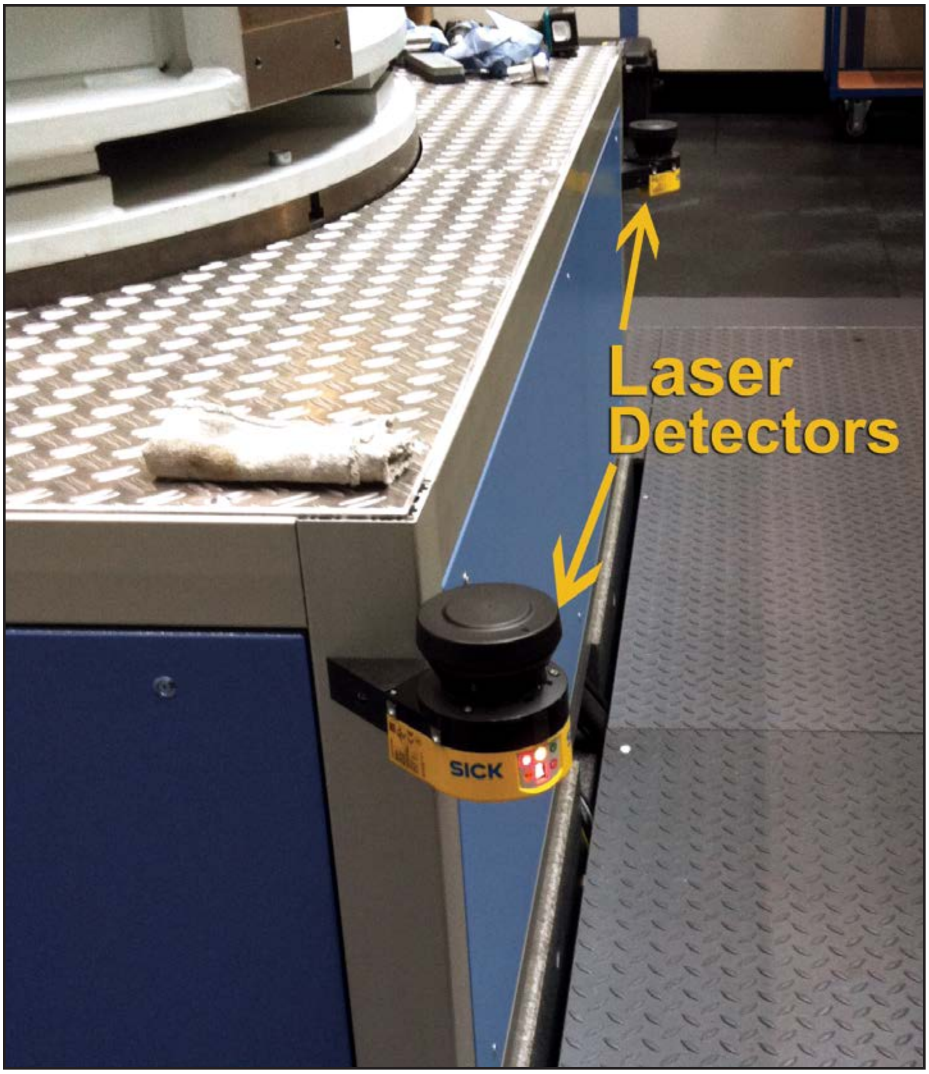


Figure 6 **Intelligent laser safety system.**

Customized Versions – One Size Does Not Fit All

Any gear measuring machines for workpieces with diameters of 1,600 mm and larger should fulfill all individual demands of the customers and special requests of the operators.

To provide this customization, R&P Metrology GmbH uses modular design by which individual machines are configured:

- Individualized design and size of granite base plate to fit into existing facility
- Several options for vertical measuring lengths
- Rotary tables with different load capacities using air bearings or hydrostatic bearings as appropriate
- Various tailstock lengths in fixed or movable positions
- Intelligent solutions for centering, holding and clamping workpieces
- Complete software library including 3-D software packages

Big Gears Require Special Solutions

From the above discussion, it is clear that large gear metrology requires careful application of appropriate technologies rather than just "supersizing" a standard smaller gear checker.

Taken together as a system, the use of intelligent algorithms for automatically determining and controlling polar inertia, granite guideways and air bearings, hydrostatic rotary tables, thermal stability/compensation and vibration isolation combine to form the benchmark for the current state-of-the-art in large gear metrology. Controlling the large masses involved, thermal stability, part centering and alignment, safety and customization all need to be taken into consideration and addressed to produce accurate and reliable gear inspection systems for large gears.

Dwight Smith is the Sales Manager for Kapp Technologies, and product manager for the R&P Metrology line of large gear measuring equipment, which Kapp Technologies distributes in North America. He has over twenty-five years of experience in the gear industry and gear metrology. He chairs an AGMA committee and is an instructor for the AGMA Basic Gear School, and presents at the annual Kapp Niles Rocky Mountain Gear Finishing School. For more information about R&P Metrology machines please contact Dwight Smith at 734-516-1365.

The Basics of Gear Theory, Part 2

Hermann J. Stadtfeld

Bevel Gears: By the Book

Introduction (Chapter 1, Part 2)

The first part of this publication series covered the general basics of involute gearing and applied the generating principle of cylindrical gears analogous to angular gear axis arrangements the kinematic coupling conditions between the two mating members have been postulated in three rules. Entering the world of bevel gears also required to dwell somewhat on the definition of conjugacy. The second part is devoted to the different generating gears and the chain of kinematic relationships between the gear - gear generator - pinion generator and pinion. In addition to the kinematic coupling conditions, the five geometric flank form rules are discussed in order to lay the grounds for the understanding of the different kinds of generating gears. After this, the generating gears of the most common bevel and hypoid gear design and manufacturing methods are discussed. The lineup covers face milling and face hobbing as well as generated and non-generated pinion-gear systems. This section will provide the reader with the deeper understanding of the strength of the different systems and their limitations. It is also explained, how some of the limitations can be overcome if certain measures are applied. **—Hermann J. Stadtfeld**

Geometric Flank Form Rules

1. Flank lines 2 times steady differentiable
   - No steps
   - No edges (elbows)
   - No steps in curvature changes
2. Steady monotonic rising or falling flank lines
   - No inflection points
   - No maxima and no minima
   - Special case straight bevel gears
3. Spiral angle rises along flank line from inside to outside
   - Special case straight bevel gears
4. Pressure angle not allowed below limit pressure angle (meshable profile)
5. Spiral angle limit not exceeded

Violation of the rules will reduce the mesh performance and can even result in the complete loss of the ability to transmit motion and torque. An example of this is Zerol gears *(see Chap. 4, General Explanations, Fig. 26*). Zerol gears have an arc-shaped flank line with zero degree spiral angle in the middle of the face width; this violates rules 2 and 3. The consequence is an instant contact zone moving during meshing from the outside to the inside, and then back to the outside. This gives a consolidated contact area (tooth bearing) with a tendency to split into parts. The results are mesh disturbances that increase with higher flank line curvatures.

Generating Gears of Bevel Gears with Parallel Depth Tooth

The kinematic requirements conditions are applied in this next section to the four methods in Figures 12–15. It has been assumed that conformance to the geometric flank form rules is given in all cases. In order to achieve congruent generating gears, certain geometric and kinematic features in the bevel gear generators are required. The greatest influence is the tooth depth characteristic along the face width (tapered or parallel).

Method "A" (Fig. 12) was already globally discussed (Figs. 9–10). The generating gear axis of the pinion is labeled in the top part of the figure as "Pinion Generating Gear Axis." The pinion cutter rotates around the "pinion cutter axis," which is parallel to the generating gear axis. The generating gear plane is equal to the pitch plane. It contains the axis Z_4 and stands perpendicular to the drawing plane. The same rules explained for the pinion generation apply also for generating the ring gear. In addition to the generating gear orientation and axis location, the blade profiles are also congruent (see pinion cut-

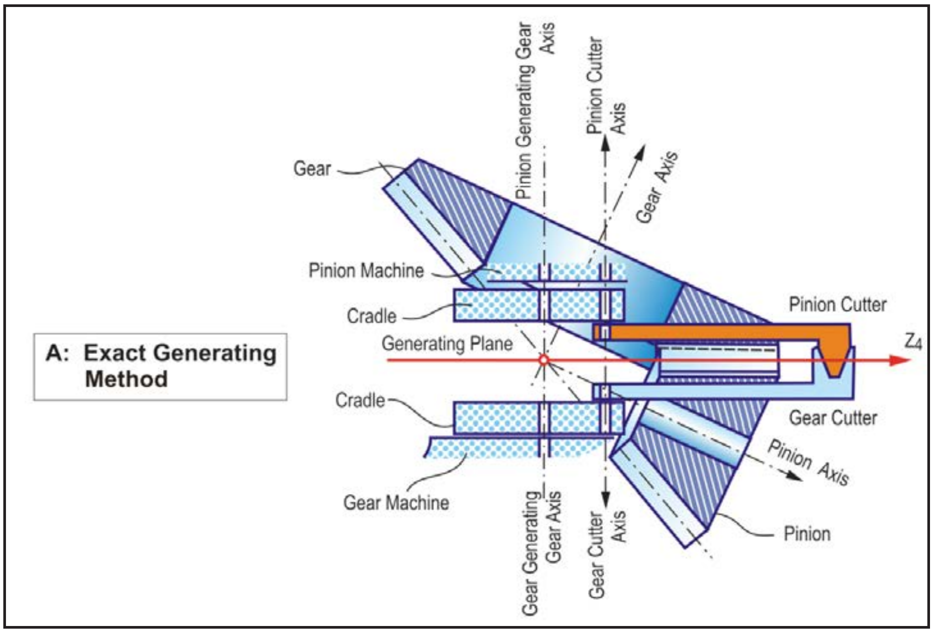


Figure 12 Generating model for bevel gears with parallel tooth depth Method A, generated pinion and ring gear.

ter and gear cutter, to the right in Figure 12). The kinematic coupling requirements for method "A" are entirely fulfilled.

The orientation of pinion and ring gear (Fig. 12) for this method is identical to the orientation of pinion and ring gear in their final gearbox assembly. Pinion and ring gear have in most cases a different number of teeth but their common generating gear does not. The number of generating gear teeth has to be calculated in order to determine its precise rotation dependent on the rotation of pinion or ring gear. The pitch surfaces of the two meshing gears roll onto each other without sliding (like a slip-free traction drive). At a certain radius R, the relationship between the circumference of the generating gear to the circumference of the work gear (on the pitch cone element) must be determined. The work gear pitch angle has the value γ; the generating gear pitch angle is 90°.

$$z_E/z_W = [R * \sin(90°) * 2 * \pi]/[R * \sin(\gamma) * 2 * \pi] \quad (10)$$

or:

$$z_E/z_W = 1/\sin(\gamma) \quad (11)$$

which delivers:

$$z_E = z_W/\sin(\gamma) \quad (12)$$

$$UDIF = z_w/z_E \quad (13)$$

where:

z_E number of teeth generating gear
z_W number of teeth work gear
R observed radius
90° pitch angle generating gear
γ pitch angle work gear

UDIF ratio of roll (generating gear rotation/work gear rotation)

The angular velocities of two meshing gears have the opposite relationship than their number of teeth; the work gear rotation is therefore calculated by dividing the generating gear rotation with *UDIF*.

Figure 13 shows a different realization, which will also satisfy the kinematic coupling requirements (method "B"). The gear cutter on the gear machine (without generating provision) cuts the ring gear (top) without generating motion. The Formate ring gear (see chapter 0) produced this way does not include any involute curvature in the flank profiles. In order to generate the pinion, a virtual version of exactly this ring gear is used as generating gear. In order to do this, the pinion cutter has to be adjusted with its axis perpendicular to the pitch angle of the ring gear and has to rotate during the pinion generation around the pinion generating gear axis (which is identical to the gear axis). The blades simulate one tooth of the generating gear, while the cutter rotates around its axis. The blade profiles (Fig. 13, right) are congruent. The generating rotation around the pinion generating gear axis (= gear axis) simulates the rotation of the generating gear. With this arrangement a conical generating gear was created that cuts and rolls conjugate pinion tooth slots if the pinion blank is placed correct relative to the generating gear. The orientation of pinion and ring gear in Figure 13 for this method is identical to the orientation of pinion and ring gear in their final gearbox assembly. Method B is called "Formate" (non-generated ring gear - generated pinion).

The ring gear requires no generating roll during its creation, while the pinion ratio of roll is calculated as:

$$UDIF = z_{Pinion}/z_{Gear} \quad (14)$$

where:

z_{Gear} number of teeth generating gear
z_{Pinion} number of teeth work gear

Both methods "A" and "B" are valid for continuous as well as for single indexing processes. The differences between those cutting processes are explained in detail in Chapters 4, 5 and 9.

The case of a hypoid offset between pinion and ring gear (*see also chap. 4.6 "Hypoid Gears"*) is investigated with the two generating models (Figs. 14 and 15; Methods C and D). If the model of the ring gear, like in the case of method "B," should be used as a generating gear (Formate), then the generating gear model is easy to understand. A pinion blank is positioned in front of the generating gear while its axis is not intersecting

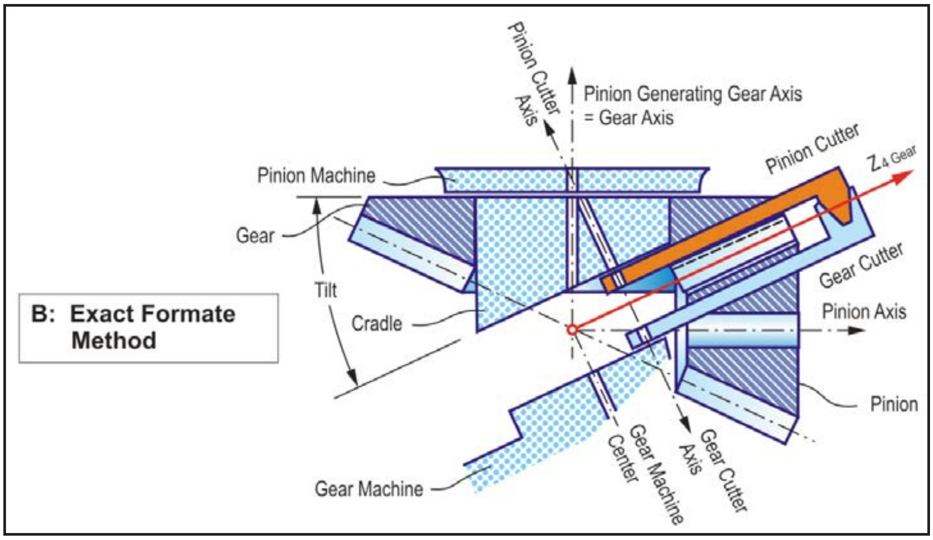


Figure 13 Generating model for bevel gears with parallel tooth depth Method B, formate.

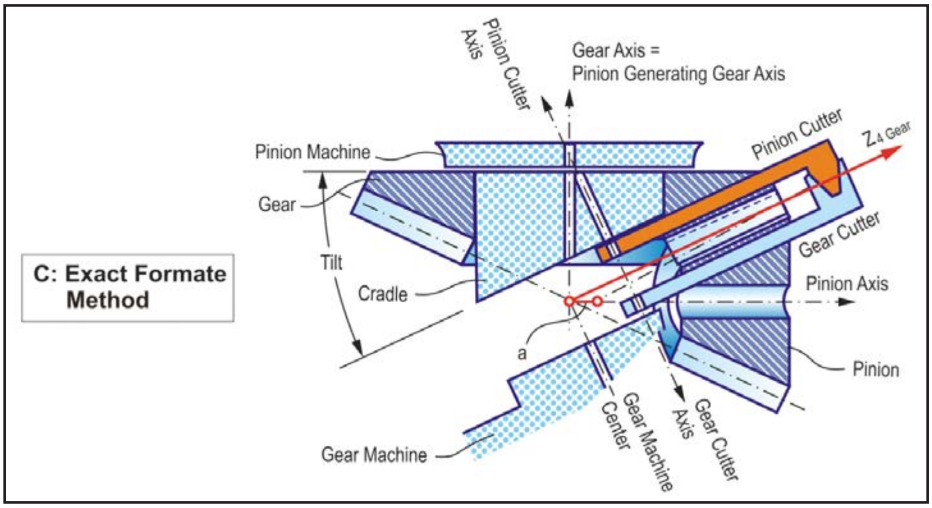


Figure 14 Generating model with parallel tooth depth and offset Method C, formate.

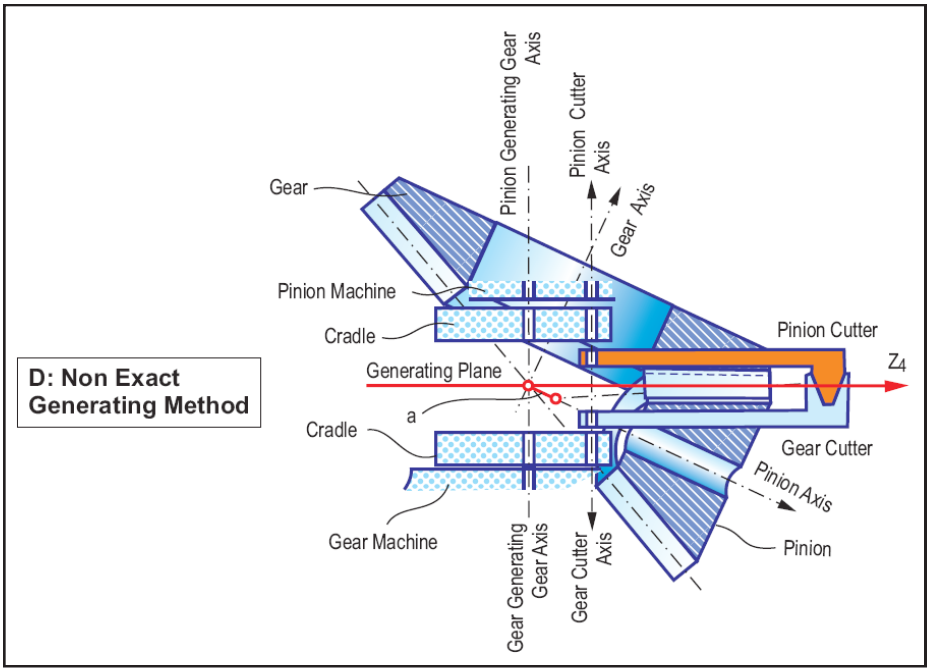


Figure 15 Generating model with parallel tooth depth and offset Method D, generated pinion and ring gear.

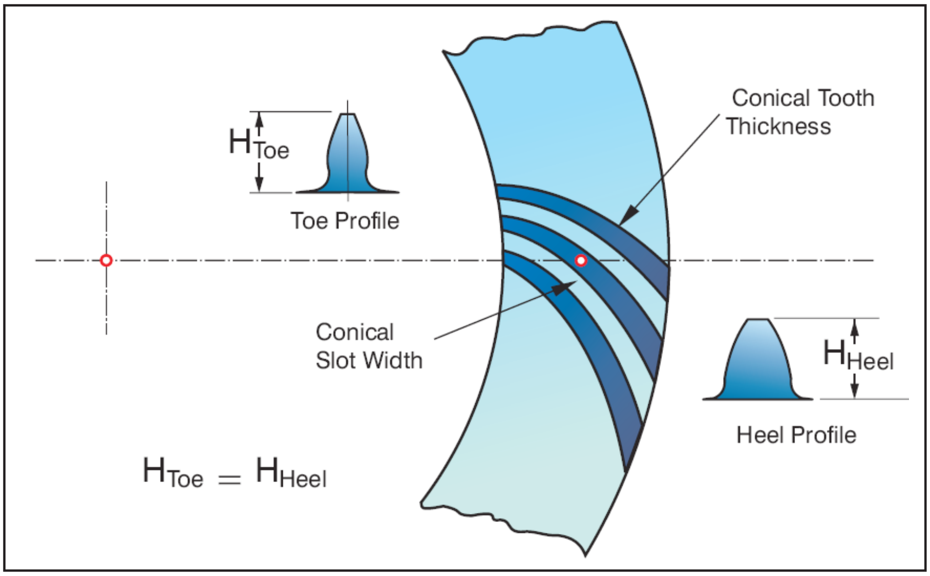


Figure 16 Conical tooth thickness change and conical change of the slot width in case of parallel-depth teeth and a continuous indexing process.

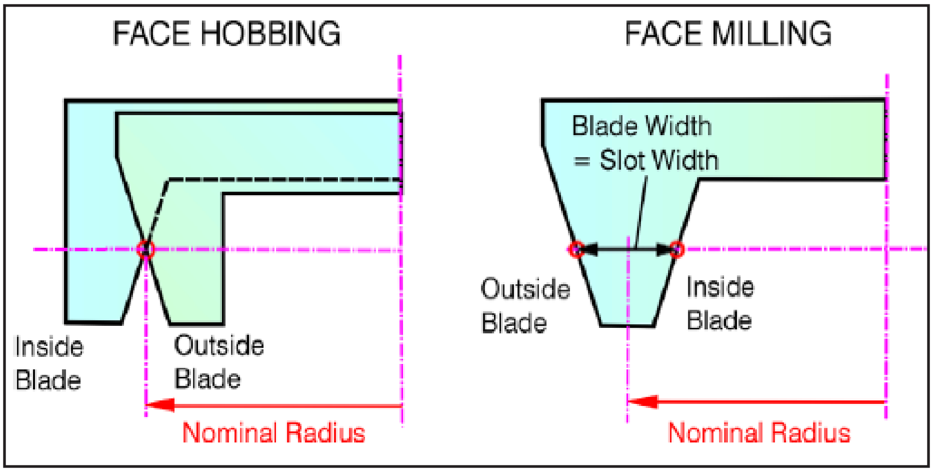


Figure 17 Blade orientation—left for face hobbed, and right for face milled bevel gears.

with the generating gear axis, but crosses it under a certain distance (offset). The difference from Figure 13 is given with the distance "*a*" between the pitch cone apexes in Figure 14. The generating gears for pinion and ring gear, as well as their axes, are identical, since with this method the generating gear is equal to the ring gear. It can be concluded that all kinematic coupling requirements for method "C" are fulfilled.

In the case where the pinion and ring gear are manufactured by rolling them both on a plane generating gear like in method "A," but with an offset between their axes, deviations from a conjugate pair will occur.

Method "D" (Fig. 15) applies the same generating gear for the generation of both, pinion and ring gear, which satisfies the first two kinematic coupling requirements. The surfaces of engagement between generating gear and ring gear and between generating gear and pinion are not congruent because they lie about the axes offset apart (in offset direction). It is possible to rotate them "into" each other, but they are still not exactly congruent. Although the blade profiles in Figure 15 are congruent, the generating gear flank surfaces will still deviate from each other due to the axes offset. The non-conformance with one of the kinematic coupling requirements causes, in this case—surface deviations—which can be compensated to a large extent by first order corrections.

The pitch line (flank line through the pitch point in Figure 5) in case of parallel depth teeth is parallel to the root line. Identical generating gear axes and congruent generating gear flank surfaces can therefore be achieved and the kinematic coupling conditions 1 and 2 can be satisfied.

In order to achieve a proportional and balanced relationship between tooth thickness and slot width along the tooth face it has been shown that bevel gears manufactured with a continuous indexing process (face hobbing) require a parallel tooth depth to fulfill those requirements and deliver at the same time conjugate flank pairs. Bevel gears manufactured by face hobbing have in general a flank line with an epicyclic form. Tooth thicknesses and slot widths are the result of an even split of the gears circumference due to the process' kinematics. Also between outside diameter (heel) and inside diameter (toe) a proportional adjustment of tooth thickness and slot width depending on the radial position occurs (Fig. 16).

Face hobbed bevel gears can be lapped after heat treatment in a short time with good results. The precise bevel gear grinding of epicycloids in a completing process to the contrary is not possible. A precisely defined flank form of the hard finished face hobbed bevel gears can be achieved by (hard) skiving (*see also Chapters 9 "Cutting Methods"; and 11 "Hard Finishing, Grinding and Skiving"*).

Generating Gears of Bevel Gears with Tapered Depth Teeth

Bevel gear sets manufactured in the single indexing process (face milling) have circular flank lines. A proportional tooth thickness and slot width split like in face hobbing is not acceptable. If the objective is a tooth thickness and slot width change along the pitch line similar to that of face hobbed gears, it is necessary to use convex and concave flanks cutter heads with different radii and also different machine settings (*see also Chapter 5, Practical Characteristics*). Cutting of convex and concave flanks has to be done in this case using two separate cutting cycles. If both sides of a slot are machined with only one cutter head having outside and inside blades (Fig.17, right side), then a parallel slot width and a conical (tapered) tooth thicknesses will result (Fig. 18).

Since this applies initially also for the mating gear, a pinion and a gear manufactured this way would not fit together. A tapered depth tooth, by lifting up the root towards the smaller diameter, will still maintain a parallel root width but also achieve a proportionally reducing (conical) slot width from outside to inside (Fig. 19).

Lifting the root up is possible via the dedendum angle (Fig. 20); this is so only with generating gear configurations different from those as previously shown (Figs. 12–15). As a result, the introduction of a dedendum angle requires also the introduction of a corresponding addendum angle. This is necessary in order to avoid interferences of the toplands with the root fillets of the mating gear (which also requires a tapered depth

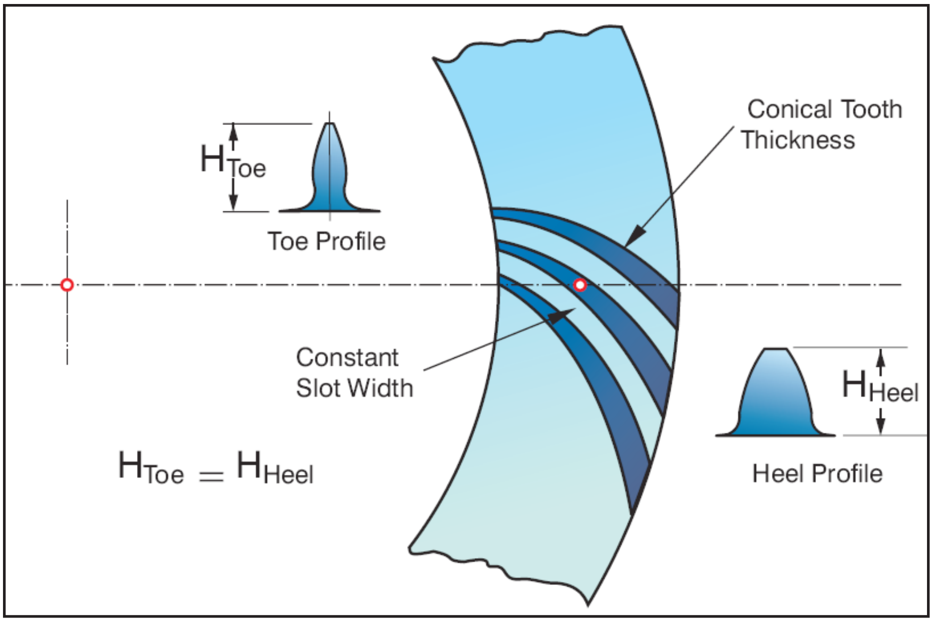


Figure 18 Tapered tooth thickness and constant slot width (face milling).

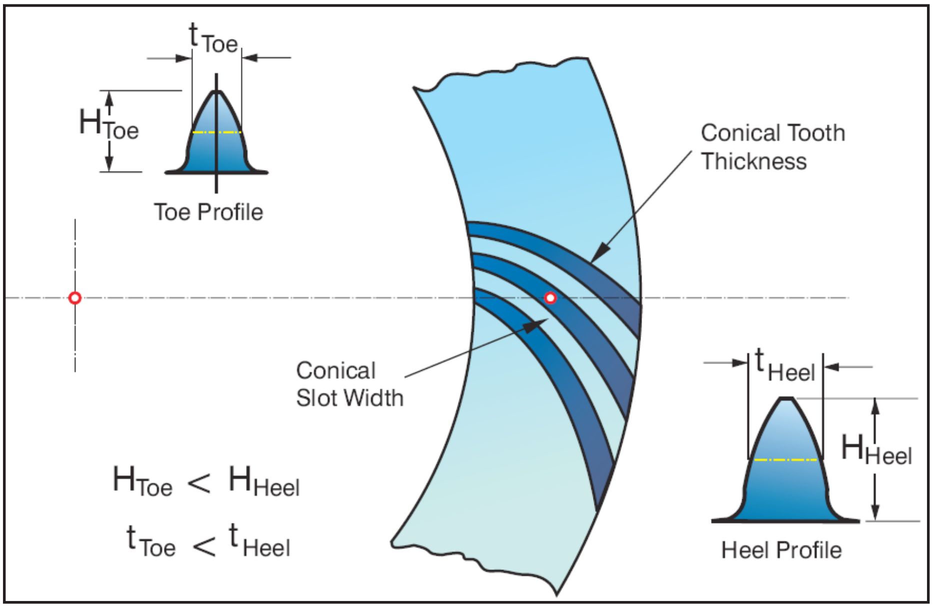


Figure 19 Tapered tooth depth change causes conical slot width along the pitch line (face milling).

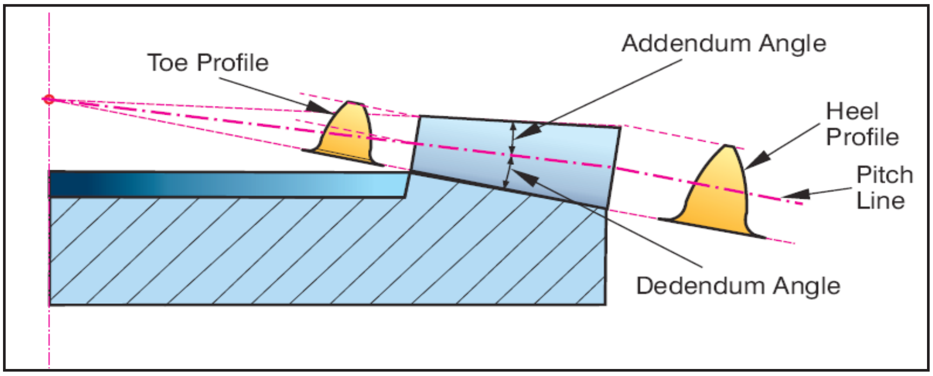


Figure 20 Tapered tooth depth change by addendum and dedendum angle.

tooth (Fig. 20). The tapered depth tooth has a number of advantages based on the original idea of the spherical involute. The tooth depths and the tooth profiles have proportions connected to the distance from the gear axes. The phenomenon known as undercut (left tooth profile, Fig. 16) is virtually eliminated or reduced.

However, the generation of bevel gears with tapered depth teeth causes conflict between the desired generating gear axis and the practical possible generating gear axis orientation. The methods E, F, G and H present different solutions for this conflict which are compared based on their kinematic coupling conditions.

Graphic "E" (Fig. 21) would require a horizontally oriented generating gear plane, which is perpendicular to the presentation plane and includes the pitch line. The employed machine design allows the tilting of the cutter head about $_K$ into the root line direction only in connection with a generating gear orientation — which is also parallel to the root line. The results are two non-matching generating gear axes for pinion and gear.

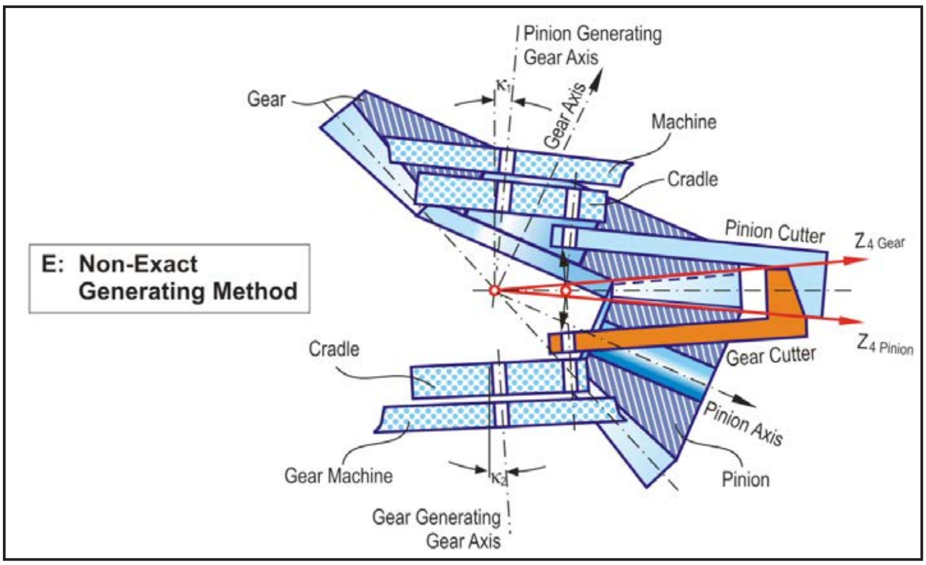


Figure 21 **Generating model for bevel gears with tapered depth teeth — Method E, octoid of the second order.**

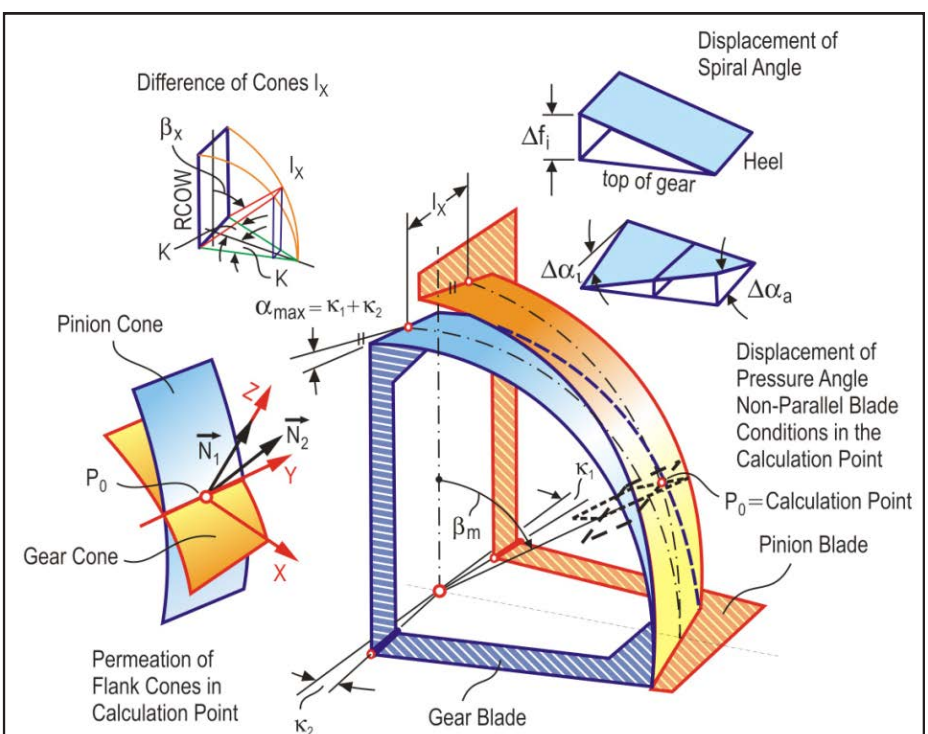


Figure 22 **Blade cone element deviation in case of different axes of rotation.**

Although both cutting edges match at the calculation point, the cone elements generated by the pinion and gear cutter deviate from each other due to a cutter axis orientation difference of $_{K1} + _{K2}$ (Fig. 22; Ref. 5). The kinematic coupling requirements 1 and 2 are not satisfied, whereas coupling requirement 3 is only slightly violated. Method "E" exists as a production process with and without a hypoid offset. The profiles of the resulting non-conjugate flank forms are octoids of the second order. The flank form deviations of method "E" are a maximum compared to the other methods discussed in this chapter. With the configuration of method "F" (Fig. 23) the attempt is made to keep the systematic errors as small as possible (Refs. 6–7). In spite of the collinear generating gear axes, both cutter heads are tilted about the angles $_{K1} + _{K2}$ in order for the blade tips to follow the root lines of the work gears. Coupling requirement 2 is fulfilled, the generating gear axes are identical, and the cutter cone elements match perfectly in the area of the calculation point. However, the cutter head tilt creates two slightly internal conical generating gears, which is why the conical generating tooth surfaces increasingly deviate with increasing distance from the calculation point. Coupling conditions 1 and 3 are not precisely fulfilled. The generated profile form is consistent with an octoid of the first order. Method "F" creates small flank form deviations that consist mostly of profile crowning.

Arrangement "G" (Fig. 24) shows the form cutting of a ring gear and the generating of a pinion with a tilted cutter head. The tilt angle $_{K1}$ is equal to the root angle $_{K1}$ of the pinion (in case of a gear box shaft angle of 90°). Although the two cutting edges match in the calculation point, the generating gear flank cone elements are deviating from each other with distance from the calculation point. Coupling requirement 2 is not satisfied, while the coupling requirements 1 and 3 are fulfilled.

By applying the artifice in Figure 25, a nearly exact bevel gear pair is created in spite of the tapered depth teeth and the plain generating gears (Method "H"). The crossing angle of the generating gear axes is like in case of method "E," or the sum of the dedendum angles. The particular artifice bases on the choice of curved blades whose radii originate in the intersecting point of the two correctly oriented cutter head axes. The result is a spherical generating gear flank surface which is perfectly congruent in the calculation point. The two surfaces of engagement intersect in this roll position along the center contact line. The coupling requirements 1 and 3 are fulfilled at the calculation point. Moving from the roll position that includes the calculation point will however, show differences in the surfaces of engagement and misalignment of the spherical generating flanks because the intersecting point of the cutter head axis is shifting during a generating cradle rotation. Eventually, none of the kinematic coupling requirements is fulfilled any longer.

The roll quality of uncorrected gearsets manufactured with method "H" (Gleason UNITOOL) is similar to the roll quality of gearsets manufactured with Method "F," but Method "H" can be performed on a less complex machine tool.

Bevel gears with tapered depth teeth present a number of advantages that are based on the balanced tooth cross-sections between heel and toe. Their manufacturing is limited until today, to face milled bevel gear sets. The reason for this is that changes in tooth thickness (i.e., slot width along the face width) cannot be compensated with a face hobbing process.

Already in the 1920s, Gleason developed mathematics for first- and second-order flank modifications via geometrical and kinematical corrections in cutting machines. These corrections made it possible to compensate flank form errors and additionally allowed the application of crowning to the flank surfaces. Crowning is necessary to avoid edge contact between the pinion and gear flanks in case of load-inflicted deformations and manufacturing tolerances.

Today's Phoenix free-form bevel gear cutting machines use a combination of cutter head tilt and helical motion (axial shifting of the generating gear during roll rotation) in order to manufacture bevel gears with tapered depth teeth and conical slot width while using a face milling completing process (Fig.20). With this technology the rolling quality of bevel gears with tapered depth teeth (cut in a single indexing process) is comparable

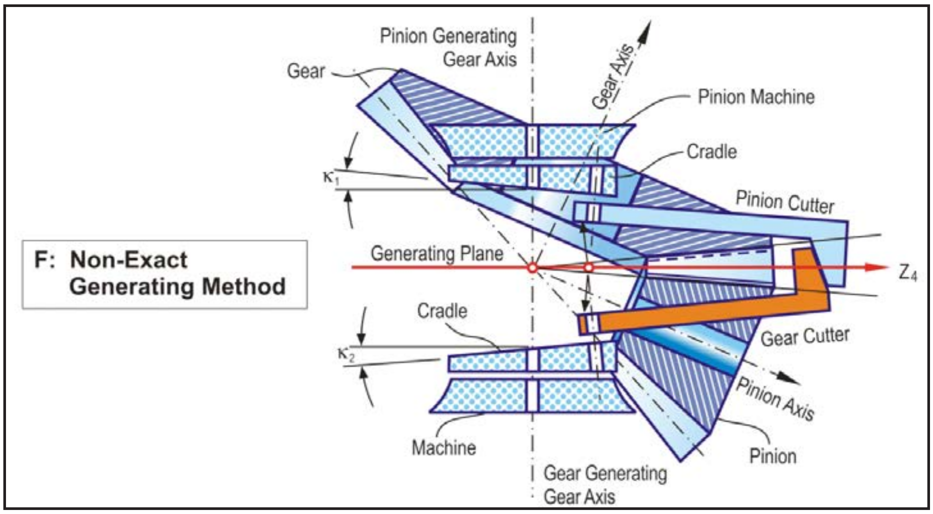


Figure 23 Generating model for bevel gears with tapered depth teeth — Method F, octoid of the first order.

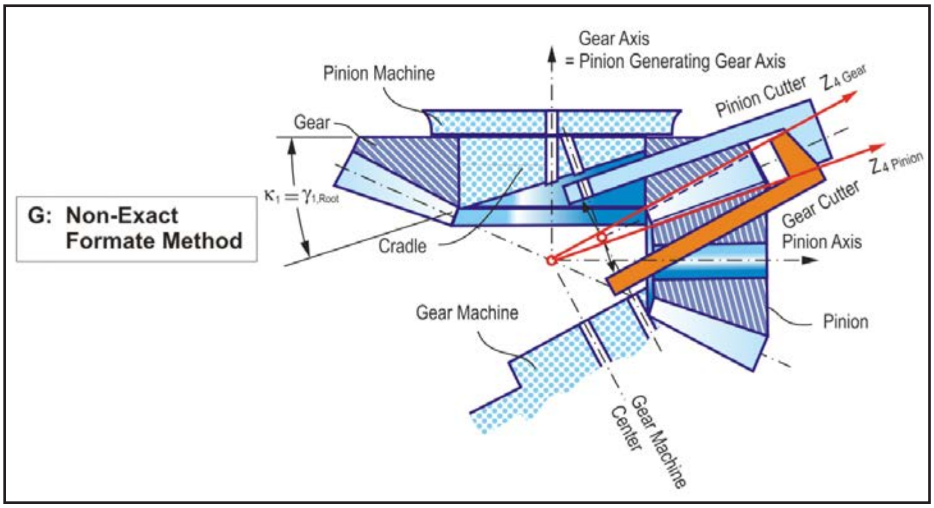


Figure 24 Generating model for bevel gears with tapered depth teeth — Method G, formate.

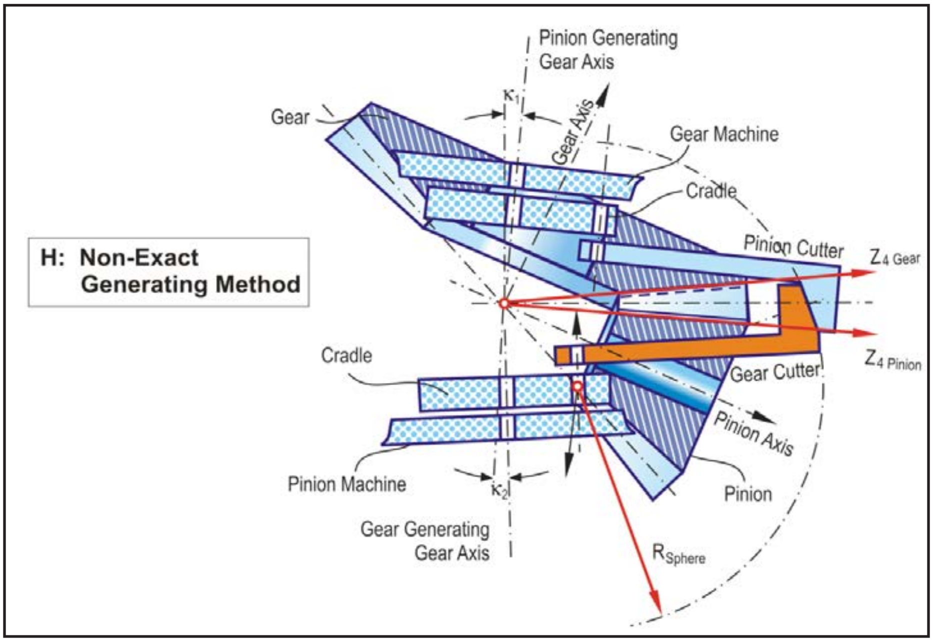


Figure 25 Generating model for bevel gears with tapered depth teeth — Method H, spherical flanks.

with the rolling quality of bevel gears with parallel depth teeth (cut in a continuous cutting process). Also, the cutting times of the two methods with modern machines and tools are basically identical.

A further advantage of the single indexing (face milling) method lies in the possibilities for hard finishing after soft cutting and heat treatment. The flank lines of face milled bevel gears are circular arcs, which make it possible to use grinding (not only lapping) as a hard finishing process. A suitable grinding wheel duplicates the silhouette of the cutting edges in a cutter head (stock allowance taken into account). The grinding wheel profile is basically dressed like the profile at the right side in Figure 17. The crossed profiles required in the continuous cutting process (face hobbing; left, Fig. 17) make it clear that it is physically impossible to dress those profiles onto a suitable grinding wheel.

Summary

- At the beginning of this chapter some thoughts about plausible explanations of the gearing law were discussed.
- Involute gearing was then presented as the consequential result of the engineering demand for a robustly functioning, easy-to-manufacture tooth form.
- A simplified explanation of the analogy between the cylindrical gear and bevel gear generating principle helps clarify things in making the bevel gear generating methods easier to understand. Based on this general understanding garnered at this point, a closer relationship of how the different bevel and hypoid gear generating methods are conducted is developed.
- The chapter continues to a deeper comprehension of the theory and understanding the pros and cons of the different methods.
- There is an acknowledgement that face hobbed bevel gears always feature parallel depth teeth and are not suitable for grinding due to their flank form and tooth thickness taper.
- Hard finishing of face hobbed bevel gears is generally done by lapping. In cases of smaller batches, a skiving with coated carbide blades is also possible.
- The goal with regards to face milled bevel gears was to convey the knowledge that they have, with only some unimportant exceptions, a tapered tooth depth form. It is possible to grind face milled gears very precisely and efficiently based on their tooth depth taper and circular flank lines. Lapping as well as skiving of face milled bevel gears are today's only exceptions — which are not often applied.

References

1. Buckingham, E. *Spur Gears*, McGraw-Hill Book Company, Inc., New York and London 1928.
2. Niemann, G. and H. Winter. *Maschinenelemente I, II & III*, Springer-Verlag Berlin, Heidelberg, New York, Tokyo, 1983.
3. Dudley, D. *Dudley's Gear Handbook*, McGraw-Hill, Inc. New York 1991.
4. Schriefer, H. "Verzahnungsgeometrie und Laufverhalten bogenverzahnter Kegelradgetriebe," Dissertation, RWTH Aachen, 1983.
5. Stadtfeld, H.J. "Anforderungsgerechte Auslegung bogenverzahnter Kegel-Radgetriebe," Dissertation, RWTH Aachen, 1987.
6. Brandner, G. "Kreisbogenverzahnte Kegelräder," *Maschinenbautechnik, 3.* Jg. Issue 5 May 1954.
7. Richter, E.H. "Geometrische Grundlagen der Kreisbogenverzahnung und ihre Herstellung," *Konstruktion*, Issue 3, 1958.

Dr. Hermann J. Stadtfeld received in 1978 his B.S. and in 1982 his M.S. degrees in mechanical engineering at the Technical University in Aachen, Germany; upon receiving his Doctorate, he remained as a research scientist at the University's Machine Tool Laboratory. In 1987, he accepted the position of head of engineering and R&D of the Bevel Gear Machine Tool Division of Oerlikon Buehrle AG in Zurich and, in 1992, returned to academia as visiting professor at the Rochester Institute of Technology. Dr. Stadtfeld returned to the commercial workplace in 1994 — joining The Gleason Works — also in Rochester — first as director of R&D, and, in 1996, as vice president R&D. During a three-year hiatus (2002–2005) from Gleason, he established a gear research company in Germany while simultaneously accepting a professorship to teach gear technology courses at the University of Ilmenau. Stadtfeld subsequently returned to the Gleason Corporation in 2005, where he currently holds the position of vice president, bevel gear technology and R&D. A prolific author (and frequent contributor to Gear Technology), Dr. Stadtfeld has published more than 200 technical papers and 10 books on bevel gear technology; he also controls more than 50 international patents on gear design, gear process, tools and machinery.

Elastohydrodynamic Lubrication (EHL): A Review

Robert Errichello

Introduction

This review of elastohydrodynamic lubrication (EHL) was derived from many excellent sources (Refs. 1–5). The review of Blok's flash temperature theory was derived from his publications (Refs. 6–9). An excellent general reference on all aspects of tribology is the *Encyclopedia of Tribology* (Ref. 10).

Gear teeth, rolling element bearings, cams, and other non-conforming Hertzian contacts are lubricated by the EHL mechanism.

Figure 1 is an enlarged view of a lubricated Hertzian contact. It shows the distribution of film pressure and film thickness between two steel cylinders in rolling contact. Note that Figure 1 exaggerates the vertical distance and shrinks the horizontal distance for purposes of illustration. An actual EHL contact is typically 1,000 times wider than the film thickness. The EHL contact starts with a slowly converging inlet region where the lubricant is entrained and hydrodynamic pressure is generated. The film pressure gradually increases in the inlet region until it reaches the leading edge of the Hertzian region where the pressure quickly builds to values that are essentially equal to the Hertzian contact stress. Under high pressure, the lubricant viscosity increases exponentially to the extent that the lubricant cannot escape because its viscosity is too high. Within the Hertzian region, the bodies are separated by a constant film thickness. At the end of the Hertzian region there is a constriction near the outlet that forms the minimum film thickness. Within the Hertzian region the film pressure follows the Hertzian pressure, except for a sharp spike in pressure just upstream from the constriction at the outlet. Within the constriction, the film pressure drops rapidly to atmospheric pressure.

The inlet region. The lubricant that is adsorbed on the surfaces of the contacting bodies is entrained into the EHL contact by the rolling motion of the bodies. Entrainment of the lubricant is greatly facilitated by its viscosity increase because the high viscosity resists flow, makes it more difficult to squeeze the lubricant out, and viscous drag forces cause it to move with the surfaces into the Hertzian region. As a result, the inlet pumps the film up to a thickness that is sufficient to separate the opposing bodies.

EHL film thickness is determined by the viscosity and pressure-viscosity coefficient of the lubricant in the inlet region. For gears, the lubricant that is entrained into the inlet is molecularly attached to the surfaces of the pinion and wheel teeth and consists of thin boundary layers that immediately take on the bulk surface temperatures of the pinion and wheel teeth. Consequently, EHL film thickness is determined by the equilibrium bulk surface temperatures of the pinion and wheel teeth in the inlet region before the lubricant reaches the Hertzian region.

Bulk surface temperature. When the pinion and wheel are running under a given load, the surfaces of the pinion and wheel teeth are heated by the sliding friction between the gear teeth and gradually increase in temperature until finally reaching the equilibrium bulk surface temperatures after many revolutions.

Inlet shear heating. In a fully flooded EHL contact, only a fraction of the lubricant can pass through the contact. Therefore, some of the lubricant is rejected and reverse flow occurs in the inlet. Furthermore, if there is sliding in addition to rolling, heat is generated by shearing of the lubricant. Churning and shearing generate heat that increases the lubricant temperature above the average bulk surface temperatures. Therefore, the temperature that controls lubricant viscosity and EHL film thickness is the temperature of the lubricant in the inlet. Empirical equations are available to correct calculations of isothermal EHL film thickness to account for inlet shear heating.

Starvation. To form full EHL film thickness, the inlet region must be fully flooded. However, the inlet region might be starved of lubricant if the lubricant supply is inadequate, if very high speed causes fling off of the lubricant, or both. Under these conditions, EHL film thickness might be reduced. Empirical equations are available to correct calculations of isothermal EHL film thickness to account for lubricant starvation.

The Hertzian region. By the time the lubricant enters the Hertzian region, its viscosity has increased by a factor of 1,000 and it is trapped in the contact. At maximum Hertzian pressures typical in gears and rolling element bearings, the lubricant undergoes a phase transition into a solid glassy state. From this point on the lubricant no longer behaves as a Newtonian fluid, and it can be considered a pseudo-solid. In the Hertzian region the surfaces of the bodies are parallel and separated by the central film thickness that has an essentially constant thickness. The film within the Hertzian region is extremely stiff; therefore if the load increases, the bodies deform more than the central film thickness decreases. Consequently, EHL contacts are relatively insensitive to changes in load and the main effect of increasing load is to deform the surfaces, which increases the area of the Hertzian region, but does little to alter the shape of the inlet zone where the EHL film is formed. Bottom line, increasing load leaves the film thickness virtually unchanged.

Sliding within the Hertzian region. Sliding friction within the EHL film increases the bulk temperature of the gear teeth from a cold start by accumulating heat from each tooth engagement. The bulk temperature of the gear teeth increases until the heat input is equal to the heat loss to the surroundings. Once the bulk temperature reaches equilibrium, there is no further change in gear tooth bulk surface temperature — unless the operating conditions change. The heat input is confined to the immediate area of the Hertzian region and its duration is only a fraction of a millisecond long. Consequently, the heat produced by frictional heating within the EHL film is removed by conduction through the film into the tooth surfaces, and by convection as the hot oil exits the outlet region. Due to the short contact time the heat penetrates only a shallow distance into the gear teeth and is rapidly dissipated. Consequently, as the contact point moves on, the heat input disappears immediately and the surface temperature of the gear teeth returns promptly to the equilibrium bulk temperature. After one revolution of the gear, a particular point on the gear flank comes into engagement with essentially the same bulk temperature as the previous engagement. Although frictional heating does not directly alter the film thickness within the Hertzian region, any increase of the bulk surface temperatures due to frictional heating indirectly reduces film thickness by decreasing the viscosity of the lubricant in the inlet region.

The sliding is significant because it generates traction forces that result in energy losses. If the lubricant behaved like a Newtonian fluid, the high viscosity would lead to extremely high traction force. Fortunately, however, when subjected to high shear stresses the lubricant behaves like a plastic pseudo-solid with limited shear strength that is characterized by its traction coefficient. The bulk surface temperatures are controlled by heat generated in the Hertzian region and

As the lubricant passes through the three regions, it viscosity increases exponentially, and the lubricant changes phase from a freely flowing fluid in the inlet region, to a pseudo-solid within the Hertzian region, and back to a freely flowing fluid in the outlet region — all within a matter of milliseconds.

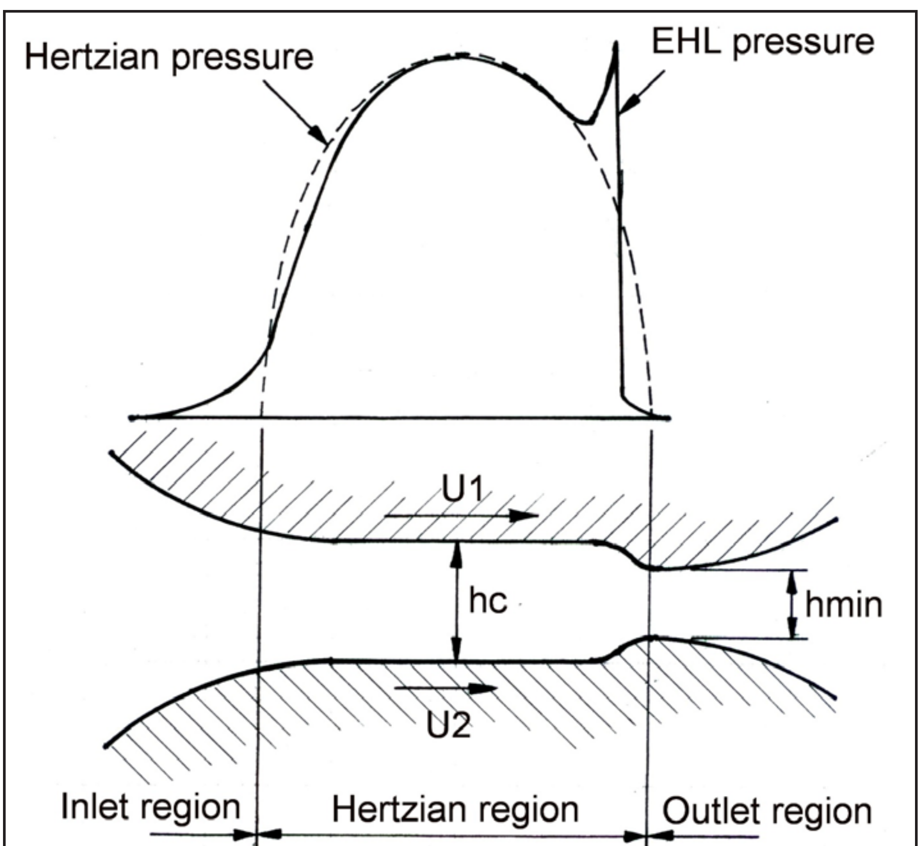


Figure 1 Enlarged view of a lubricated Hertzian contact. Note distribution of film pressure and film thickness between two steel cylinders in rolling contact. Also note exaggeration of the vertical distance and shrinkage of horizontal distance for purposes of illustration. An actual EHL contact is typically 1,000 times wider than the film thickness.

the temperatures can vary significantly, depending on the molecular structure of the lubricant base stock, which influences a lubricant's solidification pressure, shear strength, and traction coefficient. Furthermore, depending on anti-wear and anti-scuff additives that may be in the lubricant, the sliding and heat generate boundary tribofilms that help to prevent adhesive wear.

Shear thinning. Lubricants containing high molecular weight polymers, which are additives known as a viscosity index (VI) improvers, may lose viscosity under the high shear rates that occur in the Hertzian region and reduce the EHL film thickness. This is known as shear thinning.

The outlet region. As the lubricant leaves the Hertzian zone, film pressure tends to boost the lubricant flow toward the outlet region. The amount of lubricant within the contact is controlled by the inlet and continuity of flow can only be maintained if there is a local restriction in the outflow, which causes a constriction to form at the outlet. This is the position where the minimum film thickness occurs. A sharp spike is generated in the film pressure next to the constriction on the upstream side. The pressure drops abruptly to atmospheric pressure downstream of the spike and the lubricant viscosity returns to its atmospheric viscosity. Consequently, the contact pressure between the surfaces is negligible in the area of the minimum film thickness. The divergent region of the outlet generates negative pressure that causes dissolved gases in the lubricant to come out of solution. This ruptures the lubricant film, which cavitates and forms a wavy wake that consists of separate lubricant streamers intermixed with air.

Significance of the three regions. The inlet region pumps the film up, the Hertzian region rides the film, and the outlet region discharges it. As the lubricant passes through the three regions, it viscosity increases exponentially, and the lubricant changes phase from a freely flowing fluid in the inlet region, to a pseudo-solid within the Hertzian region, and back to a freely flowing fluid in the outlet region — all within a matter of milliseconds. Lubricant emerging from the wake of the outlet region is indistinguishable from lubricant that entered the inlet region. The film forming capability of the hydrodynamic pressure generated in the inlet region is governed by the local viscosity of the lubricant in the inlet, which is controlled by the bulk temperatures of the surfaces and inlet shear heating. Consequently, the central film thickness is established by the lubricant properties in the inlet region. Once in the Hertzian region, the lubricant acts as a pseudo-solid where it influences the traction coefficient, boundary tribofilms, and bulk surface temperatures. Finally the outlet region adjusts the minimum film thickness to maintain continuity of lubricant flow.

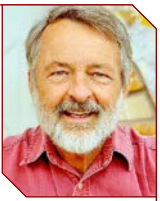
Blok's flash temperature theory. In 1937 Blok published his flash temperature theory in a series of papers (Refs. 6–8). Blok defined the contact temperature as the sum of the bulk surface temperatures of the gear teeth, and the flash temperature rise associated with frictional heating in regions of asperity contact. Blok in 1969 published the thermal-network method (Ref. 9) for calculating the equilibrium bulk temperatures of gear teeth. Blok's theory of scuffing proposes that scuffing occurs when the maximum value of the contact temperature reaches a critical temperature. He predicted the surface temperature based on the following assumptions:

1. The surfaces are in intimate contact or perfectly insulated
2. All the heat is removed by one-dimensional conduction straight down into the surfaces
3. The two bulk temperatures are identical

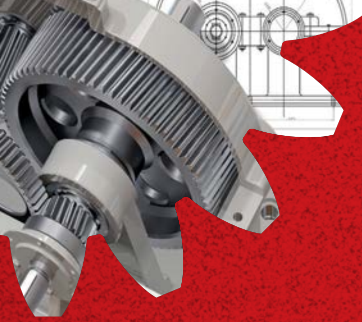
Assumption (1) is violated if an EHL film is present; assumption (2) is violated if the speed of either surface is too slow; and assumption (3) is violated for a high gear ratio because the pinion typically runs hotter than the wheel. Therefore, Blok's flash temperature theory applies only to the boundary lubrication regime where the EHL film is non-existent, and the only protection against scuffing is any tribofilm deposited by lubricant additives. But once the tribofilms fail, the lone remaining protection is the natural oxide layer on the gear teeth. Consequently, Blok's flash temperature is not applicable to the mixed-film or full EHL regime.

References

1. Tallian, T.E. "Elastohydrodynamic Hertzian Contacts," *Mechanical Engineering, Part 1*, 1971, Nov, pp 14–18, *Part: 2*, Dec, pp 17–22.
2. Wedeven, L.D. "What is EHD?" *ASLE Lubrication Engineering*, 1975, 31, 6, pp. 291–296.
3. Mobil EHL Guidebook. "Elastohydrodynamic Lubrication (EHL)," 1992, pp 1–13.
4. Olver, A.V. "Gear Lubrication: A Review," *J. Engr. Tribology, Proc. Inst. Mech. Engrs.*, 216, 2002, J, pp 255–267.
5. Larsson, R. "EHL Film Thickness Behavior," *Encyclopedia of Tribology, Vol 2*, 2013, pp 817–827.
6. Blok, H. "Les Temperatures de Surface Dans des Conditions de Graissage Sous Pression Conditions," *Proceedings of Second World Petroleum Congress* (Paris), 4, 1937 a, pp 151–182.
7. Blok, H. (1937b) "Measurement of temperature flashes on gear teeth under extreme pressure conditions," *Inst. Mech. Engrs, Proceedings of the General Discussion on Lubrication and Lubricants*, 2, 1937 b. pp 14–20.
8. Blok, H. "Theoretical study of temperature rise at surfaces of actual contact under oiliness lubricating conditions," *Proceedings of the General Discussion on Lubrication and Lubricants*, 2, 1937 c, pp 222–235.
9. Blok, H. "The Thermal-Network Method for Predicting Bulk Temperatures in Gear Transmissions," *Proc. 7th Round-Table Discussion on Marine Reduction Gears*, 1969, Stal-Laval, Finspong, Sweden, pp 3–25.
10. Wang Q.P. and Y.W. Chung. (Eds.) Springer Reference, *Encyclopedia of Tribology*, Volumes 1–6.



Robert Errichello, PE, heads his own gear consulting firm—GEARTECH—and is a founder of GEARTECH Software, Inc. A graduate of the University of California at Berkeley, he holds B.S. and M.S. degrees in mechanical engineering and a master of engineering degree in structural dynamics. In his more than 30 years of industrial experience, Errichello worked for several gear companies; he has also been a consultant to the gear industry for more than 20 years and has taught courses in material science, fracture mechanics, vibration and machine design at San Francisco State University and the University of California at Berkeley. He is also a member of ASM International, STLE, ASME Power Transmission and Gearing Committee, AGMA Gear Rating Committee and the AGMA/AWEA Wind Turbine Committee. Errichello has published dozens of articles on design, analysis and the application of gears, and is the author of three widely used computer programs for the design and analysis of gears. He is also a longtime technical editor for Gear Technology magazine and STLE Tribology Transactions, and has presented numerous seminars on design, analysis, lubrication and failure analysis of gears. Errichello is a past recipient of the AGMA TDEC award and the STLE Wilbur Deutch Memorial award.



GEAR EXPO 2015

THE DRIVE TECHNOLOGY SHOW

POWERFUL CONNECTIONS

OCTOBER 20-22, 2015
DETROIT, MI
COBO CENTER

REGISTER TODAY

FOR THE DRIVE TECHNOLOGY SHOW OF THE YEAR!

GEAR EXPO 2015 brings together the full range of experts — design, manufacturing, application engineering — all in the same place — **Cobo Center, Detroit**, **October 20–22.**

From a single gear to complex drives, **GEAR EXPO** covers it all...

- **More exhibitors** — from gear making machines to systems integrators.
- **More education sessions** — from metallurgy and geometry to maximizing power density.
- **More attendees** — buyers and makers from automotive, construction, mining, agriculture — wherever high-performance drives are critical.
- **More solutions** — meet the experts and evaluate your capabilities.

GEAR EXPO 2015 is the place to be.

- Make powerful connections.
- See the latest technology in action.
- Get the inspiration you need to take your products to the next level.

Improved drive technology is critical to achieving higher efficiencies and longer service life. All the information and answers you need will be at **GEAR EXPO 2015**.

VISIT

www.gearexpo.com/GETECH

Use code **GETECH** when registering.

Impact of Indexing Errors on Spur Gear Dynamics

Murat Inalpolat, Michael Handschuh and Ahmet Kahraman

A transverse-torsional dynamic model of a spur gear pair is employed to investigate the influence of gear tooth indexing errors on the dynamic response. With measured long-period quasi-static transmission error time traces as the primary excitation, the model predicts frequency-domain dynamic mesh force and dynamic transmission error spectra. The dynamic responses due to both deterministic and random tooth indexing errors are predicted.

Introduction

Every manufactured gear contains certain types and magnitudes of errors depending on the quality level imposed. Such errors often contribute to the loaded transmission error to affect the meshing dynamics of gears. Consequently, understanding the impact of different gear design and manufacturing based errors and tolerances on the dynamic transmission error of gears is crucial. One of the most significant contributors to the gear transmission error is the tooth indexing errors. Gear tooth indexing error (deviation) is defined as the displacement of any tooth flank from its theoretical position relative to a reference tooth flank (Ref. 1). In relation to it, tooth spacing error is defined as the circumferential position error of one gear tooth flank with respect to the previous tooth flank. Ideally, a particular gear with Z number of teeth has identical involute profiles equally spaced around the pitch diameter. Existence of indexing error means that some of the tooth profiles are angularly misplaced from their ideal position with respect to one randomly chosen reference profile (index tooth or profile), say Tooth-1 without loss of generality. The right-hand side flank of Tooth-1 is the reference profile (flank) when certain amount of torque acting in the clockwise direction is assumed to exist on this gear. The circular distances, S1 and S2, between the right flanks of Tooth-1 and Tooth-2, and also between Tooth-2 and Tooth-3, where both flanks intersect the reference diameter, are both equal to a circular pitch p ($p=\pi m$, where m is the module) for a gear with ideal geometry. If S1 deviates from the nominal circular pitch p ($S1 \neq p$), then the difference is interpreted as the spacing error ε_1 for Tooth-2. Similarly, if S2 has a different value than p, then it is interpreted as the spacing error ε_2 for Tooth-3. The value $\varepsilon_1+\varepsilon_2$ will then be the indexing error for Tooth-3. If spacing error of any Tooth-N is $\varepsilon_{N\text{-}1}$ on a gear, then the corresponding indexing error for Tooth-N will be $\Sigma_{j=1}^{N\text{-}1}$, where j is the indexing error index.

Gear tooth indexing errors arise during manufacturing, causing deviations related to the cutting or heat treatment process in addition to the random components (Ref. 2). Indexing errors modify the transmission error as they cause a certain gear tooth profile to be misplaced on the reference diameter, thus either coming into contact earlier or later with the corresponding tooth on the other gear in mesh compared to its expected timing under ideal conditions. This essentially shifts the contact in time that can significantly change the dynamic behavior of the gears, as the dynamic excitation phase continuously changes and instantaneous contact ratio becomes lower or higher than expected at different times, causing either overloading or contact loss of the tooth in mesh. Consequently, complicated indexing error patterns that interact with each other on gears in mesh could significantly alter the resultant life of gears under operation. The frequency spectra for the gears with indexing error show significant increase to the non-harmonic orders, making the spectra broad-band (Ref. 3).

The main reason for these non-harmonic orders to exist is the non-periodic transmission error values due to spacing/indexing errors. Therefore, it is not sufficient to use limited Fourier series amplitudes of transmission error to simulate the meshing dynamics anymore. The proper means of simulating indexing errors would be to apply the errors over multiple revolutions of both pinion and gear, covering their total hunting period. Worst-case spacing errors occur when the respective errors of the pinion and gear match up.

The published work on the effects of indexing errors on gear dynamics is rather sparse. Remmers (Ref. 2) developed an analytical method to study the effect of tooth spacing errors, load, contact ratio and profile modifications on the gear mesh excitations. He indicated that random tooth spacing errors may be used to reduce the gear mesh excitations at certain frequencies. Mark (Refs. 3–4) derived expressions for Fourier series coefficients of all components of static transmission error, including harmonic and non-harmonic coefficients of gear defects of concern. He used two-dimensional Fourier transforms of local tooth pair stiffness and tooth surface deviations from perfect involute to come up with these expressions and used them to study mesh transfer functions of gears with different surface and profile deviations. Kohler and Regan and later Mark (Refs. 5–7) discussed components of the frequency spectrum for gears with pitch errors based on analytical approaches and agreed on the fact that existence of the components depends on loading conditions, and if the only deviation from perfect tooth geometry is due to pitch errors, then frequency spectrum of corresponding transmission error function will have no components at the mesh frequency harmonics. Padmasolala et al. (Ref. 8) developed a model to understand the effectiveness of profile modification for reducing dynamic loads in gears with different tooth

This paper was originally presented at the 2014 International Gear Conference, Lyon Villeurbanne, France and is republished here with the authors' permission.

spacing errors. They showed that linear tip relief is more effective in reducing dynamic loads on gears with relatively small spacing errors, whereas parabolic tip relief becomes more effective when the amplitude of the spacing error increases. Wijaya (Ref. 9) studied the effects of spacing errors and run-out on the dynamic load factor and the dynamic root stress factor of an idler gear system. He employed an analytical approach that defines the static transmission error and static tooth forces and predicted dynamic mesh force spectra of an idler gear system using a linear, time-invariant model. Spitas and Spitas (Ref. 10) also investigated overloading of gears and effect of tip relief on the dynamics of gears with indexing errors. They employed a geometry-based meshing analysis with a multi-degree-of-freedom dynamic model and reported simulated load factors and transmission error functions for gears with assumed indexing errors. Milliren (Ref. 11) and Handschuh (Ref. 12) investigated the influence of various gear errors on the quasi-static transmission errors and root stresses of spur gears experimentally. They used the same test rig to investigate the effects, such as spacing errors and lead wobble on the transmission error. Moreover, they compared experimental results with the results of a contact model and showed that the results are highly correlated.

In this study, the impact of indexing errors on dynamics of spur gears is investigated. A test setup and its encoder-based measurement system are used to measure loaded transmission error excitations. A dynamic model capable of including long-period transmission error excitations is proposed to demonstrate the effect of indexing errors on the resultant dynamics of gears.

Dynamic Model Formulation

In this study, impact of indexing errors on spur gear dynamics is demonstrated via a simplified lumped parameter dynamic model, as shown in Figure 1. It is a 6 degree-of-freedom (DOF) dynamic model that assumes both gears can translate in x and y coordinates and also can rotate about their own local rotational axis represented by the *z* coordinates. However, the line of action (LOA) is selected such that it is coincident with the global *x*-axis of the inertial frame which uncouples the gear dynamics observed in the *y* directions for both gears from the rest. Consequently, the model can essentially be seen as a 4 DOF gear pair model that represents the LOA and torsional dynamics, as

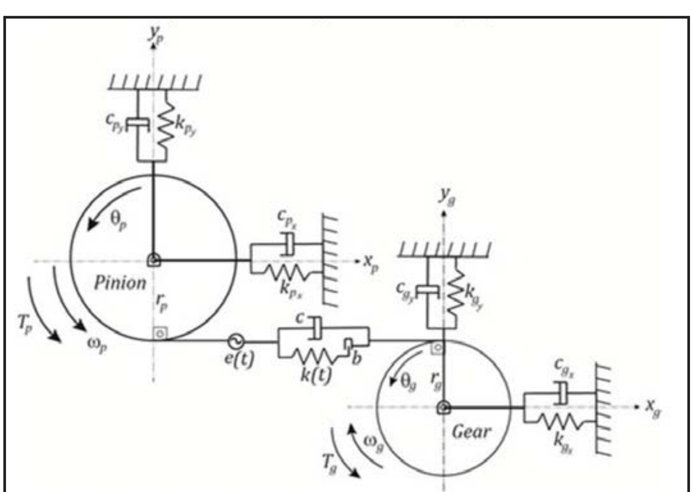


Figure 1 The discrete dynamic model.

shown by several other researchers to be a valid model when not including the effects of friction (Refs. 13–15).

The proposed model can accept broadband quasi-static transmission error excitations both in the form of long-period time domain input and also in the form of multi-harmonic frequency domain input. This is shown to be superior when certain time-varying and nonlinear effects act on the dynamics of the gear pair simultaneously, such as the cyclo-stationary, quasi-static transmission error.

In the lumped parameter dynamic model developed here, gears are represented by rigid disks which are connected to each other through a number of elements. A spring element is used to represent gear mesh flexibility $k(t)$, which is a time-varying function that is evaluated at different input torque levels by using a contact model (Ref. 16). Here, $k(t)$ includes the parametric excitations due to the mesh stiffness variation caused by the fluctuation of number of teeth in contact into account. A clearance element b accompanies the mesh stiffness function to account for tooth separations caused by backlash or other tooth profile deviations. A viscous damper element c is employed to represent the gear mesh energy losses. A displacement element $e(t)$ is used to represent the quasi-static transmission error that acts along the line of action. The quasi-static transmission error function is also load-dependent and thus should be evaluated at each different input torque level.

Each disk that represents a gear body has a mass of m_i, a mass moment of inertia of J_i and a base radius of r_i, where $i = p$ or g. Gear bodies are supported by stiffnesses k_{ix}, k_{iy}, and also by dampers c_{ix}, c_{iy}, which represent the combined elasticity and damping of supporting bearings and shafts that carry the gears. The equations of motion are given below where x_i, y_i are the translational, and θ_i are the rotational degrees-of-freedom for the gear bodies:

$$m_p\ddot{x}_p(t) + c\delta(t) + k(t)\,\delta(t)\,g(\delta(t)) + c_{p_x}\dot{x}_p(t) + k_{p_x}x_p(t) = 0 \tag{1a}$$

$$m_p\ddot{y}_p(t) + c_{p_y}\dot{y}_p(t) + k_{p_y}y_p(t) = 0 \tag{1b}$$

$$J_p\ddot{\theta}_p(t) + cr_p\dot{\delta}(t) + k(t)r_p\delta(t)g(\delta(t)) = T_p(t) \tag{1c}$$

$$m_g\ddot{x}_g(t) - c\dot{\delta}(t) - k(t)\,\delta(t)\,g(\delta(t)) + c_{g_x}\dot{x}_g(t) + k_{g_x}x_g(t) = 0 \tag{1d}$$

$$m_g\ddot{y}_g(t) + c_{g_y}\dot{y}_g(t) + k_{g_y}y_g(t) = 0 \tag{1e}$$

$$J_g\ddot{\theta}_g(t) + cr_g\dot{\delta}(t) + k(t)r_g\delta(t)g(\delta(t)) = T_g(t) \tag{1f}$$

In Equation 1 a-f, δ_t represents the relative mesh deflections defined as:

$$\delta(t) = x_p(t) - x_g(t) + \theta_p(t)r_p + \theta_g(t)r_g - e(t) \tag{2}$$

and gear mesh contact loss is mathematically induced by the unit step function, $g(\delta(t))$. The unit step function, $g(\delta(t)) = 0$, if $\delta(t) < 0$ and $g(\delta(t)) = 1$, *if* $\delta(t) \geq 0$. An over-dot means a time derivative of the corresponding variable in the equations of motion. In Equation 2 $e(t)$ is the time-varying transmission error function that represents the motion errors caused by gear mesh deflections due to load and also due to manufacturing deviations. Transmission error is a periodic function and is simulated via discrete Fourier series amplitude and phase of this periodic function in modeling schemes that exist in literature (Refs. 16–17). The model can employ the transmission error function either in a time-series form or in a broad-band frequency domain form from a measured or a simulated time-series. In other words, this model makes it possible to use measured time-domain or broad-band frequency domain representation of the long-period quasi-static transmission error to predict dynamic transmission errors and dynamic mesh forces. Moreover, the dynamic model developed in this study accepts measured transmission error signal with any resolution, automatically synchronizing the time resolution of the transmission error signal with the time resolution of the dynamic model equations for the right solution. A simplified mathematical representation of such a signal in the time domain can be given as:

$$e(t) = \sum_{k=1}^{\infty} \sum_{i=1}^{\infty} A_i \sin\left(i \frac{\omega_m}{k} t + \varphi_i\right) \tag{3}$$

This transmission error function assumes an infinite number of harmonics, sub-harmonics and overtones of the mesh frequency ω_m that are superimposed to obtain a broadband transmission error function as in Equation 3. The quasi-static transmission error, $e(t)$, when measured from a gear set with indexing errors, is periodic over the complete rotation of the gears in mesh if the gears are unity-ratio gears. If two gears with different number of teeth are used, then transmission error signal is periodic over Z_1Z_2 rotations of both gears, where Z_1 and Z_2 represent the number of teeth on the pinion and the gear, respectively. On the contrary, for a gear set with no indexing errors, transmission error is periodic over a mesh cycle and over a complete rotation of the gear. This difference constitutes the main mechanism for the signals to convolute.

This dynamic model employs a number of assumptions. First of all, gear wheels are assumed to be rigid, with flexibility only coming from the gear mesh. Gear motions in some other directions, such as rotations about x and y, and translations in z, are excluded for the sake of simplicity. Bearings are assumed to be linear. Finally, simplified damping elements are used. The model presented here makes use of Rayleigh damping, where $[C] = \alpha [M] + \beta [K]$. Here, $[C]$, $[M]$ and $[K]$ represent the overall system damping, mass and stiffness matrices and αand β are constant coefficients, and $\alpha = 479$ and $\beta = 1.4 \cdot 10^{-7}$ was used here.

Experimental Transmission Error Measurements

Test machine and set-up. In this study, an open-architecture gearbox with drive and load capacity was used to measure the quasi-static transmission error of spur gears with different indexing errors. This rig (Fig. 2) is designed to operate gears under high-load and low-speed (loaded quasi-static) conditions. A small DC motor is connected to a 100:1 ratio harmonic drive to reduce the speed significantly while increasing the torque delivered to the gear pair. The harmonic drive was directly connected to a torque-meter to monitor the input torque provided to the gearbox. The output side of the torque-meter was connected to one of the test gear shafts via a flexible elastomer coupling. The test gearbox consisted of a unity-ratio spur gear pair held by relatively rigid shafts. Concentric bearing housings holding the bearings were mounted on two massive split pedestals. The shaft of the driven (output) gear was connected to a magnetic particle brake with a maximum torque capacity of 400 Nm by means of another flexible coupling. The shafting permitted optical rotary encoders to be mounted on their free ends outside the bearing pedestal for measurement of TE. The rotary encoders (Heidenhain, RON 287) have the capability to measure 18,000 positions per complete rotation.

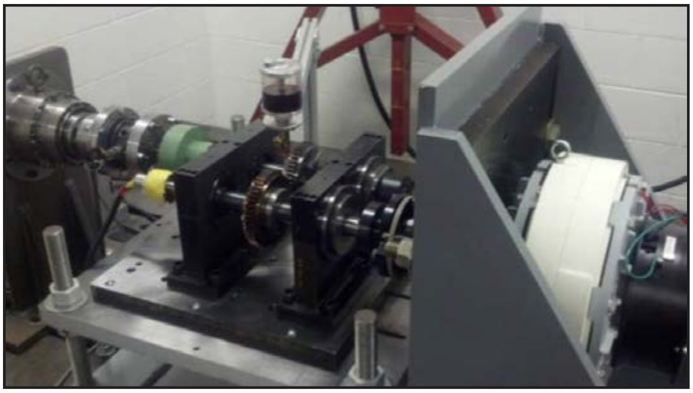


Figure 2 Quasi-static transmission error (TE) measurement set-up.

Test specimens and test matrix. Unity-ratio spur gear pairs were used in this study with an operating center distance of 150 mm. A total of 7 gears with tightly controlled indexing errors formed the inventory for the test matrix. Two gears with no indexing errors, three test gears with discrete spacing errors at a few teeth along with two additional gears with randomized spacing errors were procured specifically for this study. Basic gear parameters for these gears are listed in Table 1. Each gear pair had a theoretical contact ratio of 1.8. The test gears had the following intentional spacing errors:

- Gears #1, #2: Gears with no indexing errors (in reality, errors exist but less than 2 μm).
- Gear #3: Gear with a single tooth having a negative 15 μm spacing error.
- Gear #4: Gear with two consecutive teeth having negative 15 μm spacing errors.
- Gear #5: Gear with two teeth having negative 15 μm indexing errors with the tooth in between them in the correct position.
- Gears #6, #7: Gears with all teeth having spacing errors with randomly distributed magnitudes between the range [-15, 15] μm.

Table 1 Basic parameters of the gears used in this study

Parameter	Value
Number of Teeth	50
Normal Module [mm]	3
Pressure Angle [deg]	20
Pitch Diameter [mm]	150
Base Diameter [mm]	140.954
Major Diameter [mm]	156
Minor Diameter [mm]	140.68
Circular Tooth Thickness [mm]	4.64
Active Facewidth [mm]	20
Tip Relief Magnitude [mm]	0.01
Starting Roll Angle of Tip Relief [deg]	20.9

Table 2 Test matrix used in the test runs

Test #	Driver	Driven	Description
1	#2	#1	Baseline - No error
2	#3	#2	Gear with indexing error meshing with a no-error gear
3	#5	#2	Gear with indexing error meshing with a no-error gear
4 to 19	#7	#8	Gear with random spacing errors meshing in 16 different positions

Each error was placed in the negative direction, which refers to the tooth entering mesh later than expected. The negative direction of the errors was chosen because they were obtained by removing additional material from the teeth with no error. Likewise, if a tooth enters mesh earlier than nominal compared to the previous tooth, the spacing error is positive. Moreover, a more realistic case where two gears having various different indexing errors was studied utilizing the gears designated as Gears #6 and #7. These gears were machined to have random spacing errors within the range [-15, 15] μm. Furthermore, with right and left flanks having different error values, these two gears represented four different random spacing error conditions.

Transmission error measurements. The transmission error measurement system used two separate encoders to determine the angular position of each gear shaft. The signals from the encoders were passed into encoder conditioners. A commercially available transmission error measurement system was used to process the conditioned encoder signals to compute the transmission error. The software has high and low pass filtered TE time histories of the same data to quantify shaft and mesh frequency content of the TE signal. This system considers either the driving pinion (input) encoder or the driven gear (output) encoder as the reference signal to determine output and input TE signals (both in μrad), or as a linear displacement along the line of action as:

$$e(t) = r_p\theta_p(t) - r_g\theta_g(t) \tag{4}$$

These TE measurements are processed using data segments corresponding to 16 complete gear rotations. In Figure 3 a representative measured loaded $e(t)$ is presented from Test #2 at 200 Nm. In Figure 3 (a, b), the unfiltered (raw) and the high-pass filtered versions of the measured $e(t)$ is presented. Although the measurements were recorded for 16 complete gear rotations, only one complete rotation, including 50 gear mesh cycles, is presented in these figures. These representative measurements clearly exhibit the intentional indexing error amount generated on the test gears. With the gear rotational speed of 10 rpm, gear mesh frequency is 8.33 Hz. Accordingly, the measured TE signal is put through a high-pass filter with a cut-off frequency slightly higher than 0.167 Hz, removing frequencies equivalent to one shaft order frequency and below. A family of baseline tests of the two no-error gears was run first at all of the operating conditions specified. The sinusoidal waviness in Figure 3(a) is the once-per-revolution amplitude caused by the pitch line run-out error of the gear pair. Once such low-frequency content is removed by the high-pass filter, the remaining TE is seen to be dominated by the gear mesh orders (50, 100, 150, etc.), and the resultant TE time histories reveal mesh frequency components of TE. It is also observed from the filtered data that certain tooth-to-tooth variations exist. This is a direct result of a small amount of spacing errors present in these no-error gears — despite all efforts to minimize them. As a result, the frequency spectrum has other non-zero, non-mesh harmonics — especially below the fundamental gear mesh order of 50.

Results and Discussion

In this section, impact of indexing errors on dynamic response of gear pairs is presented utilizing predicted dynamic transmission error time histories and corresponding frequency spectra. Dynamic transmission error (DTE) predictions are compared to the baseline measured quasi-static transmission error $e(t)$ time histories (STE) and frequency spectra of the corresponding gear pairs. Here results are reported in terms of dynamic transmission error, as it provides the most direct physical link between the measured STE and predicted DTE while the model is also capable of predicting dynamic mesh force, dynamic bearing forces and displacements and gear motions, as defined in all 6 degrees of freedom.

First, a linear time-invariant (LTI) version of the same dynamic model was run to determine the resonant frequencies and corresponding mode shapes of the baseline gear pair with no indexing errors. The resonant frequencies were found to be at 3,370, 5,940 and 7,300 rpm. As pointed out earlier, dynamics regarding the y-direction motions were excluded from the discussion, as they can be treated independently. Later, dynamic response and mesh force spectra of the same baseline gear pair were predicted utilizing both LTI and NTV versions of the model to identify the off-resonant speed (frequency) ranges of the system within which specific frequencies were chosen for further analysis. The off-resonant frequencies that were used for further analysis are 400 rpm, 1,300 rpm and 2,100 rpm. At these off-resonance speeds, magnification on the TE due to the gear pair dynamics can be clearly observed without the extra amplifications due to mesh resonances and nonlinear effects.

Although numerous test cases, along with the corresponding dynamic model predictions, were carried out, a subset of the results is reported here to illustrate the importance of incorporating indexing errors into the gear design and dynamics considerations. Results from Test #1 that included the baseline gears with no indexing errors along with the corresponding dynamic transmission error predictions are shown in Figure 4.

In Figure 4(a) a measured STE time history is given between gear mesh cycles 100 to 200. This mesh cycle range was intentionally chosen to exclude any transient effects taking place — especially for the initial cycles for the predicted DTE. In addition, measured STE and predicted DTE traces were pre-

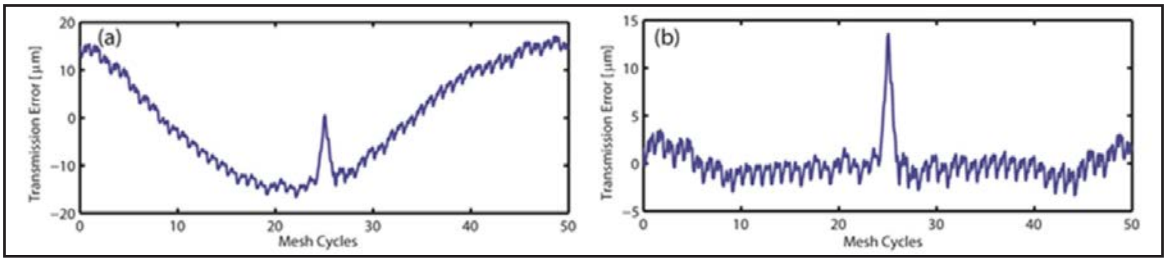


Figure 3 Measured static TE from Test #2: (a) unfiltered TE; (b) filtered TE.

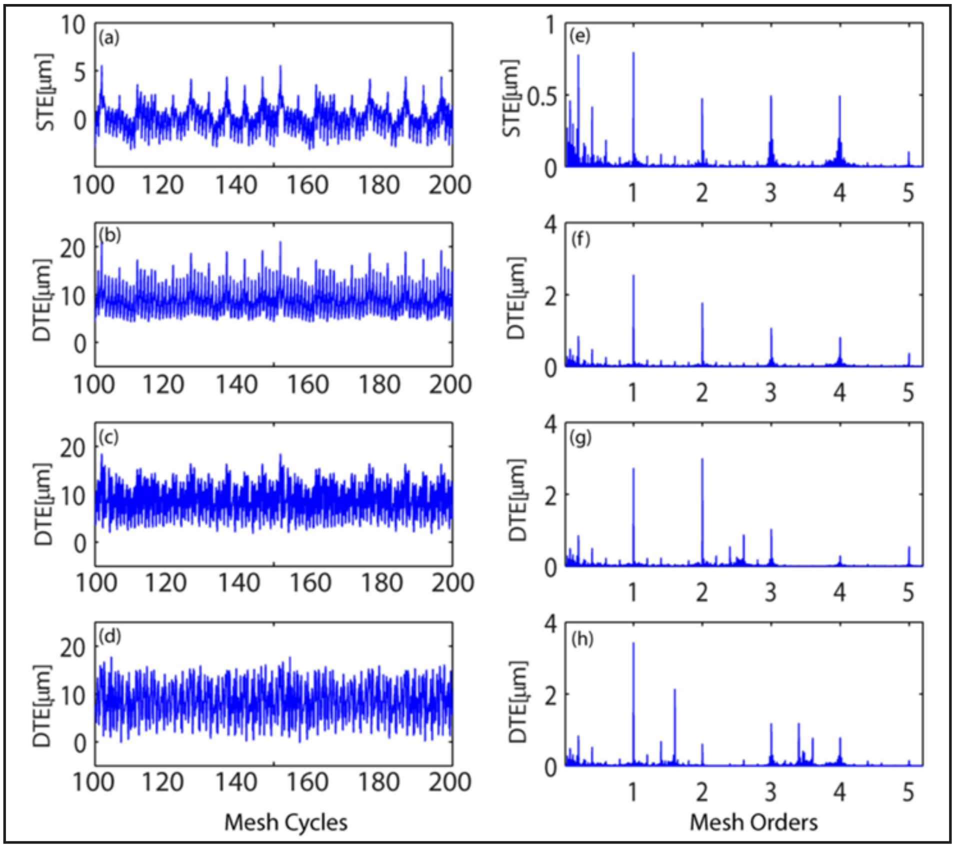


Figure 4 **Measured static TE (a) along with predicted dynamic TE time histories from Test#1 at 400, 1,300 and 2,100 rpm (b-d); and corresponding frequency spectra (e-h).**

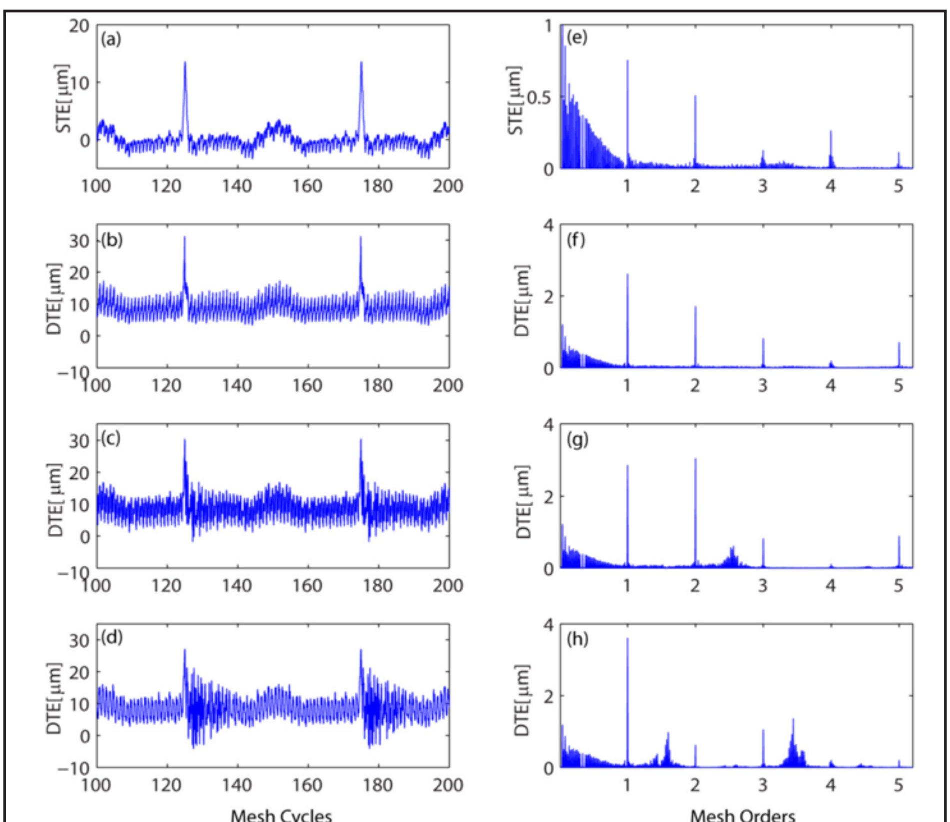


Figure 5 **Measured static TE (a) along with predicted dynamic TE time histories from Test #2 at 400, 1,300 and 2,100 rpm (b-d); and corresponding frequency spectra (e-h).**

sented within the same mesh cycle ranges to clearly demonstrate the dynamic magnification effects. It is apparent from both Figure 4(a), and its corresponding frequency spectrum Figure 4(e), that even baseline gears had limited indexing errors on them along with teeth deflection and other geometric imperfections that contributed to the STE. Predicted DTE at 400 rpm, 1,300 rpm and 2,100 rpm is presented in Figure 4(b-d), along with their corresponding frequency spectra in Figure 4(f-h). Although it is clearly observed from the comparison between time histories for STE and DTE at different speeds how dynamics play a significant role on TE, note also that frequency spectra are intentionally presented in gear mesh orders to help us detect the changes at main mesh order and its harmonics, shaft orders and also gear pair resonances at once. For instance, in Figure 4(g), when gears were rotating at 1,300 rpm, resonant peak revealed at the mesh order 2.8 that corresponded to the second mode (torsional mode) of the system. Similarly, in Figure 4(h), when gears were rotating at 2,100 rpm, resonant peaks at the mesh orders 1.6 (torsional mode) and 3.6 (coupled torsional-translational mode) were revealed due to gear pair dynamics.

In Figure 5 STE and DTE time histories, and corresponding frequency spectra from Test #2, are given following the same sequence in Figure 4. Test #2 included a gear with no indexing errors and a pinion with an intentional spacing error of 15 μm at tooth #25 (Fig. 3). The peaks that are seen in the filtered STE time trace (Fig. 5(a)) exhibit how the discrete spacing error of the particular tooth on the gear comes in mesh and causes a sudden increase in the measured STE once during a complete rotation of the gear; i.e. — once in every 50 mesh cycles. The fact that one of the teeth has discrete spacing error, whereas all the other teeth have almost perfect spacing, breaks the cyclo-symmetricity of the meshing action and thus mistunes the dynamics of the gear pair. This can essentially be observed in Figure 5(a-d), when the same gear pair was used but was run at sequentially increasing speeds. Especially, in Figure 5(c) and (d), when the gears were rotating at 1,300 and 2,100 rpm, respectively, time traces exhibit how more than once mesh cycle was affected due to a single discrete spacing error on the gear.

Under dynamic conditions, teeth do not only deflect due to the applied external load, but also oscillate about this pre-deflected position under the influence of dynamic loads. Moreover, indexing errors cause time and phase lags (or leads) on the mesh timing that can cause dynamic loads to become magnified and also modify them to where they become more impulsive. Gear contact ratio reinvigorates the gear pair dynamics as it is the most important design parameter that decides upon when and how much load will a particular tooth carry. Moreover, mesh damping strongly influences the dynamic response of the gears, but is hard to determine and also to change. The other important point to stress here is the energized broad-band shaft order regime and its physical reasoning. In Figure 5(e-h), although run-out effects were excluded from the measured STE, shaft order (orders lower than the fundamental mesh order) peaks and integer multiples are revealed. The main reason is that the teeth with deterministic spacing error repeats and leaves its signature once in every rotation of the gear, leaving a strong mark on the frequency spectrum. This effect becomes even more significant when dynamics come into play at higher speeds. The peaks revealed at integer multiples of the shaft orders are due to the amplitude and phase modulation of the TE and will be further studied in a subsequent study by the authors. In Figure 6, results from Test #3, where an intentional deterministic indexing error pattern of 0-15-0-15-0 μm was used on the pinion, is presented. In this case there is a single, intentionally misplaced tooth on the gear that was followed by both a perfectly positioned and misplaced tooth. This error pattern introduced another layer of complications to the resulting frequency spectra by strength-

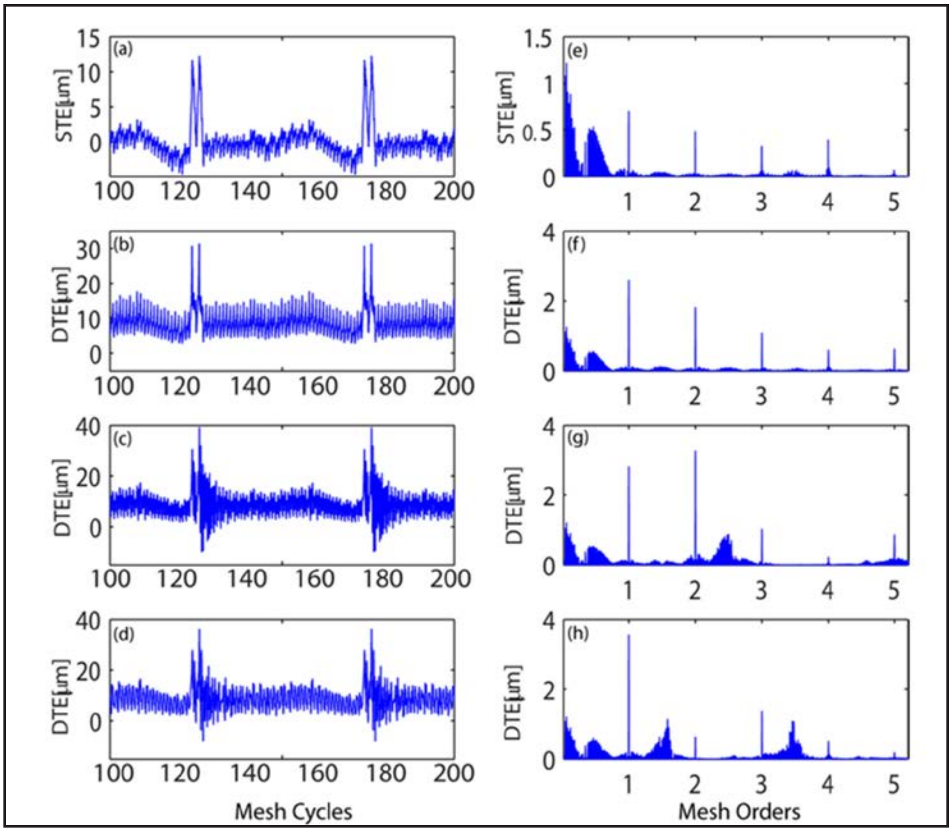


Figure 6 Measured static TE (a) along with predicted dynamic TE time histories from Test #3 at 400, 1,300 and 2,100 rpm (b-d); and corresponding frequency spectra (e-h).

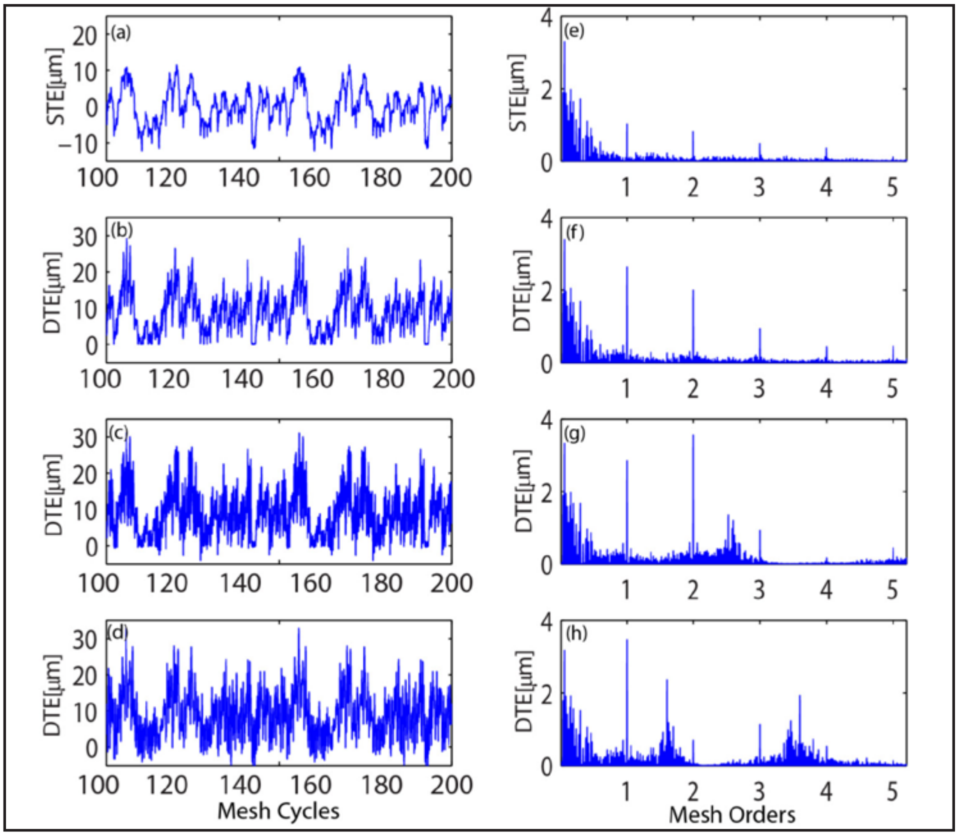


Figure 7 Measured static TE (a) along with predicted dynamic TE time histories from Test #4 at 400, 1,300 and 2,100 rpm (b-d); and corresponding frequency spectra (e-h)

ening 2× shaft orders along with 1× shaft orders. The complicated phasing relationship affected the resulting peak distribution in the frequency spectra, which is more clearly visible from orders lower than the fundamental (1×) gear mesh order.

The measured STE and predicted DTE thus far have included intentionally created, deterministic indexing errors on the gear. In Figure 7, however, results from Test #4, which included random spacing errors on the pinion, are presented. The random error pattern is definitely more representative of the real indexing error patterns observed on the gears manufactured using cutting, shaping etc. operations. Although randomly distributed, spacing errors were kept between the range [-15, 15] μm. As seen both from the time traces and the corresponding spectra (Fig. 7), random spacing error patterns cause a truly broadband excitation, thereby activating the resonant peaks and shaft orders — regardless of gear speeds.

Conclusions

In this study a model to predict gear pair dynamics using measured, long-period quasi-static transmission error of a gear pair was developed. This model uses measured, broadband static TE excitation with any time and frequency resolution and predicts dynamic gear mesh force, dynamic TE and bearing forces. Both time domain and frequency domain results can be obtained at steady-state and transient speed conditions. First, gears were intentionally mistuned through tightly-controlled deterministic and stochastic indexing errors, and resulting gear pair dynamics were compared against a baseline gear pair with no indexing errors (minimized, but not zero). Cases with gears having limited, discrete indexing errors increase the dynamic response during a limited number of mesh cycles, thus increasing the dynamic response. Their frequency spectra are enriched by additional shaft order peaks due to amplitude and frequency modulations caused by the perturbed transmission error excitation. Cases with random indexing error exhibit even greater broadband response, having peaks at shaft order and its integer multiples along with mesh order and its harmonics. They exhibit significantly more energy content at frequencies lower than the fundamental mesh frequency. The broader spectrum is caused by the broadband TE excitation causing a frequency modulated dynamic mesh force spectrum.

References

1. AGMA Technical Standards. ANSI/AGMA 2015-1-A01.
2. Remmers, E.P. "Gear Mesh Excitation Spectra for Arbitrary Tooth Spacing Errors, Load and Design Contact Ratio," *J. of Mech. Des.*100, 1978, 715–722.
3. Mark, W.D. "Analysis of the Vibratory Excitation of Gear Systems, Part I: Basic Theory," *J. Acoust. Soc. Am.* 63 (5), 1978, 1409–1430.
4. Mark, W.D. "Analysis of the Vibratory Excitation of Gear Systems, Part II: Tooth Error Representations, Approximations, and Application," *J. Acoust. Soc. Am.* 66 (6), 1979, 1758–1787.
5. Kohler, H. and R. Regan. "The Derivation of Gear Transmission Error from Pitch Error Records," *J. Mech. Eng. Science*, 199 (C3), 1985, 195–201.
6. Mark, W.D. "The Role of the Discrete Fourier Transform in the Contribution to Gear Transmission Error Spectra from Tooth Spacing Errors," *J. Mech. Eng. Science*, 201(C3), 1987, 227–229.
7. Kohler, H., R. Regan and W. D. Mark. "The Derivation of Gear Transmission Error from Pitch Error Records," Discussion, *J. Mech. Eng. Science*, 201(C3), 1987, 230–232.
8. Padmasolala, G., H. H. Lin and F. B. Oswald. "Influence of Tooth Spacing Error on Gears With and Without Profile Modifications," *NASA/TM* 2000-210061 PTG-14436, October 2000.
9. Wijaya, H. "Effect of Spacing Errors and Runout on Transverse Load Sharing and Dynamic Factors and Idler Gear Dynamic Analysis," MS Thesis, The Ohio State University, Columbus, Ohio, 2001.
10. Spitas, C. and V. Spitas. "Calculation of Overloads Induced by Indexing Errors in Spur Gearboxes Using Multi-Degree-of-Freedom Dynamical Simulation," *J. Multi-Body Dynamics* 220, 2006, 273–282.
11. Milliren, M. "An Experimental Investigation Into the Various Errors on the Transmission Error and Root Stresses of Spur Gears," MS Thesis, The Ohio State University, Ohio, 2011.
12. Handschuh, M. An Investigation into the Impact of Random Spacing Errors on Static Transmission Error and Root Stresses of Spur Gear Pairs," MS Thesis, The Ohio State University, Ohio, 2013.
13. Tamminnana, V.K., A. Kahraman and S. Vijayakar. "On the Relationship between the Dynamic Factors and Dynamic Transmission Error of Spur Gear Pairs," *J. Mech. Des.*, 129, 2007, 75–84.
14. Kahraman, A., J. Lim and H. Ding. "A Dynamic Model of a Spur Gear Pair with Friction," *12th IFToMM World Congress*, Besançon (France), June 18–21, 2007.
15. Blankenship, G. W. and A. Kahraman. "Steady State Forced Response of a Mechanical Oscillator with Combined Parametric Excitation and Clearance Type Nonlinearity," *J.Sound and Vibr.*, 185, 1995, 743–765.
16. *Windows LDP*. The Ohio State University, Columbus, Ohio, 2013.
17. *External2D*. Advanced Numerical Solutions, Inc., Hilliard, Ohio, 2013.

Dr. Murat Inalpolat is an Assistant Professor in the Department of Mechanical Engineering at The University of Massachusetts Lowell. He has more than a decade of experience in the areas of structural health monitoring; damage detection/diagnostics and prognostics; structural dynamics and acoustics; and signal processing — along with rotating machinery dynamics and noise. His most current research focuses on structural health monitoring, diagnostics, prognostics, and operational damage detection and identification. He also has extensive project experience working on aircraft- and rotorcraft-related projects obtained while working at the General Electric Global Research Center. He has been a reviewer for such publications as the Journal of Sound and Vibration; Mechanical Systems and Signal Processing; and ASME Journal of Mechanical Design. Inalpolat was previously awarded by CTI (Car Training Institute of Germany) the prestigious "Young Drive Experts" award based on his quality of the research

Michael J. Handschuh is a PhD student in the Mechanical and Aerospace Department of The Ohio State University, having received his BS and MS degrees in Mechanical Engineering in 2011 and 2013, respectively. Handschuh is a Graduate Research Associate with the Gear and Power Transmission Research Laboratory.

Ahmet Kahraman is Howard D. Winbigler Professor of Mechanical and Aerospace Engineering at the Ohio State University. He is the Director of Gleason Gear and Power Transmission Research Laboratory. He also directs the Pratt & Whitney Center of Excellence in Gearbox Technology.

Hermann J. Stadtfeld

RECEIVES 'DISTINGUISHED INVENTOR OF THE YEAR AWARD'

Dr. Hermann J. Stadtfeld, vice president of bevel gear technology and R&D at Gleason Corporation, and a frequent contributor to *Gear Technology* and *Power Transmission Engineering* magazines, was recently named a recipient of the annual RIPLA "Distinguished Inventor of the Year Award."

RIPLA Award winner Dr. Hermann J. Stadtfeld with wife Hedy and Gleason colleagues: (From left) Markus Bolze, manager, bevel gear software design; John J. Perrotti, president and chief executive officer; Uwe Gaiser, director product management, bevel gear technology; (Stadtfeld); Robert L. McDowell, senior patent agent; and Mrs. Hedy Stadtfeld.

New York-based RIPLA (Rochester Intellectual Property Law Association) began conferring the award in 1976 to those of particular merit in the local research community of the Rochester area. Former winners include Ernest Wildhaber (gearing mechanisms), Robert Rose, William Bonnez and Richard Reichman (HIV vaccine), and Steve Sasson (digital camera).

Although the award is given in recognition of a body of work, Stadtfeld, who possesses numerous patents, was particularly cited for "developing a process for manufacturing gears without the use of oil to cut the metal which saves energy and materials as well as making a safer work environment." He was one of two winners announced from a group of 11 nominees celebrated at a June 4 banquet.

Stadtfeld's name was put in nomination for the award by a fellow Gleason employee, Ronald Rietz. Joining Stadtfeld for the event were his wife, Hedy, Gleason CEO and President John Perrotti, Gleason patent agent Robert McDowell, and Gleason colleagues Markus Bolze, Uwe Gaiser and Rietz.

"Gleason is extremely proud of Dr. Stadtfeld's award of Distinguished Inventor of the Year from the Rochester Intellectual Property Law Association (RIPLA)," said Al Finegan, Gleason director of marketing. "The contributions of Stadtfeld and the entire gear technology development team reinforce his position as a leading worldwide authority in the field of bevel gear theory and technology, but are also a key element in Gleason's position as The Total Gear Solutions Provider."

(Every technical paper of Dr. Stadtfeld's that has appeared in *Gear Technology* is available unabridged, at no charge, at *www.geartechnology.com*.)

Samputensili

ACQUIRES GEAR SHAVING MACHINE TECHNOLOGY COMPANY SICMAT

Samputensili recently announced the acquisition of Sicmat S.p.A., a company specializing in gear shaving machine technology.

Due to the synergies between the two Italian companies, customers worldwide will benefit from an enhanced level of technology and service support. The continuity of the actual product portfolio will therefore be ensured, as well as the supply of original spare parts, updated technical drawings and machine software that will be distributed by Samputensili and Star SU globally.

As a result of the acquisition, a strategic opportunity has been created by combining Sicmat's decade-long years of expertise in the gear shaving process with Samputensili's product development capabilities as manufacturer of machine tools and gear cutting tools. Gear shaving machines and related equipment will be produced at the Technology Centre in Bentivoglio, near Bologna (Italy).

The purchase of Sicmat allows Samputensili to broaden the company's product portfolio and therefore move closer to the corporate mission of becoming a leading supplier of complete solutions for the gear manufacturing industry.

At the upcoming EMO (Milan, Italy, Oct. 5-10), Samputensili will display the RASO 200 gear shaving machine.

Excel Gear

FORMS NEW RETROFIT DIVISION FOR THE REPAIR AND UPGRADE OF OLD MACHINERY

Excel Gear recently formed a new division of their company, Excel CNC Retrofit LLC, devoted to upgrading and remanufacturing older gear machines to meet today's performance standards.

The company offers services to enhance the performance of most major gear machine models; from converting manual or outdated CNC machines to the use of the latest Fanuc CNC; to restoring older machines to their original OEM specifications; to increasing productivity to levels comparable to those found on the latest machines.

"We're unique to this industry because we combine an exceptional level of rebuild and remanufacturing experience with Excel Gear's renowned gear manufacturing and engineering capabilities," said Excel CNC Retrofit's Bill Ramsay. "Ultimately, our customers benefit with a more economical machine solution delivering exceptional performance."

True Gear & Spline

INCREASES GRINDING CAPACITY WITH NEW NILES ZP 12

True Gear & Spline Ltd. recently added CNC gear profile grinding to their machining capabilities with their newly installed Niles ZP 12 gear grinder.

The recently redesigned Niles ZP 12 will allow True Gear & Spline to grind larger gears with higher tolerance and increased speed, expanding their services to their customers in the machinery, mining, energy, oil and gas, and prototyping sectors.

True Gear & Spline serves a worldwide customer base from their 20,000 sq. ft. manufacturing facility located in Cambridge, Ontario. In 1998, they purchased a Niles ZSTZ 315 gear grinder that still exists in the facility. In 2008 they became ISO 9001 certified.

Historically, True Gear & Spline made gears to AGMA class 8; using the older Niles machines, they achieved AGMA 10. Today, AGMA 11 and 12 are routinely achieved on the Niles ZP 12.

"Continued growth and investing in the latest technology has always been a driving force at

True Gear & Spline," said vice president Mark Bizjak. "This latest purchase is in keeping with our commitment to accommodate our customers with any type of gear."

True Gear's Niles ZP 12 will allow them to grind internal and external gears up to 1.2 meters in diameter. Dual dressers reduce dressing time and gear inspection on the machine. Optional software makes worm grinding possible, too.

"We have had our ZSTZ 315 in production for close to 20 years and look forward to many more years of using the advanced technology offered by the ZP 12 machine," Bizjak said. "We have had positive experiences with the Kapp Niles staff, and with their sales representative, John Manley of Machine Tool Systems, Inc."

Broaching Machine Specialties

TO SUPPLY EKIN PRODUCTS IN NORTH AMERICA

Broaching Machine Specialties (BMS) recently announced a strategic partnership with Ekin of Bilbao, Spain.

Ekin has been supplying machinery and tools for broaching and spline rolling operations for over 50 years. Ekin works primarily in the automotive, power generation and aerospace industries.

BMS will act as the U.S. and Canadian sales and service organization for the following product lines: helical broaching machines and tools; hard broaching machines and tools; dry broaching machines and tools; CNC broach sharpening machines; and spline rolling machines and tools.

Ekin machines are electromechanically powered and are available in various stroke lengths and tonnages.

Gleason's Gear Solutions Forum

TO INCLUDE OVER 40 DEMONSTRATIONS ON PRODUCTS AND TECHNOLOGIES

Gleason Corporation will host the Gear Solutions Forum (GSF) at its worldwide headquarters in Rochester, NY on Sept. 23-24. This year also marks the company's 150th anniversary, which will be recognized as part of certain events during the two-day forum.

GSF will include more than 40 demonstrations of products and technologies, covering all major gear production processes addressing a wide variety of industry needs. The event will be accompanied by presentations from international industry experts, who will present their research in a wide range of gear-related subjects relevant to gear design and manufacturing.

GSF is an educational opportunity, giving visitors an in-depth look at many of the advanced new technologies that will shape the future of gear manufacturing. Demonstrations and presentations will cover technologies for spiral and straight bevel gears, including grind from solid and SmartLap corrective lapping. For cylindrical gears, live exhibits will include high-speed hobbing, power skiving, grinding and optimum processing of large gears both on dedicated gear machines and five-axis machining centers.

August 4–6—Ipsen U Cherry Valley, IL. This three-day course provides attendees with a broad overview of furnace equipment, processes and maintenance, as well as a hands-on approach to learning while receiving qualified tips directly from the experts. Participants in Ipsen U courses throughout 2015 have come from across North America, including Colorado, Illinois, Massachusetts, Mexico, Michigan, New Jersey, New Hampshire, Pennsylvania and Texas. Throughout the course, attendees will learn about an extensive range of topics—from an introduction to vacuum and atmosphere furnaces to heat treating, furnace controls, subsystems, maintenance and more. For more information, visit *www.IpsenUSA.com/IpsenU.com*.

August 6–8—Asia International Gear Transmission Expo 2015 As Asia's most influential, professional and authoritative gear industry event, GTE has been held 10 years in a row and during that time has obtained the affirmation of a large number of exhibitors and buyers. The exhibition will work with multiple marketplace platforms to create the Asia gear industry's most influential international showcase. With a planned area of 45,000 square meters, the exhibition expects more than 500 exhibitors and 40,000 professional visitors from home and abroad. For more information, visit *www.gte-asia.com*.

August 10–12—MPIF's Basic PM Short Course Penn Stater Conference Center Hotel, State College, PA. This intensive 3-day course is designed especially for you, if you are starting out in the field and looking for an introduction to powder metallurgy (PM); updating your knowledge of recent developments in PM; seeking to expand your current knowledge of the PM industry; a user of PM parts or are considering PM. This course is designed for engineers, tool designers, metallurgists, supervisors and technicians. For more information, visit *www.mpif.org*.

September 13–15—TECHINDIA 2015 Bombay Exhibition Centre, Mumbai, India. TECHINDIA will be the ultimate facilitator for b2b cooperation between manufacturers and consumers of all hues connected to the engineering, machinery and manufacturing industry. This leading business event is co-located with five other industry events to make it an extended platform for metal, engineering, manufacturing and machine tools industry: World of Metal—International Exhibition on Metal Producing, Metal Processing and Metal Working Industry; CWE—International Exhibition on Cutting and Welding Equipment; IMEX—International Exhibition on Machine Tools and Engineering Products; UMEX—International Exhibition on Used Machineries; Hand Tools and Fasteners Expo—International Exhibition on Hand Tools and Fasteners. The co-location of industry events will maximize business opportunities for industry professionals. For more information, visit *techindiaexpo.com*.

September 21–23—Gear Failure Analysis Big Sky Resort, Big Sky, MT. Explore gear failure analysis in this hands-on seminar where students not only see slides of failed gears but can hold and examine those same field samples close up. Experience the use of microscope and take your own contact pattern from field samples. Cost is $1,600 for members and $2,100 for non-members. For more information, visit *www.agma.org*.

September 29-October 1—2015 Gear Manufacturing Seminar Hyatt Regency, Rochester, NY. This seminar provides the gear design engineer with a broad understanding of the methods used to manufacture and inspect gears and how the resultant information can be applied and interpreted in the design process. Following this seminar, participants will be able to identify methods of manufacturing external and internal spur, single and double helical, and bevel and worm gears, describe the methodology ad underlying theory for basic manufacture and inspection of each, and much more. Cost is $1,430 for member and $1,930 for non-members. For more information, visit *www.agma.org*.

October 4–7—Euro PM2015 Congress & Exhibition Reims Congress Centre, Reims, France. Europe's annual powder metallurgy congress and exhibition, organized and sponsored by the European Powder Metallurgy Association, will return to France in 2015. The combination of a world class technical program and state-of-the art exhibition will provide the ideal networking opportunity for suppliers, producers and end-users. The program of plenary and keynote addresses, oral and poster presentations and special interest seminars will focus on: additive manufacturing; hard materials and diamond tools; hot isostatic pressing; new materials and applications; and more. Alongside the technical sessions the Euro PM2015 Exhibition will be an excellent opportunity for international suppliers to the PM industry to network with new and existing customers from the powder metallurgy and associated sectors. For more information, visit *www.europm2015.com*.

October 7–8—Design & Manufacturing Philadelphia Pennsylvania Convention Center, Philadelphia, PA. As the United States' first major industrial city, Philadelphia continues to grow into a leading design and manufacturing hub across a wide variety of industries. With manufactured goods exports increasing twice as fast as the overall state economy, accounting for 87.8% of all state total exports, now is the time to bring everyone within this community to one dedicated design and manufacturing event to spark new ideas and partnerships. See the newest technologies, equipment, products, and services and get real time answers to your questions from industry experts. The educational offerings bring you the latest in manufacturing trends and applications with comprehensive full day conferences across both days of Design & Manufacturing Philadelphia. For more information, visit *www.dmphilly.designnews.com*.

October 27–29—Modern Furnace Brazing School Brazing Engineering Center, Wall Colmonoy Aerobraze Division, Cincinnati, OH. The late Robert Peaslee's tradition continues with the return of the brazing school. The Brazing Engineering Center provides engineering services and training, as well as offering new practical experience on the shop floor. For over 60 years, Wall Colmonoy engineers have gained practical experience on actual problems in brazing plants around the world. Knowledge and practical application will be taught by industry-leading brazing experts. In 1950, Peaslee developed the first nickel-based brazing filler metal, Nicrobraz. Modern Furnace Brazing School will allow you to apply workable solutions to your brazing needs. For more information or to register, contact *brazingschool@wallcolmonoy.com*.

gear TECHNOLOGY

FREE SUBSCRIPTION

Gear Technology, is the world's leading publication on gear design, engineering and manufacturing. To sign up or renew your **FREE SUBSCRIPTION** use the form below.

FIND THE EXACT SUBJECTS YOU ARE LOOKING FOR: Subscribers now get access to the 30 years of searchable articles in our online archive.

1

☐ **YES** I want to receive/continue to receive ***Gear Technology*** FREE**!**

☐ No thank you.

Name:______________________________ (print) JobTitle/Position: ____________________

Signature ____________________________________ (required) Date: ____________

2 E-mail ______________________________________
(So we can notify you when the online edition is available)

SAVE a TREE!
☐ I ONLY want the digital edition (e-mail required)

3 Please update your contact information or attach you business card here. (No Staples)

Company Name: ______________________________
Address: ______________________________
____
____
City: ______________________________
State/Province: ____________ ZIP+4/Postal Code: ____________
Country: ______________________________
Telephone: ______________ Fax:______________
E-mail: ______________________________
Website: ______________________________

4 What is your primary JOB FUNCTION (check ONLY one)

- ❏ Corporate Executive (B)
- ❏ Manufacturing Production Management (C)
- ❏ Manufacturing Production Department (D)
- ❏ Manufacturing Engineering Management (E)
- ❏ Manufacturing Engineering Department (F)
- ❏ Product Design, R&D Management (H)
- ❏ Product Design, R&D Department (I)
- ❏ Purchasing (L)
- ❏ Marketing, Sales or Communications (M)
- ❏ Quality Control Management (P)
- ❏ Quality Control Department (Q)
- ❏ Other (Please describe) (N)

____

5 My company's principal product or service is:____________________

6 How are YOU involved with gears (check all that apply)?

- ☐ My company MAKES gears (20)
- ☐ My company BUYS gears (22)
- ☐ I DESIGN gears (23)
- ☐ I am a SUPPLIER to the GEAR INDUSTRY (please specify product or service) (24)

____

Is there a colleague within your company that might benefit from having their own copy of ***Gear Technology*** magazine?

Name: ______________________________
Title: ______________________________

DIGITAL SUBSCRIPTIONS

- ❏ **YES,** I want to be notified each issue when *Gear Technology* is available online
- ❏ **YES,** I would like to receive *Gear Technology*'s monthly e-mail newsletter
- ❏ **YES,** I would like to receive *Gear Technology*'s twice monthly e-mail Product Alerts
- ❏ **YES,** I would like to receive 3rd-party e-mails from *Gear Technology*'s advertisers and partners
- ❏ **YES,** I would like to receive subscription renewal notices by e-mail
- ❏ **No,** I do not wish to be contacted by e-mail.

FAX TO 1-847-437-6618 or
VISIT US ONLINE AT
www.geartechnology.com/subscribe.htm

Correction

During the editing process, an error was introduced into the conclusion of the article "An Approach to Pairing Bevel Gears from Conventional Cutting Machine with Gears Produced on 5-Axis Milling Machine," by Dr. Inho Bae from KISSsoft AG (June 2015).

As the method presented in the paper can be applied in many new cases, the correct conclusion should read "These are very important features in practice, and were unresolved issues in the 5-axis milling process."

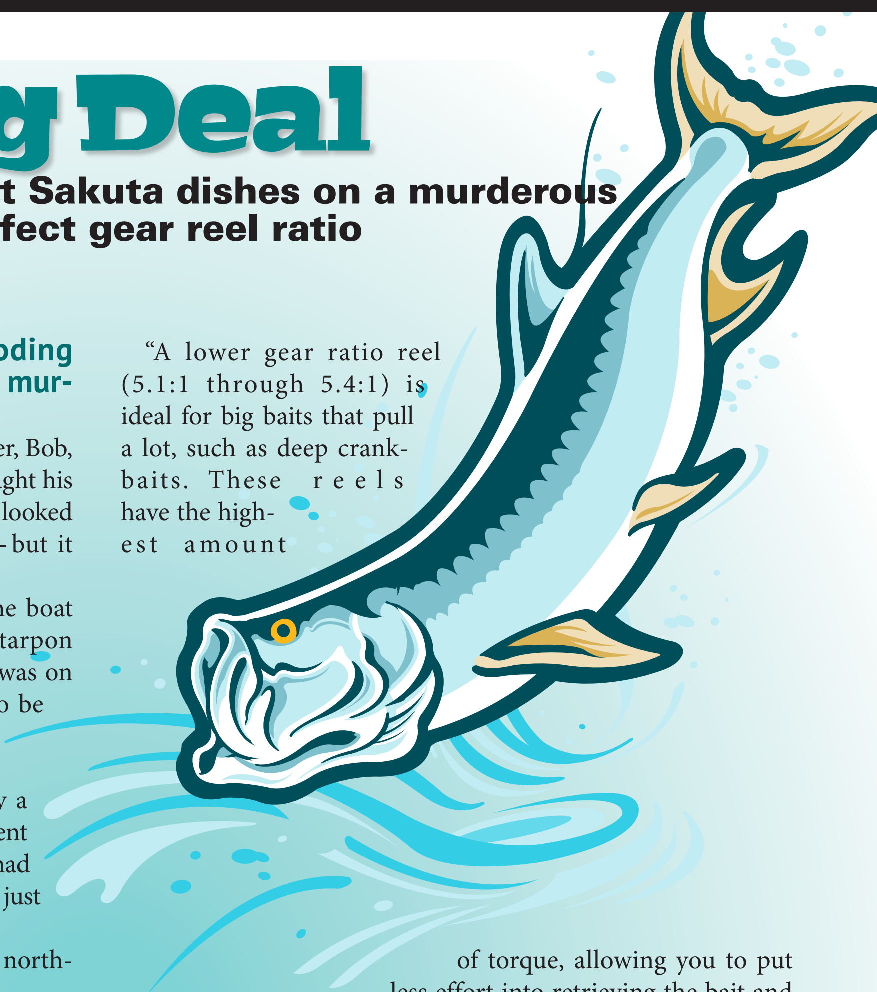
The online version of the article has been corrected.

Gear Technology regrets the error.

A Reel Big Deal

Delta Gear Vice President Scott Sakuta dishes on a murderous tarpon and how to find the perfect gear reel ratio

Erik Schmidt, Assistant Editor



One time not long ago, under the foreboding glow of moonlight, Scott Sakuta was almost murdered by a tarpon.

It was late at night and Sakuta was fishing with his father, Bob, and several other employees at Delta Gear, when they caught his would-be executioner and wrangled it into their boat. It looked like a normal tarpon — a sizeable, silvery Megalops — but it wasn't. It had hate in its little fishy heart.

As Sakuta positioned the 150-pound leviathan on the boat for revival before casting it back into the water, the tarpon tilted its head — mouth agape — right at him. This fish was on a scaly, slippery warpath, and poor Sakuta happened to be directly in its line of fire.

"I'm leaning over the boat, and the fish jumped and hit me right in the face — it was like being punched by a 150-pound punch," he says. "I saw stars and I almost went over the side of the boat. When I came back to work, I had a big black eye and a big scratch, because this tarpon had just punched me in the face."

The moral of the story: Forget the Kumite, fishing is northeastern Michigan's real bloodsport.

That's not to say that the idealistic image of fishing — lounging on a lawn chair at the edge of a serene lake, pole in one hand, lite beer in the other — is untrue. Some people *do* do that. Sakuta just isn't one of them.

"I've caught a lot of the saltwater trophy fish: blue marlin, sailfish, tarpon — the one I haven't had the opportunity to catch is the white marlin," Sakuta says. "My dad's an outdoorsman, so we fished for years and years together on the lakes and rivers in Michigan. Right around 2003 or 2004 we started to investigate some of the more exotic, warmer climate fish.

"At one point we considered possibly making reels — we have the machines and the gears, so we toyed with that for a while."

It's not quite as troublesome as a terrible tarpon but, according to Sakuta, finding the right gear ratio reel can be — pardon my pun — fishy business.

Sakuta says that choosing the right gear ratio reel is one of the more confusing aspects of fishing, but with some basic understanding of what the numbers really mean, it gets much easier to understand.

It's not something the average lawn chair fisherman contemplates while he's angling for bite-sized bluegill, but choosing a reel with the right gear ratio can often be the difference between a Friday night fish fry and eating dinner out of a paper sack.

Fishing website *Wired2Fish.com* outlined what you should be looking for in a 2013 article:

"The gear ratio of a reel is measured by how many times the spool turns for each single turn of the handle. For instance, if a reel has a gear ratio of 6.4:1, every time you turn the handle, the spool inside turns exactly 6.4 times.

"A lower gear ratio reel (5.1:1 through 5.4:1) is ideal for big baits that pull a lot, such as deep crankbaits. These reels have the highest amount of torque, allowing you to put less effort into retrieving the bait and more energy towards finding the fish.

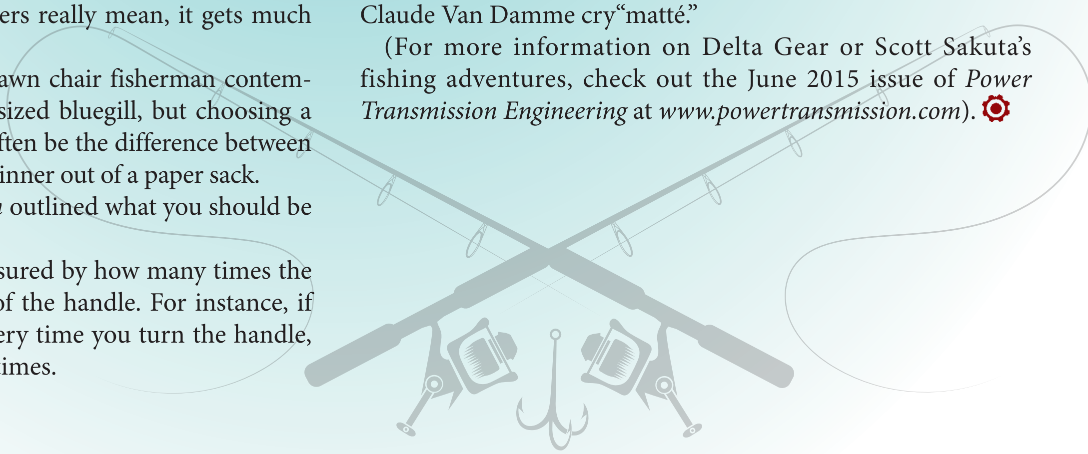
"Medium gear ratio reels (6.1:1 through 6.4:1) are great for multiple techniques and presentations, making them very popular among bass anglers. Whether you're plowing through nasty cover with a squarebill during the prespawn or bombing spinnerbaits on shallow flats in the fall, a medium gear ratio reel will do the job.

"If you're fishing any lure that you primarily work with your rod, a high gear ratio reel (7.1:1 through 8.1:1) is the way to go. You're often pulling the bait with your rod tip, but you need to have the ability to quickly take up your slack when you get a bite. A fast reel also helps when fighting a big bass—you need all the speed you can get in order to quickly pull it away from any line-fraying hazards."

Unfortunately for Sakuta, the "ideal" gear ratio reel couldn't save him from getting round-housed by a homicidal tarpon.

But then again, that fish probably would have made Jean-Claude Van Damme cry"matté."

(For more information on Delta Gear or Scott Sakuta's fishing adventures, check out the June 2015 issue of *Power Transmission Engineering* at *www.powertransmission.com*).



KLINGELNBERG

MEASURING TECHNOLOGY

OERLIKON

BEVEL GEAR TECHNOLOGY

HÖFLER

CYLINDRICAL GEAR TECHNOLOGY

A winning team has a system. It is perfectly tuned and balanced. It acts with the utmost precision. The strengths of each individual work together like perfect gears.

Klingelnberg's divisions are true team players:
Ground breaking solutions for an optimal production process of Bevel and Cylindrical Gears give manufacturers worldwide a leading edge in gear manufacturing.

NEXT FAIRS:

EMO MILANO
Oct 5-10, 2015
Hall 4 Booth D23
Milano, Italy

GEAR EXPO
Oct 20-22, 2015
Booth 1410
Detroit, USA

Win the match:
www.klingelnberg.com
www.hofler.com

KLINGELNBERG America, Inc.
p. +1 734 470 6278
e. usa.sales@klingelnberg.com

SUBJECT TO CHANGES.

MEASURING TECHNOLOGY | BEVEL GEAR TECHNOLOGY | CYLINDRICAL GEAR TECHNOLOGY