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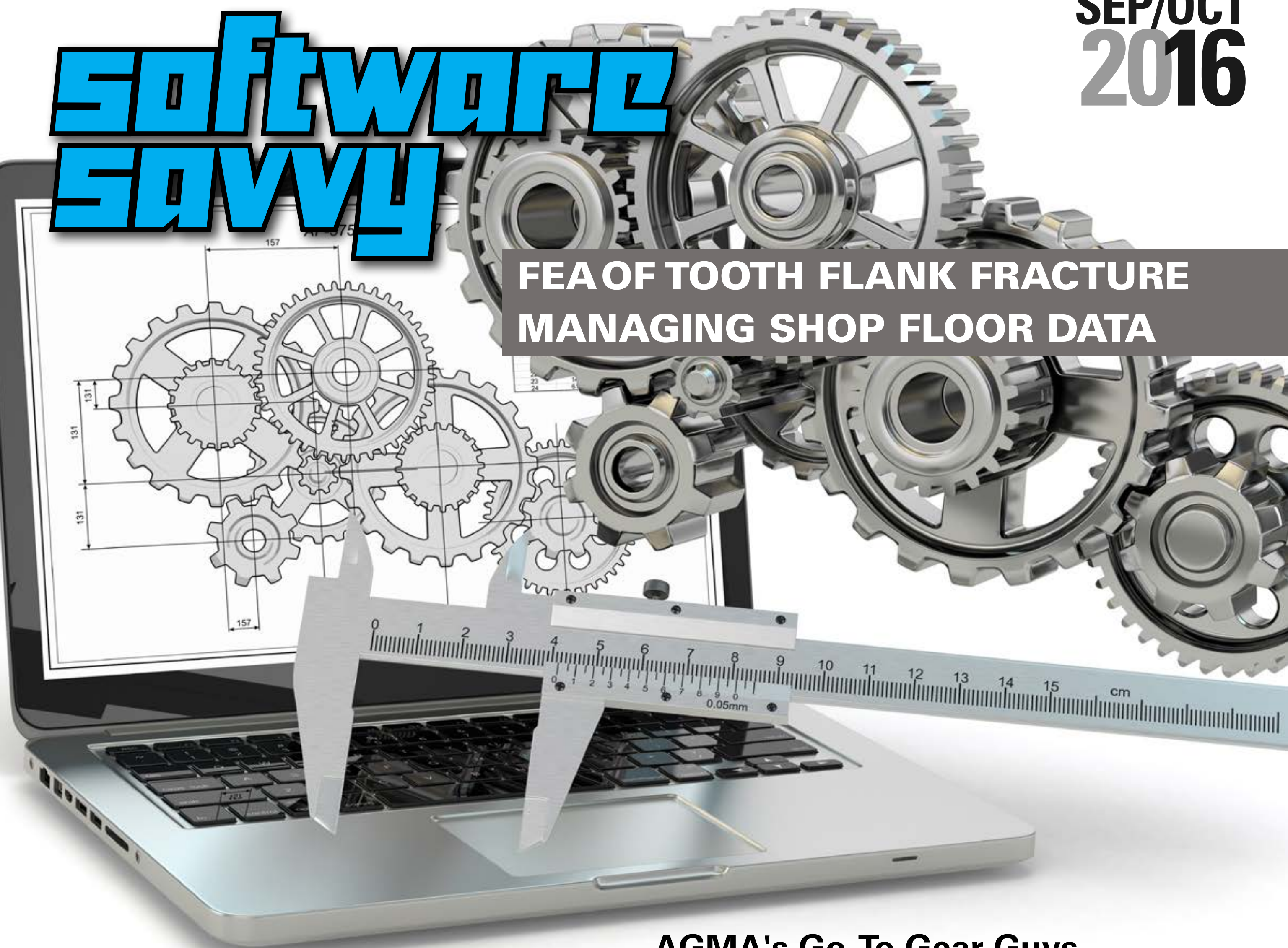

FEA OF TOOTH FLANK FRACTURE
MANAGING SHOP FLOOR DATA

AGMA's Go-To Gear Guys
Advances in Spline Manufacturing


Star SU
SICMAT
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SEP/OCT 2016

24

20

features

technical

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Vol. 33, No. 7

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THE GEAR INDUSTRY'S INFORMATION SOURCE

www.geartechnology.com

Product Spotlight
Kennametal Mill 4-11 Series

This October Kennametal (*www.kennametal.com*) will release the Mill 4-11, a new series of indexable milling cutters designed for smaller machining centers. The Mill 4-11 accommodates 40-taper CAT and BT, HSK50 and similarly-sized spindles, according to Tim Marshall, senior global product manager for indexable milling.

Get Social

Have you browsed our Twitter page recently? We've added the latest gear news and product information from companies like Mahr Federal, Holroyd Precision and Maprox. Check out these and other gear manufacturing topics here: *https://twitter.com/Gear_Technology*.

Facebook is another resource that features updates on the ***Gear Technology*** Blog, additional Ask the Expert resources and a quick and convenient way to renew your GT magazine subscriptions.

Unlock the Vault

The GT Library offers more than 2,000 technical articles published in the last 32 years. For a quick, hassle-free way to look up the gear information you need by subject. Check out the **Subject Index** at (*www.geartechnology.com/subjects/*).

IMTS 2016 Measures Up

Not only did IMTS 2016 attract more than 115,000 registered attendees (slightly higher than the 2014 show), but it also "measured up" by presenting a wide variety of new technologies for gear inspection. Our online newsletter includes a quick rundown of some of the technologies *Gear Technology* editors saw at the show.

www.geartechnology.com/newsletter/0916.htm

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BPA WORLDWIDE BUSINESS

IMTS Cuts the Mustard

Publisher & Editor-in-Chief
Michael Goldstein

Boy! The combination of Chicago style hot dogs and machinery seems to have been irresistible.

IMTS, held bi-annually here in Chicago, ended September 17. It was the third-largest IMTS ever, both in terms of registrations (115,612) and exhibit space occupied (1,370,256 square feet). Also, this year's show had more exhibiting companies (2,407) than any previous IMTS.

Many of the exhibitors came to Chicago with a certain amount of apprehension. After all, certain manufacturing industries (energy and mining, to name two) have been struggling for some time. But despite rumors to the contrary, manufacturing is alive and well, as evidenced by the steady foot traffic through the exhibit halls during most of the show.

Most of the major gear industry exhibitors told me they were happy with the show. Executives at Gleason, Star SU, Klingelnberg, Kapp-Niles, Liebherr, Mitsubishi, Koepfer America and others all told me they were pleased with the number and quality of leads they were getting from attendees.

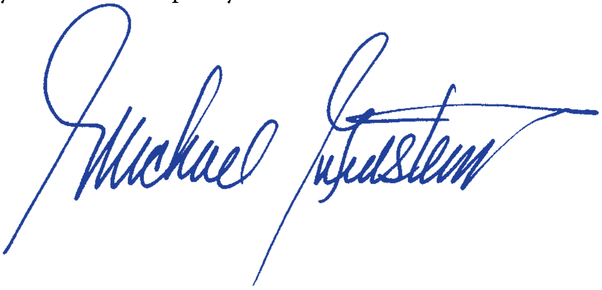
To me, this means there is significant activity brewing in the gear industry. Gear manufacturers are preparing for future work enquiring about machine tools and upgrading their technology. And some of them were even in a buying mood. I saw more than a couple instances of machine tools with "sold" signs on them. Hopefully, this bodes well for a strong gear industry in 2017.

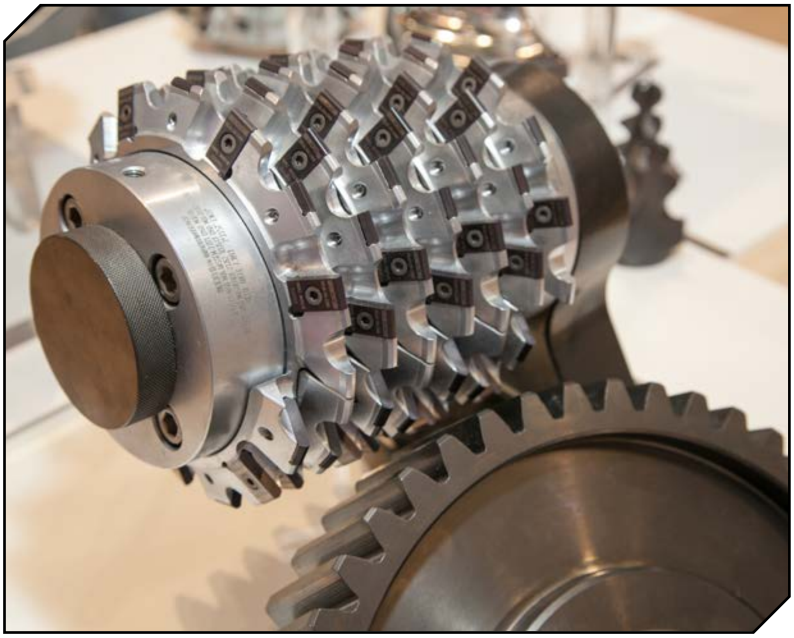
If you didn't make it to IMTS this year, you missed out on a lot of new technology. But don't worry. We were there for you. In addition to myself, we had four editors at the show (Randy Stott, Jack McGuinn, Matt Jaster and Alex Cannella), plus our digital content manager (Kirk Sturgulewski), our *geartechnology.com* blogger and technical editor (Chuck Schultz), our art director (Dave Ropinski) and associate publisher Dave Friedman. All of them were extremely busy throughout the week, talking to the experts about how gear manufacturing is changing. This wasn't MY staff who went to the show, it was YOUR staff. Their goal is to help you learn what's going on in the industry and to give you the information you need to make gears and related products better, faster and at a lower cost.

Some of that information has already been published. The article "IMTS Measures Up," for example, appeared in our e-mail newsletter. That article covered many of the newest technologies in gear inspection that were on display at IMTS. You can read the newsletter by visiting *www.geartechnology.com/newsletter/*. You can subscribe by visiting *www.geartechnology.com* and clicking on "Subscribe" for your free subscription at the top of any page.

We will continue to share everything that's important from the show over the next several issues. For example, in this issue, you can read Alex Cannella's article on the latest advances in spline manufacturing. Next issue, we plan to cover one of the hottest topics in gear manufacturing: gear skiving. And we'll also have a feature on the gear shop of the future.

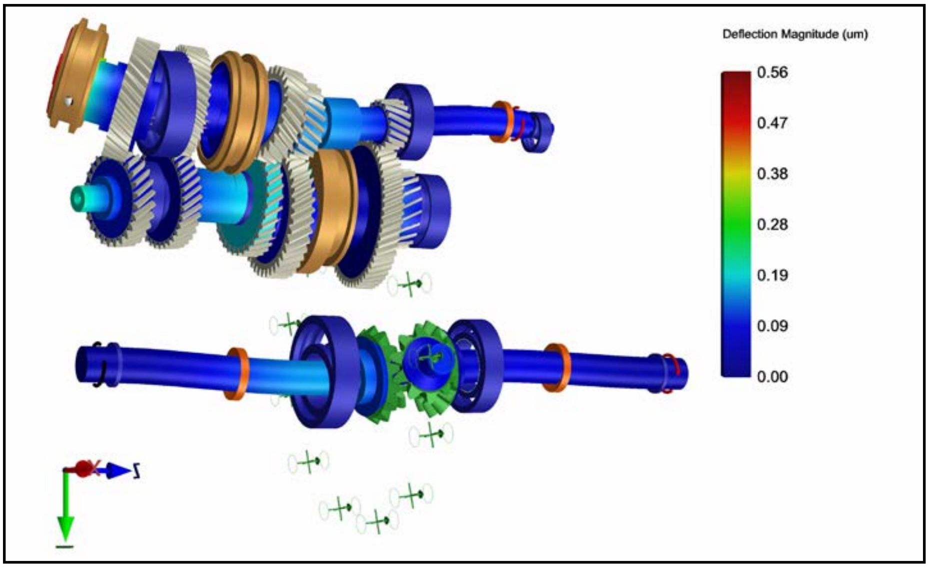
So stay tuned. There's plenty more to come.





Romax Technology

SUPPORTS VOLKSWAGEN GROUP TO OPTIMIZE THEIR TRANSMISSION DESIGN



Example of mode shape analysis in *RomaxDesigner*.

With continued pressure to reduce development time and costs, along with ever-higher consumer expectations and the rise of electric vehicles (EVs) and hybrids, the need to understand and solve noise, vibration and harshness (NVH) issues has become greater than ever. Companies want to quickly and accurately assess the effects of design changes as early as possible in the development cycle — before the process moves into expensive physical prototypes and testing.

Romax Technology offers simulation software that allows designers and manufacturers to assess and optimize NVH characteristics while also maintaining or improving efficiency and durability, and in a single environment: a powerful integrated approach optimized for design improvements. Issues can be identified at the design stage, checking basic analyses for NVH at a concept level, saving time and money.

Volkswagen Group is one such company that Romax Technology have supported to provide a holistic approach to their design process using *RomaxDesigner* software for gearbox simulation, to perform fast and accurate NVH analysis at each stage of the design and validation.

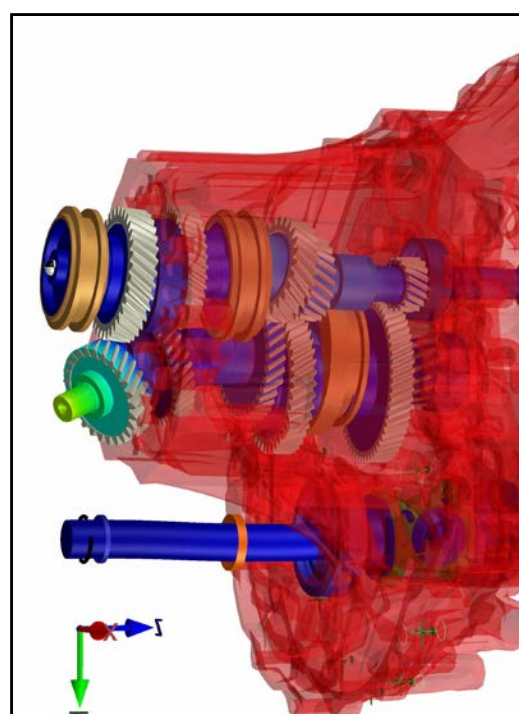
Europe's biggest carmaker, Volkswagen Group delivers over 10 million cars to customers each year. Almost one in four new cars (24.8 percent) in Western Europe are made by Volkswagen, a group which comprises 12 leading brands from seven European countries: Volkswagen Passenger Cars, Audi, SEAT, ŠKODA, Bentley, Bugatti, Lamborghini, Porsche, Ducati, Volkswagen Commercial Vehicles, Scania and MAN.

Its challenge was to develop a reliable validation strategy for gearbox NVH to allow design changes to be made with confidence, and satisfying the demanding needs of the market-leading brands. Romax's whole system simulation environment offers both a prevention and cure strategy for transmission NVH issues. Design from the earliest concept stages with NVH in mind for confidence from the start or use advanced analysis and optimization processes to improve the NVH performance of existing designs, whilst never having to compromise on efficiency or durability.

Employing over 15,000 people, Volkswagen's primary transmission site at Kassel, Germany, supplies about four million manual and automatic transmissions every year. Volkswagen engineers at Kassel have used *RomaxDesigner* software for more than five years, to support the effective production of gearboxes and to ensure the required NVH quality is achieved. Kassel's Acoustics and System Simulation department focuses on NVH correlation and simulation: "Our main challenge is gear whine, and the need to support our high acoustic standards," said Carsten Schmitt, Ph.D. student of Volkswagen's postgraduate program. "NVH is such an important issue in the industry today because of the rise in electric motor developments, and the simultaneous increase in the production of complex gearboxes. We use *RomaxDesigner* so that we can perform accurate simulation of these new gearbox designs, and assess the NVH performance."

From trial and error to simulation for development

Previously, sporadic correlation studies on the main parts of a gearbox would be conducted based on eigenfrequencies, which allowed for little correlation guarantee. "We have a requirement to develop simulation models that are representative of the real world, so that our design changes can be made with confidence," said Schmitt. "This gave rise to the need for an integral validation strategy, which we investigated in *RomaxDesigner*. We have already



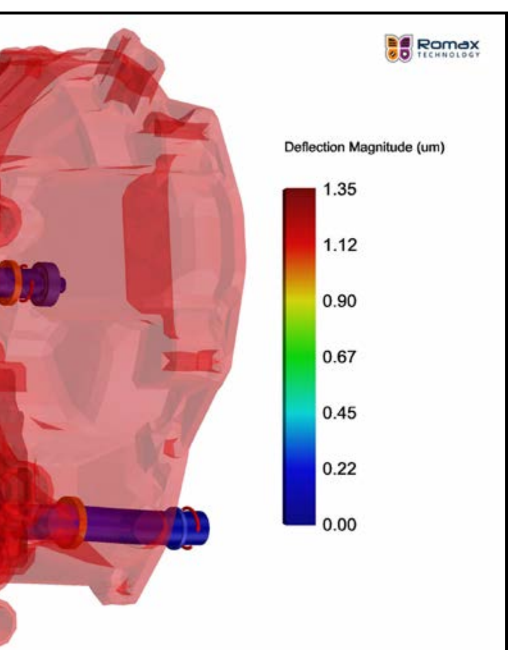
used the software for over five years on multiple projects. The speed and unique system-level simulation which *RomaxDesigner* offers stand it apart from other products currently available on the market."

An integral validation strategy

The strategy developed by Volkswagen focuses on a step-by-step process, allowing correlation between measurement and simulation along the acoustic transfer path at each of the following stages: gear excitation, shaft systems, bearings, gearbox housing, and whole vehicle testing. "If test and simulation are compared only at the end of the system development, then it is not possible to work out where discrepancies may arise, hence the need to perform correlation at each level. This gives us an understanding of exactly where problems are occurring, so that we can resolve validation errors quickly and easily, and avoid time-consuming investigatory work," Schmitt explains. "And Romax software plays a big part in this investigation. Only with *RomaxDesigner* can we quickly and accurately investigate gear whine phenomena on a system level — looking deeper into models to work out where the problems are. This is what allows us to meet high expectations for NVH within even the most cutting-edge system designs. Romax's unique system level view is a huge benefit to us, as well as its easy-to-use bearing catalogues, which make it easy to model gearboxes even if you are not a bearing expert, and its reliable and accurate transmission error calculations."

A step-by-step process

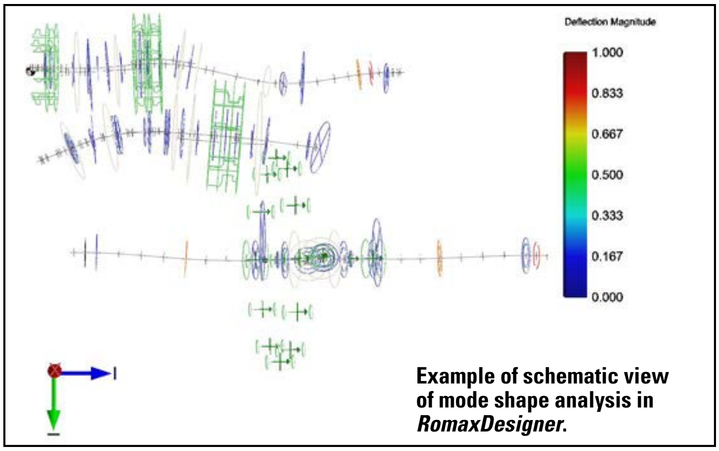
The gears are validated first, with testing and simulation performed across a range of loads. The gear contact pattern is checked; poor correlation indicates either incorrect micro-geometry in the simulation, or deviations in the manufacturing process. The next stage is shaft system validation, which consists of modelling single parts and assemblies, then performing finite element analysis (including pretest analysis and experimental modal analysis, if necessary). This is again validated against test data, and if this is unsuccessful the model must be updated in *RomaxDesigner*. Whenever correlation is not successful, changes can be made which will improve



Example of mode shape analysis including housing in *RomaxDesigner*

the process for the future, as Schmitt explains: "In the first run we did, we found that the model did need updating. The updates that we performed, including accounting for Young's modulus and part-to-part stiffness connections, improved the correlation significantly."

The third stage is correlating the bearing stiffness, and the final step is the correlation for the gearbox housing, for which there are two options, as Schmitt explained: "The validation can be performed by building up the components separately using different tools and testing each individually, and then adding together to make the final model. Alternatively a single model can be created in *RomaxDesigner*, which means just one experimental modal analysis, one correlation analysis, and only one model to update. We found that there was little difference between the methods, so the full housing assembly was done in order to save time and effort — this is a very useful way of doing the correlation."



Example of schematic view of mode shape analysis in *RomaxDesigner*.

"Now we have developed the framework, we are confident that the work that we have put into this implementation will enable time and cost savings for future projects, as well as maintaining our customer's trust in our ability to deliver their requirements," Schmitt concluded. "We have developed a clear strategy to perform straightforward model updating procedures, and extended the validity and trust of our Romax gear whine models. Our design changes are not reliant on trial and error, but are based on proven, trustworthy simulation."

For more information:
Romax Technology
Phone: (248) 220-1201
www.romaxtech.com

Gleason

EXPANDS P-SERIES HORIZONTAL HOBBING MACHINES

Gleason Corporation introduces the P90CD and P90iC horizontal hobbing machines. Now, two new machines from the P-Series enhance the current model line to provide solutions for specific applications which require high productivity and efficiency. Based on the P90 hobbing machine, the new P90CD hobbing machine with an integrated chamfering/deburring module has been designed for disc-type workpieces, like automotive pinions and short shafts.

The P90CD features an integrated CNC chamfering/deburring station which works in parallel to the cutting process; hence productivity is not compromised by the added auxiliary process. The P90CD can hob parts up to a diameter of 60 mm and module 3 mm; larger diameters are available on request. Cycle times are as short as 10 seconds for planetary pinions. The chamfering/deburring station includes an auto-meshing feature which avoids tooth-on-tooth cut workpieces and chamfering tools for consistent cycle times and to avoid damage to workpieces.

The new P90iC hobbing machine with an integrated chamfering/deburring unit has been specifically designed for the hobbing and quality chamfering/deburring of geared shafts, but can be used for disc-type workpieces as well. The P90iC features an integrated chamfering/deburring unit which eliminates burrs and creates even and precise chamfers. The P90iC handles any workpiece up to a diameter of 100 mm and module 3 mm; larger diameters and modules are available on request. The P90iC is an excellent solution to employ one or two-cut processes for finish hobbing or to create a quality base for subsequent hard-finishing operations.

The two-cut process in particular is executed in a single setup (cutting-chamfering/deburring-cutting) and eliminates secondary burrs and residue on the gear flanks. This process is very beneficial for subsequent hard-finishing processes, protecting the tool life of expensive finishing and dressing tools, especially if a honing process is applied. For both machines direct-driven hob head options are available which offer the optimum adaption to many applications. For both machines Gleason provides hobs from state-of the-art materials, with the latest wear coatings and high quality rotary chamfering/deburring tools with extremely long tool life as well as clamping fixtures to secure high machining quality.

For more information:
Gleason Corporation
Phone: (585) 473-1000
www.gleason.com

Hardinge

DEVELOPS LATEST 5-AXIS VERTICAL MACHINING CENTER

Hardinge Inc. has announced the release of their newest 5-axis vertical machining center, the Bridgeport XT 630 5-Axis.

Bridgeport's new generation full 5-axis vertical machining center is a fully digital, high-quality machine tool designed to achieve maximum capacity and performance in the aerospace, mold and die, medical and automotive industries and many other manufacturing sectors. This machine has been developed to provide a powerful and precise solution to meet the requirements of the most demanding metal cutting user.

The machine offers a highly sophisticated yet user-friendly Siemens 840D control with a 19" LCD. Axis travels are X: 762 mm (30"), Y: 630 mm (24.8"), Z: 610 mm (24"), A: 30 to -120 and C: 360

Standard machine equipment includes items such as the Big Plus CT40 spindle, 15,000 rpm direct-coupled spindle with oil chiller, coolant chip flush system, three color stack light, 24 tool swing-arm ATC, through-ball screw chiller, preparation for through spindle coolant (with rotary union), remote MPG hand wheel, auto central grease lubrication and an A-axis encoder.

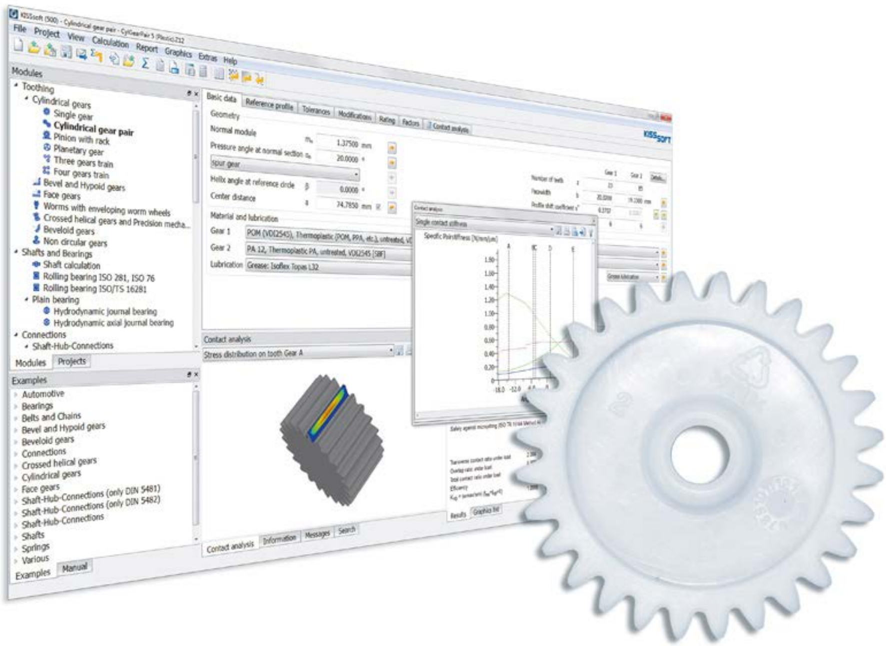
Additional options include through-spindle coolant (280 psi), X/Y/Z linear scale, coolant system with chip conveyor, part probe, tool probe, C-axis rotary encoder, dynamic collision monitoring, tool changer option 40, 48, & 60 and a mist collector.

"The Bridgeport XT 630 5-axis machining center is priced as a productive 5-face (4+1 axes) vertical machining center, although it includes the high degree of sophistication and functionality of the simultaneous 5-axis machining platform it is," states Brooke Sykes, director of sales and customer services North America.

For more information:
Hardinge Inc.
Phone: (607) 734-2281
www.hardinge.com

KISSsoft

OFFERS PLASTIC GEAR SUPPLEMENT



The application area of plastic gears has grown significantly in the last decade. The requirements from the industry to design stronger, lighter, quieter and more efficient gears have also motivated plastic manufacturers to produce custom made materials.

In the last two years, the new VDI 2736 guideline for the design of plastic gears was introduced. Unfortunately, its material data for lifetime calculations is limited. In order to design gears with custom made materials, it seems necessary to measure the fatigue data and temperatures in gear tests prior to the gear design. If you are interested in this topic, this KISSsoft paper

(*http://www.kisssoft.ch/english/downloads/pdf/article_kisssoft_vdi_2736_gear_temperature.pdf*) presents an accelerated testing procedure for plastic gears that is based on different levels of testing. As a supplement to the VDI 2736 guideline, the calculation of plastic deformation and wear of plastic crossed helical gears (according to Pech) has been implemented in the KISSsoft *Release 03/2016.*

For more information:
KISSsoft USA LLC
Phone: (815) 363-8823
www.kisssoft.com

Cortec Corporation

DEVELOPS SUPER BARRIER VERSION OF ITS ECOSHIELD VPCI-144

Cortec Corporation has developed a new Super Barrier version of its EcoShield VpCI-144 paper for protecting metal parts from corrosion. EcoShield VpCI-144 Super Barrier combines the corrosion protection of VpCI paper coating with a high gloss water-based barrier coating that prevents moisture from reaching metal parts wrapped inside the paper. The enhanced moisture barrier of EcoShield VpCI-144 is an excellent environmentally friendly alternative to polyethylene and waxed papers. Under recent ASTM E-96 testing, EcoShield VpCI-144 Super Barrier exhibited a water vapor transfer rate (WVTR) highly comparable to that of polycoated paper.

Past testing has also shown EcoShield VpCI-144 Super Barrier to rival the moisture barrier properties of polycoated paper and commercial waxed paper. This is an important advantage since poly and wax coatings are not recyclable through normal channels and therefore create an environmental problem. In contrast, EcoShield VpCI-144 Super Barrier paper is environmentally safe and fully recyclable into other types of paper products such as boxes, cardboard, and other corrugated materials.

EcoShield VpCI-144 Super Barrier combines corrosion protection, moisture barrier properties, and oil and grease resistivity into one material to protect both ferrous and non-ferrous metals. This eliminates the need to inventory multiple types of papers for different types of metal. Protected metals include carbon steel, stainless steel, galvanized steel, cast iron, aluminum alloys, copper, brass and solder.

VpCIs on the inside face of the EcoShield VpCI-144 Super Barrier paper vaporize and condense on metal surfaces to form a thin protective film that doesn't influence physical properties of most sensitive electrical and electronic components, including conductivity and resistivity. The protective film does not need to be removed prior to further surface finishing or coating application, and protected parts can be painted, welded, or soldered.

EcoShield VpCI-144 Super Barrier is useful in a variety of different applications including metal production, metal forging and die casting, metalworking, finished products and electrical and electronic products.

For more information:
Cortec Corporation
Phone: (651) 429-1100
www.cortecvci.com

Sumitomo Electric Carbide

OFFERS COOLANT THROUGH MODELS FOR WDX DRILL SERIES

Sumitomo Electric Carbide, Inc.'s WDX Series large diameter drill line now includes WDX-L coolant through models. Reducing the cost per part, the WDX-L Indexable Drill features side port coolant. Because not all spindles are coolant-thru, the WDX-L allows for the use of coolant through the side port. WDX-L diameters range from 0.5625"–2.5000" in 4XD drill body lengths. The WDX-L uses the same inserts as Sumitomo's conventional WDX Series (WDXT inserts). Diameters of the conventional WDX range from 0.5625"–2.625" in 2XD, 3XD and 4XD drill body lengths. The WDX Series is available in inch and metric sizes.

Sumitomo's vast selection of WDX and WDX-L bodies deliver a proven design for stable drilling. All drill bodies come with a four-cornered indexable insert design to provide a cost effective drilling solution and easy tool management. The series uses two inserts per drill, regardless of diameter size. Insert grades for a longer tool life include ACP300 for steel, ACK300 for cast iron and DL1500 for aluminum.

For a limited time, the WDX and WDX-L are available at special IMTS promotional savings. Through December, Sumitomo is offering kit pricing (including the drill body, 10 inserts, and a $10 gift card) and buy two drills get one drill free.

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

Mitutoyo

RELEASES LATEST GENERATION OF LINEAR SCALE

Mitutoyo is pleased to announce the Absolute AT1100, the latest generation of assembly-type linear scales that incorporate Mitutoyo's electromagnetic induction technology, which provides resistance against contaminants such as cutting fluids, oil and water. The ABS AT1100 was featured in the Mitutoyo booth during IMTS. The innovative shape and location of the detector track in the aluminum frame provides a highly effective defense against the contamination of the scale and sensor, even in harsh environments. The sensor-to-scale air gap is approximately 0.4 mm, about 4x as wide as that of conventional optical or electromagnetic induction systems,

thereby providing additional protection against dust or oil contamination. ABS AT1100 Linear Scales are available up to an effective range of 3,040 mm and are compatible with FANUC Corporation's serial interface and Mitsubishi Electric Corporation's high-speed serial interface. ABS AT1100 series scales are a special order item.

For more information:
Mitutoyo America Corporation
Phone: (630) 820-9666
www.mitutoyo.com

PTG Holroyd

RELEASES CUTTER GRINDING MACHINE

A newly developed 'large diameter' CNC cutter grinding machine from Holroyd Precision Ltd. is all set to bring considerable advantages to organizations that wish to achieve even greater levels of accuracy and repeatability when finish milling some of the world's largest helical rotors.

Called the CS700E Tool Management Center, the new machine has been designed especially for the grinding of high-accuracy profile forms on finish milling cutters of up to 700 mm in diameter, and follows on from the company's highly successful CS500E (500 mm max. diameter) model.

"The new CS700E machine is generating considerable interest," comments Holroyd Sales Manager, Mark Curran. "Indeed, a major compressor manufacturer that has only recently purchased its second Holroyd 8EX rotor milling machine (for producing rotors of up to 850 mm in diameter), has already placed an order for a CS700E."

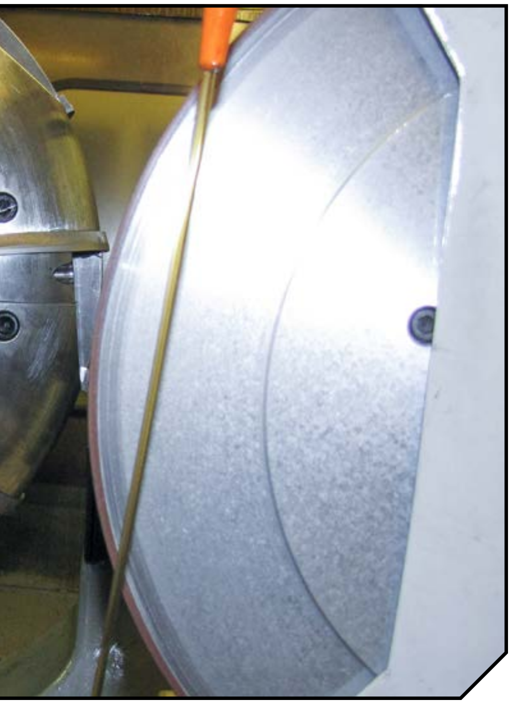
Curran says that the accuracy of any helical screw form relates directly to the accuracy of the cutting tool that is used to create it. "Through the development of the CS700E Tool Management Center, we are enabling producers of larger helical components to benefit from complete control of all aspects of the cutting tools they use. The CS700E produces high accuracy profile forms on a wide range of tool materials, from traditional high-speed steel, to exotic materials such as carbide and 'AS' finish machining systems," Curran said.

The CS700E Tool Management Center incorporates automatic dressing stations to ensure optimum integrity of the grinding wheel at all times. The stations are equipped to dress aluminum oxide, CBN and diamond grinding wheels, while use of HSK-A160 arbors (as fitted to 8EX rotor milling machines), enable rapid wheel changeover times and further enhance accuracy.

At the heart of the CS700E is Holroyd's user-friendly advanced touch screen programming system. This provides complete control over production accuracies, as well as invaluable on-screen modification of profile forms for the rapid evaluation of new profile shapes during the development and prototyping stages. Profile modifications are easily made thanks to the menu-driven system, which provides control of the cutting profile, with on-screen modification of profiles.

Automatic touch probing of the tool profile during the grinding cycle is another major benefit provided by the CS700E. Following probing, the actual tool profile is graphically superimposed over the theoretical profile and displayed within user-defined tolerance bands to ensure that all cutter blades are ground within tightly controlled limits. Indexing and multi-pass grinding/trimming cycles are also fully automated.

For more information:
PTG Holroyd
Phone: +44 (0) 1706 526 590
www.holroyd.com



Managing Shop Floor Data

First Gear Utilizes New Tooling and Machining Processes with Assistance from KISSsoft

Matthew Jaster, Senior Editor

First Gear utilizes KISSsoft software to verify robust design principles to determine the proper cutter to produce the desired tooth profiles.

There's no substitute for a good software package in gear manufacturing. It's a critical shop floor tool that provides practical engineering services that customers appreciate. When you're in the business of specifying and procuring high quality gears, the software needs to meet many objectives including the consideration of all tolerances of center distance, tooth thickness and tip diameters, root diameters, fillets, etc. It's also imperative that the software updates include the latest revisions to the gear standards being used in the industry.

Many companies, including some of the largest aerospace customers that use gears, may not always have extensive exposure to gear design or gear manufacturing processes today, according to Greg Leffler, president, First Gear Engineering & Technology.

"Our customers often need help with their designs and getting the proper information on their prints to guarantee what they order is correct and will function as desired," Leffler said. "This requires us to use *KISSsoft* software every day for verifying robust design principles to determine the proper cutter to produce the desired tooth profiles."

One of First Gear's medical customers is borrowing technology from the gear industry to produce a specific medical device. Some of the device's components are being manufactured on First Gear's CNC gear machinery. Although it is a non-standard design, First Gear employs *KISSsoft* software to provide solid models that can further be evaluated in FEA and other analyses.

"We are able to design new tooling and machining processes that have not been done before (to our knowledge) to produce these components," Leffler added. "KISSsoft has been a huge help in this program."

A Quality Start

First Gear serves many industries including medical, aerospace, military, motors and drives, automation, food and general commercial. The company offers services such as hobbing, shaping, finish grinding and finish rolling of gears and splines and provides assistance in the design and development of prototypes to hundreds of thousands of gears annually.

An emphasis on quality began from day one, when the company took delivery of an M&M Analytical Gear Inspection Machine.

"We continually invest in new equipment to advance our capabilities. We have state-of-the-art gear inspection and analysis equipment from Penta Gear (formerly known as PECo and now part of the Kapp-Niles group) and Pratt & Whitney. We have been AS9100C and ISO 9001:2008 certified for many years and this year we received our accreditation for ISO 17025 to re-calibrate master gears, splines and gages, and we are seeing business for this service from many industries. This is another segment of our business where *KISSsoft* software is helpful," Leffler added.

"When we need answers, we can talk to KISSsoft engineers immediately and have been impressed with their product knowledge and response time. We have found their software to be very intuitive and user friendly, it is updated frequently and it is very simple to manage," Leffler said.

New Software Solutions Equals New Business

The solution KISSsoft provided for the medical customer is out of the ordinary in gear manufacturing and has allowed First Gear a unique opportunity to broaden the scope of its manufacturing services. For example, the First Gear staff doesn't claim to know everything there is to know about gears, but they are blessed to know a handful of people who have spent their entire lives in this industry and they can go to these people for answers when needed.

"These engineers have had exposure to almost all types of gearing and have provided a wealth of information when needed," Leffler said. "We have used other commercial software packages, but KISSsoft is the first to provide an all-encompassing package that includes solid modeling and other features that have not been available to us before."

In addition to the current standards found in the software, First Gear engineers were able to design non-standard gearing using the standards as a guideline. The flexibility of the software allowed them to do several "what-if" designs in a very short period of time.

The more comfortable the engineers get with the software solutions, the more opportunity they have to utilize it in other areas. The software is used on a daily basis at First Gear when reviewing the accuracy of data on prints that they are quoting. Most all international standards are available in the software which is very useful today.

"We see prints from all over the world. We want to confirm that all the data is correct and our customer will be able to use their gears as they intended. We also work with customers on new designs, often providing a complete analysis," Leffler added.

KISSsoft has allowed First Gear engineers to evaluate many design options for its customer's applications and select the best solution that minimizes the potential for noise, heat generation and tooth wear. "The detailed manufacturing evaluation that follows allows us to either design the proper tool or choose an available tool and calculate the details of the final product. This saves our customer time, money and provides them quality, confidence and assurance," Leffler said.

Training

The tutorials present within the software and the help from KISSsoft's engineering support staff makes it easier to learn the ins and outs of the software package. Another great reference tool is the instructional videos KISSsoft produces online.

"Attending one of KISSsoft's training programs is another great way to get deeper into utilizing more of KISSsoft's capabilities through class design exercises," Leffler said.

First Impressions

In an ever-competitive gear market, First Gear wants to be the first choice when it comes to specifying and helping buyers and engineers in procuring gears. The right software package, one that keeps up with the faster and more efficient pace in manufacturing today, significantly helps the organization meet its daily objectives.

First Gear expects to more than double the size of its current facility in the next year (the company wants to expand to offer complete design and manufacturing capabilities). They are also looking to acquire or build other gear facilities where opportunities exist.

"We've had the luxury of utilizing several CNC machines since day one. We have a database of all our hobs, shaper cutters and their exact geometries, material properties and coatings available," Leffler said. "This information, coupled with *KISSsoft* software, confirms that we can produce the geometry the customer prints specify and in turn makes it quick and accurate to quote jobs. This gives us the confidence that we are doing everything possible to make our customer's job easier."

For more information:

First Gear
Phone: (260) 490-3238
www.first-gear.com

KISSsoft USA LLC
Phone: (815) 363-8823
www.kisssoft.com

Latest KISSsoft 03/2016 Updates

The latest KISSsoft *Release 03/2016* debuted earlier this year at the Industrial Fair in Hannover. This version included enhanced evaluation of planet carrier deformation, bevel gear contact analysis, and the following improvements:

Plastic Gears

The application area of plastic gears has grown significantly in the last decade. The requirements from the industry to design stronger, lighter, quieter and more efficient gears have also motivated plastic manufacturers to produce custom made materials.

In the last two years, the new VDI 2736 guideline for the design of plastic gears was introduced. Unfortunately, its material data for lifetime calculations is limited. In order to design gears with custom made materials, it seems necessary to measure the fatigue data and temperatures in gear tests prior to the gear design. As a supplement to the VDI 2736 guideline, the calculation of plastic deformation and wear of plastic crossed helical gears (according to Pech) has been implemented in the KISSsoft *Release 03/2016*.

3D Modeling

Modeling in *KISSsys* has been simplified in the latest KISSsoft *Release 03/2016*: Now, for example, when elements are added, the part geometries are prefilled with default values. At the same time, the shafts are positioned intelligently, to suit the gearing types involved, such as cylindrical gear pair, bevel gear or planetary stage. The user can now see the modeling progress immediately in the 3D view. Another new feature is the option of adding assemblies (such as planetary stages) to a model, and also adding shafts, if required.

Roller Bearings

The new version now contains the very latest data from the "Rolling Bearings" catalog. By cooperation between SKF and KISSsoft, all future releases of *KISSsoft* will include an updated bearing database from SKF. The previous *KISSsoft* database contained 4,000 SKF bearings — the new release contains 11,500 bearings as shown in the latest catalog. In case of major changes to the catalog by SKF, KISSsoft can deliver a new database to the customer with patch files. This collaboration ensures that *KISSsoft* users are able to design gearboxes using up-to-date bearing range and bearing catalog data. (*www.kisssoft.ch*).

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Innovating Against the Tide

Despite the soft market, suppliers are providing more and better ways to manufacture splines.

Alex Cannella, News Editor

During a year with a strong dollar, tanked oil prices and a number of soft markets that just aren't buying, one might expect spline manufacturers to be experiencing the same tumult everyone else is. But when I got a chance to speak with some of the suppliers to spline manufacturers at IMTS about how business is going, many of the manufacturing industry's recent woes never came up, and instead were replaced by a shrug and an "eh, business is doing pretty well." That casual optimism might have been influenced by an IMTS that was, all things considered, a success during a quiet year for industry growth (the third largest IMTS ever, and with more exhibitors than any previous year, to boot), but there are other signs of growth as well, most notably the fact that people are still coming up with new products instead of setting up camp for the winter.

Dry Broaching from Across the Sea

One of the biggest events in the spline industry that's happening right now is a partnership between Broaching Machine Specialties (BMS) and Ekin, an event catalyzed by their initial meeting at 2014's show. According to Matt Egrin, president of BMS, the two companies quickly took a shine to each other.

"Ekin's a large company," Egrin said. "They've got operations in five different countries, however have no facilities in the US. So when they came to the show two years ago, they came looking for somebody to partner with. And we met them, and immediately on a personal level, had a very good rapport with their personnel and decided that we like each other and maybe we should dance together."

Though they've been working together for a little over a year, Ekin and BMS's partnership is only just now coming together. This year's IMTS was BMS's first time showing off Ekin products, and Egrin says that he's seen a positive reception to them.

"This IMTS is kind of the formal coming out party of our agreement, if you will," Egrin said. "We've got some very good interest already."

With the new partnership in place, BMS will be supplying machines made by Ekin, which is based in Spain, to the US market. Most notable amongst those products is the Rishem-10X1250X400 dry broaching machine.

Though dry cutting has been a part of the industry for over a decade, dry broaching itself has only come about in the last few years. The technology behind the Rishem is only a few years old for the broaching community and not yet widespread.

"Ekin, to my knowledge, is only one of two companies in the world that have this technology," Egrin said.

The Rishem is a table up internal dry broaching machine. Instead of using coolant to keep temperatures down, the broach tool utilizes a special coating that dissipates heat and ensures chips don't stick to the workpiece. It also uses a vacuum system to suck chips away from the workpiece in much the same manner coolant washes it away.

Much like its dry cutting counterparts, dry broaching offers significant advantages over standard coolant-driven methods. There's the obvious benefit of not having to spend money on coolant or any other related costs, such as the draining and disposal of said coolant. Beyond saving on coolant costs, Ekin's Rishem also boasts a cutting speed of 40 meters-per-minute, four times faster than that of standard broaching systems.

The Rishem won't be the only Ekin product BMS will be selling going forward. They will also be selling all of Ekin's broaching machines that don't already overlap with their own products, including hard broaching, helical broaching and spline rolling machines.

"Right now, we're just acting as sales

BMS is now selling a number of Ekin's broaching products, including the dry broaching machine pictured here.

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2301 CURTISS STREET, DOWNERS GROVE, IL 60515 | 630-969-7640 | ARROWGEAR.COM and service for the machine and tool product lines," Egrin said. "But the long term goals, hopefully within the next 12-18 months, are going to be to open a JV operation near BMS in Detroit for the manufacture of broach cutting tools and service of broach cutting tools and spline rolling racks."

Establishing the Standard

Another relatively new product spline manufacturers can take advantage of is Vargus's stock milling cutters. In the four years they've been on the market, Vargus has expanded production of their stock cutters twice, with a third expansion in the works, just to keep up with demand.

"When this product first came out, we were quoting 10 a month," Joe Magee, product manager at Vargus, said. "Now we're up to 400 a month."

Vargus's stock cutters were inspired by necessity. As demand increased for their custom cutters, the company found their lead times stretching longer and longer, going from weeks to months. In an effort to cut down on lead times, the company established their line of stock cutters, which could still handle many of their orders while decreasing turnaround time.

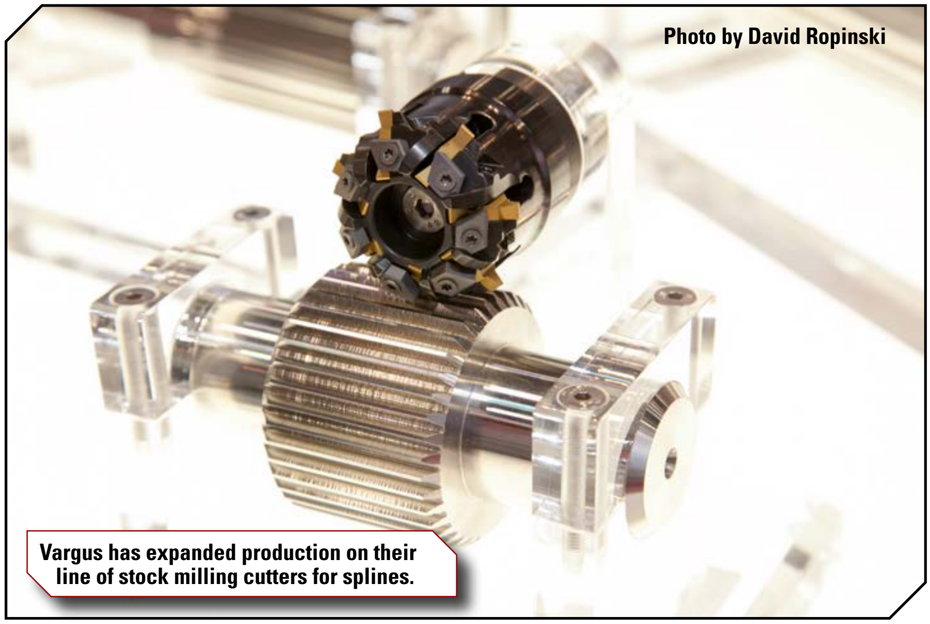
Photo by David Ropinski

Vargus has expanded production on their line of stock milling cutters for splines.

"Lead time went from six to eight to 10, sometimes 12 weeks. And the customer would then wait 12 weeks to get their tool…" William Jubran, mechanical engineer and marketing and business manager at Vargus, said. "To improve service, to improve support to the customer, we have tools that we carry as standard tools in stock in the States."

"You don't need to recreate the wheel any time somebody wants something," Magee said. "So that's why we're going to standard."

The system enables customers to decide what spline dimensions they need for a particular job, then select the proper cutters needed to produce what they're looking for. Vargus's stock cutters can cut most standard splines up to 6 inches in diameter. According to Magee, the system can cover 60 percent of splines manufactured today in America.

"It doesn't replace all broaching," Magee said. "If you have to come up to a sharp corner, you still have to broach it, but 65-70 percent of [spline jobs] do not include a sharp corner. When you want to reduce your manufacturing costs, you can go to a standard milling process and reduce your costs by 30, 35 percent."

Beyond the convenience of a system of standardized cutters, Vargus's cutters themselves boast some advantages for spline manufacturers such as lower setup costs and full profile grinding. The cutters are also accurate, reaching DIN 3962 Class 7 tolerances. The system also puts an emphasis on simplicity and ease of use with simple programming and design.

"This design is so simple, you can run it just on a standard milling cutter with an indexing hand..." Magee said. "You can buy one body and run 20 different spline sizes with that one body by buying stock standard inserts."

The automotive industry has been the primary field to take advantage of Vargus's standard milling cutters. Aerospace and defense manufacturers, meanwhile, have stuck with Vargus's custom cutters, which they will continue to provide alongside their new stock line.

Practicing the Basics

While some spline manufacturers' suppliers aren't feeling the effect of market forces such as the cost of oil or the strength of the dollar as keenly as other companies might be, they are wrestling with the same ever-present, industry-wide difficulties that have dogged the industry for years: tightening tolerances and a lack of talent in the workforce.

The General Broach Company is a good example of how to deal with those nigh-existential problems. They've managed to keep up with demands for class 5 and even class 4 splines, meeting tolerances of up to .0002 inches, and their process for doing so is surprisingly straightforward.

"You either have to upgrade your equipment, or you have to upgrade the process," Larry Stover, general manager at General Broach, said.

Upgrading your company's capabilities to meet increasing demand is a simple, almost no-brainer plan in theory, but also easier said than done. It's easy enough to buy a better, more accurate machine or find somewhere you can streamline your process, but all of that requires talent, people who know how to use the machines you're buying, and that puts your company on a collision course with existential problem number two.

"Just like everybody else, we find the same problems," Stover said. "There are not enough trained, talented people to be able to fill all the positions that's needed. In our business, it takes a good

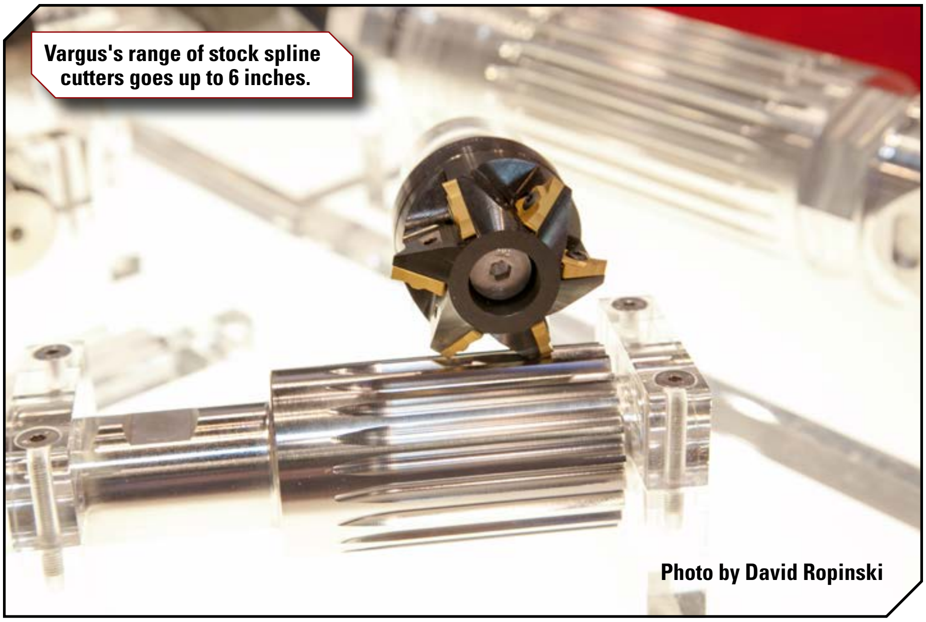
Vargus's range of stock spline cutters goes up to 6 inches.

Photo by David Ropinski

year to two years, maybe sometimes even three or four years to get somebody trained that can handle that. It's more than just pushing a button on a machine. You have to understand what's behind that move that that machine is making."

General Broach is taking multiple routes to deal with the skills gap, focusing not only on training the future generation, but also on retaining their current talent.

On the training front, General Broach is working alongside three colleges and 30 manufacturing businesses in Lenawee County, Michigan to build curriculums for high school students and get them interested in the industry. The program is gaining momentum and expanding, and has recently added mechatronics classes.

"We're getting kids interested in manufacturing, and that's what we need," Stover said. "That's the future."

General Broach also maintains common sense tactics to retain their employees. They make sure pay's competitive. They continue to offer training for their employees. They focus on providing a family-oriented work culture.

"We don't want them to feel like they're in a dead end position," Stover said. "We don't want them to ever feel that way. Because the problem is, if they end up feeling that way, they move on. And then you've lost the knowledge that they had."

Like a lot of General Broach's solutions, their method for retaining their workforce sounds like standard, common sense stuff, but it's all too easy to neglect the basics of maintaining a healthy business, particularly when the clamps are on and business is tight as it is now for some manufacturers. Just like with our basics articles on more technical topics, sometimes it helps to be reminded of the basics, and General Broach's even-keel, steady mastery of them provides a good example of how to properly nurture your business.

Steady and even-keel are, in fact, good words for describing spline manufacturing as a whole right now. We may have had a soft year so far, but for spline makers, it's been business as usual, and business has been good. They've had a good IMTS, and they continue to move forward, slowly but surely.

For more information:

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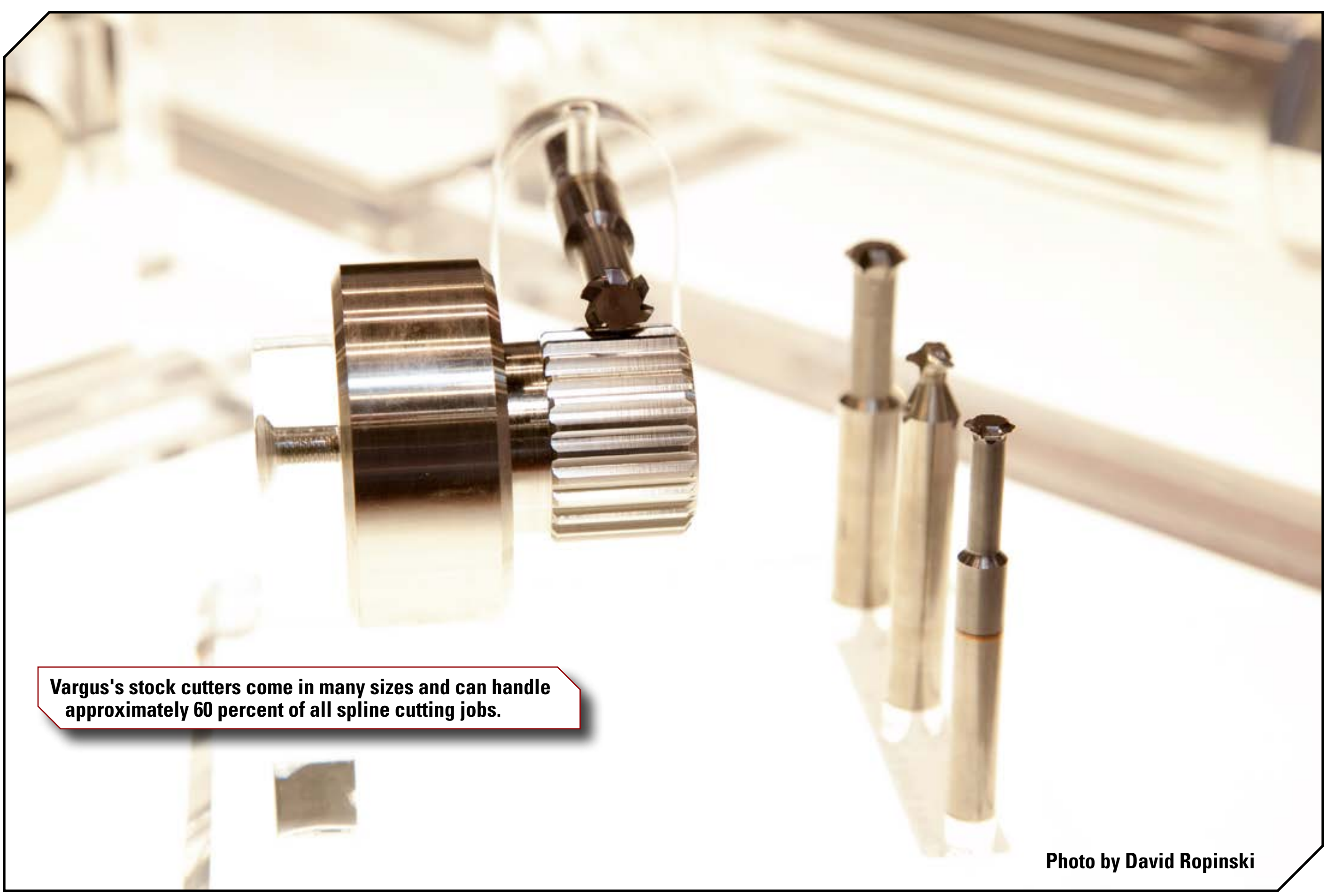

Vargus's stock cutters come in many sizes and can handle approximately 60 percent of all spline cutting jobs.

Photo by David Ropinski


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AGMA's Go-To Gear Guys

Ray Drago and Bob Errichello Discuss Their Tenures as AGMA Gear Instructors

Jack McGuinn, Senior Editor

"He who can, does; he who cannot, teaches."

So goes the pithy George Bernard Shaw observation.

And then you have gear guys like Bob Errichello (Geartech — *geartech.com*) and Ray Drago (Drive Systems Technology (DST); *GearDoctor@Verizon.net*), who have been teaching *and* doing for many years. Indeed, collectively they total roughly a century's worth of in-the-field gear technology experience as well as unparalleled classroom instruction. And while they have or have had at one time or another other teaching gigs (U. Wisconsin Madison-Drago; U. California-Berkley-Errichello), it is their extended runs presenting AGMA gear courses that get the most attention here. Drago's and Errichello's AGMA courses continue to be the most highly attended (or viewed) classes available of AGMA coursework, and attract worldwide interest.

While the two have taught a variety of AGMA courses over the years, without question their most popular courses are *Gear Failure Analysis* (Errichello with longtime colleague Jane Muller) and *Gearbox CSI: Forensic Analysis of Gear & Bearing Failures* (Drago). Drago currently teaches *Manufacturing & Inspection* (with AGMA instructor Joseph W. Lenski, Jr.) and *Gearbox System Design: The Rest of the Story… Everything but the Gears and Bearings* (with AGMA instructor Steve Cymbala) as well.

Please note the remaining AGMA instructors giving their valuable time and priceless expertise to students include: Allen Bird; Roy Cunningham; Steve Cymbala; Peter Grossi; Jane R. Muller; and Dwight Smith. In addition, AGMA's gear school at Chicago's Daley Community College continues to provide invaluable instruction to those in or seeking to enter the gear industry. (Go to *agma.com* for more information regarding these folks and the important work they do.)

With that, we were able to slow down Drago and Errichello long enough to answer some questions for us regarding their long tenure and the material they teach.

Gear Technology (GT). Which is your most popular class, and why do you think that is the case?

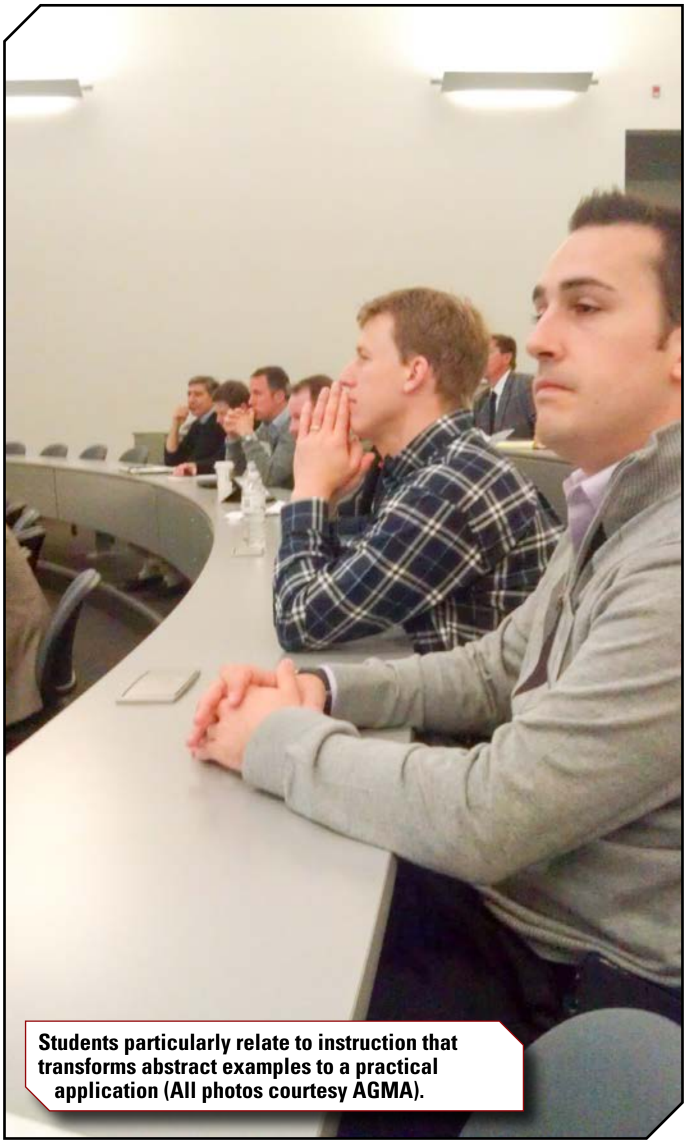
Bob Errichello (BE). The AGMA Gear Failure Analysis Seminar is by far AGMA's most popular seminar. It has been taught twice a year for the past 26 years and is always sold out. It is the premier course because it was carefully constructed, using the best teaching techniques and directed to the students' needs.

Ray Drago (RD). That is a tough question. I teach or co-teach more than a dozen courses, however, the mainstays of the group are three that I teach at the University of Wisconsin and seven that I teach for AGMA on a regular yearly schedule. Of these two groups, it appears that the AGMA Detailed Gear Design and UWM PC Applications in Parallel Axis are the most popular in each grouping.

GT: How do you update your course over time to keep abreast of today's technology and sophisticated "smart" requirements?

BE: We continuously revise the course based on student evaluations and the latest achievements in our knowledge gained from our consulting practice.

RD: We have a very active consulting practice that involves gear systems used in everything—from electric toothbrushes to space craft and just about anything in



Students particularly relate to instruction that transforms abstract examples to a practical application (All photos courtesy AGMA).

“We, the citizens of each school district, must pressure our school boards to include shop or, more politically correct — ‘technology’ classes — back into the mainstream curriculum.”

Ray Drago

between. We use this experience to generate “examples” that we use in our course presentations. So as we gain new experiences we add or update material through these real-world, sometimes very unusual, experiences. We find that the students react with extreme interest when we can relate a “dry” fact to an actual application, especially if it is an unusual one. We have a two-page comment sheet that we hand out with the printed notes for every course. We guarantee return of this comment sheet from every participant by holding their Certificate of Completion “hostage” until each person turns in a comment sheet on the last day of the seminar. This sheet covers everything from meals and facility to the applicability of the course material to the participant’s work and the level of the seminar presented. It also has a section for suggestions regarding (new) topics.

GT: What pre-qualifications are required of the students?

BE: We don’t insist on pre-requisites, but recommend the AGMA webinar on gear metallurgy.

RD: My AGMA courses are generally single topic, in depth, focused, and require a good solid basic gear technology understanding. My AGMA “Gear Materials: Selection, Metallurgy, Heat Treatment, and Quality Control” course is an in-depth treatment unavailable anywhere.

GT: What is the general age demographic of the class?

BE: All ages attend, starting with new hires through retirement-age engineers. Often, young, non-degreed students do so well that they are awarded prizes for their achievements.

RD: The class makeup is very diverse age-wise, ranging from young engineers typically out of college a few years to much older engineers who have many years’ experience, but not necessarily in gear technology. We also see folks who have lots of gear technology experience, but in a limited area (e.g., parallel axis gearing but not bevel gearing, etc.). Since the Design Systems Technology (DST) group’s experience ranges over almost every type and size of gearing, our group is a good resource for transitional technology. In addition to my “adult” classes, over the last 10 years or so, I have also been teaching classes to school kids as young as preschool! These are, beyond any doubt, my very favorite classes.

GT: What significant differences exist between now and when you began teaching these courses decades ago?

BE: We started with a one-day seminar, increased it two days, and finally settled on three days. All these changes were based on student input from their evaluations. We added a written test, a practical test, and a case history. The practical test and the case history include student presentations of their failure analyses.

RD: Without a doubt—presentation technologies. When I started I used overhead transparencies! I had some "video" that I used which was on 16 mm reel-to-reel film and the printed handouts were all simple black and white copies of the transparencies. Today, everything is on my computer via PowerPoint presentations in full color, using digital projectors. Several years ago, we committed to bringing the printing of our course handout materials in-house. Although we choked on the price of the printer, it actually paid for itself in about one year in savings from outsourcing this expensive part of any class. We have also done some webinars and one video (for AGMA).

Another major change over these years has been the appearance of women at my courses! When I was in college there were virtually *no* female engineering students. Similarly, when I first started teaching there were *no* women in any course. Today, virtually every course has at least one woman and usually several in the group. In my courses for little kids I take special pains to talk directly to the girls (without singling them out of course) so that they simply grow up thinking that engineering is one of many career choices for them.

GT: What has stayed pretty much the same?

RD: I know it sounds like a cliché but I very much enjoy teaching and courses such as these allow me to have the satisfaction of teaching and the thrill of "passing it on." This has remained the same. Gear system technology is part of just about all mechanical engineering programs, but the treatment is almost universally very superficial—typically, a few one-hour class sessions in total. A skilled gear technologist (will also) gain a vast amount of the required knowledge by virtue of experience and association.

BE: The teaching technique; I spent many years as a student in undergraduate and graduate courses, and carefully selected the best teachers. This experience, together with my own teaching experience as adjunct professor at San Francisco University and the University of California at Berkeley, allowed me to hone my teaching skills to what is most effective for student learning.

GT: What have you had to do to keep current with today's technologies in order to teach your classes? Like, how do you keep current with all the machinery updates, and bearings improvements, to name just a couple?

BE: I gain state-of-the-art knowledge from my consulting practice of failure analysis. Because failure analysis requires expertise in diverse disciplines such as stress analysis; kinematics; dynamics; material science; solid mechanics; tribology; applied lubrication; condition monitoring; manufacturing technology; and design of gears, shafts, bearings, and housings, it is a continuous learning experience. In other words, the best way to learn something is to try to teach it.

RD: Easy one! Earlier in my career, when my "primary" employer was Boeing, I was in a new group (whose) focus was gear system technology development and application to the aerospace environment. This allowed us to work on a variety of both research and design development projects, each of which provided somewhat unique insight and technology development. As our consulting practice grew, and certainly now that consulting and teaching are our two sole "occupations," those varied consulting projects provide all the current knowledge updating that is needed to keep current.

GT: Are today's classes any harder to "reach" than say, 10-15 years ago?

BE: Not at all; I find students are entirely attentive, and they continually impress me with their insights and cleverness.

RD: I would not say that today's students are "harder to reach," but today's students certainly have much higher expectations regarding the quality of the presentations at each session, especially the visuals. PowerPoint is a great asset in this regard; adding videos and, depending on the course, in-class workshop-type problems add to the "drama" of today's

> "We use many techniques to ensure the student absorbs the material so that when they take their written and practical tests they nearly always "ace" them. If we have a struggling student, we give him or her extra assistance."
>
> **Bob Errichello**

tech-savvy engineering student. Today's student also wants the opportunity to interact with the instructors, not just in class but also informally, outside of the formal class setting. In our classes, for example, we take questions during the presentations but we also take frequent breaks during which we can interact individually with each student. We have found that many students will not ask a question out loud in class but will do so while holding a cup of coffee just standing around. We also have lunch together and we try to sit at a different table each day to foster individual discussions.

GT: What else would you like to see happening in gear education in the industry?

BE: I would like to see gearing seriously taught at all schools including, high school, technical colleges, and universities—programs similar to those in European schools. Furthermore, there should be more emphasis on STEM programs for young women.

Ray Drago holding forth at one of his AGMA gear education classes.

RD: I'd say that better management support of younger engineers' desire to obtain practical knowledge offered by courses such as ours (and many others as well!) is a key element. In conversations with many students I sometimes hear that they were "sent" to the class by their supervisor or manager, but more often I hear stories of the challenge it was to get management to approve the class. During major meetings, in conversations after I have presented a formal technical paper (not a course), I often hear management types lamenting how difficult it is these days to hire skilled gear system technologists. These are often the same folks who are reluctant to approve or, even better, advocate, attendance at courses such as ours so that they can train their own people. I find this dichotomy most astounding.

GT: Employer-financed apprenticeship programs are regaining some traction here in the States — a good thing? Any drawbacks?

RD: Unfortunately, I think that the word "apprenticeship," when associated with a beginning engineering career here in the U. S., often has a somewhat negative connotation, thus gaining traction may be difficult. A better approach may be to encourage field and shop experience that is directly related to the engineer's basic job. I do very firmly believe, however, that hands-on experience is an excellent teacher. In fact, every course that I teach includes this "guidance" as part of the introductory material.

BE: Apprenticeships are a wonderful way to stimulate interests in gearing. Furthermore, work-study programs and summer employment for young engineers should be encouraged.

GT: Which of your courses do you find the most difficult to teach — and is that because the material is the most difficult or is it something else?

BE: I only teach what I know, so it is never difficult. And always a very enjoyable, learning experience.

RD: I don't find any course more difficult to teach than any other. I do find, however, that each class has its own set of requirements, including presentation style. For example, the PC Applications in Parallel Axis Gear System Design course is actually mostly a workshop. I lecture for only about 3 or 4 hours on the first day, while the remaining 2.5 days are devoted to guiding the class through their efforts in working out "problems" of increasing difficulty. Each student has a copy of our gear analysis/optimization software (*PowerGear*) and the class typically breaks into self-defined work teams of two or three to work on the problems. A typical problem would deal with a large internal slewing gear driven by several pinions as would be used in a wastewater treatment plant clarifier system.

> "The best way to learn something is to try to teach it."
> **Bob Errichello**

GT: What happens if someone fails your course? What are their options? How does it affect their job prospects, if at all?

RD: None of my courses is actually graded and there are no tests. Each participant who completes the course is awarded a "Certificate of Completion," indicating the number of contact hours and the subject.

BE: In 26 years of teaching the AGMA Gear Failure Analysis Seminar, I've never had to fail a student. We use many techniques to ensure the student absorbs the material so that when they take their written and practical tests they nearly always "ace" them. If we have a struggling student, we give him or her extra assistance.

GT: On the other hand, how beneficial is a Certificate of Completion from your course in finding work in the gear industry?

BE: Most the gear industry knows about the AGMA Gear Failure Analysis Seminar, so a Certificate of Completion is valuable for applying for a job in the gear industry.

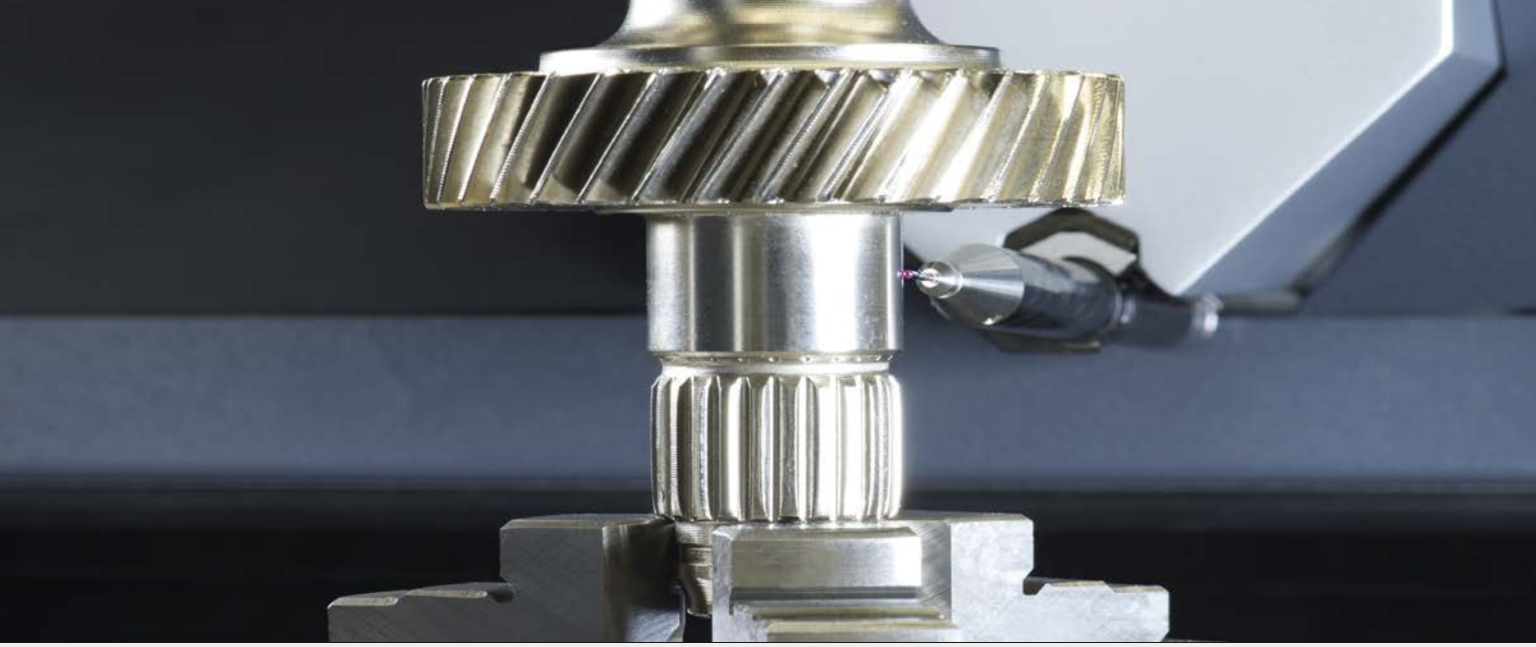
RD: We have received feedback that the Certificates of Completion are valuable "chips" in the annual evaluation process, and a number of students have indicated that their raise was positively affected by their earning the certificate. Of much greater impact, however, is the AGMA-awarded "Advanced Gear Engineering Certificate" (requires taking five AGMA-sponsored gear technology courses). Feedback from students indicates that these certificates carry significant "influence" during annual performance reviews. AGMA has also taken the educational movement to heart, having created the "Advanced Gear Engineering Academy." Though we have worked with the new AGMA Education Director (taking over for recently retired Jan Alfieri) **Cassandra Blassingame** for just a month, it is clear that she has a good handle on the task of continuing the advancement of educational opportunities within the AGMA umbrella.

GT: What—if anything—can be done to bring "shop" classes back to the high schools?

Errichello colleague Jane Muller (partially obstructed) sharing some nuts-and-bolts thoughts with a student.

BE: It will take a major restructuring of the educational system in the U.S.

RD: We, the citizens of each school district, must pressure our school boards to include shop or, more politically correct—"technology" classes—back into the



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mainstream curriculum. We are becoming a generation of virtual experiences with smart phones and computer-generated materials taking the place of real hands-on experience. This *must* change. *Hands-on*, not virtual, computer world experience starting at the earliest possible time is essential to the development of great future engineers, both in the gear technology field and in general.

GT: Silver-lining the fact that a great many experienced workers have retired or are retiring, might there now be a ready reserve of recently pensioned gear guys willing, able and available to teach shop-type courses — even if the courses are not accredited?

RD: Ah — good question. It is not nearly as simple as it appears, to be an effective teacher. In addition to the basic requirement that the teacher be knowledgeable, he or she must also be able to relate to the student and convey information in a way that the students can understand and with a delivery that will cause the students to actually remember what is taught. All of my teaching efforts in the schools are completely uncompensated. In fact, our company, DST, underwrites all costs associated with these presentations. This includes purchasing prizes used during hands-on experiments and exercises to reinforce the concepts taught, and all materials required for the presentation, including color-printed handout materials of the same quality as those provided for our engineering courses.

BE: I believe so. Certainly I would be willing to teach such courses.

> "These (management types) are often the same folks who are reluctant to approve or, even better, advocate, attendance at courses such as ours so that they can train their own people. I find this dichotomy most astounding."
>
> **Ray Drago**

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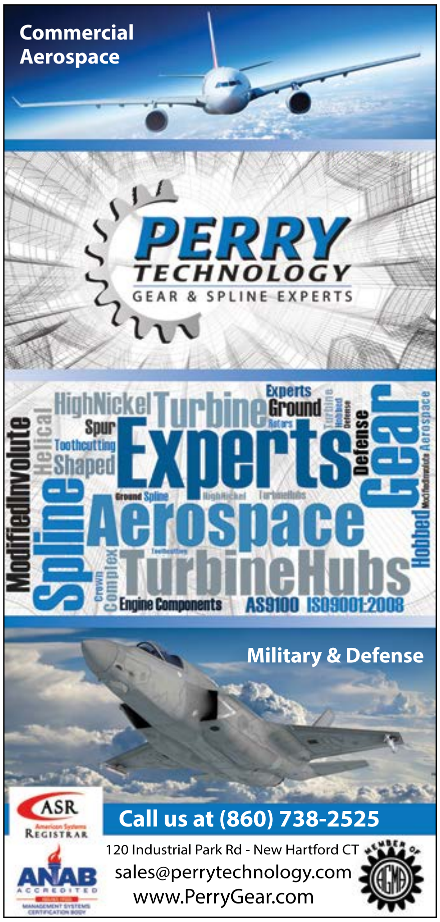
Raymond J. Drago is Chief Engineer of Drive Systems Technology, Inc. (DST), a mechanical power transmission consulting organization that he founded in 1976. Prior to this, Drago worked for the Boeing Company — Helicopters Division until his retirement after 37 years of service. Currently Mr. Drago is involved in the analysis, design, manufacture, assembly, and testing of many gear systems. In his role with DST Drago is active in all areas of mechanical power transmission, including the design and analysis of drive systems in a very diverse field of application — from heart pumps to very large mining and mill gears. Drago also prepared and delivered more than 150 seminars dealing with various aspects of gear design and analysis.

In a career spanning more than 40 years, **Robert Errichello** has earned a reputation for being the go-to person for instruction on gear failure analysis. Bob heads his own gear consulting firm, GEARTECH, and is founder of GEARTECH Software, Inc. He is a registered Professional Engineer who holds BS and MS degrees in Mechanical Engineering and a Master of Engineering degree in structural dynamics from the University of California at Berkeley. He is author of more than 60 articles on design, analysis, and application of gears, and has written three widely-used computer programs for the design and analysis of gears. He is a recipient of AGMA's Lifetime Achievement Award in addition to other awards from AGMA, AWEA, and STLE. Students come from all over the world to attend his course, and AGMA is proud to be able to extend this learning experience to you. Last, but certainly not least, Bob is also a longtime Gear Technology magazine Technical Editor.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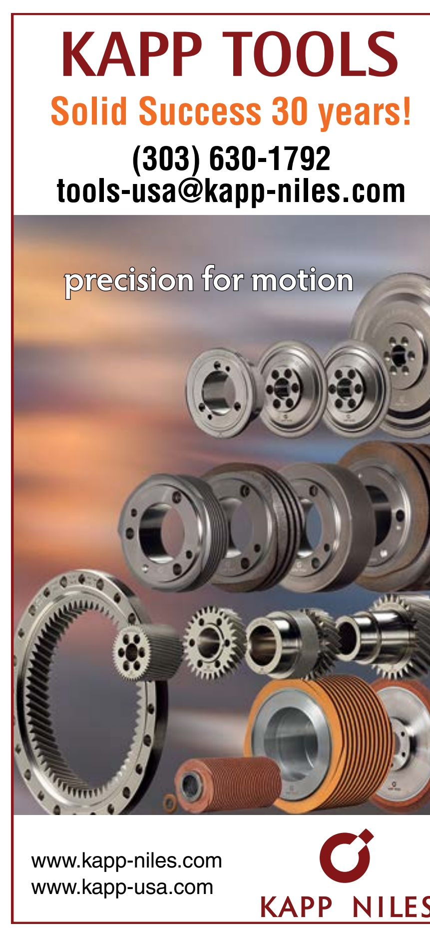
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Same Hob for Pinion and Gear

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

I make all the double helical gears that go into a gearbox — four different gears in this unit. If the gear module for the bull gear and the intermediate gear are the same (these are the two individual gears that mate), and the gear module for the high-speed pinion and high-speed gears are the same (these are the other two individual gears that mate in the gear box as well), is it then possible to just use two hobs in this setup to make all four gears, since they mate together with each other? We are currently using a different gear hob for each gear.

Expert Response Provided by Bob Wasilewski, Arrow Gear

Different hobs are often used to manufacture mating gears. Just because the module is the same does not mean that the same hob can be used to make both parts and still meet blueprint requirements. The hob controls more than the module. Tooth depth, thickness and root fillet conditions often require very different cutting tools. These items are interrelated. The same hob used to cut a wheel with a large number of teeth will produce a radically different root condition on a pinion with a small number of teeth. (But) without specifics it really is not possible to say if the same hob can be used in your application (See also Fig. 1).

Robert Wasilewski is engineering services manager for Arrow Gear Company.

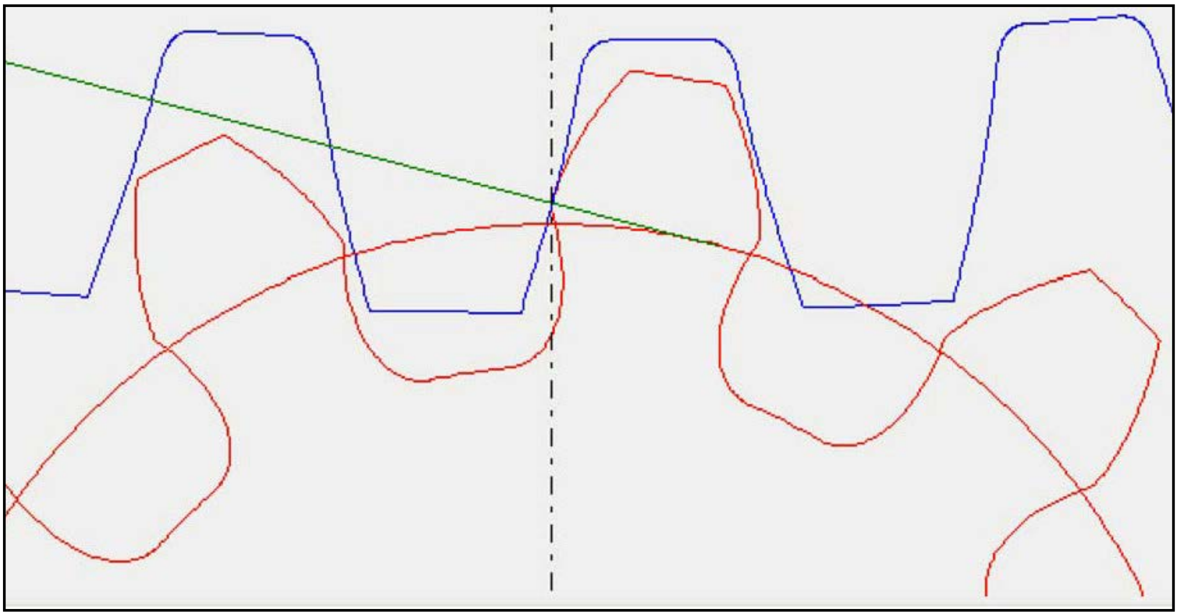

Figure 1 The same hob used to cut a wheel with a large number of teeth will produce a radically different root condition on a pinion with a small number of teeth, as shown in this illustration prepared using AGMA's *Gear Rating Suite 2.2* software.

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Increased Tooth Bending Strength and Pitting Load Capacity of Fine-Module Gears

Dipl.-Ing. Andreas Dobler, Dr.-Ing. Maria Hergesell, Dr.-Ing. Thomas Tobie and Prof. Dr.-Ing. Karsten Stahl

Miniaturization is one of the major trends for future drivetrain design. Hence more and more small motors and gearboxes are available at the market. Also components like gears becoming smaller. Modules of 1 mm and less are not unusual. But the common calculation methods according to ISO 6336 and DIN 3990 are mainly verified for gears with module 3 mm — 10 mm. These investigations showed a decreasing load-carrying capacity for tooth bending strength and pitting resistance with increasing gear size. But for gears with module less than 5 mm, a size effect is not thus far considered in the calculation methods.

Therefore theoretical analysis and experimental investigations were done to verify the load-carrying capacity of small-sized gears. The results prove an increased tooth bending strength and pitting resistance of approx. 30% for case-carburized gears with module 0.6 mm, compared to gears with module 5 mm. Hence a proposal for an extended size factor for the calculation method according to DIN 3990 and ISO 6336 was derived.

Introduction

Miniaturization is one of the major trends in drivetrain design. Therefore, gearboxes with gears with a module of 1 mm or less are increasingly used (Refs. 10 and 18). For industrial robots, waste-heat-recovery units, and rapidly accelerating pick-and-place applications, high power densities are required; hence these gears are often made of case-carburized steels. The design of the gears is typically based on the calculation methods according to DIN 3990 (Refs. 3–4) and ISO 6336 (Refs. 6–7) for tooth root bending strength and pitting resistance. These calculations are based on manifold tests on gears with $m_n = 5$ mm. Special research projects (Refs. 21 and 23) prove decreasing load-carrying capacity with increasing gear size. But only a thin data basis is available for smaller gears in the size range of approx. $m_n = 1.5 \ldots 5$ mm. For smaller sizes, the calculation methods are not verified at all. Therefore, no gear size influence is considered in the calculation for gears with a module of less than 5 mm. Hence small-sized gears have wasted load-carrying capacity. For this reason the load-carrying capacity of small-sized gears was theoretically and experimentally investigated.

Theoretical Influence of Gear Size on Load-Carrying Capacity

The common calculation methods according to DIN 3990 and ISO 6336 are based on a comparison of occurring stress and allowable stress. The influence of gear size on the load-carrying capacity is considered with the size factors Y_X (tooth root bending) and Z_X (pitting), but there are further influences, which should be considered.

In the following, major influences of gear size on the load factors as well as on the permissible tooth root bending and contact stress will be discussed.

2.1 Influence of gear size on the load factors K_v, $K_{H\alpha}$, $K_{H\beta}$, $K_{F\alpha}$, $K_{F\beta}$. The influence of gear size on the occurring stress is limited to the load factors. The common calculation methods for tooth root stress and contact stress are presented in Equations 1 and 2.

$$\sigma_F = K_A \cdot K_V \cdot K_{F\beta} \cdot K_{F\alpha} \cdot \frac{F_t}{b \cdot m_n} \cdot Y_F \cdot Y_S \cdot Y_\beta \tag{1}$$

$$\sigma_{H½} = \sqrt{K_A \cdot K_V \cdot K_{H\beta} \cdot K_{H\alpha}} \cdot Z_{B/D} \cdot Z_\varepsilon \cdot Z_\beta \cdot Z_H \cdot Z_E \cdot \sqrt{\frac{F_t}{d_1 \cdot b} \cdot \frac{u+1}{u}} \tag{2}$$

The application factor K_A considers externally induced overload. The dynamic factor K_v takes into account internal dynamic loads, while the transverse load factor $K_{F\alpha}/K_{H\alpha}$ and the face load factor $K_{F\beta}/K_{H\beta}$ consider the influence of uneven load distribution at the meshing teeth respectively along the face width; detailed descriptions of all factors are summarized in ISO 6336.

The dynamic factor K_v mainly depends on the operating conditions (here: resonance ratio N), tooth deviations (from manufacturing and profile modification, here: factor K) and the variation of the meshing stiffness (Eq. 3).

$$K_V = N \cdot K + 1 \tag{3}$$

For a smaller gear with constant main geometry, the resonance ratio N is proportional to the module m_n and the revolution speed n_1 (Eq. 4). Hence the subcritical operating range is becoming wider with decreasing gear size. Thus for small-sized gears, higher speeds are acceptable.

$$N \sim m_n \cdot n_1 \tag{4}$$

The factor K in Equation 3 is a function of the ratio of manufacturing tolerances to load-induced deviations. While the load-caused deviations are proportional to the module, the manufacturing tolerances remain the same due to technological limitations. This may lead to a worse dynamic behavior and increased stresses.

This paper is an excerpt of research project No. 410 II (Ref. 11) of the German Research Association for Power Transmission (FVA). It was funded by the Federal Ministry of Economics and Technology (BMWi) through the German Federation of Industrial Research Associations.

The manufacturing tolerances also influence the load distribution, considered with the factors $K_{F\alpha}$, $K_{H\alpha}$, $K_{F\beta}$ and $K_{H\beta}$. Furthermore, shaft deviations, bearing displacements and housing deformations, have an increasing effect on the load distribution with decreasing gear size. Hence uneven load distribution is a major problem of small-sized gears. Therefore extraordinary high manufacturing quality, adequate flank modifications and the application of bearings with reduced clearance are recommended for small-sized gears. For evaluating gear quality, the used standardization has to be considered. According to DIN 3961 (Ref. 1) and DIN 3962 (Ref. 2), the limiting values for gear quality are the same for gears equal or less than $m_n = 1$ mm. In most cases no extrapolation is used. It is recommended to use ISO 1328 (Ref. 8) to determine the quality of small-sized gears. This standard gives different limiting values down to $m_n = 0.5$ mm. However, there is no linear relation between the limiting values and gear size, so small-sized gears with quality 5 may have more deviations compared to the gear size than larger gears of the same quality.

2.2 Influence of gear size on the permissible tooth root bending stress. The permissible tooth root bending stress σ_{FP} is calculated as:

$$\sigma_{FP} = \frac{\sigma_{Flim} \cdot Y_{ST} \cdot Y_{NT}}{S_{Fmin}} \cdot Y_{\delta relT} \cdot Y_{RrelT} \cdot Y_X \quad (5)$$

Thereby σ_{Flim} is the allowable bending stress number of the reference test gears with a module of $m_n = 5$ mm for 1% failure probability. $Y_{\delta relT}$ is the relative notch sensitivity factor that compares the notch sensitivity of the actual gear with that of the reference gear. The size factor Y_X considers the influence of gear size on the tooth root strength. It depends on the material, heat treatment and module. For module sizes $m_n \leq 5$ mm, the size factor $Y_X = 1$, according to DIN 3990 and ISO 6336.

However, according to the general mechanics of materials local notches like the tooth root fillet lead to locally increased stresses. But the stress peaks are reduced by the support of the surrounded material through plastic micro-deformations. This leads to increased gear strength. The range of support n_χ can be calculated:

$$n_\chi = 1 + \sqrt{\rho' \cdot \chi^*} \quad (6)$$

It is depending on the relative stress gradient χ^*:

$$\chi^* = \frac{d\sigma}{dy} \sim \frac{1}{m_n} \quad (7)$$

The slip-layer thickness ρ' is a function of the material (Ref. 7), but there are further results of research projects that determine higher values (Ref. 19).

The supporting properties of the material are also depending on the notch parameter q_S. For $q_S = 2.5$ the calculated values according to the different calculation methods are shown (Fig. 1).

While DIN 3990 shows no effect for decreasing the gear size from module 5 mm to 0.5 mm, the other calculation methods promise increased gear strength of 20% – 55%, depending on the assumed slip-layer thickness.

The relative surface factor Y_{RrelT} is taking into account the influence of surface roughness at the tooth root fillet on the local stresses. Depending on whether the tooth fillet is ground, the roughness is comparable to that of the gear flank. However, the resulting roughness of common grinding processes is limited to values of approx. $Ra = 0.2$ µm. Hence the influence of roughness is increasing with decreasing gear size. This may lead to an additional stress increase, caused by roughness notches in the tooth fillet.

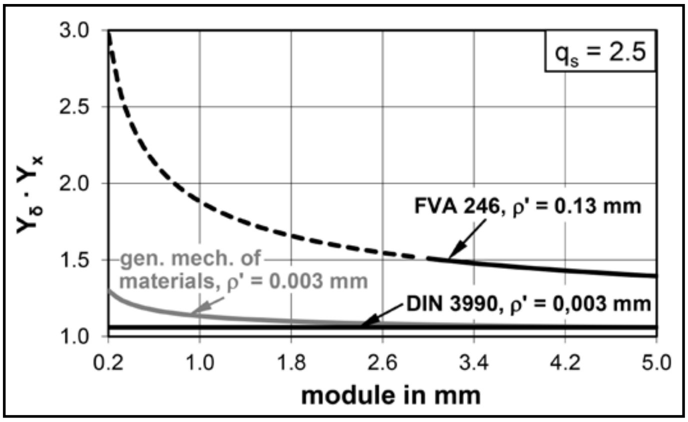

Figure 1 Supporting properties of the material depending on gear size according to DIN 3990 (Ref. 4), general mechanics of materials (Ref. 17), and FVA 246 (Ref.19).

2.3 Influence of gear size on the permissible contact stress. The permissible contact stress is calculated:

$$\sigma_{HP} = \frac{\sigma_{Hlim} \cdot Z_{NT}}{S_{Hmin}} \cdot Z_L \cdot Z_V \cdot Z_R \cdot Z_W \cdot Z_X \quad (8)$$

The allowable contact stress σ_{Hlim} of the reference test gears is explained in DIN 3990 and ISO 6336. It is dependent upon the material and heat treatment. The size factor Z_X considers the influence of gear size on the permissible contact strength. The major parameters on this factor are the material quality and heat treatment (statistical influence), as well as the radius of flank curvature and the case depth (supporting effect). The size factor $Z_X = 1$ for $m_n \leq 10$ mm (Ref. 3) respectively, $Z_X = 1$ for all modules (Ref. 6). But with decreasing gear size the stress gradient at the gear flank is increasing as well. On one hand, the depth of maximum shear stress beneath the surface is proportional to the relative radius of curvature. Hence with decreasing gear size the depth is decreasing. On the other hand, theoretical studies show that with decreasing gear size the friction coefficient increases. This leads to higher shear and thermal loads at the flank surface. Therefore the shear stresses at the surface, as well as near to the surface, are increased as well. The main influence on the pitting resistance is the first one because of the depth of pitting cracks; it leads to increased permissible contact stresses for small-sized gears.

The complex loading conditions at the tooth flank can only be calculated with adequate software, e.g. — *ROSLCORHR* (Ref. 13). This was done for the reference test gears as well as for gears with the same geometry — but with a module of 0.5 mm (Ref. 12). Thus the relative radius of curvature at the pitch point was 10 mm compared to 1 mm. The operating conditions were assumed as constant. The results show that the supporting properties depend on the local point at the gear flank. For the calculated gears with module 0.5 mm, an increase of 10% – 20% can be derived.

The factors Z_L, Z_v and Z_R consider the influences of lubricating conditions (oil viscosity, circumferential velocity, surface roughness) on the permissible contact stress. The roughness factor Z_R depends on the flank roughness and the center distance (Ref. 3), respectively the relative radius of curvature (Ref. 6). Because flank roughness is almost independent of gear size, this factor worsens with decreasing gear size. Furthermore, the circumferential velocity is decreased for small-sized gears, even if they are operated at higher rotational speeds. Hence the velocity factor Z_v is also decreased. These effects reduce the pitting resistance of small gears.

The worse lubricating conditions at small-sized gears lead to a higher risk for micropitting and wear, which should be considered.

Table 1 Geometry of the test gears

Test Aim		Tooth Bending Strength				Pitting Load Capacity			
Test Rig		Back-To-Back		Pulsator Test Rig		Back-To-Back			
Designation		BR06K	BR06U	FL06P	FL045(U)P	FL06K	FL06U	FL045K	FL045U
Module	m_n / mm	0.60		0.60	0.45	0.60		0.45	
Centre Distance	a / mm	35.28		-		15.00		11.25	
Tip Diameter	d_a / mm	36.38 / 36.45		19.38	14.54	13.08 / 19.38		9.81 / 14.54	
Face Width	b / mm	9.00		9.00	6.75	10.00 / 9.00		7.75 / 6.75	
Number of Teeth	z_1 / z_2	57 / 58		29		19 / 29			
Profile Shift Coefficient	x_1 / x_2	0.903 / 0.500		0.500		0.450 / 0.689			
Helix Angle	β / °	0							
Profile Crowning	C_h / µm	5 / 5	-	-		5 / 5	-	3.5 / 3.5	-
Lengthwise Crowning	C_b / µm	1 / 0	-	-		3 / 0	-	3 / 0	-
Helix Angle Correction	C_p / µm	-	-	-		-13 / 0	-	1 / 0	-

Table 2 Operating conditions

Test Aim	Tooth Bending Strength	Pitting Load Capacity	
Test Gears	BR06	FL06	FL045
Module	0.6	0.6	0.45
Rotational Speed Of Pinion	2000 rpm	7000 rpm	9000 rpm
Tangential Speed At Pitch Point	3.66 m/s	4.18 m/s	4.03 m/s
Lubricant	mineral oil ISO VG 100 +EP-additive		
Lubricating Conditions	regulated circulationg dip lubrication @60°C (140°F)		
Limit Load Cycles	10 million	50 million	
Failure Criteria	tooth root breakage	pitting area > 4 % of tooth flank	

Experimental Investigations of Gear Size Influence on Load-Carrying Capacity

In a first research project (Refs. 9, 14 – 15) an increased load-carrying capacity of involute gears within a module range of 0.3 – 1.0 mm had already been basically confirmed, although there were some difficulties with the heat treatment of the test gears. And so a second project (Ref. 11) was started to determine reliable values for this potential increase.

Test gears and operating conditions. For the determination of the load-carrying capacity two special gear geometries were designed. The gear ratio for tooth root breakage testing was 57/58, for pitting testing 19/29. The tests were performed with modified and unmodified test gears of module $m_n = 0.45$ mm and $m_n = 0.6$ mm on a FZG small-gear, back-to-back test rig. The detailed gear geometry is presented in Table 1. In addition, pulsator tests were performed with the wheels of the pitting gear design, partly with unground tooth fillet.

All test gears were made of 16MnCr5. After gear hobbing they were case-carburized to 700–750 HV surface hardness and 0.1 – 0.2 mm case hardening depth (limit hardness 550 HV). After heat treatment the gears were ground. The gear quality acc. to DIN 3962 was Q ≤ 5. The tooth root fillets were ground due to lack of small enough protuberance hobbing tools. Only the gears of FL045UP had unground fillets. They were only hobbed and heat treated. The test conditions for the different tests at the FZG small gear test rig are shown (Table 2). Prior to the tests all gearsets were run a two-stage running-in procedure.

The pulsator tests were run at an electromagnetic resonance pulsator, as it is described, e.g., in (Ref. 21). Each gear was clamped over 6 teeth. Hence the force was applied near the outer point of single tooth contact. The test frequency was in a range of 50 to 60 Hz. The pulsator tests were run until tooth root breakage occurred or a maximum of 6 million load cycles.

Evaluation of Test Results

Tooth root bending strength. For the analysis of running tests the equations of DIN 3990 (Ref. 4) or ISO 6336 (Ref. 7) are typically used to determine the tooth root stress, depending on the applied torque. But the special test gear geometry leads to transverse contact ratios of two or even higher for the loaded contact. Hence the tooth root stresses have been calculated using the FZG-FVA software *RIKOR* I (Ref. 22). Furthermore, the actual tooth fillet geometry was considered. The tooth root stress for the endurable torque was multiplied with the factor of 0.86 (Ref. 20) to convert the stress for 50% failure probability to 1% failure probability $\sigma_{Flim,exp}$. This stress was compared to the allowable bending stress σ_{Flim} of the reference test gears with 5 mm module.

For evaluation of the pulsator tests, pulsator stress was calculated depending on the pulsator force as:

$$\sigma_{F,pulsator} = K_A \cdot K_V \cdot K_{F\beta} \cdot K_{F\alpha} \cdot \frac{F_{PN} \cdot \cos(\alpha_n)}{b \cdot m_n} \cdot Y_F \cdot Y_S \tag{9}$$

Of course the real tooth root fillet geometry was considered for the calculation of the form factor Y_F and the stress correction factor Y_S. The pulsator stress for the endurable force was multiplied with the factor 0.90 to convert the pulsator result to one of a running test (Ref. 20). This stress was additionally converted to a failure probability of 1%.

Evaluation of the pitting load capacity. For the evaluation of the pitting load capacity, the endurable contact stress was calculated according to Equations 2 and 8. This stress was multiplied with the factor 0.92 (Ref. 20) to convert the result for 50% failure probability to 1% failure probability $\sigma_{Hlim,exp}$. Subsequently the stress was compared to the allowable contact stress of the standard reference test gears.

Test results. Figures 2 and 3 show typical examples of tooth root breakage at the test gears after running tests. The damage is comparable to those at gears with a module of 5 mm. The cracks start at the surface near the 30° tangent at the tooth root fillet. As expected for a gearset without flank modifications, the crack initiation at BR06U is near the face side of the pinion. For the gearsets with adequate flank modifications the crack starts near the middle of the face width. The tooth root breakages at the pulsator test rig are comparable to those of BR06K.

The pitting damages at the pinions with module 0.45 mm and 0.6 mm are also comparable to those of gears with module 5 mm. The pitting occurred preferentially in the flank area with negative specific sliding. The test gears with adequate flank modifications show uniform pitting damage along the whole face width (Fig. 4). In contrast, the pitting at the gears without modifications occurred near one face side. Therefore, the adjustable bearing plates of the test rig were insufficient to compensate for the load-caused deformations and deflections. Furthermore, micropitting was observed on all test gears (Fig. 5).

Figure 6 shows major results for the tooth bending strength. The presented S-N curves of the pulsator tests are in good accordance with the results of the additional running tests with modified and unmodified test gears (Fig. 8) when considering the load-caused, real transverse ratio.

For the pitting load capacity, exemplary S-N curves of the test gears with module 0.6 mm are shown (Fig. 7). The results are comparable and correspond well with those for module 0.45 mm. The allowable stress numbers σ_{Hlim} are approximately equal for all pitting test variants and on a high level. The determined allowable stress numbers for tooth root breakage σ_{Flim} and for pitting load capacity σ_{Hlim} of all test variants are shown (Fig. 8). Additionally, the stress numbers acc. to DIN 3990 (Ref. 5) (material quality MQ) are presented for comparison.

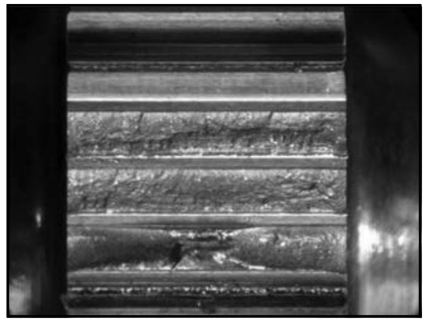

Figure 2 **Tooth root breakage at the BR06K pinion (T_1 = 53 Nm, 5.01·10^6 LC_{Pi}).**

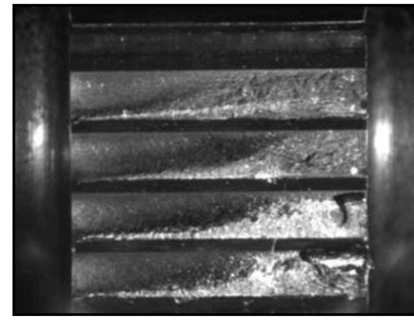

Figure 3 **Tooth root breakage at the BR06U pinion (T_1 = 60 Nm, 474849 LC_{Pi}).**

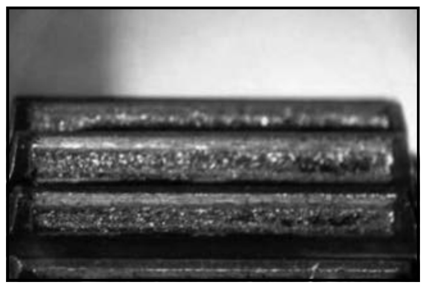

Figure 4 **Pitting at the FL06K pinion (T_1= 8.4 Nm, 26.0·10^6 LC_{Pi}).**

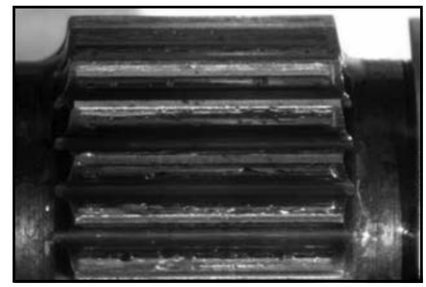

Figure 5 **Micropitting at the FL06U pinion (T_1 = 10 Nm, 27.0·10^6 LC_{Pi}).**

Proposals for Extended Size Factors

As seen in Figure 8, the experimentally determined allowable stress numbers of the gears with modules of m_n = 0.45 and 0.6 mm are significantly higher than those given for standard reference gears. The values are comparable to those which were expected from the theoretical investigations (see again sections *Influence of gear size on the permissible tooth root bending stress,* and *Influence of gear size on the permissible contact stress*). Hence, there is a need for new, extended size factors. These factors should consider the higher strength of small sized gears.

For the derivation of the new size factors, additional experimental data of further research projects were analyzed (Refs. 9, 16, 21 and 23). In these projects case-carburized spur gears with different module sizes were tested. The proposals for the

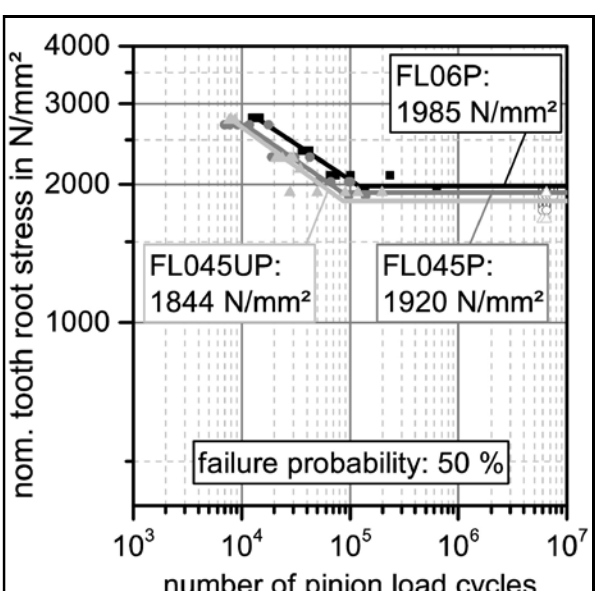

Figure 6 **S-N curves for tooth bending strength pulsator tests *only*.**

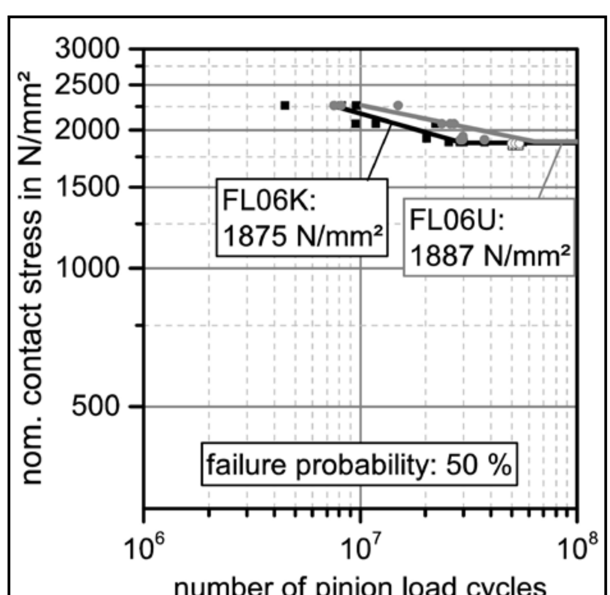

Figure 7 **S-N curves for pitting load capacity of FL06K and FL06U.**

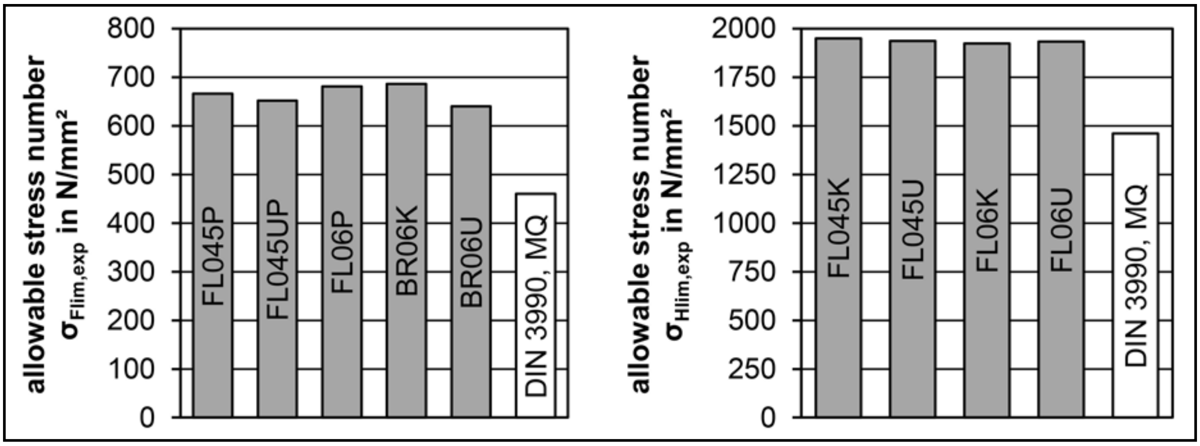

Figure 8 **Experimentally determined allowable stress numbers for tooth root breakage (left) and pitting (right) in comparison to reference gears (Ref. 5).**

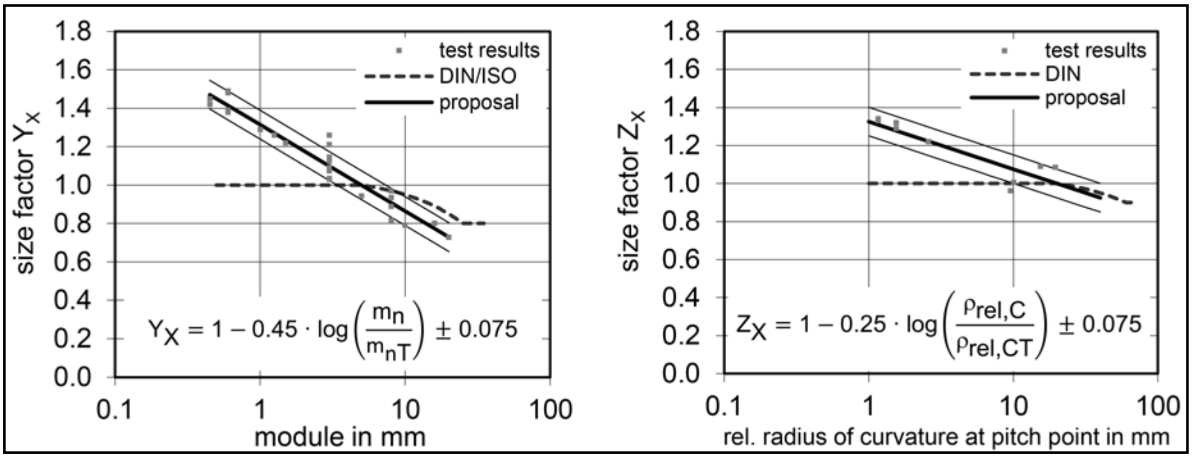

Figure 9 **Proposed size factors for tooth bending strength (left) and pitting load capacity (right) for case-hardened steel 16MnCr5 based on test results.**

new extended size factors Y_X and Z_X, as well as the existing ones of DIN 3990/ISO 6336, are presented (Fig. 9). The test results of different research work are included, too.

For tooth root breakage, the relevant size parameter is the module (see again *Influence of gear size on the permissible tooth root bending stress*). Hence the calculation method for Y_X is depending on the module of the actual gear m_n, compared to the module of the reference test gears $m_{nT} = 5$ mm. For pitting resistance, the relevant size parameter is the relative radius of curvature at the pitch point C. Therefore the calculation of the size factor Z_X considers the relative radius of curvature of the actual gear $\rho_{rel,C}$ as well as the one of the standard reference gears $\rho_{rel,CT} = 10$ mm.

Since reliable manufacturing and heat treatment of high-quality gears with module sizes $m_n < 0.45$ mm is extremely demanding, the validity range of the proposals should be limited to module sizes $m_n \geq 0.45$ mm. For high gear quality and proper heat treatment, the upper range of tolerance can be used. If there are uncertainties, it is recommended to use the size factors according to the lower range of tolerance.

Summary

Tooth bending strength and pitting load capacity increase with decreasing gear size. Since no comprehensive verification of the carrying capacity of fine module gears has been available thus far, common calculation methods do not state a positive size effect for gears with module sizes smaller than 5 mm. Increased tooth bending strength and pitting load capacity are proven theoretically and experimentally in this work. On this basis proposals for extended size factors for calculation acc. to DIN 3990 / ISO 6336 are given.

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A Proposed Pre-Finish Cylindrical Gear Quality Standard

Peter E. Chapin

It is quite common to specify a gear class for in-process quality requirements, usually calling for a lower quality class than is required for the finished gear quality. Although it is appropriate to have lower expectations for a pre-finish gear condition, it is not appropriate to subject the pre-finish gear to the same level of scrutiny as a finished gear. Gears in a pre-finish condition may have large feed scallops or generating flats, which are desirable for productivity and may be conducive to the finishing process. However, such features will be evaluated as errors when subjected to the full analysis as required by the finished gear class inspection. Therefore, the use of a finished gear quality specification is not recommended or even appropriate for pre-finish gear quality evaluation, even if the quality class has been adjusted to pre-finish expectations. Additionally, in-process requirements often require non-zero target helix and profile slopes, which necessitate a new method of analysis to determine the achieved quality class. Therefore, a pre-finish evaluation method and standard is proposed. This proposed standard would not make any recommendations regarding the required quality for any application. The intent is to establish standard pre-finish quality classes for typical finishing operations, which only include the inspection elements that are important to properly evaluate pre-finish gear quality as it applies to the finishing operation.

Introduction

It is quite common to reference a gear class for in-process quality requirements. For example, a gear may need to meet ANSI/AGMA/ISO 1328 Class 6 when it is finished, but to control the in-process quality, the pre-finish hobbing quality may be specified at a lower quality, such as Class 8. (ANSI/AGMA/ISO 1328 is the current AGMA standard, replacing the older 2015, 2000-A88, and 390 quality class specifications.) Although it is appropriate to have lower expectations for a pre-finish gear condition, it is not appropriate to subject the pre-finish gear to the same level of scrutiny as a finished gear. The ANSI/AGMA/ISO 1328 gear quality specification, which is intended to be applied to a finished gear, covers numerous parameters, including: helix (slope, form, and total), profile (slope, form, and total), and pitch (single and cumulative). However, many of these quality characteristics are so drastically affected by typical pre-finish conditions that they should not even be applied. To illustrate this issue, consider a pre-grind hobbing operation where a large feed rate is used, which produces rather significant feed scallops on the tooth flanks. Such scallops are not only typical, but are often desirable. Features such as helix slope, profile slope, and cumulative pitch (if properly measured)—likely unaffected by feed scallops—are considered important features to evaluate pre-grind quality. However, features such as helix total and form, profile total, and form will be negatively impacted by the feed scallop, which may cause the gear to fail pre-finish specifications even with lowered pre-finish expectations (see Figs. 1, 2 and 4). Therefore, the use of a finished gear quality specification such as ANSI/AGMA/ISO 1328 is not recommended or even appropriate for pre-finish gear quality evaluation — even if the quality class has been adjusted to pre-finish expectations. In addition, in-process requirements often require non-zero target helix and profile slopes, and a method of analysis is required to handle this. Therefore, a pre-finish evaluation method and quality standard is proposed.

Goals of this proposed pre-finish standard:

- Provide a standard method of evaluation for pre-finish quality.
- Reduce the evaluated parameters to those that directly impact final gear quality for the given finishing process.
- Avoid penalizing evaluated quality for inherent, typical, and often desirable pre-finish geometries (such as feed scallops, generating flats, thread-to-thread induced pitch errors, intentional off-lead conditions, leaning profiles, etc.).
- Promote a common understanding of typical pre-finish characteristics and how they affect evaluated quality.
- Reduce in-process costs by eliminating unnecessary and overly restrictive tolerances.
- Create pre-finish quality classes for the most common finishing operations.

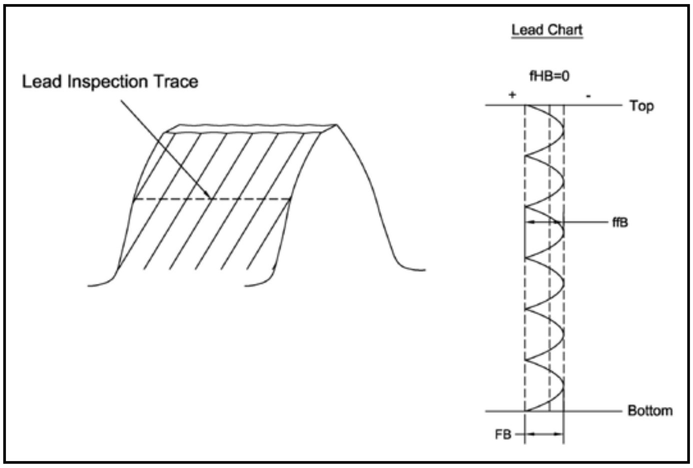

Figure 1 **While it is often desirable to have a significant hobbing feed scallop, such scallops will negatively impact helix total and form errors. There should be a minimal effect on lead slope unless there are not enough scallops to construct a decent best-fit line.**

This proposed standard would not make any recommendations regarding the required quality for any application. The intent is to establish standard pre-finish quality classes for typical finishing operations, which only include the inspection elements that are important to properly evaluate pre-finish gear quality as it applies to the finishing operation. It would be the responsibility of the manufacturing/process engineer, quality engineer, or other responsible individual to establish the required pre-finish quality class for their application.

The intent of this standard is not to create a rigid pass/fail evaluation system. Because the gears that would be evaluated according to this standard are still in an unfinished state, the final pass/fail decision cannot necessarily be made. However, as would be expected, the likelihood of producing an acceptable gear through the finishing process decreases with increased pre-finish errors. Therefore, this standard should be viewed as a manufacturing tool and not a strict quality standard.

Since the various finishing processes have different pre-finish quality requirements, it makes sense to establish different criteria for the pre-finish conditions of each finishing process. The five most common finishing processes are grinding, shaving, honing, rolling, and skiving (hard hobbing), and guidelines would be provided for each. These guidelines are based on the existing ANSI/AGMA/ISO 1328 tolerances and tolerance classes (with certain elements excluded), and also with some newly defined elements that may be important to pre-finish quality but are not currently covered in the finished gear standard.

Elements included in ANSI/AGMA/ISO 1328:

- Helix slope (individual teeth): $fH\beta$
- Helix slope tolerance: $fH\beta T$
- Helix form: $ff\beta$
- Helix form tolerance: $ff\beta T$
- Helix total: $F\beta$
- Profile slope (individual teeth): $fH\alpha$
- Profile slope tolerance: $fH\alpha T$
- Profile form: $ff\alpha$
- Profile form tolerance: $ff\alpha T$
- Profile total: $F\alpha$
- Single (individual) pitch: fp
- Cumulative pitch: Fp

NEW elements NOT included in ANSI/AGMA/ISO 1328:

- * Runout: Fr
- Average helix slope (mean): $fH\beta m$
- Helix slope variation (difference from high to low $fH\beta$): $\Delta fH\beta$
- Helix slope upper tolerance limit: $fH\beta Tmax$
- Helix slope lower tolerance limit: $fH\beta Tmin$
- Helix slope target (nominal): $fH\beta nom$
- Helix slope error (difference from nominal): $fH\beta'$
- Average helix slope error: $fH\beta m'$
- Average profile slope (mean): $fH\alpha m$
- Profile slope variation (difference from high to low $fH\alpha$): $\Delta fH\alpha$
- Profile slope upper tolerance limit: $fH\alpha Tmax$
- Profile slope lower tolerance limit: $fH\alpha Tmin$
- Profile slope target (nominal): $fH\alpha nom$
- Profile slope error (difference from nominal): $fH\alpha'$
- Average profile slope error: $fH\alpha m'$

New Elements Defined:

***Runout (Fr).** Technically, runout is not a newly defined element; however, it is not a required element under ANSI/AGMA/ISO 1328 because gears do not typically function with double-flank contact. Because gear finishing processes typically do use double-flank contact and can be significantly affected by pre-finish radial runout, it is important to include runout in certain pre-finish evaluations. Therefore, runout is included in this proposed pre-finish standard.

Average helix slope **(*fHβm*)** is the numeric average (mean value) of all measured individual tooth helix slope values (including sign) per flank.

Example: four (4) $fH\beta$ values: -4, 2, 7, -1

$$fH\beta m = (-4 + 2 + 7 - 1) / 4 = 1$$

Helix slope variation **(Δ*fHβ*)** is the difference between the maximum and minimum (including sign) of all measured individual tooth helix slopes per flank.

Example: four (4) $fH\beta$ values: -4, 2, 7, -1

$$\Delta fH\beta = 7 - (-4) = 11$$

In-process helix slope. It is quite common to require an off-lead condition, where zero helix slope is not the ideal condition in the pre-finish state. This is most often done to compensate for subsequent distortion in heat-treatment, where the two most common helix changes that occur are tapering and unwinding of the helix. To allow for this desirable off-lead condition, the target left and right flank $fH\beta$ values may be offset from zero, with each flank possibly having different target helix slopes. Therefore, the upper and lower $fH\beta T$ tolerance limits are calculated from the target $fH\beta$ instead of zero. As a result, several new elements must be defined to allow for a non-zero target helix slope.

Target helix slope **(*fHβnom*)** is the target (nominal) in-process helix slope ($fH\beta$). This value is set by the manufacturing/process engineer, quality engineer, or other individual responsible for the finishing process. The target helix slope value should be chosen to optimize the conditions for the finishing process. Target values may be different for each flank, denoted with a LF or RF preceding $fH\beta nom$ (i.e., LF $fH\beta nom$). If the left flank and right flank have different target helix slopes, then the part orientation and flank designations need to be clearly identified.

Helix slope tolerance **(*fHβT*)** is the helix slope ($fH\beta$) tolerance per ANSI/AGMA/ISO 1328.

Helix slope, upper limit **(*fHβTmax*)** is the upper tolerance limit for helix slope.

$$fH\beta Tmax = fH\beta nom + fH\beta T$$

Helix slope, lower limit ($fH\beta Tmin$) is the lower tolerance limit for helix slope.

$$fH\beta Tmin = fH\beta nom - fH\beta T$$

Helix slope error ($fH\beta'$) is the difference between the target helix slope ($fH\beta nom$) and the measured helix slope ($fH\beta$) value

Example: $fH\beta nom = -20\mu m$ and $fH\beta = -17\mu m$

$$fH\beta' = -17-(-20) = 3\mu m$$

***Note*:** The use of prime (') in any element identifies that value as an error value (difference from target), and not a raw measurement value.

Average helix slope error ($fH\beta m'$) is the numeric average (mean value) of all measured individual tooth helix slope error values (including sign) per flank.

Example: four (4) $fH\beta'$ values: -4, 2, 7, -1

$$fH\beta m' = (-4 + 2 + 7 - 1) / 4 = 1$$

Average profile slope ($fH\alpha m$) is the numeric average (mean value) of all measured individual tooth profile slope values (including sign) per flank.

Example: four (4) $fH\alpha$ values: -10, 5, -5, -2

$$fH\alpha m = (-10 + 5 - 5 - 2) / 4 = -3$$

Profile slope variation ($\Delta fH\alpha$) is the difference between the maximum and minimum (including sign) of all measured individual tooth profile slopes per flank.

Example: four (4) $fH\alpha$ values: -10, 5, -5, -2

$$\Delta fH\alpha = 5 - (-10) = 15$$

In-process profile slope. It is quite common to target a non-zero profile slope, where zero profile slope is not the ideal condition in the pre-finish state. This is often done either to compensate for distortion in heat treatment, or to improve meshing conditions with the finishing tool. To allow for this desirable non-zero profile slope condition, the target $fH\alpha$ value may be offset from zero. Therefore, the upper and lower $fH\alpha T$ tolerance limits are calculated from the target $fH\alpha$ instead of zero. As a result, several new elements must be defined to allow for a non-zero target profile slope.

Target profile slope ($fH\alpha nom$) is the target (nominal) in-process profile slope ($fH\alpha$). This value is set by the manufacturing/process engineer, quality engineer, or other individual responsible for the finishing process. The target profile slope value should be chosen to optimize the conditions for the finishing process. Target values may be different for each flank, denoted with an LF or RF preceding $fH\alpha nom$ (i.e., LF $fH\alpha nom$). If the left flank and right flank have different target profile slopes, then the part orientation and flank designations need to be clearly identified.

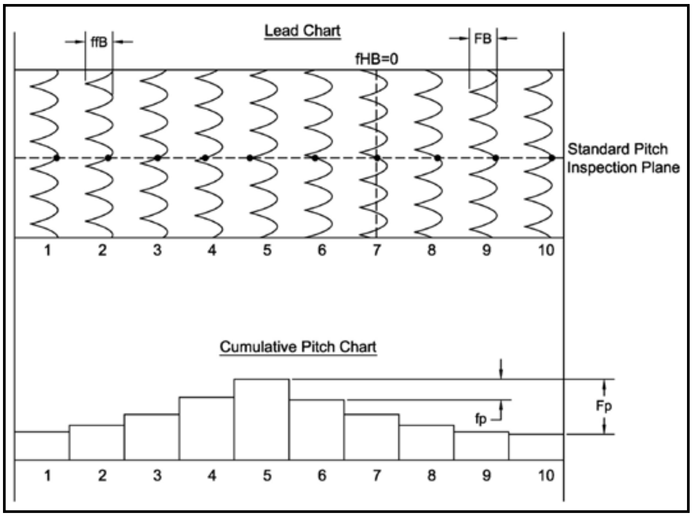

Figure 2 **It is obvious that hobbing scallops will directly affect helix total and form errors. But helix slope error can also be affected — especially when the number of feed scallops on the face width is very low — or if a best-fit evaluation is not used. What may *not* be obvious is that pitch inspection can also be affected by feed scallops (as shown); checking pitch in a fixed plane will traverse the feed spiral, including inducing erroneous cumulative and single pitch errors (as well as calculated runout).**

Profile slope tolerance ($fH\alpha T$) is the profile slope ($fH\alpha$) tolerance per ANSI/AGMA/ISO 1328.

Profile slope upper limit ($fH\alpha Tmax$) is the upper tolerance limit for profile slope.

$$fH\alpha Tmax = fH\alpha nom + fH\alpha T$$

Profile slope lower limit ($fH\alpha Tmin$) is the lower tolerance limit for profile slope.

$$fH\alpha Tmin = fH\alpha nom - fH\alpha T$$

Profile slope error ($fH\alpha'$) is the difference between the target profile slope ($fH\alpha nom$) and the measured profile slope ($fH\alpha$) value.

Example: $fH\alpha nom = 10\mu m$ and $fH\alpha = -5\mu m$

$$fH\alpha' = -5 - (10) = -15\mu m$$

Average profile slope error ($fH\alpha m'$) is the numeric average (mean value) of all measured individual tooth profile slope error values (including sign) per flank.

Example: four (4) $fH\alpha'$ values: -4, 2, 7, -1

$$fH\alpha m' = (-4 + 2 + 7 - 1) / 4 = 1$$

Proposed pre-finish quality classes

Five (5) pre-finish quality classes are proposed to cover the most common finishing operations:

1. Pre-grind “PG”
2. Pre-shave “PS”
3. Pre-hone “PH”
4. Pre-roll “PR”
5. Pre-skive (pre-hard hob)“PSk”

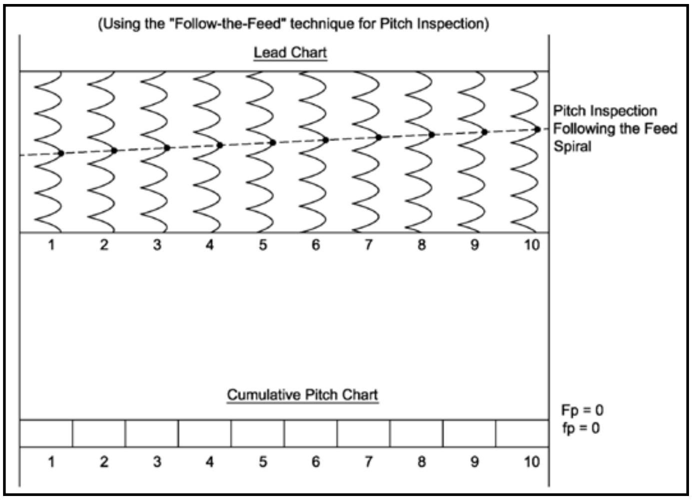

Figure 3 **By following feed spiral during pitch inspection, scallop depth is not factored into cumulative or single pitch errors (or calculated runout). Caution must be exercised with this inspection method for gears with significant crown or taper, as the pitch evaluation finishes one scallop above or below from where it started, which may show a mismatch from start to end.**

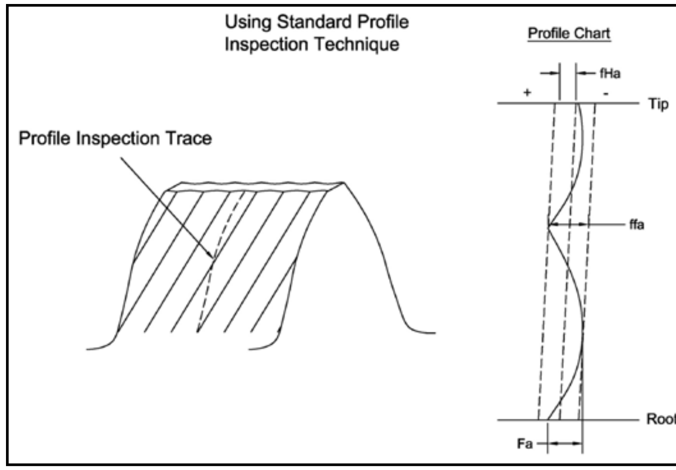

Figure 4 **The standard method of measuring involute profile on a hobbed helical gear will traverse one or more hobbing scallops. Scallop depth will factor directly into evaluated profile total and form errors, and may falsely induce a profile slope error, depending on how the best-fit line is constructed for partial scallops traversed. Note: Spur gears are typically not affected by this because the scallop angles are nearly parallel to the profile inspection trace.**

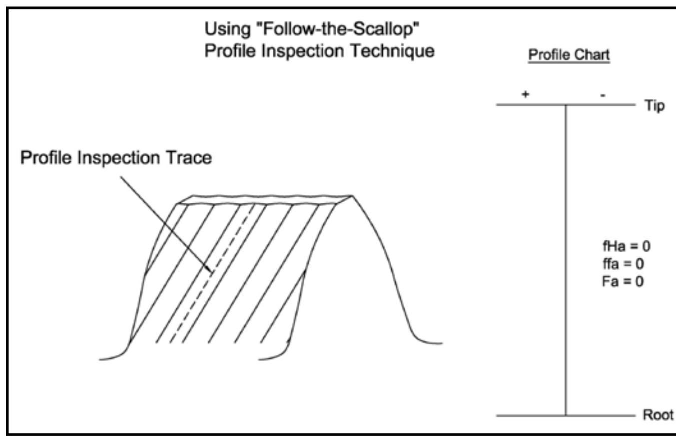

Figure 5 **By following the scallop angle on a hobbed helical gear, the profile inspection does not include scallop geometry in the quality evaluation. This is important because scallop geometry should not be confused with profile quality, especially for a pre-finish quality evaluation.**

The tolerances for each pre-finish class generally follow the equivalent ANSI/AGMA/ISO 1328 quality class (i.e. pre-finish class PG8 tolerances generally follow ANSI/AGMA/ISO 1328 class 8), but with a modified scope. ANSI/AGMA/ISO 1328 includes tolerance classes 1–11. However, classes 1–5 are rather high precision to be included in a pre-finish standard. Therefore, pre-finish classes have been limited to classes 6–11.

Each pre-finish quality class includes specific elements required to meet the defined pre-finish quality class number. All other elements are excluded from evaluation, unless specifically added to the requirements. For example, the pre-grind quality class does not include helix form, but this can be added separately to the pre-finish quality requirements (i.e., class PG9, plus helix form ($ff\beta$) 18µm max.) if deemed necessary by the responsible manufacturing/process engineer, quality engineer, or other responsible individual.

As this is currently in the proposal stage; all elements and proposed tolerances given below for each pre-finish quality class are open for discussion and change.

Pre-grind (class PG6– PG11)

- Average helix slope error ($fH\beta m'$) (average of inspected $fH\beta'$ values)
  - The tolerance for $fH\beta m'$ is equal to $fH\beta T$ per ANSI/AGMA/ISO 1328, using the same class number.
  - Helix slope variation ($\Delta fH\beta$) (difference between max and min $fH\beta$ values) (Tolerance = $2 \times fH\beta T$? (TBD))
  - Average profile slope error ($fH\alpha m'$) (average of inspected $fH\alpha'$ values). Note: If significant hobbing scallops are present, measurement should follow the feed scallop angle (Figs. 4 and 5).
- Cumulative pitch. Note: If significant hobbing scallops are present, measurement must follow the feed spiral (Figs. 2 and 3).

Pre-shave (class PS6– PS11)

- Average helix slope error ($fH\beta m'$) (average of inspected $fH\beta'$ values)
  - The tolerance for $fH\beta m'$ is equal to $fH\beta T$ per ANSI/AGMA/ISO 1328, using the same class number.
- Helix slope variation ($\Delta fH\beta$) (difference between max and min $fH\beta$ values) (Tolerance = $2 \times fH\beta T$? (TBD))
- Average profile slope error ($fH\alpha m'$) (average of inspected $fH\alpha'$ values). Note: If significant hobbing scallops are present, measurement should follow the feed scallop angle (Figs. 4 and 5).
- Profile form. Note: If significant hobbing scallops are present, measurement should follow the feed scallop angle (Figs. 4 and 5). (Tolerance = TBD)
- Cumulative pitch. Note: If significant hobbing scallops are present, measurement must follow the feed spiral (Figs. 2 and 3).
- Runout. Note: If significant hobbing scallops are present, measurement should follow the feed scallop angle (Figs. 4 and 5). The tolerance for runout (FrT) is defined as 90% of the cumulative pitch tolerance per ANSI/AGMA/ISO 1328.

Pre-hone (class PH6– PH11)

- Average helix slope error ($fH\beta m'$) (average of inspected $fH\beta'$ values)
- The tolerance for $fH\beta m'$ is equal to $fH\beta T$ per ANSI/AGMA/ISO 1328, using the same class number.

• Helix slope variation ($\Delta fH\beta$) (difference between max and min $fH\beta$ values)
  - ($\Delta fH\beta T = 2 \times fH\beta T$? (TBD))

• Helix form to limit scallop height? Maybe.
  - $ff\beta T$ = (TBD)
  - Average profile slope error ($fH\alpha m'$) (average of inspected $fH\alpha'$ values). Note: If significant hobbing scallops are present, measurement should follow the feed scallop angle (Figs. 4 and 5).

• Profile form. Note: If significant hobbing scallops are present, measurement should follow the feed scallop angle (Figs. 4 and 5). (Tolerance = TBD)
• Cumulative pitch. Note: If significant hobbing scallops are present, measurement must follow the feed spiral (Figs. 2 and 3).
• Runout. Note: If significant hobbing scallops are present, measurement should follow the feed scallop angle (Figs. 4 and 5). The tolerance for runout (FrT) is defined as 90% of the cumulative pitch tolerance, per ANSI/AGMA/ISO 1328.

Pre-roll (class PR6–PR11)

- Average helix slope error ($fH\beta m'$) (average of inspected $fH\beta'$ values)
- The tolerance for $fH\beta m'$ is equal to $fH\beta T$ per ANSI/AGMA/ISO 1328, using the same class number.
- Helix slope variation ($\Delta fH\beta$) (difference between max and min $fH\beta$ values) (Tolerance = $2 \times fH\beta T$? (TBD))
- Helix form ($ff\beta T$)
- $ff\beta T$ = (TBD)
- Average profile slope error ($fH\alpha$m') (average of inspected $fH\alpha'$ values). Note: If significant hobbing scallops are present, measurement should follow the feed scallop angle (Figs. 4 and 5).
- Profile form. Note: If significant hobbing scallops are present, measurement should follow the feed scallop angle (Figs. 4 and 5).
- Cumulative pitch. Note: If significant hobbing scallops are present, measurement must follow the feed spiral (Figs. 2 and 3).

Runout. Note: If significant hobbing scallops are present, measurement should follow the feed scallop angle (Figs. 3 and 4). The tolerance for runout (FrT) is defined as 90% of the cumulative pitch tolerance, per ANSI/AGMA/ISO 1328.

Pre-skive/pre-hard-hob (class PSk6– PSk11)

- Average helix slope error ($fH\beta m'$) (average of inspected $fH\beta'$ values)
- The tolerance for $fH\beta m'$ is equal to fHβT per ANSI/AGMA/ISO 1328 using the same class number.
- Helix slope variation ($\Delta fH\beta$) (difference between max and min $fH\beta$ values) (Tolerance = $2 \times fH\beta T$? (TBD))
- Average profile slope error ($fH\alpha$m') (average of inspected $fH\alpha'$ values). Note: If significant hobbing scallops are present, measurement should follow the feed scallop angle (Figs. 4 and 5).
- Cumulative pitch. Note: If significant hobbing scallops are present, measurement must follow the feed spiral (Figs. 2 and 3).

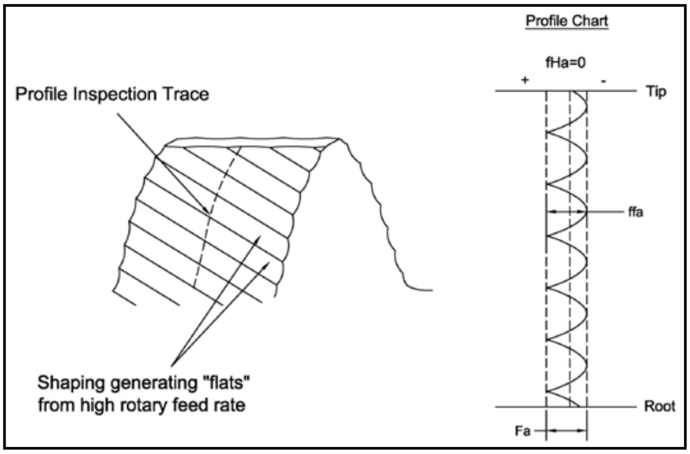

Figure 6 Using a high rotary feed rate during shaping may be desirable for optimizing productivity; but this can leave significant generating flats that are similar to hobbing scallops, yet are more pronounced in the profile direction. Such generating flats will influence profile inspection, causing significant total and form errors, but should have minimal effect on profile slope. Note: Hobbed gears with a low teeth-to-thread ratio can also exhibit similar generating marks on the profile.

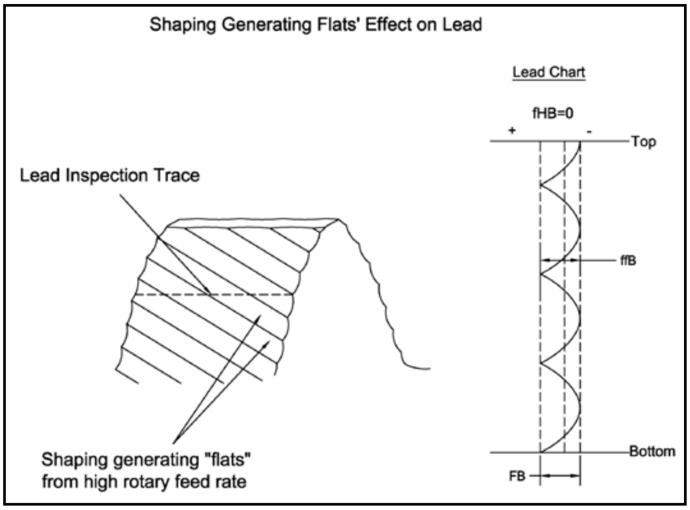

Figure 7 Using a high rotary feed rate during shaping may be desirable for optimizing productivity, but this can leave significant generating flats similar to hobbing scallops. Because these generating flats may cross the tooth flank diagonally, they can also appear on lead traces.

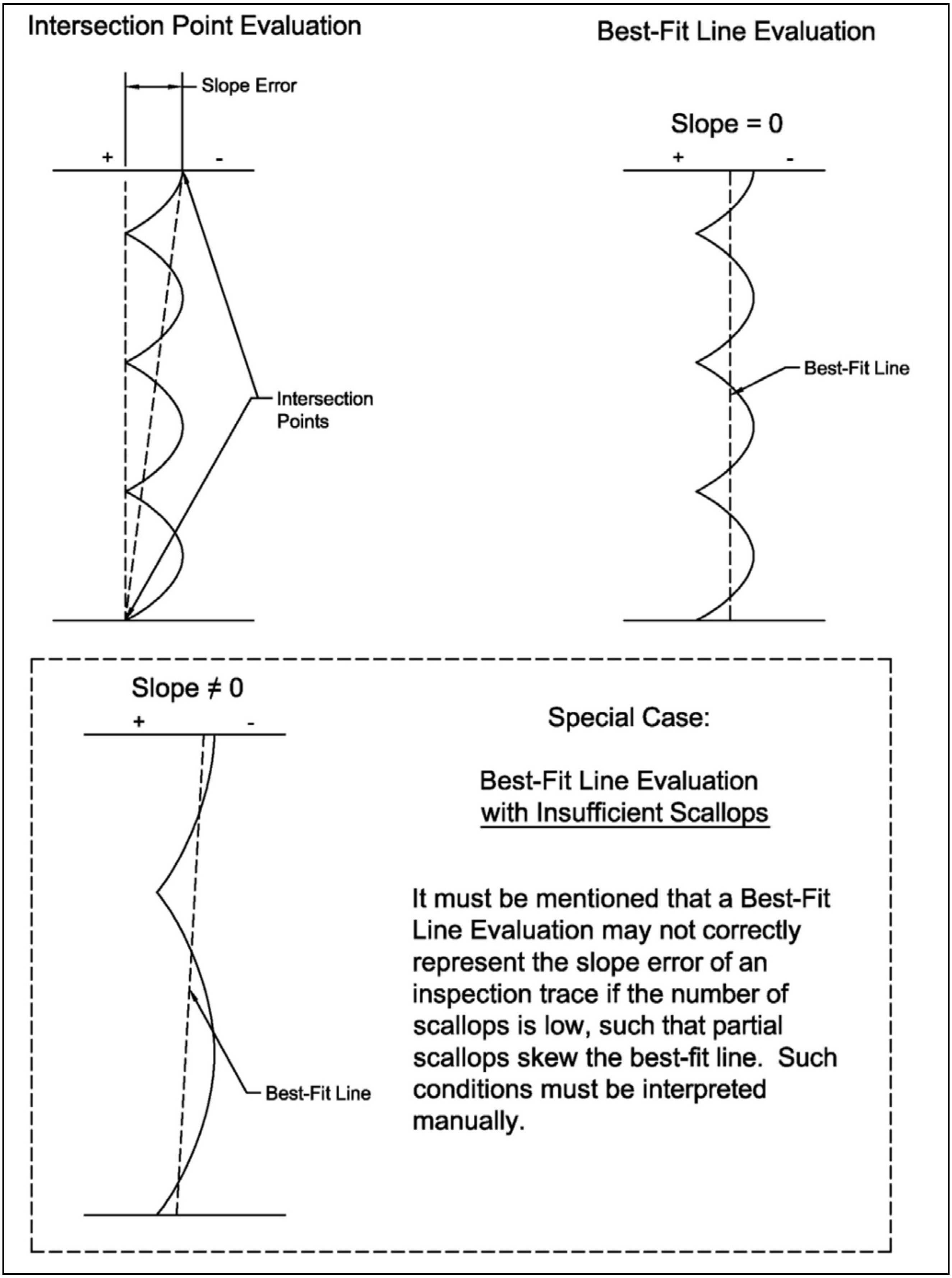

Figure 8 **It is extremely important to use a best-fit evaluation for slope deviation, as illustrated above. The use of a simple intersection point evaluation may incorrectly induce slope error and slope variation from tooth to tooth, depending on where the inspection trace crosses the evaluation zone lines.**

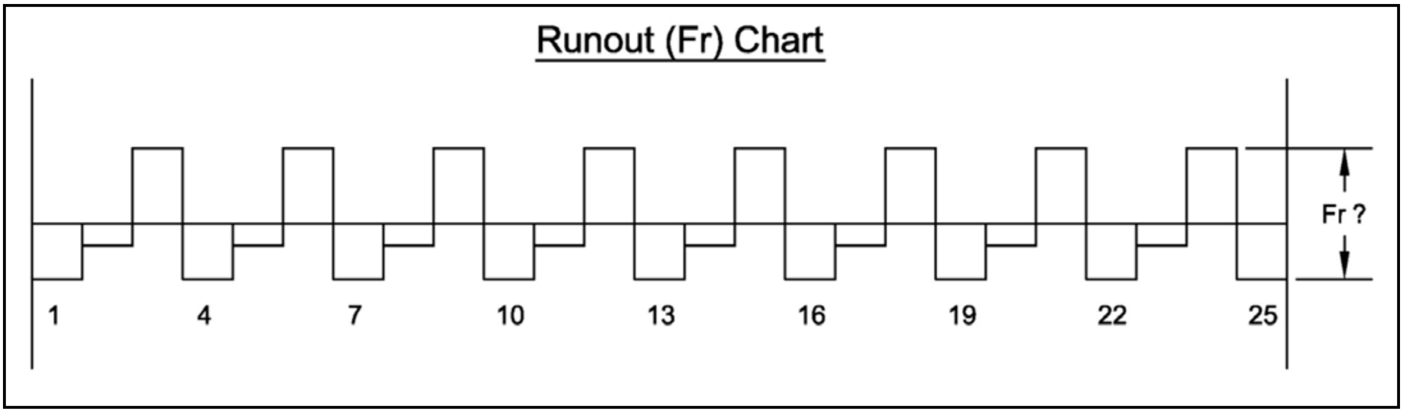

Figure 9 This example shows a gear with *no actual runout*. However, because it was cut with a 3-thread hob, an analytical inspection machine will interpret errors induced by the hob's thread-to-thread errors as runout (*Fr*). Note: The above chart could just as equally represent individual pitch (*fp*) or cumulative pitch (*Fp*) error of a gear cut with a 3-thread hob, and may not accurately represent true spacing errors.

Best Practices for Properly Evaluating Typical Pre-Finish Gears

- ***Scallops and generating flats are not "errors."*** On pre-finish gears, scallops and generating flats are expected—and often desirable—geometric features. As such, they should not affect the evaluated quality (Figs. 1, 2, 4, 6 and 7).
- ***Use the "follow-the-feed" technique for pitch inspection***. Hobbing produces a feed scallop that inherently spirals up or down the gear face width, with the scallops repositioned incrementally from tooth to tooth. One full revolution of the gear reveals that the scallop has spirally advanced according to the process feed rate (distance per gear revolution). A problem arises when a standard pitch inspection is performed on such a gear, especially with large pre-finish feed scallops. If the pitch inspection is performed in a simple plane, then each tooth's pitch measurement will be taken in a different relative location within the scallop as the scallops incrementally advance (Fig. 2). If left uncorrected, the pitch measurement will display the topography of a feed scallop, superimposed on top of the actual gear pitch results. To correct for this, the pitch measurements can be incrementally advanced along with the feed scallop so that all pitch measurements are taken in the same relative location within each scallop. This technique is often called "follow-the feed" (Fig. 3).
  Special case note: When "follow-the-feed" pitch inspection is used in conjunction with an intentional off-lead condition, a phantom pitch error component will appear, due to the fact that the feed spiral is progressing along a leaning helix; after completing the pitch inspection of all teeth, the inspection probe has progressed one full scallop above or below where it began. This will appear as a small pitch error per tooth around the gear, but it will appear as a large pitch error from last tooth to first tooth, with the phantom pitch error being equal to the helix difference from one scallop to the next. Because analytical inspection machines use pitch inspection results to calculate runout, evaluated runout will also have a phantom component as well.
- ***Use the "follow-the-scallop" technique for profile inspection.*** The generated scallop produced by hobbing typically follows a scallop angle that is not perpendicular to the root. This effect is much more pronounced on helical gears. Because a standard profile inspection trace will simply follow up the tooth in a single plane, the profile measurement may traverse partial or multiple feed scallops. As a result, the profile inspection chart will reveal the topography of the scallops traversed, superimposed on top of the actual profile measurement (Fig. 4). To eliminate the scallop geometry from the profile inspection, the profile inspection trace can be taken at the scallop angle so that the entire profile trace is performed at the same relative location within the scallop. This technique is often called "follow-the-scallop" (Fig. 5). Because the scallop angle on spur gears is usually perpendicular to the root, this technique is generally not necessary for spur gear profile inspection.
- ***Use best-fit lines for evaluating profile and helix slope.*** When analyzing either profile or helix for slope, it is important to use a best-fit line (or curve). If the slope is determined by simply identifying the intersection points of the actual trace and the evaluation lines, the presence of large feed scallops or generating flats will artificially induce slope error and slope variation from tooth to tooth (Fig. 8).

If there are enough scallops present between the evaluation lines, then a best-fit line should accurately represent the overall trace. However, in special cases where there are very few scallops present, the best-fit line may be skewed.

- ***Beware of profile evaluation range***. For a pre-finish gear, it may be necessary to modify to the evaluated range of the profile inspection. There are two primary reasons for this. First, a semi-topping pre-finish cutter may transition from flank to tip chamfer below the finished gear's required EAP (end of active profile). However, after more stock has been removed in the finishing process, the tip chamfer fall-off should be above the EAP, as required. Therefore, in the pre-finish state, the upper evaluation line may need to be adjusted below the required EAP to avoid evaluating the tip chamfer fall-off as profile error. Second, a pre-finish cutter may produce an undercut that begins above the TIF (true involute form) or SAP (start of active profile) diameter. Similar to the upper evaluation line, it may be necessary to adjust the lower evaluation line above the required TIF or SAP to avoid evaluating undercut as profile error.
- ***Be aware of hob thread-to-thread error's effect on runout and pitch errors***. Analytical gear inspection machines usually measure the left and right flank indexes and then calculate a runout chart and runout value based on how the center distance of a theoretical ball or pin would vary within all spaces around the part. Gears cut with multiple-thread hobs often have a regular tooth spacing error that repeats in a pattern that matches the number of threads in the hob. This regular pattern will usually also appear on a calculated runout chart, and the cyclical highs and lows will also be included in the

reported runout value (Fig. 9). However, calculated runout induced by hob thread-to- thread is not true runout, does not function as true runout, and will not affect finishing processes in the same way that true runout would. Eliminating this effect from calculated runout is important because it is not productive to be chasing a runout problem that does not exist, and the use of multiple-thread hobs should be encouraged (when appropriate), and not penalized, for optimizing productivity. Therefore, a solution is needed to remove such cyclical highs and lows from calculated runout. Possible solutions include, but are not limited to:

- Apply a low-pass filter to the index values prior to calculating runout.
- Apply a low-pass filter to the runout chart.
- Determine the cyclical pattern and subtract its effect from the calculated runout value.
- Perform a lead scan for each tooth's pitch measurement to take an average of all represented hob threads on each flank.

Note that the hob's thread-to-thread errors will also appear as individual pitch error (*fp*) as well as cumulative pitch error (*Fp*). If the number of hob threads is not evenly divisible into the number of gear teeth, and they do not share a common multiple, then the pitch errors reported by an analytical inspection machine will not accurately represent true spacing errors. This is due to the fact that all hob threads will be represented on each flank (in preceding or subsequent feed scallops), given sufficient face width.

Special case: helical gears with very few teeth. Hobbed gears with very few teeth often have pronounced generating flats. These generating flats will appear as scallops on the profile trace. (Note: A profile inspection turns a curve into a straight line. Interestingly, flats that are present on the profile will appear as curves or "scallops" on a profile inspection.) On helical gears, the relative position of these generating flats ("scallops") will change from tooth to tooth — just as feed scallops progress from tooth to tooth. As a result, a pitch inspection of the gear done at a constant diameter will traverse these "scallops" and the index chart will show the topography of the "scallop" progression, in a manner nearly identical to the phenomenon shown in Figure 2, even if a slow feed rate is used and actual feed scallops are not visible. This will lead to phantom pitch and runout readings. At this time, no solution is offered for this phenomenon. This is mentioned purely for informational purposes.

Evaluating Profile and Helix with Modifications

Gears are frequently designed with intentional modifications in order to improve their rolling characteristics. Profiles may have crown, tip relief, and root relief, which deviate from the perfect involute to compensate for tooth deflections under load. Similarly, helix crown or edge reliefs are often specified to avoid edge contact. However, these intentional modifications may be measured as errors if they are not handled properly in inspection. ANSI/AGMA/ISO 1328 illustrates how profile and helix deviations should be measured as they relate to "design profile" and "design helix." This is pertinent to this proposed pre- finish standard due to the fact that pre-finish gears will often have different modifications than the finished gear, or may have no profile or helix modifications whatsoever. This is due to the possibility that the prescribed modifications may be intended to be achieved with the finishing process, or that different modifications are desired in the pre-finish state due to changes that may occur during heat treatment. Therefore, the in-process "design profile" and "design helix" used for evaluation in the pre-finish state may differ from the final gear requirements and should be defined by the responsible manufacturing/process engineer, quality engineer, or other responsible individual.

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Finite Element Analysis of Tooth Flank Fracture Using Boundary Conditions from LTCA

Baydu Al, Rupesh Patel and Paul Langlois

This paper demonstrates an application of the tooth interior fatigue fracture (TIFF) analysis method, as implemented in SMT's MASTA software, in which loaded tooth contact analysis (LTCA) results from a specialized 3-D contact model have been utilized to determine the load boundary conditions for analysis of tooth flank fracture (TFF). In contrast to existing TFF methods, which use analytical Hertzian contact stress formalisms, 2-D finite element analysis has been utilized. This method allows for the full stress field to be analyzed while retaining quick analysis times (compared to full finite element contact analysis) for the calculation of stress history and the estimation of residual stresses leading to fast optimization. This method also allows calculation of residual stresses by applying a transformation strain profile or the input of analytical profiles available in the open literature. This paper also demonstrates differences between the calculated stress profile based on MackAldener's methodology and an empirical calculation method initially proposed by Lang and adapted for TFF calculation. The paper reproduces TFF results obtained by Witzig using the described methodology and a reasonable agreement of the trends has been presented.

Introduction

Gears are case hardened to produce residual stresses at the surface, improving wear resistance, bending fatigue, and contract fatigue strength. These beneficial, compressive stresses are balanced by tensile stresses within the core. This poses an increased risk of fatigue crack initiation in the material below the surface. Both tooth flank fracture (TFF), also known as tooth flank breakage (TFB), and tooth interior fatigue fracture (TIFF), describe a failure mode where a subsurface fatigue crack initiates close to case core boundary, at approximately mid-height on the tooth. Previous research (Refs.1-8) has established that the direction in which the crack propagates and the appearance of the associated fracture is dependent on the flank loading (i.e. — single-stage loading vs. idler usage). Although there does not appear to be total agreement in the literature, TIFF (failure with reverse loading) and TFF (failure with single flank loading) appear to have very similar characteristics and crack initiation mechanisms. However, as shown in Figure 1, the final fracture shape is different, due to TIFF having near symmetric total stresses along the tooth centreline (with two possible initiation points per tooth).

Due to their similar characteristics both TIFF and TFF can be analysed using similar techniques. TIFF and TFF failures can appear at loads below the allowable loading conditions for pitting and bending fatigue failure modes based on internationally accepted calculation procedures (such as ISO 6336 (Ref. 9) and AGMA 2101(Ref. 10)). Therefore, an understanding of TIFF and TFF failure modes is required at the design stage to avoid durability issues in the field. Previous research has shown that TFF and TIFF risk is dependent on the gear macro geometry, loading, and hardening properties. At this writing there is no currently standardized method to assess the probability of this type of failure and the relative importance of the influencing factors. It is worth noting, however, that TFF is an active topic within the ISO gearing committee that is currently working on a draft standard — ISO/DTR 19042 — for the calculation of tooth flank fracture performance.

In this paper we provide a brief summary of the current calculation methods found in the literature for both TFF and TIFF, as recently discussed in Al and Langlois (Ref. 11). The currently proposed approaches for TFF and TIFF all have similar, fundamental approaches consisting of four stages:

1. Calculation of stress history
2. Calculation of specification of residual stresses
3. Calculation of equivalent stresses using some fatigue criterion
4. Comparison with some initiation thresholds based on field experience or experiments

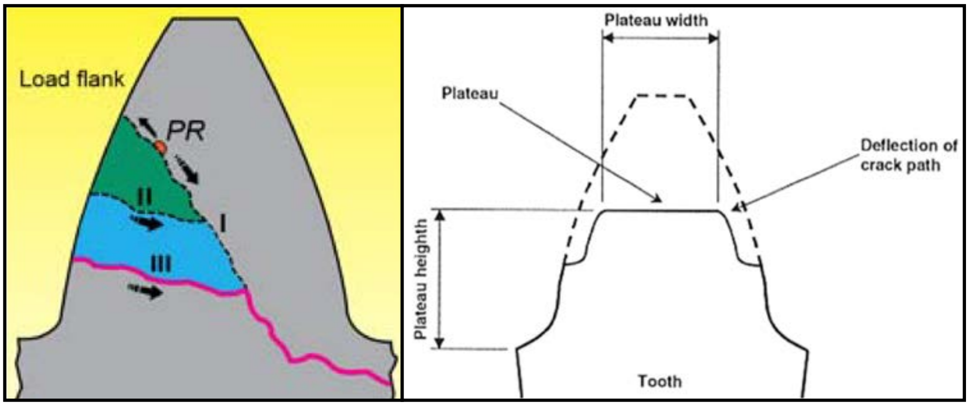

Figure 1 Expected crack propagation paths for TFF (Ref. 4) (left) and TIFF (Ref. 1) (right).

This paper was first presented at the 2016 CTI Symposium USA.

Each of the calculation methods described below differ in some of the details of the above steps. Further, the applicability of the methods depends on the assumptions made and implementation details of each stage. These calculation steps could be interchanged between methods creating a number of permutations of possibilities.

Tooth flank fracture calculation methods. To the authors' knowledge, there are two main TFF load capacity calculation methods proposed in the literature.

The first model was developed by FZG. This method has been published in Witzig (Ref. 8) and Tobie, et al. (Ref. 6) and Boiadjev, et al. (Ref. 7); and relies on calculation of the local stress history based on a shear stress intensity hypothesis of Hertter (Ref. 12). The method has significant empirical contributions and is limited in applicability due to the empirical nature of the equation used in calculating local material exposure. In the literature this method has been presented for single flank loading only; it could, in theory, be extended to consider double flank loading (i.e. idler usage), but this is not trivial. As described by Witzig, this method requires Hertzian contact stresses as inputs which can then be calculated using a gear load distribution program or via simplified analytical calculations such as those available in the standards. The method as published is also restricted to case hardened gears due to the assumptions related to the residual stress calculation. It should be noted that this method in its current form can underestimate the critical fatigue stresses if resulting residual stresses within the core are beyond negligible, since these tensile stresses within the core are currently not taken into account. This assumption for the residual stresses is only valid when the core section is much larger when compared to the thickness of the case. This introduces limitations on applicability for slender teeth and extensive case hardening depths.

Ghribi and Octrue (Ref. 5) proposed an alternative calculation method for TFF load capacity. This method is more generic than that of Witzig (Ref. 8) and can be applied to both TFF and TIFF. The method proposes use of a multiaxial fatigue criterion and considers the importance of including tensile stresses in the core. The stress history is calculated using the Hertzian contact stress calculations of ISO/TR 15144-1 (Ref. 13) micropitting load capacity calculation standard, together with a proposal of Johnson (Ref. 14) to calculate stress at a depth inside the tooth. Method A of ISO/TR 15144-1 (Ref. 13) is based on using the results of a 3-D gear loaded tooth contact analysis, however only Hertzian contact stresses calculated by the standard have been considered in the analysis. Addition of stresses due to bending has been mentioned as planned future work, as these stresses could have an impact on the calculated stress states.

Neither of these methodologies is based on finite element analysis (FEA), although they clearly could be adapted to do so. However, using general FE packages requires considerable time and computational power to set up and run analyses.

Tooth interior fatigue fracture calculation methods. MackAldener (Refs. 1-3) has shown that an analysis method based on 2-D FEA can be utilized to analyze the risk of TIFF and determine optimum macro geometry, material, and case hardening properties. In this analysis MackAldener used the gear load distribution analysis program *LDP* (*Ohio State University Load Distribution Program*) to calculate the total force on one tooth at different phases within the mesh cycle. The calculated force was then applied to a 2-D FE model of a single pair of teeth in contact as a torque after normalizing with the face width. A contact analysis was then run on the 2-D FE model in order to calculate the stress history. MackAldener's papers show the evolution of the methodology used to estimate residual stresses and material properties. While early papers (Ref. 1) described a methodology where transformation strain and material fatigue properties were assumed constant throughout the case, in later papers these assumptions were replaced with a non-homogeneous transformation strain profile and material fatigue properties that depend on the hardness profile were used within the case.

Due to the complexity of setting up and running MackAldener's FE-based method within a general FE package, MackAldener (Ref. 2) also proposed a simpler semi-analytical method. This method was proposed for rapid calculation, design parameter studies, and optimization — but with some compromise in the accuracy of the results. In the analysis of results presented in MackAldener (Ref. 2), the crack initiation risk factor result was seen to be over-predicted, as compared to MackAldener's FE-based method, by a maximum of 20%.

MackAldener used a factorial design to evaluate the effect of gear design parameters on TIFF risk and concluded that TIFF failure can be avoided if the slenderness ratio is reduced, tensile residual stresses are reduced, the gear is not used as an idler gear, and optimum case and core properties are used.

Recently Al and Langlois (Ref. 11) demonstrated a modification to the analysis of TIFF based on MackAldener's FE based method, as implemented in SMT's *MASTA* software, in which the loaded tooth contact analysis (LTCA) results from a specialized 3-D elastic contact model have been utilized to determine the load boundary conditions for TIFF analysis. This replaces a computationally expensive, explicitly modelled FE contact analysis with simple load boundary conditions obtained by a separate, specialized gear LTCA. This method has been validated against MackAldener's results and it is this method that is used throughout this paper.

In the following section the models to be used are introduced. Justification is given for model selection. The results section demonstrates the application of the described methodology as a means to investigate the risk of fatigue crack growth beneath the surface of a single flank loaded gear. Results are compared with those available within the literature (Ref. 8). Good correlation is shown for the prediction of the risk of TFF fatigue crack growth.

Methodology and Analysis

The methodology used for the analysis of TIFF and TFF has been implemented in SMT's *MASTA 7*, and previously described by Al and Langlois (Ref. 11).

MASTA's implementation is derived from MackAldener's finite element method, but the need for a full FE tooth contact analysis has been removed by using loading conditions calculated using *MASTA's* specialized loaded tooth contact analysis. MackAldener also simplified his FE analysis in a later stage, not for calculation of crack initiation risk factor, but when investigating the crack propagation mechanism during the TIFF (Ref. 16). This method removes the complexity of the contact analysis and speeds up the calculation while reducing the computational requirements.

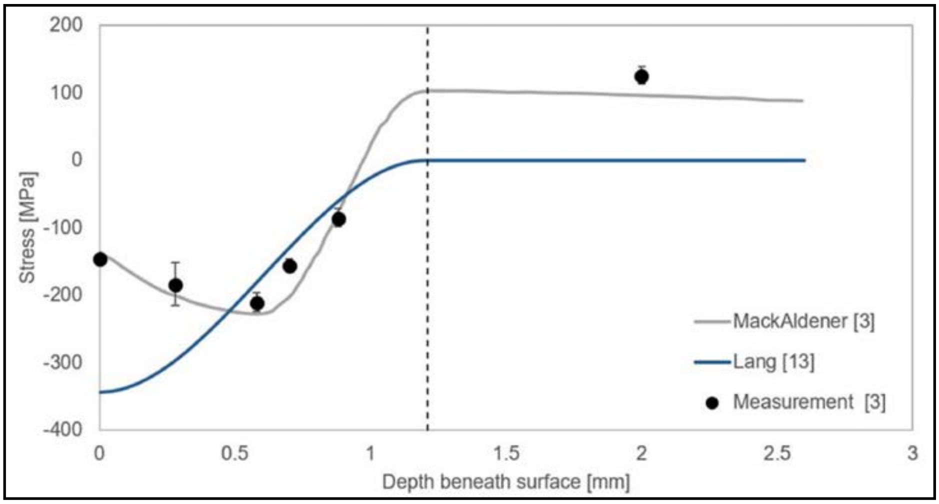

Figure 2 Experimentally measured hardness profile and curve fits of MackAldener, together with a number of empirical models available in the literature. The total case depth of 1.2 mm is marked by a dashed line; effective case depth where hardness drops below 550HV is 0.68 mm (required for empirical models). See AI and Langlois for more information.

Analysis of the stress history. *MASTA's* 3-D loaded tooth contact analysis (LTCA) model combines an FE representation of bending and base rotation stiffness of the gear teeth and blank with a Hertzian contact formalism for the local contact stiffness. This calculation includes the effect of extended tip contact where the effective contact ratio is increased under load due to tooth bending. This effect can be particularly important for slender tooth gears that are also more at risk of TIFF.

This model is used to determine load boundary conditions at a selected number of time steps through the mesh cycle. At each time step the load distribution between and across the teeth is calculated and at each of the contact lines, load positions, load magnitudes and Hertzian half-widths are obtained.

A separate, fine 2-D mesh of the gear tooth is then built automatically, using plane strain elements. At each time step within the mesh cycle the position and distribution of the load is determined from the results of the 3-D tooth contact analysis and applied to the 2-D FE mesh using the average load position and Hertzian half-width. In the results presented below, the finite element mesh was sized according to the Hertzian half-width and a refinement study was performed to check the convergence of the results.

Hardness profile and material properties. The variation of the material properties within the case and core play an important role in TIFF; however, many assumptions have been made in previous analyses in this area. Since the analyzed gear is case hardened, the material properties are not constant throughout the tooth. The critical shear stress and fatigue sensitivity to normal stress, in the critical plane criterion, are also expected to vary with location. As with MackAldener, for our analysis we have assumed that these properties vary in the same way as an assumed hardness profile.

Hardness profile definition used by MackAldener (Ref. 3):

$$H(z) = H_{surface} \cdot g\left(\frac{z}{\bar{z}}\right) + H_{core} \cdot \left(1 - g\left(\frac{z}{\bar{z}}\right)\right) \quad (1)$$

$$g\left(\frac{z}{\bar{z}}\right) = 1 - 3 \cdot \left(\frac{z}{\bar{z}}\right)^2 + 2 \cdot \left(\frac{z}{\bar{z}}\right)^3$$

where, $H_{surface}$ and H_{core} are the hardness at the surface and core, respectively, g is a function that determines the variation between the case and the core defined by MackAldener, z is the normal depth at the point considered and $\bar{z}$ is the total case depth.

This method relies on measurement of the total case depth which is often neither measured nor known. Therefore, as an alternative, a hardness measurement at a defined effective case depth is used. In such cases a different hardness profile may be utilized, i.e. Lang (Ref. 17), which in fact is the hardness profile used by Witzig (Ref. 8).

Hardness profile definition used by Lang (Ref. 17) *and adopted by Witzig* (Ref. 8):

$$H(z) = H_{surface} \cdot g\left(\frac{z}{CHD}\right) + H_{core} \cdot \left(1 - g\left(\frac{z}{CHD}\right)\right) \quad (2)$$

$$g\left(\frac{z}{CHD}\right) = 10^{-0.0381 \cdot \left(\frac{z}{CHD}\right) - 0.2662 \cdot \left(\frac{z}{CHD}\right)^2} \quad (3)$$

in which CHD is the effective case depth where hardness drops below 550 HV.

Comparison of hardness profile definitions. Figure 2 shows a comparison of the hardness profile measurement and curve fit proposed by MackAldener with other hardness profile models found in the literature. Unless otherwise stated, for this article MackAldener's curve fit has been used. It is interesting to note that the hardness profile model proposed by Thomas (Ref. 19) has been found to give the best comparison against MackAldener's measurement, and that of Tobe et al. (Ref. 20) is also close. The Lang method could lead to a difference in the CIRF, since fatigue properties are expected to differ near the case-core boundary.

Determination of material properties required by multiaxial fatigue analysis. In the current implementation, the material properties are assumed to vary continuously between case and core in the same manner as the hardness profile. This assumption is not required if variations of the material properties are known.

$$\sigma_{crit}(z) = \sigma_{crit,surface} \cdot g\left(\frac{z}{\bar{z}}\right) + \sigma_{crit,core} \cdot \left(1 - g\left(\frac{z}{\bar{z}}\right)\right) \tag{4}$$

$$a_{cp}(z) = a_{cp,surface} \cdot g\left(\frac{z}{\bar{z}}\right) + a_{cp,core} \cdot \left(1 - g\left(\frac{z}{\bar{z}}\right)\right) \tag{5}$$

The surface fatigue resistance of a gear flank and root can be improved by shot peening. This improvement is due to an increase in the compressive stresses in a thin layer close to the surface; this layer is very thin compared to the case hardening layer. Shot peening properties for depth and the effect on the critical shear stresses are required. In the current implementation both the depth and the effect on the critical shear stresses are assumed to be constant, the latter specified via a shot peening factor.

Residual stress analysis. Residual stresses influence the stress states within the gear tooth. These stresses are not load dependent and assumed to be constant over time. Residual stresses due to case hardening and shot peening are superimposed.

Residual stress calculation according to MackAldener:

Utilizing the 2-D mesh used to calculate the stress history due to flank loading, residual stresses are predicted by performing a separate FE analysis.

The transformation strain profile is isotropic and measured relative to the core. This profile has been presented as a piecewise polynomial with smooth connections by MackAldener (Ref. 3):

$$\varepsilon_t(z) = \begin{cases} \varepsilon_1 + 4\cdot(\varepsilon_2 - \varepsilon_1)\cdot\left(\left(\frac{z}{\bar{z}}\right) - \left(\frac{z}{\bar{z}}\right)^2\right) & \text{if } 0 \le z \le \frac{\bar{z}}{2} \\ -4\cdot\varepsilon_2\cdot\left(1 - 6\cdot\left(\frac{z}{\bar{z}}\right) + 9\cdot\left(\frac{z}{\bar{z}}\right)^2 - 4\cdot\left(\frac{z}{\bar{z}}\right)^3\right) & \text{if } \frac{\bar{z}}{2} \le z \le \bar{z} \\ 0 & \text{if } \bar{z} \le z \end{cases} \tag{6}$$

where, ε_1 is the transformation strain at the surface; and, ε_2 is the maximum transformation strain.

The volume expansion in the surface layer due to the case hardening process is modeled by applying a temperature profile to the FE model. The temperature profile applied is the same as the transformation strain profile when the coefficient of thermal expansion is set to 1. All side nodes are allowed to move only in the radial direction.

Residual stress analysis according to Lang (Ref. 17) and modified by Witzig (Ref. 8):

The calculation method proposed by Lang simply requires the heat treatment type and depth from the surface to be known in order to calculate tangential residual stresses. As can be seen from the equations, only compressive residual stresses are calculated via this method. Note that $HV(z)$ in the equation refers to Lang's hardness profile, as opposed to MackAldener's.

$$\sigma_{residual}(z) = \begin{cases} -1.25\cdot(H(z) - H_{core}) & \text{if } H(z) - H_{core} \le 300 \\ 0.2857\cdot(H(z) - H_{core}) - 460 & \text{if } H(z) - H_{core} > 300 \end{cases} \tag{7}$$

This model has been used by both the TFF calculation methods proposed by Witzig (Ref. 8) and Ghribi-Octrue (Ref. 5). The implementation described by Ghribi and Octrue can, however, also calculate tensile residual stresses by considering a force balance across the teeth. (*Note*: this improvement on Lang's model has not been considered for this article.)

Comparison of residual stress calculation methods. Figure 3 shows the results presented by MackAldener for the variation of residual stresses with depth beneath the surface — both using the analysis method above and from measurements carried out by MackAldener. Figure 3 further compares this residual stress profile with that proposed by Lang (Ref. 17) and used by Witzig (Ref. 8) in the investigation of TFF. Interestingly, the profiles differ quite notably. This may be due to a significant material dependency not considered, but the exact reason is currently unknown and further understanding is required. It should be noted that the resulting calculated residual stresses can change from one mesh position to another due to the variation in tooth thickness.

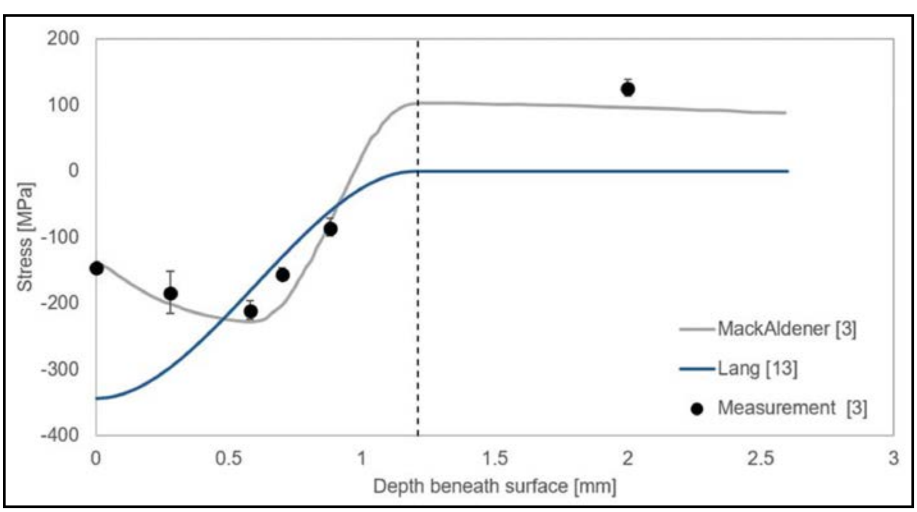

Figure 3 Variation of residual stresses with increasing depth for the original gearset defined by MackAldener. The total case depth is marked by a dashed line. See Al and Langlois for more information. The residual stress profile that is the result of the strain profile has been utilized with = 0.000833 and = 0.00114, as determined by MackAldener (Ref. 3).

Final stress state and fatigue crack initiation criterion. The effective stress state within the gear teeth during its load cycle is calculated, without calculating residual stresses at each step, by superimposing the calculated stress history states and the initially estimated residual stresses.

The Findley multiaxial fatigue criterion (Ref. 18) is then used to analyze the stress history and assess the possibility of failure. Within our analysis the Findley critical plane stress has been calculated for every 5 degrees of inclination at each node. The value of 5 degrees was chosen, instead of every 1 degree used by MackAldener (Ref. 2), as results did not show a significant dependency on this value. This is confirmed by the cases presented in the results section of this paper where differences between using an inclination increment of

2.5 degrees over 5 degrees is less than 0.05%.

The Findley stress is calculated as: $\sigma_F = \tau_a + a_{cp} \cdot \sigma_{n,max}$
where, τ_a is the shear stress amplitude and $\sigma_{n,max}$ is the maximum normal stress.

Variation of the material properties within the tooth are related to the hardness profile, as described above.

The ratio between the maximum Findley critical plane stress and critical shear stress is a measure of the risk of crack initiation. This metric is called the crack initiation risk factor (CIRF).

Table 1 Gear and cutter data for the designs considered by Witzig (Ref. 8)

Spur gear set designation	40/41		67/69	
	Pinion	Wheel	Pinion	Wheel
Number of teeth	40	41	67	69
Centre distance [mm]	200		200	
Module [mm]	5		3	
Pressure angle [°]	20		20	
Profile shift coefficient	-0.23	-0.2456	-0.61	-0.6169
Tip diameter [mm]	205.6	210.2	201.2	207.16
Face width [mm]	18	18	18	18
Number of teeth measured for chordal span	5	5	7	6
Average chordal span measured [mm]	68.287	68.304	59.02	50.237
Assumed finish stock [mm]	0.04	0.04	0.04	0.04
Normal thickness of the cutter [mm]	7.854		4.712	
Protuberance [mm]	0.2		0.15	
Protuberance height [mm]	2.853		1.383	
Addendum for cutter [mm]	7.4		4.71	
Dedendum for cutter [mm]	5.6		4.54	
Cutter tip edge radius [mm]	2		0.81	
Cutter fillet radius [mm]	1		0.6	
Core Hardness HV	405		410	
Surface Hardness HV	695		695	
Effective Case Depth, HV550 [mm]	0.69		0.5	

Table 2 Estimated transformation strains and total case depth for gears defined in Table 1, assuming stresses at the surface can be obtained using Lang (Ref. 17).

Spur gear set designation	40/41		67/69	
Location	Surface, ε_1	Mid-case, ε_2	Surface, ε_1	Mid-case, ε_2
Transformation strain	0.000609	0.001163	0.000618	0.00121
Estimated total case depth	1.38 mm		0.98 mm	
Critical stress at surface	235 MPa			
Critical stress in core	675 MPa			

Validation Results

Details of gear tooth geometries and cutters specified by Witzig (Ref. 8) are provided in Table 1. Gearset 67/69, with pressure angle 15° (details not provided here), could not be created from supplied tooth thickness and center distance information within Witzig's (Ref. 8) thesis. For each design, the CIRF throughout the tooth was calculated and trends have been compared to those obtained by Witzig (Ref. 8). Both of the gear tooth geometries used within this article do not include any profile modification other than a generous tip relief.

Estimation of material fatigue properties for Findley multiaxial fatigue criterion. Fatigue properties of the core and surface material required for Findley multiaxial fatigue criterion were estimated. Fatigue sensitivity to normal stress was assumed 0.3 within the core, and 1 at the surface. The local material shear strength definition used by Witzig (Ref. 8) — $\tau_{crit} = 0.4 \cdot HV(z)$ — was used. The calculation of the Findley critical shear stress has been carried out according to MackAldener (Ref. 3).

Estimation of transformation strains for MackAldener's residual stress calculation. MackAldener (Ref. 3) gives a clear description of how transformation strain values can be estimated at the surface, and the mid-case depth using two residual stress simulations. First, simulation with transformation strain 1 on the surface and 0 at the mid-case depth was performed. The surface stress calculated by Lang (Ref. 17) was then compared to the calculated residual stress; linear interpolation was used to calculate the transformation strain at a surface point leading to the surface stress of Lang. A similar analysis was carried out at the mid-total case depth. The transformation strain profile definitions used within the current analysis are provided in Table 2.

Analysis Results

Witzig (Ref. 8) has run numerous experiments with test gears and validated their calculation model, suggesting a critical value of 0.8 for material exposure. These gearsets were designed to fail due to tooth flank fracture, and results were reproducible. It is important to note that failure analysis of these gearsets showed that, in the majority of cases, initial crack initiation occurred at an inclusion near the case-core boundary. However, the size and effect of these inclusions are not included within the analysis in that the material is considered homogeneous.

Figure 4 summarizes the results from Witzig (Ref. 8) for spur gear set 40/41 (Fig. 4a) and 67/69 (Fig. 4b). It should be noted that the y-axis on the right, for the maximum material exposure, is shifted to give comparable results, as the critical value for the Findley criterion is expected to be 1 and 0.8 for Witzig.

Figure 4 also shows the calculated CIRF using the proposed methodology with and without tensile residual stresses. Although not clearly seen in the figure, the gradient of calculated crack initiation risk factor, with respect to torque, grew by 10.9%, and 11.8% when residual tensile stresses are included for spur gearset 40/41 and 67/69, respectively. Further, the gradient of a line fit of the results obtained by Witzig is found to be lower than that for the proposed method.

Witzig (Ref. 8) has observed that experimental test gears fail when maximum material exposure exceeds 0.8, but this value does not consider the effect of the inclusion on the stress field. Therefore further understanding of CIRF-initiated TFF failure is required.

Given the assumptions made — including the assumption that a critical CIRF of 1 can be compared with a maximum material exposure of 0.8 — the results obtained show similar qualitative behavior, but also some significant difference in the torque, which leads to the critical metric values. Further investigation is required to understand whether this difference is down to the assumptions made in our inputs or a more fundamental difference in the formalism of the methods.

Figure 5 shows a comparison of experimental failure shapes from (Ref. 8) and crack initiation points with calculated crack initiation risk factors from the proposed method for both residual stress calculation methods considered. It should be noted that experimental failures originated from inclusions which act as stress concentration regions. Therefore the calculated maximum crack initiation risk factor may not necessarily coincide with the experimental crack initiation point. Figure 5 does, however, show good correlation between the position of crack initiation in the tests and the calculated region of high CIRF.

Furthermore, results show that the area with the highest crack initiation risk is smaller when tensile residual stresses are considered due to the profile of the residual tensile stresses into the tooth i.e.—highest risk is within the case-core boundary.

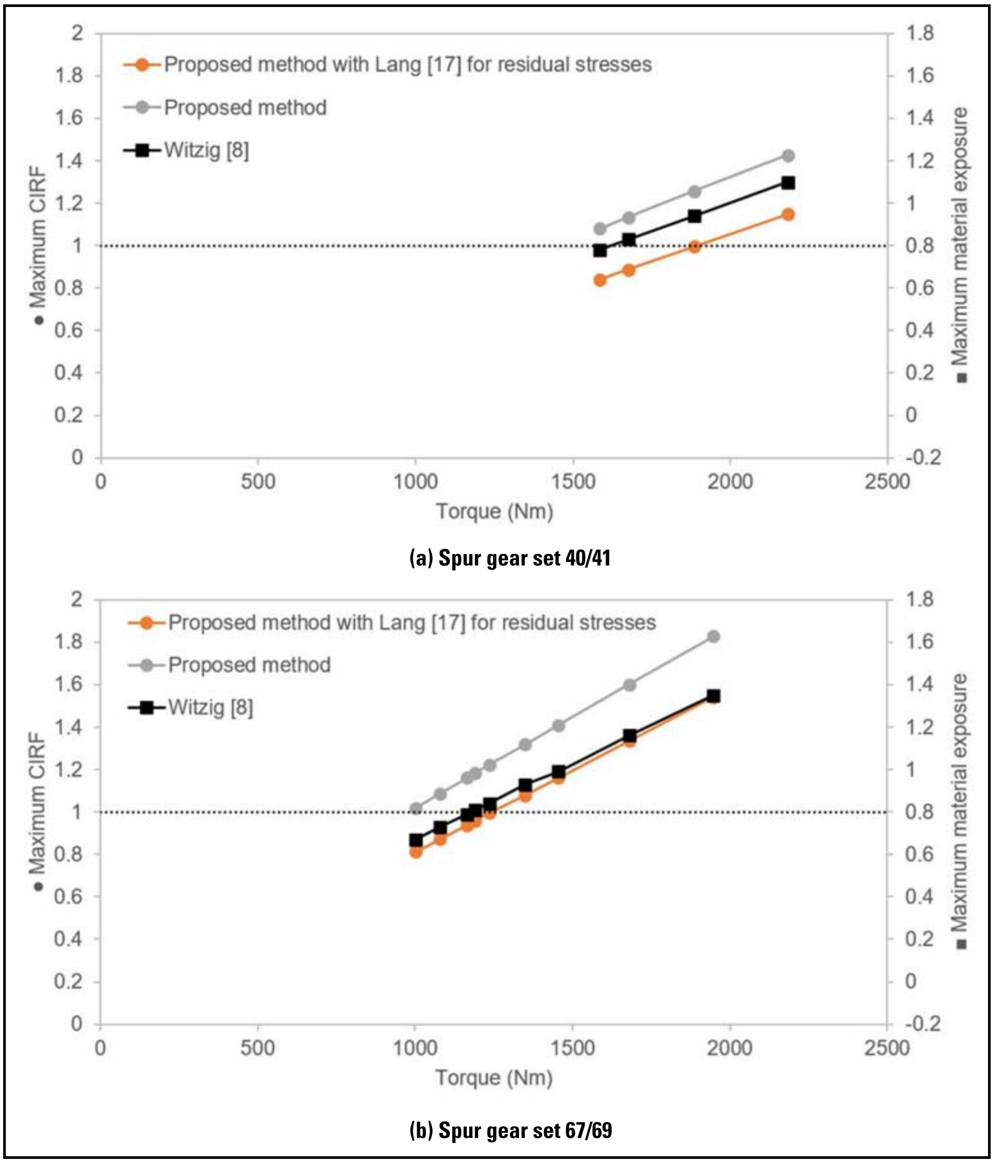

Figure 4 Comparison of the calculated maximum CIRFs from MASTA and Witzig's (Ref. 8) method. It should be noted that the *y*-axis of the maximum material exposure is shifted to give comparable results (i.e., critical value for Findley criterion is expected to be 1, and 0.8 for Witzig). The dotted line represents the expected critical value for the fatigue criterion used.

Conclusions

Application of an FEA-based analysis technique to analyze risk of TIFF crack initiation has been applied to TFF, with results presented and compared against open literature.

The conclusions from this paper are as follows:

- It is possible to analyse the risk of tooth fatigue fracture by applying a methodology based on MackAldener where computationally expensive, explicitly modelled, FE-based contact analysis is replaced with simple load boundary conditions obtained by a separate, specialized gear-loaded tooth contact analysis.
- It has been shown that there is a linear relationship between torque and material exposure, as calculated by Witzig, within the range investigated. A linear relationship has also been observed between torque and calculated crack initiation risk factor.
- As is to be expected, thresholds obtained from Witzig's calculation method and Findley are different. The critical value has been found to be close to 1, but requires further investigation.
- The calculated crack initiation risk factor is higher when residual stresses are estimated with MackAldener (i.e. when tensile residual stresses are considered within the core), compared to Lang. It has been found that the effect of the residual stresses increases with torque.
- Further studies are required to evaluate thresholds and material properties used within different fatigue criterions.

Further work on comparison of TIFF and TFF load carrying capacity with other failure modes, such as bending and pitting fatigue, are planned for future work.

It is the authors' opinion that the critical effect of material quality and inclusions is the key factor missing in the types of analyses previously presented. We would expect that this could be addressed as a factor applied to, for example, the material thresholds; however, significant field experience and further experimental studies are required to address this point.

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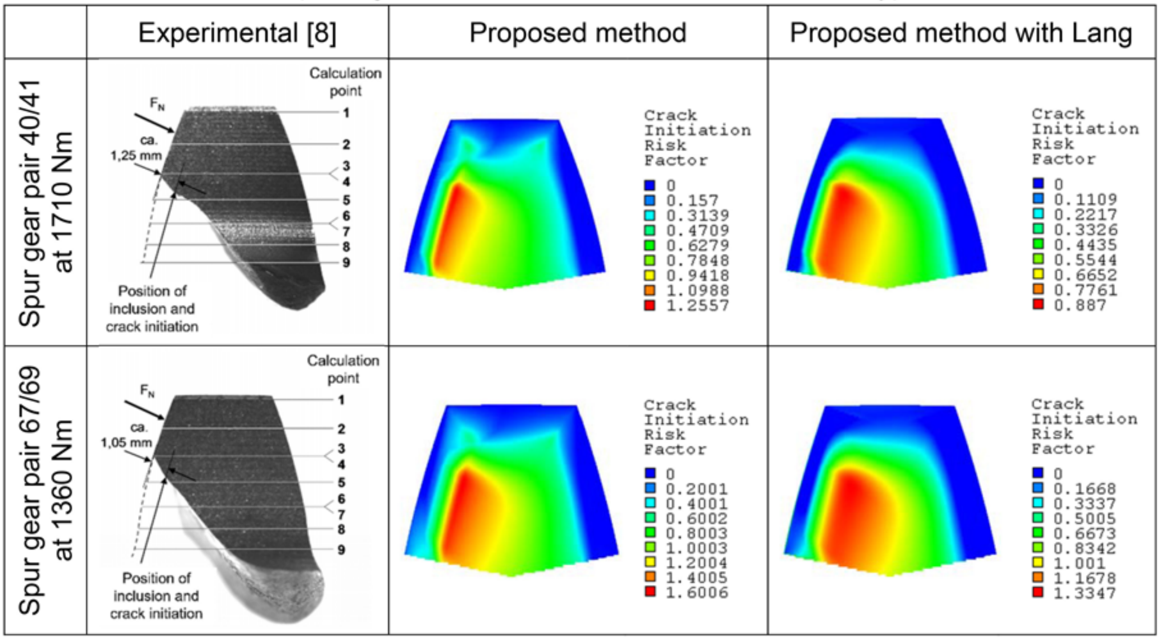

Figure 5 Comparison of experimental failure shape (Ref. 8) and calculated crack initiation risk factor; CIRF for both residual stress calculation methods considered.

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Baydu Al has been an analyst software engineer at Smart Manufacturing Technology (SMT) since October 2014. Prior to joining SMT he worked as a researcher at Nottingham University (gas turbine and transmission systems), specializing in efficiency and oil management. Since joining SMT Al has been busy contributing to MASTA's loaded tooth contact analysis, and analysis of tooth interior fatigue fracture.

Dr. Rupesh Patel has since January 2016 worked at Smart Manufacturing Technology (SMT) as a systems analyst/software developer. During this time he has been actively involved in research and development of functionality for the analysis of tooth interior fatigue fracture and for gear macro-geometry optimization. Prior to employment at SMT, Patel worked for several years as a post-doctoral researcher at the University of Nottingham, specializing in structural dynamics and focusing on energy harvesting from natural sources of vibration. Patel was also employed in Japan in relation to this work and has published numerous scientific papers over his career.

Dr. Paul Langlois is the CAE products development department manager at SMT. Having worked for SMT for 10 years, he has extensive knowledge of transmission analysis methods and their software implementation. He manages the development of SMT's software products and is a main contributor to many aspects of the technical software development. As a member of the BSI MCE/005 committee, Langlois contributes to ISO standards development for gears.

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Trescal

ACQUIRES PRECISION METROLOGY INC.

Trescal recently announced the acquisition of Precision Metrology Inc., a calibration services provider based in Wisconsin and Florida. This transaction consolidates Trescal's geographical footprint and technical coverage in the United States and has been completed with the support of Ardian, its majority shareholder.

Founded in 1980 and headquartered in Milwaukee, Precision Metrology is A2LA-accredited and has strong technical skills in dimensional, DC/Low Frequency, thermodynamics, and mechanical/dimensional repair. With a turnover of around $11 million, the company employs around 90 people and is mostly active in the Utilities, Aeronautics and Life Science sectors.

Precision Metrology's founder, Carol Shipley, will remain in her current position to continue leading the growth of the business in the coming years. This acquisition will bring Trescal's portfolio in the United States to 15 calibration laboratories, with over 400 employees.

It is the 16th acquisition since Ardian became its majority shareholder in July 2013. Deputy CEO Guillaume Caroit, at Trescal said, "The acquisition will allow us to strengthen our position in the Great Lakes area as well as in Florida. The strong technical skills and the leading and young management team are key assets for our development in the USA."

Shipley added: "We are a robust and mature calibration laboratory with a strong history in the industry serving our customers for 36 wonderful years. We are proud to become part of Trescal and will continue to strive and exceed our customers and our new owners expectations."

Kathy Boyd, president Trescal Inc. said: "We are very excited to have Precision Metrology as part of the Trescal U.S. organization. The years of experience, breadth of capabilities and services they provide to their customers will be a great addition to our offering."

Thibault Basquin, managing director Ardian Mid Cap Buyout, added: "This transaction fits perfectly with Trescal's development strategy in North America. Both Ardian and Trescal follow an ambitious roadmap, with a view to accelerating external growth. This has continued to prove its efficiency with what is the 16th acquisition since we invested in the company."

Wenzel America

PROMOTES DREW SHEMENSKI TO PRESIDENT

Wenzel America is proud to announce the promotion of **Drew Shemenski** to position of president of Wenzel America. Having spent the last two years as regional sales manager – Midwest for Wenzel, Shemenski succeeds Andy Woodward, who has been promoted to Wenzel Group marketing director. "I am eager to continue the growth and innovation which have been hallmarks of Wenzel America's success over the past several years," says Shemenski. "We also look forward to the launch of new products and technologies which will support the growth of Wenzel not only as a trusted, local partner, but also a global leader in our market."

Drake Manufacturing and Sanwa

ESTABLISH SALES AGREEMENT FOR JAPAN

Drake Manufacturing Services Co., LLC and Sanwa Seiki Co., Ltd. have formally established a sales agreement for Japan and for other designated regional Asian countries excluding China, Taiwan and Korea. This agreement provides sales, marketing and support related to all thread grinding solutions offered by Drake.

Drake is recognized globally for designing, building, automating, and servicing state-of-the-art production systems for manufacturers of parts with threads. Specifically, it provides turn-key solutions for production of ultra-high precision internal and external threads, steering gears, taps, gages, ball nuts and screws, and worms.

Anthony Tully, Drake general manager Asia, states, "Our partnership with Sanwa Seiki directly follows the opening of our new Regional Sales and Service office in Tokyo to expand Drake's Asia presence in the steering systems, aerospace, cutting tool, speed reducer, ball screw, and linear motion industries."

Established in 1950, Sanwa Seiki has a solid reputation in automotive and other key industries, and has the technical and product knowledge that customers have come to expect from the Drake brand. With its principal office in Nagoya, Japan, Sanwa Seiki is well positioned to bring Drake's service to its Japanese customer base.

LMC Workholding

INTRODUCES NEW GLOBAL PARTNER ROTOMOTORS AT IMTS 2016

LMC Workholding is introducing its newest global partner, Rotomors of Turin, Italy, at IMTS. LMC will be showing a new 2.2-meter automatic self-centering, down-clamping chuck from Rotomotors. Also in Booth No. W-1314, will be a variety of LMC's workholding product line, including face drivers, wheel chucks, cylinders, steady rests, specialty workholding products and more.

LMC Workholding celebrates 100 years of engineering and manufacturing in 2016. LMC makes all types of workholding with a heavy emphasis currently on Rotational Workholding and Aluminum Wheel Manufacturing Systems. LMC's products include standard and special chucks and cylinders up to 7 meters in diameter, manual and hydraulic steady rest systems and integration, face drivers, centers and many other types of standard and super precision chucks, cylinders and fixtures. LMC has global reach and partnerships with additional manufacturing and service in Sweden, Germany, Austria, Italy, Taiwan and China.

Ajax TOCCO

APPOINTS GENERAL MANAGER OF AFTERMARKET SALES

Ajax TOCCO Magnethermic welcomes **Joe Hawkins** as General Manager of Aftermarket Sales. Hawkins brings over 20 years of induction melting and heating expertise to the company. His entire career evolved within the metals industry, beginning with Lectrotherm, Inc. in 1991. Hawkins most recently served as Director of Sales for ReMelt Scientific.

A graduate of Kent State University, Hawkins later earned his MBA from Malone University. Joe and his wife Lore reside in North Canton, Ohio with their three children.

Ajax TOCCO proudly maintains a team of highly skilled technicians strategically located throughout the world to repair and maintain Ajax TOCCO equipment as well as competitors' equipment. In addition, the Company houses a large inventory of parts for all of their equipment including older units and some competitor equipment.

DOE & IHEA

RELEASE HEAT TREATING SOURCEBOOK

Improving Process Heating System Performance: A Sourcebook for Industry, 3rd Edition is now available for download. This Sourcebook is part of an initiative under the U.S. Department of Energy (DOE) Industrial Technologies Program (ITP) and the Industrial Heating Equipment Association (IHEA) which began more than ten years ago when the first Sourcebook was published. The ITP and IHEA undertook this project as part of a series of Sourcebook publications on industrial utility systems. Other topics in this series include compressed air systems, pumping systems, fan systems, steam systems and motors and drives.

This sourcebook describes basic process heating applications and equipment and outlines opportunities for energy and performance improvements. It also discusses the merits of using a systems approach in identifying and implementing these improvement opportunities. It is not intended to be a comprehensive technical text on improving process heating systems, but serves to raise awareness of potential performance improvement opportunities, provides practical guidelines, and offers suggestions on where to find additional help.

IHEA's mission is to provide services that will enhance member company capabilities to serve end users in the industrial heat processing industry and improve the business performance of member companies. Consistent with that mission, IHEA supports energy efficiency improvement efforts that provide cost savings, performance benefits, and other competitive advantages that enable success in the global marketplace.

Improving Process Heating Performance Sourcebook is available for download at no charge on the IHEA website, *www.ihea.org*. Click the Publications tab and scroll down to Books.

FANUC America

EXPANDS CNC TRAINING PROGRAM

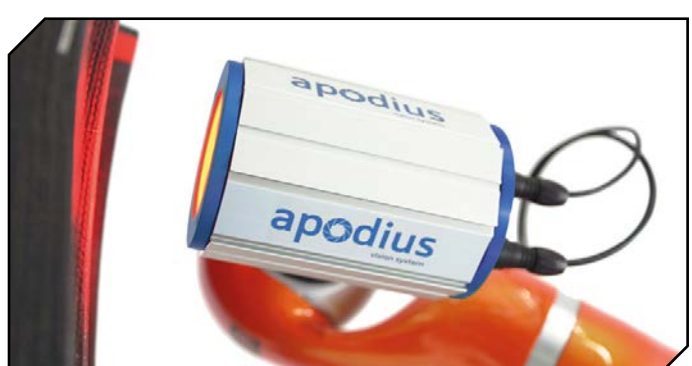
FANUC America, a provider of CNC, robotics and factory automation solutions, is expanding its CNC training program with additional course offerings and significant facility upgrades. FANUC currently offers a wide range of training courses covering CNC maintenance, operations and programming, servo and laser maintenance, Ladder/PMC editing, FANUC Picture Development, G-code programming, custom macro programming, conversational programming and more. FANUC training courses include hands on instruction and real-world troubleshooting and are taught by experienced professionals who have been trained at FANUC headquarters in Japan.

FANUC training program improvements include: New online training courses including mill, lathe, and custom macro programming courses as well as general maintenance and integrator training, more than 50 new CNC systems were purchased for classroom training, new training facility in Auburn, WA (Seattle) and classroom upgrades at the Huntersville, NC (Charlotte) and Rochester Hills, MI (Detroit) facilities, machine tools with FANUC controls installed at several FANUC locations for use in select classes.

Hexagon AB

ACQUIRES APODIUS GMBH

Hexagon AB recently announced the acquisition of Apodius GmbH, a start-up specialized in measurement solutions for fiber composite components. Based in Germany, Apodius was founded in 2012 and serves original equipment manufacturers (OEMs) as well as suppliers in the automotive, aerospace, electronics and appliances industries with the development, production and integration of measurement solutions for carbon fiber. Carbon composite materials are enablers for lightweight structures in various high-growth fields like aerospace, automotive and renewable energy. Furthering Hexagon's strategy to expand its expertise and portfolio to support current and



future manufacturing trends, the Apodius acquisition adds core capabilities to support the implementation of measurement solutions in multi-layered textile reinforced structures for extremely light, rigid and robust fibre composite components. "The requirements for composite fibre inspection go beyond the classical dimensional metrology which Hexagon's current offering provides," said Hexagon Manufacturing Intelligence President Norbert Hanke. "Apodius' solutions offer a perfect complement to our portfolio, in terms of technology and application expertise." Apodius Co-CEOs, Alexander Leutner and Jonathan Roberz, added, "Providing highly-accurate positioning data, Hexagon's solutions fit perfectly to our sensors. Furthermore, joining Hexagon is a great opportunity to bring our technology to composite production lines all over the world."

GMTA

CELEBRATES 25TH ANNIVERSARY

German Machine Tools of America (GMTA) celebrated the company's 25th anniversary during the IMTS (International Manufacturing & Technology Show) in Chicago.

(left to right) Buchholz, Knoy, Hambleton and Friedrich.

The celebration took place aboard the Anita Dee II, a private yacht charter, on Wednesday evening after the show concluded for the day. A crowd of over 150 customers and business associates boarded the yacht on the east end of McCormick Place for an evening of wining & dining while cruising the waters of Lake Michigan. The evening was hosted by Walter Friedrich, president of GMTA, Scott Knoy, vice president of sales, and Claudia Hambleton, CFO.

Friedrich welcomed the excited crowd with a short speech thanking them for their support over the last 25 years. He gave a bit of company history and mentioned the recent opening of their newest office in Queretaro, Mexico.

Guests enjoyed a lavish dinner buffet, open bar and the music of a live band playing tropical party music while cruising Lake Michigan on the elegant Anita Dee II. The ship cruised along the illuminated Chicago skyline, past Buckingham Fountain and along Navy Pier for several hours before returning guests to Navy Pier.

Maprox

WELCOMES NEW CEO

As a planned successor, **Adrian Zwirner** took over from the long-time owner **Max Maurer** as CEO of Maprox Gmbh effective September 1, 2016.

Zwirner has been working in mechanical and plant engineering for nearly 20 years and has been able to work up a solid track record in international sales and in the management of renowned, globally-active Swiss companies.

The graduate mechanical engineer is therefore extremely well-prepared and is looking forward to his new challenge. "I have considerable interest in managing the company as sustainably and carefully as Max Maurer has done up to now. I'm particularly pleased that we complement each other so well and that Max will remain in the company for awhile to deal increasingly with the subject of innovation. Our mutual level of trust is extremely high," said Zwirner.

Maurer and Zwirner got to know each other at the beginning of the year and have carefully prepared the changeover at the top to guarantee a new start with the least-possible problems and to set up considerable reciprocal trust. With minimal employee fluctuation over the years, the Maprox team has mastered a crisis or two and even increased its strength after the so-called Swiss Franc shock in January 2015.

Maurer will be comprehensively familiarizing the new CEO over the coming period and will continue to drive product development forward during a transition phase within the company.

For example, Maprox has reworked its pneumatic clamping chuck (PM ER32) by popular demand. It is now 25 percent narrower, still only pneumatically-operated (without springs) and at only 5 micron repeat accuracy can also be used with rotation on request. It is particularly suitable for laser processing.

Maprox GmbH produces customer-specific and high-precision clamping devices in the international market. Long-term customers include well-known automobile suppliers, clock manufacturers, instrumentation manufacturers and optic specialists. Maprox is represented in the United States by Rotec Tools, Mahopac, New York,

SMT

ANNOUNCES FORMATION OF SMART MANUFACTURING TECHNOLOGY JAPAN LTD.

SMT is proud to announce the formation of Smart Manufacturing Technology Japan LTD. (SMTJ), a subsidiary of SMT. The new office in Japan is the latest installment of the company's expanding reach worldwide.

A lot of Japan's excellent development and manufacturing companies in aerospace, automotive, bearing, marine, off-highway, rail and wind, were founded by engineers full of monozukuri spirit. That spirit will be the bedrock on which bonds will be built and nurtured between SMT and its customers. SMTJ will support these engineers and the Southeast Asia area from this new Japanese hub.

David Beedan, director at SMT, says "The opening of a new business unit on the Asian continent will substantially help keep close ties and build stronger relationships with our customers across the globe. With growth comes exciting opportunities and I have full confidence that the new and experienced team in Japan will deliver and exceed the SMT values."

"As well as provide support and services to customers in Japan , the team will help open more doors to culturally proximate markets as well as provide greater access to sectors on the Australasian continent," he added.

Kapp Technologies

TAKES MAJORITY OWNERSHIP IN PENTA GEAR METROLOGY

Kapp Technologies announces it has become majority partner in Penta Gear Metrology, LLC.

Penta Gear Metrology LLC builds both functional and analytical gear measuring systems used in most gear manufacturing plants from automotive transmissions to industrial drives. The Dayton, OH company is known for innovative design and modern intuitive software. Penta Gear Metrology (formerly Pentagear Products and PECO) pioneered the gear analyzer technology with advances in software and motion control systems beginning in 1999. Penta Gear Metrology proudly serves Pentagear and PECO customers on four countinents and is highly regarded for support. President Marvin Nicholson has led the company through the transition and remains a partner.

"The partnership with Kapp Technologies expands opportunities for us and provides resources to accelerate our new developments," said Nicholson. "We are excited to introduce the PGM400 at IMTS 2016 It is the larger and faster brother to the smaller ND products."

Penta Gear Metrology also offers a retrofit or "REPOWER" of certain used gear analyzers. The PGM motion system and software is adapted to provide a cost effective solution for gear manufacturers. Penta Gear Metrology LLC is ISO 9001, ISO14001 & QS 9000/TE Registered.

"This was a natural progression for us," says Bill Miller, vice president sales at Kapp Technologies, "With Penta Gear Metrology we now serve our entire market with analytical gear inspection systems. Penta Gear is a perfect fit with the larger systems of R&P Metrology Systems."

Bodycote

ACQUIRES NITREX METAL TECHNOLOGIES AND RELOCATES TEXAS FACILITY

Bodycote has acquired Nitrex Metal Technologies. Nitrex Metal Technologies specializes in precision gas nitriding and ferritic nitrocarburizing in both batch and continuous forms. Continuous gas nitriding and ferritic nitrocarburizing are unique in the industry and are particularly suited to high-volume automotive work. The addition of Nitrex Metal Technologies to the Bodycote Group broadens the range of thermal processing services that Bodycote offers, which already range from conventional atmosphere heat treatments like batch IQ, vacuum, and induction to more exotic specialty technologies like LPC, BoroCote, and Corr-I-Dur.

"Nitrex Metal Technologies is a great addition to the Bodycote Group. Along with the rest of Bodycote's existing service offerings, this acquisition really cements our position as the go-to expert source for all things nitriding," said Dan McCurdy, president of Bodycote Automotive and General Industrial Heat Treating in North America and Asia.

Additionally, Bodycote has announced plans to open a new Dallas/Fort Worth, Texas location in Haltom City. The existing Fort Worth business, on Montgomery Street, will be relocated due to redevelopment of the area. A new heat treatment facility is being constructed in Haltom City and a ground breaking ceremony took place at the new location on August 26, 2016, attended by The Honorable Mayor David Averitt and other Haltom City Council members.

Bodycote is in a strong position to continue supporting its supply chain partners in the Dallas / Fort Worth area, with ample capacity and a diverse set of capabilities to meet their specific needs. The new facility will hold international quality standards including Nadcap, AS9100 and ISO9001, as well as OEM approvals.

October 12–14—Rocky Mountain Gear Finishing School Boulder, Colorado. Rocky Mountain Gear Finishing School covers advancements in profile and generating grinding technology in a classroom setting but also through hands-on demonstrations. Participants learn about the principles and mechanics of different gear finishing processes through classroom presentations, and then see them demonstrated on a Kapp or Niles grinding machine. Additional workshops provide further opportunities to explore, and network with others in the field. Machine demonstrations reinforce the theoretical training and help people retain more of the information taught in the classroom. Workshops offer even further opportunity for review, as well as a platform to explore your own applications with Kapp experts, as well as others in the group. Included, is an in-depth review of non-dressable CBN solutions coupled with a tour through the CBN tool production plant at Kapp Technologies. A presentation on diamond dressing tools complements the tool technology. For more information, visit *www.kapp-usa.com*.

October 19–20—AGMA Fall Marketing & Forecasting Conference 2016 Rosemont, Illinois. So many important factors to watch in the gear industry including rising inventory levels, emerging markets in the late stage of the credit cycle, large impact of trade agreements and the looming election. What will directly impact the gear industry in the next 12 months? Speakers Tom Runiewicz (Global Insight) and John Maketa (KGC Direct) will discuss projections and challenges now and in the future. For more information, visit *www.agma.org*.

October 23–27—Materials, Science and Technology 2016 Salt Lake City, Utah. MS&T16 is the most comprehensive forum for materials science and engineering technologies. Attendees learn from materials specialists, explore diverse materials applications and experience the synergy of this materials community. MS&T crosses the boundaries of most materials events by bringing together a broad range of technical sessions and expertise through the strengths of six major materials organizations: The American Ceramic Society (ACerS), Association for Iron & Steel Technology (AIST), ASM International, Metallurgy and Materials Society of CIM (MetSoc), NACE International, and The Minerals, Metals & Materials Society (TMS). Topics include additive manufacturing, ceramic composites, failure analysis, light metal technology, next generation biomaterials, surface protection, high performance metals and more. For additional information, visit *www.matscitech.org*.

November 2–3—Advanced Engineering 2016 Birmingham, England. Bringing together OEMs and Tier 1 manufacturers to meet and do business with all levels of the engineering supply chain, Advanced Engineering addresses the supply chain needs of the aerospace, automotive, motorsport, transportation, civil engineering industries and more. In addition to its show floor containing 700+ exhibiting suppliers, partners and industry bodies, Advanced Engineering hosts an open conference providing expert industry intel, latest technology and innovations case studies and supply chain opportunities delivered by OEM program managers and industry experts. Co-located events include Aero Engineering, Composites Engineering, Automotive Engineering and Performance Metals Engineering. For more information, visit *www.easyfairs.com*.

November 15–17—Detailed Gear Design-Beyond Simple Service Factors Las Vegas, Nevada. This course explores all factors that go into good gear design from life cycle, load, torque, tooth optimization, and evaluating consequences. Students should have a good understanding of basic gear theory and nomenclature. Interact with a group of your peers and with a talented and well-respected instructor who will push your thinking beyond its normal boundaries. Gear engineers, gear designers, application engineers, people who are responsible for interpreting gear designs, technicians and managers that want to better understand all aspects of gear design should attend. Raymond Drago is the course instructor. For more information, visit *www.agma.org*.

November 16–18—Fabtech 2016 Las Vegas, Nevada. More than 28,000 attendees and over 1,300 exhibiting companies are expected to gather once again to celebrate metal manufacturing. The event also provides educational sessions and expert-led presentations covering the latest trends and technology in the metal forming, fabricating, welding and finishing industries. The show features 100+ technical, operational, economic and managerial sessions allowing attendees to exchange best practices and explore new product advancements. A variety of technology pavilions feature technologies like arc welding, cutting, hydroforming, lasers, lubrication, robotics, roll forming, safety, tooling, waterjet and more. Fabtech 2016 is co-sponsored by the Fabricators & Manufacturers Association International, SME, Precision Metalforming Association, Chemical Coaters Association International and the American Welding Society. For more information, visit *www.fabtechexpo.com*.

November 29–December 1—Composites Europe 2016 Dusseldorf, Germany. Experience the entire process chain of the composites industry and gain a complete overview of the market and the major exhibitors—in just three days. As the international industry meeting point in Europe's biggest composites market, Composites Europe combines tried and tested solutions and efficient innovations. The trade fair reflects the variety of goods and services and the innovative strength of the entire industry. Major topics at the fair are state-of-the-art production and processing technologies focusing on concepts for lightweight construction and automotive applications. Industries involved include automotive, aerospace, energy, marine, electronics, medical, wastewater, agriculture and general engineering. For more information, visit *www.composites-europe.com*.

December 5–8—CTI Symposium Berlin 2016. Berlin, Germany. More than ever, the development of transmissions and drivetrains is being shaped by the trends of electrification, connectivity and automation. The focus rests on significantly increasing driving efficiency and comfort while reducing emissions, and on providing a brand-appropriate driving experience – as well as on 'ongoing demands' in terms of costs, package, weight and modularity. The planned introduction of regulations to measure Real Driving Emissions (RDE) in 2017 will heighten the technical challenges these criteria involve. The only way to meet the ambitious targets is by intelligent, effective transmission and powertrain integration. These and other topics will be front and center during the CTI Symposium Berlin 2016. For more information, visit *www.transmission-symposium.com*.

gear TECHNOLOGY

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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)

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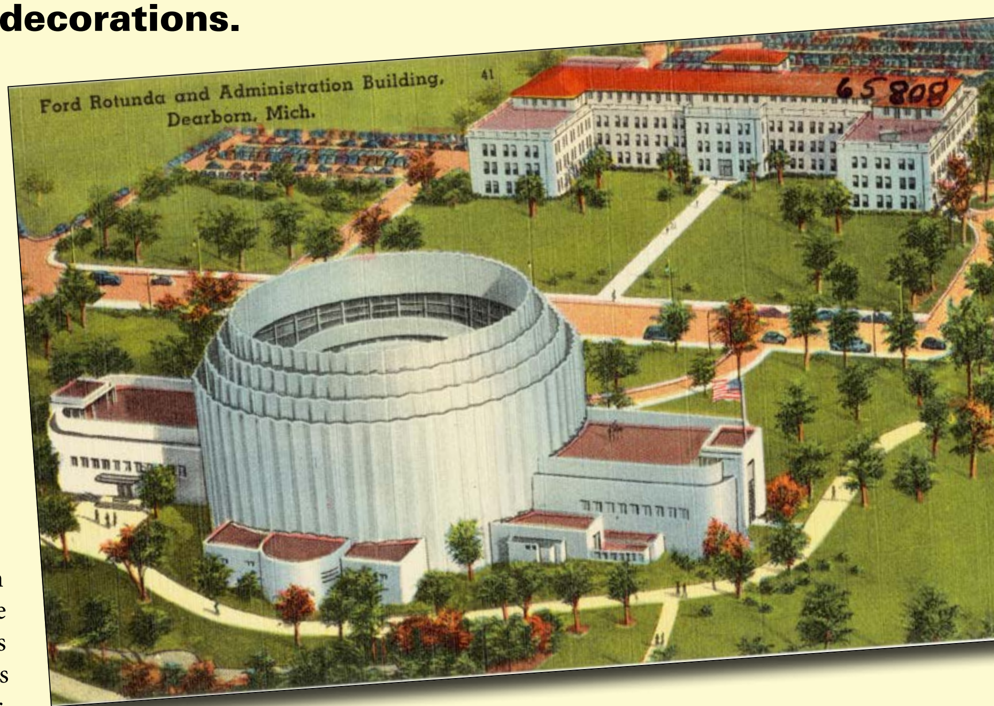
Motown's Most Famous Gear

The Ford Rotunda used to be one of the most popular tourist attractions in the United States, but its main draw wasn't its cars, but its Christmas decorations.

Alex Cannella, News Editor



It's hard for me to think of a massive Christmas exhibit as being the fifth largest tourist attraction in the entire country. I mean, sure, it's still a tradition to show up at the local Macy's to check out the Christmas decorations, but for my generation, the idea that a Christmas exhibit could draw out 1.5 million visitors, more tourists than either Yellowstone Park or the Statue of Liberty received, is stunning. But at the height of its popularity, that's exactly what the Ford Rotunda was.

The Rotunda was, of course, more than just a Christmas display. It also featured the Ford Motor Company's latest car models alongside other industrial exhibits and was regularly used to host events and for advertising purposes. But the Ford Rotunda's main draw would eventually become its Christmas Fantasy exhibit, a massive, yearly display that featured thousands of individual pieces from dolls to animatronic scenes.

The building itself was also a unique sight. It was designed to look like a giant set of gears.

Initially constructed in 1934 for the Century of Progress World's Fair in Chicago, the Rotunda didn't follow any architectural style of the time. Instead, it was built to resemble a graduated cluster of internally-meshed gears. The "gears" were 12 stories tall and surrounded a 92-foot-wide courtyard. For a temporary structure, it was mammoth. The building was built by Albert Kahn, the architect behind two of Ford's factories amongst other buildings in the Detroit region.

The Rotunda became one of the centerpiece exhibits at Chicago's World's Fair until it ended a few months later, but Ford wasn't done with its now famous gear-shaped building yet. Instead, they decided to build a new, permanent version of the building near their Rouge Plant in Dearborn, Michigan. In its new location, the Ford Rotunda became a showcase of the company's latest car models alongside other displays. The Rotunda eventually underwent renovations in 1952, during which one of the very first geodesic domes in the world was installed over the courtyard. The building wasn't designed to support the weight of a conventional roof, and so Buckminster Fuller's dome design was utilized.

A year later, the Ford Rotunda underwent a grand reopening just in time to celebrate Ford's 50th anniversary. Over the next nine years, the yearly Christmas Fantasy events started kicking off and the building reached the peak of its renown. For almost a decade afterwards, the Rotunda was not only a well-known landmark, but also a continued platform to leverage Ford's latest car lines.

Less than a decade later, however, the Ford Rotunda caught fire while preparing for the 1962 Christmas Fantasy. The fire started while workers were applying weatherproofing sealant to the dome, some of the sealant's vapors catching fire. An hour later, the building had burned to the ground, though there were no casualties. The only part of the Christmas Fantasy display that had survived was the 15,000 piece miniature animated circus, which had only evaded the fire by virtue of still being in storage.

And just like that, the Ford Rotunda vanished almost as quickly as it had appeared. Ford opted not to rebuild the Rotunda, as its reconstruction was estimated to cost a staggering $15 million.

Though best remembered for its Christmas displays, the Ford Rotunda was also one of the most well-known symbols of the automotive industry. Its gear-inspired architecture and industrial displays wowed crowds since its inception at the 1934 World's Fair. The building was at the center of some of Ford's marketing efforts and was often used to photograph and introduce car models, including the ill-fated Ford Edsel.

During the short time it was in operation, the Ford Rotunda saw over 18 million visitors, and no doubt many of the people in our industry were among them. All that's left of the Rotunda is a street that carries its name, but the fact remains that it was an iconic cornerstone of industrial America's history, our history, and as those who were around to actually visit the Ford Rotunda move on and retire, it should not be forgotten.