New gear skiving machine LK 300-500
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In the LK 300 and 500 gear skiving machines, process, tools and machine including tool changer and automation system come from a single source because in skiving³ the delivery of an integrated solution for the customer is of primary interest. Skiving³ is especially suited for internal gears of medium size and quantity, as it is much faster than shaping and more economical than broaching. The machine can be operated using the touch-based LHGe@rTec control system.
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ROOT CAUSE FOR NOISE EXCITATION IN GEOMETRICALLY-ACCURATE (PERFECT) PARALLEL-AXES GEARING

Theoretically, geometrically-accurate gearing that features zero base pitch variation promise to be noiseless, as no noise is produced by the gear mesh. In reality they can be noisy.

By STEPHEN P. RADZEVICH

USING SIMULATION TO EVALUATE PART GREEN SHAPE TO REDUCE DISTORTION DURING PLUG QUENCHING

This case study follows an example of applying the commercially available heat-treatment simulation software, DANTE, to evaluate a real-world challenge.

By JUSTIN SIMS, CHARLIE LI, and B. LYNN FERGUSON

FILLING SOME GAPS IN SPLINE DESIGN GUIDELINES

This paper provides an accurate method for calculating radial loads transmitted by straight-sided splines by means of the effective pressure angle calculation, enabling more accurate hoop-stress calculation for these splines.

By STEPHEN McKENNY

COMPONENT DESIGN FOR FUNCTIONAL AND COMMERCIAL OPTIMIZATION

COMPANY PROFILE With advanced engineering at the core of its forging and gear divisions, Sona BLW is constantly tackling demands while developing and implementing solutions for its customers.

By KENNETH CARTER
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WILLIAM P. NEBIOLO
TIPS TO INCREASE VIBRATORY PERFORMANCE DURING GEAR POLISHING
An attentive eye and some simple fixes will keep things going when there are issues during the process.

Brian Dengel
Gear Mesh and Tooth Modifications
The involute gear form allows for a fluid mesh of the teeth as they slide along one another.

D. Scott Mackenzie
Quench System Monitoring Requirements in AMS 2759G
Determining the minimum hardness required by AMS 2759G requires attention to the specific hardenability curve of the specimen material and proper equipment maintenance.

McKernin named sales, marketing director at United Stars.

Klingelnberg to present non-gear solutions at MPT Expo.
MPT Expo: New name, new opportunities, new horizons

It may have a new name, but it’s time for the gear industry’s biggest trade show — Motion+Power Technology Expo 2019 in Detroit, Michigan.

You saw our pre-show issue in September, and our October issue is still in major show mode.

There’s more to this year’s show than just a name change. For the first time, the show will feature exhibitors from the fluid-power industries in addition to the mechanical-, gear-, and electric-power industries.

By adding this eclectic mix of businesses, the MPT Expo is geared to guarantee attendees will gain the best practices and new ideas from like-minded colleagues as well as cross-sector collaboration. End-users can shop the latest technology, gear products, and services, and compare benefits side-by-side. Prominent exhibitors will conduct demos and host information-rich seminars as well as offer up technical expertise.

And odds are, that if you’re reading this, you might be walking MPT’s massive show floor already during the three-day event.

Since this is a show issue, Gear Solutions is offering up a lot of interesting information to help build that show enthusiasm. To help you out as you make your way around the show floor, our Industry News section and Q&A feature highlight several companies who’ll be showcasing some of their products. You’ll find summaries of what to expect at their booths as well as their booth numbers for easy access.

But there’s even more inside to keep those brain gears turning. Frequent contributor Stephen Radzevich shares his insights on the root cause for noise excitation in geometrically-accurate (perfect) parallel-axes gearing.

Justin Sims, Charlie Li, and B. Lynn Ferguson with DANTE Solutions Inc. co-author an article on using simulation to evaluate heat-treatment geometry to reduce distortion during plug quenching.

And GM’s Stephen McKenny has written a fascinating article on an accurate method for calculating radial loads transmitted by straight-sided splines by means of the effective pressure angle calculation, enabling more accurate hoop-stress calculation for the splines.

Our monthly columnists are also offering up a range of interesting subjects from the quenchability and system monitoring requirements in AMS 2759G to troubleshooting tips to increase vibratory performance during gear polishing to gear mesh and tooth modifications.

I hope you enjoy all the new opportunities the MPT Expo has to offer. And, please stop by Gear Solutions’ booth (#4036) and say hey. I’d love to discuss editorial opportunities with you, so I can feature you in a future issue.

As always, thanks for reading, and I’ll see you in Detroit.

KENNETH CARTER, editor
editor@gearsolutions.com
(800) 366-2185 x204

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WE OWN WHAT WE SELL, AND WE KNOW WHAT WE’RE SELLING!
McKernin named sales, marketing director at United Stars

United Stars, Inc. appointed Michael McKernin as the Gear Group Sales and Marketing Director during the summer. With 35 years in the gear industry, McKernin brings experience in all aspects of gear engineering, manufacturing, and sales. He will lead the value-added strategy to continue customer relations, offer solutions and products along with broadening the end-user markets and industrial applications.

McKernin has a degree in manufacturing engineering technology from Bradley University. He has been involved in all aspects of gear manufacturing, from machine tool operation to corporate management — including being the president at his previous company. McKernin serves as the chairman of the Business Management Executive Committee (BMEC) for the American Gear Manufacturers Association (AGMA) where he oversees the managerial and educational programs along with committee activity for AGMA members in manufacturing.

“I look forward to working with the great people at United Stars,” McKernin said. “The company vision and quality products, coupled with my experience in the industry, will make a great team.”

“Having Mike join our team is going to push our products even further,” said Richard Van Lanen, President of United Starts, Inc. “His passion for the industry and unparalleled customer service skills will help to grow our brand and the customer base we currently have.”

MORE INFO www.ustars.com

Klingelnberg to present non-gear solutions at MPT Expo

Machine manufacturer Klingelnberg will be using the U.S.-based Motion + Power Technology Expo (formerly known as Gear Expo) platform for an intensive professional exchange of equals.

The trade show will be October 15-17 in Detroit. Not only will visitors to Booth #3631 encounter a team of experts who are available for technical discussions, but Klingelnberg will also unveil leading-edge technology with the P 26 precision measuring center and optical measuring technology. At the same time, Klingelnberg will shine a light on its expertise in the “non-gear” sector in this forum.

To be equipped for all measurement tasks, Klingelnberg combines HISPEED OPTOSCAN optical measuring technology with the established, high-precision 3D NANOSCAN probing system. This system incorporates maximum precision with the required robustness for a production environment. Thanks to the high-speed changeover feature, the optimal measurement system can be used for every task. Optical measuring technology is a valuable addition for many tasks in precision measuring technology, such as digitizing an entire, possibly unknown, component or measuring splines quickly yet completely. The high measurement speed of the optical system with a high point density allows for measurements that would be time-intensive if performed with a tactile system. This enables cost-effective measurements in an industrial setting. The tactile 3D NANOSCAND is a must any time individual, high-precision measured values are required for measurement tasks. For a rapidly generated, dense point cloud, HISPEED OPTOSCAN is available. Combined with the available roughness measuring technology, Klingelnberg preci-
G variant precision measuring centers: Complete measurement of complex components

Stringent precision requirements in series production and increasing component complexity — both call for the best available measuring technology. The G series precision measuring centers are perfectly suited to production processes that require not just dimensional measurement tasks but also complex form and surface measurements in large numbers. Klingelnberg G variant precision measuring centers are designed for use in the production environment and are therefore ideal for measurements on the shop floor.

This makes the G variant appealing for the automotive and commercial vehicle industry, for machine building and plant engineering, and for manufacturers of rolling bearings. Like all precision measuring centers from Klingelnberg, the G variants stand out for their high-precision 3D NANOSCAN probing system as well as their easy-to-use roughness probe systems for external and internal measurements.

Furthermore, visitors can also look forward to additional highlights from Klingelnberg. The expo team will also be informing visitors to the stand about the exhibit highlights presented at EMO Hannover 2019. Among others, these include the OERLIKON G 35 bevel gear grinding machine for the aviation industry and the HÖFLER VIPER 500 MFM cycloid grinding machine, which is of special interest to the robot industry.

G 35: Optimal for requirements in the aviation industry

Specifically tailored to the requirements of the aviation industry, the newly developed OERLIKON G 35 bevel gear grinding machine takes aviation gear manufacturing to a new level with its two vertical grinding spindles. Tried and tested concepts and components are used to ensure optimal functionality and the best possible availability of spare parts for this special-purpose machine. The technology has been enhanced to include new features that allow for efficient work practices on the machine.

VIPER 500 MFM: Increasing productivity and precision in the robot industry

With the VIPER 500 MFM, Klingelnberg is strengthening its presence in the market outside the gearing sector. This machine makes it possible to mass-produce high-precision cycloid gearings without requiring complex pairing of components. In combination with the precision measuring centers and Closed Loop, Klingelnberg now has a complete system that makes the production of highly accurate cycloid gearings very easy: in the cycloid grinding cell, the processing machine and the precision measuring center are connected by automation. Thanks to the use of GearEngine®, this cycloid grinding cell is “fit” for Industry 4.0 processes. Combined with Closed Loop, this gives rise to an autonomous, self-optimizing production system that makes it possible to use the machining and measuring capacity of the machines to optimal effect.

Potential customers at MPT are invited to experience all of this for themselves — the Klingelnberg specialists will be on hand at stand 3631 to assist all visitors with their expert knowledge.

MORE INFO www.klingelnberg.com/en

Gear Motions to introduce non-backdriveable gearbox technology

Gear Motions, a leading precision gear manufacturer, will exhibit at the Motion + Power Technology Expo (MPT Expo) Oct. 15-17, 2019. During the event, the company will introduce new, non-backdriveable gearbox technology. Attendees can view demonstrations of the never-before-seen, patented technology at the Gear Motions’ Booth #3531.

At MPT Expo, Gear Motions will also unveil its plan for a substantial new equipment investment. The precision manufacturer plans to add three new Reishauer robotic gear grinders to its facilities in Syracuse, New York, and Buffalo, New York, over the next two years. The first, a Reishauer RZ 260, is expected to be installed by the end of 2019. Already known as gear-grinding specialists, the new machines will further expand the company’s gear grinding capacity and add new capabilities such as twist control and super finishing.

Gear Motions’ top sales and engineering staff will be on-site and available to discuss customers’ precision manufacturing needs and project specifications.

MORE INFO www.gearmotions.com
Liebherr offers economic alternatives for grinding processes

For many applications, generating grinding with corundum is a good solution, but this abrasive also has certain disadvantages for some applications. Grinding special geometrical modifications could, for example, have a negative effect: The modifications would then have to be integrated in the tool completely or partially via the dressing process in the machine. On one hand, however, profiling a grinding worm needs time and, on the other, it alters the geometry of the tool. Depending on the choice of grinding process, it has to be repeated in very short intervals in order to guarantee the production at the same level of quality.

This is just where Liebherr comes in, offering a CBN tool with an implemented modification and demonstrating on an example workpiece that extremely economic production is possible with these tools.

CBN stands for cubic boron nitride. Next to diamond, it is the second-hardest cutting material in the world. It consists of a three-dimensional matrix made of boron and nitrogen atoms that can develop a broader spectrum of crystal forms than diamond. It has high thermal conductivity and a low coefficient of friction. In this way, the workpiece heats up much less than grinding with corundum, for example. It is possible to machine very hard materials reliably with CBN. CBN grinding worms can also be
smaller, which means that the range of applications is greater than with corundum.

CBN tools are experiencing a comeback. They may be expensive to procure but make gains with the unit costs on modifications. Dr. Andreas Mehr, the leading grinding technology expert at Liebherr-Verzahntechnik GmbH, explains the differences of the grinding materials:

“We have been using galvanically coated CBN since 1988. It is a highly durable grinding material. A significant increase in the grinding performance of modern corundum has been achieved in recent years, but, compared with CBN, it comes with the disadvantage of the amount of effort required for dressing, which is encountered most prominently on topological grinding processes.”

With these processes, the number of workpieces per dressing cycle is significantly reduced due to the limited shift possibility, which in turn raises tool costs and also cycle times. It may be possible on corundum tools (e.g. for distortion-free generating grinding) to increase the workpiece number for each dressing interval through new mathematical solutions but this also applies in the same way to the use of CBN tools.

With CBN, the dressing times can be dispensed with completely, which means that cycle times and thereby manufacturing costs can be reduced. CBN is highly machineable and generates an extremely low measuring complexity. A CBN grinding worm is clamped in and the grinding process begins straightaway — with there being no need at all to make corrections beforehand.

The unit costs of a test workpiece (m=1.53 mm, z=81) included a special width modification with corundum at 4.25 euros with a cycle time of 114.6 seconds, while the same grinding process with CBN was 3.38 euros cheaper and considerably faster with a cycle time of 78 seconds.

Where the number of producible workpieces per dressing cycle with corundum is in the two-figure range, it can occasionally reach well into the four-figure range with a CBN coating.

“Each situation needs to be assessed individually to determine which grinding worm is the most viable. We are happy to advise our customers on whether CBN is the better alternative for their numbers and application scenarios,” Mehr said.

In a dressing-free CBN process, all parameters are predefined and “frozen.” This is a crucial difference between corundum and CBN processes. All corundum processes are subject to changes through dressing, which can impair the grinding worm quality. Examples are wear of the dressing tool or diminishing worm diameter — as the worm diameter diminishes, the length of the active worm spirals are shorter, which reduces the number of active abrasive grains.

A consequence of this is an increase in the roughness factor on the tooth flank, which should remain constant throughout the worm tool life. “There is a limit to which this can be counteracted through finer dressing processes,” Mehr said.

Additional processes, such as dressing, present fault sources that simply cannot occur with CBN. CBN processes are extremely robust and quality assured, which makes them particularly interesting for the economic production of high-quality gear teeth, such as in the area of electric mobility. “Attempts are being made to reduce noise, particularly on the very sophisticated electric gears in the automotive sector, by changing the macro and micro geometry,” Mehr said.

Liebherr manufactures CBN tools at the
“Our aims in production are high performance and top quality in a very sturdy process,” said Haider Arroum, regional sales manager for gear-cutting tools. The production of CBN grinding worms and discs is therefore carried out in a closed loop process in which measurement results flow immediately back into the production parameters as corrections.

“We have to carry over the configuration accuracy to manufacture as close to 1:1 as possible,” Arroum said.

In most cases, the new generation of electric cars is built in new factories where the focus is on reliable and clean processes. At EMO 2019, Liebherr exhibited a generating gear-grinding machine that meets the requirements of a clean factory in full: the LGG 180 with integrated centrifuge. The centrifuge station for the removal of burrs and coolants is at the pocket of the ring loader pointing toward the operator side. It is mounted decoupled from the machine so that oscillations or vibrations from the centrifugation process do not have any effect on the gear quality. Spinning during the machining process is therefore possible. There is no loss of oil; the media remain in the machine, and the cleaned components can be transported further in any automation system.

Advantages of integrated centrifuge unit:
- Clean factory, thanks to integrated centrifuge: dry workpieces, no oil loss, reliable production.
- Decoupled centrifuge unit with no transfer of vibrations to grinding machine.
- Can be connected to any conventional automation solution.

Advantages of CBN tools:
- Tool mounting, pre-profiling, and dressing are dispensed with.
- No adjustment of profile angle necessary.
- Easy operation.
- Significantly reduced measuring and testing effort.

Luren Precision Co.
changes name to Matrix
as part of rebranding

In order to advance the development of the company and improve brand awareness, Luren Precision Co. has changed the company name to Matrix Precision Co., Ltd. The change took place in June.

Company officials see this as an opportunity to serve its customers better, while appreciating the support they have gotten as Luren Precision. The address, phone numbers, and website will not change.

Big Kaiser moves production of CAT50 Smart Damper

In order to offer more flexibility to customers, Big Kaiser, a global leader in high-performance tooling and systems for metalworking industries, has moved production of its CAT50 milling adapters to the U.S. The product performance and design will not change, but the move allows for more variation and faster delivery of special lengths.
for specific applications.

“The market has spoken,” said Alan Miller, engineering and product manager. “As awareness of Smart Damper has spread, more and more customers have asked for specials for their unique projects. This allows us to do more of that without the delay of coordinating with our partners overseas and additional shipping time.”

With both a patented counter weight and friction damper, the Smart Damper is a dynamic damping system that eliminates vibration for higher productivity. It provides quiet, vibration-free milling with long projection tools, ultimately making it easier to achieve fine surface finishes and higher metal removal rates.

Smart Damper adapters are also available for turning and boring applications.

MORE INFO www.bigkaiser.com
During each rotation, presenters gave an overview of their profession, answered questions, and led students in hands-on activities. Approximately 160 seniors attended the morning session, with 162 juniors participating in the afternoon. They were divided into six groups and rotated among six “cluster areas.” These areas included agriculture, food, and natural resources; arts, communications, and information systems; engineering, manufacturing, and technology; health and life sciences; human services; business, management, and administration. Students were able to see the list of employers in each cluster and chose two from each cluster.

Exact Metrology attended this career expo to expand its relationship with schools and businesses in the area, recruit potential employees, and educate students on Exact Metrology’s technology offerings. The latter was achieved using Artec 3D, a world-renowned developer and manufacturer of professional 3D scanners and software. The Artec Eva™ is a handheld scanner ideal for quick, textured, and accurate scans. Artec Eva doesn’t require markers or calibration, and the scanner captures 16 frames per second. These frames are automatically aligned in real time, making scanning easy and fast.

“We are focusing on giving back to the local communities by partnering with more schools. These students will be tomorrow’s workforce, and we want to make sure they know who we are and what we do,” said Dana Green, the inside sales person at Exact Metrology.

Exact Metrology is an ISO 9001:2008, AS9100, FFL and ITAR Certified Company.

Heller Machine Tools partners with Ellison Technologies

Heller Machine Tools is changing its business model in the North American market. For the first time, Heller has moved to both direct and distribution sales and service — providing customers with a more local interface that can provide sales, engineering, and customer support via a new distribution network and still maintain Heller’s
existing direct key account business.

Heller Machine Tools L.P. is pleased to announce a new distribution partnership with Ellison’s Advanced Technologies Business Unit in the U.S., with specific representation in the following states: Alabama, Delaware, Georgia, Illinois, Maryland, Mississippi, Nebraska, North Carolina, South Carolina, Tennessee, and Virginia.

The Ellison Company was founded by James O. Ellison in 1955. The company was based in Southern California to distribute machines tools and provide after-sales service. In 1965, W.J. (Jim) Ellison joined his father's company and started to grow and expand the family business, representing several machine tool builders and expanding coverage into other states.

Over the following years, Ellison developed a full solution-based business including applications and process engineering capability. This added value for customers by supplying machine tools and also demonstrating the most efficient methods of producing components including full turnkey solutions.

In 2004, Ellison started to expand further into new U.S. locations, which included the purchase of an automation company and other machine tool distribution companies. In 2006 the official company name was changed to Ellison Technologies to reflect more accurately the company’s capabilities.

After continued expansion and organic growth, in 2012 Ellison reached a half-billion USD in sales. The company celebrated its 60th anniversary in 2015.

Today, Ellison Technologies is part of Mitsui & Co., Ltd. of Japan, which has allowed the company to increase its business footprint, as a provider of advanced machining solutions to the North American metal cutting manufacturers and their global affiliates.

Ellison Technologies priority is to introduce technologies that can strengthen the customer’s ability to compete in the markets they serve. Whether the solution involves a single stand-alone machine, multi-process equipment, or a fully integrated manufacturing system with automation, the goal remains the same: to optimize production throughput and component quality at the lowest per-part manufacturing cost.

“Ellison Technologies can provide our customer base with a high level of engineering and sales support; they provide excellent geographical coverage with their network of offices and technical centers. This all means a better, more focused customer support experience for both existing and new customers,” said Steve Pegram, director, Heller Machine Tools.

With new distribution partnerships now in place, Heller will be introduced to new markets and business sectors, offering high-quality performance machine tools. In addition to Ellison Technologies, current distribution partners include Ellison Machinery, Maruka, and Compumachine, offering complete coverage in North America.

MORE INFO  www.heller-us.com

Gary Brown celebrates 30 years at helm of Heule Tool Corporation

Gary Brown, president and general manager of Heule Tool Corporation in Loveland, Ohio, recently celebrated his 30th anniversary with the company. Brown began working for Heule in August 1989, shortly after earning his engineering degree from Miami University in Oxford, Ohio. At that time,
Heule Werkzeug AG had just established its first subsidiary in North America, which was near Detroit, Michigan (until 1994, when it was relocated to the Cincinnati area). The subsidiary was established in order to better serve customers with local technical support, as well as to grow sales of its automated deburring tools to manufacturers in the United States, Canada, and Mexico.

Brown spent a year in Switzerland learning about the business before he returned to the U.S. to begin growing sales to North American manufacturers. Modeling Heule’s business approach of slow growth “one customer at a time,” Brown was able to quickly get a foothold in the U.S. market with Heule’s high-quality Swiss-made precision cutting tools, uniquely designed to machine both the front and back of a through-hole in one pass.

Because of the ability to streamline processes for manufacturers, reduce cycle time, and produce higher quality parts off the machine, Heule tools soon became the preferred deburring/chamfering tool for high-volume automotive manufacturers in the U.S. By the mid-‘90s, Brown had also successfully established Heule in the aerospace manufacturing market.

Under his leadership for the last 30 years, Heule Tool Corporation has grown from Brown as the only full-time employee to a sales organization with 11 full-time employees and 16 independent sales agencies representing the Heule line of tools in the U.S., Canada, and Mexico. Heule tools are found on the production lines of major manufacturers across all industries including heavy construction equipment, oil and gas, energy, military, and defense, and medical.

Speaking on behalf of the Heule family as well as Heule Werkzeug AG, Ulf Heule congratulated Brown on the anniversary, expressing gratitude for his commitment to their family-run business. “Not only has Gary succeeded in making our products known in the manufacturing industry in North America, but as the head of Heule Tool Corporation he has also succeeded in contributing a large proportion of overall company sales. Aside from this success, what is also valued and appreciated is the long-term personal relationship we have built. This cooperation between Heule Switzerland and HTC North America is one of the main pillars on which a prosperous business relationship has developed and will continue for years to come,” he said.

MORE INFO www.heuletool.com

Mazak to showcase latest in advanced manufacturing at SEMA

Mazak Corporation will feature the latest of its advanced machine tool technology at the Specialty Equipment Market Association (SEMA) show in Las Vegas November 5-8. Visitors to Booth 10521 will be able to experience how Mazak’s simple but innovative machine tool technology, namely the VC-300A/5X and the QUICK TURN 200MSY, can transform part-production operations and expand manufacturers’ capabilities.

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compact package, the VC-300A/5X Vertical Machining Center features a wide range of solutions for high-value small-parts production across all industrial sectors. Its 5-axis rotary/tilt trunnion table uses roller gear cam technology for the utmost in precision and positioning speed, and the rigid, powerful spindle comes in various maximum speed configurations to meet manufacturers’ specific application needs.

The QUICK TURN 200MSY Multi-Tasking Turning Center brings together advanced technology, productivity, and value to deliver exceptional performance for shops large and small. It features the MSY configuration to offer milling capabilities, Y-axis functionality, and a second turning spindle for true DONE IN ONE® processing. And with the MAZATROL SmoothG CNC, operators can use both conversational and EIA/ISO programming, as well as a full suite of diagnostic and monitoring functions.

MORE INFO  www.mazakusa.com

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- Gear Rack & Pinion
- Worms
- Wormshafts
- Spline Shafts
- Serrated Shafts

United Tool Supply, which started as an industrial supply business, transformed seemingly overnight in the mid-1980s to a manufacturing company with the birth of the Unite-A-Matic™ PD inspection gauge, a turn-of-the-decade concept at the time. The original Unite-A-Matic OD DOB gauge evolved into an ID model and a bench top model for OD gear inspection. These three models sustained the business from the mid-'80s through the first decade and a half of the 2000s.

Going back to the 2015 Gear Expo at Cobo Hall, the company’s first trade show in many, many years, it was a yearlong blitz to prepare for the show. The company designed a brand-new booth, built inventory to demonstrate products, had new marketing collateral to design, and they had to show how they had innovated. This is only a peek into the internal workings of what needed to be done as employees worked to transform the company after its founder and president, Rusty
Young, passed away in 2014, having been the face of the company for 41 years. At times, the task list seemed monumental. United Tool Supply was a fledgling company down to four employees. It had lost touch with many engineers and key contacts as restructuring took place over the years. There was a shrinking customer base and the business wasn’t selling the amount of new product that sustained the company for decades. Day in and day out, employees showed up and worked hard.

At its height, United Tool Supply was selling 100-plus new gauges a year and boasted dozens of employees. It was seemingly unfathomable to imagine how a company went from sustaining a livelihood for so many families turn into a shell of what it once was. Our conference room had the same blood red carpet from the ‘70s when the facility was built. The conference room turned into an oversized storage closet. The manufacturing shop was cluttered, full of antiquated machines and memories of what use to be — the memories where you can recall the smell of the air, the energy in the shop, the music you remember listening to some days. These are memories that left people inspired, memories unknowingly recreated.

United Tool Supplies took on the painstaking process of a cosmetic overhaul. The blood red carpet was replaced with a neutral blue. Down came the storage shelves, which archived the company’s history for decades. Out went the old drafting table, replaced with a new computer and software. Down came the wallpaper that lined the office walls. Out went the old, outdated, worn office furniture. And with this, out went the company’s old ways. This was a difficult time. A company so proud of its roots, at times, too proud to make a change. As the saying goes, “if it isn’t broke, don’t fix it.” Truth is, the business model was broken, and it was left for the employees to fix.

Next came the manufacturing facility, a facility that was never supposed to be. An oversized basement designed to be the storage area for the company’s supply business in the late ’70s turned into a manufacturing mecca for the Unite-A-Matic™ DOB gauge. Mills and lathes lined walls that were manually operated by what’s now hard-to-find talent. Outdated inventory management system cluttered working areas.

The question was where to start? Organization was the answer. Workers began to sort and separate important from not, good vs. bad inventory, then reorganizing in ways that made sense. Machine component inventory was labeled and stored accordingly, a workflow process was created.
ated, and efficiencies were gained. In came new CNC equipment, out went multiple machines with each replacement. Not only was new equipment brought in, but there was the ability to keep work in-house that was contracted out in the past. What took weeks now takes days. Now that there were newly created efficiencies when it came to manufacturing our Unite-A-Matic DOB gauge, next came the innovation.

Staying in its wheelhouse, United Tool Supplies developed a bench top gauge for 20mm to 75mm internal spline inspection, the Unite-A-Matic™ Model 2020. This was a natural fit as it fell within the Unite-A-Matic™ product line. With a new mindset to “own the gauge bench,” the company set off to develop a product line for a shop hardened surface finish gauge and a shop hardened PD runout gauge. This led to the birth of the SurfA-Matic™ surface finish gauge and Roll-A-Matic™ PD runout gauge.

The company is focused on what it needs to deliver to customers. Product innovation has been key for over the last five years. While the Unite-A-Matic™ PD gauge built the business, it is now time for other products that will grow the business. In order to continue success moving into 2020 and beyond, United Tool Supplies needs to continue to listen to its customers and cater to their needs. Without the influence of our customers, the birth of additional product lines never would have happened. Without aiming to pick up that information to understand how the influence of modern manufacturing relates to the advancements needed to meet these metrics, the company wouldn’t be where is today. Moving into the new decade, United has its sights set on owning the gauge bench in each and every facility visited.

Importantly, it is creating strategic partnerships to support its customers in any way, including prototype gears and technical staffing. United Tool Supply is seemingly a company that was never meant to be, building a product that had never existed previously, in an environment originally meant to store inventory. It is delivering superior products to OEMs and top-tier suppliers worldwide all from a Cincinnati, Ohio, headquarters with a growing staff of seven employees truly delivering a world-class product with the attention to detail and service of a family-owned business. The offering is a quality product, from a quality staff.

MORE INFO  www.united-tool.com

Richard Geiss GmbH puts focus on solvent monitoring and care

No fewer than three new products will be presented by Richard Geiss GmbH, Offingen (Bavaria), at the parts2clean 2019 trade fair.
October 22-24 in Stuttgart: the newly developed Cleanstab S stabilizer, the GEISS Digital Indexer, and the RG PROTECT 160 corrosion protection concentrate. In addition to its latest innovations, the solvent specialist will also be showcasing established products for solvent monitoring as well as its high-purity solvents and contract degreasing services at the leading international trade fair for parts and surface cleaning (hall 7, stand C52).

“I’m absolutely delighted that we will be presenting as many as three new products at this year’s parts2clean”, said Bastian Geiss, CEO of Richard Geiss GmbH. “Industrial parts and surface cleaning face many challenges as almost all sectors require ever-higher levels of cleanliness for components. To meet this demand, we offer not only high-end solvents, but also full-package solutions for a safe and stable cleaning process.”

The absolute highlight of the trade fair is the newly developed Cleanstab S stabilizer for modified alcohols. With this product, Richard Geiss GmbH offers a sump stabilizer that stops acid formation in the cleaning system and reduces any acid that has already formed. This prevents acidification or even self-decomposition of the solvent used and significantly increases the durability of modified alcohols. The Cleanstab S thereby prevents corrosive damage and surface oxidation. The requirements are analyzed and determined individually for every system and every customer in the Geiss laboratory at the company’s headquarters in Offingen.

Another new product featured at the parts2clean is the RG PROTECT 160 corrosion protection concentrate, which can be used in perchloroethylene, hydrocarbon solvents, and in modified alcohols. “We have developed the RG PROTECT 160 specifically for companies in industrial parts and surface cleaning that need to conserve very large numbers of parts and require extended corrosion protection,” said Dieter Ortner, sales manager surface cleaning at Richard Geiss GmbH. “RG PROTECT 160 has an even higher concentration than our established RG PROTECT 180, enabling companies to save not only corrosion protection concentrate, but ultimately also money.”

The third new product, the GEISS Digital Indexer for modified alcohols and hydrocarbons, rounds off the corrosion protection package. It determines the concentration of corrosion protection oils in the solvent bath, allowing companies to efficiently monitor whether their parts are sufficiently conserved for later storage. “The GEISS Digital Indexer is basically the counterpart to our GEISS PER Density Test for perchloroethylene, which was already unveiled in 2017 at the parts2clean and received an excellent response,” said CEO Bastian Geiss.

Besides the new and established products for solvent monitoring and care, Richard Geiss GmbH’s high-purity distillates will, of course, also be showcased in Stuttgart. The globally operating recycling company specializes in the processing of solvents and their return into a functioning circulation.
system. The distillates achieve 100 percent of the original product’s quality while saving up to 90 percent in carbon emissions.

Contract degreasing, first introduced as a major feature at parts2clean 2017, is another key topic. Given the growing demand for professional contract degreasing, the solvent specialist has since launched an additional parts cleaning system, consistently expanding its range of high-end degreasing services. At the company’s headquarters in Offingen, Richard Geiss GmbH offers reliable contract degreasing for small parts and components measuring up to 1,200 mm x 800 mm x 970 mm in size.

Richard Geiss GmbH, headquartered in Offingen in the Günzburg district, is one of Europe’s leading specialists for recovered solvents. The company supplies solvents for industrial surface cleaning and textile cleaning as well as for the chemical and pharmaceutical industry.

MORE INFO  www.geiss-gmbh.de

The ASTES4’s integrated automated sorting solution allows for less manual loading, unloading, and sorting for higher productivity and efficiency. (Courtesy: MC Machinery)

MC Machinery to demo several new machines at FABTECH show

New automation and machines will be on display at MC Machinery Systems’ booth A2923 at this year’s FABTECH show, November 11-14 in Chicago.

The Advanced 800 eX-F Series fiber laser will be on display with upgraded ASTES4 automation. This fiber laser features key technologies to amplify overall machine sophistication and speed, including the M800 control and Mitsubishi’s all-in-one Zoom Head. The ASTES4’s integrated automated sorting solution allows for less manual loading, unloading, and sorting for higher productivity and efficiency.

“We are very excited to unveil the ASTES4 system at FABTECH,” said Shane Herendeen, North American sales manager — fabrication division. “The new automation system will change the way companies use their machines and sort their parts and is a huge step forward for our industry as a whole.”

The BH13530 press brake will also be on display with all-new automatic tool changer. The BH13530 includes a larger control, improved cycle times and a friendlier user interface, as well as the Diamond BH “Dual Drive” system which controls high-speed up and down movement with a ball screw and bending movement by the servo-hydraulic piston. The new automatic tool changer provides quicker and more precise tool setup for maximum machine uptime.

Additional press brakes, the BB6020 with VIDERE and the all-new compact BB306 will also be on display. Both BB Series electric press brakes offer solid performance and include 100 percent Mitsubishi electronics, motors,
and servo drives with an AC Servo motor and ball screw mechanism for high-speed productivity and repeatability. The VIDERE operator support system, showcased on the BB6020, simplifies press brake operations by clearly displaying functional process information that is easy-to-understand — setup and work handling that once required a skilled operator is now streamlined by VIDERE, improving productivity by reducing bending defects and eliminating non-value-added steps in daily machine interactions. The BB306 is the newest BB Series offering that, with its smaller footprint, can be configured to fit any facility.

InspecVision, the high-speed integrated 2D and 3D Measurement System will be on display. InspecVision brings speed, accuracy and reliability to your measurement quality process. The 2D system can be used to simply and quickly verify product quality by performing 2D inspection, CAD comparisons and reverse engineering. This 3D system is the world’s first to measure surfaces and edges in 3D by using a high-speed, high-resolution industrial camera and an LED DLP projector to scan the surfaces of an object.

**Starrett salutes goods made in America in special edition brochure**

The L.S. Starrett Company, a leading global manufacturer of precision measuring tools and gages, metrology systems, and more, has recently released an engaging special-edition brochure that heralds the company’s history and commitment to American-made quality and innovation. The 18-page brochure includes overviews of Starrett’s five U.S. manufacturing facilities where thousands of Starrett Precision tools, metrology systems, gages, shop tools, and saw blades are made. An introduction from Douglas A. Starrett, overview of the company’s key competencies, photos of skilled employees, and a laminated pull-out Starrett/U.S. flag poster are also included.

“Spanning most of our storied 139-year history, Starrett has been and still is, a global company. But the hallmark of Starrett quality and innovation is rooted in America,” said Starrett, president and CEO. “And so it is today, where the lion’s share of Starrett tools are manufactured. True to our tradition, we are committed to skilled, dedicated American workers and making tools in the United States of America.”

The printed brochure, with a laminated pull-out American flag poster, is available free by request online at www.starrett.com/american-made. In addition, a digital flipbook of the brochure can be viewed at the same link.

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**MORE INFO** [www.mcmachinery.com](http://www.mcmachinery.com)

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**MORE INFO** [www.starrett.com](http://www.starrett.com)

**Hauser GTS-1 Broach grinder for manufacturing and resharpening flat, round and spirex broaches.**
- Available probe and dressing.

**Hauser GTS-2 Skiving sharpening machine for regrinding skiving tools.**
- Available probe.
- Four CNC axes.

**MORE INFO** [www.hauserincorp.com](http://www.hauserincorp.com)
With 70 years of experience, we can handle all your custom workholding needs. Drewco Corporation is a family run business led by a team of engineers and machinists. We are backed by original patents, years of experience, and proven effective designs.
AGMA courses focus on operator-level training to help with recruitment, retainment

A large part of the responsibility of the Education Services department at AGMA is evaluating our course offerings in an attempt to facilitate courses that are aligned with the needs of the industry. In February 2017, the department corralled a group of instructors with diverse backgrounds and levels of expertise to perform a gap analysis on the AGMA course offerings. We organized the courses according to the gear-design cycle from the application to the finished product. In addition, we analyzed the Skills Assessment Tool to determine if there were, in fact, courses that could be developed from a tool that was constructed by a dedicated group of industry professionals in previous years. The answer was yes.

In addition to gathering data to evaluate our courses, the AGMA Foundation conducted a survey in August 2017. Members were finding it difficult to recruit and train operator level employees. “Finding qualified, hourly, plant staff” was identified as the top problem facing U.S. gear manufacturers in that survey. Entry level training surfaced as another major issue. In response to the instructor-led group’s findings and the AGMA Foundation’s survey, four new courses were born for operators: hobbing, grinding, inspection, and heat-treat equipment operator.

This will be the first time AGMA offers courses on this fundamental level. The Basic Training for Gear Manufacturing course, also in Chicago, is where individuals learn the fundamentals of gear design and how to actually cut a gear. This course is taught on a series of about seven manual machines. As we move to expand the center for the new operator courses, our plan is to accommodate various learning styles and integrate updated machinery to accomplish the much-needed industry training needs.

Today’s manufacturing workforce is rapidly aging out. Industry Week estimates 10,000 manufacturing employees retire every day. To solve the recruitment problem, gear manufacturers need to overcome the “dirty, dumb, and dangerous” myth by educating new recruits on the high-tech, clean-shop, good-jobs reality of today’s manufacturing environment. To do this, the AGMA Foundation released the “Get Into Gears” employee recruitment toolkit in December 2018.

The kit is bright and lively, with photos and text touting the importance of our work and the good jobs and benefits available in our shops. The kit contains traditional hard copy materials — a brochure, poster, and postcard templates for manufacturers to customize, as well as social media graphics and a PowerPoint presentation to appeal to today’s tech-savvy youth. Capping off the package is a two-minute video on the industry which, again, highlights our vital work performed in a clean, safe environment. The “Get Into Gears” toolkit is a free download on the Foundation’s website: www.agmafoundation.org.

The development of the courses and toolkit is a pro-active response to the need to recruit and train people from the ground up to fill the transient manufacturing pool. With the funding provided from the AGMA Foundation, the AGMA Education Services department’s instructional team is completing the development of these courses. Classes will be offered as a part of the 2020 class schedule at the new AGMA National Training Center in collaboration with Richard J. Daley College in Chicago, Illinois.
Upcoming AGMA Courses

Gear Failure Analysis
November 6-8, 2019 | St. Louis, Missouri
Explore gear failure analysis in this hands-on seminar where students not only see slides of failed gears but can hold and examine those same field samples close up. Use of a microscope to examine field samples.

Epicyclic Gear Systems: Application, Design & Analysis
December 3-5, 2019 | Seattle, Washington
Learn and define the concept of epicyclic gearing, including some basic history and the differences among simple planetary gear systems, compound planetary gear systems, and star drive gear systems. Cover concepts on the arrangement of the individual components including the carrier, sun, planet, ring, and star gears, and the rigid requirements for the system to perform properly. Topics covered include critical factors such as load sharing among the planet or star gears, sequential loading, equal planet/star spacing, relations among the numbers of teeth on each element, and the calculation of the maximum and optimum number of planet/star gears for a specific system. Provides an in-depth discussion of the methodology by which noise and vibration may be optimized for such systems and load sharing guidelines for planet load sharing.

Online Education
Don’t have the ability to come to one of AGMA’s fantastic face-to-face courses? We understand that you are busy, and that is why we offer online education to meet your schedule. Now you can grow your gear knowledge, get the same quality AGMA education, and save money on travel by learning directly at your own computer.
AGMA’s online education courses include:
- Gear Failure Analysis.
- Gearbox CSI: Gears Only.
- Detailed Gear Design–Beyond Simple Service Factors.
- Fundamentals of Gearing.
- Hobbing.
- Parallel Gear Inspection.
Supply Chain Management Courses

Through funded research from the AGMA Foundation, AGMA and Ranken Technical College have teamed up to provide the gear industry with Supply Chain Management courses that are conveniently online!

Topics covered in this six-course program include:
- Integrated Supply Chain Management.
- Inventory Management.
- Manufacturing and Service Operations.
- Order Fulfillment and Customer Service.
- Transportation and Warehousing.

The program curriculum will also prepare students for the Council of Supply Chain Management (CSCM) SCPro Certification. To find out more, go to: www.agma.org/education/online/supply-chain-management-certification-program.

SAVE THE DATE

2020 AGMA Events: Registration is now open

JANUARY 28-30
Manufacturing & Inspection | TBD (Check back on website)

FEBRUARY 8-20
Worm Gear | Alexandria, Virginia

MARCH 19-21
AGMA/ABMA Annual Meeting | Lake Buena Vista, Florida

MARCH 24-26
Steels for Gear Application | Alexandria, Virginia

APRIL 13-17
Basic Training for Gear Manufacturing | Chicago, Illinois - Daley College

APRIL 21-22
Basic Gear Inspection for Operators | Chicago, Illinois - Daley College

MAY 12-13
Operator Hobbing & Shaper Cutting | Chicago, Illinois - Daley College

MAY 19-21
Gearbox CSI | Concordville, PA

JUNE 23-24
Operator Gear Grinding | Chicago, Illinois - Daley College

JUNE 16-18
Gear Failure Analysis | St. Louis, Missouri - Ranken Technical College

JULY 14-15
Basics of Gearing | Chicago, Illinois - Daley College

JULY 21-23
Detailed Gear Design | St. Louis, Missouri - Ranken Technical College

AUGUST 11-13
Fundamentals of Gear Design and Analysis | Chicago, Illinois - Daley College

AUGUST 25-27
Epicyclic Gear Design | Chicago, Illinois - Daley College

SEPTEMBER 9-10
Heat-Treat Equipment Operator | Chicago, Illinois - Daley College

SEPTEMBER 21-25
Basic Training for Gear Manufacturing | Chicago, Illinois - Daley College

OCTOBER 13-15
Gearbox Systems Design | Clearwater Beach, Florida

OCTOBER 19-21
Fall Technical Meeting | Rosemont, Illinois

NOVEMBER 10-12
Gear Failure Analysis | St. Louis, Missouri - Ranken Technical College

DECEMBER 1-3
Gear Systems Design for Minimum Noise | Phoenix, Arizona
CALENDAR OF EVENTS

Whether you're looking for technical education, networking opportunities, or a way for your voice to be heard in the standards process, AGMA has something to offer you. If you would like more information on any of the following events, visit www.agma.org or send an email to events@agma.org.

October

October 14-16 — Fall Technical Meeting — Detroit, Michigan
October 15-17 — Motion + Power Technology Expo — Detroit, Michigan
October 17 — Powder Metallurgy Committee — Detroit, Michigan
October 22 — Helical Gear Rating Committee Meeting — WebEx
October 23 — Lubrication Committee Meeting — WebEx
October 24 — Metallurgy and Materials Committee Meeting — WebEx
October 28 — Cutting Tools Committee Meeting — WebEx
October 30 — Flexible Couplings Committee Meeting — WebEx
October 30 — Nomenclature Committee Meeting — WebEx
October 31 — Gear Accuracy Committee — WebEx

November

November 5 — Wormgearing Committee Meeting — WebEx
November 6 — Plastics Committee — WebEx
November 21 — Lubrication Committee Meeting — WebEx

AGMA LEADERSHIP

John Cross: Chairman
ASI Drives

Greg Schulte: Treasurer
Bonfiglioli USA

Michael McKernin: Chairman, BMEC
Gear Group at United Stars, Inc.

Todd Praneis: Chairman, TDEC
Cotta Transmission Company, LLC

Jim Bregi: Chairman Emeritus
Doppler Gear Company

Matt Croson: President

Amir Aboutaleb: VP, Technical Division

Jenny Blackford: VP, Marketing

Jill Johnson: Director, Member Services

Casandra D. Blassingame, M.Ed.: VP, Education Services

Zen Cichon: Innovative Rack & Gear Company

Michael Engesser: Reishauer Corporation

Bent Hervard: CFT

Ruth Johnston: Croix Gear & Machining

David R. Long: Chalmers & Kubeck Inc.

Jack Masseth: Meritor, Inc.

Scott Miller: Caterpillar, Inc.

Gary Neidig: ITAMCO

Shawn O'Brien: McNees Rolled Rings

Cory Oyen: Global Gear & Machining, LLC

Carl D. Rapp: The Timken Company

Sara Zimmerman: Sumitomo Drive Technologies

General requests: webmaster@agma.org | Membership questions: membership@agma.org | AGMA Foundation: foundation@agma.org
Technical/Standards information: tech@agma.org | MPT Expo information: mptexpo@agma.org
AIM HIGH.

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Tips to increase vibratory performance during gear polishing

An attentive eye and some simple fixes will keep things going when there are issues during the process.

In our final article in this series on gear polishing using vibratory equipment, let’s take a look at some troubleshooting tips to increase the efficiency of your vibratory equipment’s performance.

VIBRATORY MEDIA OF CHOICE

Choose the least abrasive media possible, or better yet a high-density, non-abrasive media if you are using an isotropic finishing chemistry to polish your gears. The lower the abrasive content, the longer the media maintains its form, and the less media swarf solids that are generated and deposited down the drain.

Choosing media with a lower abrasive content also means that the media will have a lower attrition rate. As such, it will maintain its size and shape for a longer period of time than a media that is more abrasive in content. In applications where media size and shape are critical to contact specific locations on the parts being processed, it will be exceptionally advantageous to maintain media size and form for as long a period of time as it’s possible.

MEDIA LEVEL IN THE VIBRATORY BOWL

Media is the tool in vibratory finishing that performs the work to increase surface quality on your gears. Don’t skimp on media volume to save money. A properly loaded vibratory bowl will have the media level two inches below the O.D. rim. Running with less media mass is simply inefficient.

LIQUID GEYSERING

Does your vibratory bowl spit up after about 10 minutes of operation with the result being geyers of foam shooting into the air and onto the ground around your machine? If this occurs, this is an indication that there is a problem with the machine’s drainage. Stop the machine and pull the media away from the in-bowl drain. It’s probably plugged with debris. Take some time to clean out the holes so fluid will drain better.

What about the drain tubing? Is it clogged with debris such that good drainage is nearly impossible? Now would be a good time to clean it out or possibly replace it with new tubing.

How is the drain tubing positioned beneath your machine? The drain tubing should gravity flow for its entire length into a floor sump to efficiently remove soiled solutions from the machine. In applications where the sump may be ten feet or more away from the machine it is not uncommon to see a long stretch of drain tubing extended for some distance horizontally along the floor. In such installations, the tubing is prone to quickly clog with swarf, debris, metal fines, etc. It is a better option in such situations to drain into a small plastic tank beneath the bowl’s drain. This tank can be fitted with a float-controlled submersible sump pump. Upon filling with spent liquid, the upper level float will activate the pump which will then drain the accumulated liquid. In such situations you will have the additional advantage of directing the pump’s discharge flow to the exact location you would like to send it. In most facilities this will be a centralized spent waste liquid collection tank.

UNLOADING RAMPS AND MOTOR INVERTERS

Does your vibratory set-up include a ramped-bottom bowl that will discharge parts at the end of the processing cycle? If so, keep in mind that during the vibratory processing cycle, the media mass will effi-
ciently separate parts from one another. However, during the unloading cycle, the parts will extrude from the mass and the incidence of part-on-part damage will increase. Fully separated parts will then bounce across the media separation deck where they are completely exposed and completely in contact with one another. Should your bowl be fortunate to have a motor speed potentiometer or a motor inverter, now would be the time to consider setting the motor speed at a lower level. In North America, maximum motor speed is 60 Hz. However, during the unloading step you may find it more advantageous to lower motor speed to 48-50 Hz. This will minimize the force of part-on-part contact and reduce the incidence of part bouncing across the media separator deck.

Don’t forget, once the parts exit the separator deck they will usually gravity fall into a collect box of some type. Long drops, especially for parts weighing more than one pound, will most certainly generate part-on-part damage. Consider equipping your separation deck with an angled ramp such that parts will slide into a collection box.

SOUND COVERS
Sound covers are a marvelous convenience in locations where sound attenuation is desired. But be aware that the cover will also insulate the media mass when it is closed. Frictional force during media roll will heat the media mass, but with the cover in place, this heat will be unable to dissipate. As such, when a cover is typically in use be aware that a higher liquid flow rate may be necessary to replace fluid loss due to evaporation.

VIBRATORY BOWL MORPHOLOGICAL FORM:
THE PIE WEDGE AND PART LOADING
Consider the form of a vibratory bowl. It is in the shape of a toroid, or more commonly, the form of a bagel. The operating channel is an annulus limited on the ID by the machine’s center hub and on the OD by the machine’s outer wall. The diameter of the center hub is shorter than the diameter of the entire machine from OD wall to OD wall. This means that the hub’s circumference is shorter than the OD wall’s circumference.

As parts plunge downward at the ID wall, be aware that the bowl’s toroidal form creates channel sections that are pie-wedged in shape. If we look at the bowl from overhead and divide it into quarters, each resultant pie wedge is narrower on its ID side. Therefore, even appropriately separated parts will be in closer proximity to one another on the ID side of the channel. Load delicate parts accordingly, being mindful of this physical constraint caused by toroidal form.
Gear mesh and tooth modifications

The involute gear form allows for a fluid mesh of the teeth as they slide along one another.

As we march through this thing that we all call life, we look to find our perfect partner — that singular person that perfectly meshes with our personality, improves our weaknesses, and complements our strengths. Sometimes we will make enhancements to our physical self in order to iron out some of our perceived surface flaws. As with life, the same is true about gearing.

The most common tooth form in gearing is the involute tooth form. This form allows for a fluid mesh of the teeth as they slide along one another. In Figure 1, a pair of standard involute gears are meshing together. The contact point of the two involutes, as Figure 1 shows, slides along the common tangent of the two base circles as rotation occurs. The common tangent is called the line of contact, or line of action.

A pair of gears can only mesh correctly if the pitches and the pressure angle are the same. The requirement that the pressure angles must be identical becomes obvious from the following equation for base pitch $P_b$:

$$P_b = \pi m (\cos \alpha)$$

where

$m$ = module, and

$\alpha$ = the pressure angle.

Thus, if the pressure angles are different, the base pitches cannot be identical.

The contact length $ab$ shown in Figure 1 is described as the “Length of the path of contact.” The contact ratio can be expressed by the following equation:

Transverse Contact Ratio $\epsilon_\alpha = \frac{\text{Length of path of contact } ab}{\text{Base pitch } P_b}$

In practice, you should always aim to maintain a transverse contact ratio of 1 or greater. Module, $m$, and the pressure angle, $\alpha$, are the key items in the meshing of gears.

In order to improve the mesh, there are three common methods of modifying the tooth form: the tooth profile modification, crowning and end relief, and topping or semi-topping.

In most gear literature, tooth profile modification generally means adjusting the addendum. However, tooth profile adjustment can be done by chamfering the tooth surface in order to make an incorrect involute profile on purpose. This adjustment enables the tooth to vault when it gets the load, so it can avoid interfering with the mating gear. This is effective for reducing noise and creating a longer surface life. However, too much adjustment can create bad tooth contact as the result functions the same as a large tooth profile error. An example of this modification is seen in Figure 2.

Crowning is the removal of a slight amount of the tooth from the center of the tooth on outward to the tooth edge, making the tooth surface slightly convex. This method allows the gear to maintain contact in the central region of the tooth and creates an avoidance of edge contact. Crowning should not be larger than necessary, as it will reduce the tooth contact area, thus weakening the gears’ strength. End relief is the chamfering of both ends of the tooth surface. An example of these modifications is seen in Figure 3.

In topping, often referred to as top hobbing, the top or tip diameter of the gear is cut simultaneously with the generation of the teeth. This type of gear generation is produced usually when using rack type cutters. An advantage is that there will be no burrs on the tooth top. Also, the tip diameter is highly concentric with the pitch circle.

Semitopping is the chamfering of the tooth’s top corner, which is accomplished simultaneously with tooth generation. Figure 4 shows a semitopping cutter and the resultant generated semitopped gear. Such a tooth end prevents corner damage and has no burrs.

The magnitude of semitopping should not go beyond a proper limit; otherwise, it would significantly shorten addendum and contact ratio. Figure 5 shows a recommended magnitude of semitopping. Topping and semitopping are independent modifications but, if desired, can be applied simultaneously.

With each gear pair there is a unique mesh that will provide the optimum result. It may include any or all of the tooth modifications above in order to reach that ideal.

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**ABOUT THE AUTHOR**

Brian Dengel is general manager of KHK-USA, which is based in Mineola, New York. Go online to www.khkgears.us.
When you visit Motion & Power Technology Expo (aka Gear Expo) in Detroit, be sure you stop at booth 4439 and see GMTA, a leader in gear machine technology, as well as laser joining and parts washing equipment for the industry.

Say hi to Walter Friedrich and ask him, “OK, Walter, what’s the big secret?” He might tell you.

If you do broaching, you might want to hear his answer.

See you in the Motor City!
**Quench system monitoring requirements in AMS 2759G**

Determining the minimum hardness required by AMS 2759G requires attention to the specific hardenability curve of the specimen material and proper equipment maintenance.

AMS 2759G Heat Treatment of Steel Parts, General Requirements [1] was adopted April 23, 2019. This important aerospace specification governs the heat treatment of aerospace parts and incorporates a number of changes related to testing the quenchant and, most importantly, the quenching system. Previously, this specification only focused on the quenchant and did not look at the entire quench system, which includes the quenchant, but also includes material handling and agitation.

The Aerospace Materials Engineering Committee (AMEC) of SAE decided to examine the entire quenching process and quench system instead of focusing strictly on the quenchant. The quenchant, agitation, and material-handling system (quench system) must operate together to ensure that parts are quenched properly. Agitation and material handling are difficult to quantify. The quenchant can be suitably described by various testing methods, including cooling curves (ASTM D6200 [2]). However, there is presently no method to qualify agitation, as the presence of different parts can affect the fluid flow. In light of this, AMEC decided to base the quality of the quench system on the basis of achieving acceptable quenching of specific bars of material. The test materials are one of the following:

- SAE 4130 round bar, a minimum of 1.5” (3.81 cm) long, and 0.5” (1.27 cm) nominal diameter.
- SAE 4140 round bar, a minimum of 4.5” (11.43 cm) long, and 1.5” (3.81 cm) nominal diameter.
- SAE 4330V round bar, a minimum of 7.5” (19.05 cm) long, and 2.5” (6.35 cm) nominal diameter.

These specimens, after quenching, shall have a hardness “not less than the hardness on the end-quench hardenability curve corresponding to the diameter of the specimen when tested in accordance with ASTM A304.” [2] [3]

These samples are to be run quarterly with either a typical or simulated production load processed to AMS 2759 requirements. After the parts are quenched, a 0.5” (12.5 mm) segment from the center of the specimen is to be removed from the test specimen in the as-quenched condition.

**JOMINY END-QUENCH TESTING**

The Jominy end-quench test, is a rather ingenious test invented by Walter E. Jominy (1893-1976) and A.L. Boegehold [4]. In this test, a round steel bar 1” in diameter by 4” long is heated to 1,600°F. The bar is then removed from the furnace and placed in a special fixture. The special fixture allows water to strike one end of the sample. This quenched end experiences the fastest quench rate, and distances from the quenched end (usually measured in n/16”) cool progressively slower. Two flats are ground in the bar, and hardness is measured as a function of the distance from the quenched end. The basic apparatus is shown in Figure 1.

This test forms the basic foundation for the measurement of hardenability in steels. If you remember, hardenability is not how hard the steel gets when quenched but rather how deeply it is hardened.

Interestingly, this test has been used for aluminum [5] and titanium [6] to characterize microstructure as a function of quench rate.

The hardness data is typically plotted as a function of distance from the quenched end on standardized forms [3]. Figure 2 is an illustration of the form from ASTM A304 [3].

There are several things to notice in this graph. The first things to notice are the upper and lower hardenability bands for the specific alloy. In general, the alloy, when tested, must lie between the upper and lower limits of the hardenability curve.

The upper and lower hardenability limits are in HRC as a function of distance from the quenched end of the Jominy end-quench specimen. Due to end effects, the hardness measurements are usually started at n=2/16”. Occasionally, the end quench hardness at n=0 is taken on the end of the specimen.

The second thing to notice is the table at the top of the figure showing the diameters of rounds with the same as-quenched hardness for a mild water quench and a mild oil quench. Radial distances of surface (r=R), three-quarter radius (r = 3/4R), and the center of the bar (r = 0) are described. While not clear in the specification, it is these values that are used to establish the acceptable quench system limits in AMS 2759.

The distances relating to the diameters of bars with the same as-quenched hardness remain the same and are a function of the distance from the quenched end. Plotting the bar diameters for surface, 3/4 radius, and center for a mild oil quench as a function of distance from the quenched end yields a graph (Figure 3). We will use this graph to establish the Jominy distances for each of the bar diameters cited above, in AMS 2759.

Ignoring for a moment the specific speci-
Men hardenability used in AMS 2759, it is possible to detail the limits of minimum hardness and the location on the Jominy end-quench curve. Using the graph in Figure 3, we remember that the SAE 4130 bar has a specimen size of 0.5". These specimens, after quenching, shall have a hardness “not less than the hardness on the end-quench hardenability curve corresponding to the diameter of the specimen when tested in accordance with ASTM A304” [2] [3]. Where is the Jominy distance corresponding to this requirement?

Looking at Figure 3, we see that the bar diameter, 0.5", measured at the center of the bar, corresponds to a Jominy distance of J = 3.5/16" for SAE 4130. For the SAE 4330V bar, it has a nominal diameter of 2.5".

Looking at Figure 3 for the distance corresponding to the center hardness for a bar 2.5” in diameter, this corresponds to a distance from the quenched end of 14.5/16”.

The case for SAE 4140H (Figure 2) is a bit different. In this case, two hardesses at different locations are specified. Also, the hardness at the center of the bar shall not be less than HRC 44, and the 3/4 Radius shall not be less than HRC 50. However, while appearing to be more stringent, the test requirement is not more stringent.

Looking at the Jominy curve for SAE 4140H with a diameter of 1.5”, the center of the bar corresponds to J = 9/16", and the 3/4 Radius corresponds to a Jominy distance of J = 6/16”. Taking the hardness of HRC 44 at the center, and HRC 50 at the 3/4 Radius, we see that these values correspond to the minimum hardenability curves for SAE 4140.

The values for the specific Jominy end-quench distances are specific in Table 1.

When raw material is obtained for performing the quench system monitoring test, the user should request the specific Jominy end-quench data for the heat of steel. Using the Jominy distances in the specific heat lot certification, the minimum acceptable hardness can be determined. If the hardness, when tested quarterly, falls below the minimum hardness, then corrective action should be taken. This corrective action could be maintenance on agitators or material handling systems. This also assumes that the quenchant used meets the requirements and is similar to original specifications. As long as the proper oil has been chosen, then the quenchant is usually not the problem.

CONCLUSIONS

In this short article, the methodology for determining the minimum hardness required by AMS 2759 for the quench system monitoring is described. It requires attention to the specific hardenability curve of the specimen material, as well as proper maintenance of equipment.

This is the first attempt to ensure that the entire quench system (quenchant, agitation, and handling) is up to the task of heat-treating aerospace components.

REFERENCES


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ROOT CAUSE FOR NOISE EXCITATION IN GEOMETRICALLY-ACCURATE (PERFECT) PARALLEL-AXES GEARING
Theoretically, geometrically-accurate gearing that features zero base pitch variation promise to be noiseless, as no noise is produced by the gear mesh. In reality they can be noisy.

By STEPHEN P. RADZEVICH

Gears are extensively used to transmit a rotation from a driving shaft to a driven shaft. Depending on the gear application, various configurations of the gear-shaft axes are distinguished. In the general case, gear-shaft axes cross one another; in one particular application, the gear shaft axes intersect one another; finally, in another particular application, the gear shaft axes are parallel to one another.

Gears used to transmit a rotation from a driving shaft to a driven shaft parallel to one another are commonly referred to as “parallel-axes gearing” (or just “Pa — axes gearing,” for simplicity). Spur and helical gearing, double-helical and herring-bone gearing, as well as a few more kinds of gearing, are perfect examples of “parallel-axes gearing.” Figure 1 illustrates a trivial application of “Pa — axes gearing.” Gearing of only this particular kind, that is Pa — axes gearing, is considered in this article.

The root cause for noise excitation in gearboxes and in gear transmissions is discussed in our earlier paper [1]. It is shown that the deviation of the angular base pitch of a gear (and that of a mating pinion) from operating base pitch of the gear pair, is the root cause for an excessive gear-noise excitation and vibration generation. It is also shown that only in geometrically-accurate gearing (or, in other words, in perfect gearing), the angular base pitch of a gear (and that of a mating pinion) both are equal to the operating base pitch of the gear pair. Does it mean that perfect gear pairs are noiseless, and do not generate noise and vibrations? No, it doesn’t. So, what is the root cause for the noise excitation in geometrically accurate gearing when the gear mesh produces no noise?

To answer this question, let’s begin with the analysis of the interaction of a gear and a mating pinion tooth flank.

INTERACTION OF A GEAR AND A MATING PINION TOOTH FLANK

In a parallel-axes gear pair, an input rotation and an input torque are transmitted from a driving shaft to a driven shaft by means of forces acting within the plane of action, PA.

The transmission of rotation in geometrically-accurate Pa — gearing is illustrated in Figure 2. Here, the input rotation, \( \omega_{input} \), and the input torque, \( T_{input} \), along with the output rotation, \( \omega_{output} \), and the output torque, \( T_{output} \), are shown.

In Figure 2, the input rotation, \( \omega_p \), is transformed to the corresponding output rotation, \( \omega_g \). The correlation
between the rotations can be expressed by the following formula:

\[ \omega_g = \frac{\omega_p}{u} \]  

Equation 1

where the gear ratio is designated as \( u \).

Magnitude, \( F_t = |F_t| \), of the acting tangential force, \( F_t \), is equal:

\[ F_t = \frac{T_p}{2d_{bg}} \]  

Equation 2

where base diameter of the driving pinion is designated as \( d_{bg} \).

The acting tangential force, \( F_t \), is evenly distributed, \( f_{t,pa} \), along the face width, \( F_{pa} \), of the gear pair. An equation:

\[ f_{t,pa} = \frac{F_t}{F_{pa}} \]  

Equation 3

can be used for the calculation of the distributed load, \( f_{t,pa} \).

When one pair of the teeth is engaged in mesh as illustrated in Figure 3a, the applied load is evenly distributed along a single line of contact, \( LC \), the resultant (or, the equivalent) force, \( F_t \), is applied at point, \( c_g \), at the middle of a single line of contact, \( LC \). The force \( F_t \) is along a line of action, \( LA \), through the point \( c_g \).

Equation 2 is valid for the calculation of a distributed load in spur parallel-axes gear pairs. In a case of helical parallel-axes gearing the distributed load, \( f_t \), equals:

\[ f_t = \frac{F_t}{F_{pa}} \cos \psi_b \]  

Equation 4

where the base helix angle is designated as \( \psi_b \).

Both, the normal distributed load, \( f_{t,n} \), and the axial component, \( f_{t,a} \), of the distributed load can be expressed in terms of the distributed load, \( f_t \), (see Equation 4).

Equation 3 is valid in cases when a gear and a mating pinion tooth flanks, \( G \) and \( P \), contact each other along a single line of contact, \( LC \). As the total contact ratio, \( m_t \), in a gear pair is always greater than one \( (m_t > 1) \), in reality two or even more lines of contact, \( LC_i \), are observed.

Let’s consider the load transmission for the cases when the plane of action, \( PA \), is shaped in the form of a rectangle.

When two (or more) pairs of the teeth are engaged in mesh as illustrated in Figure 3b, the applied load is evenly distributed along the corresponding number of the lines of contact, \( LC_i \). The distributed load, \( f_t \), is shared among \( n_f \) pairs of teeth engaged in mesh at that same instant of time. The distributed load per a pair of teeth, \( f_{t,nf} \), equals:

\[ f_{t,nf} = \frac{f_t}{n_f} = \frac{F_t}{n_f F_{pa}} \]  

Equation 5

The resultant force, \( F_r \), is equally shared among all the lines of contact, \( LC_i \). The resultant force per a pair of teeth, \( F_{t,nf} \), equals:

\[ F_{t,nf} = \frac{F_t}{n_f} \]  

Equation 6

All the forces \( F_{t,nf} \) are along a line of action, \( LA \), through the middle of the lines of contact, \( LC_i \).

The resultant (or the equivalent) force, \( F_r \), is applied at point \( c_g \). Point \( c_g \) corresponds to the middle of the lines of contact, \( LC_i \), and between the first, \( LC_1 \), and the last, \( LC_n \), lines of simultaneous contact. When two lines of contact are observed, point \( c_g \) is located in the middle in between the lines of contact. When three lines of contact are observed, point \( c_g \) is located at the center of the second line of contact, and so forth.

Similar to the case of a single line of contact, in cases of multiple lines of contact, the force \( F_t \) is along a line of action, \( LA \), through point \( c_g \).

When the gear rotate, \( \omega_g \), the plane of action, \( PA \), travels straight, \( V_{pa} \) (Figure 4). The arm, \( R_{cg} \), of the resultant (or the equivalent) force, \( F_r \), with respect to the bearing support remains the same and equals:
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where:

\[ r_{bg} \] is the base radius of the gear.

\[ a \] is the distance of the gear from the bearing support (Figure 4).

Point, \( c_g \), does not migrate in the axial direction of the gear. Therefore, the force, \( F_t \), creates a bending moment of a constant value with respect to the bearing support. Thus, no variation of the deflection of the shafts, of the gear housing, and so forth, are observed in the case under consideration. Spur gears create no additional source for noise excitation.

A more general case of contact between tooth flanks, \( \beta \) and \( \phi \), of a gear and a mating pinion is observed in geometrically-accurate helical \( P_a \)-axes gearing.

When the gears rotate, commonly either one or two to three lines of contact, \( LC_i \), are observed in geometrically-accurate helical \( P_a \)-axes gearing. In two different cases, it is convenient to distinguish in helical gearing, namely:

(a) when all the lines of contact, \( LC_i \), are of a full length, and
(b) when one or more line(s) of contact is of a reduced length.

When the only pair of gear teeth is engaged in mesh, the entire applied load is evenly distributed along a single line of contact, \( LC \). This case is illustrated in Figure 5a.

The distributed load, \( f_t \), is calculated from the equation:

\[ f_t = \frac{F_i}{F_{pa}} \]  

The resultant (the equivalent) force, \( F_t \), is applied at point \( c_g \) exactly at the middle of a single line of contact, \( LC \). The force \( F_i \) is along a line of action, \( LA \), through the point \( c_g \).

Point, \( c_g \), where the load, \( F_t \), is applied does not migrate in the axial direction when a gear rotates. Therefore, bending moment of the force, \( F_t \), with respect to the bearing support, is of a constant value.

Normal force, \( F_{tn} \), is perpendicular to the line of contact, \( LC \). The component \( F_{tn} \) of the resultant (the equivalent) force, \( F_t \), is applied at that same point, \( c_g \), as the load \( F_i \) is applied. The magnitude, \( F_{tn} \), of the component \( F_{tn} \) of the resultant (the equivalent) force, \( F_t \), equals to:

\[ F_{tn} = F_t \cos \psi_b \]  

Equation 9

Normal distributed load, \( f_{tn} \), can be calculated from the expression:

\[ f_{tn} = \frac{F_i}{F_{pa}} \cos^2 \psi_b \]  

Equation 10

When two (or more) pairs of the teeth are engaged in mesh, the applied load is shared among a few number of lines of contact, \( LC_i \). This case is illustrated in Figure 5b. The distributed load, \( f_t \), is shared among \( n_\phi \) pairs of teeth engaged in mesh at that same time. The distributed load per a pair of teeth, \( f_{tn \phi} \), equals:

\[ f_{tn \phi} = f_t = \frac{F_i}{n_\phi F_{pa}} \]  

Equation 11

The resultant force, \( F_t \), is shared among all the lines of contact, \( LC_i \). The resultant force per a line of contact, \( F_{tn \phi} \), equals:

\[ F_{tn \phi} = \frac{F_i}{n_\phi} \]  

Equation 12

The resultant (the equivalent) force, \( F_t \) [as well as the resultant (the equivalent) normal force, \( F_{tn} \)], is applied at point \( c_g \). Point \( c_g \) corresponds to the middle of the lines of contact, \( LC_i \), and between the first, \( LC_i \), and the last, \( LC_j \), lines of simultaneous contact. When...
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two lines of contact are observed, point \( c_g \) is located in the middle in between the lines of contact. When three lines of contact are observed, point \( c_g \) coincides with the center of the second line of contact, and so forth.

The normal distributed load, \( f_{ln} \), can be calculated from the expression:

\[
f_{ln} = \frac{F_t}{F_{pa}} \cos^2 \psi_b
\]

Equation 13

Point, \( c_g \), does not migrate in the axial direction of the gear. Therefore, the force, \( F_t \), creates a bending moment of a constant value with respect to the bearing support. Thus, no variation of the deflection of the shafts, of the gear housing, and so forth, is observed in the case under consideration. No additional source for noise excitation is created in the case under consideration.

Ultimately, one or more lines of contact, \( LC_i \), can be not of a full length.

As illustrated in Figure 6a, the line of contact, \( LC_i \), is not of a full length, while the next line of contact, \( LC_{i+1} \), is of a full length (that is, the inequality \( LC_i < LC_{i+1} \) is observed). In the example under consideration, tangential load, \( F_t \), is shared between the lines of contact, \( LC_i \) and \( LC_{i+1} \). The load \((|F_{t,1}| + |F_{t,2}|)\) equals (see Figure 6a):

where:

\[
|F_{t,1}| + |F_{t,2}| = \frac{F_{pa,1}}{F_{pa}} |F_t|
\]

Equation 14

\[
|F_{t,1}| = \frac{F_{pa,1}}{2 \cdot F_{pa}} |F_t|
\]

Equation 15

The loads \( F_{t,1} \) and \( F_{t,2} \) are applied at points 1 and 2 within the lines of contact. Points 1 and 2 correspond to points within the lines of contact, \( LC_i \) and \( LC_{i+1} \), at the middle of the portion \( F_{pa,1} \) of the face width \( F_{pa} \).

The load \( |F_{t,3}| \) equals (see Figure 6a):

\[
|F_{t,3}| = \frac{F_{pa,2}}{F_{pa}} |F_t|
\]

Equation 16

The load \( |F_{t,3}| \) is applied at point 3 within the line of contact, \( LC_{i+1} \). Point 3 corresponds to point within the line of contact, \( LC_{i+1} \), at the middle of the portion \( F_{pa,2} \) of the face width \( F_{pa} \).

The resultant load, \( F_r \), is applied at point \( c_g \). The coordinates of point \( c_g \) can be determined using the principle of the center of gravity, that is, point \( c_g \) is the “center of gravity” of the points 1, 2, and 3 having the “weights” \( F_{t,1} \), \( F_{t,2} \), and \( F_{t,3} \) correspondingly.

As the gears rotate, the lines of contact, \( LC_i \) and \( LC_{i+1} \), travel together with the plane of action, \( PA \). Points 1, 2, and 3, as well as point \( c_g \) migrate within the plane of action, \( PA \). The migration of point \( c_g \) with a linear velocity, \( V_{cg} \), in the axial direction of the gear pair results in a corresponding alteration of the arm, \( R_{cg} \), of the resultant force, \( F_r \), in relation to the bearing support of the gear shaft as schematically illustrated in Figure 7. When point \( c_g \) migrates between points \( c_g^* \) and \( c_g^{**} \), the arm \( |R_{cg}^*| \) of the resultant force, \( F_r \), changes in the range of:

\[
|R_{cg}^*| \leq |R_{cg}| \leq |R_{cg}^{**}|
\]

Equation 17

A variation of magnitude, \( \Delta R_{cg} \), of the arm, \( R_{cg} \), results in a variable in time torque that bends the gear shaft, and deforms the gear housing, and can cause an extensive unfavorable vibration of the gear housing, and an extensive noise excitation.

Another configuration of the lines of contact, \( LC_i \) and \( LC_{i+1} \), is illustrated in Figure 6b.

The lines of contact, \( LC_i \) and \( LC_{i+1} \), shown in Figure 6b , are not of full length (that is, the inequality \( LC_i < LC_{i+1} \) is observed). In the example under consideration, the tangential load, \( F_t \), is shared between the lines of contact, \( LC_i \) and \( LC_{i+1} \). The load \( |F_{t,1}| \) equals (see Figure 6b):

\[
|F_{t,1}| = \frac{F_{pa,1}}{F_{pa,1} + F_{pa,2}} |F_t|
\]

Equation 18

Point 1 is located in the middle of the line of contact, \( LC_i \). The load \( |F_{t,2}| \) equals (see Figure 6b):

\[
|F_{t,2}| = \frac{F_{pa,2}}{F_{pa,1} + F_{pa,2}} |F_t|
\]

Equation 19

Figure 7: Variation of the bending torque in helical parallel-axes gearing.

Figure 8: Forces acting in a section of a parallel-axes involute gear pair.
Variation of the gear loading along with a variation of point at which the resultant load is applied is the root cause for an excessive noise excitation in geometrically-accurate parallel-axes gearing.

Point 2 is located in the middle of the line of contact, LC_{t+1}. The load F_{t,2} is applied at point 2.

No load is transmitted by the portion F_{pa,3} of the face width, F_{pa}, as no lines of contact are located there. Therefore, this portion of the gear face is excluded from the analysis.

When calculating contact stress in a parallel-axes gear pair, use of the so-called “Total length of lines of contact” (or just TLLC, for simplicity) is often helpful. “Total length of lines of contact” means a summa of the length of all the lines of contact that occur at a specified instant of time. When two or more lines of contact do not overlap one another, the importance of the total length of lines of contact is vital. In such a case, an applied load is equally distributed along the TLLC.

The resultant load, F_t, is applied at point c.g. The coordinates of point c.g. can be determined using the principle of the center of gravity, that is, point c.g. is the center of “gravity” of points 1 and 2, having the “weights” F_{t,1} and F_{t,2}, correspondingly.

As the gears rotate, the lines of contact, LC_1 and LC_{t+1}, travel together with the plane of action, PA. Points 1 and 2, as well as point c.g. migrate within the plane of action, PA. The migration of point c.g. with a linear velocity V_{cg} in the axial direction of the gear pair results in a corresponding alteration of the arm, R_{cg}, of the resultant force, F_t, in relation to the bearing support of the gear shaft as schematically illustrated in Figure 6.

When point c.g. migrates between points c.g. and c.g., the arm |R_{cg}| of the resultant force, F_t, alters in the range of |R_{cg}^*| ≤ |R_{cg}| ≤ |R_{cg}^{**}| (Figure 7).

A variation of magnitude, |ΔR_{cg}|, of the arm, R_{cg}, results in a variable in time torque that bends the gear shaft and deforms the gear housing and can cause an extensive unfavorable vibration of the gear housing and an extensive noise excitation.

The disclosed approach for the determining of loads acting in every pair of teeth in geometrically-accurate spur and helical parallel-axes gearing can be enhanced to gear pairs with lines of contact of an arbitrary geometry, that is, to gear pairs with lines of contact in the form of arcs of a circle, of a spiral curve, and so forth. Generally speaking, for this purpose the plane-of-action face width, F_{pa}, is subdivided onto several segments within each of them, either zero, or one, or two and so forth, lines of contact occur. In a case of line of contact in a form of a planar curve, the corresponding portion of plane-of-action face width, F_{pa}, is sliced on infinite number of infinitesimally narrow slices. A portion of a line(s) of contact within each infinitesimally narrow slice is considered a straight line segment. In such a manner, loads in geometrically-accurate parallel-axes gear pairs with the lines of contact of an arbitrary geometry can be determined.

**FORCES ACTING IN TRANSVERSE SECTION OF GEOMETRICALLY-ACCURATE PARALLEL-AXES INVOLUTE GEAR PAIR**

In addition to forces that act in the plane of action, friction forces act in transverse section of the gear pair as illustrated in Figure 8.

At an instant of time when two contact points, K_1 and K_2, are observed, the resultant tangential force, F_t, is equally shared between the points, K_1 and K_2, that is, the forces F_{t,1} and F_{t,2} are equal to one another. The forces, F_{t,1} and F_{t,2}, create the input torque, T_p.

Friction forces, F_{f,1} and F_{f,2}, act perpendicular to the line of action, LA. As the instant relative motion of the driving gear and the driven pinion is an instant rotation about the pitch point, P, and the contact points, K_1 and K_2, are located from the opposite sides of the pitch point, the friction forces, F_{f,1} and F_{f,2}, are pointed oppositely to each other. The friction forces, F_{f,1} and F_{f,2}, are equal:

\[ F_{f,1} = \mu \cdot F_{t,1} \]  
\[ F_{f,2} = \mu \cdot F_{t,2} \]  

|Equation 20|  
|Equation 21|

Here, friction coefficient is designated as \( \mu \). Variation (due to lubrication) of magnitudes of the forces, F_{f,1} and F_{f,2}, can also produce noise.

Friction forces, F_{f,1} and F_{f,2}, create a friction torque, T_{f,friction}, that is calculated as:

\[ T_{f,friction} = F_{f,1} \cdot K_1 P + F_{f,2} \cdot K_2 P \]  

|Equation 22|

Friction torque, T_{f,friction}, is opposite to the input torque, T_p.

When the gears rotate, the distances, K_1 P and K_2 P, alter. However, the summa (K_1 P + K_2 P) remains of a constant value. Therefore, in geometrically-accurate parallel-axes involute gearing, the friction torque, T_{f,friction}, also is of a constant value.

**CONCLUSION**

Noise excitation in perfect parallel-axes gearing is the subject of the article. As it is shown in our earlier published article [1], base
pitch variation is the root cause for an excessive noise excitation in $P_n$ -- gearing. Theoretically, geometrically-accurate gearing that features zero base pitch variation promises to be noiseless, as no noise is produced by the gear mesh. In reality it can be noisy.

Variation of the gear loading along with a variation of point at which the resultant load is applied is the root cause for an excessive noise excitation in geometrically-accurate parallel-axes gearing. This results in a corresponding variation of the deformation of the gear shafts, of the gear housing, and so forth. Ultimately, perfect parallel-axes gearing can produce an excessive noise. Noise of this kind is also referred to as the "gear noise": however, it is produced not by the gear mesh, but by the other components of the gear mechanism instead: by the shafts, by bearings, as well as by the gear housing, and so forth (all of these components vibrate due to the variation of the resultant force in gear mesh. This source of noise has to be considered as a "gear noise," as control over the level of the noise can be achieved by the proper selection of design parameters of the gears: profile angle, contact (total) ratio, helix angle, and so forth.

Gears with a low tooth count (that is, LTC -- gears, for simplicity) are affected more by the load variation.

A variation of the tangential force (a) in magnitude, and (b) point (and line) of application of the resultant force can also be caused by angular oscillation of the plane of action, $PA$, around the axis of instant rotation, $P_{in}$.

In a multiple-stage gear drive, every shaft (and every gear pair) causes its own source of noise/vibration. All the vibrations superpose each other.

The disclosed approach can be used for intersected-axes gearing ($I_a$ -- gearing, for simplicity), as well as for crossed-axes gearing ($C_a$ -- gearing, for simplicity).

REFERENCES

FOOTNOTES
1 Geometrically-accurate gears are those that meet three fundamental laws of gearing [2]. Gears only of this particular kind are capable of transmitting a uniform rotation of an input shaft to a uniform rotation of an output shaft, that is, with a constant angular velocity ratio.

2 It is important to stress here that the gear teeth loading strongly depends on the total length of the line of contact ($TLLC$), as well as of the contact ratio in the gear pair. The influence of the total length of the line of contact, and of the contact ratio on the gear teeth loading is complex: It could happen that when the pitch helix angle of a gear, $\gamma_g$, goes up, the total length of the lines of contact goes down, that is, the contact ratio can be greater in this case, but the total length of the line of contact can be shorter. The latter results in a higher contact stress, and a higher bending stress in the gear teeth.
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USING SIMULATION TO EVALUATE PART GREEN SHAPE TO REDUCE DISTORTION DURING PLUG QUENCHING
The commercially available heat-treatment simulation software, DANTE, has been used successfully to improve heat-treat processes and steel-part characteristics; this case study follows an example of applying the software to evaluate a real-world challenge.

By JUSTIN SIMS, CHARLIE LI, and B. LYNN FERGUSON

Determining the heat-treatment geometry (green shape) for complex gears is not a trivial task. Mass distribution becomes critical in determining the uniformity of heat removal, and subsequently, the uniformity of solid-state phase transformations. The uniformity of transformation directly affects the distortion after quenching. For decades, distortion control during heat treating has been based on experience. However, simulation of complex processes and geometries is now feasible due to improved material models, improved computational power, and the reduced cost of computer hardware [1]. Heat-treatment processes are no longer black boxes but become transparent and pliable with the use of heat-treatment software.

The commercially available heat-treatment simulation software, DANTE, has been used successfully to improve heat-treat processes and steel-part characteristics [1-3]. The following is an example of applying DANTE to evaluate a real-world challenge. Several green shapes are explored in an effort to reduce distortion of a complex gear shape that is quenched in oil using a plug. The ability to simulate the effects of quenching on the gear shape quickly, cheaply, and accurately cannot be overstated. The time and cost to perform physical trials on various gear shapes would be a daunting task. Simulation can help guide the physical trials, substantially reducing the total number.

HEAT-TREATMENT PROCESS MODELING

Accurately modeling the heat-treatment process requires the solution to several physical phenomena: mass diffusion for the carburization process, heat transfer for heating and cooling processes, stress and strain for the prediction of deformation and residual stress, and solid-state phase transformations for microstructural evolution predictions. The commercially available heat-treatment simulation software DANTE accounts for all of these phenomena [4]. These models, resident in DANTE, exist as libraries that link with commercial finite element packages ANSYS Mechanical or ABAQUS/STANDARD. Necessary process data, defined as heat-transfer coefficients as functions of part-surface temperature, are contained in supplied databases to describe the heating and cooling processes. The effects of the oil-fill rate also can be considered, as they are here.

COMPONENT GEOMETRY

An internal gear with a spline and a smooth outer diameter was suffering from excessive radial growth and bow distortion of the spline during a plug quenching operation. A redesign of the green shape was proposed. DANTE heat-treatment simulation software was used to evaluate several different green shapes and how the green shape affects the distortion modes and magnitudes of the spline. The gear was simplified for this initial study by replacing the spline and teeth with an equivalent thermal mass, as shown in Figure 1. This allowed the model to be reduced to 2D axisymmetric for quicker computation times and the evaluation of many geometric changes in a short period of time. A 3D model would be needed to validate the 2D results, but it is not discussed here. The component has a 37 mm OD, 5.7 mm wall thickness, and 90 mm height. The wall thickness at the spline and gear teeth is 8.7 mm. There is also a flange at one end that adds to the difficulty in controlling the distortion during quenching. The component is made from AISI 9310.

The four main geometries evaluated and discussed in this article are shown in Figure 2. From left to right:

- The original geometry described in Figure 1.
- A geometry with 2 mm added to the entire outer diameter, increasing the total mass to reduce the overall cooling rate.
- A geometry with 2 mm added to the outer diam-
A geometry that removed material near the spline, making the cross-section more uniform throughout the part.

These geometry modifications were for the initial trial and should not be considered optimized for each case. Once a geometry proves better than the current green shape, further optimization can be performed. This optimization was not considered in this study.

HEAT TREAT PROCESS CONDITIONS
Selective carburization was performed on the spline only and was the same for each geometry modification, as shown in Figure 3. The component has a surface carbon value of 0.86 percent and an effective case depth of 0.4 mm.

The quenching process consists of several steps, as shown in Table 1. All steps were modeled and consist of:
- Austenitize at 830°C for 1 hour.
- Transfer from the heating furnace to the quench vessel in 7 seconds.
- Immerse the part into the 55°C oil.
- Hold for 1 sec before oil flow begins.
- Flow the oil to quench the component.
- Cool to room temperature after removing the part from the oil.

The 400°C transfer temperature is due to the close proximity of the hot-fixture mass, which acts to substantially heat the air around the part during transfer. It should be noted that during the immersion and hold processes, the plug blocks oil from contacting the spline and traveling past it up the bore. This has an impact on the temperature profile when the quenching starts, and hence, the transformation timings.

Figure 4 shows contour plots of temperature in Celsius at the end of each step in the process: heating, transfer, immersion, quench hold, quench, and cool to room temperature. Of particular note, the temperature at the tip of the flange after the transfer from the furnace to the quench vessel has dropped below 800°C. After the 1-second hold, the tip of the flange has dropped below 400°C, while the spline is still more than 800°C. The martensite start temperature for 9310 is approximately 425°C. Martensite has started to form in the flange before the oil flow even begins. Once the oil flow is initiated, the oil is able to flow between the component and the plug.

GREEN SHAPE SENSITIVITY ANALYSIS
The bow distortion and radial growth of the spline were the main concerns, with the radial growth taking precedent due to the component not mating to other parts properly without too much finish machining. Understanding where these distortions came from and the sensitivity of the geometry to the quenching conditions were the main goals of these models. Figure 5 shows a plot of the radial displacement of the spline for the four geometries. A radial displacement of zero is the original green-shape dimension. Positive displacement is growth, and negative displacement is shrinkage. Figure 5 shows that all four geometries experience radial growth. Figure 6
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shows contour plots of the radial displacement for the four geometries. The lower limit of the contours is set at 0.040 mm, and the upper limit is set at 0.090 mm for all geometries to better illustrate the differences between them.

Figure 5 shows that adding additional material to the outer diameter can significantly reduce the overall growth of the spline while maintaining approximately the same bow distortion. Removing material, as is the case with the scallop geometry, made the radial growth and the bow distortion worse than the original geometry. Thanks to the DANTE model, the causes leading to these outcomes can be ascertained. The radial growth differences shall be examined first.

The component is restricted at the green-size inner-diameter dimension during quenching by the plug, which means that growth will always occur since the transformation to martensite should allow the part to expand off the plug. Figure 7 shows the predicted radial displacement (solid lines) and temperature (dashed lines) histories for the four geometries during the cooling steps of the process at a single node. The node is on the surface and at mid-height of the spline. Figure 8 is a close-up of the time when the part makes contact with the plug.

As can be seen in Figure 8, the four geometries contact the plug at different times. This seemingly minor difference is ultimately the cause of the different final radial dimensions of the four geometries. An interesting point can be gleaned from Figure 8: Different geometries clearly contact the plug at different times and for different amounts of time, with the smallest growth occurring for the geometry that remains in contact with the plug for the longest period of time. The scallop geometry remains in contact for approximately
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4 seconds, the original geometry for approximately 6 seconds, and the two geometries with added mass for approximately 7 seconds. This means that, in order to keep the radial growth to a minimum, the component should remain in contact with the plug for as long as possible. Since the part appears to contact the plug at the same temperature, the slower the part can cool, the longer it will stay in contact with the plug and the smaller the overall radial growth. This is why adding material to the part near the spline helped, because it slowed the cooling rate of the spline. By adjusting the size of the green shape and the plug size, the part could be made to expand to the required dimension instead of being machined to the final shape. Heat-treatment modeling could help determine these sizes.

The bow distortion is strictly a result of the unbalanced mass in the axial direction, resulting in the top and bottom of the gear transforming to martensite before the center section. Figure 9 shows snapshots at key times for the progression of the martensite and temperature histories. All figures have a displacement magnification of 25X. Figures labeled A# are of temperature in Celsius, and those labeled B# are of martensite in volume fraction.

Figures A1 and B1 show the profiles at the end of the transfer step. A minor temperature gradient exists at this stage with no martensite transformation. The part is also straight at this time. Figures A2 and B2 show a snapshot at 1.8 seconds into the quench. At this stage, a large temperature profile exists, and the transformation to martensite has started at the very edges of the flange and at the top corner. The bow distortion shown is strictly from the thermal effects and is mostly elastic. Figures A3 and B3 show a snapshot at 8.3 seconds into quench. At this point, the martensite transformation in the flange and at the top of the part bow the part in the opposite direction and begin to straighten the spline. There is still no phase transformation happening in the core of the spline. At approximately 14 seconds into quench, Figures A4 and B4 show the martensite transformation has started in the core. The expansion in the core pushes outward and actually make the component straight. The transformation is nearly complete at the ends of the component at this time, yet is only approximately 35 percent complete in the core. The transformation to martensite is completed in the core at approximately 30 seconds, shown in Figures A5 and B5. The 65-percent martensite transformation in the core that occurred over the previous 16 seconds with the ends already completely transformed has acted to bow the component in to the final shape. There is still a small temperature gradient at this point, but it is not large enough to significantly change the shape once the component has cooled to room temperature. Figures A6 and B6 show the component at room temperature.

**CONCLUSIONS**

The analysis showed that additional material is needed to reduce the overall radial growth. The increase in mass also slightly improved the bow distortion. Removing material from the green shape, as was the case with the scallop geometry, increased the radial growth and bow distortion. The next step would be to optimize the mass needed to reduce the radial growth to manageable levels while keeping the bow distortion to a minimum. Once optimized, a 3-dimensional model would be executed to confirm the 2-dimensional modeling results. These conclusions were reached through the use of heat-treatment simulation software DANTE and were relatively inexpensive trials.

**REFERENCES**


In order to keep the radial growth to a minimum, the component should remain in contact with the plug for as long as possible. Since the part appears to contact the plug at the same temperature, the slower the part can cool, the longer it will stay in contact with the plug and the smaller the overall radial growth.

Figure 9: Contour plots of A) Temperature, in Celsius, and B) Martensite, in volume fraction, at various times during the hardening process.
FILLING SOME GAPS IN SPLINE DESIGN GUIDELINES: CENTERING, FRICTION, AND MISALIGNMENT
This paper provides an accurate method for calculating radial loads transmitted by straight-sided splines by means of the effective pressure angle calculation, enabling more accurate hoop-stress calculation for these splines.

By STEPHEN McKENNY

International spline standards such as ISO 4156, ISO 14, ANSI B92.2, and SAE J499a have detailed definition of two-dimensional spline geometry but cover only the most basic axial effects such as helix error. Other widely used published documents, such as those written by Dudley, and Cedoz & Chaplin, provide information about stresses, including some axial effects such as misalignment. Many recent studies of load distribution have been published, but a general approach is not defined. Even armed with all these documents, the engineer is not provided with adequate guidance regarding several factors that influence how a splined interface functions. Additionally, for some characteristics, small-diameter spline interfaces behave very differently from larger diameter splines.

This paper presents information that is not found — or is not satisfactorily covered — in current standards and existing papers. These include: how to calculate the effective pressure angle of straight-sided splines that is needed to accurately determine normal and radial loads; how to calculate the effective centering force of a spline pair; an update to the Dudley misalignment factors that can be applied to splines of any size; and an update to the calculation of the maximum axial force that a spline can transmit via friction.

Results from analytical studies of centering forces and misalignment factors are provided, and an experimental study of centering force is discussed.

The effective pressure angle calculation method is based on the tooth thickness and profile angle and can be included in future standards and guidelines. The centering force analysis is supported by results from prior published measurement data (Medina, Olver) and new measurements. The misalignment factors are an update to the published table (Dudley) that covers a very limited — and undefined — spline size range. The friction force calculation method is a correction to widely published (Maag, and others) formulae that do not include the cam effect of normal force.

1 INTRODUCTION

This paper presents some practical approaches to some details of spline design that are not covered in widely used spline-design guidelines or are not clearly covered in standards. The results provide insight into how splines behave in non-dynamic conditions.

All splines described in this paper have a side fit with positive backlash and consist of a shaft and a hub as the interfacing parts. Tip-to-root clearances are larger than the backlash and form-clearance is large enough to prevent tip-to-root fillet contact.

Straight-sided splines come in many varieties, from parallel straight-sided splines defined by ISO14 [1] to non-parallel straight-sided splines that are not defined by an industry standard. Square and hex drives can even be considered and analyzed as splines. ISO 14 defines geometry but does not describe how to determine the radial loads associated with transmitting torque. A definition of, and calculations for, the effective pressure angle of a straight-sided spline is provided, and backlash effects are discussed to enable effective design and analysis of these splines.

The radial centering force provided by both involute and straight-sided splines is studied, including the behavior of off-center rotation. A description of the centerline motion during off-center rotation is provided. The effect of unequal tooth loading on centering for a hub with an offset load is explored. An update is provided for the formula for centering torque to include the effects of the shaft-rotation angle.

Angular misalignment affects spline contact width and stresses. A study of misalignment shows additional factors beyond those described by Dudley [2] in his load-distribution factor table are important.

Spline torque lock is discussed, and a formula is provided that considers the cam effect of the pressure angle, friction, and applied-axial load.

2 EFFECTIVE PRESSURE ANGLE OF STRAIGHT SIDED SPLINES

Anyone who has had the experience of a flat-head screwdriver slipping out of the screw head slot while tightening the screw should be able to appreciate the importance of having the driving and driven members rotate on the same center while transmitting torque. We call this “centering.” Since external splines essentially are a set of integral keys in a shaft, and internal splines are the mating set of keyways in the bore of the mating hub, a flat-head screwdriver can be considered a small diameter shaft with two radially tall straight-sided spline teeth. A Phillips-head screwdriver can be considered a four-tooth splined shaft with parallel straight-sided splines. The Phillips-head design ensures centering of the drive torque on the screw axis. These screwdriver examples both have an unusual characteristic for a spline: The minor diameter is nearly equal to the tooth thickness. Continuing with fasteners, square and hex shapes often used in bolts and nuts (as well as...
other polygon drives) also can be considered variants of splines. In the square and hexagon examples, the minor diameter is tangent to the tooth flanks.

A common characteristic of straight-sided splines, whether parallel straight-sided splines as defined by ISO14 or non-parallel-sided flat-tooth splines that have a non-zero included angle between drive and coast flanks, is that they do not have a defined pitch diameter that is useful for calculating tangential and radial forces. Another characteristic is that, although the external and internal tooth geometry is defined without regard to backlash, the contact zone is highly dependent on backlash. These splines have a contact zone that diminishes (becoming more concentrated toward the major diameter) as backlash increases. As the profile contact zone decreases, the effective pitch radius used for force calculations increases — whether that is considered the center of the profile contact zone or the location of mean compressive stress in the profile pressure distribution. The profile-contact zone size and effective pitch radius also are affected by the tip radius.

### 2.1 NOMINAL PROFILE ANGLE FOR STRAIGHT-SIDED SPLINES
The nominal profile angle for various examples of straight-sided splines is defined as the half-included angle from the center of the external tooth to the flank of the tooth. The same angle exists in the mating internal spline between the center of the tooth space and the tooth flank. Parallel straight-sided splines have a zero-degree profile angle. Figure 1 contains examples of straight-sided splines and polygon drives showing their nominal profile angles: a) a parallel straight-sided spline (one tooth shown) such as those defined by ISO14, b) a non-parallel straight sided spline, c) a square drive, and d) a hex drive. Although square and hex drives are not commonly thought of as splines, they function essentially as straight-sided splines since the drive faces are an integrated set of driving features and the width of the contact zone in the profile direction is determined by the backlash with the mating part.

### 2.2 EFFECTIVE PRESSURE ANGLE FOR STRAIGHT-SIDED SPLINES
Once a reference diameter is selected for a straight-sided spline, whether it is the mid-tooth height or an arbitrary diameter within the contact zone, tangential force can be calculated. But to calculate the normal tooth force and its radial component correctly, the effective pressure angle, not the nominal profile angle, must be used. The radial component of the normal force is important for understanding hoop stress and radial deflections of thin wall parts.

The effective pressure angle is the sum of the nominal profile angle (zero for parallel straight-sided splines) and the half-tooth thickness in degrees (Figure 2). For the effective-pressure angle of the internal spline space, substitute the space width in place of the tooth thickness since the drive faces are an integrated set of driving features and the width of the contact zone in the profile direction is determined by the backlash with the mating part.

### 2.3 MISMATCH IN EFFECTIVE PRESSURE ANGLE
Since straight-sided spline geometry is defined by the profile angle (half-included angle) and width of the tooth or space, backlash (difference between internal-tooth space width and external-tooth thickness)

---

**Figure 1**: Straight sided splines and polygons with normal force vectors.

**Figure 2**: Profile angle $\alpha$ and effective pressure angle $\alpha_e$ of a straight sided spline.

**Figure 3**: Mismatch in effective pressure angle between internal and external splines.
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angle and friction coefficient. Straight-sided splines, however, due to magnitude to an involute spline that has a similar effective pressure spline. Thus, straight-sided splines should provide centering similar in force, resulting in unequal tooth load sharing, as with an involute is the same — an applied radial load will be supported by a centering gram analysis of a straight-sided spline system would show the physics so, whether the amount differs from involute splines. A free body dia-

Since the normal force on the external tooth must be of the same magnitude and angle as the corresponding normal force on the mat-
ing internal tooth, this difference in angle and its effect on the normal force must be properly considered. In a straight-sided spline with a small diameter and large backlash, the difference between using the external-tooth thickness vs. internal-space width to calculate the effective pressure angle, and therefore the normal force, is greatest. Since the external spline flank-to-tip radius likely contacts the flat flank of the internal spline, it is most appropriate to use the internal-spline space width for the calculation. Using the space width results in a slightly larger normal force, so this should be conservative.

For small diameter splines, this mismatch in angles causes a small profile-contact zone, so careful attention should be paid to the back-
lash specification for small-diameter straight-sided splines.

3 RADIAL CENTERING FORCE OF A SPLINE INTERFACE UNDER TORQUE LOADING

Involute splines are commonly understood to provide a centering force — but there are few references that provide comprehensive information regarding the amount of centering and the behavior of off-center motion. ISO 4156-1 [3] (Figure 3, page 12) shows that the centering force at any tooth is the radial component of the normal tooth force, which suggests that a spline carrying an applied radial load from imbalance or the weight of a supported hub results in unequal tooth forces (unequal load sharing). A free body diagram leads to the same conclusion. There are few, if any, explanations for whether straight (e.g. parallel straight-sided splines) provide any centering force, and if so, whether the amount differs from involute splines. A free body dia-

Guo, et al. [4] provide a formula for restoring torque, which is the torque required to restore the supported part to the center of rotation. (In their system, the hubs on each end of the shaft are supported with bearings, and the shaft is supported by the hub splines.) Their for-

The center of the hub is not static — in view “a” it is below the center of shaft rotation. In view “b” it has slid to the left and below the shaft center. In view “c” it is again below the center. In views “d” through “f” it is below and to the right of the center. When the rotation reaches an angle at which the friction force is overcome by the hub weight, the hub will slide from its position in view “f” to a position similar to view “b” (but with the next tooth in the upper right quadrant). This pattern of motion indicates that there is more off-center hub motion normal to the direction of load (horizontal in this case) than motion aligned with the direction of load. The hub centerline moves in an arc from view “b” to view “f” (while tooth #1 has both drive and coast-side contact) and their tendency to have edge contact, may have a higher friction force than an involute spline.

3.1 BEHAVIOR OF A HUB ROTATING OFF-CENTER

We return to the square drive as a simple spline system to investigate the behavior of a splined hub as it rotates off center due to its own weight on a shaft. This evaluation assumes motion is slow enough that there are no dynamic effects from hub inertia. Figure 4 shows several time steps of rotation. The hub has substantial backlash, allowing the motion of its centerline to be visualized. As the prior section demonstrated, this square drive is essentially a straight-sided spline with 45-degree nominal profile angle. In each time step, the shaft (black square) defines the center of rotation. The brown square represents the hub internal spline. The hub is forced downward by its own weight. Backlash is represented by clearance between the two squares. The center of the hub is a brown dot near, but not on, the center of rotation.

One quadrant has the number “1” in it to show shaft rotation in the counter-clockwise direction. Normal forces in the positive torque direction on the hub are in black. Reaction forces on the coast side of the tooth are in red. Drive flank forces that support the hub weight are higher in magnitude than those that do not (e.g. in view “f” the normal force shown in the upper left quadrant only contributes to hub rotation. The center of the hub is not static — in view “a” it is below the center of shaft rotation. In view “b” it has slid to the left and below the shaft center. In view “c” it is again below the center. In views “d” through “f” it is below and to the right of the center. When the rotation reaches an angle at which the friction force is overcome by the hub weight, the hub will slide from its position in view “f” to a position similar to view “b” (but with the next tooth in the upper right quadrant). This pattern of motion indicates that there is more off-center hub motion normal to the direction of load (horizontal in this case) than motion aligned with the direction of load. The hub centerline moves in an arc from view “b” to view “f” (while tooth #1 has both drive and coast-side contact) and
then slides to contact the next tooth. This cycle of rotation followed by sliding occurs once per pitch rotation. The amount of centerline motion and sliding can be reduced by increasing the number of teeth or reducing backlash (or both). Once the torque is sufficient to lift the weight of the hub at all rotation angles (Figure 5), the rotation will be smooth since the hub spline is centered on the shaft spline, which is the center of rotation, and all drive flanks will be in contact.

Since the rotation of the hub is affected by friction (i.e., higher friction would cause it to slide at a larger angle than low friction) any comprehensive model of centering behavior would need to include friction.

### 3.2 TORQUE REQUIRED TO PROVIDE CENTERING

There may be an intermediate state during which the hub is lifted and dropped depending on the rotation angle of the spline. To investigate this potential intermediate state, consider the three shaft rotations in Figure 5 at the instant when the hub is lifted. In view “a” there is one tooth carrying the weight of the hub, and the force shown at tooth 1 is the vertical force reacting the weight (and providing all the torque). In view “b,” which is rotated 45 degrees from view “a,” there are two teeth in contact, and we can determine from the sum of forces in each direction that they carry equal load. The sum of the vertical components of their normal force will support the hub weight. In view “c” the vertical component of the normal force from tooth 1 will only carry a small fraction of the hub weight, so the next tooth carries most of it. Table 1 shows an example of a square drive (45-degree nominal profile angle spline) with tooth forces and torques that are required at several rotation angles through one pitch rotation to support the weight of a hub. The tooth load sharing (second and third columns) assumes tooth loads rise and fall in a sinusoidal pattern through 180 degrees of rotation, reaching the maximum load at the rotation angle where the normal force vector is opposite the direction of the hub weight. Since there is not a single torque value that lifts the same weight at all rotation angles, it is evident that if the torque is at a constant value between the minimum and maximum values in the table, the hub will lift and fall during each pitch rotation. Since this spline varies from one to two drive flanks in contact during off-center rotation and has 90 degrees of shaft rotation per pitch, it is an extreme case intended to demonstrate the phenomenon of slow-speed operation during the transition to (or from) centering. More typical splines used to support hubs would be expected to have a smaller torque range during the transition from onset of lift to the onset of complete centering.

Based on this analysis, the torque to provide centering is expressed by Equation 3, which is a modification of the formula provided by Guo, et al. [4]. In this formula, coast-flank forces should be zero, since the formula describes the onset of centering, and there are no drive forces in the same direction of the hub force (vertically downward in this case since the hub force is due to its own weight). Only drive flank forces that oppose the hub force are non-zero.

\[
T = \frac{\sum_{i=1}^{j} T_i \cos(\gamma_i - \alpha_e)}{2000 \cos \alpha_e}
\]

where

- \( T \) = shaft torque (Nm).
- \( T_i/T \) = fraction of load carried by drive flank of tooth \( i \).
- \( w \) = radial load (N).
- \( d \) = pitch diameter (mm)
- \( i \) = tooth number.
- \( j \) = number of teeth in 180 degrees.
- \( \gamma_i \) = shaft angle at tooth \( i \) contact point with hub, with zero defined as normal to the direction of the hub offset force (see Figure 6), (degrees).
- \( \alpha_e \) = effective pressure angle (degrees).

This formula can be used to determine the centering torque with any number of lifting teeth (up to the limit of half of the teeth lifting the hub), and to plot the theoretical lifting force vs. shaft-rotation angle. Figure 6 shows an involute spline with the tooth normal forces that contribute to centering (supporting the weight of the hub). The horizontal components of these tooth forces sum to zero net horizontal forces.
force, and the sum of tangential tooth forces times the pitch radius equals the shaft torque (and is opposite to the hub reaction torque). At the onset of centering, the sum of the vertical components of the lifting tooth forces equals the hub weight.

3.3 EXPERIMENTAL MEASUREMENT
When conducting fully-reversed or partially-reversed torsional testing of shafts with splines that have backlash, it is common to see a rotation discontinuity in the output compared to the sine wave torque vs. time input (Figure 7). The cause of this discontinuity is that the shaft has free motion through backlash, which takes a finite amount of time, and occurs when the torque changes from positive to negative and vice versa. During this transition, the shaft, which is the part that needs to have its weight supported by the test fixtures, falls off-center until the torque is sufficient to lift it again.

A measurement activity was undertaken to determine the torque required to center a shaft with an applied radial load near the spline at one end. The spline for this shaft has 21 teeth, 0.75 mm module, 30-degree pressure angle.

Testing was done on an MTS servo hydraulic rotary actuator with a controller running MTS 793 software with 100 Hz data collection and a rotary displacement transducer. Radial load was applied by hanging a known weight on one end of the shaft using a ball bearing to minimize rotational friction. Dial indicators were used as a secondary measure when the weighted shaft was centered or off-center. The test set-up is shown in Figure 8. Early in the testing, it became clear the calibration of the torque meter was not at a low enough torque to be able to detect the onset of centering with sufficient accuracy that the analytical model could be confirmed or rejected. Figure 9 shows a magnified trace of the transition from negative to positive torque, showing the difference between commanded torque from the actuator (black dashed line) and measured torque at the fixed end of the test stand (solid blue line). There appear to be two sources of the difference: a relatively stable offset due to wind-up of the system and the discontinuity both before and after passing through zero torque.

4 MISALIGNMENT LOAD FACTORS
Different methods are used to account for how misalignment affects spline stress. Some people use a measurement of the observed contact pattern — or an analytical estimate of it — and calculate contact pressure over the face width that seems to be in contact. A review of many published papers found that several authors present examples of stress vs. face width for splines under load, in some cases just from torque loading and in other cases including misalignment. Analytical studies were done by Volfson [5], Medina et al. [6, 7], Hong, et al. [8], Guo, et al. [4]; and experimental measurements by Dudley [2, 9], and Hong, et al. [8]. The stress vs. face width curves generally follow an exponential slope, and the maximum stress often is greater than three times the average value, but rarely greater than five. Dudley [2] provides a load-distribution factor table that indicates the stress increases as a function of two factors: misalignment and face width. Dudley's load factor table covers a limited face width range of about 12 mm to 100 mm, and he does not indicate what pitch diameter, module, or pressure angle he used. A plot of the data from his chart is shown in Figure 10. The data follows a power function.

4.1 ANALYTICAL STUDY USING SPLINE LDP
An analytical study was done using Spline LDP to check whether splines with a range of diameters exhibit the same pattern of misalignment load factors vs. face width that Dudley published. Another purpose of the study was to determine whether this tool could be useful for estimating misalignment load factors. Several splines were evaluated: three pitch diameters and two length/diameter ratios. All are 1.0 module involute splines with a 30-degree pressure angle. The results of the study are in Table 2 and Figure 11. The data falls into two patterns: splines with a length/diameter ratio of 1.0 follow a power function (top three curves). Data with length/diameter ratio 0.2 are nearly linear over the range studied (lower three curves). It is apparent from the different slopes and shapes of the curves that L/D is a relevant factor, and diameter may be a relevant factor. Spline LDP appears to be a useful tool for estimating load distribution factors based on stress.
We have all heard the phrase *WORK SMARTER, NOT HARDER.* Makes sense, right? In times of economic uncertainty, it’s SMART to maximize the efficiency of every one of your resources. Workholding technology that allows you to go from O.D. to I.D. to 3-jaw clamping in a matter of seconds without readjustment can maximize the production—and the profits—of your existing machines. Now that is WORKING SMARTER.

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**Booth 3203**
A series of shaft splines was designed to enable comparison of the effects of pitch diameter and face width. The basic spline designs are all 1.0 module and have three different pitch diameters. The nominal contact pressure based on a simple force/area calculation with uniform pressure was set at 360 MPa for the designs with 1.0 length/diameter ratio. Spline LDP was used to determine the peak stress for the baseline designs and for designs with 0.2 length/diameter ratio. The misalignment load factors were calculated based on the peak contact stress in the misaligned condition vs. the condition with perfect alignment (which included the non-uniform stress caused by torsional stiffness difference between the shaft and hub). The contact stress values are highly influenced by edge effects: For aligned splines, there is a stress peak at the inner contact diameter (corresponding to the internal spline minor diameter), and it is concentrated at one end. A 5-micron profile crown was added to the base designs to attempt to duplicate the stress profiles shown by Hong.

4.2 USING SPLINE LDP
Any finite element-based stress calculation method produces a very different stress value than the traditional force/area calculations available in Dudley’s time, so the specific stress values from Spline LDP cannot be compared to Dudley allowable stresses.

5 THRUST LOAD FROM SPLINE TORQUE LOCK
Crease [10] discusses axial tooth forces transmitted by a spline due to misalignment, axial sliding, and a combination of the two. He describes conditions such as small misalignments causing sliding, which result in effectively lower global-friction coefficients, and conditions such as perfect alignment causing lubricant to be pushed out of the interface, which results in higher global-friction coefficients. The paper includes a set of formulas for tooth friction force — but not of the interface, which results in effectively lower global-friction coefficients, and conditions such as perfect alignment causing lubricant to be pushed out of the interface, which results in higher global-friction coefficients. The paper includes a set of formulas for tooth friction force — but not a clear formula for total shaft axial force due to spline torque lock.

Transmitted thrust force from spline-torque lock can be from two phenomena: transmittal of an applied axial force on one part to the mating part through a spline or axial force generated in the spline coupling interface due to misalignment. In the first case, the force transmitted is the smaller of two values: the axial force due to misalignment, axial sliding, and a combination of the two. He describes conditions such as small misalignments causing sliding, which result in effectively lower global-friction coefficients, and conditions such as perfect alignment causing lubricant to be pushed out of the interface, which results in higher global-friction coefficients. The paper includes a set of formulas for tooth friction force — but not a clear formula for total shaft axial force due to spline torque lock.

The coefficient is static until motion occurs, and it is dynamic during sliding motion. If a spline has angular misalignment or a centerline offset that causes constant sliding during shaft rotation, the dynamic coefficient of friction (or an even smaller value per Crease) would be most appropriate. If the spline is static or rotating in very good alignment, the static coefficient (or possibly even the static-friction coefficient for dry contact) should be used. In either case, since friction coefficients vary depending on many application factors such as surface texture, lubrication type, lubrication amount, contact stress, temperature, and vibration, the user is advised to measure the friction force and then back-calculate an observed friction coefficient, then select a suitably conservative coefficient to use for future calculations of that system. In many applications, the user only needs to know the maximum friction force (the maximum axial force that a spline can transmit) so other parts of the system can be designed. In this case, a conservative (high) value for the friction coefficient can be used.

For an involute spline, the normal force is calculated from the torque, pitch diameter, and pressure angle. Along the profile contact zone, there is a range of pressure angles. In splines with a small number of teeth, the pressure angle at the tip can be multiple degrees of roll.
larger than at the pitch diameter. A calculus approach could be used to integrate loads along the profile to get a more accurate prediction, but this typically is not done since the friction coefficient has such a large uncertainty that the error from using the nominal pressure angle is not noticed.

For a straight-sided spline, the normal force calculation should be based on the effective pressure angle (a parallel sided spline per ISO 14 with six teeth has an effective pressure angle of approximately 13 degrees, not zero degrees).

6 CONCLUSION

Calculations for the effective pressure angle of straight-sided splines are established. Two formulas are provided: one for splines defined by a chordal tooth thickness and another for those defined by circumferential tooth thickness. This effective pressure angle, rather than the half included angle, should be used to calculate the normal force on the tooth. The effective pressure angle for the internal spline tooth space, rather than the external tooth, should be used to determine normal forces for both internal and external teeth if the backlash is large and/or the spline diameter is small. Square, hex, and other polygon drives can be analyzed as though they are straight-sided spline teeth if the contact width is known or can be estimated.

Centering behavior, even without considering dynamic effects at high speed, is complex. While rotating off-center, the center of the hub has a predictable, but irregular motion affected by friction. During the transition from eccentric rotation to centered rotation, the hub may oscillate between being centered and not centered until the torque is sufficient to maintain its center through all rotation angles. Straight-sided splines provide centering fundamentally similar to involute splines of a similar effective pressure angle.

Misalignment load factors are not simply a function of face width, they also are affected by spline-length-to-diameter ratio and probably by other factors. They can be estimated using tools such as Spline LDP.

Spline LDP is useful for comparative stress studies but not for absolute stress values, since it does not always produce the correct geometry and does not have the flexibility to adjust the hub constraints.

A formula for spline-torque lock is provided, which includes the cam effect of the spline pressure angle.

This paper provides an accurate method for calculating radial loads transmitted by straight sided splines by means of the effective pressure angle calculation. This enables more accurate hoop stress calculation for these splines. The centering-torque formula provides a useful calculation to determine the shaft torque required to center a mating hub without needing to know how many teeth are in contact. The method suggested here for determining misalignment factors covers splines in a wide range of sizes and L/D ratios.

Dudley’s table of misalignment factors is shown to under-predict the misalignment factor for some long L/D splines. Finally, the torque-lock formula provides a calculation of the maximum thrust that can be transmitted through a spline interface. This can be important for designing mating parts that react the thrust load transmitted through the spline.

The author would like to thank Ying Wang for running many Spline LDP iterations and Michelle Kehrig, Mark Owen, and Gary Eure for their creativity and assistance with the torsional bench testing.

BIBLIOGRAPHY


ABOUT THE AUTHORS

Stephen McKenny is a technical fellow at General Motors with 36 years of experience in transmission design. He has worked on automatic, manual, hybrid, CVT, DCT, and electric drive transmissions. For 15 years, he led the Gear Systems Group at General Motors and is GM’s global technical specialist for shafts and splines.

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COMPONENT DESIGN FOR FUNCTIONAL & COMMERCIAL OPTIMIZATION

Sona BLW’s precision forged components provide the optimum combination of the functional requirements that modern high-performance drivelines place on component design and safety, while maximizing quality and minimizing cost (Courtesy: Sona BLW)
With advanced engineering at the core of its forging and gear divisions, Sona BLW is constantly tackling demands while developing and implementing solutions for its customers.

By KENNETH CARTER, Gear Solutions editor

Sona BLW CEO Sunjay Kapur proudly says his father, the founding chairman, had gears in his blood.

And for a company whose offerings include the sale of 60 million gears and more across the globe, that statement might be the perfect way to describe the force behind the company’s worldwide success.

“Today, we make 147 types of gears,” Kapur said. “We build our own dies and tooling fixtures. And we’re very technology oriented.”

Products Sona offers include high-strength, ready-to-assemble bevel gears for automobile applications, differential assemblies for automobile axles with precision forged bevel gears and pinions, a wide range of driveline components for transmission and axle assembly applications, driveline components for non-automobile applications, rotavator gears, and more.

“We have the capability to design gears,” Kapur said. “And this is something that we’ve developed over the years.”

GLOBAL FOOTPRINT

Sona has eight manufacturing plants: four in India, three in Germany and one in Hungary. The company employs more than 2,700 people worldwide, and it has 100 engineers in R&D who have created 12 active patents.

Sona’s main product focus includes its differential bevel gears, differential assemblies and synchronizer rings, as well as speed gears with finish-forged and back-tapered dog toothings, reverse gears, and reverse-idler gears.

This extensive product range is supplemented by Hatebur-forgings, cold-extruded parts, steering knuckles, and sophisticated precision-forged parts for the truck industry.

The company’s precision forged components provide the optimum combination of the functional requirements that modern high-performance drivelines place on component design and safety, while maximizing quality and minimizing cost, according to Kapur.

“Our main goals consist of more than just the serial production of transmission components,” he said. “Component design and optimization with regard to function and costs are also essential for fulfilling our customers’ needs.”

WARM FORGED BEVEL GEARS

“Our mainstay product was warm forged bevel gears and we control 12 percent of the global market,” he said. “We supply to car manufacturers – from passenger cars, to commercial vehicles as well as off highway vehicles.”

With Sona’s technology, the company offers a combination of hot- and cold-forging processes that permit the manufacturing of high-strength components with complex functional elements that are ready-for-assembly. Sona develops these components while working closely with its customers, according to Kapur.
always been at the core of Sona BLW’s forging division, according to Kapur. The company’s engineering team is constantly tackling demands while developing and implementing solutions for its customers. And it does this while searching for new techniques implemented in the design and manufacture of complex, high-strength forgings.

Sona’s products have excellent strength properties, which are necessary in modern-day drivetrains, especially highly-stressed transmissions and axles.

As the company has diversified, Sona has been able to branch out in many directions beyond gears, according to Kapur.

“We’ve been able to forge a lot of products that are traditionally being cut,” he said. “Therefore, we’ve been able to move from being a pure bevel gear maker to a differential assembly company. We’ve also been able to develop a lighter differential by welding the differential as opposed to using bolts. So, we’ve done a lot of R&D in terms of end value.”

A FAMILY BUSINESS
With all the multi-faceted design, manufacturing, and development going on at Sona, it might be hard to believe that the whole enterprise is still very much a family business.

Before Kapur’s father, Dr. Surinder Kapur, passed away in 2015, the two of them ran the entire business together, but Sona’s journey actually began about 30 years ago.

“We started with a joint venture in the steering business,” Kapur said. “When Suzuki came to India, it created an opportunity to set up component businesses, which made way for the creation of joint ventures from a technology perspective. My father set up the business in 1987 and we started production in 1989.”

Dr. Surinder Kapur started this first joint venture with a steering company from Japan. It began life as Koyo Steering and is now called JTEKT.

“That’s how we started our automotive journey,” he said. “In the ’90s, we were part of several other joint ventures, one of them being...”
Sona, what today is Sona BLW. BLW invented warm-forging technology for bevel gears in Germany and licensed a few companies. One of the companies they licensed in the 1960s was Mitsubishi, and we partnered with them in this technology.”

In 1995, the Sona Group and Mitsubishi Materials signed a joint venture agreement to set up a manufacturing unit in India to produce warm forged bevel gears and synchronizer rings. The Sona Group owned 75 percent of the company while Mitsubishi Materials owned 25 percent, according to Kapur.

In the early 2000s, Dr. Surinder Kapur decided he wanted to be the largest producer (by volume) of automotive bevel gears in India. In 2008, the Sona Group acquired the forging division of ThyssenKrupp (who had earlier acquired BLW) and became the largest manufacturer of forged gears by acquisition. In 2017, the Sona Group acquired the 25 percent share of Mitsubishi Materials.

THE FUTURE OF SONA
The production of electric vehicles is where Kapur sees the future of gearing and Sona.

“There will be a big push toward electrification, and we’re ready for that,” he said. “Just the fact that we already have a healthy supply to electric vehicles makes us future ready. And we’re going to continue to see newer products. Our acquisitions in the future will also be technology oriented, whether it be in the connected space or in the electric space.”

Kapur is proud that Sona products exist in electric vehicles, and he expects that trend to continue.

“As long as we continuously work on designs, we can continuously grow the business,” he said. 

MORE INFO
www.sonablw.com
Klingelnberg gains major reduction in cylindrical gear measurement time

Klingelnberg Optical Metrology successfully combines the advantages of tactile and optical measurement in one system. With the precision of the tactile 3D Nanoscan and the speed of the optical HIspeed optoscan, Klingelnberg precision measuring centers are ideally equipped to handle all measurement tasks. Now, the development engineers at Klingelnberg have gone one step further. Through an ingenious combination of optical and tactile measurement, the total measuring time for cylindrical gear measurements can be reduced by up to 30 percent without compromising accuracy.

Decades of the system supplier’s experience have shown that tactile measurement cannot be fully replaced by an optical measurement method. That is why Klingelnberg has opted for a hybrid solution combining the best of both measurement types. A prerequisite for such a system is a rapid changeover from one method to the other. Thus, the precision measuring centers combine conventional and modern analysis methods for gearings with fast, automated sensor changeover.

The two methods are now also applied in rapid succession for cylindrical gear measurements. Specifically, this can be illustrated using an application example taken from e-mobility: For a gear with 48 teeth, the total measuring time can be reduced from 2.5 minutes to 90 seconds. To achieve this, the time-intensive measurements are performed in a very targeted manner with the optical sensors and are significantly reduced, as a result, by up to 90 percent in the ideal case. Measurement tasks in which tactile measurement presents advantages with respect to measurement time, flexibility, or accuracy are performed in turn using the tactile method.

The fact that this operational sequence has been integrated into the world-renowned Klingelnberg cylindrical gear software is particularly noteworthy. As a result, the measurement methods are combined in a fully automatic, time-optimized manner simply by making the appropriate selection. No special knowledge of optical measuring technology is required for data input and operation.

More info: www.klingelnberg.com
Hexagon showcases world class 3D scanning solutions at WESTEC 2019

Hexagon’s Manufacturing Intelligence division scheduled a display of its world class 3D scanning solutions at WESTEC 2019. Hexagon featured the new-to-market RS6 Laser Scanner and the 3D scanning laser tracker, the Leica Absolute Tracker ATS600. The RS6 Laser Scanner delivers extremely high-density point-cloud data collection at high speeds and high accuracy. With a scan line of 150 millimeters wide at mid-range, the scanner achieves a greater than 30 percent increase in surface area covered by each scan movement. Attendees got a close look at the ATS600, developed to digitally pinpoint and inspect large components and surfaces with metrology-grade accuracy, addressing parts that were previously out of the range of a traditional handheld 3D laser scanner, touch probe, or reflector. The ATS600 can accurately locate a point in 3D space within 300 microns from up to 60 meters away without the need for a reflector at the point of measurement.

Hexagon thought leader Beshoy Dauod joined the speaker roster at WESTEC 2019. He was set to give a Knowledge Bar presentation entitled “How Metrology-Grade Laser Tracker Scanning is Disrupting the Status Quo.” Dauod is a commercial solutions engineer at Hexagon. In the large-scale manufacturing realm, scanning is an area that has been ripe for disruption in terms of accuracy, timelines, usability, and more. Metrology-grade laser tracker scanning has reimagined what productive quality control should look like for manufacturers wanting to simplify problem solving and get a handle on large manufacturing backlogs.

Daoud’s presentation covered scanning laser tracker fundamentals and the unique workflows enabled by laser tracker accuracy and integrated 3D scanning capabilities. He also explained how the technology best applies to different applications such as aerostructures, transportation, shipbuilding, energy, and general assembly. The key concepts discussed in this presentation included implementing a metrology data driven closed-loop manufacturing process, transforming the current production and manufacturing workflows, and cutting production and operation costs.

MORE INFO  www.hexagon.com

Dillon Jaws’ easy-locating features reduce set-up times, aids performance

Soft jaw blanks from Dillon Jaws have an easy-locating feature which helps to reduce set-up times, aid concentric performance, provide better chuck balance, and ensure longer jaw life due to the increased accuracies. A milled surface on the jaw end, opposite to the clamping surface, serves as a location point for positioning the counter-bored...
holes and the jaws’ serrations, thus resulting in exact dimensional matching of Dillon jaw sets. As a result of having established surfaces in relation to the jaw serrations, Dillon chuck jaws consistently perform as a standard workholding component for turned or bored parts.

Available in 1018 steel and 6061 aluminum as well as 4140, 8620 and A2, brass, delrin, or stainless steel to suit virtually any CNC or manual chuck project involving turned or bored parts, they are ideal for second operation finishing work such as shafts, gear blanks, wheel hub and bearings, and more.

Dillon Manufacturing, Inc. manufactures a complete line of standard and custom workholding solutions including chuck jaws, chucks, vises jaws, soft jaws, hard jaws, collet jaw systems, chuck lubrication, and more. All products are made in the United States, and Dillon is ISO 9001:2015 registered.

MORE INFO  www.dillonmfg.com

Bevel gear calculation more precise with KISSsoft and GEMS

Calculation of scuffing for bevel and hypoid gears has been enhanced in KISSsoft Release 2019 (module ZC2, ZC9) according to the current edition of the draft ISO/DTS 10300-20:2018. For the first time, this standard uses a more precise calculation of the equivalent cylindrical gear and thus enables a significantly more accurate prediction of scuffing damage, especially for hypoid gears.
Another new feature is the calculation of flank fracture for bevel and hypoid gears according to the draft standard ISO/DTS 10300-4 (2019)(module ZZ4). The calculation is based on the approach of Dr. Jochen Witzig, FZG Munich, and determines the flank fracture risk over the entire active flank, which leads to a thorough evaluation of the teeth.

It is now also possible to export the bevel gear geometry into Gleason’s GEMS software or import data from this software (module CD3). This allows a bevel gear to be designed in KISSsoft and imported directly into GEMS for production to check the blanks, foot radii, and other parameters. The interface allows a much more efficient procedure in the design of bevel gears. If you would like to learn more about the possibilities in KISSsoft and our interface to GEMS, you can consult the company’s current documentation.

MORE INFO  www.kisssoft.ch/news

Innovative software offers monitoring for CNC transfer machines

Innovative new application software from NUM enables users of high throughput, multi-process CNC machine tools to implement process monitoring without incurring any additional hardware costs.

The software is likely to be of especial interest to users of high-end production systems such as transfer machines, where even relatively small operating issues can rapidly escalate into much larger, more expensive problems unless proactive remedial action is taken.

Through real-time monitoring of the power/current values of the electric motors on a transfer machine throughout its milling, turning or grinding processes, it is possible to minimize system downtime and maintain production quality by guarding against faults. Typically, these would include worn or damaged tools and undersize or oversize workpiece blanks.

Known as NUMmonitor, the software initially operates in “learn” mode to acquire the varying loads and drive currents of motors when the CNC machine tool is running at optimal performance levels and with a sharp new tool. Up to eight motors can be monitored simultaneously throughout the machine’s operating cycle, and the software accommodates up to 11 different error detection criteria per motor. In the case of multi-NCK systems, a further eight motors can be monitored for each additional NCK.

On transfer machines, it is generally sufficient to monitor just the load (power) of the spindle motors.

Both the level and duration of each load event that occurs during the learn cycle are measured and recorded, and the process can be repeated to obtain average values. There is no limit to the number of different load events that can be accommodated during a complete machine cycle. Minimum and maximum curves are automatically generated from the learn cycles, with the user able to define the types of error detection and the logic (for combining different error criteria, if desired).

These machine cycle-time related operating parameters form “known good” event references that can then be used for comparison purposes against data sampled during subsequent production runs. The user-programmed amplitude, duration, and inte-
NUM’s new NUMmonitor software enables users of CNC transfer machines to implement process monitoring without incurring any additional hardware costs. (Courtesy: NUM)

One of the key design aims behind NUM’s new process monitoring software was it should be entirely self-contained, so machine designers and users can implement relatively sophisticated process monitoring schemes without incurring additional hardware costs.

NUMmonitor capitalizes on the inherent flexibility of NUM’s latest-generation Flexium+ CNC platform. As standard, every Flexium+ CNC system includes a PC that can handle data from the servo drives’ measurement points, a PLC that has direct access to machine parameters, and an NCK oscilloscope feature capable of reading values in real-time. All system communications are handled by FXServer, using fast real-time Ethernet (RTE) networking.

Production can commence as soon as NUMmonitor has acquired the machine’s “known good” performance parameters. The same part program is used for both the “learning” phase and the production phase. All active values are stored in the CNC system’s solid state PC memory to facilitate fast access, while a second parameter in the part program defines precisely when each comparison should start. If a discrepancy on any of the monitored motors is detected, then a signal is sent to the PLC, which decides what action should be taken: from a simple
warning message to an emergency disen-
gagement.

The new NUMmonitor software option can be installed and used on any Flexium+ CNC system running NUM's Flexium software version 4.1.10 or higher.

NUM also intends to produce a lower cost version of NUMmonitor software, limited to two traces, for use on less complex CNC machine tools.

MORE INFO  www.num.com

Hardinge introduces Bridgeport XR1000 milling solution

Hardinge, Inc., a leading international provider of advanced metal-cutting tool solutions and accessories, offers the new and improved Bridgeport® XR1000 vertical machining center. As an upgrade to its Bridgeport vertical milling machine line, the XR1000 offers a high-quality, rugged, and powerful machining center developed for tough machining applications well-suited to meet any manufacturing challenge you might be facing.

The XR1000 is a market-leading Advanced Performance CNC Machining Center that’s creating new benchmarks for quality, productivity, and reliability, said Michael Marshall, Hardinge’s global milling product manager. “Our machining centers can produce highly accurate and precision-detailed parts helping you to improve your overall performance and competitive advantage, regardless of the sectors that you serve.”

Whether you are an OEM or a sub-contract engineering company, the upgraded XR1000 is a fully digital, high-quality machine tool designed to achieve maximum capacity and performance in today’s demanding production sectors, from OEM, aerospace, automotive, medical, mold and die, to other demanding markets.

XR1000 features include:

- **Easy and intuitive new touch screen control options**: Equipped with the optional 19” Heidenhain control with multi-touch display screen, Intel Dual Core processor and an integrated USB hub. Customized HMI design to include Hardinge guidance screens, thermal compensation screens, and option bit display screens.

- **Superior machine accuracy and repeatability with dual ballscrews**: The XR1000 comes complete with oversized high-class 45mm double nut ballscrews, fixed and pre-tensioned, and large 45mm linear high-quality linear guideways supported by six trucks on the Z axis.

- **Confident cutting**: The new XR1000 has a Big Plus, 40 taper, 12,000-rpm direct drive spindle powered by a dual-wound Heidenhain spindle motor. A quad set of 70mm angular contact bearings and a 60mm rear taper roller bearing provide superior thermal stability, significant radial and axial stiffness, and high accuracy coupled with faster cutting speeds to increase throughput, providing better reliability and meeting the requirements to run the machine more profitably.

- **Adjustable vibration suppression**: Frequency shock absorber technology helps to control and improve the low frequency vibrations, eliminate the various micro movements seen in typical C-Frame structures, extends tool life, and achieves higher surface finishes.

MORE INFO  www.hardinge.com
L. S. Starrett expands benchtop hardness testing line

The L.S. Starrett Co., a leading global manufacturer of precision measuring tools and gages, metrology systems and more, has significantly expanded its line of benchtop hardness testers, adding seven Rockwell systems, eight Vickers systems, and one Brinell system — a total of 16 new testers. “From basic analog and manual control, to advanced digital and fully automated systems, our new hardness lineup offers customers a complete and comprehensive range of solutions for any or all of their hardness testing needs,” said Emerson Leme, vice president industrial products — North America.

The new Starrett Rockwell Hardness Systems include two regular Rockwell Digital Testers, two Superficial Rockwell Testers (one dial and one digital), two Twin Rockwell-Superficial Rockwell Testers (one dial and one digital), and two Twin Rockwell-Superficial testers with a Dolphin Nose design that are fully automated digital systems with output to PC and capable of measuring 30 different Rockwell scales.

New Starrett Vickers Hardness Testers include six Micro Vickers Testers for handling a testing range of 1HV-2967HC and eight test forces, two with Digicam Basic Manual Software for manually selecting edges of indentation, two with Digicam Auto Software for automatically detecting edges of indentation, and two testers with Auto Turret control (one with basic software, one with auto software). In addition, there are two Macro Vickers Testers for handling up to 17 test forces; one featuring Digicam Basic Manual software and one featuring Digicam Automatic Software.

The new Starrett Digital Brinell Hardness Tester features automatic loading and can handle 10 scales.

MORE INFO www.starrett.com

FROM ROCKWELL TO BRINELL AND VICKERS, 16 NEW BENCHTOP HARDNESS TESTERS SYSTEMS HAVE BEEN ADDED. (COURTESY: L.S. STARRETT CO.)

From Rockwell to Brinell and Vickers, 16 new benchtop hardness testers systems have been added. (Courtesy: L.S. Starrett Co.)

Testers, (one dial and one digital), two Twin Rockwell-Superficial Rockwell Testers (one dial and one digital), and two Twin Rockwell-Superficial testers with a Dolphin Nose design that are fully automated digital systems with output to PC and capable of measuring 30 different Rockwell scales.

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The new Starrett Digital Brinell Hardness Tester features automatic loading and can handle 10 scales.

MORE INFO www.starrett.com

Vanta™ Element handheld XRF analyzer offers fast ID at an affordable price

The new Vanta™ Element X-ray Fluorescence (XRF) analyzer offers the essential features that the Vanta series is known for — speed, reliability, ruggedness, connectivity, and smartphone-like ease of use — in a cost-effective model. Easy to learn, fast to use, and weighing 2.9 pounds (1.32 kilograms), the Vanta Element analyzer is up to the challenge of all day, high-throughput testing.
The Vanta Element analyzer offers speed and ease of use in a variety of testing environments, including scrap recycling and metal manufacturing. Users can obtain clear material and grade ID in seconds and compare alloy grades on the instrument’s screen. With a dual-core processor and powered by Olympus’ proven Axon Technology™, the Vanta Element analyzer has the same high-count rate and stability as the rest of the Vanta series for rapid sorting and a fast ROI.

Built for use in demanding environments, Vanta Element analyzers are IP54 rated for resistance to dust and moisture and constructed to pass a 4-foot drop test (MIL-STD-810G) to help keep you working in case of an accidental drop or impact. For additional protection, a stainless-steel faceplate is paired with a thick (50 µm) Kapton® window that can be easily attached and removed for tool-less window changes in the field. Whether work is in hot or cold environments, users can count on the Vanta Element analyzer to perform in temperatures from -10°C to 45°C (14°F to 113°F).

The analyzer’s ruggedness features are paired with optional wireless connectivity to connect to the Olympus Scientific Cloud™ for wireless data sharing and access to convenient fleet management tools, the Olympus mobile app or your network, helping future proof it for Industry 4.0. The analyzer also has a 1 GB microSD™ card for storing results and two USB ports for easy data export. For added flexibility, the Vanta Element analyzer is compatible with accessories including the Vanta field stand, soil foot, probe shield, and holster.

**MORE INFO**
www.olympus-ims.com

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ZF Electronic launches hinged guard safety interlock switches

ZF Electronic Systems is introducing Steute’s Series ES 95 SB safety interlock switch designed expressly for monitoring hinged machine guards or access doors which must be closed to ensure safety.

Positive-break NC contacts assure circuit interruption upon opening of the guard/access door. Units feature a rugged, fiberglass-reinforced, UL94-V0 thermoplastic housing; a 4 x 90° rotatable actuating head; and IP67, EN ISO 13849-1 and cCSAus-compliance.

 Typical applications include stamping equipment, machine tools, and work cells.

**MORE INFO**
www.switches-sensors zf.com/us

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The Vanta Element analyzer offers speed and ease of use in a variety of testing environments, including scrap recycling and metal manufacturing. (Courtesy: Olympus)
From Design to Finished Gear

A visit to the Gleason booth connects you to powerful gear and transmission design tools, economical Chamfer Hobbing, hard finishing innovations like Combi Honing™ and polish grinding, in-process laser inspection as well as productive new tooling and services. All working together to connect your gear manufacturing to maximum efficiency, from design to finished gear.

“We very much look forward to being participants and contributors this year at the MPT Expo.”

The Motion + Power Technology Expo 2019 is scheduled to hit Detroit, Michigan, October 15-17. The show will bring thousands of industry experts and insiders to the Motor City, where they will be on hand with knowledge to take gear manufacturing and more into 2020 and beyond. Gear Solutions reached out to several exhibitors and asked them to share their plans for M+PT Expo 2019. If you’re going to be at the show, be sure and stop by their booths for more information.

**Gleason Corporation**

**Booth #3400**

Christian Albrecht, Director, Global Marketing

Visitors to the Gleason booth will experience the power of Connected Systems, including design, manufacturing, and metrology, optimizing gear manufacturing at every step, from design to finished gear. The KISSsoft® gear and transmission design software interface is used to develop a better EV transmission. With GEMS® Cloud, application services are now available online, including real-time software updates and upgrades. Advanced user interfaces, Closed Loop design, and manufacturing help customers do business seamlessly and manage assets conveniently. New Genesis® 160HCD hobbing machine has the most efficient chamfer hobbing to produce any chamfer form desired, which is ideal for preparing the tooth flank of eDrive gears in advance of hard-finishing operations. Closed Loop, high accuracy, and excellent surface finish is available with the Genesis® 260GX Threaded Wheel Grinding Machine, including twist control and polish grinding. It is completely automated, with integrated in-process laser measuring. The GRSL Gear Roll Testing System with Laser Scanning revolutionizes gear measuring with double flank roll testing as well as analytical index and involute measurement on all teeth for in-process inspection in a matter of seconds. It’s also available with Lead Measurement. The 300GMSL Gear Metrology System combines standard tactile probing with laser scanning of tooth flank form, associated 3D graphics and CAD interface, which is ideally suited for gear noise detection and analysis. Digital service technologies, Gleason Fingerprint™, and Gleason Connect+ can solve service problems on the fly. All work together in a Closed Loop system so that gear development and optimization is fast, efficient, and continuous.

**Induction Tooling, Inc.**

**Booth #707**

William Stuehr, President/CEO

Experts from Induction Tooling Inc., a world leader in the design and fabrication of selective hardening (heat treating) inductors, will be available in Booth 707 to discuss: new inductor designs and fabrication, inductor repair needs, inductor redesign for increased life, quick-change inductors to reduce set-up time, research & development programs using our extensive induction lab, process troubleshooting, and metallographic lab testing requirements. Based in the Cleveland, Ohio, area, we tackle some of the most challenging problems in the industry. Stop by and let’s talk induction!

**Proto**

**Booth #4328**

Taylor Thompson, Application Scientist/Sales Representative

Proto’s exhibit will illustrate the benefits of using X-ray diffraction (XRD) to measure residual stress and retained austenite. In addition, we will highlight our diverse and comprehensive product line, from large laboratory systems to novel portable systems, showcasing the capabilities of our instruments in gathering data on safety-critical and life-limited gears and components. In order to make good engineering decisions, you need highly accurate and reliable data. Proto has been manufacturing stress systems and offering lab, process troubleshooting, and metallographic lab testing requirements. Based in the Cleveland, Ohio, area, we tackle some of the most challenging problems in the industry. Stop by and let’s talk induction!

**Nagel Precision**

**Booth # 4308**

Sanjai Keshavan, Manager of ECO Hone & Microfinish Systems Division

Nagel Precision, Inc. of Ann Arbor, Michigan, has developed a new series of flexible stone super-finishers to finish inner and outer bearing races. The machines are custom designed to meet customer specifications and can be equipped with either Siemens or Allen Bradley controls. The ultra-light finishing stone holder is mounted directly on a heavy-duty servo oscillator and can attain oscillating speeds of up to 45 Hz. Directly coupled servo part driver can reach speeds of up to 3,000 rpm. Multiple finishing heads can be mounted to achieve desired quality and output. The key features of the new machine are: ability to change over rapidly from one part type to another and minimize non-cutting idle time (part load/unload) during the machine cycle. The superfinishing heads are mounted on an X and Z servo axis for automatic positioning to accommodate part diameter and thickness change. Patent pending part drive system can handle a range of part diameters without any changeover. The new system encompasses a number of industry features to make the machine more flexible and productive. For more information, stop by our booth at Motion Power Expo 2019.
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measurement services for 30 years; our team has the experience to help you optimize part performance and make components more reliable. Our application scientists will be available at booth #4328 to consult with you about your specific residual stress and retained austenite needs and find you the best equipment for your application.

REM SURFACE ENGINEERING
BOOTH #4510
Bill Nebiolo, Sales Engineer/Product Manager

REM Surface Engineering is a proud member of AGMA and has for decades been a presence at the former AGMA Technical Conferences and now the MPT Exhibition. As the developer and services provider for isotropic superfinishing services we understand that the companies who can benefit most from the engineering advantages isotropic superfinishing provide are represented at this exhibition. The world-class nature of these companies and their personnel as well as the professional atmosphere with which the exhibition will be presented make for an ideal atmosphere for the appropriate discussions and development of the camaraderie required to be successful in our endeavors. We very much look forward to being participants and contributors this year at the MPT Expo.

STAR SU
BOOTH #4013
Mark Parillo, Director of Marketing

Star SU is excited to showcase the latest in gear manufacturing solutions at the 2019 Motion + Power Technology Expo. Star SU Booth 4013 will be presenting advanced manufacturing solutions from both machine tools and cutting tools. Our gear-cutting tool solutions will showcase their usual wide variety of gear-cutting tools, precision tool re-sharpening services, and advanced coatings from Oerlikon Balzers; along with our innovative Scudding® cutters manufactured to produce gear and spline teeth for reduced cycle times and tool costs. Addressing the fluid-power industry, we will showcase a selection of cavity/port tool solutions including one shot cavity machining. Our machine tool solutions will introduce our hobbing and shaping machine portfolio through the newly created company Samputensili CLC S.r.l. In addition, view the ground-breaking concept that totally eliminates the need for cutting oils during the grinding of gears after heat treatment with a significant reduction in the cost of consumables and a considerable improvement of environmental impact with the Samputensili SG 160 SKYGRIND — the first gear dry grinding machine in the world with a patented process.

Gear Solutions magazine will be at AGMA’s biannual gear manufacturing and technology show. We hope you’ll stop by our booth (4036) to chat with our staff and to register for our daily giveaway of a Stor-Loc toolbox. We look forward to seeing you there.
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