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First Gear Engineering & Technology
By Kenneth Carter

First Gear Engineering & Technology continues its growth in gear manufacturing to many markets, including aerospace, medical, food, industrial fields, and contract gear inspection.

Fast, Flexible Bevel Gear Cutting
From Gear Solutions staff reports

Companies interested in increasing capacity and agility to their precision gearing operations will be interested in the new Gleason bevel gear cutting, lapping, and testing machines. Here’s one company’s experience.

Numerical Thermal 3D Modeling of Plastic Gearing
By Niranjan Raghuraman, MS, Dr. Donald Houser, and Zachary H. Wright, MS

In this exercise, a numerical thermal 3D model is used to predict the surface and body temperature of spur and helical plastic gears.

Measuring Root/Flank Stress in Plastic Gears
By Dr. A. Pogacnik and Dr. S. Beermann

This paper provides an overview of the testing procedure for plastic gears according to the VDI 2736-4.
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THE ZEN OF GEAR DESIGN
Exploring the ‘hows and whys’ of choosing the proper center distance for your gearing requirements.

In this section, the premier supporter of gear manufacturing in the United States and beyond shares news of the organization’s activities, upcoming educational and training opportunities, technical meetings and seminars, standards development, and the actions of AGMA councils and committees.

19 TOOTH TIPS

Brian Dengel
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Exploring the ‘hows and whys’ of choosing the proper center distance for your gearing requirements.

20 MATERIALS MATTER

Patrick I. Anderson & Mike Burnett
ENHANCED STEEL PERFORMANCE DURING VACUUM CARBURIZING OF Gears
There are many benefits that can be gained from vacuum carburizing, but material selection is critical to producing a consistently high-quality carburized gear.

22 HOT SEAT

D. Scott MacKenzie, Ph.D., FASM
CONTAMINATION OF POLYMER QUENCHANTS IN INDUCTION HARDENING
Some think coolants or other contamination do not harm the quenchant. This perception couldn’t be farther from the truth.

American Gear Manufacturers Association
Another successful Gear Expo in Columbus

It may be December to you, but to me, it’s only been a month since I returned from Gear Expo 2017 in Columbus, Ohio.

It was another successful show filled with massive energy, fascinating presentations, and demonstrations from some amazing high-tech machines.

I met a lot of dedicated people in the gear manufacturing industry, and I hope I can count many of you among them.

If I didn’t get a chance to talk with you about editorial opportunities with Gear Solutions, then please feel free to call or email me. I’d love to chat about what you can share with our readers.

You no doubt noticed that your issue was polybagged and included our annual wall calendar for 2018. The calendar includes reminders for AGMA committee members, AGMA training dates, Gleason training dates (design/theory, cutting tools, inspection, machine operations, maintenance) as well as classes dealing with specific Gleason machines. You’ll also find dates for tradeshows of gear interest (materials selection, heat treating, quality/inspection, cutting tools, machining, windpower, etc.) and local contract manufacturing tradeshows from d2p.com (design2part). Make sure you hang it in a handy location.

Swinging back to editorial, make sure you spend some time with our December issue. You’ll find it full of interesting articles, particularly focusing on plastic gears and bevel gears.

On the subject of plastic gears, Niranjan Raghuraman and Zachary Wright from Romax Technology and Donald Houser from The Ohio State University present a technical paper on numerical thermal 3D model to predict the surface and body temperature of spur and helical plastic gears.

And from the experts at KISSsoft, they share their insight on measuring the root/flank stress in plastic gears.

On the bevel gear side, we have an article that discusses fast, flexible bevel gear cutting. Some of our regular features also spotlight First Gear Engineering & Technology in our company profile, and Tom Treuden, general manger/engineering and sales with Butler Gear Enterprises LLC, talks about what his company does for the gear industry and his role in that.

You’ll find that and much more in this issue, so in between sipping eggnog and roasting chestnuts (does anyone really do that anymore?), enjoy some interesting articles on where the gear industry is going and what’s going on to get it there.

Have a super-safe holiday season, and here’s to a prosperous new year.

Thanks for reading!

Kenneth Carter
Editor
Gear Solutions magazine
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Penta Gear Metrology receives calibration accreditation

In October, A2LA labs accredited Penta Gear Metrology ISO17025 for technical competence in the field of calibration. Penta Gear President Marvin Nicholson said the company is excited to be able to offer these services for its North American customers.

“We are hopeful this offers our customers faster turnaround on their recertification of their master gears and spline gauges,” he said. “We look forward to providing excellent service to our customers.”

Ohio-based Penta Gear Metrology provides gear measurement and inspection including contract inspection, metrology services, metrology services contract inspection, and metrology services gauge certification. It provides a wide variety of functional and bench top gauges to meet customers’ gear and spline inspection needs.

A2LA is the only independent, 501(c)(3), nonprofit, internationally recognized accreditation body in the United States that offers a full range of comprehensive laboratory and laboratory-related accreditation services and training. A2LA, established in 1978 as a nonprofit, public service membership society, is dedicated to the formal recognition of competent testing and calibration laboratories, inspection bodies, proficiency testing providers, and reference material producers.

FOR MORE INFORMATION: gearinspection.com

Hexagon CMM technology part of new Fullerton College program

Hexagon Manufacturing Intelligence announced its portable and stationary coordinate measuring machine (CMM) technologies will be integrated into a new Metrology Program offered at Fullerton College, North Orange County, in Fullerton, California. Fullerton College is one of the largest and best equipped machining trade schools in California. Starting in fall 2018, students will learn how to use an array of measurement and inspection tools including ROMER Absolute arms, laser scanners, CMMs, and more. The new curriculum will enable students to study the science of measurement and acquire job skills that are in high demand by science laboratories and industries using advanced manufacturing technologies such as aerospace, defense, automotive, medical, and power generation. Students can begin work toward a Metrology Certificate of Achievement with several compulsory courses available.

The metrology program is a natural extension of the machine technology curriculum already in place at Fullerton College. Course study will cover fundamental metrology concepts and offer hands-on usage of shop floor CMMs and portable measuring machines (PCMMs) for practical measurement and inspection operations conducted in machine shops and manufacturing cells. The ROMER articulating arm is a versatile measurement tool designed to meet the needs of almost any measurement application, whether scanning or touch-probing. This portable CMM provides the foundation for Fullerton College’s rigorous program of electives designed to prepare students for scientific research and today’s data-driven manufacturing environments. The ROMER Absolute arm with integrated RS4 laser scanner will provide the means for precision 3D data capture across a range of surfaces and applications.

“Our new metrology program will add additional certificates and skills competencies, making our students even more valuable and employable within our local industries,” said Dan O’Brien, instructor and Machine Technology Department Coordinator at Fullerton College. “Manufacturing companies are challenged with filling employment positions that utilize new digital manufacturing and measuring technologies. This program intends to help fill this critical gap.”

“Hexagon Manufacturing Intelligence is very pleased to support the new metrology program being offered next year at Fullerton College, a premier academic institution that strong-
ly supports vocational programs,” said Zvonimir Kotnik, business unit manager, portable products, Hexagon Manufacturing Intelligence. “Measurement professionals will be in very high demand for the foreseeable future. Metrology education can open many career doors and introduce students to data-driven manufacturing and other diverse applications requiring 3D data for analysis, measurement, automation alignment, visualization, and more. The ROMER Absolute Arm with Integrated Scanner is the ideal teaching toolset as industry uses this technology for point-cloud inspection, product benchmarking, reverse engineering, rapid prototyping, virtual assembly, and CNC milling.”

PER MORE INFORMATION:
HexagonMI.com or machine.fullcoll.edu

Starting in fall 2018, Fullerton College students will learn how to use an array of measurement and inspection tools, including ROMER Absolute arms, laser scanners, CMMs, and more. (Courtesy: Hexagon Manufacturing)
Auction to feature complete state-of-the-art heat-treat line

Heat Treat Equipment, in conjunction with Robert Levy Associates, will conduct an auction to sell a complete state-of-the-art heat-treat line no longer required in the continuing operations of Autoliv-Nissin Brake Systems. Autoliv-Nissin, headquartered in Yokohama, focuses on the design, development, and production of "brake control and brake apply systems" for the global light vehicle market, with manufacturing facilities in Japan, China, and the United States. The company started its operation April 1, 2016.

A C-0109 Dowa Thermotech Company TFC-80-ERT/TDR-264824 3 Stage Batch Type Heat Treat System (10/2010) is among the equipment to be auctioned. It has high heat load, high heat quench and unload, atmosphere supply panel, washer load, washer unload, temper load, temper unload, and main control panel.

Heat Treat Equipment specializes in the sale of used furnaces equipment. Robert Levy Associates Inc. specializes in creating successful monetization programs for industrial assets around the world. Robert Levy’s core industries include automotive, electronics, and manufacturing.

FOR MORE INFORMATION: heattreatequip.com and rlevyinc.com

Romax Technology offers workshop on electrified powertrains

Gain a hands-on experience of a novel analysis design solution that helps motor and/or gear design engineers determine the efficiency and durability of integrated electrified powertrain systems more quickly.

The successful design of integrated electric powertrains is built on three fundamental requirements:

- Holistic understanding of interactions within the powertrain system.
- Tools that allow engineering decisions to be made quickly throughout the development process — especially early on.
- Mutual understanding between transmission and electric machine engineers.

This Romax Technology workshop is aimed at electric vehicle powertrain engineers, transmission designers, and electric machine specialists who want to deliver better quality products more efficiently.

Through hands-on experience of a novel CAE-led design solution for electric powertrains, attendees will gain understanding of:

- System-wide issues of noise and vibration, durability, and efficiency caused by interactions between transmission and electric machine.
- The importance of a system-wide approach to design that incorporates these interactions.
- The tools available that can lead the design from the start (where engineering and design decisions have the most impact) right through to fully detailed design verification.

What: Designing Electric Powertrains for Durability and High Efficiency
Date: March 7, 2018
Location: 2025 Gateway Place Suite 390
San Jose, California 95110

FOR MORE INFORMATION: romaxtech.com/media/workshops

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ALD is a subsidiary of AMG Advanced Metallurgical Group N.V.
Ipsen USA shipments remain strong in third quarter 2017

The third quarter has proven to be yet another strong quarter for Ipsen USA with 14 vacuum furnaces shipped to seven states, as well as to Indonesia and Japan. This equipment will be used in support of the additive manufacturing, aerospace, automotive, and commercial heat-treating industries.

The shipments included:
• A vertical (bottom-loading) MetalMaster® vacuum furnace with an 84” (2,100 mm) diameter work zone and 8,000-pound (3,600 kg) load capacity.
• Four standard TITAN® vacuum furnaces with options ranging from spare parts kits and a thermocouple upgrade to the PdMetrics® predictive maintenance software platform.
• Three HIQ (horizontal internal quench) and HEQ (horizontal external quench) furnaces from the VFS® product line that will process parts for the aerospace and commercial heat-treating industries.
• Several custom-built, horizontal MetalMaster® and TurboTreater® vacuum furnaces.

Among these customers, more than half also ordered one or more spare parts kits to help minimize future downtime. Ipsen’s global ICS (Ipsen Customer Service) team provides many comprehensive aftermarket offerings, making it easy for customers to rely on them for ongoing support throughout the equipment’s life span (for any brand of furnace). This includes facilitating system installations, as well as providing startup assistance, expert training, spare parts, and maintenance programs.

FOR MORE INFORMATION: IpsenUSA.com

Klüber studies hydro lubricants as lubricants of the future

How would it be if maintenance of industrial gear units or automobile transmissions were to consist simply of topping up with mains water? If contaminations by lubricants could be eliminated without any problems using clear water? Not only would that render work in industrial plants safer and cleaner, but at a stroke, all worries regarding the sustainability of the lubricant being used would be dispelled. This was the thinking behind Klüber Lubrication’s development of a concept that has the potential to revolutionize the future of specialty lubricants. The tribology experts have succeeded in developing homogeneous lubricants with functional water contents: the Hydro Lubricants.

“Compared with the present-day oil-based lubricants, water offers numerous important functional advantages: It is sustainable, globally available, non-toxic, and non-flammable,” said Dr. Stefan Seemeyer, head of Research and Product Development at Klüber Lubrication. “As a lubricant, it had, up to now, been of limited utility, since water is subject to certain physical and biological limitations, like evaporation and freezing points, oxidation or microbiological growth. By means of additives in the lubricant or technical solutions at the component concerned, however, these limits can be shifted, and the water-specific effects rendered beneficially usable. With a water-based product concept, we’ve even been able to reduce the friction involved far enough to bring the range of ‘super lubricity’ within reach.”

Water as a deliberately selected lubricant component offers entirely new possibilities, not least with the newly acquired sheer...
diversity of raw materials. Since water-soluble substances are not usually soluble in oil, they were nearly irrelevant in classical lubricant development. With innovative lubricant components of this kind, it is now possible to achieve performance parameters previously unattainable, like very good cooling properties or energy-savings, thanks to significantly reduced friction.

The water-specific effects can likewise be rendered beneficially usable by technical solutions at the component itself. For example, evaporated water can be retained in the circuit inside an enclosed component, and even be used for cooling purposes. Another line of approach is to render usable the obvious properties of water, such as electrical conductivity or cooling effects. This opens up entirely new options for many application categories, not least in the field of e-mobility.

“The specialty lubricant of the future will have to solve hitherto unknown challenges,” said Michaela Wiesböck, group leader at Klüber Lubrication in the Research and Product Development Department. “In view of progressively more stringent expectations of lubricants in terms of performative capabilities, energy-efficiency and eco-compatibility on the one hand and the finitude of fossil raw materials on the other, the demand for innovative lubrication concepts based entirely on renewable raw materials is already becoming apparent.”

The Hydro Lubrication technology is meanwhile being used in the Klüberplus C 2 series. In this lubricant, conceived for conveyor belts, water and water-soluble oils form a homogeneous solution, which leads to improved metering of the lubricant, and thus reduces the incidence of malfunctions in the production operation. This means, however, that Hydro Lubrication technology is only just beginning to explore its potentials. Together with cooperation from different sectors, Klüber Lubrication is working on Hydro Lubricants for applications in gear units, bearings, chains, and other components.

FOR MORE INFORMATION: klueber.com

Obituary: Alan Lawley, distinguished PM industry professor

Alan Lawley, Emeritus Professor, Drexel University, died October 17 at the age of 84. A long-time friend and supporter of PM technology and the industry, Lawley made significant contributions to research and development in PM and particulate materials and guided the professional development of undergrad and graduate PM students, many of whom are working in the industry today.

Lawley received BSc, Physical Metallurgy and PhD, Metallurgy degrees from the University of Birmingham in the U.K. He had worked
at the University of Pennsylvania’s School of Metallurgical Engineering (Post-Doctoral Fellow) and at the Franklin Institute’s Solid State Research Laboratory. He joined Drexel University in 1968 where he initiated a PM program. Over the years, his PM teachings, research, and consulting activities affected the academic world, industry companies, and national laboratories as well as federal government and state agencies. While at Drexel, he was appointed Department Head/Materials Engineering and the A.W. Grosvenor Professor of Metallurgy. Hoeganaes Corporation, Cinnaminson, New Jersey, endowed a professorship in PM at Drexel where two dedicated PM laboratories were established. Lawley published more than 300 articles in archival journals, conference proceedings, and books — more than 200 of which embrace PM and particulate materials.

He was named editor-in-chief of APMI International’s International Journal of Powder Metallurgy in 1985 and served in this capacity until 2015. Lawley was co-chairman of the 1993 International Conference on Powder Metallurgy & Particulate Materials (Nashville), co-chaired the MPIF/APMI PM2008 World Congress (Washington, DC), and was a long-time member of the MPIF Technical Board. A recipient of numerous professional and societal awards, Lawley was among the first class of Fellows of APMI International (1998). Additionally, he served on APMI’s Panel of Fellows, Awards Committee and Publications Committee. He received the MPIF Distinguished Service Award in 1991.

Lawley’s distinguished career in the powder metallurgy industry was recognized in 2012 when he received MPIF’s Kempton H. Roll PM Lifetime Achievement Award.

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The actuation force is provided by a spindle and is linear. By turning the spindle, the moveable jaw is moved to the fixed jaw. The clamping force depends on the torque, and the clamping range depends on the jaws being used. High clamping forces can be achieved with low torque. Due to the fact that the unit uses tension to clamp, the bending load at the base body is minimized, making it easy to use with the whole VERO-S modular system from Schunk.

For more information: www.us.schunk.com

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Heat treatment touches the lives of everyone around the world in one form or another. From the tools and technological devices we use to the cars we drive and the planes in which we fly, heat treatment plays a role in the creation of each of these items. As such, the quality and safety of these products is of the utmost importance to both the companies that make them and the consumers that use them.

The Nadcap accreditation process and Aerospace Material Specification (AMS) standards play a key role in ensuring those utilizing special processes and heat treating parts for the Aerospace industry adhere to consistent, high-quality standards for Aerospace products.

Maintaining global quality standards can not only help ensure the safety of all who utilize these final Aerospace products, but can also help you continually improve and refine your heat treatment processes to provide all of your customers with the best product quality possible.

Many of our customers have found each Nadcap audit to be a unique process (depending on their specific Prime specifications, processes, equipment and more), but the overall path to receiving Nadcap accreditation typically involves a set number of steps.

Whether you are participating in a Nadcap audit for the first time, or going through the reaccreditation process, many have found that they are constantly refining their audit process as they keep track of what did and did not work for future reference. To assist with this multiplicity of steps and processes, here are some recommended best practices and steps for …

Read the full technical article: go.IpsenUSA.com/Nadcap-Download

Instrument Calibrations and TUS

Ipsen utilizes Performance Review Institute (PRI) along with extensive internal pyrometry training so our technicians can provide calibration and survey services that adhere to strict Nadcap requirements. Our technicians will calibrate and test your furnace’s temperature and vacuum control systems, as well as verify that those systems are operating properly.

Find out more: go.IpsenUSA.com/Calibrations
AGMA welcomed more than 3,500 attendees from 30 countries and 41 states to Gear Expo, The Drive Technology Show in Columbus, Ohio. The Fall Technical Meeting had a record attendance of 180 participants and speakers making it the largest AGMA technical meeting to date. Both the show and technical meeting reflected AGMA’s vision of innovation and global strategy.

Exhibitors brought their newest machines and products that filled the 59,500-square-foot show floor at the Columbus Convention Center. Attendees saw the latest technology in automation and IIoT as it relates to the gear industry and found new solutions to increase productivity and profit in their businesses.

"Gear Expo 2017 was a great platform for attendees to see the latest solutions in the marketplace," said Jenny Blackford, vice president of Marketing for AGMA. "We are pleased with our attendance numbers but are even more excited to hear that exhibitors were meeting quality leads for new programs and making sales."

With focused promotional efforts on social media as well as geo-fencing and geo-tagging outreach, the attendees could immediately share their experience while at the show and were put in touch with the exact supplier needed. These tactics allowed AGMA to expand its promotional audience, while also leveraging traditional media partners, and exhibitors, to get the word out.

“We offered additional ways for exhibitors to promote themselves and the show this year,” Blackford said. “Companies, attendees, and speakers were able to share their Gear Expo experience right away so those that did not come to the show were still connected to what was happening. This kind of marketing allows exhibitors to get the most out of their investment.”

The Fall Technical Meeting (FTM) was one of the greatest successes of the week. The quality of the papers and the engagement of the large audience made this year’s meeting the most attended yet. Discussions about emerging technologies and the standards process brought great interest from members and non-members for future involvement.

“AGMA focused heavily on keeping the integrity of the FTM and the papers presented,” said Amir Aboutaleb, vice president of AGMA’s Technical Division. “People know that when they come to this meeting, they will be hearing about the latest in the gear industry from the foremost technical experts from around the world. This popular event will continue to grow as AGMA continues to be content driven for the changing industry.”

Additionally, the Solutions Center schedule was filled with more than 30 presentations that mirrored the newly established Emerging Technology Pavilion hosted by AGMA. Attendees connected directly with players in the additive and IIoT industries like Jay Rogers from Local Motors and Paul Boris from Vuzix to discuss the direction of manufacturing and the potential disrupters the gear industry might face in the future.

“Thank you to all of the AGMA members and Gear Expo for a successful show. We were pleased with the growth in the buying audience,” said Matthew E. Croson, president of AGMA. “We look forward to working with our innovative members as we collaborate to grow our show to include the full gamut of power transmission solutions including: mechanical, electrical, and fluid power. It will be an exciting time to be a part of this effort in 2019 and beyond.”

Information about the 2019 tradeshow will be coming out soon, and you won’t want to miss out on the exciting event. Make sure to check in to www.agma.org to find out what is going on with AGMA and to see all the new classes and events that we will have for 2018.
AGMA awards
10 scholarships to future industry engineers

The following students with a focus in the gear industry received the AGMA Foundation Scholarship awards for the 2017-2018 academic year:

**Technical School**

JACK JAEGER  Waukesha (Wisconsin) County Technical College

**Undergraduate**

ISAAC BOWSER  Taylor University (Indiana)
NOLAN MANERS  Western Kentucky University
AGGASH SIVASOTH  University of Waterloo (Ontario, Canada)
JOSHUA SMITH  Colorado State University
MICHAEL USTES  University of Michigan, Dearborn
JACOB VANDERVEEN  Grand Valley State (Michigan)

**Graduate**

TIFFANY LIM  The Ohio State University
YUE PENG  University of North Carolina at Charlotte
MUSTAFA TURHAN  University of Waterloo (Ontario, Canada)

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Johannesburg Gauteng, South Africa

**KBE+ Inc.**
www.kbeplus.com
Fair Haven, New York

**Ranken Technical College**
www.ranken.edu
St. Louis, MO

**Sierra Nevada Corporation — Space Systems**
www.sncorp.com
Sparks, Nevada

**Toyoda Americas Corporation**
www.toyoda.com
Arlington Heights, Illinois

**Wenzel America Ltd.**
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www.boughey.com
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**AshtonTucker**
www.ashtuck.com
Fruit Heights, Utah

**C.C. Jensen**
www.ccjensen.com
Newnan, Georgia

**Durban University of Technology**
www.dut.ac.za/
Durban, South Africa

**Gamesa Energy Transmission**
www.gamesagearbox.com
Zamudio, Vizcaya, Spain

**Jemms-Cascade, Inc.**
www.jemms-cascade.com
Troy, Michigan

**Kikuchi Gear Co., Ltd**
www.kikuchigear.co.jp/en/
Ashikaga, Japan

**PAI Industries, Inc.**
www.paiindustries.com
Suwanee, Georgia
FOLLOW AGMA ON LINKEDIN
We have enjoyed connecting to many of you on social media and love seeing all the pictures and stories that our members share about their businesses and employees. We want to continue to grow our following and see all that is going on in the gear industry. Follow “American Gear Manufacturers Association” on Linkedin and see all the opportunities AGMA has to offer with education, conferences, foundation items, and emerging technology. Stay connected to AGMA so we can be there for you!

GUEST BLOGGERS NEEDED FOR AGMA’S GEARS MATTER BLOG
We have had a great response to the Gears Matter blog, and we want to keep the information flowing for our readers. We are looking for 300-400 words on any topic that might relate to the gear industry, and what better way than to have our members share their own thoughts or ideas? We have entries on foundations, communications, standards, and emerging technologies, to give you some ideas. If you are interested in becoming a guest blogger, please contact Rebecca Brinkley at brinkley@agma.org.

2018 AGMA Education Events
AGMA is excited to bring you some new education events this year that offer enhanced learning opportunities taught by some of the best experts in the gear industry. Registration opens for these events October 24, 2017

Gearbox System Design
January 23-25 | Las Vegas, Nevada

Gear Materials
February 5-7 | Clearwater Beach, Florida

Gearbox CSI
March 20-22 | Concordville, Pennsylvania

Basic Training for Gear Manufacturing
April 3-6 | Chicago, Illinois

Detailed Gear Design
May 8-10 | Detroit, Michigan

Fundamentals of Worm & Crossed Axial Helical Gearing
May 31-June 1 | Alexandria, Virginia

Epicyclic Gear Systems: Application, Design & Analysis
June 5-7 | Chicago, Illinois

Gear Failure Analysis
June 12-14 | St. Louis, Missouri

Gear Manufacturing & Inspection
July 10-12 | Ontario, Canada

Basic Training for Gear Manufacturing
September 11-14 | Chicago, Illinois

Fundamentals of Gear Design and Analysis
October 24-26 | Houston, Texas

Gear Failure Analysis
November 6-8 | St. Louis, Missouri

Steels for Gear Applications
December 5-7 | Clearwater Beach, Florida

There are some great events planned for 2018; be sure to visit www.agma.org for the full schedule.
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.

Events are open to AGMA members only. Not a member? Send an email to membership@agma.org.

DECEMBER
December 5 — Aerospace Gearing Committee Meeting — WebEx
December 5 — Enclosed Drives for Industrial Applications Committee — WebEx
December 5-7 — Working Group 2 Meeting — Shanghai, China
December 6 — Fine-Pitch Gearing Committee Meeting — WebEx
December 6-8 — Gear Failure Analysis — San Francisco, California
December 11 — Helical Gear Rating Committee Meeting — WebEx
December 12 — Cutting Tools Committee Meeting — WebEx
December 13 — Computer Programming Committee Meeting — WebEx
December 14 — Helical Gear Rating Committee — subcommittee 925 Meeting — WebEx
December 19 — Wormgearing Committee Meeting — WebEx

JANUARY
January 9 — Technical Division Executive Committee — WebEx
January 10 — Wind Turbine Gear Committee — GB Review
January 11 — Metallurgy and Materials Committee Meeting
January 16 — Nomenclature Committee Meeting — WebEx
January 16 — Spline Committee Meeting — WebEx
January 17 — Bevel Gearing Committee Meeting — WebEx
January 18 — Gear Accuracy Committee Meeting — WebEx
January 23-25 — Gearbox System Design — Las Vegas, Nevada
January 23-24 — Working Group 13 Meeting — Alexandria, Virginia
January 25 — Lubrication Committee Meeting — WebEx
January 29 — BMEC Meeting — Orlando, Florida
January 30 — Helical Gear Rating Committee Meeting — WebEx

AGMA LEADERSHIP

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THE FEELING OF ZEN IS ONE THAT WE ALL can aspire to. Through practice and self-reflection, we can work toward being centered and grounded. Unfortunately for gears, too many designers overlook the need for gears to be properly centered.

When set at the proper center distance, gears will mesh at the pitch point. In doing so, they will perform as designed. The proper center distance allows for the tooth of one gear to engage with the mating gear at the optimal intersection point; it allows for the proper flow of lubricant into the mesh; it allows for the gears to operate with the designed backlash allowance, and it minimizes gear noise.

The formula for calculating the center distance of two parallel axis gears is:

\[
\text{Center Distance} = \frac{\text{Pitch Diameter of Gear A} + \text{Pitch Diameter of Gear B}}{2}
\]

This formula calculates the nominal value of the center distance. It does not reflect the tolerance of that value. For most parallel axis gear applications, a tolerance of ±25-30 µm is ideal. This range reflects the inherent variations in pitch through one rotation of each gear.

The accompanying table shows the center distance tolerance for parallel axis gears. The tolerance values in this table are quoted from JGMA1101-01 (2000), and are applicable for involute spur and helical gears made of iron and steel.

When designing a gear pair, the consideration of center distance is often one of the last values calculated. In some instances, the center distance between the parallel shafts is different than the optimum center distance of the gears. This situation can arise if you replace diametral pitch gearing with metric gearing. As these two systems are not 100-percent interchangeable, the center distance of standard metric gears would not fit an inch dimensioned shaft center distance. For example, a 10DP, 20-tooth spur gear mated with a 25-tooth spur gear would have a designed center distance of 2.25 inches. If these gears are replaced with a Module 2.5, 20-tooth spur gear mated with a 25-tooth spur gear, its design center distance is 56.25 mm (2.2146 inches).

One of the ways in which these metric gears can be adjusted in order to match the shaft center distance is to design the pinion with an enlarged addendum. A modification factor, known as the coefficient of profile shift, can be applied to a gear during manufacture, which will increase the pitch point of the pinion such that its effective pitch diameter is larger than the nominal pitch diameter. By selecting a positive profile shift coefficient that increases the effective pitch diameter of the pinion by 1.8 mm, these gears would now fit the 2.25-inch center distance.

The second way for these gears to meet this center distance requirement would be to introduce a helix angle. Since the pitch diameter of a helical gear is calculated as:

\[
\text{Pitch Diameter} = \left(\frac{\text{Module} \times \text{Number of Teeth}}{\cos \beta}\right)
\]

where \(\beta\) is the helix angle, the pitch diameter increases as the helix angle increases.

For this example, if you make the pinion a left-hand gear with a helix angle of 10° 10’ 54”, and make the mating gear a right-hand gear with a helix angle of 10° 10’ 54”, then these gears will have a center distance of 2.25 inches.

Although both of these manufacturing methods will achieve the desired operating center distance for the gear pair, both have their limitations. For gears with addendum modification, the maximum amount of shift is limited. The value for the coefficient of profile shift can vary from +1.0 to -1.0 where a value of +1.0 is equivalent to increasing the pitch diameter by one tooth and a value of -1.0 is equivalent to decreasing the pitch diameter by one tooth. When applying a positive profile shift, the pinion’s tooth tip becomes pointed and is weaker than a gear with a standard addendum. When applying a negative profile shift, the pinion’s tooth root is weakened due to undercutting.

For gears with a helix angle, there is a limitation on the helix angle, and there are additional forces on the gear tooth that are introduced due to the twist. A traditional spur gear has a helix angle of zero degrees. Once a helix angle greater than zero is applied, a thrust force is imparted by the gear mesh in the transverse direction. This thrust force requires the introduction of thrust bearings into the application.

One common issue with center distances is the use of undersized distances in order to reduce the backlash of two gears in mesh. In doing this, the assembler is forcing the gears to mesh at a point other than the pitch point. If the mesh is reduced to a point where the addendum of the pinion is hitting the tooth root of the mating gear, the results will be an increase in gear noise, premature wear, and a significant reduction in gear life. When properly centered, life is quiet and peaceful.

ABOUT THE AUTHOR: Brian Dengel is general manager of KHK-USA, which is based in Mineola, New York. Go online to www.khkgears.us.
ENHANCED STEEL PERFORMANCE DURING VACUUM CARBURIZING OF GEARS

There are many benefits that can be gained from vacuum carburizing, but material selection is critical to producing a consistently high-quality carburized gear.

MANY REFERENCES COVER THE HISTORY AND DEVELOPMENT of low-pressure carburizing, colloquially known as vacuum carburizing, so we’ll start our discussion with the motivations of gear manufacturers in choosing this process [1, 2]. The vacuum carburizing benefits commonly cited are the elimination of intergranular oxidation (IGO) for increased fatigue life, minimization/increased consistency in distortion response, elimination of an oil quench, energy efficient furnaces, better gas-quenching flexibility, and reduced carburizing times due to high-temperature furnace capabilities. Of this long list of potential benefits, we will discuss three key areas: high-temperature carburizing, fatigue performance (eliminated IGO), and response to high pressure gas quenching (HPGQ). Understanding how to optimize each of these conditions during vacuum carburizing can influence the proper alloy design and heat-treat methodology.

GRAIN COARSENING RESISTANCE

Vacuum carburizing furnaces are capable of operating at higher carburizing temperatures than traditional gas carburizing furnaces. An exponential relationship between carburizing temperature and case depth exists, shown in Figure 1, allowing for reduced cycle times. Figure 1 shows that, for an equivalent case depth, you can achieve a 43 percent or 55 percent reduction in carburizing time by increasing the carburizing temperature by 50°C or 70°C, respectively. However, most steels and processing schemes are not capable of maintaining a fine grain size at elevated carburizing temperatures.

A fine grain size is required to ensure optimal part performance after carburizing. Steel composition and prior processing can be used to increase the temperature at which austenite grain coarsening begins. The higher this temperature, the higher the carburizing temperature that can be used — thus capturing an increased benefit of reduced carburizing cycle time. Figure 2 shows composition and processing modifications to a traditional 5120 steel can increase the grain coarsening temperature from 980°C to above 1,050°C.

FATIGUE IMPROVEMENT

Fatigue testing is a primary performance measurement for carburized applications. Attainment of case and core characteristics is paramount to developing the necessary fatigue properties for a part.

Vacuum carburizing results in the elimination of intergranular oxidation and other related near-surface effects prevalent in gas carburizing. Four-point bending fatigue testing has been used to simulate gear root bending fatigue through the use of a notched test sample [3, 4]. These fatigue results can be used to compare various grades and process conditions. Figure 3 highlights the improvement in fatigue life runout when IGO is removed or prevented. The middle pair of conditions shows a 283 MPa increase in runout stress for identically heat-treated 8620 before and after mechanical removal of the IGO produced in traditional gas carburizing. The left and right side pairs show the comparison in runout stress between gas carburized and vacuum carburized conditions for 5120 and 8620, respectively. These two pairs show a similar boost in runout stress (239 MPa for 5120, 311 MPa for 8620) attributed primarily to the prevention of IGO formation during vacuum carburizing.

ROBUST HPGQ RESPONSE

Some early adopters of vacuum carburizing and HPGQ soon found the grades traditionally used in their gas carburizing process did not have sufficient hardenability to meet properties with a less severe gas...
quench. Because of this reason, many existing and proposed new grades of steel were analyzed for their ability to produce robust core hardness results in an HPGQ process. A designed experiment that varied composition (a 5120Mod and three experimental grades), sample size (0.75-inch, 1.5-inch diameter cylinders), HPGQ pressure (10, 15 bar nitrogen), and location with a furnace load (top, middle, bottom) showed the ability to link standard Jominy data to the expected core hardness given the cooling rate produced at the core of a gear [5]. Figure 4 shows the results of this study, plotting cooling rate (as calculated from measured temperature data from embedded thermocouples) versus the hardness immediately next to the embedded thermocouple. The plotted points represent the measured data, while the solid lines are the associated Jominy data for each grade plotted against cooling rates for each J-position.

A fairly good match is observed between the experimentally measured data and the standard Jominy results. The slope of the data sets can be interpreted as measurement of robustness with respect to a known variation of HPGQ processing conditions. A more robust quenching response (lower slope) would allow a range of cooling rates to consistently meet core property requirements.

CONCLUSIONS

Vacuum carburizing is a technology that is gaining broad acceptance in the industry in Europe and North America. Significant value can be brought to vacuum-carburized applications through steel-grade selection, development, and testing. Today, available steels provide consistent hardenability for each application, robust material response to varying quench rates, high-temperature grain coarsening resistance, and a cost-effective, lean alloy design. All of these aspects can help gear heat-treaters achieve the many cost reduction and performance enhancement opportunities available with vacuum carburizing heat-treating technology.

REFERENCES:

1. D. Herring, “Vacuum Carburizing Technology for Powertrain Gears,” presentation made in Port Huron, MI, USA, July 27, 2006. that, for an equivalent case depth, you can get a significant reduction in carburizing time by increasing the carburizing temperature.


ABOUT THE AUTHORS: Patrick Anderson is manager of the advanced modeling group at TimkenSteel Corporation. Mike Burnett is a technologist at TimkenSteel. Learn more at www.timkensteel.com.
CONTAMINATION OF POLYMER QUENCHANTS IN INDUCTION HARDENING

Some think coolants or other contamination do not harm the quenchant.
This perception couldn’t be farther from the truth.

INDUCTION HARDENING IS A COMMON AND EFFECTIVE
method of achieving a hard, wear-resistant surface without carburizing or nitriding. It is used extensively to harden gear teeth, as well as wear surfaces. A compressive residual stress at the surface contributes to excellent fatigue resistance. Heating is rapid, and well suited for automation. The preferred process sequence is shown in Figure 1.

In induction hardening, contamination is a common occurrence.

Polymer quenchants are commonly used for induction hardening. Contamination is always a problem. The most common contaminants in induction hardening applications are organic contaminants from hydraulic fluid leaks, coolants or forming lubricants, or machine swarf. Rust preventatives and cleaners are also found on parts. Solids, such as soot or high water hardness, can also be present. High inorganic salt content from either rust inhibitors or very hard water can also be present in the quench tank. Parts usually come from prior processes and may be covered with different types of coolants such as oils, synthetic coolants, emulsions, and semi-synthetic. If the parts are cleaned prior to heat treated (recommended, but rarely done), the parts may contain residual soaps. Dirt from the environment, and parts such as machining swarf, are always present. Many parts per hour are processed, with each part carrying a small amount of contaminant. This is compounded by the fact that many induction systems have a very small quench tank — often 100-200 gallons.

Typical customer samples of contaminated polymer quenchants used in induction hardening are shown in Figure 2. During the machining operations, coolant is used for lubrication and to help carry away chips and swarf. As the part moves to the induction step, parts should be properly washed and rinsed to remove chips, coolant, and machining swarf. However, often the cleaning step is omitted as “non-value added.” The perception is that the coolants or other contamination do not harm the quenchant. This perception is inaccurate.

Contamination by cutting fluids, rust preventatives, cleaners, or hydraulic fluids can provide nutrients for biological growth. These contaminants may also prolong or stabilize the vapor phase of the quenching process, which can contribute to low hardness and inadequate case depth. The effect of oil contamination on the cooling curve of a 5 percent solution of a typical PAG polymer quenchant used in induction hardening is shown in Figure 3.

At lower concentrations of polymer quenchant, more typical of those used in induction hardening, the presence of organic contamination (mineral oil) was found to be significant in increasing the time to 600° C and 400° C, and decreasing the cooling rate at 300° C. This has the effect of effectively increasing the concentration of the bath by several percent. Depending on the hardenability of the part, this can result in spotty hardness, or a case hardness that does not meet specification.

Often the use of an air knife prior to the heat-treating operation can remove most of the coolant and reduce the contamination. While cleaning the parts prior to heat treatment is always preferred, the use of an air knife is an acceptable alternative. Properly designed skimmers and filtration systems will help minimize this source of contamination from the quench tank (Figure 4).

Sediment and particulate matter such as scale have little effect on the quench rate but can increase the overall quench rate by providing nucleation sites for bubble formation and destabilize the vapor phase. This is not usually a problem in induction hardening because of the direct impingement of quenchant on the parts. However, this particulate matter can hinder concentration control by making the refractive index difficult to measure. This contamination also can affect the cleanliness of the quenched component. Filtration is always recommended to remove this particulate. Typically, filtration of the quenchant to approximately 10 microns is adequate to remove most particulates.

Most bacteria contamination found in polymer quenchants is associated with anaerobic bacteria. This is a common problem for all polymer quenchants. This bacterium is not a health hazard but more
of a smell issue. This smell is caused by large numbers of anaerobic bacteria decaying. Hydrogen sulfide (H₂S) is formed by the decay. Hydrogen chloride (HCl) can sometimes appear as a green cloud over the quench tank when the equipment is first started after a weekend or longer shut-down. Hydrogen sulfide, H₂S, is the rotten-egg smell. Both of these chemical compounds decrease the pH and contribute to increased corrosion of parts and equipment, and increased bacteria growth. It is this bacteria that contribute to “Monday Morning Smells.” This is typically cured by sump additions of a biocide or dumping the system. This is a costly, and potentially dangerous (with regard to biocides), solution to the problem of bacteria growth.

Anaerobic bacteria thrive in oxygen-depleted environments, such as is found at the bottom of a quench tank. Stagnant solutions contribute to localized oxygen depletion, which in turn increases potential for bacteria growth. To prevent stagnant solutions, the quenchant should be kept moving, and the quenchant free from rust and solids. It is these solids that provide much of the food source for anaerobic bacteria.

Bacteria can be controlled without the use of biocides. Cleanliness of the quenchant and quench tank is critical to preventing the growth of bacteria and the subsequent use of biocides. The quenchant should be kept agitated to eliminate stagnant solutions. Once the equipment is shut off for the weekend, the quenchant should be kept agitated. This can be done by keeping the filtration pump “on” over the weekend, or by intermittently initiating the agitation system on a periodic basis. A couple of minutes every hour should be adequate.

Filtration is also effective in reducing bacteria growth. Filtration serves two purposes. First it eliminates particulate, scale, and other debris from the quenchant that can act as a food source. Secondly, it maintains the quenchant cleanliness, so proper quenching can occur. Bag and cartridge filters are commonly used in this application. Typically bag sizes down to approximately 15-20 microns are used. However, these types of filters are not recommended for this application, as the low fluid flows through the filters and the high concentration of food sources can contribute to bacteria growth. Further, this type of filtration, once contaminated, can spread the contamination to other locations in the quench tank.

Sand filters are effective filtration systems for polymer quenchants. The nature of sand filtration has high flow velocities so stagnant solutions do not occur. Further, they are also effective at filtration, with filtration levels often at 6-8 microns or better. They are also cost-effective, as the filtration media is inexpensive, clean white sand.

Quenchants have been developed that will not sustain bacteria growth, and “will not stink” under most circumstances. These quenchants are specially designed for induction hardening applications with enhanced corrosion inhibitor packages. These are proprietary quenchants designed to have long life and not support bacteria growth. Examples are Houghton’s Aqua-Quench 245 and Aqua-Quench 145 polymer quenchants.

Proper control of the polymer quenchant tank is important to ensure long, consistent bath life. Contamination must be held to a minimum for proper quenching. Contamination is controlled by appropriate washing of parts prior to heat treatment. This yields consistent case hardness and depth, while extending bath life.

First Gear Engineering & Technology prides itself in putting the customer first. (Courtesy: First Gear Engineering & Technology)
First Gear Engineering & Technology continues its growth in gear manufacturing to many markets, including aerospace, medical, food, industrial fields, and contract gear inspection.

By Kenneth Carter | Editor | Gear Solutions

WHENEVER CUSTOMERS VISIT FIRST GEAR Engineering & Technology’s shop floor, they rave about how clean it is, according to company President Greg Leffler. That outward representation of a clean and orderly business helps to demonstrate the extra effort First Gear puts in to ensure that its customers are its highest priority. “We’ve always tried to approach business the way we would want to be treated,” Leffler said. “And that’s giving the customers more than they expect — every time. That’s our main drive.”

First Gear prides itself in putting the customer first, according to Quality Manager Greg Leffler Jr. First Gear began as a manufacturer of internal and external spur and helical gears, splines, and shafts. First Gear expanded its capability to include finish grinding and skiving. Finish rolling of planetary pinions (service parts) for the automotive industry is also a service that not many gear companies offer. Job quantities range from one-piece prototypes or long-term contract manufacturing up to several hundred thousand pieces per year.

First Gear is a client-focused company that strives to be the best partner to all customers. It works closely with customers going over all details of prints to ensure the end product meets or exceeds all expectations in all aspects of quality and meets time requirements. “We utilize the latest gear software suites from KISSsoft, UTS, and Ash Gear as well as custom software to assist customers with any gear or design analysis. Our customers can benefit from our years of experience,” Leffler said.

Here is what one customer had to say about First Gear: “Thanks for everything you do for us,” said Andrew Hines, five-time NHRA Pro Stock Motorcycle Champion from Vance & Hines Racing. “The transmission shafts and gears that you knocked out for us in that short time-frame in the middle of the season really allowed us to make the most of our starting line launches. Without having the availability to change ratios in different track conditions, we wouldn’t have been as good as we became.”

With modern, up-to-date, automated CNC machines, First Gear can take advantage of all the latest cutting tool materials and coatings as well as carbide to provide the highest quality gears at fast cycle rates to keep costs down, Leffler said.

GEAR BLANKING

Although a large part of the business is “cut teeth only” for other manufacturers, First Gear has always produced gears complete to print. Until recently, all gear blanking was outsourced, Leffler said. “We have been blessed to be able to work with good suppliers who have made quality blanks, but some markets today demand that all work is done in house for a variety of reasons but mainly efficiency and traceability,” he said. “We have purchased the highest quality lathes with bar feeders, live tooling, Y-axis, and sub-spindles to be able to achieve what some customers/jobs demand. We will continue to outsource or make in-house, whichever meets the customer’s needs best.”

First Gear Engineering & Technology is now an ISO 17025 accredited laboratory for master gear calibration, allowing for detailed recalibration of master gears and gauges. (Courtesy: First Gear Engineering & Technology)

ISO 17025 RECALIBRATION & INSPECTION

First Gear is now an ISO 17025 accredited laboratory for master gear calibration, which allows for detailed recalibration of master gears and gauges. This accreditation complements the company’s continued commitment to the AS9100 Rev. D/ISO 9001:2008 quality management system as well as its registration to ITAR. It also offers a state-of-the-art inspection lab where it monitors quality closely as well as performs contract outside inspection services.
FOLLOWING A JOB
When a potential customer comes to First Gear, the company makes sure to approach a job with the confidence to see it through from beginning to end.

“We're fortunate to have (General Manager) Mike Goza, who does most of the communications with our customers,” Leffler said. “He receives all the requests for quotes, and he reviews it with a team of people at First Gear to understand what it's going to take to get the job done.”

Part of that attention to detail comes from a shop floor management system that meticulously tracks a job’s status.

“We've got an electronic system, and every piece of information for each inquiry goes into this electronic file,” Leffler said. “We will quote it, and if we win the job, those details print out and are flowed down to the shop floor — every little detail about it, whether it be a micrometer or certain bolt, to finish the job.”

BEVEL GEARS
Over the years, First Gear has had numerous inquiries for bevel gears.

“Bevel gears are a whole new discipline, so we stuck to what we knew until we were confident we could take on bevel gears and do it right,” Leffler said. “As with everything, technology and machines have progressed to the point that we feel we need to be involved. We are now working on our first project, and when it is successful, we will be expanding into that type of gear business as well.”

None of these innovations will affect what First Gear is doing now and what it’s been doing for years.

As First Gear looks toward the future, it will continue to do what it does best for the gear industry, and that includes remembering its roots.

“My background is in racing,” Leffler said. “The race is going to start whether you're ready or not. So, we plan smart and execute as necessary to be ready when the green flag waves. It is inevitable that the unexpected can happen, but it’s how we react that sets us apart. We have always done whatever it takes to get the job done and be there on time, and we still operate that way. We’ll figure out a way to be successful and get it done.”

FOR MORE INFORMATION: www.first-gear.com
Phoenix® 280G is the only bevel gear grinding machine in its class that doesn’t compromise on quality, productivity or ease of use. Now available with integrated automation, made by Gleason.

www.gleason.com/280G
Numerical Thermal 3D Modeling of Plastic Gearing

In this exercise, a numerical thermal 3D model is used to predict the surface and body temperature of spur and helical plastic gears.

By Niranjan Raghuraman, MS, Dr. Donald Houser, and Zachary H. Wright, MS

This paper will primarily focus on the prediction of gear temperature of plastic gears using a numerical heat transfer model based on 3D Finite Difference (FDE) method. It is quite common that in most of the applications, the plastic gears are self-lubricated. Tooth surface wear is an important failure mode in plastic gears, and this primarily occurs because, at significantly higher loads, the surface temperature might increase to a value close to the melting point of the material, thereby changing the surface behavior. Thus, it is critical to compute the temperature of the gear pair in an accurate fashion. The heat source is the frictional heat dissipation due to sliding of the gears. The model is capable of solving for both helical and spur gears.

Since the gear tooth experiences a repetitive heating and cooling cycle for every rotation, the heat input is averaged over one rotation of tooth and is independent of time. The program computes both the surface and body temperature of the gears as a function of space and time. Because of the inherent nature of the implicit FDE method, there is no restriction on the discretization both in the time and space domains. This reduces the simulation time to a great extent without much compromise in accuracy. The results are correlated with experimental data, and the good agreement is achieved between the test and simulation results for different cases of load and speed. This simulation was developed for plastic gears and can be extrapolated to metal gears, with the greatest challenge being obtaining the accurate heat transfer coefficients for lubricated gears.

1: INTRODUCTION

The popularity of plastic gears is slowly rising in the gearing community, and they are constantly in the realm of replacing metallic gears in many of the low-load applications. This is primarily because of their low cost and weight, their quietness in operation, and ability to function in the absence of any lubrication. But to its downside, their performance under higher loading conditions are questionable. Adding to this, their complex thermo-mechanical and visco-elastic behavior pose a great challenge when it comes to the design of plastic gears.

Much documented research was published from the latter half of the 20th century on plastic gear design. Gauvin et al. [1] worked on the prediction of peak surface temperature of plastic gear teeth. Park et al. [2] made an experimental verification of the surface temperature and wear of plastic gears after a number of million cycles and compared them with other computer models developed for polymer gears. Tooth surface temperature of gear pairs has been experimentally verified by various authors [3-5]. Koffi et al. [6] developed a rudimentary model to predict gear temperature by considering both frictional and hysteresis heating. He concluded that the contribution of hysteresis heating in plastics is much less than that of frictional heating. Hooke et al. [7-8] developed a model to predict the surface temperature in polymer gears and its relationship to gear wear — along with comparison with experiments. Takanashi et al. [9-10] studied the heat generated in plastic gears due to friction and predicted the equilibrium temperature of a plastic gear pair.

The numerical models to predict gear temperature, specified in the above mentioned literature are mainly empirical relationships based on experimental data and are not mathematical models that involve solving physics-based heat equations. The primary disadvantage of having an empirical model is to justify its use over a wide range of operating conditions. The accuracy of the model is highly influenced by the sample of operating conditions considered in the experiments. Doll et al. [11] has done significant work on developing numerical FEA models to compute temperature of gears in contact using ANSYS, but the downside of that method is the simulation time. This paper talks about an efficient Finite Difference-based approach that provides a numerical solution to the heat equations that govern the heat transfer mechanism in a gear pair.

Previous work done by the authors [12-13] involves the development of a calculation procedure to account for transient temperature and humidity in the load distribution analysis of plastic and metal gears. This work acts as a pre-processor to that study by providing the temperature distribution of the gears in mesh.

The temperature distribution predicted by this model is compared with test results published by Hooke et al. [7-8]. The steady state results predicted by the model is in good agreement with the experimental measurements obtained using infrared camera and thermocouples. Case studies that show these comparisons are elaborated in sections below.

METHODOLOGY

This section explains the methodology involved in this model. It is important to identify the modes of heat transfer for a gear pair in mesh in order to predict the transient thermal behavior of the system. In this model, friction is assumed to be the only source of heat generation. Since most plastic gear applications are self-lubricated, the heat is lost due to convection to the atmosphere and conduction to the supporting shaft. Another assumption in this model is that a...
temperature distribution is imposed on the shaft that can either be time-dependent or time-independent.

2.1: BOUNDARY SPECIFICATIONS
Figure 1 shows a schematic of the tooth section. Each tooth is divided into eight sections, which has its own set of governing equations to define the heat transfer mechanism. Based on Figure 1, the following sections are elaborated below.

2.2: GRID LAYOUT
The gear tooth is discretized in space in \( r \), \( \theta \), and \( z \) coordinates which are radial, angular, and face-width directions, respectively. As far as this model is concerned, the domain is mapped by cylindrical grids as shown in Figure 2.

The mesh is generated using the given tooth geometry, and the nodes form the center of each finite control volume. This process is repeated in all the three directions. The mesh can be non-uniform, which means that the nodes need not be equi-spaced. Figure 3 shows an example generated mesh diagram of the tooth section. The number of nodes along the active profile, top and bottom, land and face width, are specified by the user. The involute shape of the active and inactive profile is discretized based on the number of points specified, and a straight line connection is assumed between the nodes.

2.3 GOVERNING EQUATIONS AND DISCRETIZATION
As stated previously, different regions of the tooth section exhibit different boundary conditions. Each of these boundary conditions are discretized with respect to space and time, and the resulting equations are solved simultaneously to obtain the temperature distribution. As far as the discretization is concerned, implicit finite difference method [14] is used in this model. Some of the main advantages of this method are:

- The method is unconditionally stable. Unlike explicit method, implicit method is stable for all values of time and space discretization steps.
- This method is not restricted by the discretization lengths in both space and time domains. This increases the computational efficiency, as the time steps can be higher — to quickly attain steady state condition.

Figure 4 shows the nodal arrangement for an interior node \((i, j, k)\). Let \( i \), \( j \), and \( k \) denote the nodes along the angular, radial, and face-width direction.

In the above figure, “L” represents the distance between any two nodes in a specific direction. “A” represents the cross-sectional area (2D) of the heat transfer face. i.e.:

\[
A_i = \left( \frac{L_r}{2} \right) \cdot \left( \frac{L_s + L_b}{2} \right)
\]

Equation 1

All the interior nodes lose/gain heat only through conduction. The heat transfer mechanism can be described by the following equation:

\[
\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{1}{r^2} \frac{\partial^2 T}{\partial \theta^2} = \frac{1}{\alpha} \frac{\partial T}{\partial t}
\]

Equation 2

Where, \( \alpha \) is the thermal diffusivity of the gear material. Superscript “n” represents the temperature at \( n \)th time instant. Equation 1 can be discretized as follows:

\[
\frac{\partial^2 T}{\partial r^2} = \frac{T_{i+1,j,k}^{n+1} + T_{i-1,j,k}^{n+1} - 2T_{i,j,k}^{n+1}}{\Delta r^2}
\]

Equation 3

\[
\frac{1}{r^2} \frac{\partial T}{\partial r} = \frac{T_{i+1,j,k}^{n+1} + T_{i-1,j,k}^{n+1} - 2T_{i,j,k}^{n+1}}{r^2 \Delta \theta^2}
\]

Equation 4

\[
\frac{\partial^2 T}{\partial \theta^2} = \frac{T_{i,j,k+1}^{n+1} + T_{i,j,k-1}^{n+1} - 2T_{i,j,k}^{n+1}}{\Delta \theta^2}
\]

Equation 5

\[
\frac{\partial T}{\partial t} = \frac{T_{i,j,k}^{n+1} - T_{i,j,k}^{n}}{\Delta t}
\]

Equation 6

Equations 3-6 are combined to obtain \( T_{i,j,k}^{n+1} \) as a function of the temperature in the next time step. Combining the above equations, we get:
Where

\[ R_m = \frac{L_m}{A_m \Delta t} \]

\( "V" \) is the control volume expressed in SI units. Equation 7 gives the temperature of every interior node as a function of mesh geometry and the temperature in the next time step. Similarly, the temperature of every node on each of the boundaries are obtained as a function of mesh geometry and temperature of that node in the next time step. So, if the mesh contains 500 nodes, we will be having 500 equations of the form:

\[ T_{ij,k}^n = T_{ij,k}^{n+1} \left( 1 + \sum_{m=1}^{4} \frac{1}{R_m} \right) - T_{ij,k+1}^{n+1} \left( \frac{1}{R} \right) - T_{ij,k-1}^{n+1} \left( \frac{1}{R} \right) - T_{ij+1,k}^{n+1} \left( \frac{1}{R} \right) - T_{ij-1,k}^{n+1} \left( \frac{1}{R} \right) - T_{ij+1,k+1}^{n+1} \left( \frac{1}{R} \right) - T_{ij+1,k-1}^{n+1} \left( \frac{1}{R} \right) - T_{ij-1,k+1}^{n+1} \left( \frac{1}{R} \right) - T_{ij-1,k-1}^{n+1} \left( \frac{1}{R} \right) \]

**EQUATION 7**

Where

\[ C_i \]

is the coefficient of neighboring nodal temperature.

For a node along active profile, the governing equation can be found using the energy balance method. Figure 5 shows the nodal arrangement of a node along the active profile. It can be seen that there is external heat that is being added to the node \( (i, j, k) \).

This external heat is because of the friction, due to sliding between the contacting surfaces. It is a function of friction coefficient, sliding velocity, and the contact pressure. The frictional heat can be computed using the relation:

\[ Q = \mu \cdot V \cdot \sigma_{cs} \]

**EQUATION 10**

It is important to note that the value of “\( Q \)” varies for every node along the profile. Since sliding is zero at pitch point, \( Q \) will be zero at a node located in the operating pitch point of the gear. Consecutively, \( Q \) is maximum either at EAP or SAP. The contact pressure distribution as a function of roll angle and face width can be obtained from WindowsLDP, which is a gear contact solver developed by The Gear and Power Transmission Laboratory at The Ohio State University [15]. Figure 6 shows a sample heat rate distribution along the active profile (no micro-geometry modification).

The heat rate constantly traverses across the teeth depending on the speed of rotation of the gear. A single tooth undergoes cyclic heating and cooling for one mesh cycle. In this model, the heat rate that is applied on the active profile is assumed not to change with time. The benefit of this approach is that the model is insensitive to the time step that is chosen for transient temperature calculation. But for this approach, the time step involved in the analysis will be in the order of the mesh cycle, and this would drastically increase the computational time. Essentially, by this idea, the boundary conditions on the active profile are made to not vary with time, whereas in reality, it does, as the gear makes one full rotation. In order to make the heat flux independent of time, a net heat rate is calculated, which is a time-based average of the heating and cooling that is seen by the nodes on the active profile. The calculation method is shown in Equation 11:

\[ Q_{avg} = \frac{t_1 Q_{heat} + t_2 Q_{cool}}{t_{meshcycle}} \]

**EQUATION 11**

Where \( t_1 \) is the time taken for the node to heat up and \( t_2 \) is the time taken for the node to cool down. \( t_{meshcycle} \) is the time taken for one complete rotation of the gear. It can be deduced that \( t_{meshcycle} \) is essentially the sum of \( t_1 \) and \( t_2 \). The cooling rate can be calculated based on the convection relation as shown in Equation 12:

\[ Q_{cool} = h_{conv} \cdot A_{cross-section} \cdot (T - T_{atm}) \]

**EQUATION 12**

The shape of the active profile is assumed to be a perfect involute, ignoring any micro-geometry modifications that might be incorporated in the gear pair. However, this modification is ignored only during mesh generation, and its effects are considered in the temperature prediction calculations. The contact stress profile generated by WindowsLDP accounts for the micro-geometry modification, and this directly affects the temperature distribution on the surface of the gear pair.

By solving the energy balance for every node on the active profile, the governing equation can be found out and it is shown in Equation 13:

\[ T_{ij,k}^n = T_{ij,k}^{n+1} \left( 1 + \sum_{m=1}^{4} \frac{1}{R_m} \right) - T_{ij,k+1}^{n+1} \left( \frac{1}{R} \right) - T_{ij,k-1}^{n+1} \left( \frac{1}{R} \right) - T_{ij+1,k}^{n+1} \left( \frac{1}{R} \right) - T_{ij-1,k}^{n+1} \left( \frac{1}{R} \right) - T_{ij+1,k+1}^{n+1} \left( \frac{1}{R} \right) - T_{ij+1,k-1}^{n+1} \left( \frac{1}{R} \right) - T_{ij-1,k+1}^{n+1} \left( \frac{1}{R} \right) - T_{ij-1,k-1}^{n+1} \left( \frac{1}{R} \right) - Q_{avg} \left( \frac{1}{R} \right) \]

**EQUATION 13**

Where

\[ R_m = \frac{L_m}{A_m \Delta t} \]

\[ R \]

\[ k_{cond} \]

being the thermal conductivity of the gear material. For plastics, the thermal conductivity is in the order of 0.5 W/mk. This low thermal conductivity causes most of the heat generated to be retained at the surface of the tooth. This is even more critical when it comes to predicting the gear failure modes like tooth wear, which is caused by the tooth surface melting at high operating temperatures.

For the nodes along the inactive profile, the methodology is essentially the same, except for the fact that the nodes are always cooled. So, the term \( Q_{avg} \) in Equation 13 is replaced by \( Q_{cond} \) from Equation 12. Figure 7 shows the nodal arrangement of the nodes in the inactive profile.
It can be seen that the heat is constantly removed from the surface through convection. Using energy balance method, the governing equation can be written as shown in Equation 15:

\[ T_{i,j,k} = \frac{1}{R} \left( T_{i,j,k}^{n+1} - \frac{1}{R} \left[ Q_{cool} + \sum_{m=i}^{5} R_m \left( T_{i,j+k}^{n+1} - T_{i,j-k}^{n+1} \right) \right] \right) \]

**EQUATION 15**

Where the terms \( R_{1-5} \) are explained in Equation 15. For the nodes along the top land and bottom land, the governing equations resemble the ones for the nodes on inactive profile, as the heat transfer mechanism is the same in both cases — convective cooling. Figure 8 shows the nodal arrangement of the nodes along top land and root fillet boundaries.

The discretized form of the PDEs for the nodes in this boundary is given in Equation 16:

\[ T_{i,j,k} = T_{i,j,k}^{n+1} - \frac{1}{R} \left( \frac{1}{R} \left[ T_{i,j+k}^{n+1} - T_{i,j-k}^{n+1} \right] \right) - Q_{cool} \]

**EQUATION 16**

Where

\[ R_m = \frac{k_m}{\rho_m c \Delta T} \]

\[ R_e = \frac{k_{cool}}{\rho c \Delta T} \]

It is assumed that all the nodes that are in contact with the shaft are at a specified temperature at any given instant of time. This specified temperature can either be a constant number or a time-dependent distribution. For the nodes along the shaft:

\[ T_{i,j,k} = T_{shaft} \]

**EQUATION 18**

For the nodes along the front face of the tooth (“B” from section 2.1), the nodal arrangement is shown in Figure 9. The nodal arrangement for the nodes along the rear face is exactly the same but for the node \((i, j, k+1)\). Instead, the node \((i, j, k-1)\) will take its place.

Again, the heat transfer mechanism is exactly the same as the ones on the top and bottom land. The discretized form of the equations goes like this:

\[ T_{i,j,k} = T_{i,j,k}^{n+1} - \frac{1}{R} \left( \frac{1}{R} \left[ T_{i,j+k}^{n+1} - T_{i,j-k}^{n+1} \right] \right) - Q_{cool} \]

**EQUATION 19**

Where the terms \( R_{1-6} \) has their original meaning, it is important to keep in mind that the \( Q_{cool} \) is computed using Equation 12. The heat transfer coefficients for each of the boundaries can either be specified by the user, or the values that are computed by the program for a given gear can be used. The methodology to compute heat transfer coefficients is elaborated in the next section.

Finally, the nodes along the boundaries C, F (from Section 2.1) exhibit symmetry in the angular direction, i.e. the temperature profile for every tooth section is identical. The discretized form of the governing PDEs are shown in Equation 20:

**EQUATION 20**

Where the suffix \((n_{ang}-1)\) represents the node before the last node in the angular direction as shown in Figure 10.

After obtaining the equations for all the nodes at current time steps, they are solved simultaneously to compute the temperature of a given node at every time step. The connectivity matrix is generated by filling out the right rows and columns that match the spatial dependence of every node according to each of its governing equations. The connectivity matrix looks like the one shown in Equation 21:
The above equation is of the form $KX = F$, where $X$ is the unknown. In the above case, $T_{ij}$ is the unknown, and the $K$ matrix can be inverted to get the necessary solution at every time step. This process is repeated for every time step, and the transient temperature is obtained.

It is important to have knowledge on the heat transfer coefficients and the friction coefficients that are involved in the process. The next section elaborates on how the convective coefficients are obtained.

### 2.4 HEAT TRANSFER COEFFICIENT

For plastic gear applications, Takanashi et al. [9-10] formulated an empirical relationship for the heat transfer coefficients that essentially depend on the speed and the properties of the cooling medium. Since most of the plastic gearing applications are self-lubricated, the cooling medium is assumed to be air. The convective heat transfer coefficient depends on the following:

- **Flow Geometry:** It is the shape of the surface over which the fluid flow occurs.
- **Material Property:** It includes the property of the material such as surface roughness, etc., and also the property of fluid — the Prandtl number. It is defined as the ratio of momentum and thermal diffusivity of the fluid.
- **Reynolds Number:** The boundary layer conditions are strongly influenced by this parameter. It is important to predict whether the flow is laminar or turbulent.

It can be intuitively stated that the gears cool faster when they run at greater speeds. However, to ascertain this statement, works from the past have been studied, which predict the relation between heat transfer coefficients and the rotational speed of the cooling. Takanashi et al. [9-10] have come up with an equation that relates the heat transfer coefficient, the rotational speed and other fluid properties. The overall heat transfer coefficient is calculated as shown in Equation 22:

$$ h = 1.75 \frac{h_{diff}}{m} \left( \frac{m}{\mu} \right)^{0.5} \left( \frac{V_{el}}{V_{0}} \right)^{0.64} \left( \frac{V_{0}}{V_{eff}} \right)^{0.64} \text{ EQUATION 22} $$

Where $R_P$ is the pitch radius of the gear, $w$ is the rotational speed, $\nu_0$ is the kinematic viscosity of air and $\alpha_{eff}$ is the thermal diffusivity of air. $m$ and FW represent the module and face width of the gear respectively. The next section elaborates the functionalities of the program by analyzing a gear pair and comparing it with the experimental data found in the literature search.

### 3: CASE STUDY – RESULTS AND DISCUSSION

Mao et al. [7–8] conducted a series of experiments to predict the gear temperature of plastic gear pairs in mesh. These results have been used as a metric for validating the temperature profiles predicted by the model presented in this paper. Some of the other capabilities of the program are also presented in this paper which include:

- Prediction of transient gear bulk temperature.
- Effect of micro-geometry modification on the surface temperature distribution of the gear.
- Prediction of micro-geometry due to temperature change based on previous work by the author [12-13] and Kashyap [16].

Table 1 summarizes the gear geometry that was used in this paper. As mentioned before, it is the same gear pair that was used by Mao et. al [7-8].

The ambient temperature conditions are summarized in Table 2.

The ambient and the shaft temperature values are taken from the experiments conducted by Mao et al. and are imposed in the model. The gears were tested at various levels of speed and torque and measured the non-contact surface temperatures (inactive flank) using infrared video cameras. Figure 11 shows the comparison between their experimental results and the temperature values generated by the program. The values shown are the maximum surface temperature of the non-contacting flank of the gear. The predicted surface temperatures of the inactive flank are in good agreement with the experimental data.

For the 500, 1,000, and 1,500 rpm speed cases, the experimental data shows a change in slope at about 7 Nm torque. This might be due to change in boundary condition with respect to ambient temperature during the testing. The airflow in the gap between the gear teeth as the gears come out of mesh will be turbulent in nature. It is fair to assume that the temperature of the air within each pocket is a constant. Theoretically, the heat transfer coefficients will vary across the gear flank as the gears act like an air pump. But, capturing this physics is beyond the scope of this work, and one might need to incorporate detailed CFD models to accurately predict the heat transfer coefficients. One other important assumption of this work is that the coefficient of friction is assumed to be 0.11 for Acetal-Acetal surface pair based on the Delrin data sheet [17]. In reality, the coefficient of friction depends on load and speed conditions, but it is assumed to be constant in this model. The model relies on the user to provide reasonable coefficient of friction values.

<table>
<thead>
<tr>
<th>Parameter (mm)</th>
<th>Gear 1</th>
<th>Gear 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. of teeth</td>
<td>30</td>
<td>30</td>
</tr>
<tr>
<td>Module</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>Pr. Angl</td>
<td>20°</td>
<td>20°</td>
</tr>
<tr>
<td>Tip Diameter</td>
<td>64</td>
<td>64</td>
</tr>
<tr>
<td>Root Diameter</td>
<td>54</td>
<td>54</td>
</tr>
<tr>
<td>Center Distance</td>
<td>60.24</td>
<td>60.24</td>
</tr>
<tr>
<td>Face width</td>
<td>17</td>
<td>17</td>
</tr>
<tr>
<td>Shaft OD</td>
<td>40</td>
<td>40</td>
</tr>
<tr>
<td>Material</td>
<td>Acetal</td>
<td>Acetal</td>
</tr>
</tbody>
</table>

Table 1: Spur gear geometry used in this analysis.

<table>
<thead>
<tr>
<th>Zones</th>
<th>Temperature (deg C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ambient</td>
<td>35</td>
</tr>
<tr>
<td>Initial</td>
<td>25</td>
</tr>
<tr>
<td>Shaft Temperature</td>
<td>35</td>
</tr>
</tbody>
</table>

Table 2: Temperature conditions.

The steady state temperature values of the inactive profile as predicted by the model is shown in Figure 11. It can be observed that they are close to being linear with respect to torque for a given speed. This can be attributed to the fact that the change in torque level is reflected only in the contact stress distribution that is fed to the model to predict the heat flux generated due to friction. The model takes approximately six seconds to compute the temperature distribution for a single load case and is therefore a computationally efficient method. This comparison checks for the following:

- Reasonable temperature prediction within the realm of experimental data.
- Dependence of temperature on load and speed of the gear pair.

This comparison is to verify the model behavior and not to correlate with this set of experimental data because of the unknowns that exist with respect to boundary conditions, surface parameters, etc. Figure 12 shows the heat transfer coefficients that were used in this model as a function of gear speed. This model assumes the heat transfer coefficients are the same for all boundaries of the tooth section that experience convective cooling. However, the user can override the program to specify different heat transfer coefficients for different boundaries. The relationship with the speed is non-linear, as can be seen from Equation 22.

To illustrate the capabilities of the model, a speed of 1,000 rpm and a torque of 7 Nm was chosen. Figure 13 shows the steady state temperature distribution of the active flank of the gear pair.

The maximum temperature occurs at the EAP of the teeth due to the peak sliding
velocity occurring at either EAP or SAP. At the SAP, heat is conducted to the shaft and convectively cooled by the ambient air. This causes the EAP to have higher temperatures than the SAP. The temperature at the edges of the teeth are approximately 10 degrees less than the temperature at the center. This can be attributed to the fact that nodes at the edges cool faster because of the convective heat loss to the atmosphere. The minimum temperature on the active profile is observed at the pitch point. This is congruent to the fact that sliding is zero at pitch point. This can be seen from the heat flux distribution at the center of the teeth, as shown in Figure 14.

In order to study the effect of micro-geometry on the surface temperature, a profile crown of 25 microns and a lead crown of 40 microns is used on the active flank to predict the contact stress distribution for input to the heat transfer model. Figure 15 shows the heat flux distribution as a result of the micro-geometry modification.

The effect of lead crown can be seen in the flux distribution. However, the effect of the profile crown is not that evident because of the multiplication of sliding velocity with contact stress. Since sliding is maximum at SAP, EAP, and zero at pitch, this type of flux distribution is seen. Figure 16 shows the temperature distribution after applying the micro-geometry modification.

It is clearly observed that temperature at the edges of face width are drastically lower than the center, and the overall temperature of the surface is lower when compared with the case of no micro-geometry. This can be attributed to the reduction of overall contact zone in the gear mesh. These results show the sensitivity of the model to micro-geometry and its effect on the temperature distribution.

Due to increase in surface temperature of plastic gears and subsequent thermal expansion, the tooth form will be modified from its nominal involute form at ambient temperature. The resulting modification to the new micro-geometry can be predicted because of the thermal expansion. This algorithm was initially structured by Kashyap [16] and was further developed as a part of the author's master's thesis. The new modification, along the surface normal direction, is calculated for every point on the involute profile based on its temperature. Figure 17 shows the micro-geometry modification at steady state because of thermal expansion.

The negative sign indicates the growth in material from pure involute form. However, the main assumption in this calculation is that the tooth surface is intact and did not undergo any form of wear or any other surface abnormality during the operation. It can be seen that the zone with the highest temperature has expanded at a higher rate than the other regions. Ideally, one approach is to iteratively compute contact stress distribution due to this micro-geometry and re-calculate temperatures at regular intervals. This is one of the future goals of this work.
4: SUMMARY
The plastic gear surface temperature distribution can be predicted using the developed model. The finite difference method was used to perform thermal analysis on a gear pair for predicting surface and bulk temperatures. This is a physics-based model and does not use any empirical relationship to compute temperature.

The model predicts that the steady state temperature increases with torque, which confirms what is expected. The relationship between rotational speed and temperature is non-linear, due to the speed dependency of heat transfer coefficients. The model also predicts the effect of micro-geometry modification on temperature distribution, and vice versa. Profile or lead crown on the gear teeth can cause the maximum temperature to increase on the active flank, but at the same time, reducing the temperatures at the edges.

The scope of this work can be expanded by developing simple analytical models to predict the heat transfer and friction coefficients for various speed and load conditions that can be fed as an input to this model.

REFERENCES
17. Product Information, DuPont Delrin 100 NC010 Acetal Resin

ABOUT THE AUTHORS: Niranjan Raghuraman, MS, and Zachary H. Wright, MS, are with Romax Technology Inc. Visit www.romaxtech.com. Dr. Donald Houser is on the faculty of The Ohio State University, at www.osu.edu. This paper was presented at the American Gear Manufacturers Association’s technical meeting. The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the AGMA.
Measuring Root/Flank Stress in Plastic Gears

This paper provides an overview of the testing procedure for plastic gears according to the VDI 2736-4.

By Dr. A. Pogacnik and Dr. S. Beermann

The number of plastic gears used in different applications is increasing every year, mostly due to their cost effectiveness for large series production and lubrication-free running. With a rapid development of plastic materials in terms of strength and allowable temperature range, plastic gears are finding their way also into more demanding applications, where high transmittable torque at elevated temperatures is required. Unfortunately for gear design engineers, “gear fatigue data” (S-N curves) are rarely measured for new materials.

If the decision is made to measure gear-fatigue data, it is worth doing it correctly so that the measured data can be used. This paper provides an overview of the testing procedure for plastic gears according to the VDI 2736-4, which was published in 2016. The procedure to measure temperature and other challenges that arise are discussed. Furthermore, the statistical procedure to evaluate test results is explained.

Measured data according to the VDI 2736-4 must be converted correctly to become usable for strength calculation according to the VDI 2736-2. To help test engineers, a software tool was developed. The measured data is introduced, then the software performs the evaluation of the test results automatically and calculates the permissible root/flank stress data for different temperatures.

In the conclusion, some remaining problems are discussed. For the calculation of achievable lifetime or for loads based on duty cycles, the S-N curves (Wöhler lines) should be defined up to $10^{10}$ cycles, which is far beyond the obtainable test results. Therefore, an extrapolation of the measured data to higher cycle numbers should be considered.

INTRODUCTION

Due to the well-known advantages of plastic gears, their use is increasing every year, especially in the automotive industry where lubrication-free running, low noise, and high serial production are required [1,2]. With a rapid development of plastic materials in terms of strength and allowable temperature range, plastic gears are finding their way also into more demanding applications, where high transmittable torque at elevated temperatures is required [3,4]. But this is a problem for gear engineers as gear data (permissible stresses) are not being measured at the same rate as new plastics are developed. In fact, there are just a few new materials measured every year, for which permissible root/flank stresses are publicly available. The initiative is mainly coming from the major plastic material companies, which want to promote their materials also for gear applications.

As an engineer, how can you design plastic gears if no reliable fatigue data is available for the material that is used in the application? It is possible to calculate root/flank safety factors with fatigue data from a material, which has gear data available. But due to uncertainties between the two materials in question, the safety factors should be increased/decreased. For high temperatures, there is very little gear data available. Often, extrapolation is used to project permissible stresses at high temperatures, but this can be very inaccurate. At the end, all these uncertainties lead to gear designs that are not optimized in terms of strength and can be far from an optimal solution in terms of price.

An alternative to the above-mentioned procedure is to generate gear data for a material that is being used in the application. Generating root/flank fatigue data on actual gears (on a gear test rig) is a huge effort in terms of time and money spent. Generating permissible root stresses for three different temperatures and for five different numbers of cycles (between $0.1 \cdot 10^6$ and $5 \cdot 10^9$) can easily take between 3-4 months and can cost up to 50,000 euros.

If decision is taken to measure gear data, it is worth doing it correctly so that the measured data can be used at the end. In this paper, an overview of the testing procedure for plastic gears according to the VDI 2736-4 [5], which was published in 2016, is provided. The procedure to measure temperature and other challenges that arise are discussed. The statistical procedure to evaluate test results is also explained. A comparison of the permissible tooth root strength, measured on different test gears, will be discussed, indicating a large scatter between the measured permissible stresses.

Measured data according to the VDI 2736-4 must be converted correctly to become usable for strength calculation according to the VDI 2736-2 [6]. The calculation of tooth root and flank safety factors requires S-N curves, which are temperature dependent. The wear calculation requires wear factors.

To help test engineers, a tool was developed based on the VDI 2736-4. The measured data is introduced, then the tool performs the evaluation of the test results automatically and calculates the permissible root/flank stresses. This information is documented in a text file, which can directly be used by a calculation tool for the calculation of plastic gears according to the VDI 2736-2.

For the safety factor calculation with duty cycles, the S-N curves should be defined up to the $10^{50}$ cycles, which is far beyond the obtainable test results. Therefore, an
extrapolation of the measured data to higher cycle numbers should be considered.

**CONVERTING CYCLES TO FAILURE**

According to the VDI 2736-4 guideline, each test condition (T(torque), ϑ(temperature)) must be measured at least three times. To measure a S-N curve for one temperature, cycles to failure should be measured at four different loads, each being repeated at least three times. In total, at least 12 measurements are necessary to calculate a S-N curve for 1 temperature. Cycles to failure for every (T, ϑ) condition and every repetition can be written as

\[ N_j(T_i, \vartheta_i), \]

\[ 1 \leq i \leq n_{T-\vartheta \text{ pairs}}, \quad 1 \leq j \leq n_T. \]

Since the S-N curve has a logarithmic scaling for the number of load cycles, a variable for the logarithm of the number of load cycles is introduced for simplifying the equation.

\[ L_{ij} = \log_{10} N_j(T_i, \vartheta_i). \]

For every test condition (T_i, ϑ_i), the mean value of cycles to failure is calculated as

\[ \overline{L}_{i,50\%} = \frac{1}{n_i} \sum_{j=1}^{n_i} L_{ij}, \quad 1 \leq i \leq n_{T-\vartheta \text{ pairs}}. \]

The mean value is related to 50 percent damage probability. The standard deviation is estimated through

\[ s_i = \sqrt{\frac{1}{n_i-1} \sum_{j=1}^{n_i} (L_{ij} - \overline{L}_{i,50\%})^2}, \quad 1 \leq i \leq n_{T-\vartheta \text{ pairs}}. \]

To improve the quality of estimates, standard deviations of all (T,ϑ) pairs are normalized and averaged, as shown in

\[ \bar{s} = \frac{1}{n_{T-\vartheta \text{ pairs}}} \sum_{i=1}^{n_{T-\vartheta \text{ pairs}}} \frac{s_i}{L_{i,50\%}}. \]

The logarithm of the number of cycles with 10 percent failure probability (assuming normal distribution) are calculated as

\[ L_{i,10\%} = L_{i,50\%} - 1.28 \cdot \bar{s} \cdot L_{i,50\%} = (1 - 1.28 \cdot \bar{s}) \cdot L_{i,50\%}, \]

and the statistically determined cycles to failure at 10 percent damage probability as

\[ N_{i,10\%} = 10^{L_{i,10\%}}. \]

The VDI 2736-4 always calculates with the 10 percent damage probability. If necessary, cycles to failure can easily be calculated also for lower damage probabilities.

In Table 1, a statistical calculation of cycles to failure at different damage probabilities for POM gear is shown. Measured cycles to failure were 143300, 100780, and 94020 cycles (calculated standard deviation of cycles to failure: 26715). It can be seen that, at 10 percent damage probability, the statistically calculated cycles to failure are 30 percent lower than the calculated average value of cycles to failure. Without recalculating cycles to failure to lower damage probabilities, 50 percent of the gears would fail even before the desired lifetime (112700 cycles) is achieved. It is evident that statistical evaluation of the results is necessary to have a reliable lifetime calculation of plastic gears.

To have a valid gear test, the test must run until failure, otherwise it should not be used in the statistical evaluation.

**GEOMETRY OF THE TEST GEARS**

The VDI 2736-4 guideline defines three sets of possible test gears, ranging from normal module 1 mm to 4.5 mm. For testing small, injection molded plastic gears, “size 1” gears is preferred. Parameters of the size 1 gears are shown in Table 2. The material for the pinion (DIN quality 6) is steel and for the gear (DIN quality 10), the plastic material that we want to test. A possible material for the pinion is 100Cr6, tempered and hardened to 55 HRC. However, other materials can also be used.

Nevertheless, it is not always possible to use the test gear geometries defined in the VDI 2736-4. The reason could be that the gears are too “strong” for the given test rig (would not be able to get any failure in reasonable testing time) or that the center distance does not fit. In such cases, it is recommended to design your own test gears with similar properties to the VDI defined gears (similar theoretical contact ratio and contact ratio under load, similar specific sliding). This should keep the measured data comparable with the measurements on the VDI defined gear geometries.

It is also very important to check the operating backlash at expected maximum temperatures to avoid gear jamming as this can lead to “wrong” cycles to failure. If jamming occurs, instead of having contact just on the working flanks, additional contacts occur also on the non-working flanks, which results in additional forces, and heating. In cases where contact occurs on both flanks, the gears usually fail after a short number of cycles.

If root or flank fatigue is investigated, it is necessary to prevent excessive wear on the plastic gear as it can affect the outcome of the tests significantly. The surface roughness Ra of the steel pinion should be around 0.3 µm or lower (Rz < 1.5 µm) [7]. However, smaller roughness does not necessarily lead to lower wear; there might be an optimum with minimal wear. For plastic/plastic material combinations, the presence of fiber reinforcements (glass, carbon) has bigger influence on the wear of the gears than surface roughness [8].

**MEASURING AND CONTROLLING TEMPERATURE**

When conducting gear testing, it is important to have the correct temperature measurements. Here it is crucial that not the ambient...
temperature is measured, but the relevant material temperature. The most convenient way to measure gear temperatures is using a thermal camera; however, other options are possible (thermal couples). But measuring with a thermal camera is only possible for dry running gears. If oil or grease is used for lubrication, other methods must be used for temperature measurements (measuring oil temperature for instance).

Based on the design of the test rig, there are usually two options to measure gear temperatures: from the top (Figure 1a) and from the side (Figure 1b). If the temperatures are measured from the top, then the flank temperature is measured on the working flank, and the root temperature on the non-working flank. When measuring temperatures from the side, the root temperature can be measured just below the root diameter.

Depending on the expected failure mode (root, flank, wear), the corresponding root and/or flank temperature (see Table 6) must be controlled.

To get reliable fatigue data, a climate chamber is necessary to control the temperatures. If the goal is to measure fatigue data at room temperature (20°C), then it might be necessary to set the ambient temperature in the chamber to e.g. -10°C (or lower, depending on the testing conditions), so the temperature control system must also enable negative temperatures.

**COMPARISON OF CALCULATED TOOTH ROOT STRESSES**

The calculated permissible root/flank stresses are calculation-method dependent. An example below will show how much the results can differ if different calculation methods are used for calculating the permissible stresses and for the calculation of the lifetime. The comparison is done for the following calculation methods: VDI 2736 (YF-C), VDI 2545 (YF-C), and VDI 2545 (YF-B) [11]. The difference between methods YF-B and YF-C is in the location of the applied bending force. For method C, the force is applied at the tip of the tooth, whereas for method B the force is applied at the point of single tooth contact. For a manual calculation, method C is much easier to apply, but method B is potentially more accurate. Both VDI guidelines originally follow method C, so applying method B is a modification of the VDI method.

Table 3 shows the calculated safety factors for size 1 test gears made from POM. If permissible stresses and safety factors are calculated with the same calculation method, then the safety factors are 1 (results

<table>
<thead>
<tr>
<th>Temperature (°C)</th>
<th>50°C</th>
<th>50°C</th>
<th>70°C</th>
<th>70°C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cycles to failure</td>
<td>9.00x10⁵</td>
<td>1.35x10⁶</td>
<td>5.00x10⁵</td>
<td>8.00x10⁵</td>
</tr>
<tr>
<td>POM (VDI 2736)</td>
<td>90.2</td>
<td>86.4</td>
<td>85</td>
<td>79.6</td>
</tr>
<tr>
<td>Delrin 100</td>
<td>49.8</td>
<td>48.2</td>
<td>54.8</td>
<td>44.8</td>
</tr>
<tr>
<td>Delrin 100P</td>
<td>84.6</td>
<td>72.6</td>
<td>84.6</td>
<td>72.6</td>
</tr>
</tbody>
</table>

Table 4: Permissible tooth root stresses for POM.

<table>
<thead>
<tr>
<th>Temperature (°C)</th>
<th>20°C</th>
<th>20°C</th>
<th>80°C</th>
<th>80°C</th>
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</thead>
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<tr>
<td>Cycles to failure</td>
<td>5.00x10⁵</td>
<td>1.00x10⁶</td>
<td>5.00x10⁵</td>
<td>1.00x10⁶</td>
</tr>
<tr>
<td>PA66 (VDI 2736)</td>
<td>112</td>
<td>98.0</td>
<td>57.2</td>
<td>50.0</td>
</tr>
<tr>
<td>Ultradur A4H</td>
<td>51.4</td>
<td>44.3</td>
<td>44.8</td>
<td>43.2</td>
</tr>
</tbody>
</table>

Table 5: Permissible tooth root stresses for PA66.
on the diagonal from top left to bottom right). But if methods are different, then the results deviate from 1. Safety factor below < 1 indicates that the calculated lifetime is smaller than the actual achievable lifetime (254 h), so the results are on the safe side. In contrary, if the safety factors are > 1, then the calculated lifetime is higher than the achievable lifetime. In worst-case conditions for this given example, the calculated lifetime is by a factor 2.1 higher than the actual achievable lifetime.

**COMPARISON OF MEASURED PERMISSIBLE TOOTH ROOT STRESSES**

The VDI guidelines contain permissible stress data for various plastic materials. Compared to measured, values this data fits more or less OK to the measurements.

Table 4 shows the comparison of the tooth root stresses for different POM materials at given temperature and cycles to failure. It can be seen that the values for Delrin 100 deviate significantly from the VDI and Delrin 100P measurements. The values for VDI and Delrin 100P are very similar.

Table 5 shows the permissible tooth root stresses for material PA66. At 20° C, the difference between the materials is factor 2, but at 80° C, the difference is much smaller. Still, however, the VDI material has higher permissible stresses.

The results from Tables 4 and 5 show that there can be big differences between the measurements and the values in the VDI. The problem is that there is hardly any description available on how the permissible stresses in the VDI were measured. But it seems that, with proper testing procedures, it is possible to achieve the VDI measured values for POM material.

With dry running steel/plastic gears, there are a lot of factors that influence the test results (quality of the gears, temperature of the gears, injection molding process, humidity, different test rig layouts, etc.). Consequently, there will always be differences in measured permissible stresses, however they should be within the reasonable limits.

The data contained in the guidelines should only be used if no other data is available. The closer the measured data fits to the real material used, the more reliable the results will be.

**DATA EXTRAPOLATION**

The permissible stresses for plastic materials are usually measured until \(2 \cdot 10^6\) or \(5 \cdot 10^6\) cycles. However, if the S-N curves are not measured until \(10^{10}\) cycles [12], it is not possible (in KISSsoft) to calculate lifetime safety factors with defined load spectrum or if the number of cycles exceed the measured number of cycles. To make load spectrum calculation possible, measured data should be extrapolated according to some principles.

To our knowledge, there are no rules available for extrapolating data from plastic gear tests.

Several proposals were evaluated at KISSsoft. The one shown above was selected as the best. We are proposing a combination of Corten/Dolan approach and Haibach approach.

From the last measured point, we extrapolate the curve with an average slope until \(10^{10}\) cycles. Between \(10^{10}\) and \(10^{30}\) cycles, extrapolation is performed with half the calculated average slope. An example is shown in Figure 2.
ANALYZING TEST RESULTS
Table 6 shows an overview of the parameters that must be measured (or monitored) during gear testing. The measured parameters depend on the desired outcome of the tests (root fatigue, flank fatigue, wear, etc.). A software program was developed that calculates the corresponding S-N curves based on the results from the gear fatigue measurements. Test results can be imported from a txt file. Figure 3 shows the input fields for the measured data, calculation settings, and test gear geometry. Torque and cycles to failure must be defined on the failed gear.

The extrapolation of the measured points is also possible. As an additional option, the measured S-N curve at lowest temperature can be extended to even lower temperatures using a factor to scale the permissible stresses. Extrapolation can be independently carried out for different lubrication regimes (dry, grease, oil).

Figure 4 shows the S-N curves that were calculated from the measured points. The values at 20°C were extended from 32°C with a factor of 1.2.

CONCLUSIONS
The number of materials for which gear fatigue data is available is limited. To measure fatigue data for new materials, gear testing is necessary. However, gear testing can be expensive and time consuming, so it should be done correctly, so the measured data can be used for gear calculations. This paper gives a brief overview of the measuring and calculations procedures for plastic gear testing described in the VDI 2736-4.

When measuring cycles to failure, it is important to scale the number of cycles to 10 percent (or lower) damage probability. As shown in Table 1, this results in lowering the average number of cycles for 30 percent, depending on the deviation of the data. When comparing gear-fatigue data, it must be compared at the same damage probability; otherwise this can cause significant inconsistencies.

If designing test gears for the specific test rig, a VDI 2736-4 defined gear geometry should be used. If this is not possible (due to the load capacity or center distances), gears with comparable properties (similar theoretical contact ratio and contact ratio under load, similar specific sliding) should be used. This should keep the measured data comparable with the measurements on the VDI defined gear geometries.

The same calculation method should be used for the calculation of the permissible fatigue stresses from the measured data and for the calculation of safety factors. If this is not the case, then significant differences in lifetime can occur (resulting in under- or over-dimensioned gears), as shown in Table 3.

Fatigue data for plastic gears is usually measured until 2·10^6 or 5·10^6 cycles. The reason is to obtain results in reasonable time. But to calculate lifetime safety factor with load spectrum, it is necessary to have fatigue data available until 10^10. To make load spectrum calculation possible, measured data should be extrapolated. A proposed extrapolation method for plastic gears is briefly discussed (Figure 2).

To help test engineers, a program was developed based on the VDI 2736-4. The measured data is introduced, and then the software performs the evaluation of the test results automatically and calculates the permissible root/flank stresses. This information is documented in a text file, which can directly be used by a calculation tool for the calculation of plastic gears according to the VDI 2736-2.

REFERENCES:
7. Internal KISSsoft documents.
12. Internal KISSsoft documents.

Fast, Flexible Bevel Gear Cutting

Companies interested in increasing capacity and agility to their precision gearing operations will be interested in the new Gleason bevel gear cutting, lapping, and testing machines. Here’s one company’s experience.

From Gear Solutions staff reports

WELL OVER 100 BEVEL GEAR CUTTING MACHINES crowded this Chicago factory floor just a few years ago, all busy producing spiral bevel gear sets by the thousands. Now those machines are gone. If you guessed that this is yet another example of a rust belt manufacturer losing its competitive edge, or shipping jobs offshore, you’d be wrong. In fact, bevel gear manufacturing at this US Gear facility of AxleTech International is not only alive and well, but running two and a half shifts a day, five days a week and producing more gear sets than ever for the high-performance gearing and aftermarket axles that the company is known for. If ever there was a testament to the power of new technology this is it, says US Gear Manufacturing Engineering Manager Mike Lobaugh, pointing to the cell of three Gleason Phoenix 600HC bevel gear cutting machines now doing the work of the 100 machines they replaced.

THE HIGH COST OF WRENCH AND IDLE TIME

“This was a ‘sea of green,’” Lobaugh said, alluding to the distinctive green color of the Gleason “five-cut” mechanical machines that once occupied much of this factory floor. “But the old tried-and-true five-cut face milling process (requiring two machines for ring gear roughing and finishing, three machines for pinion roughing and finishing of the convex and concave sides of the tooth) becomes a liability when today’s customers expect shorter lead times on smaller batches of our fast-growing product families.”

According to Lobaugh, it could take an experienced machine operator as long as two days of tedious, labor-intensive ‘wrenching,’ cutting parts and making adjustments just to change over his five machines from one gear set variant to another. Then, once the first ring gear was cut, the machines could sit for days waiting for the ring gear to return from heat treat. Only then could the pinion be cut to match the gear, and full-scale production begin.

“This process was acceptable when lot sizes were in the many hundreds or thousands,” Lobaugh said. “But today we need to accommodate batches of as few as 25 from dozens of different gear set product families and gear ratios. Fortunately, what once took many days now can be done in just a few hours with our new Gleason machines.”

FROM FIVE-CUT TO PHOENIX

US Gear began making the transition from its older Gleason five-cut machines in 2011, when it installed two Gleason Phoenix 600HC machines. Based on the success of these two machines, a third was installed in late 2016, enabling US Gear to move all of its spiral bevel gear set production to the three-machine Phoenix cell. These gears range in size from 2 inches to 23 inches in diameter, and include hot-selling products like their new Ford Super 8.8-inch IRS gear set.

“The difference between changing over a five-cut versus a Phoenix is like night and day,” Lobaugh said. “Steps that used to require hours of wrench time now reside in a gear summary and the CNC. Once you’ve verified the first part, every run thereafter is plug-and-play. With the mechanical machines, you started from scratch each and every time.”

Lobaugh also said the Phoenix machines have cut cycle times in half using Gleason’s Dry Power Cutting process. The 600HC’s clean work chamber ensures that high volumes of hot chips produced are easily collected.
away from the cutting zone. It’s an ideal platform for Gleason’s highly productive Pentac Plus, a cutter system that couples Gleason’s AlCroNite Pro-coated carbide stick blades with an innovative design that prevents the chip packing common in dry, high speed cutting applications. The cutter systems are also easy to build, using a Gleason CB cutter build machine. The CB is used to build and “true” different diameter Pentac Plus cutters in as little as 45 minutes. Blade re-sharpening and re-coating is performed at the Gleason Cutting Tools facility in Loves Park, Illinois.

Finally, The Phoenix machines make parts load/unload and tooling changeovers much faster and less fatiguing and significantly reduce the non-productive time typically required to perform these operations. Their unique monolithic column puts the pivoting cutter spindle and work spindle in close proximity to the operator, and the latest Gleason bevel gear quick change tooling greatly eliminates costly nonproductive time.

Gear sets then are lapped and tested on two Gleason 600HTL Turbo Lappers and two Gleason 600HTT Turbo Testers that have also replaced mechanical counterparts.

“Turbo Lapping is of course much faster, but what’s really significant is the Turbo Tester,” Lobaugh said. “The old roll testers only checked contact pattern, and the operator had to rely a lot on ‘tribal knowledge’ — and what he sees and hears. Now, we’ve got a lot more actual data to work with, such as single-flank measurement of transmission error.”

This inspection data can be easily networked with Gleason’s Engineering and Manufacturing System (GEMS) and its CAGE Gear Design Software to calculate machine corrections and make summary changes. Lobaugh said the new machine selection process was weighted not only on machine capabilities but also on US Gear’s 50-year association with Gleason.

“AxleTech gears operate in some of the most demanding applications including off-highway and performance coupled with the high-mix nature of our business, which necessitates lots of changeovers. We therefore needed a partner that we could rely on and Gleason has been that company for us for 50 years.”

— Bill Gryzenia / Chief Executive Officer / AxleTech International

“There’s no learning curve,” he said. “And their service is exceptional. If there’s an issue we pick up the phone. We know them, and they know us.”

Mark Kay, AxleTech senior director of operations, agreed.

“In 2017 with increased internal and external demand for our spiral bevel gear sets and to support our long-term growth strategy, we again turned to Gleason to provide the technology and equipment we need to expand our business,” he said. “I am very pleased with the results and Gleason’s commitment and look forward to continuing our partnership.”

Gleason, Phoenix, AlCroNite, Pentac, Power Cutting, Turbo Tester, Turbo Lapper, GEMS, and CAGE are trademarks of Gleason.
EMUGE OFFERS STEP DRILLS FOR THREADING, CIRCLE SEGMENT CUTTERS

Recognizing that most threaded holes require some type of chamfer, Emuge has developed a comprehensive carbide step drill offering of stocked cut and form thread drill diameters with varying step lengths. Customers can use a convenient on-line portal to select the correct drilling diameter for UNC, UNF, and M & MF thread forms. The web portal allows customers to view the range of step lengths available. Stock levels are available for immediate shipment, as well as quantities available from Emuge’s quick-shipment program with less than four-week delivery. EF Drill-C tools are coolant-fed carbide designs, and customers can order up to 10 or more pieces at the same price. They are great for short run jobs or production threading.

The advent of 5-Axis machining centers has opened up a new horizon for shops that are tasked with producing complex surfaces. When machining components for injection molds and turbine blades, ball nose and torus nose carbide cutters are the traditional method for milling large smooth surfaces. Emuge has developed a full line of tools for these 5-axis milling applications called circle segment cutters. These include solid carbide end mills with a taper form that can simulate a 250 to 500 mm radius cutter. Think of the difference between using a 1-inch diameter torus nose end mill as opposed to the same end mill with a 10-inch diameter.

Circle segment and taper form cutters can reduce cycle times by 70 percent or more. (Courtesy: Emuge)

KLÜBER LUBRICATION INTRODUCES SPECIALTY GREASE

Klüber Lubrication, a worldwide manufacturer of specialty lubricants, introduces Hydrokapilla NBU 20 HFE US, designed for the lubrication of plain and rolling bearings, and chains.

Hydrokapilla NBU 20 HFE US is designed for applications that experience high humidity and/or aggressive media. The grease can be used in drive and carrying chains for wet amusement park rides, carrying chains for textile steamers, and any chains subject to aggressive environments (hot water, chlorinated water, steam, alkaline and acidic solutions).

Hydrokapilla NBU 20 HFE US, with its excellent penetration into pin and bushings, resists corrosion and prevents chain seizure. It can be applied manually, via brush, or by an automatic spraying system.

Klüber Lubrication has approximately 2,000 employees in more than 30 countries.

FOR MORE INFORMATION: klueber.com

KISSSOFT MODEL CREATION

KISSsoft Release 03/2017 has a new function, the “Groups box,” which lists a selection of predefined gear stages. These gear stages can be added to the model tree structure and combined in any way you like. Gear stages such as a Ravigneaux set or a Wolfrom set are also available, to make it easier to model complex wind-turbine gearboxes or shifting transmissions. You can also define your own gear stages and save them in the software. This saves a great deal of valuable time when you’re modeling complex drive systems.

The way in which the model tree structure is handled in KISSsys has also been improved, so that changes (deletions, renamings etc.) can be performed at a later point in time, without any restriction.

FOR MORE INFORMATION: kisssoft.com

GLEASON METROLOGY SYSTEMS INTRODUCES NEW GEAR ROLLING SYSTEM WITH LASER

The Gear Rolling System with Laser (GRSL) revolutionizes gear measuring and sets a new standard for throughput where high speed, high volume testing is required. It provides both double-flank roll testing as well as analytical index and involute measurement on all teeth during the same inspection cycle in a matter of seconds.

The Gear Rolling System with Laser provides both double-flank roll testing as well as analytical index and involute measurement on all teeth during the same inspection cycle in a matter of seconds. (Courtesy: Gleason)
This new technology is available in manual, semi-automatic, or fully automatic configurations depending on the needs of the customer. The index and involute measurements are analyzed using Gleason Metrology’s GAMATM gear analysis software, which allows the operator to see common charting between the GMS analytical inspection machine and the GRLS gear rolling system. With GAMA, more than 50 analysis packages are available for customers with all major industry standards such as AGMA, DIN, ISO, etc., along with customer-specific analysis requirements developed for industry in the GAMA platform.

This patent-pending design is unparalleled in inspection speed and capability. It measures external, cylindrical gears up to 250 mm in diameter and in a range of 0.4 to 7.2 module. The double-monitor option provides a simple view of ongoing trends in the high-speed inspection environment where one monitor can display results of several hundred parts inspected over time while the other can show real time results of the gear being inspected.

FOR MORE INFORMATION: gleason.com

The new Vero-S WDB clamping technology modular system transfers the efficiency levels of the Schunk Vero-S quick-change pallet system specifically to machine directly clamped workpieces. By using clamping pillars with a modular structure, freely molded parts and other workpieces can be clamped directly on the machine table, without the need of additional equipment.

The modular system consists of the Vero-S WDB basic modules, clamping modules, and stacking modules that all can be freely combined. The stacking modules are 30mm and 50mm, so that the height of the clamping pillars can be varied in intervals of 10mm to 80mm without needing any special solutions. Height differences can be bridged using the stepless, adjustable, hydraulically clamped Vero-S WDA compensation element.

FEATURES AND BENEFITS
All interfaces have a taper centering mechanism that is free from play and guarantees a repeat accuracy of < 0.005mm between the individual components.

The clamping modules can be actuated and the workpiece can be monitored by an integrated air feed-through, without the need for an external media supply.

The actual workpiece clamping takes place by spring force, and is a self-locking, form-fit process.

The workpieces remain safely clamped even if the air pressure suddenly drops.

A pneumatic system of six bar opens the clamping module.

FOR MORE INFORMATION: schunk.com

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With a re-engineered design, the Seco Perfomax indexable insert drill delivers higher drilling parameters and longer tool life as well as exceptional chip control and evacuation. The drill’s innovative features include new flute designs with special anti-friction surfaces, and laser-hardened fronts for added strength, stability, and accuracy.

Perfomax flutes feature improved helix angles along with smoother chip flute exits and Seco’s engineered wave pattern that minimizes contact between chips and flute surfaces. The drill generates shorter chips that evacuate quickly and easily to significantly reduce the risk of chip jamming for higher application security.

Seco laser hardens the fronts of Perfomax’s flutes for up to 140 percent longer drill body tool life. A hardness of HRC 60 allows the drill to withstand chip erosion for much longer periods of time. The re-engineered design of the Perfomax drill also features a bigger radius at the bottom of the drill’s insert pockets for added rigidity.

In tandem with Perfomax, Seco’s DS2050 and DS4050 insert grades are especially well suited for heat-resistant materials such as titanium and titanium alloys. The grades enhance productivity and extend tool life thanks to recently developed free-cutting MP and MC geometries. DS2050 are inserts for periphery cutting, and DS4050 are those for center cutting.

Perfomax drill bodies are available in diameters ranging from 0.594” to 2.375” (15 mm to 59 mm); in length-to-diameter ratios of 2xD, 3xD, 4xD, and 5xD; and in most spindle interfaces.

FOR MORE INFORMATION:
secotools.com

SANDVIK’S HIGH-FEED SIDE MILLING CUTTERS ADDRESS PRODUCTIVITY

To enhance milling performance on ISO S materials, cutting tool, and tooling system specialist Sandvik Coromant is introducing a series of end mills featuring unique geometries and grades. The CoroMill® Plura HFS (High-feed Side Milling) ISO S cutters deliver reliable and productive results on workpieces made from titanium and nickel-based alloys, bringing benefits to aerospace engine and frame applications.

To help address the predicted growth in the aerospace industry in the coming years, the CoroMill® Plura HFS range comprises two end mill families optimized for titanium alloys, and one for nickel alloys. As chip evacuation and heat are specific challenges when machining titanium, Sandvik Coromant has developed a solid version for normal chip evacuation.

Unmatched. Steve and the men and women he works with make sure no shop surpasses Schafer Industries’ ability to generate and grind the quietest precision gears. We’ve set the quality benchmark high for spur and helical gears and cutting straight bevel gears. Orders ship on time with zero back log, too. Steve says his manufacturing team is ready to prove its expertise to you. Let’s meet.
evacuation conditions, and another featuring internal coolant and a new cooling booster (patent pending) for optimum swarf and temperature control.

The end mills for titanium are available in GC1745 grade, which is based on a tough, fine-grained (sub-micron), cemented-carbide substrate with sharp, controlled edges for tough milling operations. Furthermore, a new multi-layer coating that contains silicon provides excellent wear resistance and low thermal conductivity. The geometry of the cutters is based on a six-flute concept with no center cut and uneven tooth pitch. Additionally, the core dimension has been optimized for higher stiffness in titanium alloys, while the corner radius, rake angle, and relief are all designed specifically for machining these challenging materials.

For nickel alloys, grade GC1710 is deployed, which also features sharp, controlled cutting edges. A hard, wear-resistant, fine-grained substrate is optimized to resist high working loads when machining hard, highly adhesive, work-hardened materials such as aged Inconel 718. Here, a new coating produced with innovative HIPIMS (high power impulse magnetron sputtering) technology also offers adhesion-reducing properties to avoid the formation of BUE (built up edge) and increase tool life.

“The new cutters are designed to offer high-feed side milling with large axial depths of cut (ap) and low radial depths of cut (ae) along with a controlled maximum chip thickness so that the cutting forces are managed and provide a smooth cutting action,” said Tiziana Pro, Global Product Manager Solid End Mills at Sandvik Coromant. “The result is twofold: Increased productivity provides higher output, while greater tool life and reliability reduce scrap rates in what are typically high-value components. Additional customer benefits include reduced tool cost per component and greater safety levels.”

Target aerospace components include titanium wings and pylon parts, as well as engine cases made from Inconel 718. Applications in sectors such as oil and gas, medical and automotive, where titanium and nickel-based alloys are becoming increasingly prevalent, will also benefit.

To highlight the potential gains of this offer, a customer trial was performed, involving an LPT (low-pressure turbine) case made from aged Waspaloy 420 nickel-based alloy. Using a horizontal machining center, axial depth of cut was increased, and radial depth of cut was reduced (high radial forces are known to create deflection issues). Comparing a 12 mm diameter CoroMill® Plura HFS end mill against a competitor cutter of the same size, metal removal rate increased substantially, leading to an impressive 198 percent increase in productivity. As a result of this success, the customer has committed to ordering the new cutters.

FOR MORE INFORMATION:
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The new CoroMill® Plura HFS ISO S cutters from Sandvik Coromant are ideal for machining aerospace components made from titanium and nickel-based alloys. (Courtesy: Sandvik Coromant)
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Butler Gear Enterprises, LLC, is under new ownership, correct?

Correct, we are happy to announce we are under new ownership as of February 15, 2017. Our new president, Andrew Kattula, and his staff have made significant positive changes in our processes as well as updating machinery as needed. We have become a more efficient company from RFQ to paid invoice.

What’s a typical day like at Butler Gear Enterprises, LLC?

Butler Gear Enterprises, LLC is a fun place to work, so coming to work is a joy most days; other days it can be challenging. What I mean when I say challenging: busy non-stop, quoting, customer service, and putting out fictitious fires here and there and can consume the day. Typical day is busy, which is a great problem to have.

I get involved with many areas in the front office as well as the shop. My main job is quoting, but I also do engineering and reverse engineering. I work very closely with Larry, the plant manager, to help guide the employees out on the shop floor as well. I love designing parts, finding new ways to make things work; it’s where my passion falls. But all of those things put together — along with answering questions, customers calling, providing total customer service — it’s always something. This is what I have been doing for 44 years. I really enjoy all of it.

You talked about guiding the folks on the floor — are they happy to see you coming? Or do they get that trapped look on their faces?

I am pretty happy when an employee on the floor asks for my advice. I’ve been doing this since 1974; I’ve been around for a while. I implore them to ask me questions or to challenge me, “Tom, this isn’t working. What can I do to make it work better? Faster?” Or ask me my opinion on something. Whether they do it or not, both of us learn. It’s always a learning experience with the guidance of the people out on the floor. Larry handles all of the production and is very intelligent, but I love to offer my input, and people respect that. I like coming up with solutions; that’s my passion.

What products and services does Butler Gear offer?

We’re a complete gear job shop. We offer tooth cutting, internal and external. We do reverse engineering, general machining. When a customer comes in with a PO, we’ll deliver a finished product, from teeth cutting only to supply complete. We provide the whole gamut.

We also have a lot of control over quality. Because we offer so many services, most of the parts stay under our roof, so we have complete control of the part. We deliver quality parts, on-time — this is Butler Gear’s passion — to please the customer with exceptional customer service and deliver top-notch quality parts.

What is Butler Gear doing to advance the gear industry?

We are constantly looking at inventing new ways to manufacture. We have our head on a swivel, looking for newer, more advanced machines to increase the class of the gears we manufacture, the productivity, and overall efficiency in the entire company. Butler Gear is not afraid of investing into newer machines to make our customers happy. After all, it really is all about the customer, right?

What are some of Butler Gear’s proudest moments?

Acquiring our big shaper, separating us in the industry with a niche capability. It’s a rare machine to have. Your typical gear company does not have anything close to this machine. It was amazing that we were able to pull it off and to acquire this unit at the time we did. It’s a Fellows Model 88-16 heavy-duty gear shaper.

This was a difficult question because everyone gets excited when you get big orders, bring new customers, etc. Those are great moments, also.

What sets Butler Gear Enterprises, LLC apart when it comes to what you can offer a customer?

Diversification. We’re not scared to step out of the box to produce machine adaptions, or reverse-engineering projects, things that some companies wouldn’t feel good about or want to do. We thrive in these areas and take great pride in that. I love inventing and making machine adaptions. We can do projects for customers that the competition won’t.

Where do you see Butler Gear Enterprises, LLC in 10 years?

I think there are a lot of people who say the same thing: advancement in technology, organic growth, ability to service the entire industry, as well as other gear companies, and to continue tuning our efficiency in all of processes. We are paving the road to success with the new ownership’s increased capabilities as they continue to introduce J&J Spring Enterprises and Spring & Wire Liquidators to our consumer base. In 10 years, I see Butler Gear Enterprises, LLC offering much, much more than just gearing and machining needs. We have a bright future ahead of us.

Would you like to add anything we haven’t covered?

I’m under a lot less pressure than I used to be, so I’m happy about that. But then again, I miss sticking my nose into everything. I guess once you’re a gear nut, you’re always a gear nut.
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