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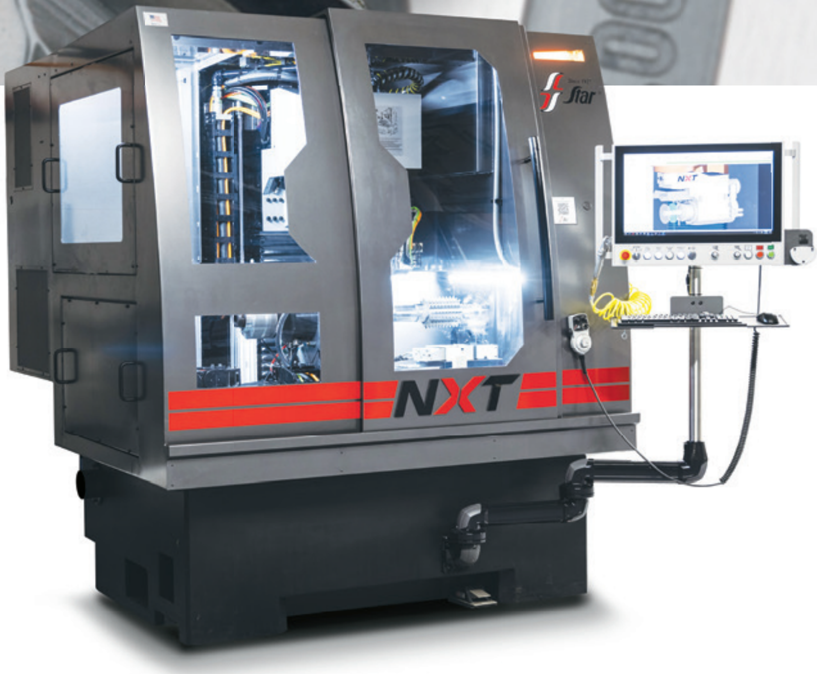
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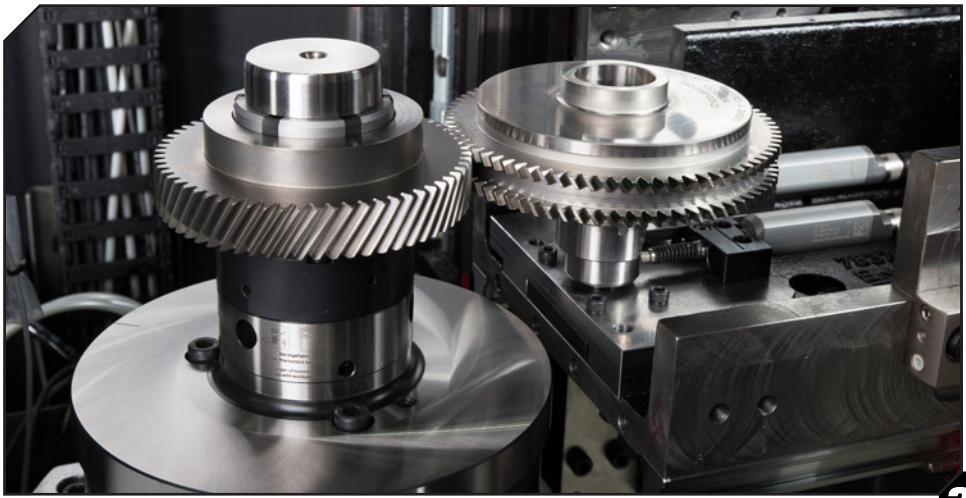

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features

technical

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THE GEAR INDUSTRY'S INFORMATION SOURCE

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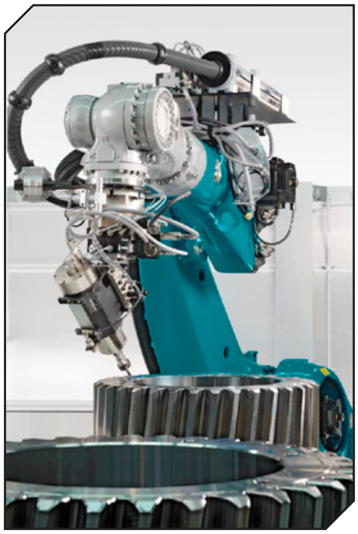
GT Revolutions

Metal Cutting Robots



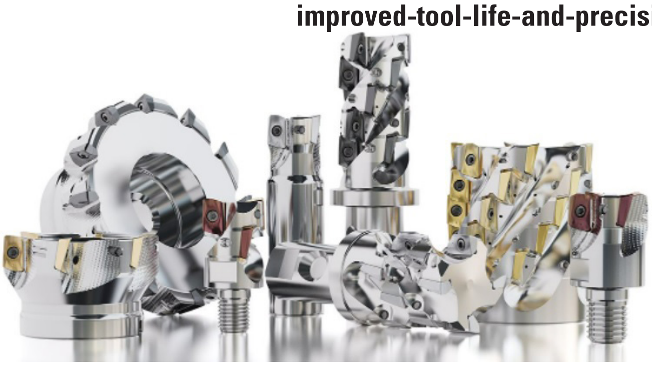
Kadia has been designing deburring robot cells based on 6-axis industrial robots for many years. In the meantime, a new trend is now emerging, solutions with an even higher value-added component, i.e., with general machining processes such as milling, drilling or thread cutting. The robot is thus no longer just part of a deburring machine.

geartechnology.com/blogs/4-revolutions/post/29907-kadia-offers-deburring-robot-cells

Cutting Tools Improve Tool Life and Precision

Cutting tools are basic to gear manufacturing. Whether it's a hob, broach, shaper cutter, or skiving tool, the object of cutting tools remains the same: material removal that is fast, accurate, and cost-effective. The field tends to evolve gradually in the machines, materials, and coatings that make cutting tools even more useful. Reliable cutting tools are essential to production-process efficiency, and recent solutions from Kennametal, Star SU, and Seco offer improved tool life and precision.

geartechnology.com/blogs/4-revolutions/post/29884-recent-cutting-tools-from-kennametal-star-su-and-seco-offer-improved-tool-life-and-precision



Event Spotlight: WZL Gear Conference—USA

The Eighth WZL Gear Conference—USA is being hosted by Reishauer Corp. and will provide the opportunity for North American companies to connect with WZL and learn about current research activities. For more than 50 years the annual WZL Gear Conference in Aachen, Germany, has been fostering technical collaboration and communication among the members of the WZL Gear Research Circle. The two-day conference is devoted exclusively to the presentation of the latest research in gear design, manufacturing and testing. The event takes place July 20–22, 2022. Speakers include Christian Brecher and Thomas Bergs.

wzl.reishauer.com

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Michael Goldstein founded Gear Technology in 1984 and served as Publisher and Editor-in-Chief from 1984 through 2019. Thanks to his efforts. the Michael Goldstein Gear Technology Library. the largest collection of gear knowledge available anywhere. will remain a free and open resource for the gear industry. More than 36 years' worth of technical articles can be found online at *www.geartechnology.com*. Michael continues working with the magazine in a consulting role and can be reached via e-mail at *michael@geartechnology.com*.


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Helios Gear

INTRODUCES DEBURRING MACHINE

The Helios TM 250-CNC machine from Tecnomacchine brings together two fundamental strategies that empower manufacturers to deburr gears more productively. "For many job shops, smaller lot sizes and a higher mix of part numbers make a fully automated solution difficult to justify," said David Harroun, vice president of Helios Gear Products. "But with the right feature set, such as the TM 250-CNC, manufacturers can deburr gears more profitably than ever before."

The TM 250-CNC evolves from the proven "250 strategy," which uses a rotary table with twin work spindles to minimize idle machine time, thus maximizing potential productivity. Machining is performed on one work spindle while the other is simultaneously unloaded and loaded manually. Interestingly, this configuration allows the machine to easily adapt in the future to an automatic loading strategy via robot, cobot (collaborative robot), pick-and-place, or other systems. The TM 250 shifts the paradigm from a fully manual deburring process to one equipped with a machine tool that allows next-level quality and productivity.

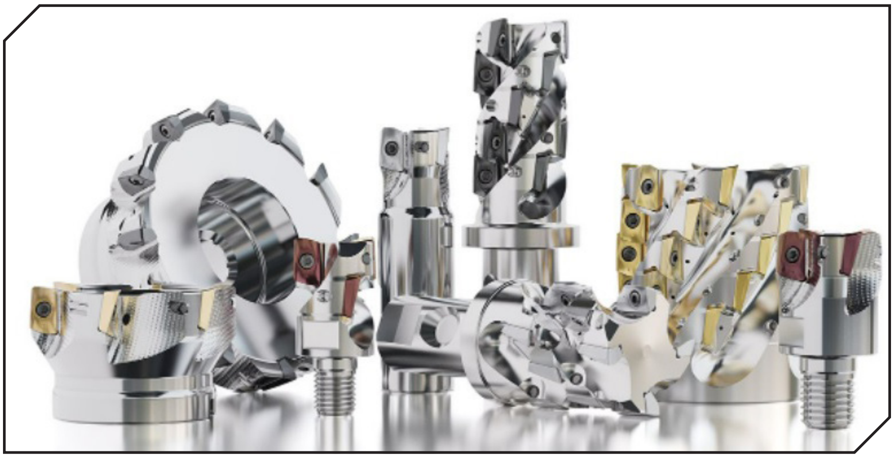
Secondly, the TM 250-CNC adds a "CNC strategy," which enables setup personnel to store all tool spindle parameters. These include radial, axial, and tangential positions, as well as tool speed, direction, and pressure. Crucially, this allows a single setup per part number, which can be consistently and reliably repeated simply by loading a program. With CNC ability, the TM 250 becomes a versatile workhorse for any gear manufacturers with a high mix of part types.

"For manufacturers of medium- to high-mix gear types, the TM 250-CNC offers a versatile, productive solution for chamfer-deburring," said Adam Gimpert, president of Helios Gear Products. By using this machine tool, quality can be improved along with reliability and consistency. Lastly, because of the 250's rotary table feature, the machine can easily integrate automation in the future, making it a sound investment.

heliosgearproduct.com

Seco Milling Cutters

REDUCE TOOLING INVENTORY COSTS



Manufacturers who look for versatility and precision in machining can pair up Seco Turbo 16 square shoulder milling cutters and Helical Turbo 16 milling cutters to reduce tooling inventories and costs. Both series offer high material removal rates in steel, stainless steel, cast iron, nonferrous metals, superalloys and titanium. Scannable Data Matrix tags on cutter inserts store product and batch information that the new Seco Assistant app can read.

Versatile Turbo 16 square shoulder milling cutters feature optimized insert pocket angles that enhance the cut and deliver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outstanding surface finishes. With a high helix angle for smoother workpiece entry and exit, these cutters also feature efficient chip evacuation. Lower cutting forces reduce power consumption, tool wear and noise levels.

"Manufacturers can achieve production efficiencies and enhance machining performance with these tools in virtually any material," said Michael Davies, global product manager, square shoulder milling. The eco-friendly design uses corrosion-resistant steel with no nickel coating.

Next-generation Helical Turbo 16 milling cutters combine top performance with ease of use, benefiting from a comprehensive range of Seco insert grades and geometries for exceptional material removal rates (MRR) and extended tool life. Larger cuts and higher feeds reduce cycle times and speed up production, with optimized coolant channels, flutes and cutting rakes for stable machining and optimal chip formation. Helical Turbo 16 ensure that lead and helix inserts are not mixed up when using large radii. The replacement of nickel cutter coatings with PVD increases sustainability.

"These features produce smoother cuts, increased tool life and process reliability for faster production. Helical Turbo 16 uses fewer inserts than equivalent tools that provide the same MRR," said Benoît Patriarca, global product manager helical milling.

secotools.com

Gleason

RELEASES SUB-MICRON GEAR INSPECTION SYSTEM

With its brand-new 300GMS nano Gear Metrology System, Gleason presents the capability of measuring gears at sub-micron level, executing advanced waviness analysis and evaluating gear noise using the most advanced analytical tools, make the 300GMS nano ideally suited to support automotive e-drive production with minimum noise requirements.

The 300GMS nano covers the full range of modern gear inspection capabilities, as well as fine pitch gear inspection and CMM measurement. With the 300GMS nano, users are now able to measure surface finish at sub-micron level with a skidless probe, analyze waviness for profile, lead and pitch, and execute noise analysis with sophisticated software tools. The 300GMS nano is equipped with high accuracy SP25 3D scanning probes with a broad range of styli, including its own stylus calibration library, meeting the most common roughness parameters as defined by DIN, ISO, ANSI and others. The 300GMS nano also integrates 3D measurement and analysis capabilities typically offered by a CMM.

In addition to gears and other rotary parts, the 300GMS nano inspects a wide range of gear cutting tools including hobs, gear skiving, shaper and shaving cutters, bevel cutter stick blades, most broaches and rack cutting tools. The system comes with Gleason's patented Advanced Operator Interface (AOI), which enables users to record video memos, leave voice messages, monitor environmental conditions and read bar and QR code information directly into the machine.

The machine uses the latest GAMA 3.2 applications and control software, fully compatible with Windows, connecting easily to customers' servers. GAMA also includes SPC data acquisition software and Gleason ConnectRemote Support Services.

The GRSL Gear Rolling System with integrated Laser Technology provides both double flank roll testing as well as laser inspection to provide measurement on all teeth in just seconds. GRSL is available in manual, semi-automatic

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- Selection of extensive data profiles with over 1000 signals
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- LHWebPlatform for increasing productivity using data-based reports
- Digital data exchange between gear cutting machine and gear inspection machine via GDE (Gear Data Exchange)

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or fully automatic configuration depending on the manufacturer's requirements and measures external cylindrical gears up to 250 mm in diameter and ranging from 0.4 to 7.2 module. It provides analytical output on all teeth for profile, lead and index with functional characteristics including nick detection, total composite variation, total runout, tooth to tooth average, DOP, average circular tooth thickness and more. Capabilities also include integrated gear noise analysis tools, making the GRSL ideally suited to meet automotive e-drive requirements.

Gleason's Closed Loop enables both the GMS Series and GRSL to send inspection results directly to Gleason production machines, including gear skiving, grinding and honing machines, for auto-correction of production variables without the need for operator intervention.

gleason.com

Verisurf and IBS Quality

OFFER PORTABLE INSPECTION SYSTEM

IBS Quality GmbH has launched the new PAM-System (Portable Automated Measurement System), a portable cobot-based 3D scanning and measurement solution. The launch is in collaboration with Verisurf Software, Inc., and its integrated 3D measurement software for automated quality inspection, reporting, scanning, and reverse engineering.

The PAM-System is a portable scanning, measurement, and inspection solution that uses cobots (collaborative robots) to augment repetitive user routines while adding flexibility, dexterity, and accuracy. All hardware (including fixtures for sensor attachments to the robot), software, installation, service, and support are included with the comprehensive system. The entire solution is self-contained on a specially designed inspection cart that can be easily rolled from the quality lab to the shop floor. The PAM-System can be used for first article inspection, volume production inspection, reverse engineering, and prototyping.

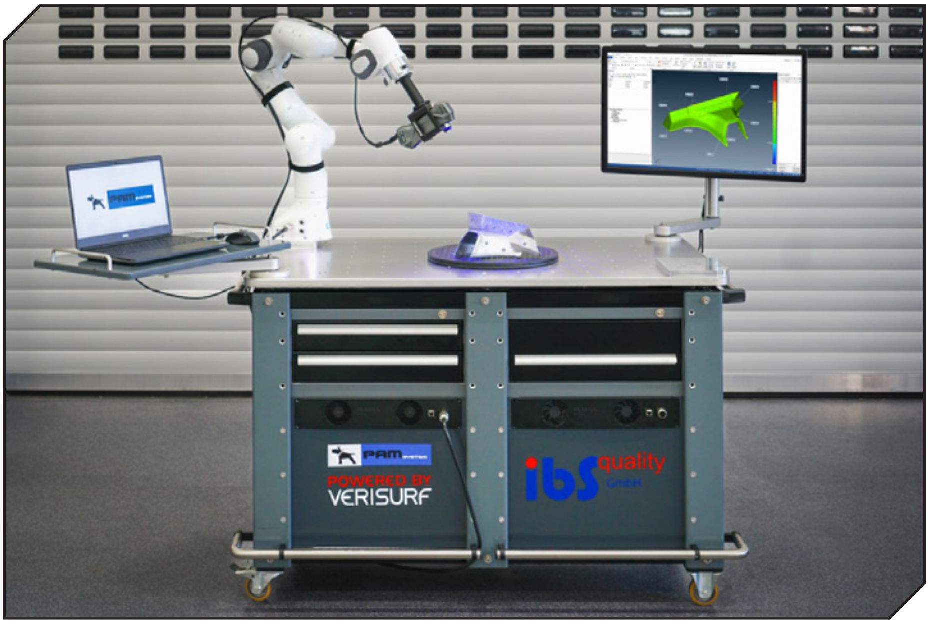
"We have designed the best of today's robotic, scanning, measurement, and inspection automation technologies into a single mobile solution; one that meets the practical needs of job shops and OEM manufacturers adapting to the realities of Industry 4.0," said Philipp Schmid, IBS Quality GmbH.

The PAM-System includes robotics automation from 11 Dynamics, 3D measurement, inspection, and reporting integration with Verisurf Software, and a choice of leading 3D laser scanner brands or other scanner technologies.

The integrated Verisurf 3D Scanning

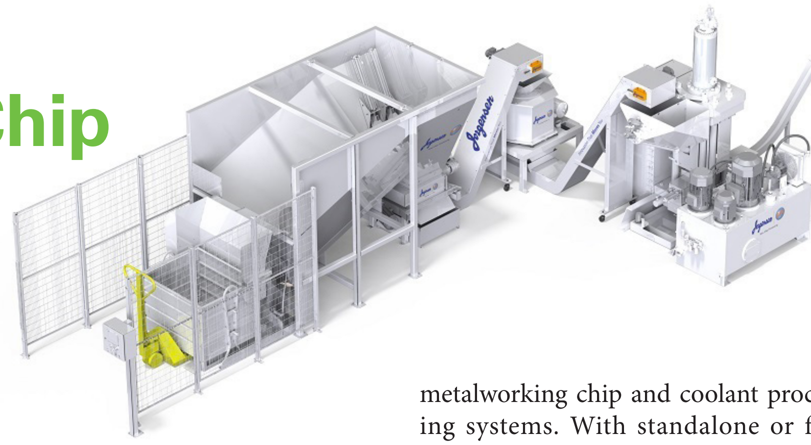
and Inspection application is built on a CAD platform and is rooted in Model-Based Definition (MBD). Inspection plans are easily created by importing any CAD model, using drawings, a golden part, or key features. Inspection plans can be used across the manufacturing enterprise on any CMM, providing repeatable process control. "The Verisurf workflow enhances productivity and enables repeatable process control, which is a key tenant of QA; the automated PAM-System supports both," said Nick Merrell, executive vice president of Verisurf.

ibs-quality.de
verisurf.com



Jorgensen Conveyor Total Chip Processing

BOOSTS RECYCLE VALUE



As a result of its recent partnership with S.F.H., Jorgensen Conveyor and Filtration Solutions now serves manufacturers as a single source for total metalworking chip and coolant processing systems. With standalone or fully integrated systems that transport, shred, wring and briquette chips and sludge, manufacturers can reduce chip volumes by as much as 90% and significantly increase their chip recycling value.

The volume reduction of aluminum, magnesium, copper, brass, steel and other types of metal chips represents a significant cost savings in labor, disposal, transport and storage. Jorgensen/SFH single and double-shaft shredders efficiently reduce chip size in preparation for centrifuging or briquetting. The volume reduction resulting from implementing a shredder also lessens the frequency of bin removal from a machine, giving a shop's employees additional time to focus on other more productive work. Jorgensen/SFH offer a variety of shredder size options constructed per a customer's material and desired chip output needs.

For maximum extraction of coolants and/or cutting oil, chip wringers from Jorgensen/SFH provide continuous, automatic operation at extremely low costs. Once collected, used coolant or oil can be processed through one of Jorgensen's filtration units and is put back into the coolant system for reuse. In terms of chips, a lower moisture content also further increases their recycling value.

Jorgensen/SFH chip compactors collect chips directly under conveyors or in centralized treatment areas. Chips and fines enter the press then are compacted under hydraulic pressure to form solid briquettes and separates out residual coolant fluids for reuse. The compactors are a robust, space-saving solution for briquetting a variety of ferrous or non-ferrous metals in the form of chips, sludges or dust.

jorgensenconveyors.com

Klingelnberg

P 152 LATEST ADDITION TO MEASURING CENTER FAMILY

The increasing cost pressure on large components for wind power requires new technologies that will enable proven principles for high-volume and mass production of smaller components to be transferred over to large components. To meet this need, Klingelnberg has developed a new precision measuring center. The P 152 is the latest addition to the family of Klingelnberg precision measuring centers. It is capable of measuring components with a maximum outside diameter of 1,520 mm and workpiece weights up to 8,000 kg with the usual precision. Despite this high workpiece weight, no special foundation is required because the Klingelnberg design engineers succeeded in extending the machine concept of the small and medium series to the large component dimensions. The inherently rigid machine bed with a 3-point support plays a key role here. The bed design and floor support have been so cleverly selected that even when loaded with workpiece weights of up to 8,000 kg, the angular position of the individual machine axes to each other does not change significantly. With its broad range of varied workpiece diameters and measurement tasks, this measuring device is also ideal for contract gear manufacturers. This ensures high precision across the entire component spectrum from smaller to larger gears, and also dimension and form measurements. The 3-point support enables the integration of an active vibration platform into the machine bed. This means that even the low-frequency vibrations from the shop floor can be safely absorbed.

Because the machine is isolated from these vibrations, the P 152 can be located on the normal shop floor without having to build an inherently vibration-isolated foundation. Despite this, it is ensured that all changes visible in the measuring results do come from the component and are not induced by the ambient conditions. The P 152 thus acts as a bridge for medium-sized gears such as planetary gears used in wind power and combines dimension, form, and position measurements with gear measurement, making the operating principles used in high-volume and mass production applicable to large gears as well. Form measurements, such as roundness and cylindricity measurements, are becoming increasingly important on machine elements. The P 152 offers all the possibilities of the Done-in-One principle in the mid-sized diameter range as well.

klingelnberg.com

Park Engineering

UPGRADES INSPECTION EQUIPMENT WITH LK METROLOGY

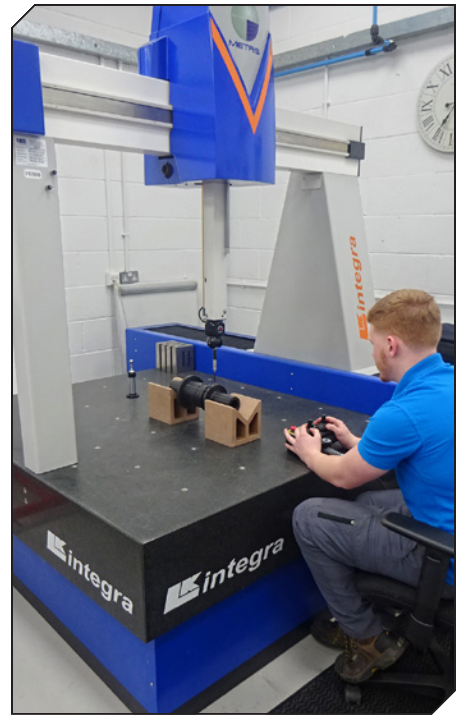
Park Engineering has significantly increased the size of parts it can machine and subsequently measure on a coordinate measuring machine (CMM). It follows the installation at the start of 2022 of two new machines, a Spanish-built Correa Fox 50 5-axis machining center having a 5.0 x 3.25 x 1.6 m working envelope, nearly 16 percent larger than the previous largest machine, and an ALTERA M 30.20.15 CMM with 3.0 x 2.0 x 1.5 m axis travels built by LK Metrology in Castle Donnington, UK. Compared with the largest of three LK machines previously operated by Park Engineering, it offers twice the measuring volume.

A member of the tooling division within the Hyde Group in the North of England, Park Engineering specializes in the design, manufacture, assembly and certification of tooling jigs and fixtures for the aerospace, nuclear, defense and automotive sectors, so the company is accustomed to dealing with large components. It also provides a subcontract machining service. When looking to invest in new technology to offer even larger capacity, there was a desire to inspect the bigger parts on a 'gold standard' CMM platform rather than rely on portable measuring systems, which may have a larger measuring volume but are usually lower in accuracy.

"The increase in speed and accuracy of the ALTERA M, which is around six times faster and 216% more accurate than the oldest of our previous CMMs, was a great selling point but we were also delighted to have a larger capacity CMM with a smaller overall footprint. It is down to the top-class LK machine design, incorporating the best of modern materials and technology," said Will Reeves, quality manager.

In day-to-day operation, the ALTERA M CMM supports the inspection of large parts with drawing tolerances of typically ± 50 µm but is sufficiently accurate to measure smaller parts down in the 10 µm region. Irrespective of tolerances, LK's software application CAMIO with its user-friendly operator interface allows CNC programs to be generated either online or off-line for the inspection of batch quantities, further reducing inspection time and cost, especially when CAD models are available.

"Across the broad spectrum of work packages, we have access to CAD models approximately 50-60% of the time, with the remainder being work-to-print. However, the provision of CAD models is something we are always encouraging our customers to provide, as it minimizes the risk of incorrect data being entered due to human error during the programming process," Reeves said.

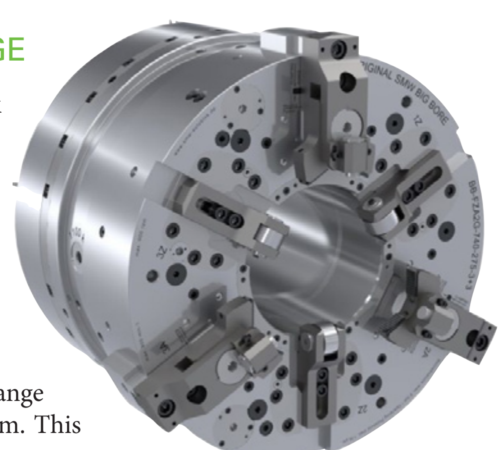


"LK's *CAMIO* software also saves time by automatically generating comprehensive reports that we can send to our customers with inspected components. The professional appearance of the documentation adds to the perceived professionalism of the service we provide."

lkmetrology.com

SMW Autoblok

FEATURES AIR CHUCK WITH EXTENDING STROKE RANGE



SMW Autoblok presents the BB-FZA2G pneumatic 3+3 sequencing air chuck featuring an extended stroke clamping range and three self-centering jaws with axial adjustment.

Ideal to handle the harsh demands of the Oil and Gas industry, BB-FZA2G allows for quick and easy setup of odd, shaped pipe for threading and re-threading. Thanks to incorporating three jaws with axial adjustment of the centering position while also using three compensating jaws with extra-long rapid and clamping stroke of 1.5, there is no need to re-shim the part. This enhances operator safety and greatly increases overall productivity.

The BB-FZA2G is available in 740, 800, and 920 mm to cover an extensive range of pipe diameter sizes, and features a large through hole of 275, 330, and 390 mm. This state-of-the-art sequencing chuck with pressure control safety system greatly increases production output without re-boring jaws.

smwautoblok.com

Amorphology

PARTNERS WITH ADDITEC FOR ROBOTIC GEAR COMPONENTS

Amorphology Inc, a NASA spinoff company founded from technology developed at the Jet Propulsion Laboratory (JPL) and the California Institute of Technology, has partnered with Additive Technologies (AddiTec), a founding partner of Meltio, an additive manufacturing company pioneering the development of affordable multi-metal 3D printing systems. Together, Amorphology and AddiTec are developing the additive manufacturing of multi-metal gear components for robotics.

Building on their previous collaboration of 3D printing large-scale strain wave gear flexsplines using directed energy deposition (DED), the partnership between Amorphology and AddiTec has now turned to the development of multi-metal, functionally-graded gear components, combining two different steels together within a single part. The three-inch diameter flexspline demonstrator is part of a zero-backlash strain wave gearbox used in robotic arms and precision-motion mechanisms. The thin-walled flexspline has competing requirements of wear resistance in the teeth and a fatigue-resistant body that motivates the use of two different materials during 3D printing. The steel in the toothed region is a precipitation hardening martensitic stainless steel typically used in high-strength applications with an average hardness of 35 Rc. In contrast, the steel below the toothed region is known for high toughness with a lower average hardness. By combining the two steels strategically in a gear, it becomes possible to tailor the mechanical properties to take advantage of the benefits of each alloy.

Through its exclusive licensing agreement, Amorphology is developing its intellectual property around multi-metal 3D printing, specifically functionally graded metals, which allows for the strategic transition between more than one metal during 3D printing to produce multi-functional parts that are free from cracks and unwanted phases. The core technology, developed over a decade ago at JPL, focuses on the design of multi-metal transitions to achieve predictable mechanical or physical properties in the printed part. Unlike conventional claddings or overlays, functional grading aims to achieve the best possible properties at the interface between dissimilar metals or composites in a 3D printed part, which is useful for preventing failures such as those caused by thermal mismatch or fatigue cracks. Tailoring the interface between different metals during printing can be achieved through diffusion at the interface by allowing the disparate metals to mix or by adding one or more intermediate compositions at the transition. The prototype developed in the current partnership was produced through wire-fed DED by printing the base of the gear from high-toughness steel and then sharply transitioning to the high hardness steel at the vertical location where the teeth begin, using the melt pool to diffuse the layers. The materials could have also been transitioned by mixing powders of the two materials in the powder-blown configuration of the DED printer.

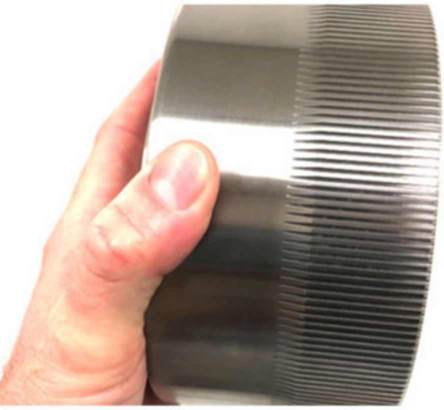
"Functional grading with multiple materials allows us to develop gear components for robotics that cannot be fabricated with conventional metallurgy. The ability to tailor the properties of a gear via alloy composition gives us an entirely new design freedom when developing precision mechanisms," said Dr. Glenn Garrett, Amorphology CTO. "Whether it's improving the wear resistance of gear teeth while maintaining toughness in the rest of the part or using high-value steel in combination with low-cost steel to save cost, multi-material additive manufacturing is allowing us to innovate in the way that we approach gears for robotics. We can tailor properties for machinability, cost, hardness, strength, corrosion resistance, even density. For large gears where it makes sense to use additive manufacturing to save machining costs, this could be a real advantage."

"Meltio's dual-wire DED technology provides an ability to change from one material to another material automatically during the fabrication of metal components. This results in gradual change in properties and functions which can be tailored for enhanced performance," said Dr. Yash Bandari, business development manager at AddiTec.

The multi-metal flexspline demonstrator from Amorphology and Additec is designed so that the high-performance high-hardness steel resides in the gear teeth and the rest of the cup is made from highly machinable tough steel. The as-printed hardness of the

gear teeth through DED was measured to be around 30 Rc while the base of the cup measured at around 7 Rc (87 Rb). A further increase of the gear teeth hardness is possible through subsequent heat treatment. Future work will focus on different combinations of steels, and the development of localized heat-treating strategies to optimize the properties of each metal in the bi-metallic gear. Amorphology is actively developing other applications for functionally graded metals in applications such as thrusters, rockets, robotics, and gears, and is seeking partners for further development and licensing.

amorphology.com



Technology Advancements in EV Inspection

Electric Vehicles Provide New Challenges as well as Opportunities

Matthew Jaster, Senior Editor

Different component characteristics in electric vehicles lead to higher noise and load requirements in the automotive industry. E-mobility—to a certain degree—is changing how gear analysis and inspection is carried out. Dissecting noise issues in gears and gearboxes requires an analytical approach like a detective. The problem could stem from the design itself, tolerancing or tip/root relief issues, tooth flank form deviations like waviness or perhaps crowning issues that directly impact noise. Every aspect of gear production needs to be examined to provide the most accurate results.

Solutions for Evolving Gear Inspection Needs

"How is the production machine influencing gear noise? Does other equipment nearby cause additional vibrations? These are questions that need to be answered to better understand gear noise," said Klaus Deininger, international sales manager, Gleason Metrology Systems.

"The constantly increasing power density of gears and the growing importance of noise behavior are leading to increasingly tight tolerances, thus placing an even greater burden on gear inspection technologies," Deininger said.

Even when all the measurements are in tolerance, components still fail due to bad noise behavior. Gleason offers two software applications to evaluate gear noise including low frequency noise, medium frequency noise and high frequency noise (crowning, tooth mesh irregularities and waviness):

GAMA's KTEPS (Kinematic Transmission Error Prediction Software) analyses the plane of contact for an entire gear rotation by deploying tooth topography measurements. Based on surface deviations transmission errors can be calculated and evaluated by FFT analysis. In a second step the operator can evaluate suspicious harmonics by extracting only the topography data representing the questionable harmonics. This data is being presented in a false color picture, making it much easier to understand the source of the waviness. This unique way of gear evaluation is an ideal tool for ghost noise studies whilst correlating with End of Line single flank testing.

Topography measurement is a relatively slow process, so customers want to obtain waviness measurements faster and more efficiently than they have in the past. Gleason has created *Advanced Waviness Analysis* to detect and mitigate critical noise behavior caused by hard-to-find tooth form irregularities. Advanced Waviness Analysis Software can be seamlessly connected with KISSsoft's *Gear Design Software for Loaded TCA* to provide multi-sensor inspection gear noise analysis based on standard profile, lead and pitch inspection, which can be tactile, optical or a combination of both.

"This is going to be a faster way to get

The HFC Hard Finishing Cell from Gleason combines gear grinding, auxiliary processes and gear inspection, with fully robotized part handling. (Courtesy Gleason) the measurements customers need in today's production environment. There's no additional time for analysis, once the measurement is completed, you'll have results," Deininger said.

The 300GMS nano is an example of a new inspection system that is suited to support automotive e-drive production with minimum noise requirements.

"This machine covers the full range of modern gear inspection capabilities, as well as fine pitch gear inspection and CMM measurement. With the 300GMS nano, users are now able to measure surface finish at sub-micron level with a skidless probe, analyze waviness for profile, lead and pitch, and execute noise analysis with sophisticated software tools," he added.

The machine also integrates 3D measurement and analysis capabilities typically offered by a CMM. The 300GMS nano provides customers with a valuable tool for e-drive gear measurements.

For many decades, Gear Noise NVH Roll Testing has been a part of bevel gear production, but cylindrical roll testing markets require more sophisticated technology.

"In order to look at electric vehicles, we must examine NVH from low to high frequency levels and provide a variety of testing features and capabilities," Deininger points out.

The GRSL (Gear Rolling System with Integrated Laser Technology) System from Gleason combines traditional roll testing with advanced non-contact laser technology. This vastly improves cycle time for index, lead and profile inspection, as well as gear noise evaluation, and provides analytical inspection of 100 percent of production output, even for the toughest gear applications.

Combining the GRSL with Advanced Waviness Analysis software offers the unique possibility to evaluate each gear produced regarding potential waviness on the tooth flanks, which is the ultimate contributor of high frequency noise in gear mesh. This means, no gears with questionable or bad noise behavior enter final assembly, which reduces the costly disassembly of gearboxes with bad noise behaviors dramatically. Today's gear inspection must also factor in the machine behavior itself. How does the machine contribute to measurements? Why are there variations from machine to machine? How can Gleason provide the most accurate inspection results moving forward?

Recently, the GRSL system was integrated into Gleason's Hard Finishing Cell (HFC). This automated system includes a robot that integrates a variety of process modules including gear grinding, washing, laser marking, measuring and part handling. During the gear inspection, the laser scanner provides measurement characteristics for each tooth. Deviations are fed back into the machine via a Closed Loop correction.

The future of gear inspection is complete, in-process measuring with integrated gear noise analysis in a Closed Loop automated system, providing customers with all the necessary means to keep tolerances at all times, minimizing scrap and rework.

gleason.com

300GMS nano Gear Metrology System with Integrated Gear Noise Analysis. (Courtesy Gleason)

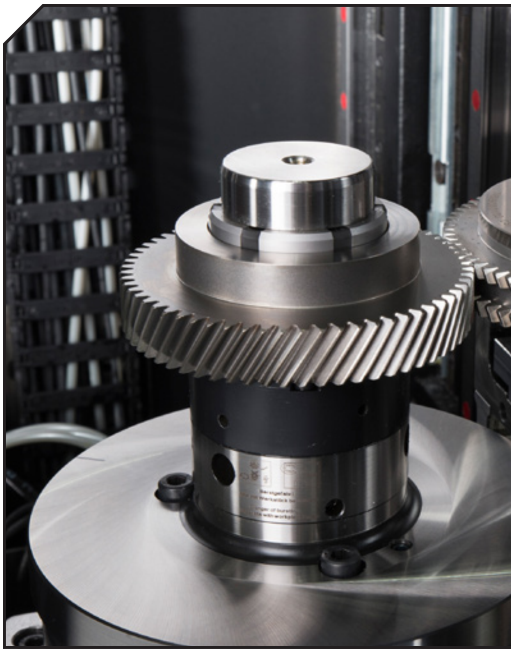
Klingelnberg Dissects Complex Inspection Requirements

Klingelnberg's R 300 roll testing solution provides 100 percent noise assessment of gears within the production cycle. (Courtesy Klingelnberg)

Dr. Christof Gorgels, director, precision metrology at Klingelnberg, discussed the reduction of gear tolerances in the automotive industry in recent years. "The need for noise performance testing has increased and is an addition to the traditional geometric quality assessment. With the *Gear Deviation Analysis (GDA)* Klingelnberg has been offering a tool for many years to analyze the expected noise behavior of a gear based just on simple gear measurement," he said.

With Klingelnberg's hybrid approach, the company offers a solution for faster measurements without compromising on accuracy. In addition, the R 300 a roll testing solution can do a fast 100 percent noise assessment of gears within the production cycle.

"Speed and flexibility both contribute to minimize the quality costs," Gorgels said. "With the combination of *GDA*, hybrid metrology and roll testing, Klingelnberg offers a solution for the



different requirements of gears concerning geometrical quality and noise.

There are two major differences in EV compared to conventional vehicles:

"The most obvious is the lack of a masking noise of the engine bringing gear noise even more into focus.

The R 300 provides five roll testing methods including single flank, structure-borne, torsional acceleration, double-flank and the helix roll test. (Courtesy Klingelnberg)

Looking into load-carrying capacity, the electric motor has a different torque characteristic increasing load on the drive flanks. In addition, recuperation leads to high loads on the coast side that is new to the automotive industry. Combining higher load and noise requirements in combination with a highly loaded coast flank, the optimization of a gear flank design becomes much more complex," Gorgels said.

Klingelnberg's *GDA* software makes gear noise visible. It comprises the four modules View, Wave Analysis, Wave Production and Produce.

"For noise analysis we are evaluating the form error in a smarter way by looking not only for the amplitudes, but also for regular structures such as waviness. With our *GDA* software we offer a tool to our customers to rate this waviness and basically receiving order diagrams that can directly be compared to the end of line test. This ensures that the acoustics engineers and the manufacturing and quality engineers speak the same language," Gorgels added.

What is important today is the advanced grinding process simulation. If a gear is found to be noisy the next question arising is how to fix the issue? Finding technical solutions for noise issues is far more complex compared to correcting standard geometrical deviations.

"With the manufacturing simulation, different process deviations, like for instance a tumbling grinding tool or a torsional vibration of the workpiece table, the result on flank waviness can be simulated and thus compared to real world measurement result. Besides making noise visible, we also help our customers to find the root cause," Gorgels said.

He is seeing more tolerances for certain orders on customer's prints today.

"These orders can be tooth mesh orders (and higher harmonics) as well as so called ghost orders. If these requirements show up on a printout, they should be measured in production. This increases the demand for standard production solutions which we offer with the *GDA* analysis as well as the R 300 roll tester," Gorgels said.

In the coming years the need to use these tools will increase. Standardization will become a major topic to ensure a common understanding between OEMs and their suppliers.

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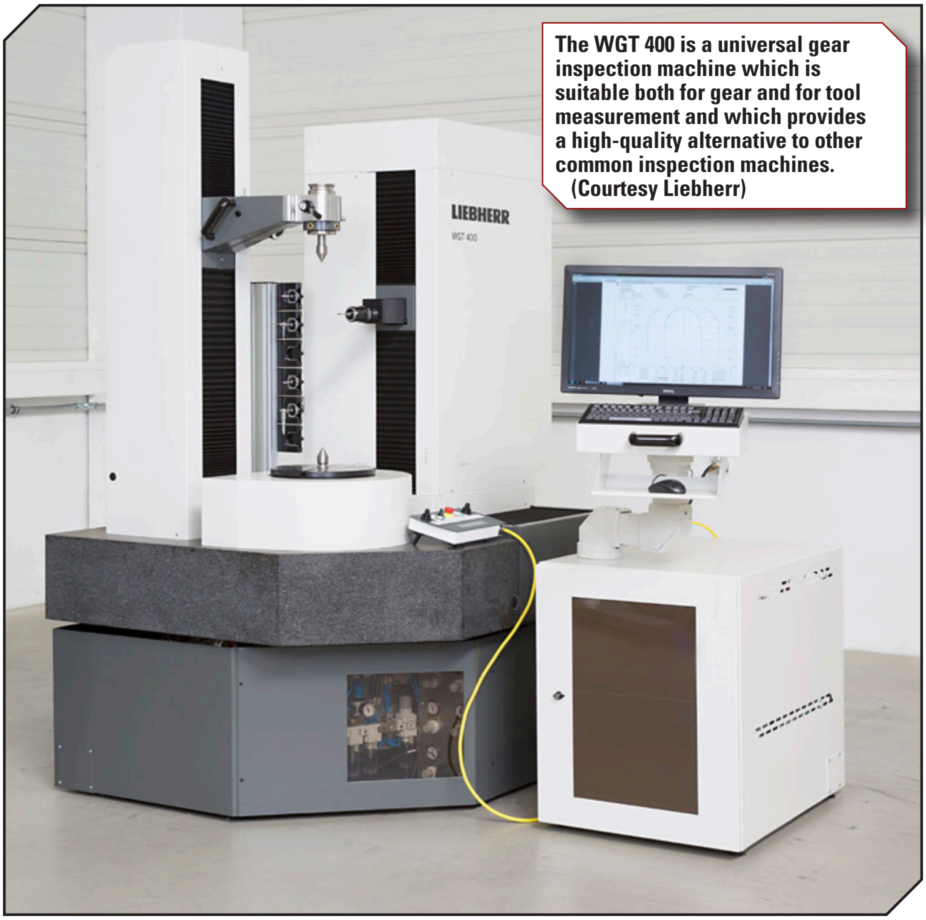


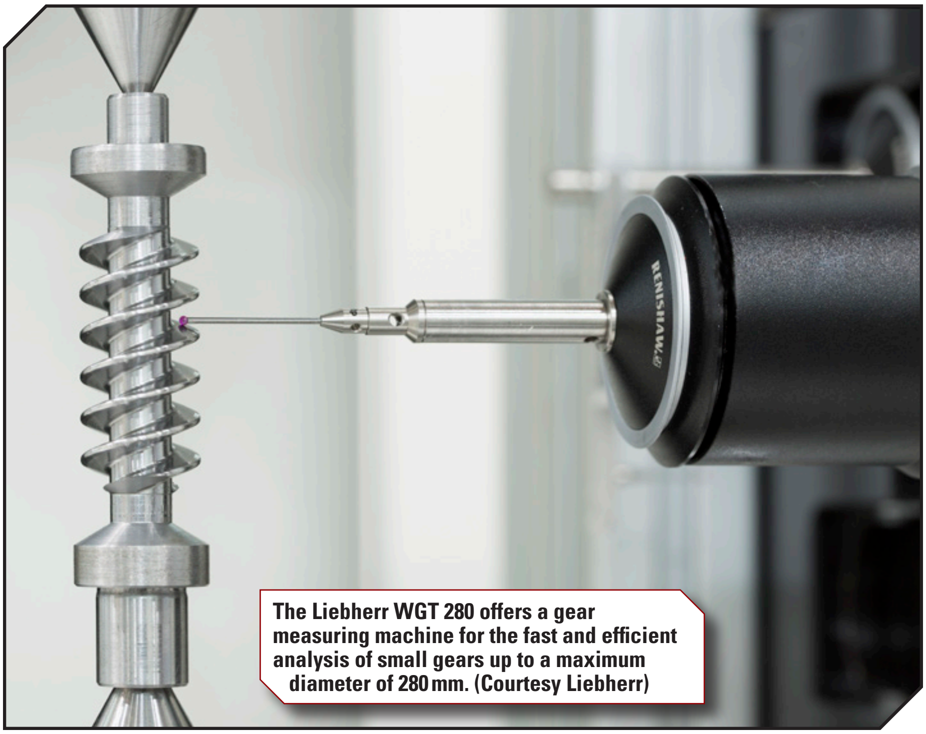
The WGT 400 is a universal gear inspection machine which is suitable both for gear and for tool measurement and which provides a high-quality alternative to other common inspection machines. (Courtesy Liebherr)

Liebherr Points to Accuracy, Liability and Reliability for Gear Inspection Results

Gears in EVs have a particular higher demand on noise generation and consequently on the surface structure and waviness of the gears. Although the measuring principle for the inspection of involute gears is still valid, EV gears are pushing the limits for gear inspection today, according to Matthias Bruederle, product manager at Liebherr.

Liebherr's WGT 400 optimizes the precision and quality of the measurement of tools used to machine very small parts, including internal gears and gears with very small modules. The supply of data in real-time is one of the machine's benefits as well as its high measuring speed.

The WGT series of measuring technology closes the gap in the closed-loop sector. The four-axis measuring devices are equipped with high-precision mechanics and electronics, which are controlled by intelligent and user-friendly software. The combination of granite guides and air cushioning creates maximum precision with wear-free mechanics.



The Liebherr WGT 280 offers a gear measuring machine for the fast and efficient analysis of small gears up to a maximum diameter of 280 mm. (Courtesy Liebherr)

Liebherr's solution "Open Connect" offers a direct link between the grinding machine and the inspection center, providing a fast and reliable solution to shorten the response time to a determined error of the gear's geometry. Data is transferred via standardized GDE-interface and work with any grinding machine.

"The use of optical sensors seems to be an appropriate solution to improve performance and quality cost, however open questions regarding accessibility and accuracy, in particular on polished surfaces, are still preventing this to be a reliable solution applicable for the inspection in industrial environment," Bruederle said.

For the most effective gear production, Bruederle said that inspection results today must provide the highest accuracy, liability and reliability. "Measuring speed and accuracy may therefore exclude each other and depend on the gear geometry. The development of a specific measuring strategy—based on gear geometry and the production method— is required."

E-mobility has expanded gear measuring solutions to meet automotive demands.

"The development of new production methods like the polishing of gear teeth with and addition grinding process with the purpose of stochastic distribution of grinding marks on the tooth flank also known as silent shift grinding find their way in the development of hardware and software components on the inspection machines," he added.

Sensors and software extensions for evaluation of surface roughness (special sensor required), evaluation of dedicated end reliefs, and evaluation and analysis of waviness and undulation will continue to play a large role for the analysis of e-drive gears.

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Higher Inspection Levels

Large scale transmission manufacturing requires a greater focus on noise behavior and quality for each gear in the powertrain. Although the electric motor provides a variety of advantages for the future of transportation, it does very little to hide gear noise. New inspection equipment and technology, therefore, must provide the tools to bring comfort, reliability and precision to all future e-mobility applications.

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Nondestructive evaluation of stresses and stress-related defects in gears

James Thomas, General Manager, American Stress Technologies

Gear manufacturing involves a number of processes that, intentional or not, affect the residual stress state of the critical surfaces. Stresses, including residual stresses from processing, are commutative, with compressive stresses typically improving fatigue life and crack initiation while tensile stresses do the opposite. Accordingly, gear designers and manufacturers often require compressive residual stresses at the surface on critical geometries such as gear teeth and roots.

Shot Peening Verification

Shot peening is a common process utilized in gear manufacturing to increase the amount of compressive stress and, consequently, increase the fatigue life of the gear. Often applied in the root area, shot peening involves blasting the surface of a component with hard "shot" (Fig. 1). A thin layer of the component is deformed and compressed while the core, or deep subsurface volume, resists this compression (Fig. 2).

The result of this process is a layer of compressive stresses from the surface through some depth often a few hundred micrometers deep (Fig. 3).

There are a variety of methods to verify the shot peening process including the Almen strip test, visual analysis including the use of tracer dyes, and direct measurement of induced stresses. It is the latter method, measurement of induced stresses, which provides the only true objective measure which can be compared to expected values from the design and modeling phases. The best and most standardized method for the measurement of stresses is X-ray Diffraction (XRD).

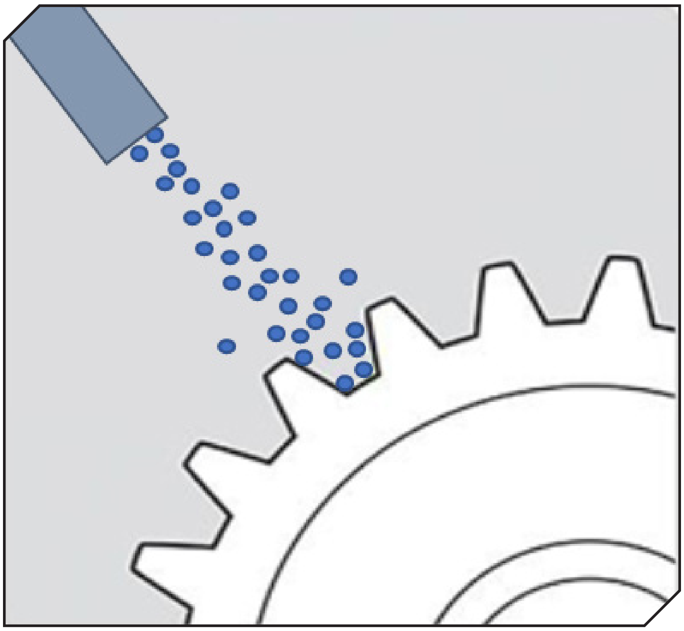

Figure 1 **Shot peening involves blasting the surface of a component with hard "shot."**

XRD for residual stresses is not a new method nor is it under the radar. It is a quantitative, standardized method with no true peers, especially when measuring nondestructively (measuring subsurface stresses requires layer removal). The XRD method has had some limitations as it can be expensive,

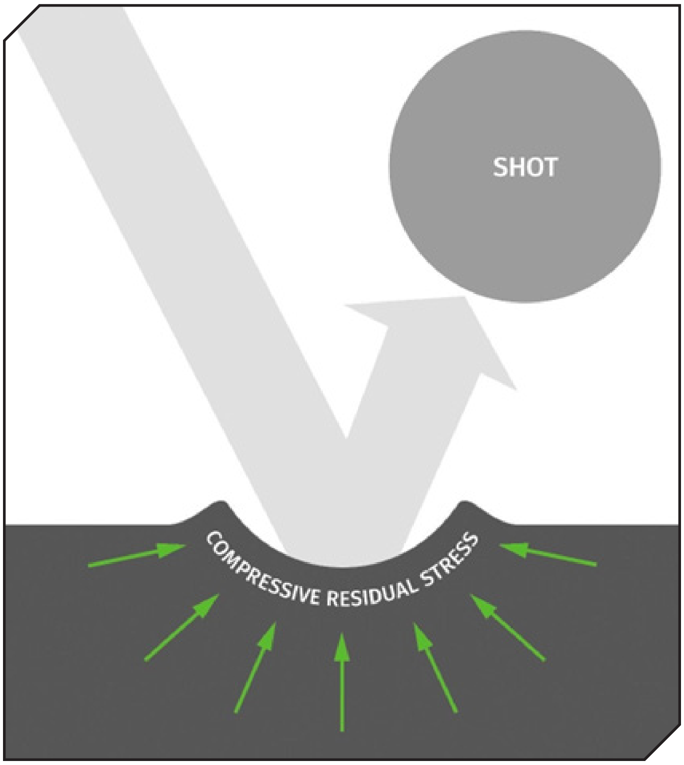

Figure 2 **Shot peening results in a thin layer of compressive residual stress.**

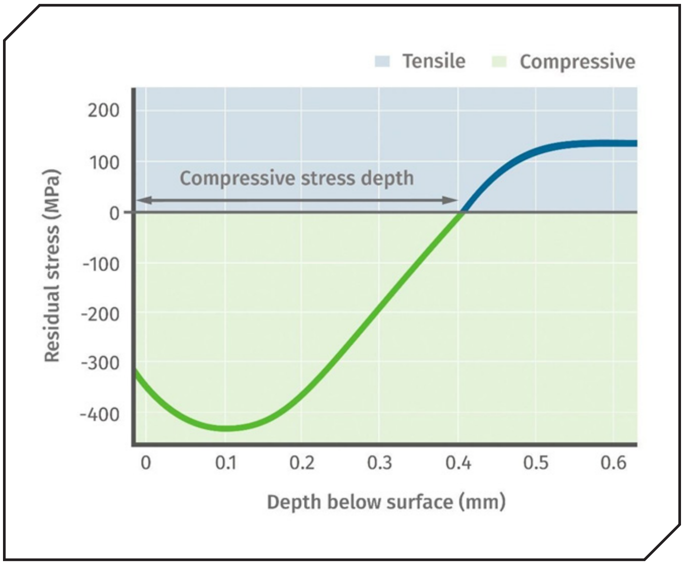

Figure 3 **The compressive residual stress from shot peening can be a few hundred micrometers deep.**

Figure 4 The Stresstech Xstress DR45 system.

time-consuming, and/or require costly sample preparation. Measurements could take minutes to hours, depending on the spot size and the equipment utilized. Generational leaps in technology, though, have changed this for the better and measurements can now be performed in seconds.

The Stresstech Xstress DR45 system (Fig. 4), utilizing state-of-the-art detector technology, is sensitive enough to make previously slow measurements lightning fast, or previously difficult measurements easy. Utilizing high-sensitivity 2D detectors, the DR45 measures so fast that it doesn't even need to stop moving to collect diffraction data.

Traditional diffractometers tailored to measure residual stresses, including Stresstech's prior offerings, often utilized 1D detectors. The use of more modern 2D detectors with improved sensitivity allows for significantly more diffraction data, as much as 100x, to be collected in the same amount of time. Sections of the 2D Debye-Scherrer ring are integrated into 1D intensity spectra for strain determination (Fig. 5).

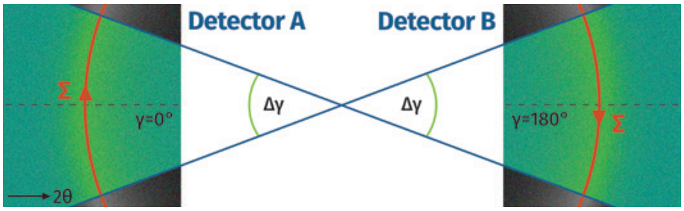

Figure 5 The use of 2D detectors allows for the collection of significantly more diffraction data.

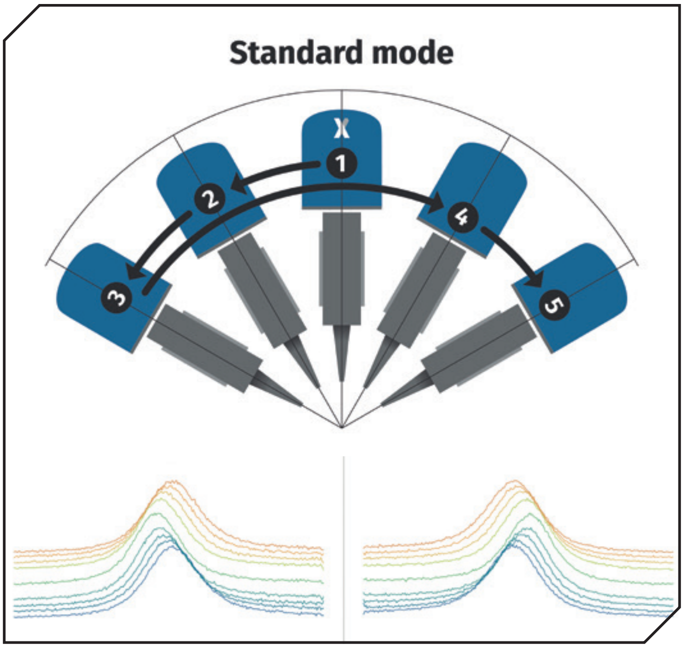

Figure 6 Standard mode x-ray diffraction systems use a time-consuming sequence of individual scans.

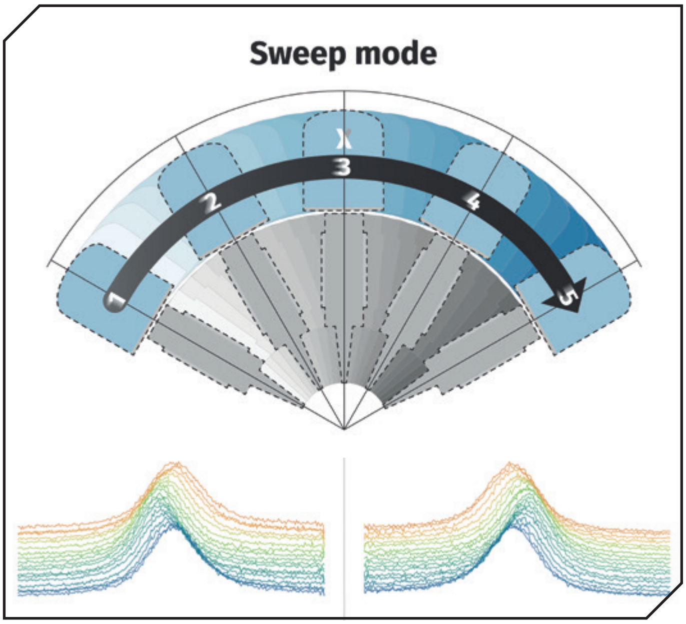

Figure 7 The Xstress DR45 is able to scan continuously, greatly improving inspection speed.

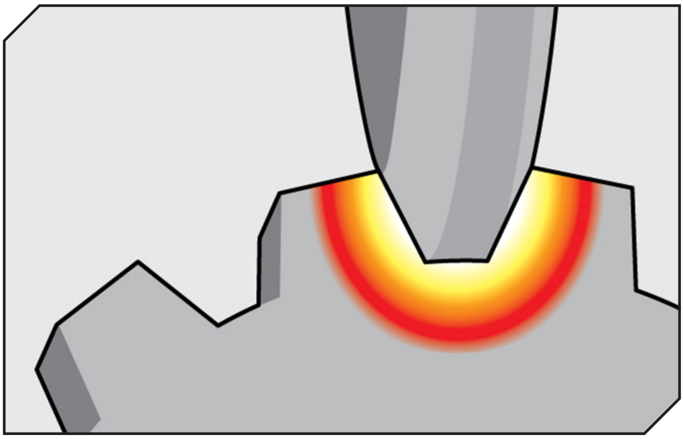

Figure 8 Aggressive feeds and speeds can result in grinding burn.

In addition to improving the sensitivity and speed of the system, the 2D detector method employed by the Xstress DR45 combines the benefits of 2D detection systems, such as mitigation of difficult diffracting conditions like textured or large grains, with the high quality and trusted results of the $\sin 2\psi$ method. Beyond that, the multiple orders of magnitude improvement in speed allows for some truly game-changing capabilities: continuous movement measurements (or Sweep mode).

Diffraction systems utilizing the tried-and-true $\sin^2\psi$ method have always followed a standard sequence of operation (or Standard mode, shown in Figure 6): 1, move X-ray incident beam and detectors to position; 2, Expose the sample the incident X-rays and collect diffraction data with detector; 3, Repeat 1–2 as necessary to satisfy the measurement specification such asEN15305. This type of measurement sequence is standard on nearly all diffractometers except for systems utilizing methods not in accordance with internationally recognized standards.

The speed of the detection system on the Xstress DR45 allows the diffractometer to collect "snapshots," much like video frames, while it is moving (Fig. 7).

The result is a large improvement in total measurement speed over the already fast Standard mode speed provided by the Xstress DR45. Measurements on spots 1 mm in diameter or larger are performed in 5 seconds. Measurements on spots less than 0.5 mm in diameter can be performed in as little as 20 seconds. These speeds open the possibility of shot peen verification on gear teeth or roots which is fast enough to keep up with production in high-volume environments.

Material	Radiation	Mode	Time	Collimator diameter				
				3 mm	12 mm	Imm	0.5 mm	0.3 mm
Ferritic steel INCONEL®	Cr Ka Mn Ka Cr Kf3	sweep	tsweep	5s	5s	5s	10s	20s
		standard	tmeas	34s	34s	34s	34s	34s
INCONEL°		sweep	tsweep	105	105	l0s	20s	40s
		standard	tmeas	34s	34s	43s	70s	115s

Grinding Burn Detection

Grinding is a crucial step in gear manufacturing, and it frequently presents manufacturers with a critical question: How fast can I grind without generating a grinding burn? Faster cycle times are always desirable but greater infeed, wheel speed, etc. can result in more energy, or heat, being deposited into the workpiece (Fig. 8).

Grinding burn occurs when the heat generated in the workpiece during grinding is great enough to act as a tempering process or, in the case of even higher temperature, a re-heat treatment of the affected surface. The result of this localized thermal event is a transformation of the microstructure, much the same as what occurs during heat treatment. The affected volume of material is transformed from a desirable microstructure, such as tempered martensite, to an undesirable mix of over-tempered martensite (or softer ferrite, etc.) and untempered martensite. The transformed microstructures have differing densities but, squeezed into the same space previously occupied by the desirable microstructure, then must be compressed and/or pulled apart in order to fit. This compressing or pulling manifests as residual stresses in the material.

The traditional method for detection of grinding burn is the Nital etch process. Nital etching involves exposing the surface of the material to different chemicals, including Nital (nitric acid and alcohol) and hydrochloric acid. The process takes advantage of differential dissolution, where the Nital mixture attacks phases such as ferrite, cementite, etc. differently. The result is that some visual contrast exists between the desirable tempered martensite microstructure and the undesirable over-tempered or untampered generated by grinding burn.

Though it is a traditional method with decades of use the Nital etching process has some downsides: it is subjective, requiring the practitioner to interpret litera l shades of gray; it requires the use of possibly dangerous chemicals, often with nontrivial handling and disposal requirements; in most cases it is destructive, as etched surfaces may not be suitable for use in service. Another limitation inherent to the Nital etch process is that it is insensitive to stresses—its mechanism of action is to reveal transformed microstructure. In the case where grinding burn occurs during a roughing pass, only to be partially cleaned up by a finish pass, the Nital etch process can be ineffective at revealing the partially cleaned up transformation products. Despite partial cleanup of the transformed layer, subsurface tensile stresses typically remain. This "hidden" burn leaves the component susceptible to early failure. Stresses induced via grinding burn typically peak at 20–50 micrometers below the surface. In the case of partial cleanup of grinding burn during the grinding process a subsurface tensile peak typically remains (Fig. 9).

An alternative method to detect grinding burn is Magnetic Barkhausen Noise (MBN). It is a repeatable, objective measure that is nondestructive (Fig. 10). Furthermore, MBN is sensitive to both stresses and microstructure in the measured volume which makes it ideally suited for detection of grinding burn in cases of partial cleanup.

Stresstech's Rollscan Barkhausen Noise analyzers take the MBN signal and reduce it to a single number measured in real-time. This allows the user to traverse an MBN sensor across a surface, manually or via automation, and get a live measurement or even a surface map.

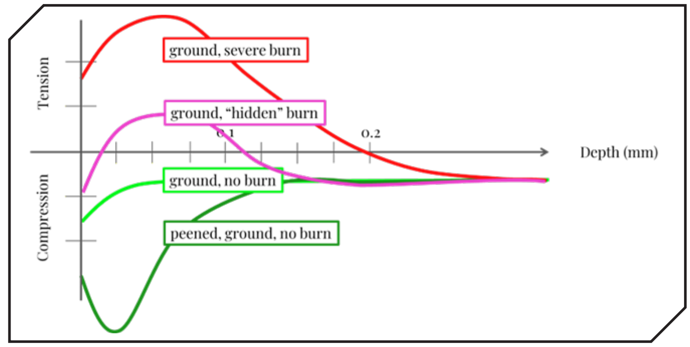

Figure 9 Nital etch can sometimes miss some forms of grinding burn.

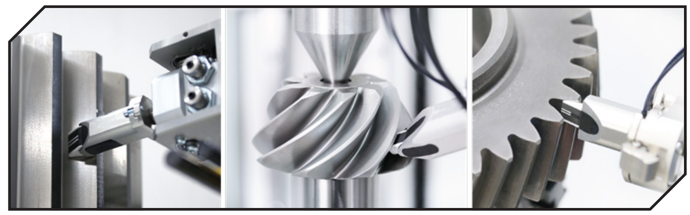

Figure 10 Magnetic Barkhausen noise analysis is an alternative to detect grinding burn.

Figure 11 Stresstech *EasyGear* software for programming of automated gear testing.

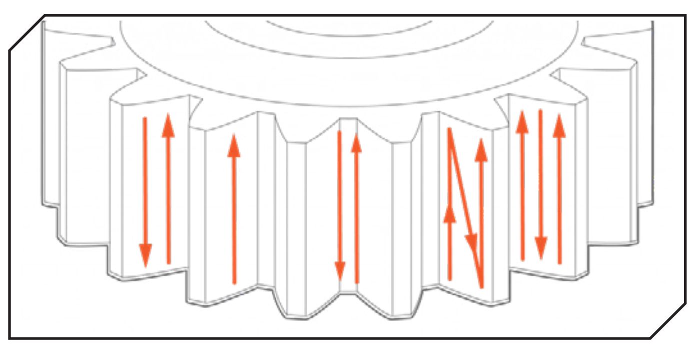

Figure 12 MBN instruments can measure gear flanks, roots, faces, ODs and IDs.

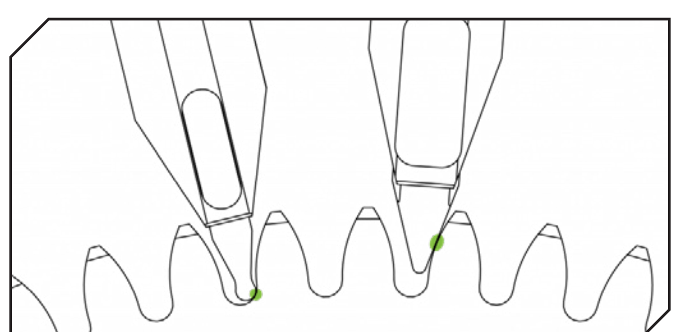

Figure 13 Customized sensors can be used, depending on the surface to be measured.

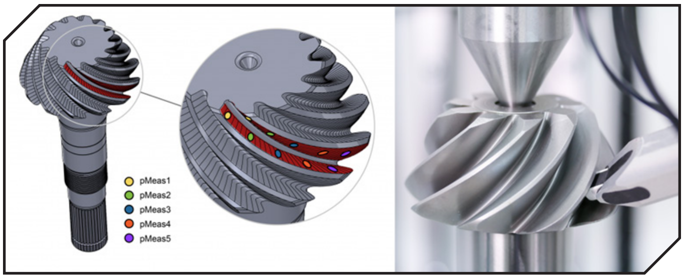

Figure 14 Complex geometries can be accommodated via special sensors and point-by-point control of the sensor path.

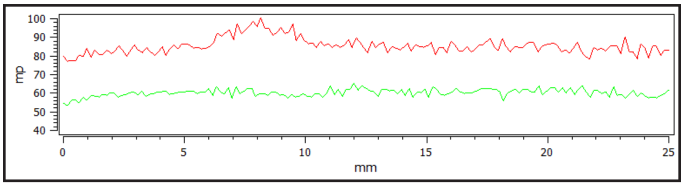

Figure 15 In this chart, the red line represents MBN signal increases, indicating the presence of grinding burn.

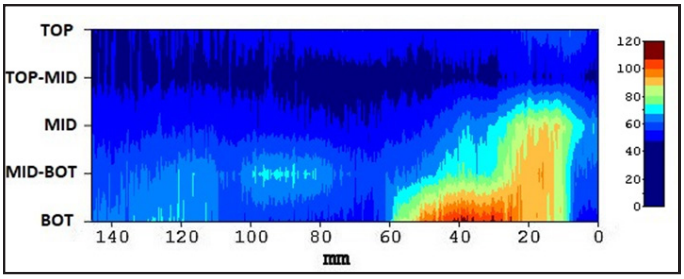

Figure 16 Multiple scans or passes per flank can be combined into a surface map.

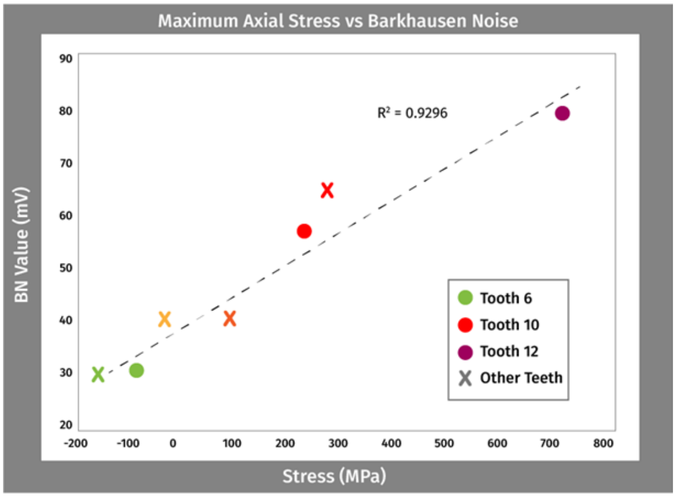

Figure 17 Data from MBN scans can be shown against simple rejection/acceptance criteria for easy evaluation.

Automated systems are programmed in the same manner as an analytical gear checker with the ability to map the surface of the flank to the level of spatial resolution required for the applicationby utilizing Stresstech's EasyGear software (Fig. 11). Gear flanks and roots, along with various other surfaces such as faces, ODs, and IDs are all measurable with MBN instruments (Fig. 12).

Customized sensors are sometimes required depending on the geometry of the surface to be inspected. For example, gear flanks and gear roots are typically measured with dedicated sensors which facilitate the sensor contacting the area of interest (Fig. 13).

In the case of more complex geometries, such as hypoid gears, custom sensors are utilized, and sensor movement paths are generated along complex curves generated point-by-point (Fig. 14).

The result of a measurement sequence is a series of scans, or plots, similar to the output of an analytical gear checker. Lower and more consistent measurement values, essentially flat scans, are typically found in acceptable parts free of grinding burn (see the green line in Figure 15). In the presence of a grinding burn the MBN signal increases (see the red line in Figure 15).

Multiple scans or passes per flank, performed at varying diameters, can be combined into a surface map. This provides the type of visual indication that Nital-etch users are accustomed to seeing, with the added benefit of objective, repeatable values (Fig. 16).

By comparing the relative MBN values to a quantitative method such as XRD the MBN values can be contextualized, and proper rejection/acceptance criteria can be developed. The most common method of developing rejection criteria involves comparing the MBN value to the maximum subsurface stress, measured via XRD, similar to the recommended practice in SAE ARP4462b. Such a comparison allows the user to choose an MBN value limit that corresponds to subsurface tensile stresses, or some other limit depending on the application and the design requirements for the component under test (Fig. 17).

Conclusion

Gears present challenges to many traditional testing methods both destructive and nondestructive. In cases where residual stress is critical, as is the case in shot-peened gears, verification of the shot-peening process via XRD can be fast enough to keep up with your production. This is especially true when utilizing state-of-the-art instruments such as the Xstress DR45.

Nearly all gears in precision applications have ground flanks. Sometimes they also have ground roots. Detecting grinding burn on these surfaces with maximum sensitivity and repeatability, all while avoiding costly scrap, is achievable using Magnetic Barkhausen Noise. Additionally, the method can be fully automated to provide measurable feedback for process control.

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Multisensor CMMs for Gear Inspection

Gene Hancz, Product Specialist, CMM, Mitutoyo America

Gear inspection has long been a highly specialized, costly investment and an overall challenging part of the gear manufacturing process. Through advances in technology and automation, this typically tedious, time-consuming process is becoming significantly more efficient as multisensor coordinate measuring machines (CMMs) gain more traction as one of the preferred methods of gear inspection.

The gear manufacturing market is forecast to grow by USD 73.66 billion between 2020 and 2025, expanding at a CAGR of 5.73%, according to market research firm Technavio. This growth will be fueled in part by the adoption of industrial automation solutions to speed time-to-market and improve quality. While standard machines still have a place on the manufacturing floor, it's not surprising that an increasing number of gear manufacturers are investing in multisensor CMMs to increase automation and streamline their inspection process.

Given that complicated gauges, testers, and CNC equipment all go into creating high quality gears, most gear-checking machines on the market today fall short because they are not designed to utilize multiple sensors beyond one or two different types. Multisensor CMM machines, on the other hand, have become a game changer for gear inspection because they combine up to five different types of sensors on just one machine: touch trigger measuring, scanning, vision sensor, laser probes, and surface roughness probes. In addition to reducing the number of machines required for gear inspection, multisensor CMMs also offer a single solution for gear measurement, production, and setup, including measurement of purchased gears. As a result, they are far more economical and cost effective compared to conventional, dedicated gear inspection equipment.

The benefits of multisensor CMMs

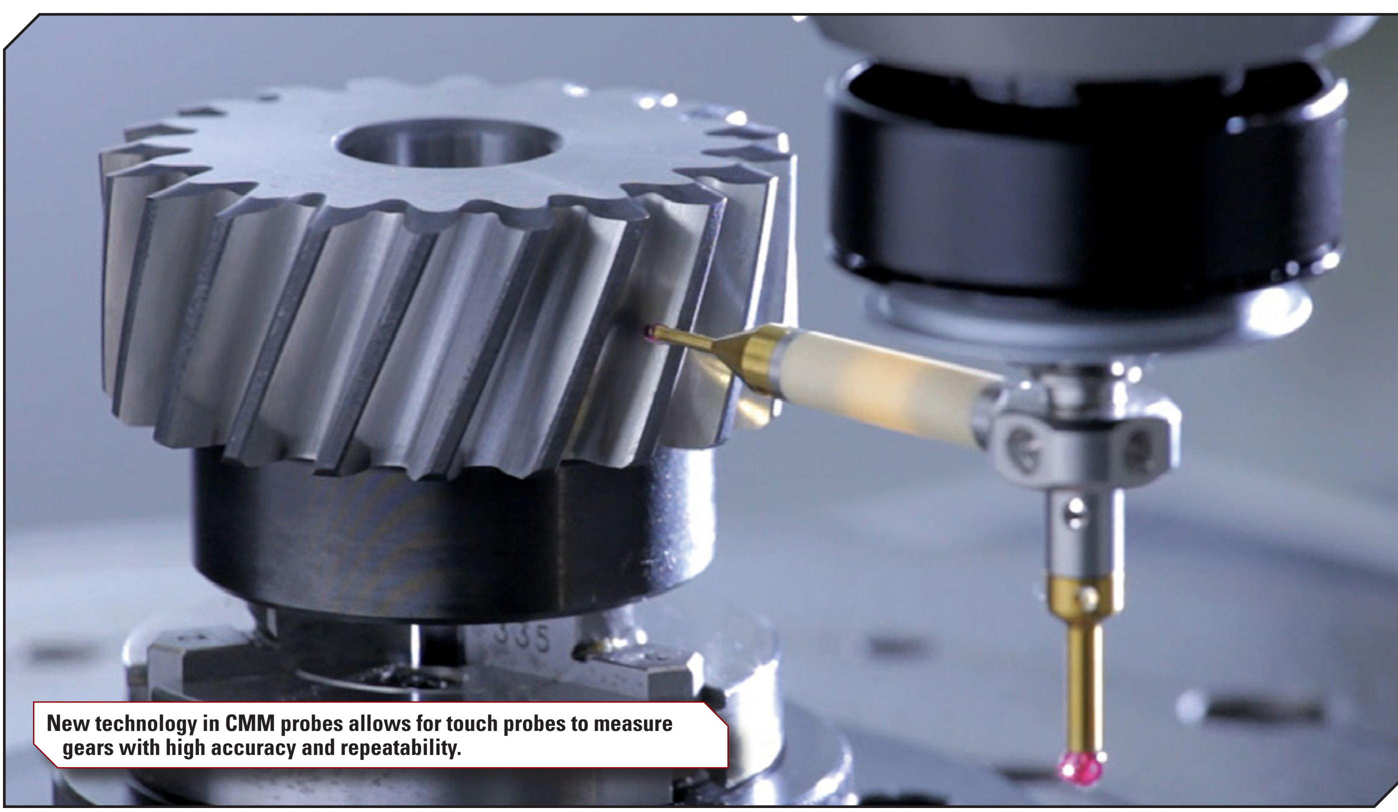
Multisensor CMMs provide a wide array of gear inspection and measurement tools on a single machine platform. This provides gear manufacturers with a number of potential benefits:

Lower costs and improved ROI

Today, the cost of programmable, automated, and sophisticated gear checking equipment can range from $300,000 to $350,000 or more. And this doesn't include the cost of master gears and artifacts needed for setting dedicated gear measurement equipment. The cost of all required components can be at least 2.5 times as much as the cost of a multisensor CMM machine equipped with a rotary table, high-speed scanning probe head and gear measurement software.

Improved productivity

In general, using the automatic measuring tools in multisensor CMMs, whether for gear checking or regular SATIC geometry, can provide time savings from 30 to 98 percent. The greater the complexity, the greater the time savings. For example, a part with 3,000 characteristics may take five hours for a typical gear



New technology in CMM probes allows for touch probes to measure gears with high accuracy and repeatability.

measurement tool to run one part. Automated tools on a multisensor CMM can bring about time savings as high as 98 percent because they can measure all those characteristics so much faster and accurately than manual methods. In addition, using multisensor CMMs increase productivity because multiple checking functions can be completed on one machine such as inspecting gear parameters, measuring microscopic gear teeth, high-accuracy 3D scanning, and more. Plus, with the surface finish probe on the same CMM, it's possible to check the surface and finish of gear teeth, the internal boar of a gear, as well as the mounting and shaft.

A positive impact on quality

The automated optical tools on multisensor CMMs never touch the gear, so they can take high precision measurements without damaging the surface of delicate gears. This type of high precision measurements of gear surfaces helps to ensure the gear will not make noise due to gear movement. Higher quality overall is also brought about by eliminating operator error. As the engineering CAD model automatically generates the program to inspect or measure the part, it is never ambiguous or subject to multiple interpretations by an operator.

Choosing the right CMM

Of course, it's important to keep in mind that the type of multisensor CMM used to measure gears will depend primarily on the gear size and weight. For large gears with diameters of more than a meter, or for those that are overly heavy, a high-precision, horizontal arm CMM with a rotary table solution is best. This kind of CMM is typically used for inspecting large-scale gears like those used in ships and heavy equipment powertrains, as well as turbine gearing and those used in nuclear and thermal power plants. Due to the open-access structure of this style of CMM, inspection of such large gears is easier.

Conversely, bridge CMMs are usually better for measuring small or medium size gears, and these machines come in two available styles. The first style has a fixed table with moving bridge, while the other has a moving table with a fixed bridge. This latter affords greater accuracy because the servo drives are located at the center of gravity, with a moving bridge the X-axis drives along one side, so the accuracy will change as the z-spindle moves in X.

Other inspection uses for CMMs

In addition to replacing dedicated inspection equipment, CMMs can also replace many of the smaller, hand-held, and functional gauges used in gear inspection. It should be noted that there can be issues around lack of repeatability between operators using manual inspection methods, as well as issues concerning slower speed of measurement in general.

It's also good to remember that with manual gauges results are written down, making them subject to human error and incorrect values being recorded. An automated CMM can also measure parts while an inspector continues to perform other duties, leading to more accurate, consistent, and repeatable results and reporting.

Utilizing a Surftest probe on a CMM allows for highly accurate measurements of complex parts, including gears and impellers.

CMMs can even be placed on the shop floor alongside production machines, as long as rapid and dramatic temperature changes, as well as vibrations, are considered. If the temperature is stable, there usually will be no major noticeable errors, even given the tolerances required for gears. And many modern CMMs today come with temperature sensors to help ensure proper compensation is made for any temperature fluctuations that do occur.

Software is important for all forms of gear measurement

Beyond the precision and accuracy of CMMs, one of the keys to accurate gear measurement, whether on a CMM or on dedicated gear checking equipment, is the software. For instance, calculation of whether an involute curve is correct based on data points extracted during measurement requires the use of high-level mathematical formulas and sophisticated algorithms.

That's why it's vital to look for CMMs with software that can manage these calculations and can even combine intuitive icon-based programming with the ability to import native CAD models.

In addition, it's important for users to be able to choose various software modules to analyze measurement results to document and present results and to archive the data in practical structures. This means software that integrates with networked systems for inline process control applications, as well as enables true enterprise-wide functionality.

One of the biggest challenges when investing in a multisensor CMM is incorporating software that supports the many diverse sensors for initial requirements but can be easily updated to accommodate new sensor technology for evolving gear requirements. For example, in the automotive industry, electric vehicles are manufactured with a constant velocity transmission that requires specific gear designs and different inspection requirements than traditional vehicles. The flexibility of multisensor CMM software makes it possible to respond to evolving industry requirements like these much faster.

The future of multisensor CMMs

As CMM machines and software continue to become more sophisticated, advancements in automated laser scanning and optical technology using Artificial Intelligence (AI) are expected to provide even higher accuracy. Moreover, multisensor CMMs will also expand their features and applications beyond gear inspection, giving manufacturers even more flexibility. Gear designs themselves are also likely to evolve and accommodating additional shapes and characteristics will require the level of automation and flexibility that only a single platform, multisensor CMM can provide for gear inspection and measurement.

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Special CMM software supports measuring several types of worm gears using a touch trigger probe or scanning probe on a CMM.

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Grinding Slow, Grinding Fine

Reducing nonproductive time increases large-gear production

Aaron Fagan, Senior Editor

While the Greek proverb "the mills of the gods grind slowly, but they grind fine," coined by Sextus Empiricus, was meant as a figurative expression of justice, it does hold true to gear grinding in the sense that there is a right way and a wrong way to do things because rushing the process only causes problems. Like justice, grinding takes the time it takes, but the grind is fine. So, the way to increase production, especially with the godlike largest gears, is to decrease nonproductive time with improved strategies.

Figure 1 Gleason P6650G.

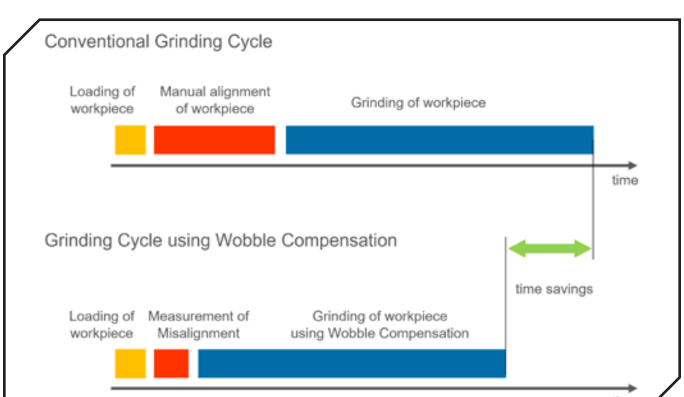

Figure 2 After measuring misalignment, a wobble compensation feature reduces nonproductive time.

Inaugurated in April 2021, CMD in France now owns Europe's largest profile grinding machine—a Gleason P6650G designed to produce gears weighing in at as much as 70 tons with diameters in excess of 6 meters. The machine arrives just in time to add much-needed capacity for the largest helical, planetary, and worm gears in industries from cement to chemical plants, mines to steel mills, water treatment to wind power. In the specialized world of giant gear production where only a relative handful of companies like CMD dare to tread, the inherent risks of equipment investments are enormous. And discontinuous profile grinding large gears has distinct and significant production challenges from their smaller counterparts—but the volume of demand for colossal gears is not huge, so, for the purposes of this discussion, *large* is defined as a diameter of 400 mm or greater. To learn more about the unique manufacturing challenges large gears present, *Gear Technology* spoke with Gleason and Norton|Saint-Gobain Abrasives about their solutions.

As with grinding at all scales, there is forever looming the threat of burning parts if production is rushed or the right parameters have not been established, so the key to success is in the reduction of costly nonproductive time because, to make high-quality parts, the runtime will be calculated as a fixed parameter. In the case of the very largest gears, many hours are typically required for workpiece setup, parts programming, parts inspection, grinding wheel dressing, and anything else that isn't actual profile grinding—and, therefore, profitmaking.

Setup time, for example, is greatly reduced with Gleason's system for wobble compensation, which automatically compensates radial runout and wobble of gears with imperfect alignment, rather than relying on an operator and a painstaking manual process for loading and adjusting the gear to the machine table. On-board inspection of both internal and external gears is fully integrated into Gleason machines, saving enormously on the time needed after grinding is complete to evaluate gear characteristics—a process that can take hours to bring heavy parts to the quality inspection department. In its drive to reduce nonproductive time, Gleason has also optimized the position of the dressing unit to shorten dressing cycles and use their dressing process, which reduces actual dressing time to just a fraction of the time conventionally required.

With respect to having a grinding wheel that's compatible with a grinding process, Josh Fairley, a product engineer for bonded abrasives at Norton|Saint-Gobain Abrasives, says, "Gleason is very good at this, the technology that goes into their equipment tells an operator what the correct tool to use is, automatically adjusting grinding parameters based on what's

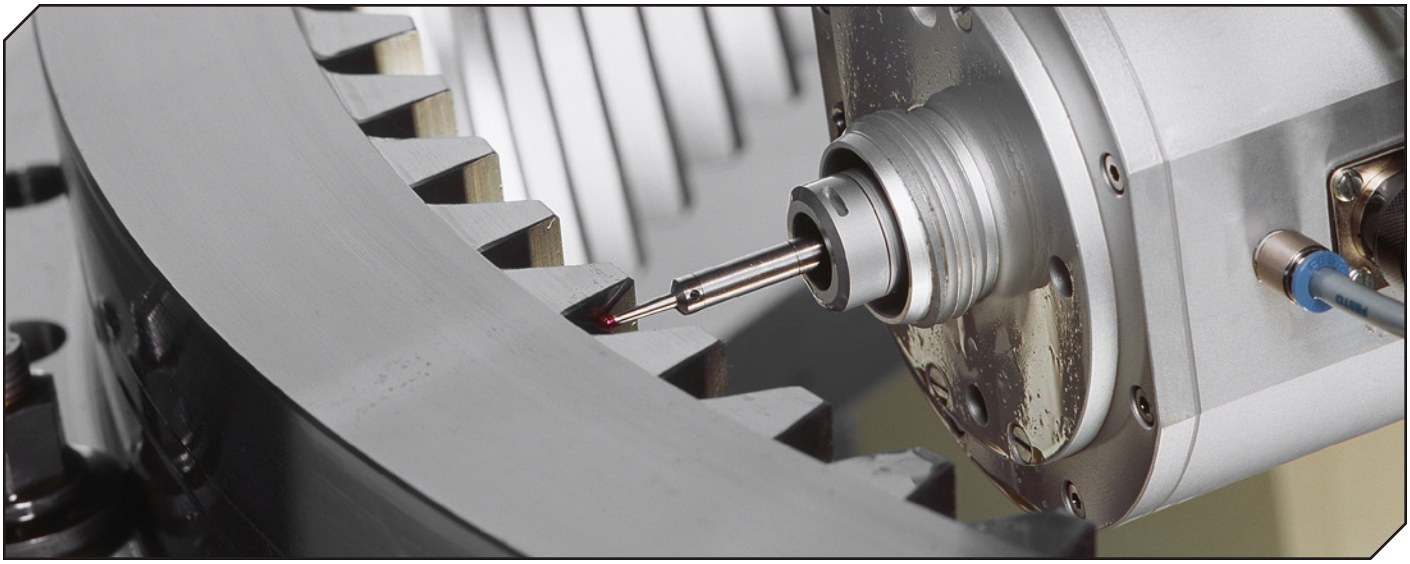

Figure 3 On-board inspection of both internal and external gears is fully integrated on Gleason machines.

happening during the process — so from the abrasive tool side, we want to deliver the most consistent product possible so that their process is repeatable."

Understanding the degree of quality the customer is looking for is Norton's main approach to assigning the right grinding-wheel technology. Quality means not introducing heat into the part along with superior formholding and surface finish. There are many different metallurgical considerations to account for when dealing with gears. Depending on what type of metal is being ground, there can be a careful selection of a secondary abrasive that's chemically the best fit for that material. While quality standards need to be met first, finding opportunities to control cost by factoring for parts per dress, wheel life, dress depth, and cycle time are all considered and optimized to save the most money for the customer. Fairley adds, "Aerospace has historically been the most precise industry, but now there is less difference between various industries because the efficiencies in the gearbox are worth the customer's time and money."

Figure 4 The new Gleason 1200G profile grinding machine.

A Large Market

Discontinuous profile grinding is the best hard finishing option for large gears because it offers high flexibility and precision, but a downside is the lower productivity inherent to the process of grinding each individual profile. The principal markets that utilize large gears include:

- Wind
- Industry
- Marine
- Construction
- Agriculture
- Mining

Most of the global commitments made to the Carbon Neutrality Coalition in 2021 are centered around 2050, so the push for net-zero emissions has become the driving force behind the wind market in particular. Windmills have very demanding quality requirements that include both internal and external gear applications. Gleason offers a wide range of products serving these different market requirements.

Figure 5 Gleason's Titan 1200H has a tool changer that allows greater versatility.

Gleason's profile grinding machines are numbered by their maximum diameter in millimeters, and the P-series ranges from P400G up to the P6650G at CMD in France. Very large machines serve lower volume markets like the mining and cement industry, but the current mainstream market for large gears is the wind industry where the grinding machines range from 800–4,000 mm. Larger sizes are mainly for the internal gears, but the most popular size is 1200 mm because it covers the majority of external gears for wind.

In the 1200 mm size, Gleason offers two machine models. The new 1200G is a highly flexible and universal machine that is intended to suit most grinding needs, particularly the demands of the wind market, and it covers a wide range of applications with one grinding wheel. It is a compact, one-piece machine bed for increased rigidity and stiffness, and it can be outfitted with one of three different internal or external grinding-head options. It also supports worm-grinding options.

The other 1200 mm option is the high-performance Titan 1200H from the Titan series which has a tool changer — a feature that remains unique to the industry — for excellent surface roughness, higher productivity, and quality by using separate grinding wheels with optimized grit sizes for roughing and finishing. The machine can also be utilized to grind two different gear profiles on a single workpiece, such as a spindle with timed gears on each end, which is ideal for this machine, because it doesn't require separate setups.

Think Big

When it comes to large gears, the parts have already passed through a long value chain in their creation by the time they come to the grinding procedure. They have generally been hobbed or milled; hardened; face and bore ground; and the very last process may be the profile grinding, which means a scratch or error on your part at this stage is detrimental. In a smaller automotive gear application, there might be some sacrificial flexibility, but not with large gears. It must be correct right out of the gate.

Getting things right begins with loading and centering the workpiece on the machine table, and conventionally, this time-consuming manual alignment requires a lot of time and effort to hammer large gears into place because they can weigh tons. A crucial aspect of reducing nonproductive idle time is the ability to load a part quickly and accurately. Workpiece changeover time is now reduced up to 70% compared to manual clamping thanks to the innovation of zero-point clamping systems for automated loading. This modular system for centering and clamping automatically pre-adjusts parts on a separate setup table to reduce idle time. It's not as rapid as it would be for a small parts changer on something which would be seconds, but this does reduce idle time in changeover significantly to 5 to 10 minutes.

Manufacturability is essential. Prior to software like Gleason's KISSsoft, there were gear designers and the people who made them — two different worlds — one trying to imagine in theory the stress peaks, writing it down into a gear drawing, and the other in production struggling to make them as they were designed in theory. This was a very expensive and time-consuming trial-and-error process, and given the relative difficulty of loading large gears, that could lead to a very protracted cycle time. So, KISSsoft and technology like it bring the abstraction of design to physical reality in terms of what's possible within manufacturing parameters.

Twist influence is a KISSsoft parameter that evaluates contact patterns of gear mesh and makes micron-level adjustments to influence the stress distribution called the normal force curve. Quality demands for surface roughness — which affects NVH, durability, and service life — in the wind market are very high, which requires tightening tolerances through complex geometrical modifications onto the basic shape of the involute gear geometry itself. That ensures the gear is twist free with good surface roughness (Rz ~ 1 μm). These modifications could include additional crowning or an end relief at the tip or the root of the profile to avoid partial overload in the stress distribution. The importance of integrated design software and production can't be overstated in achieving manufacturability.

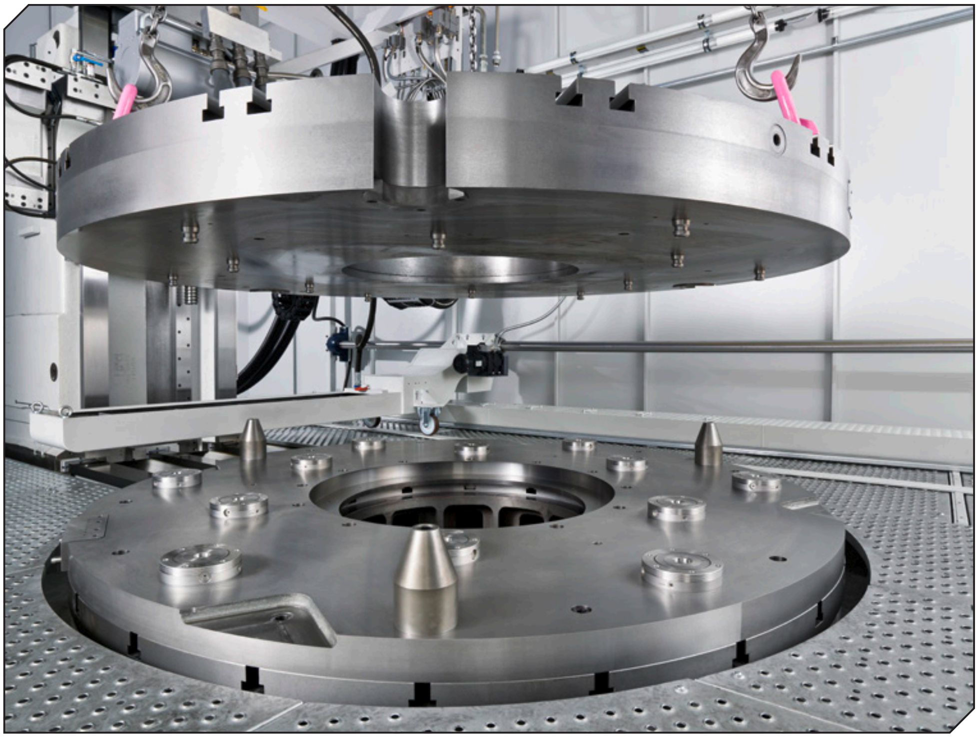

Figure 6 Zero-point clamping is crucial to reducing nonproductive time when loading and centering large gears.

Quality demands are extremely high in the wind market — not just for the required efficiency of power transmission but also because premature wear or even failure of gear drives will result in high maintenance and repair costs, especially with large turbines installed offshores. The need for the best surface qualities is one of the reasons why Gleason's Titan series was developed with a tool changer — to switch to a grinding wheel with different specifications required for extremely good surface finishes. A tool changer in other kinds of machining processes is not uncommon, but for gear grinding, the Titan machines remain unique in the industry and, by eliminating extra setups, it reduces a lot of nonproductive time.

Grinding Smarter, not Faster

Typically a machine is set up with a grinding wheel of a particular grit size, and while the dressing parameters can be adjusted to achieve different finishes from the same grinding wheel — say one for roughing and another for finishing — the finish will only be as good as its actual grit size, meaning it will not be up to the level of superfinishing allowed by switching over to a separate tool with a finer grit size. Hence, when there is only one tool to work with, the grit size is always prone to compromise with its dual task of material removal in the roughing strokes and surface finishing in the final strokes.

The temptation in grinding is to speed up the production time by adjusting the speeds and feeds of the machines, but this will invariably affect part quality and tool life, which will ultimately lead back to waste of both time, money, and other resources in the form of bad gears, energy, and undue machine wear. Stock-specific grinding eliminates any potential empty grinding strokes by using a touch probe to evaluate the orientation of the gear in relation to the grinding wheel in the event of distortion from heat treatment. The algorithm determines the high and low points from a representative sample of measurements. It averages them based on the typical ovaloid distortion of a large gear being hung in an oven for hardening. The machine will only rough pass those high teeth profiles skipping teeth that are already within range until in subsequent passes more teeth are ground until all have been brought to their finished pass dimension.

The main task of dressing is to generate the correct wheel geometry, sharpen the wheel topography, and establish a minimum runout of the wheel. The three primary dressing methods are base dressing to create a new wheel geometry, rough dressing for material removal, and finish dressing for quality surface finish and precision. The wheel can be dressed to have a more aggressive microgeometry for roughing strokes, and a finer one for finishing strokes. Smart dressing utilizes the base strokes to establish geometry by utilizing more of the dressing wheel tool surface area than just one edge, which can be preserved for dressing the grinding wheel for roughing and finishing.

With a conventional infeed strategy, a gradual amount of stock per flank is ground and hence the real material removal is increasing from stroke to stroke, but that means most of the material is removed in the last roughing stroke, which creates a high risk for thermal damage in the last stroke and greater power consumption on the grinding spindle. An infeed strategy with a distributed material removal per stroke improves productivity and process reliability. Degressive infeed means taking a more aggressive first stroke and then less with subsequent strokes hence degressive, and an A(x) infeed strategy means a slight tilt on the A-axis for the x-axis infeed ensures contact with the whole tooth profile, which means less infeed required hence fewer strokes and increasing productivity.

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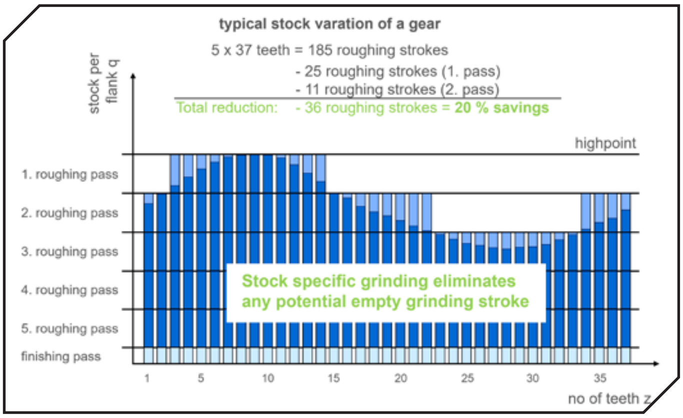

Figure 7 **Stock-specific grinding eliminates the potential for empty grinding strokes.**

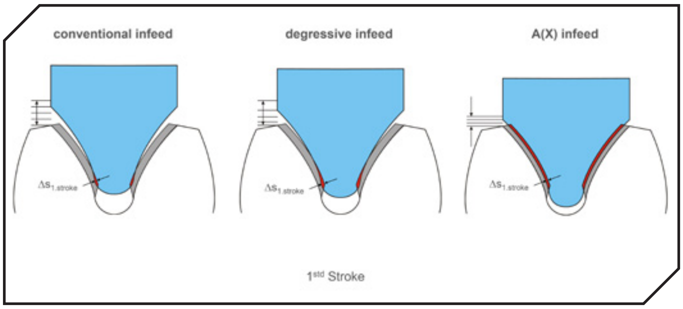

Figure 8 **Infeed strategies with distributed material removal per stroke improves productivity and process reliability.**

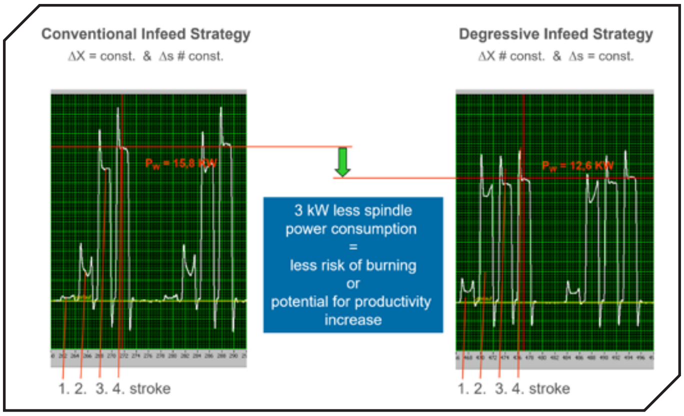

Figure 9 **Graph demonstrates the spindle power comparison between infeed strategies.**

Figure 10 Norton Xtrimium is an example of a profile grinding wheel that uses TQ and TQX technologies.

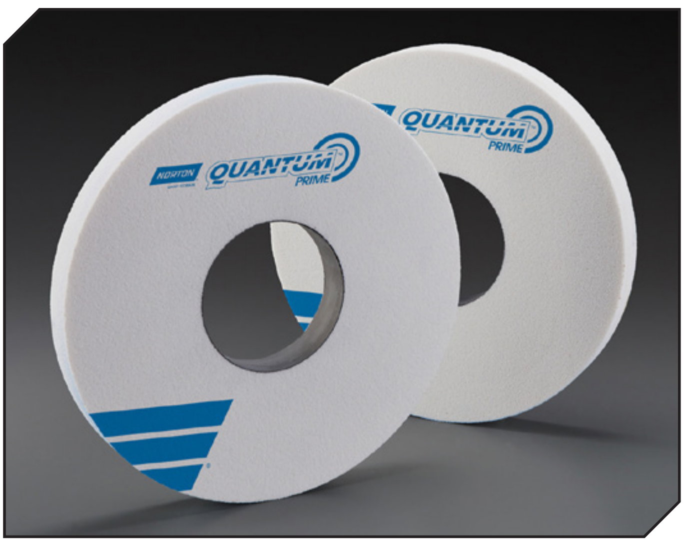

Figure 11 Norton Quantum Prime is a new ceramic grain technology designed for high productivity and quality with low power draw.

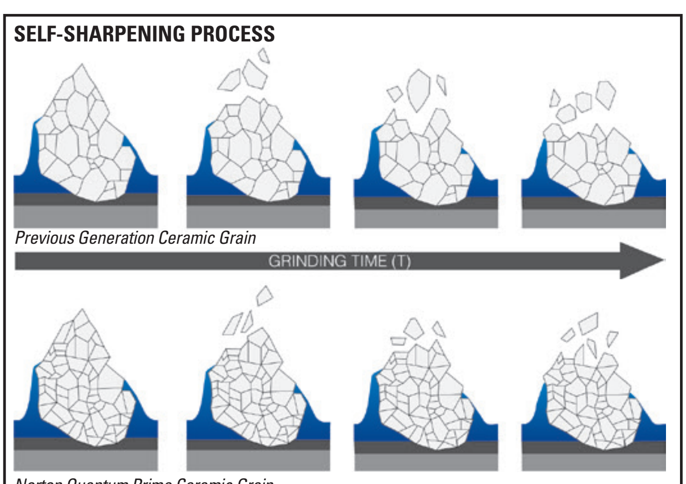

Figure 12 Ceramic grain is designed to break down smaller so it will last longer, but Norton Quantum Prime is drastically smaller than the typical ceramic grain. Illustration courtesy of Norton | Saint-Gobain Abrasives.

True Grit

Norton's straightforward approach to grinding wheel formulation is to understand what the customer is looking for, identify their pain points, and then apply three of their different technologies (TQ, Quantum Prime, or TQX), which are like ingredients because, depending on what type of metal is being ground, it's important to select the right percentage of an appropriate secondary abrasive that's both chemically the best fit for that material and cost-effective. Grinding wheels are not purchased stock, they are predominantly made to order. However, there are blank wheels of certain common compositions that are kept in an unfinished state, which allows for a quick turnaround.

TQ, Norton's go-to profile gear technology, is a shape-engineered aluminum-oxide grain that economically offers speed and quality, which is the best of both worlds. To meet the highest quality demands, Quantum Prime is used because it provides the smallest tooth-to-tooth variability in flank and profile at very low power levels. This is possible due to the combination of three critical technologies: Quantum Prime, Vortex2, and Vitrium3. Quantum Prime is a brand-new ceramic grain technology designed for high productivity and quality with low power draw, meaning a lower risk of grinding burn. Vortex2 utilizes agglomerated aluminum oxide to create an engineered porosity that maximizes coolant in the grind zone and improve formholding due to improved homogeneity of the porosity. Vitrium3 is a strong bond that allows the abrasive grains to be used to their fullest before introducing new sharp grains. Quantum Prime works well at a high speed but not as much as a shaped abrasive like TQX, which is built for speed. With its long shaped-grain abrasive, TQX allows very high removal rates while keeping cool. A high-quality surface finish is sacrificed at those speeds, but for certain applications that's acceptable.

Quality is the number one priority which means not introducing heat into the part while maintaining superior formholding and surface finish. And secondly, finding opportunities to control cost—by factoring for parts per dress, wheel life, dress depth, and cycle time—are all considered and optimized to save the most money for the customer.

The Secret Ingredient

It may seem counterintuitive, but the secret ingredient is nothing—in a grinding wheel, empty space is as important as the physical ingredients. The more consistent the size and spacing of the pores, the better the wheel will perform. Problems of variation will cause different parts of the wheel to break down at a faster or slower rate than others. You *do* want the wheel to break down—that means it is resharpening itself—but you want it to break down very consistently. Consistency ensures there is no heat buildup or excessive breakdown. These are concepts that are basic to grinding but considering the precision requirements for gears, they need to be harmonized.

Grinding wheel design has several priorities and grinding without burn is number one and formholding second. Speed affects both. The goal is to get a surface finish that minimizes the amount of variation as you grind. The chemistry of holding the abrasive grain in the bond matrix for a precise amount of time is referred to as "grain adhesion science." Bond posts — the optimal amount of bond needed to hold the grain in place without burning and to maintain the form — hold the grain in place and then the grain itself does the grinding while promoting a more controlled breakdown of the abrasive. That leads to less variation in the profile and flank and consistency in surface finish. Quantum Prime breaks down into extremely small pieces. Ceramic grain is designed to break down smaller so it will last longer, but Quantum Prime's crystal size is drastically smaller than the typical ceramic grain. With gears specifically, the finish it achieves addresses the issues of NVH very well because it is so controlled and consistent along with a low risk of burning.

Dress to Impress

Once you move away from conventional abrasives with very low or no ceramic grain, you can use normal lower-tier diamond rolls called infiltrated diamond rolls which use a certain type of metal bond to secure the diamond. But once you get into the ceramic-grain grinding wheels used for greater material removal on large gears, Norton recommends a chemical vapor deposition (CVD) reinforced diamond roll for dressing, which is larger and very precisely sized and shaped for those applications. Because the chemical nature of the CVD is tougher, it will last longer when dressing the ceramic-grain grinding wheels. That's recommended particularly on larger wheels where you have a 6 or 8 in. diamond roll but you have a 12 or 14 in. grinding wheel.

Profile gear wheels typically range from 100–450 mm in diameter. Sizing is driven by the parameters of the machines which have a minimum and maximum OD they accept. For large gears, the minimum might be 300 mm because you want to be able to dress the double bevel into the wheel itself but also reach the full depth of the gear tooth. But you also want the wheel to be large enough to last an appropriate amount of time because tool changing is time-consuming, so ideally that same wheel will complete the entire gear or multiple workpieces.

One thing is clear from speaking with Gleason and Norton: toolmakers don't sell tools, they sell process solutions to technical problems. That is an essential point of view to take into consideration when quality and precision are paramount. These relationships often begin with a part drawing and then a process to support the creation of that part is developed according to appropriate time and quality standards. Tools are not bought in isolation but as part of a whole. A process acceptance involves a proving demonstration. So it's not like buying a suit off the rack, it is like having a team of tailors make one bespoke to your exact measurements just as you need a process that suits your needs.

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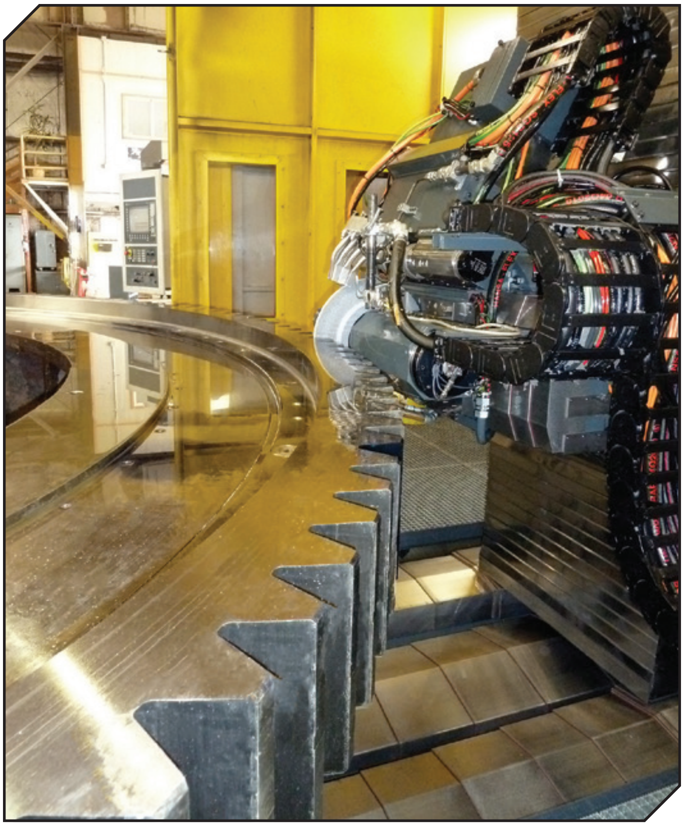

Figure 13 For large gears, the diameter of a grinding wheel is sized to reach the full depth of the gear tooth but also last long enough to avoid a mid-operation wheel change. Image Courtesy of Norton | Saint-Gobain Abrasives.

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Changing Company Culture — Meeting Goals to Achieve Improvement

Joe Arvin

Overview

In manufacturing, we often hear discussions about the importance of company culture and its impact on the overall successful operation of an organization. What does this really mean? What makes a company's culture good or bad? Finally, how do you improve company culture? I would like to make my points by highlighting the two fictional companies, BloatCo and Slimline Corp.

BloatCo

To begin, let's first hear from Jim, the president of BloatCo.

"We have some serious problems with our quality, productivity and delivery. When it comes to corrective actions, it's tough to know where to start. To be honest, our shop can be described as being in chaos most of the time. Many of the operators have radios blaring at their workstations. To make matters worse, tardiness is frequent and some of our most experienced workers are not concerned about coming in late and take their time getting back to their workstations after lunch. These people are prima donnas because they know it would be difficult or impossible to replace them. They set a bad example that others tend to follow. When there have been efforts to get them in line, the supervisors have been met with borderline insubordination. This leaves us without many choices because if we get rid of them, this will be even more detrimental to our operation.

"So, in general, our company is divided into two camps — the office and the shop. There is little cooperation between the two and we can't make any headway toward improvement. In all it's a pretty toxic environment."

All I can say is WOW! BloatCo certainly sounds like an organization with a serious company culture problem.

Slimline Corp

On the other side of town, we have Slimline Corp who is in direct competition with BloatCo. Let's listen to Phil, the General Manager, as he talks about his organization.

"When I started here at Slimline Corp, there was a pretty bad culture. It was very much like things were when I worked at BloatCo. It took a while and a lot of effort, but we have really turned things around. Today, our operators are actively involved in the forward progress of our company, working together with management. Our goals are the same and everyone works diligently to meet those goals. We are meeting our quality, productivity and delivery schedules. Things aren't perfect, but we now have a culture where positive changes can be worked on by everyone."

Now that's more like it. How did Phil do this and what can be done to modify the behavior of the employees at BloatCo? The answer can often be found in changing the culture, but what measures can a manager implement to make this happen?

The following are some points to consider for optimizing the culture of your organization.

Patience and Perseverance

The first thing you have to expect with changing a company's culture is that it will take some time and is not going to happen overnight. You must have a detailed plan and make the implementation of that plan a top priority.

A Vision for Culture Change

Next, it's important to understand what culture change really is. It does not mean transforming your company into some type of utopia where everyone is in a perpetual state of bliss. Culture change must focus on a specific goal involving something you want to improve. This goal must be something that everyone understands. For example, creating a positive work environment is not really a tangible goal. On the other hand, improving delivery is something that you can really measure, which will genuinely impact the organization.

Managing Change

I'm sure that everyone is aware that managing change can be very tricky. People tend to lean on what is familiar. I believe the best place to start change is by instilling a positive mentality into your people. You need to get all of your employees excited about what they are doing. If they begin to believe that what they are doing is important, they will be more willing to adapt to changes in order to fulfill that expectation.

Avoid Change Overload

One mistake that is commonly made when wanting to implement change is having too many objectives on the company's proverbial plate at one time. Organizations can only really effectively work on three changes concurrently, and culture change can be one of these. Just keep in mind that if there is too much going on at the same time, the efforts will become diluted.

Getting People Involved

Speaking of positive worker mindset, a good way to fortify this is to get people involved and have them feel that their thoughts and input are genuinely valued. Ask for their help. It is important to remember that production work in manufacturing can be tedious, causing operators to feel that they are just another part of the machinery. Getting them involved is a good way to effectively counter this.

I first learned how important it was to get the operator's input back when I was a supervisor at Indiana Gear. On one occasion, the company had made a unilateral decision to buy a new gear grinder with no input from the operators, who felt it was the wrong machine. When the new machine arrived, they did not have a lot of motivation to make sure the machine worked as it should. As a result, in this case, the machine never really worked successfully and it was eventually sold. I'm sure there would have been a different outcome if the operators had been consulted. I'm sure at some level there was probably a degree of subconscious psychological sabotage that took place, which is just human nature.

Bullying

In manufacturing, there is often a lot at stake in the process of getting a quality product out the door. Sometimes, tempers can flare. This is how people can be from time to time, but when aggressive behavior goes over the line of acceptability, you get bullying. If this is allowed to happen in your company, whether it is a supervisor or the CEO, just be aware that this will undermine all of your other positive measures. Nothing will kill worker enthusiasm and eagerness to contribute faster than a manager who throws his weight around while having temper tantrums. This should not be allowed to continue if you hope to improve your culture.

Communication and Access

On the other hand, positive and open communication is essential. It is important to talk to people to establish a relationship. This not only applies to formal meetings, but informal one-on-one discussions. Managers should walk the shop floor every day and talk to people, learning about them, what they're doing, and any problems they may have. This is an important step for providing the employees with the opportunity to interact with the manager, which they might not otherwise have.

Promote the Company

Another point about keeping workers enthused about what they do involves the promotion of your company. Most organizations are very adept at promoting themselves to their market and customers. In many cases, it is just as important to share this promotional message with each of your employees. Someone running a machine way in the back of your plant may otherwise have no way of accessing this information. Be sure they are exposed to this messaging by posting photos of the end products on the bulletin boards. Also, place copies of your brochures and other printed collateral material throughout the plant and office. If there is a new promotional video, make sure they are able to view it as well.

Accountability

Accountability may seem like a tired old topic from a management course, but it is essential. Having an environment of accountability is much more than holding people's feet to the fire when they mess up. In reality, it should also mean rewarding and acknowledging people's accomplishments, and providing coaching when there are problems.

Cross-Training

Cross-training may take some extra focus and effort, but it has many benefits. If an operator is off sick when a hot job needs to run, there will be someone to fill in. If at all possible, have someone trained on the equipment operated by one of the "prima donnas" who use their knowledge as a means of holding the company hostage. Cross-training will go far in helping to diffuse this. Finally, when workers have the opportunity to perform other functions on the shop floor, they will benefit from getting a broader perspective on the manufacture of your product and the importance of their primary operation. Plus, you might find that they excel in another area that you did not anticipate.

Discipline

As much as we'd all like to avoid resorting to disciplinary procedures, it is a necessary part of any environment where leaders need to ensure that everyone is working together effectively toward a common goal. However, it must be done right. This is why a Progressive Disciplinary Procedure, which is fair, must be in place and followed to the letter.

Here are some key points about this type of procedure.

- Make sure everyone in the company is aware of all aspects of the company's policies and procedures.

- Train your supervisors so they know the procedure very well and how to navigate through it effectively.
- Be consistent with following disciplinary steps.
- Realize that sometimes you might have to discipline good workers, and you might have to coach people you'd like to fire on the spot.
- Be sure that Management always has the back of the supervisors.
- Make sure all verbal and written disciplinary action records are given to HR.

Here is an approach I used when starting a disciplinary procedure for behavior. I would begin by privately explaining to the employee that the part of my job which I disliked the most was disciplinary action. I would then say, "You were hired to follow company policies, be productive, and not disruptive. I was put in my position to ensure company policies and procedures are followed. I want to do my job, so if you don't change, you will be forcing me to discipline you!"

Incentive Programs

Many organizations look at incentive programs when working to change their culture. It has been my experience that culture change can not really happen without incentives, both positive and negative. In all, the positive incentives are what you should focus on.

The go-to incentive is usually financial in nature. Remember that financial compensation is usually only a short-term motivator. For example, adding $100 to someone's paycheck, which is subject to withholding, is always welcomed, but soon forgotten. Then compare that to the president handing someone a crisp new $100 bill and personally thanking them in front of their coworkers when goals are met. The latter will have dramatically more impact. What it comes down to is that when an incentive is tied to recognition when goals are met — that is powerful.

You might also consider profit sharing, which is also a strong form of incentive, and if goals are not met, the profit sharing is gone.

Conclusion

Working with other people toward a common goal is the basic framework of a manufacturing operation. Ensuring a positive environment that runs smoothly can be very tricky, but taking measures to promote an optimal culture is a powerful step toward ongoing success.

It is my hope that these points give you some ideas that you can implement to maximize your manufacturing operation. Finally, I would like to acknowledge the following individuals for their valuable input in the development of this article. This includes Steve Carroll of Ken Mac Metals, and Wayne Hanna of Brad Foote Gear.

A Final Word

If you have any questions or comments, I would look forward to hearing from you. Also, if you missed any of my previous articles, the right column includes a list by issue number and page. They are also available on the *Gear Technology* website. If you'd like for me to send you a copy, please send me an email or just give me a call.

Joe Arvin is a veteran of the gear manufacturing industry. After 40 years at Arrow Gear Company, Joe Arvin is now president of Arvin Global Solutions (AGS). AGS offers a full range of consulting services to the manufacturing industry. His website is www.ArvinGlobalSolutions.com and he can be reached by email at ArvinGlobal@Gmail.com.

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WE WANT YOU!

Phillip Olson, Director, AGMA Technical Services

AGMA wants you to be involved in gear standards development. The creation of standards helps drive innovation and increase the market value of gear design and manufacturing—it also promotes international trade and commerce, which in turn fuels more innovation. The AGMA Gear Accuracy committee is in the early planning stages for a comprehensive review, and possible revision, of the standard ANSI/AGMA 2116, Evaluation of Double Flank Testers for Radial Composite Measurement of Gears, and we need your input. Committee meetings are a great place to network and collaborate with experts in the field, broaden your knowledge, capture technical expertise in writing, refine the standards you use and see how your influence helps shape best practices throughout America and around the world.

The condition and alignment of gear measuring instruments can greatly influence the measurement of production gears, and *ANSI/AGMA 2116* provides evaluation methods of double flank testers used to evaluate radial composite deviations of gears, as specified in *ANSI/AGMA/ISO 1328-2*. Your frontline experience is essential to describe these procedures.

ANSI/AGMA 2116 was developed to work in tandem with the information sheet *AGMA 935, Recommendations Relative to the Evaluation of Radial Composite Gear Double Flank Testers. ANSI/AGMA 2116* provides requirements, whereas *AGMA 935* provides the recommended procedures for testing the inspection equipment for those requirements. In addition to a general update, the new project also aims combine the two documents into one, which will then be proposed to ISO as the foundation for a new ISO document on the subject matter. With the general update, the committee will ensure that *ANSI/AGMA 2116* continues to provide the industry with the latest industry-accepted, state-of-the-art practices. Additional aims are to review how gauge repeatability and reproducibility are discussed for a dynamic system.

From a company perspective, being involved in standards development saves time and money in a variety of ways, including reduction of redundancy, improved quality, and safety, and better focusing of R&D resources. Also note that if your company's not at the table helping to write the latest standards, the standards that affect your business will be written by your competitors. For the health of our industry, please reach out and make your experience a part of this living record.

Perhaps you are among the many gear shops that use double flank testers and potentially comprise the majority stakeholders in this project who stand to directly benefit the most from being a part of this work. In addition to user stakeholders of this equipment, we are especially looking for companies that manufacture and calibrate double flank testers. Interested stakeholders will be invited to a virtual meeting to determine the project scope, outline the project milestones, and assess the project feasibility. For more information and to be registered as an "interested stakeholder," please contact the AGMA technical division at *tech@agma.org* before July 22.

For over 100 years, AGMA has been the facilitator for the development of American gear standards. For AGMA to make gear standards the best they can be, everyone in the industry needs to be involved. When AGMA standards-writing technical committees have open projects, they meet approximately six times per year for two-hour virtual meetings, and approximately once per year for a two-day in-person meeting.

Behind the scenes of almost every good and service, there are standards showing the industry how to make superior products. Standards provide a common language, document years of collective experience on proven and verified practices, and are the generally accepted rules, guidelines and requirements within an industry. In the United States, the stakeholders are in the driver's seat of standards development. Those who will use, and are affected by, a pro-

posed standard are the ones tasked with writing it. Standards development is a democratic, free and open process that requires consensus before publishing. After publication, a standard's fate rests with customers and suppliers to mutually, and voluntarily, agree to adopt the standard.

Double Differential for Electric Vehicle and Hybrid Transmissions — Sophisticated Simplicity

Dr. Hermann Stadtfeld and Dr. Haris Ligata

What Are the Common High Reduction Transmissions?

The duty of high reduction transmissions is reducing high input rpm into lower rpms, for example to propel the wheels of a vehicle or the rotor of a helicopter. The output rpm of such a transmission is in the range between 0 and 1,000 rpm. The input rpm can be 20,000 rpm and higher if the prime mover is an electric motor or a jet engine.

The conventional transmissions which can be operated with high input speeds and can accomplish high reductions are:

- Multistage transmissions employing cylindrical gears
- Planetary transmissions
- Bevel worm gear reductions with ratios of 20 in one stage
- Pericyclic transmissions with nutating bevel gears
- Cycloidal transmissions

Multistage Transmissions Employing Cylindrical Gears

Multi-stage transmissions with cylindrical gears require a multitude of shafts with bearings and gears. For a reduction ratio of 20, at least four stages are required. Four reduction stages require four shafts, eight bearings and four gear meshes. Only the observation of four gear meshes indicates an overall efficiency of 92.2% if the efficiency of one single stage is 98% ($0.98^4 = 0.922$). Four stage cylindrical transmissions require a rather large transmission housing envelope (Ref. 1).

Planetary Transmissions

Depending on the design, planetary transmission, in connection with conventional cylindrical gear reduction stages, can achieve high ratios and obtain high efficiency.

Bevel Worm Gear Reductions with Ratios of 20 in One Stage

Bevel worm gear drives are, for example, called High Reduction Hypoids (HRH) or Super Reduction Hypoids (SRH). The worm shaped pinions have one to five teeth and the ring gears have typically 27 to 75 teeth. The maximal achievable ratios are in the range of 75. Ratios above 15 only have a reduced back driving capability. Gear sets without back driving capability are self-locking. Self-locking gear sets cannot be used in a vehicle drive train or in a helicopter main rotor drive. Bevel worm gear drives also create high sliding velocities due to the large component in face width direction. A five-tooth SRH pinion, meshing with a 60-tooth ring gear creates 617 m/min relative sliding between the flank surfaces with a pinion speed of 10,000 rpm (equal transmission input speed). This is higher than the maximum sliding expected in a hypoid axle drive of a sports car while driving faster than 200 km/h (125 mph) with a pinion speed of 4,000 rpm. The example explains that a doubling of the transmission input will not only reduce the efficiency but also has the risk of surface damage and premature failure (Refs. 2, 3).

Pericyclic Transmissions with Nutating Bevel Gears

Pericyclic transmissions as introduced in (Refs. 4, 5, 6) can achieve very high reductions in the range of 20 to 100 without generating high relative surface sliding. As the shaft angle between two bevel gears approaches 180°, the relative sliding velocity drops down to zero. Because of shaft angles higher than 160° in the most common pericyclic transmissions, the relative sliding velocities are uncritical, even if the input speeds are 20,000 rpm or higher. Pericyclic transmissions have angled bearing seats of the nutating members and the high forces which are applied to the bearing at the angled seat have to be supported with pre-loaded tapered roller bearings. Another possible area of attention in pericyclic transmissions are the fluctuating axial mass forces the nutating members generate. High speed pericyclic transmissions require a mirror image arrangement of an even number of nutating members as well as precise timing of the gears and precise balancing.

Cycloidal Transmissions

Cycloidal transmissions are the two-dimensional analog to pericyclic transmissions. One revolution of the eccentric input shaft will rotate the output shaft by one to two tooth pitches. The radial mass forces of cycloidal transmissions cannot be compensated by a second cycloidal disk arrangement side by side. As a result, high reduction cycloidal transmissions are only used when low input speeds are reduced to very low output speeds.

If high ratios between 10 and 100 should be achieved, designers prefer multi-stage cylindrical transmissions often combined with planetary reductions. Multistage transmissions are often applied in the industry and deliver a reasonable power density.

For future high reduction transmissions, it is desirable to create a very compact high reduction transmission with easy to manufacture components and predictable operating conditions. If all involved parts are well known as standard machine design components, then the prediction of durability and endurance life is possible by applying the calculation algorithms provided by the standards of the AGMA (American Gear Manufacturers Association), ISO (International Standardization Organization) and other national standards. Those algorithms rely on tenths of thousands of fatigue life testing as well as many application factors which have been evaluated for many decades. In safety engineering, those proven algorithms and application factors are the engineer's most valuable tools.

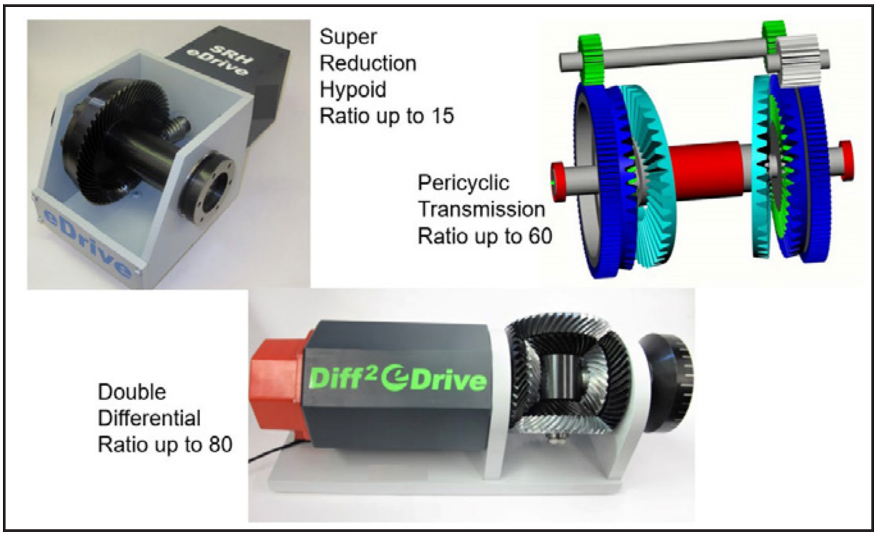

Figure 1 High ratio bevel gear transmissions.

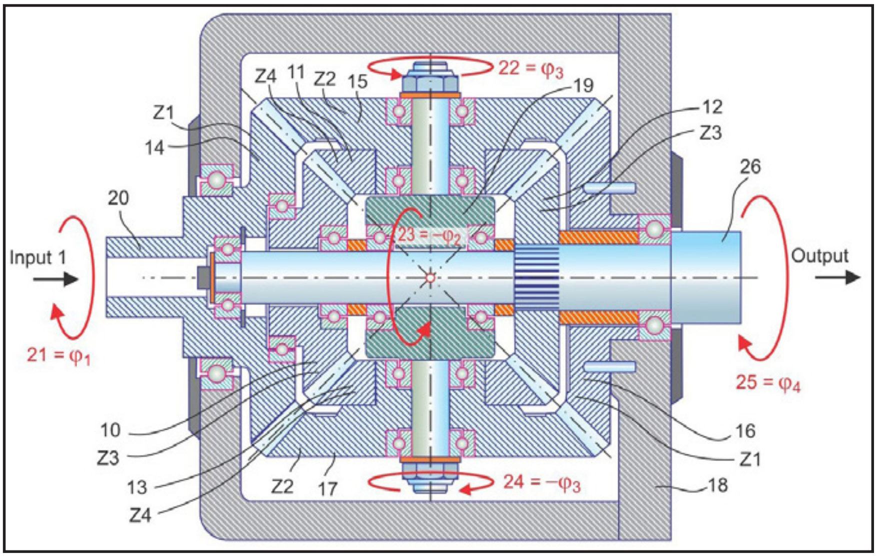

Figure 2 Double differential transmission.

Bevel Gear Based High Ratio Transmissions

Three interesting solutions of high-speed reducers which are based on bevel gears are shown in Figure 1. Automotive transmissions cannot be self-locking and must provide a good efficiency. The super reduction hypoid (SRH) on the top, left is limited to a ratio of about 15 to fulfill these two requirements. The SRH arrangement of input shaft and the two drive shaft output flanges is ideal for the adaptation between an electric motor and the driving wheels. Only two shafts and four bearings are required, which makes the SRH solution very cost effective and compact. The differential unit can be placed inside of the ring gear, similar as it is in case of a hypoid axle drive unit.

A high ratio solution realized with straight bevel gears is shown on the top right in Figure 1. This pericyclic reducer requires 6 straight bevel gears to perform two opposite nutating motions. Each revolution of the nutating (light blue) gears will rotate the green output gears by one or two pitches, depending on the number of teeth of the dark blue reaction gears. The contact ratio of this unit is very high, such that 6 to 10 pairs of teeth are always in contact.

The latest, bevel gear based high reduction e-drive is the double differential in the bottom photo in Figure 1. The double differential is very compact and can realize ratios from 5 to 80 and even higher. The following chapters will explain its functionality and its advantages as electric vehicle transmission.

What is a Double Differential?

The new developed solution for a low, medium, or high reduction transmission with high power density and the application of standard design elements is the double differential, shown in Figure 2.

The double differential transmission is symmetric and has a high-power density. The input rotation 21 from shaft 20 is transmitted to gears 15 and 17 and causes a rotation 22 of gear 15, and a rotation 24 of gear 17. Gears 15 and 17 are both in mesh with gear 16. Gear 16 is rigidly connected to the housing 18. The fact that gear 16 cannot rotate will cause a rotation 23 of the carrier 19. Gears 15 and 11 as well as gears 17 and 13 are rotationally constrained with each other, for example via a spline connection. The carrier rotation 23 gives a first component of rotation to output gear 12. The rotations 22 and 24 add a second component of rotation to output gear 12. If all eight involved bevel gears have the same number of teeth, then the output rotation 25 would be zero. The explanation is that, for example a 90° rotation ϕ_2 of the carrier 19 would rotate gears 15 and 17 by 90° in the directions 22 and 24. The output gear 12 therefore receives a 90° rotation ϕ_2 from the carrier and a 90° rotation ϕ_3 (in the opposite direction) from the gears 11 and 13 and as a result will not rotate, independent from the input rotation 21.

While this example seems not of any obvious practical interest, the example was merely used to demonstrate the interesting functionality of double differential transmissions. In the example, the ratio is $\phi_1/\phi_4 = \infty$.

A derivation of the equation for the ratio by using individual numbers of teeth provides the ability to find the variety of possible ratios by variation of the tooth numbers of the gears 14/16 versus 15/17 and 10/12 versus 11/13.

$$\phi_2/\phi_3 = z_2/z_1 \quad (1)$$
or: $$\phi_3 = \phi_2 \cdot z_1/z_2 \quad (2)$$
$$\phi_4 = \phi_2 - \phi_3 \cdot z_4/z_3 \quad (3)$$
$$\phi_1 = \phi_2 + \phi_3 \cdot z_2/z_1 \quad (4)$$
plug (2) in (4): $$\phi_1 = \phi_2 + \phi_2 = 2 \cdot \phi_2 \quad (5)$$
or: $$\phi_2 = \phi_1/2 \quad (6)$$
plug (6) in (3): $$\phi_4 = \phi_1/2 - \phi_3 \cdot z_4/z_3 \quad (7)$$
plug (6) in (2): $$\phi_3 = \phi_1/2 \cdot z_1/z_2 \quad (8)$$
plug (8) in (7): $$\phi_4 = \phi_1/2 \cdot [1 - z_1/z_2 \cdot z_4/z_3] \quad (9)$$
re-arranged: $$R = \phi_1/\phi_4 = 2/[1-(z_1 \cdot z_4)/(z_2 \cdot z_3)] \quad (10)$$

whereas:

z_1… Number of teeth gear 14 and gear 16
z_2… Number of teeth gear 15 and gear 17
z_3… Number of teeth gear 10 and gear 12
z_4… Number of teeth gear 11 and gear 13
ϕ_1… Angle of rotation gear 14
ϕ_2… Angle of rotation carrier 19
ϕ_3… Angle of rotation gear 15 (and gear 17 in negative ϕ_3 direction)
ϕ_4… Angle of rotation gear 12 (and output shaft 26)
R… Ratio of input speed divided by output speed

In the following four examples, different numbers of teeth combinations are used to demonstrate the extremely high range of ratios that can be realized with the double differential without a significant change of the transmission size:

Example 1: $z_1 = 40$; $z_2 = 39$; $z_3 = 40$; $z_4 = 40$; Ratio R = -78.000
Example 2: $z_1 = 40$; $z_2 = 41$; $z_3 = 40$; $z_4 = 40$; Ratio R = 82.000
Example 3: $z_1 = 45$; $z_2 = 50$; $z_3 = 40$; $z_4 = 40$; Ratio R = 20.000
Example 4: $z_1 = 30$; $z_2 = 50$; $z_3 = 40$; $z_4 = 40$; Ratio R = 5.000

Extended Double Differential with Two Inputs

A possible extension of the function of the double differential transmission is shown in Figure 3. In addition to the graphic in Figure 2, in Figure 3 the gears 30, 31 and shaft 32 have been added. Gear 16 is connected to a cylindrical gear 30, which is arranged rotatable to the housing 18, and in mesh with pinion 31, which is connected to a second input shaft 32. This possibility of a second input allows a variety of interesting input speed combinations with two different prime movers, e.g., electrical motors, which have different speed and torque characteristics. One motor, for example, can be a high torque and low speed motor which runs on a constant speed signal without speed regulation. The second motor would then, for example, rotate backwards if an output rpm of 0 is required. In case of quick acceleration up to a vehicle cruising speed, the second motor is first slowed down to 0 rpm and the stored kinetic energy of the differential gears and the fast-rotating carrier is used for the vehicle acceleration. Several seconds later, when the vehicle reaches half of its cruising speed, the second motor starts to rotate in positive rotational direction to accelerate the vehicle further to the desired speed. In a conventional electric vehicle drive system, high amounts of energy are drawn from the battery during this acceleration. The extended double differential allows storing kinetic energy during gentle driving periods and during deceleration and breaking actions which can be used as described before.

In the case of two inputs, there is not one particular number for the ratio which leads to the following relationship between the output rotation and the two input rotations:

$$\phi_2 = \phi_3 \cdot z_2/z_1 \quad (11)$$
or: $$\phi_3 = (\phi_2 - \phi_5) \cdot z_1/z_2 \quad (12)$$
$$\phi_4 = \phi_2 - \phi_3 \cdot z_4/z_3 \quad (13)$$
$$\phi_1 = \phi_2 + \phi_3 \cdot z_1/z_2 \quad (14)$$
insert (12) in (14): $$\phi_1 = \phi_2 + (\phi_2 - \phi_5) \cdot z_1/z_2 \cdot z_2/z_1 = 2 \cdot \phi_2 - \phi_5 \quad (15)$$
or: $$\phi_2 = (\phi_1 + \phi_5)/2 \quad (16)$$
insert (16) in (13): $$\phi_4 = (\phi_1 + \phi_5)/2 - \phi_3 \cdot z_4/z_3 \quad (17)$$
insert (16) in (12): $$\phi_3 = [(\phi_1 + \phi_5)/2 - \phi_5] \cdot z_1/z_2 \quad (18)$$
insert (18) in (17): $$\phi_4 = (\phi_1 + \phi_5)/2 \cdot [1 - z_1/z_2 \cdot z_4/z_3] + \phi_5 \cdot z_1/z_2 \cdot z_4/z_3 \quad (19)$$
second input rotation: $$\phi_6 = -\phi_5 \cdot z_5/z_6 \quad (20)$$

whereas:

z_5… Number of teeth gear 30
z_6… Number of teeth gear 31

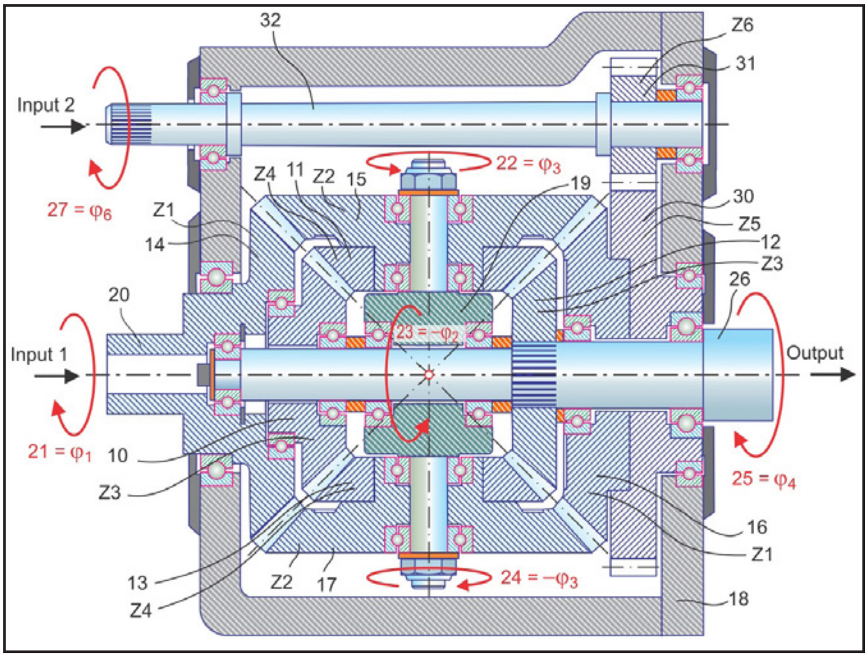

Figure 3 **Extended double differential with two inputs.**

ϕ_5… Rotation angle of gears 16 and 30

Two special cases can be encountered by applying Equation 19 for different input rotations ϕ_6. In case 1, the output speed (rotation angle ϕ_4) is equal to the speed of gear 16 (rotation angle ϕ_5). In this case, the output rotation ϕ_4 is equal to the input rotation ϕ_1, which results in a ratio of R = 1.00:

$\phi_5 = \phi_4$ inserted in (19): $\phi_4 = (\phi_1+\phi_4)/2 \cdot (1-z_1/z_2 \cdot z_4/z_3)+\phi_4 \cdot z_1/z_2 \cdot z_4/z_3$ (21)

(21) solved for ϕ_4: $\phi_4/2 \cdot (1-z_1/z_2 \cdot z_4/z_3) = \phi_1/2 \cdot (1-z_1/z_2 \cdot z_4/z_3)$ (22)

or reduced: $\phi_4 = \phi_1$ (23)

resulting in: R = 1.00 (24)

In case 2, the input rotation ϕ_5 is zero, which simplifies Equation 19, and it becomes equal to Equation 9:

$\phi_5 = 0$ plugged in (19): $\phi_4 = (\phi_1+0) \cdot (1-z_1/z_2 \cdot z_4/z_3)+0 \cdot z_1/z_2 \cdot z_4/z_3$ (25)

elimination of zero terms: $\phi_4 = \phi_1/2 \cdot [1- z_1/z_2 \cdot z_4/z_3]$ (26)

Equation 26 is equal to Equation 9. Equation 9 is based on the fact that gear 16 is rigidly connected to the transmission housing which presents the case $\phi_5 = 0$, which in turn proves that Equation 19 is conclusive.

The gears in a double differential can be straight bevel gears, spiral bevel gears, or face gears with additional helical gears for the second input. In case of high input speeds, ground spiral bevel gears will deliver the highest efficiency and the lowest noise emission in connection with a high load carrying capacity. The axial forces, which are the result of the normal flank forces, can be minimized in a double differential by using reversed spiral angles and adjusting the values of the spiral angles for the outer and the inner planets differently for the outer and inner side gears. For the bearing dimensioning of the planets, the expected centrifugal forces have to be considered.

Due to the fact that no hypoid offsets are used, the relative surface sliding has no component in face width direction but consists only of profile sliding. The relative profile sliding of a spiral bevel gear set with a ratio which is close to 1.0 and an outer diameter of 120 mm (typical for automotive double differential transmissions) with a speed of 1,000 rpm amounts to a maximum of about 84 m/min. The relative speed between the two fastest gears (14 and 15) in a double differential transmission is only about 50% of the input speed. Equation 8, $\phi_3 = \phi_1/2 \cdot z_1/z_2$ delivers a speed of gear 15 which is only 48.8% of the input speed, if $z_1 = 40$ and $z_2 = 41$ ($\phi_3 = \phi_1/2 \cdot 40/41 = 0.488 \cdot \phi_1$) The relative speed between gear 14 and gear 15 is therefore in this case $\phi_1 - \phi_3 = 0.512 \cdot \phi_1$. This means the relative speed between the fastest gears in a double differential transmission is typically only about half of the input speed. If the input speed is 10,000 rpm, then the double differential has only $10 \cdot 84$ m/min $\cdot 0.512 = 430.08$ m/min. Compared to a standard spiral bevel gear transmission, the double differential transmission has in this case only 51.2% of the sliding velocity.

The expanded double differential allows a variety of interesting applications due to the second input (input 2). If, for example, input 2 is connected to a low-speed high torque motor and input 1 is connected to a high-speed low torque motor which rotates, then it is possible to choose the speed of input 1 (e.g., 25,333 rpm) and of input 2 (e.g., 4,000 rpm) such that the output speed is 0 rpm. This example is based on the following number of teeth:

$$z_1 = 45;\ z_2 = 50;\ z_3 = 40;\ z_4 = 40;\ z_5 = 60;\ z_6 = 20$$

with a speed of input 2 (shaft 32) of $n_6 = 4{,}000$ rpm, and the first reduction $-z_6/z_5 = -20/60$ the speed of gear 30 is equal to $n_5 = -1{,}333$ rpm. The speed of the output shaft is $n_4 = 0$.

Equation 19 is also valid if instead of the angles ϕ, the rotational speeds n in rpm are used:

$$n_4 = (n_1+n_5)/2 \cdot [1-z_1/z_2 \cdot z_4/z_3]+n_5 \cdot z_1/z_2 \cdot z_4/z_3$$

becomes: $0 = (n_1 - 1{,}333)/2 \cdot [1-45/50 \cdot 40/40] - 1{,}333 \cdot 45/50 \cdot 40/40$

or: $0 = (n_1/2 - 666.7) \cdot 0.1 - 1{,}200$

resulting in: $n_1 = 25{,}333$ rpm

The practical application of this example can be a vehicle which slows down from cruising speed to a full stop in front of an intersection traffic light (Figure 4, left to center). During this deceleration the kinetic energy "moves" from the vehicle body to the carrier of the double differential. When the vehicle stands still at the red light, the high-speed motor rotates at 25,333 rpm (and the carrier with the planets with 12,666 rpm). All the kinetic energy from the moving vehicle body (less friction losses) is now stored in the fast-rotating carrier and the planets.

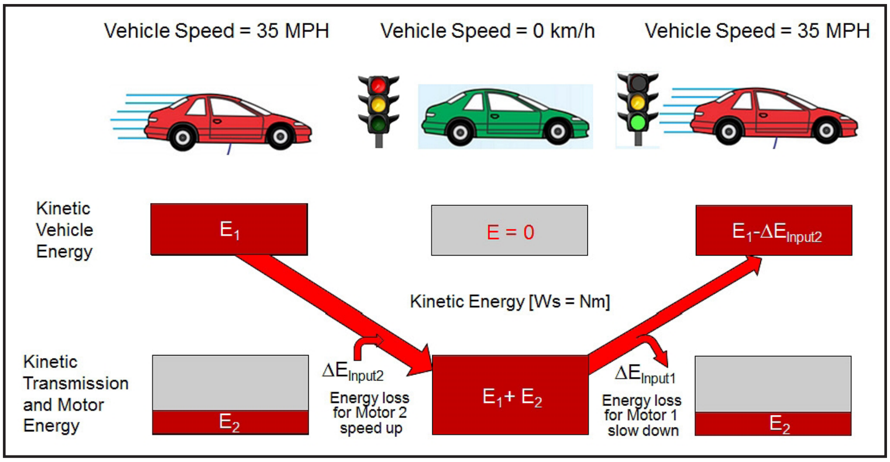

Figure 4 Energy balance—vehicle with mechanical energy storage.

After the traffic light changes to green, n_6 is reduced by the vehicle control electronics to down to zero rpm. This will create a resistance from the output shaft, which reduces n_1 from 25,333 rpm to 8,840 rpm, while n_6 reduces to zero rpm, which will accelerate the vehicle from 0 to 35 mph, less the friction losses (Figure 4, center to right). During the acceleration period, the kinetic energy of the double differential assembly with gears 10, 11, 12, 13, 14, 15, 16 and 17 as well as the carrier 19 and the motor connected to input 1 is utilized to deliver most of the acceleration energy. Driving faster than 35 mph will simply require rotating the input 1 faster. At a vehicle speed of 70 mph, the speed of input 1 will reach $n_1 = 17{,}680$ rpm. Depending on the duty cycle of a vehicle (highway or city driving), the low-speed motor can be turned off like in the example above and a not shown clutch can be applied in order to lock input 2. In this case, the motor connected to input 1 will deliver all the energy required, for example, for a light duty city driving. The two graphs in Figure 4 show that in the energy balance, a friction loss has been considered.

When attempting to constantly back-charge bursts of recuperative energy to a battery, the electrical efficiency becomes very low and the battery's chemical capacity to accept large amounts of energy within only several seconds is limited. A medium size sedan that drives at 56 km/h (35 mph) has about 0.4 kWh kinetic energy. Reducing the speed rather quickly in front of a traffic light that just turned red would require recuperating the 0.4 kWh within about two to three seconds. As a result, it is likely that not more than 0.10 kWh can be back charged to the battery and 0.35 kWh are converted to heat, either in the brake disks or in the electronic vehicle control modules. The double differential including the motor on input 1 can store about 0.24 kWh with an efficiency of about 86%, which means that 0.21 kWh are available in form of a rotation of the double differential when the vehicle comes to a full stop before the red light (0.19 kWh are converted to heat). This energy will be used only several minutes later to accelerate the vehicle after the traffic light turns green. Short term energy storage cannot be done efficiently with today's battery technology. The double differential concept allows a size reduction of the battery by maintaining the same mileage capacity.

The combination of two input speeds is allowing a wide variety of possibilities to adopt the double differential transmission to different driving conditions by achieving an optimal motor and transmission efficiency. The additional aspect of easy energy storage in a fast-rotating differential carrier unit will support the vehicle batteries especially when high energy bursts are required, for example, to accelerate a heavy truck from zero to 48 km/h (30 mph). The fact that both motors must rotate with high speeds while the vehicle is not moving requires very little energy while the external resistance is zero. In contrast, idling internal combustion engines do require considerable amounts of energy, due to the continuous combustions and the internal resistance, mainly from piston ring friction and from the constant acceleration and deceleration of the pistons and the rods.

The double differential with two inputs can also be utilized to collect and transmit the energy from an electric motor and a combustion engine to the driving wheels of a hybrid vehicle. With such an arrangement, optimal speed combinations for each of the two prime movers can be found, which also allows eliminating any additional transmission in the hybrid vehicle.

Gear 10 in Figure 3 is not required for the function of the double differential. It was used to make the transmission symmetric, and it was anticipated that in case of large tooth and transmission housing deformation (under high load), gear 10 would help to keep the torque on gears 11 and 13 equal. If symmetry and balance is not an issue, then gear 10, and in addition gears 13 and 17, can be eliminated in order to simplify the double differential transmission and reduce manufacturing cost.

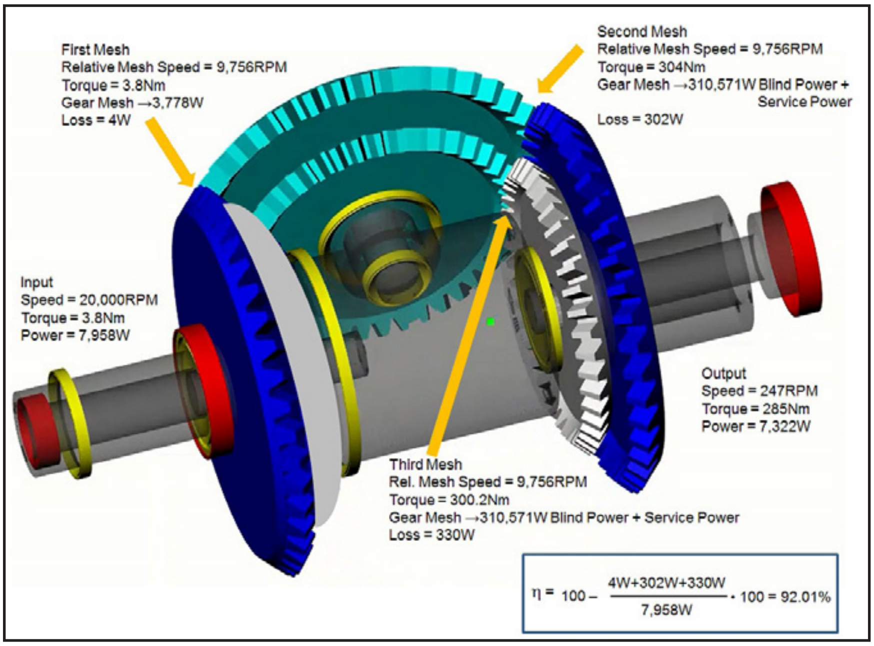

Figure 5 Energy loss per tooth mesh and efficiency calculation.

Efficiency Estimations

An efficiency calculation for 20,000 rpm input speed and 3.8 Nm input torque was conducted. The calculation was performed with the efficiency module of the Gleason Engineering and Manufacturing System (GEMS). This software considers the precise macro and micro geometry of the spiral bevel gears and uses a complex elastohydrodynamic approach to obtain good efficiency predictions (Ref. 7).

The input power of 7.955 kW actuates the outer planets with the first tooth mesh between input gear 14 and planet 15, the second tooth mesh between planet 15 and reaction gear 16, and a third tooth mesh between inner planet 11 and output gear 12. The input parameters for the efficiency calculations as well as the energy loss have been recorded in the graphic in Figure 5. The first mesh has a relative speed of 9,756 rpm and a torque of 3.8 Nm. The energy loss in the first tooth mesh as calculated is only 4 W. Also, the second tooth mesh shows a relative speed of 9,756 rpm but a high torque of 304 Nm. The energy loss in the second tooth mesh was calculated as 302 W. To establish equilibrium between the three tooth meshes, the torque of the third mesh is 304 Nm–3.8 Nm = 300.2 Nm. The efficiency calculation resulted in an energy loss of the third tooth mesh of 330 W.

In order to obtain the transmission efficiency, the sum of the lost power was divided by the input power (times 100) and subtracted from 100%, which results in a gear efficiency of 92%.

A second way of analyzing the tooth mesh losses of mesh two and mesh three appears a very realistic representation of the transmission physics in a double differential. In the discussed designs, the outer and inner planets are connected. Their respective teeth have similar normal forces while the outer planet meshes with reaction gear and the inner planet meshes with the output gear. If the outer and inner planet shown in Figure 5 were replaced with one planet with a double wide face width and if the tooth contact was doubled in length, then the efficiency calculation program shows an energy loss in the double wide mesh of 517 W. The total energy loss is then 521 W, which is equivalent to an efficiency of 100−521 W / 7,958 W · 100 = 93.5%.

The efficiency of a four-stage cylindrical gear transmission, with an approximated efficiency of 98% per stage, results in 0.98 · 0.98 · 0.98 · 0.98 = 0.922 → 92.2%. In conclusion, the efficiency of the double differential is very similar to cylindrical gearing; however, a four-stage helical gear transmission requires four shafts and builds in a larger space than a double differential. The compact design of the double differential may present several physical advantages, leading to an efficiency advantage compared to other high reduction concepts.

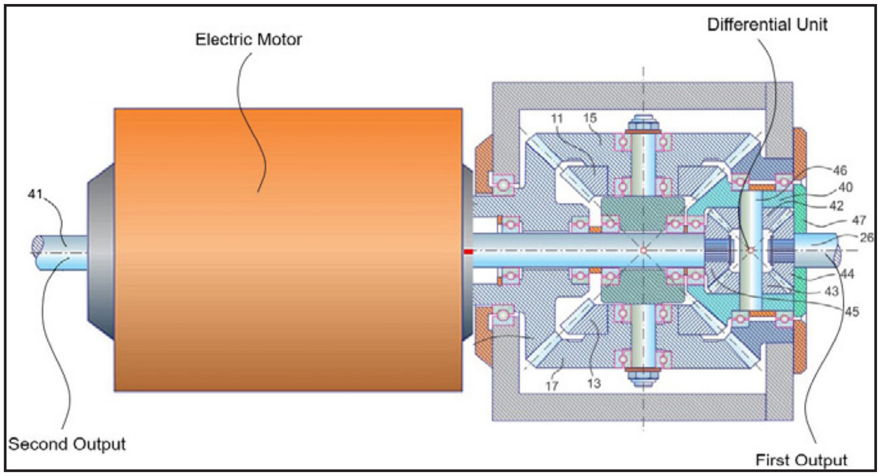

Figure 6 Double differential between the driving wheels—Solution 1..

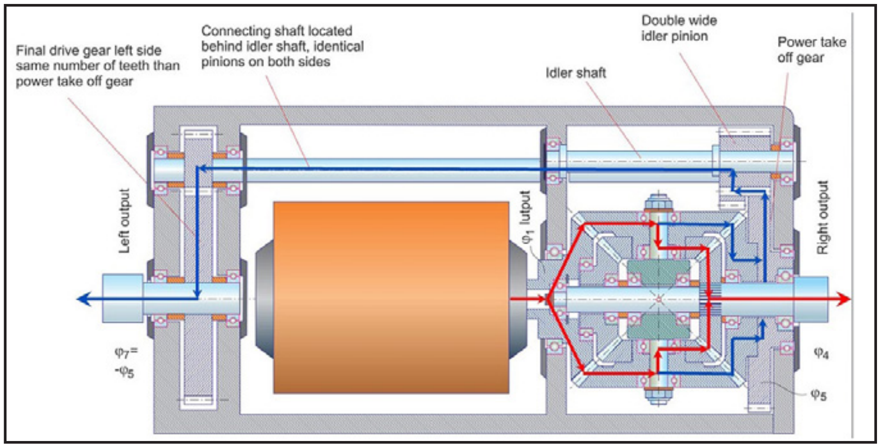

Figure 7 Double differential between the driving wheels—Solution 2.

Double Differential Inline Solution

In order to allow placing the double differential transmission between the wheels of a drive axle in a vehicle, a proposal of an additional configuration is shown in Figure 6.

The transmission in Figure 6 has an additional differential function between the two output shafts 26 and 41. Output shaft 26 remains on the right side of the transmission housing and the added output shaft 41 exits the transmission housing at the left side. Gear 10, which is not required for the correct function of the double differential, has been eliminated, and shaft 41 acts now as main transmission shaft, which was the function of shaft 26 in Figure 2. Gear 12 in Figure 2 was replaced in Figure 6 by gear 40. Gear 40 is hollow inside in order to create a space for the placement of 4 differential gears 42, 43, 44 and 45. Gears 42 and 43 are the planets which are held in position relative to gear 40 with pin 46. Pin 46 is connected to gear 40, which is the gear with the final output speed. Gears 44 and 45 are the side gears. Output shaft 26 is connected to side gear 44 and output shaft 41 is connected to side gear 45. The differential unit in Figure 6 will accomplish the differential function between the two output shafts 7 and 6 with conventional straight bevel gears. The end cap 47 closes the differential which is inside of gear 40 and acts as a radial sleeve bearing of shaft 26 and as a thrust sleeve bearing for gear 44. The walls of the hollow space in gear 40 are utilized as thrust sleeve bearings of gears 42 and 43.

The additional differential function accommodates different wheel speeds while the vehicle is, for example, driving through a curve. A differential, similar as found in conventional axle drives, has been integrated in gear 40. The transmission in Figure 6 has an output shaft 26 which could be connected to the right wheel and an output shaft 41 which could be connected to the left wheel. The input shaft 20 is still located at the left side of the transmission. If input shaft 20 is connected to an electric motor with a hollow shaft, then the transmission in Figure 6 as well as the electric motor can be in-line with the drive axle of a vehicle. This means that output shaft 26 can be connected via a first drive shaft and CV joints to the right-side driving wheel and the second output shaft can be connected via a second drive shaft and CV joints to the left-side driving wheel.

A second possibility of a double differential oriented between the driving wheels, with an integrated differential function, is shown in Figure 7. This proposed design does not require a hollow motor shaft, and the differential function is realized with cylindrical gears. The input speed and torque is transmitted via the planets to the reaction gear, which is connected to the power take off gear (speed ϕ_5). The rotation ϕ_5 is transmitted to an intermediate pinion (idler), which is in mesh with a second cylindrical pinion. The second pinion is connected to a shaft that crosses over to the left side of the motor, where it is connected to a third cylindrical pinion that is in mesh with a second cylindrical gear. Cylindrical pinions two and three require the same number of teeth as do cylindrical gears one and two. This arrangement allows for a smaller size motor (compared to the solution shown in Figure 6).

Double Differential KISSsoft Animations and First Prototype

Although the functionality of the double differential development is explained in great detail, it is difficult to visualize the kinematic of this design. In order to make the high reduction function easy to understand, KISSsoft AG provided several animated designs (Ref. 8).

A screen shot of the one of the animations is shown in Figure 8. The input on the left side rotates the large, dark blue side gear on the left side of the model. The left side gear actuates the light blue outer planets which are rigidly connected to the inner planets. The planets roll on the large dark blue reaction gear on the right side, which is connected to the transmission housing and therefore cannot rotate. This kinematic arrangement puts the carrier in rotation and the inner light blue planets roll on the light gray output gear. Each rotation of the planets causes one rotation of the carrier in the same direction the gear mesh between the outer planets, and the reaction gear advances. This is the reason why the rpm of the planets is only 50% of the input gear rpm. If the tooth count of all the gears was identical, then no output rotation would occur. As the outer planets roll on the fixed reaction gear, the inner planets roll on the output gear, which in this case (like the reaction gear) would also not rotate. If the number of teeth is changed, for example, by adding one tooth to the inner planets, then each full revolution of the planets (and the carrier) will rotate the light gray output gear by one pitch in a direction, reverse to the carrier rotation.

The ratio of the transmission in Figure 8 is +18. This was realized by using 40 teeth on all involved gears, except the two outer planets received 45 teeth. If the two outer planets had 35 teeth, then the ratio would be –14 (refer to equation 10). This example shows how flexible this transmission concept can be used to design a wide range of different ratios without a considerable change of the transmission size.

The first real size prototype of the double differential transmission is shown in Figure 9. This prototype achieves a ratio of 81. All eight gears are ground spiral bevel gears. Only two design calculations were required in order to manufacture the eight spiral bevel gears. It is interesting to mention that due to the similarity of all eight gears, only two different blade geometries (one left hand and one right hand) were required for the soft cutting of all members. The prototype has an electric motor attached, which serves to demonstrate the interesting three-dimensional motion of the planet gears and their high reduction ratio. It was very simple to assemble the transmission unit with the correct backlash and tooth contact. The gears were all ground to low flank form deviations and the shaft shoulders and axial housing dimension have been manufactured to customary tolerances. The assembly did not require any shimming, and the tooth contact resembled the prediction from the design and analysis software. Because of the symmetric design and the fact that always two opposite gears transmit the torque, no bending and warping of the transmission is expected. This fact will prevent large contact position changes under load. The symmetry makes the double differential design insensitive to contact movement and edge contact.

The axial forces, especially of the planets, can be minimized by choosing opposite spiral angle direction for the outer and the inner planets, as shown in the model in Figure 9. In the arrangement in Figure 9, all gear meshes put load on the drive side, if the input gear rotates clockwise. The input gear rotates the outer planets (teeth of the planet on top move to the right), which puts load on the convex flanks of the planets (drive side). The planets try to rotate the reaction gear counterclockwise, which is not possible, but puts the load on the convex flanks of the reaction gear (drive side). If the ratio is positive, then also the output gear rotates clockwise, which puts the load on the convex flanks of the output gear (drive side). This is an interesting phenomenon, which improves efficiency and NVH behavior of

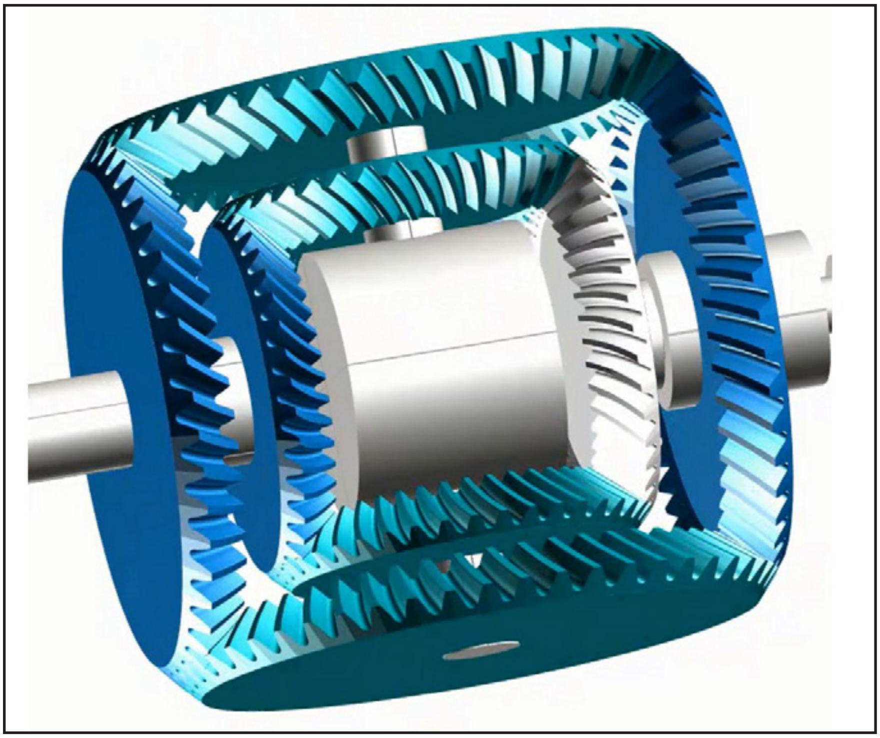

Figure 8 KISSsoft animation—Double differential with ratio 18.

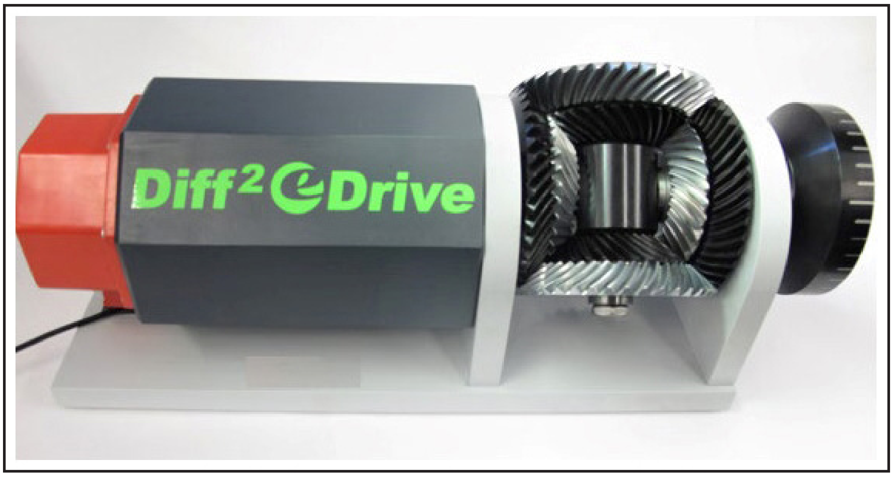

Figure 9 Prototype transmission with motor—Ratio 81.

double differential transmissions.

In order to make the transmission motion of the eight gears better visible, always two opposite pairs have been black oxidized, and the two opposite mating members were chrome plated.

Summary

The fascination of the automotive differential has led to the idea to build a second differential unit around a first center unit. Both units have the same axes around which they rotate with different speeds.

The potential of double differentials as ultrahigh reduction speed reducers is significant. Only the tooth-count of the gears in the outer differential unit must be changed in order to achieve ratios between 5 and 80 without a noticeable change of the transmission size.

Double differentials are well suited for high input speeds. The planets rotate with only about half of the input speed. This fact is attributed to the carrier, which rotates nearly with the same speed as the planets. The relative motion between the outer planets and the input gear, as well as the sliding velocity, are therefore only 50% of the value of two conventionally meshing bevel gears that roll with the same input speed, which is an ideal condition for a transmission with high input speeds.

Ground spiral bevel gears are recommended for the double differential application. Due to the load sharing of the two opposite planets, the torque of each gear is only 50% compared to a conventional bevel gear mesh. This effect results in very high-power density of this already very compact unit.

The efficiency of the double differential is comparable to multiple stage cylindrical gear transmissions comprising the same ratio, in contrast to the fact that always two pairs of gears are transmitting the rotation and torque. The compact size of the double differential can be translated to additional efficiency advantages compared to other transmission concepts, which qualifies this new transmission type very well for the speed reduction and transmission in electric vehicles and hybrids.

Although this paper concentrates on the application of double differentials to electric vehicles and hybrid cars, there are many other applications in the industry which require high ratios. Double differentials could be used in helicopters, wind turbines, agricultural equipment, and many other industrial applications.

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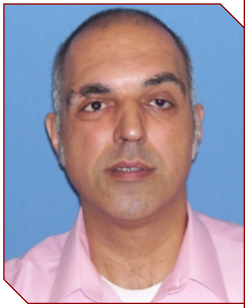

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Design and Simulation of a Back-to-Back Test Rig for Ultra High Cycle Fatigue Testing of Gears Under Fully Reversed Load

Johannes Lövenich, Moritz Trippe, Oscar Malinowski, Jens Brimmers, Stephen Neus, and Christian Brecher

Introduction and Motivation

Geared turbofans provide a significant improvement in aero-engine efficiency by allowing the fan and turbine to rotate at their optimum speeds (Ref. 15). By lowering the fan pressure ratio, larger bypass ratios (BPR) of up to BPR = 12:1 can be achieved (Ref. 28). For example, the PW1000G geared turbofan, displayed in Figure 1, which is installed in the short-haul aircraft Airbus A320neo, reduces fuel consumption and CO_2 emissions by 16% compared to the predecessor model. In addition, noise emissions can be reduced by 75% due to the larger, slower fan (Ref. 21). With the turbine and fan rotating at different speeds, a gearbox is required to transfer a high level of torque within a restricted space to minimize the size of the engine core. Epicyclic gearboxes provide a space-efficient solution leading to a very high-power transfer through the gears.

Figure 1 illustrates the occurring number of load cycles of the gears of two different geared turbofans. Lufthansa currently performs engine overhauls on its A340s after about 20,000 flight hours (Ref. 1). If this is calculated with the speed of a planet wheel in an exemplary planetary gear, the overall number of load cycles is $N_{Plane} > 10^{10}$. Assuming a percentage of 1% with the maximum take-off power (MTOP), the number of load cycles under full load is $N_{mTOP} > 10^8$. As can be seen from the S-N curve shown in Figure 1, the number of load cycles N_{mTOP} exceeds the limiting number of load cycles for tooth root bending strength according to ISO 6336 (Ref. 27) and AGMA (Ref. 25). Pulsator test stands can achieve these high load cycles in shorter test times, but they do not offer the possibility of testing gears under fully reversed bending loads. Therefore, a high-speed back-to-back test rig must be developed to investigate the tooth root bending strength of aircraft gearboxes.

The high dynamics of the test rig require design adaptations compared to a back-to-back test rig according to DIN ISO 14635 (Refs. 8, 26). In order to be able to classify the additional dynamic forces and the influence of the test speed on the load capacity, a functionally identical prototype with a lower performance class is built up. In a multibody simulation, the prototype is dynamically mapped to compare the simulated additional dynamic loads with the dynamic factor K_V from ISO 6336.

In addition to the dynamic behavior of the test rig, the focus is particularly on thermal behavior. The strong temperature differences between the bearing injection temperature T = 40°C in the test gearbox and the lubrication of the test gears at T = 140°C result in a strong temperature gradient in the test gearbox housing. For this reason, the thermal expansions of the test rig, especially at the narrow sealing gaps, are considered in a thermal simulation with Ansys.

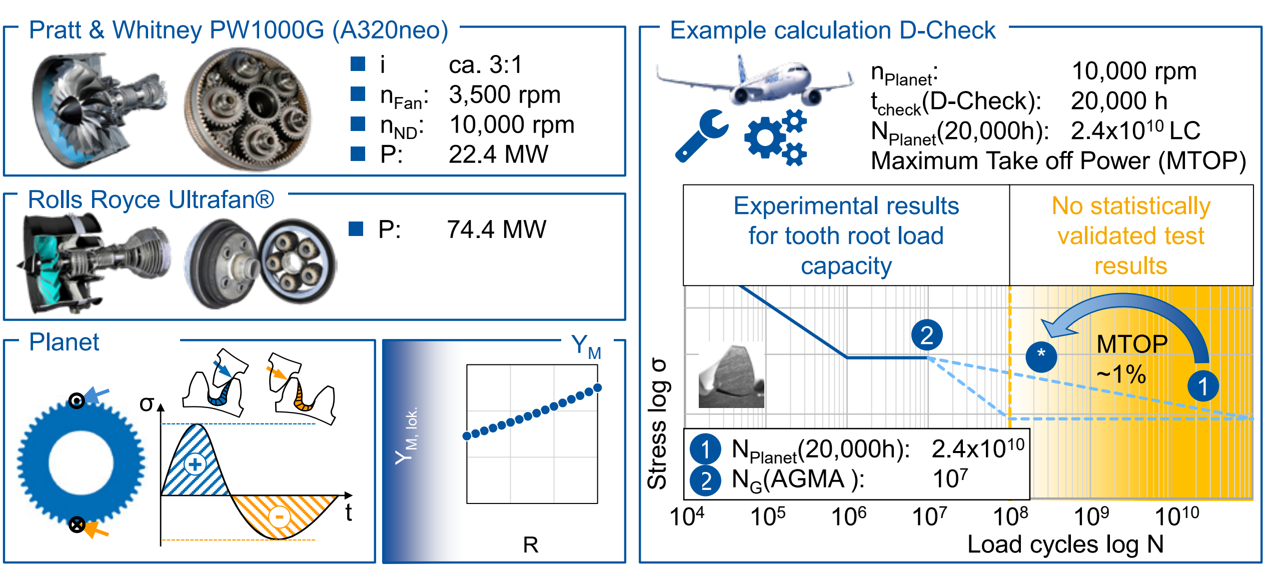


Figure 1 Load cycles of geared turbofans and standards for tooth root load capacity (Refs. 1, 8, 15, 21, 25, 27, 28).

State of the Art

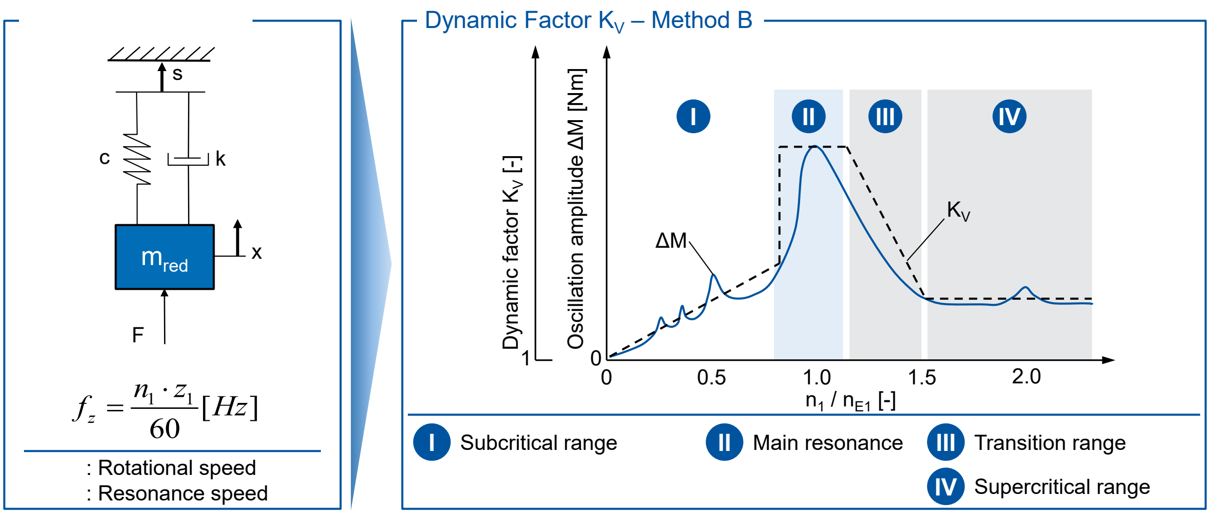
Dynamic Tooth Forces in Gearboxes

The torsionally elastic gear system is excited to vibrations by the varying tooth mesh stiffness, modified tooth flanks and the course of the external load (Refs. 4, 20, 22, 35). During operation, this vibration excitation leads to dynamic tooth forces, which are superimposed, on the load from the static torque. During the transition from single to double tooth contact, there is a jump in the tooth force due to the changed mesh stiffness (Refs. 12, 23). At high loads, the influence of premature tooth engagement resulting from tooth deformation is also superimposed. The load magnification becomes maximum when the tooth meshing frequency coincides with the natural frequencies of the entire system (Refs. 4, 7, 14, 19, 23, 27, 34, 35).

The additional dynamic loads due to the vibration excitation in the tooth mesh has been investigated in numerous works based on calculations and measurements. Bosch developed a model to calculate dynamic tooth forces considering the periodically varying tooth mesh stiffness for variable speeds (Ref. 4). The calculation results were confirmed by measurements of the dynamic tooth forces using strain gauges. Winkler carried out metrological investigations on the dynamic load overload of high-speed gears on a high-speed back-to-back test rig with up to $n_{in} = 15{,}000$ rpm (Ref. 35). Based on the results of the investigations, Winkler developed a calculation model for quantifying the dynamic additional loads of spur and helical gears. Rettig carried out investigations on a back-to-back test rig up to $n_{in} \approx 6{,}000$ rpm on the pinion (Ref. 35). Based on the results of Rettig's and Winkler's investigations, Rettig developed a simplified calculation method for determining the average additional dynamic loads in the sub- and supercritical speed range as well as the main resonance (Ref. 34). The corresponding calculation principles were later transferred into the standard calculation of DIN 3990 and are today the basis for the calculation of the K_V factor of ISO 6336 (Refs. 24, 27).

Gerber investigated the internal additional dynamic loads and the gear damping. Partial eradication of vibration energy is due to damping in the drive train. Bearing friction, flow resistance and damping in the tooth contact play a role. Damping in the tooth contact is determined by the elasto-hydrodynamic conditions and is strongly dependent on the lubricant film properties. For conventional forged steels, the material damping compared to the lubricant-damping is negligible. In this case, mesh geometry, speed ratios and lubricant viscosity are the main influencing variables (Ref. 13).

The described additional dynamic loads can lead to premature failure of the gears and to noise excitation during operation. In method B of ISO 6336, the gear stage is transformed into a



Quelle: [ISO06]

Figure 2 Influence of the rotational speed on the dynamic tooth root stress (ISO06b, BREC19).

single-mass oscillator and the vibration behavior is assessed, see Figure 2. However, this model approach is often not sufficient, since the vibrations excite the entire drive train and in turn influence the vibration amplitude and frequency. The course of the actual torque fluctuation is approximated in the calculation according to ISO 6336 by a linear course of the K_V factor. Drive train-dependent resonance points in the sub- and supercritical range are not taken into account (Ref. 27).

In addition to the standard-based calculation methods, there are numerous works on the calculation of the additional dynamic loads. Baud et al. used an electrical test rig to investigate the dynamic additional loads. The simulation program set up to calculate the additional dynamic loads was successfully validated. The comparison of the simulation and measurement results shows that a detailed model taking into account all degrees of freedom is necessary for the correct calculation of the dynamic tooth root stress (Ref. 3).

Furthermore, for the computational investigation of highly dynamic contact processes in gear drives, an FEM computational model exists that allows the consideration of impact processes but requires higher computation times than a quasi-static tooth contact analysis (Ref. 9). The program system DZP (Dynamic Tooth Forces Program) allows the calculation of the dynamic load distribution of a single stage under consideration of multidimensional rotational and translational degrees of freedom (Refs. 12, 16). In this case, the tooth meshing stiffness is calculated according to analytical approaches. The calculation of the force excitation is carried out for the limiting case of an infinite rotational speed, whereby the differential equations for calculation can be simplified (Refs. 12, 16).

The force coupling element developed by Gacka and Carl allows the mapping of the dynamic excitation in the tooth meshing for a rotational vibration model using the FE-based mesh stiffness of the gear (Refs. 5, 11). Furthermore, modules integrated in the multibody simulation are available to represent the gear using analytical approaches (Refs. 30, 33). Früh combined the analytical contact calculation in six degrees of freedom with the FE-based mesh stiffness to model the gear meshing (Ref. 10). Thereby, displacements of the gears can be converted into changed contact distances and considered in the calculation of the stiffness (Ref. 10).

In summary, it can be stated that with the help of the multibody simulation, the dynamic additional loads in operation can be quantified in detail as a function of the speed. Depending on the selected discretization of the drive train, either the calculation of the maximum dynamic additional load or the course of the dynamic additional load over a gear meshing is possible. The determination of the course of the dynamic additional load for a tooth mesh enables the additional load to be converted into a dynamic tooth root stress. Furthermore, the correct quantification of the dynamic additional loads needs the consideration of all six degrees of freedom.

Design of the Ultra-High Cycle Fatigue (UHCF) Back-to-Back Test Rig

In addition to the UHCF back-to-back test rig, a functionally identical prototype with a lower performance class is built, in order to be able to classify the dynamic additional forces and the influence of the test speed on the load capacity (Ref. 31). In the following, the mechanical structure of the prototype test rig as well as of the main test rig and the thermal conditions are described.

Mechanical Structure of the Prototype

The design of the high-speed back-to-back test rig is comparable with the standardized test rigs from ISO 14635 (Ref. 26). The test rig is divided into a test gearbox 2 and a reference gearbox 3, see Figure 3. The two gearboxes are connected to each other by shafts 7–10 and form a power circuit into which the test torque is applied. The maximum operating speed of the test rig is $n_{in,max} = 12{,}000$ rpm and the maximum operating torque $M_{test,max} = 200$ Nm. The operating parameters allow the efficient investigation of the tooth root load capacity in the UHCF range.

Due to the expected load peaks in the event of tooth root fractures during operation and the high operating speed, the test rig was equipped with journal bearings 1. As a result, the test torque

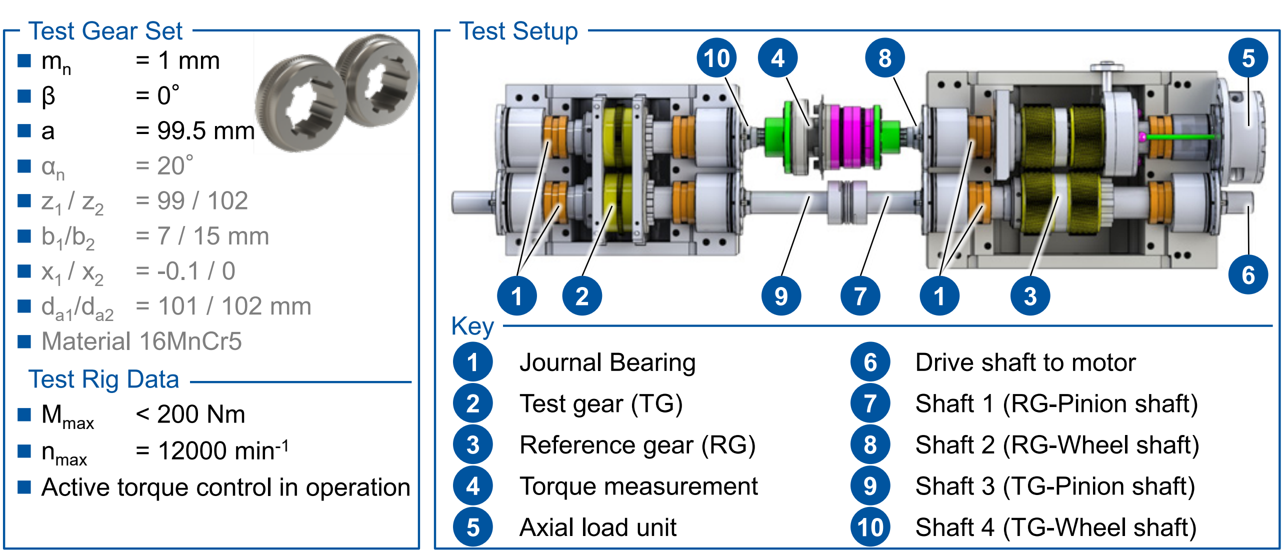


Figure 3 Design of the prototype test rig (Ref. 31).

can only be applied from the limit speed of the journal bearings, at which pure fluid friction is present. The usual load application on standardized back-to-back test rigs by rotating the shafts by means of levers and weights at standstill is therefore not used (Refs. 26, 31). Instead, the test torque is applied and controlled by the axial displacement of a helical gear in the reference gear (Ref. 31). This system allows a compact design with high torsional stiffness and has been successfully used several times for driving high-speed back-to-back test rigs (Refs. 6, 17). Double helical gears are used to compensate for the axial forces in the reference gearbox. The forces to displace the helical gears are applied by an axial load unit 5. The displacement of the gear is made possible by a splined connection with clearance fit. For the non-shiftable gears, a splined connection with press fit is used. A torque measuring flange 4 in the power circuit measures the actual torque for control. In order to achieve complete compensation of thermal axial expansions of the shafts and to avoid constraining forces in the double-helical gears, only the wheel shaft is axially supported in the reference gearbox 8 (Ref. 31).

The test gear used is designed critically with regard to tooth root fracture. The gear data is shown in the left part of Figure 3. In order to operate the test rig with an unchanged infrastructure compared to conventional back-to-back test rigs, the maximum power in the power circuit was limited to $P_{max} = 250$ kW (Ref. 31). Due to the high drive speeds, the maximum test torque is $M_{test,max} = 200$ Nm. It follows that the module and the tooth width of the test gears with $m_n = 1$ mm and $b = 7$ mm are small compared to the standardized gear geometries (Ref. 26). The gears in the reference gearbox are helical with $\beta = 18°$. To keep the number of teeth and the center distance the same, the normal modulus was reduced to $m_{n,VG} = 0.95$ mm.

Mechanical Structure of the Main Test Rig

The mechanical structure of the main test rig differs from the prototype. The main test rig has a three-shaft structure for implementing the reverse bending load, see Figure 4. There are no differences in the mode of operation, for example, the load application by axial shifting of double-helical gears or the use of journal bearings. The upscaling of the test rig allows a power increase to $P > 5$ MW at the same operating speed of $n = 12,000$ rpm by increasing the maximum torque to $M < 4,000$ Nm.

Lubrication and Cooling System (Thermal Conditions)

The main test rig has a cooling and lubrication system to absorb a total power dissipation of $P_{loss,\,total} = 390$ kW. For this purpose, a total volume flow of $Q_{oil,\,total} = 733$ l/min is divided among 3 circuits, see Figure 5.

In accordance with the requirements from aviation gearboxes, the tooth mesh of the test gearbox is operated with an oil injection temperature of $T_{warm,\,in} = 140°C$, warm circuit 1. The journal bearings of the test gearbox, the entire reference gearbox and the transmission gearbox are operated with an oil injection temperature of $T_{cold,\,in} = 40°C$, cold circuit 1 and 2. The cold circuit is divided into two parts. Cold circuit 1 includes the cooling and lubrication of the reference and transmission gears. Cold circuit 2 includes the cooling and lubrication of the journal bearings of the test, reference and transmission gearbox.

Dynamic Simulation of the Test Rig

The dynamic simulation is carried out on the prototype, which has an identical design and mode of operation, but with a lower power class, see “Design of the Ultra-High Cycle Fatigue (UHCF) Back-to-Back Test Rig,” as this will be constructed and set up before completion of the main test rig in order to investigate the influence of the loading speed on the tooth root load capacity. The results are to be used to optimize the main test rig.

Structure and Basics of the Multibody Simulation Model

The transfer of the prototype high-speed back-to-back test rig presented in “Mechanical Structure of the Prototype” to the Simpack multibody simulation requires the simplification of the test setup. Figure 6 shows the test rig topology used. All shafts of the test rig are represented as elastic bodies with their natural frequencies in Simpack. The gears are represented as rigid bodies and the dynamic excitation and vibration behavior via Simpack gear pair.

The gears are coupled to the respective shafts with one degree of freedom in the direction of rotation. In the direction of

Test rig data

- $P > 5$ MW
- $n \geq 12000$ rpm
- $M > 4000$ Nm
- $T_{Oil,\,inlet}$: 90°C – 140°C
- $Q_{oil} > 750$ l/min
- Center distance: 200 mm ≤ a ≤ 250 mm
- Gear type: spur gears

Test conditions

- Investigations up to $N = 5 \cdot 10^8$
- Active torque regulation

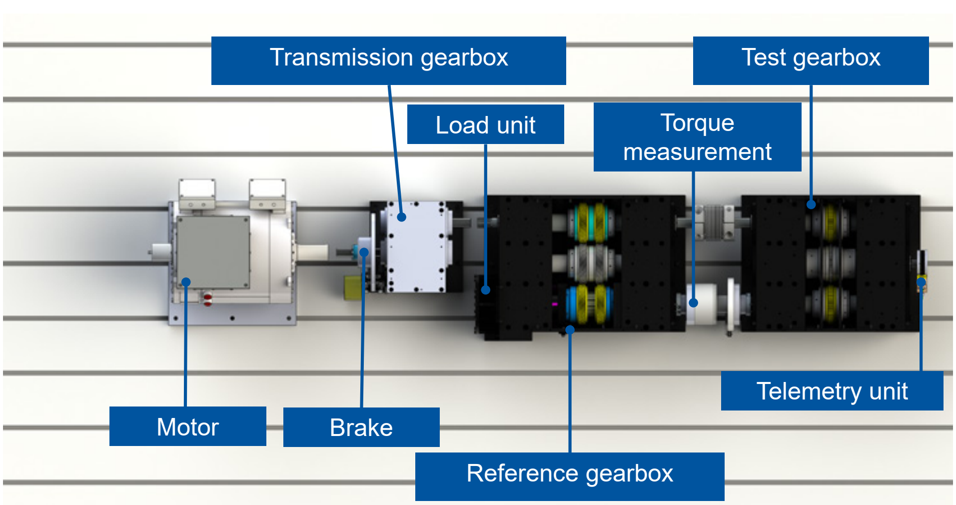
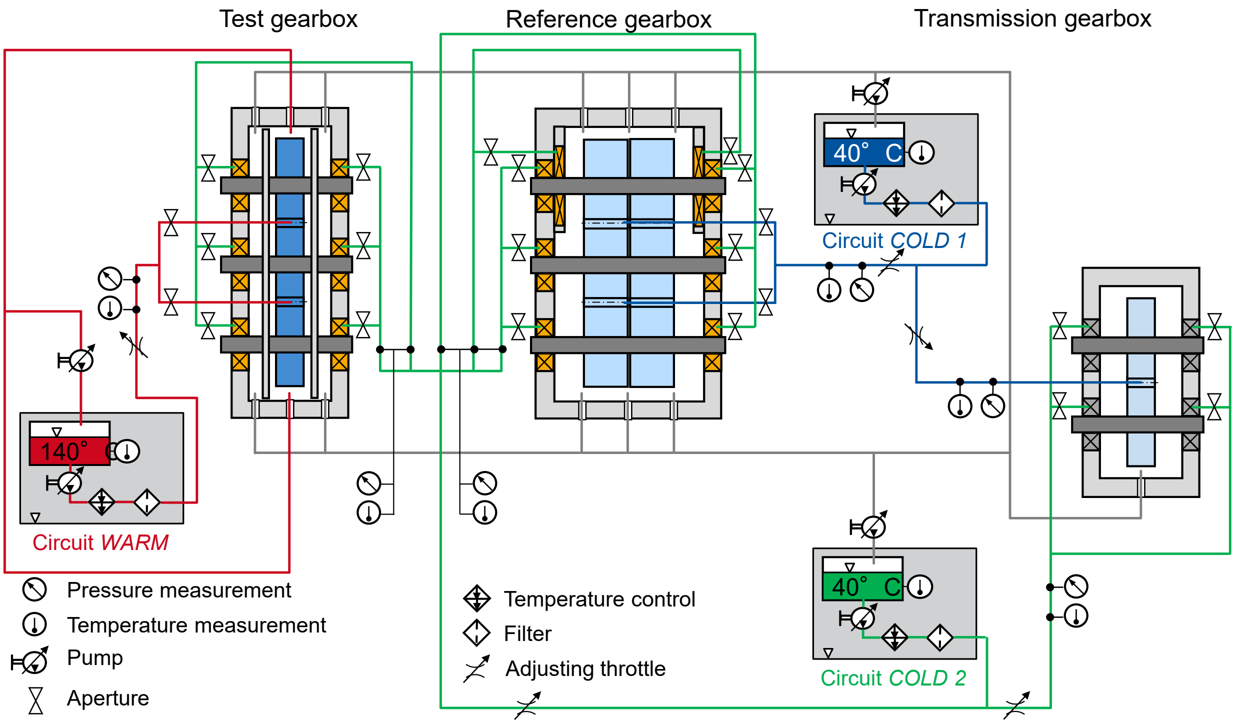


Figure 4 **Design and power data of the main test rig for UHCF investigations under fully reversed bending.**



	Circuit WARM	Circuit COLD 1	Circuit COLD 2	∑
$T_{oil, in}$	140°C	40°C	40°C	-
$T_{oil, out}$	160°C	61°C	60°C	-
P_{loss}	~80 kW	~225 kW	~85 kW	~390 kW
Q_{oil}	~190 l/min	~340 l/min	~200 l/min	~730 l/min

Figure 5 Cooling and lubrication system of the main test rig.

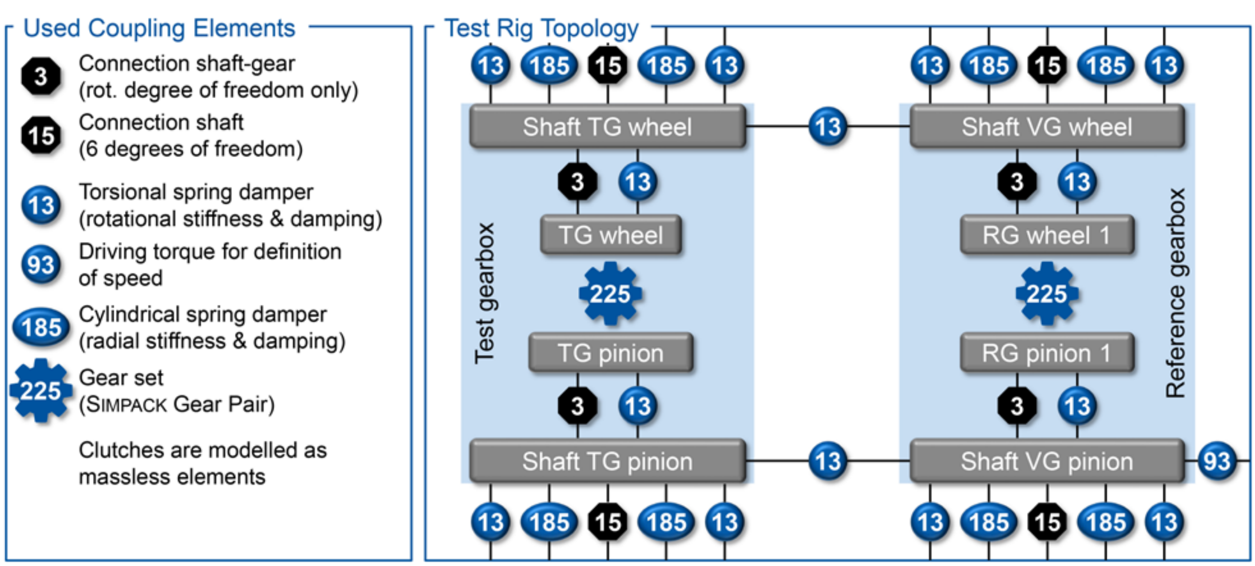


Figure 6 Topology of the prototype in the multibody simulation.

rotation, a torsion-spring-damper element is used to represent the splined connection between the gear and the shaft. The shafts are mapped with six degrees of freedom. The journal bearings of the test rig are represented in a simplified way by cylindrical spring-damper elements with radial stiffness and damping. The damping of the journal bearings about the axis of rotation is considered via a torsion spring-damper without stiffness.

In deviation from the test rig setup, only one-half of the double-helical gears is modeled in the reference gearbox. This is to avoid interference effects, for example, due to uneven load distribution or axial vibrations. This means that both shafts in the reference gearbox must be axially fixed. For this purpose, the degree of freedom of the corresponding shafts is constrained in the axial direction. The permissibility of this assumption must be verified during commissioning. The axial fixing of the shafts in the test gearbox is carried out in the same way as in the test rig via the connecting clutches on the shafts and the reference gearbox shafts. The clutches connecting the shafts of the test and reference gearboxes are modeled as massless torsion-spring-dampers.

Since load application is also possible at standstill for the simulation, the application of the bracing torque is performed analogously to standardized back-to-back test rigs by rotating a test and reference gearbox shaft relative to each other. This procedure achieves a more stable calculation in comparison to the load application in reality by shifting helical gears. The iterative adjustment of the total torsion of the shafts leads to the desired torque on the test gears. The speed is controlled analogously to the test rig by applying a torque outside the power circuit. In the simulation, the drive torque is adjusted to produce a uniform speed increase over time. The vibration behavior is analyzed in the speed ramp-up from standstill to the operating speed of $n_{in} = 12{,}000$ rpm.

The parameterization of the coupling points used determines the calculated vibration behavior. For this reason, the stiffness and damping values are taken, as far as possible, from the datasheets of the components installed in the test rig or from the literature. However, there are uncertainties, particularly with regard to the damping values used and the stiffness of the spline shaft connections and journal bearings, which have to be assessed during commissioning.

Table 1 shows the stiffness and damping values used to simulate the initial model. Where possible, the values were taken from the datasheets of the test rig components. These include the torsional stiffness of the couplings and the radial stiffness of the journal bearings.

The torsional damping of the journal bearings was calculated approximately from the moment of loss at the operating point. The torsional damping of the couplings cannot be determined without measurements of the natural frequencies and vibration behavior. As an approximation, values below the damping of the splined shaft are therefore assumed since there are fewer separation joints. Since the number of parting lines in the coupling package is doubled on the wheel side by integrating the torque measurement flange, the assumed torsional damping is also doubled. The meshing stiffness of the gears is calculated within the Simpack gear pair using the procedure of DIN 3990 (Ref. 24). The gear damping was determined according to the procedure of Gerber (Ref. 13). The torsional stiffness and damping of the spline connection was estimated using the values of Walten et al. and Barrot et al. (Refs. 2, 32). Since the system behavior is to be considered outside the resonance points in the supercritical range, only a small influence of the damping values is to be expected.

System Behavior in Relation to the Operating Speed

The amount of dynamic torque fluctuation depends on the speed of the test rig. Since different gear geometries are used in the test and reference gearboxes due to the load application concept, the dynamic torque also differs in the test and reference gearbox. The gears in the test gearbox have straight teeth and, due to the head shortening of the wheel, a total or profile contact ratio of $\varepsilon_\alpha = \varepsilon_\gamma = 1.26$. This results in a significant vibration excitation, which leads to high torque amplitudes in operation. Due to the helix angle of $\beta = 18°$ and the lack of head shortening, the gears in the reference gearbox have a total contact ratio of $\varepsilon_\gamma = 5.59$. It follows that the dynamic torque amplitudes in the reference gear are comparably small. To calculate the dynamic torque, in the first step, the system is statically clamped to a torque of $M_{R,\,pinion} = 180$ Nm at the pinion by rotating the shafts relative to each other. In the second step, a drive torque of $M_{an} = 9$ Nm is applied. An S-function is used for continuous imprinting, which prevents discontinuities in the simulation. The drive torque leads to a continuous speed increase of the test rig. The sampling rate in the simulation is $f_{sample} = 40$ kHz, so that the tooth meshing can be reproduced with sufficient accuracy, even at high speeds.

Table 1 Reference values for stiffness and damping of the bearing and coupling points.

	Stiffness/Damping type	Value
Journal bearings	Stiffness radial	10^8 N/m
	Damping radial	410^4 Ns/m
	Torsional damping	0.0007 Nms/rad
Spline-hub connection	Torsional stiffness	10^6 Nm/rad
	Torsional damping	10^3 Nms/rad
Coupling (wheel)	Torsional stiffness	290 10^3 Nm/rad
	Torsional damping	600 Nms/rad
Coupling (pinion)	Torsional stiffness	138 10^3 Nm/rad
	Torsional damping	300 Nms/rad
Gears	Gear meshing stiffness	acc. to DIN 3990
	Gear meshing damping	10^3 Ns/m

The left part of Figure 7 shows the gear data and the modifications. To reduce premature tooth meshing and to optimize load distribution on the tooth flank, profile and width crowning were applied. The dynamic torque curves in the test and reference gear for a speed ramp-up of $n_{in} = 0–12{,}000$ rpm are shown in the right part of Figure 7.

The maximum amplitude of the dynamic torque in the test gear unit with $\Delta M_{PG} \approx 150$ Nm is significantly higher than the maximum amplitude of $\Delta M_{VG} < 50$ Nm in the reference gearbox. The asymmetry of the torque curve in the test gear is due to the fact that a deflection to the maximum dynamic torque occurs only once in a tooth grip. Low dynamic torques account for the greater part of the values. In the dynamic torque curves of the test and reference gearbox, resonance points of the system are characterized by local torque maxima. The first three (TG) or four (RG) significant resonances are marked in the diagrams. The system is primarily excited from the rotational gear excitation in the test and reference gearbox. Accordingly, a large influence of the torsional natural frequencies on the system behavior is to be expected.

Table 2 shows the first three torsional natural frequencies of the shafts in the test and reference gearbox. If the speed-dependent tooth mesh frequency f_Z or multiples thereof coincide with the system natural frequencies, a local torque maximum occurs. The torque fluctuation up to the first resonance point in the test gear is due to the excitation of the bending natural frequencies of the pinion and wheel shafts by the 3rd and 4th f_Z. At the second resonance point in the test gear, the first torsional natural frequency of the pinion shaft $\omega_{T,1,TG\ Pinion}$ and the 2nd f_Z coincide. The third resonance is caused by the excitation of the first torsional natural frequency of the wheel shaft $\omega_{T,1,TG\ Wheel}$ with the 2nd f_Z. After the third resonance, the maximum dynamic torque fluctuation decreases to an increased level of $\Delta M_{PG} \approx 50$ Nm from $n_{in} = 3{,}000$ rpm and does not exhibit any further significant maxima.

Due to the overall smaller fluctuation of the dynamic torque, the resonances in the reference gearbox are more noticeable than in the test gearbox. The first resonance is caused by the encounter of the first torsional natural frequency of the pinion shaft $\omega_{T,1,RG\ pinion}$ with the 2nd f_Z. At the second resonance, the 2nd and 3rd f_Z excite the second torsional natural frequencies of pinion and wheel shaft $\omega_{T,2,RG\ pinion}$ and $\omega_{T,2,RG\ wheel}$. At the third resonance, the first torsional natural frequency of the pinion shaft $\omega_{T,1,TG}$ pinion meets the 1st f_Z. Additionally, the third torsional natural frequency of the pinion shaft $\omega_{T,3,RG\ Ritzel}$ meets the 2nd f_Z. The fourth and last significant resonance in the reference gear is due to the excitation of the second torsional natural frequency of the pinion shaft $\omega_{T,2,RG\ pinion}$ by the 1st f_Z. From $n_{in} = 4{,}000$ rpm, the maximum dynamic torque amplitudes drop to a low level without further significant maxima. The comparison of the dynamic torque curves in the test and reference gearbox shows that, due to the long gear shafts, the gearboxes are not influenced by the dynamic loads or resonances in the other gearbox. Therefore, it would be permissible to focus solely on one gearbox for further analysis.

Definition of Operating Points of the Test Rig

In the first step, the additional dynamic load calculated in Simpack in the test and reference gearboxes is compared with the K_V factor according to ISO 6336 (Ref. 27). In the second step, the operating points of the test rig and possible strategies for the run-up to the operating speed are discussed.

Comparison of Multibody-Simulation and Standardized Calculation Approach

In the load capacity calculation of gears, the dynamic additional load in operation are usually taken into account by means of the dynamic factor K_V of ISO 6336 presented in the state of the art (Ref. 27). Figure 8 compares the dynamic factors according to method B of ISO 6336 and the simulation in Simpack. For the comparison of the additional dynamic loads with the K_V factor calculation from the ISO 6336, the envelopes of the maximum dynamic torque are related to the mean torque to generate a speed-dependent K_V factor from the Simpack simulation, see Formula 1.

$$K_V(n_{An}) = \frac{M_{dyn,max}(n_{An})}{M_{dyn,MW}} \quad (1)$$

The upper diagram shows the additional dynamic loads in the test gear. According to ISO 6336, the resonance speed of the test gear is $n_{E,TG} = 2{,}953.48$ rpm. In the subcritical range

Table 2 Reference values for stiffness and damping of the bearing and coupling points.

	Shaft TG pinion	Shaft TG wheel	Shaft RG pinion	Shaft RG wheel
1s torsional natural frequency $\omega_{T,1}$	4022.7 Hz	5926.7 Hz	4887.5 Hz	3563.9 Hz
2nd torsional natural frequency $\omega_{T,2}$	5657.2 Hz	9482 Hz	7763.9 Hz	5092 Hz
3rd torsional natural frequency $\omega_{T,3}$	7776.4 Hz -	12831 Hz	9646.9 Hz	6895 Hz

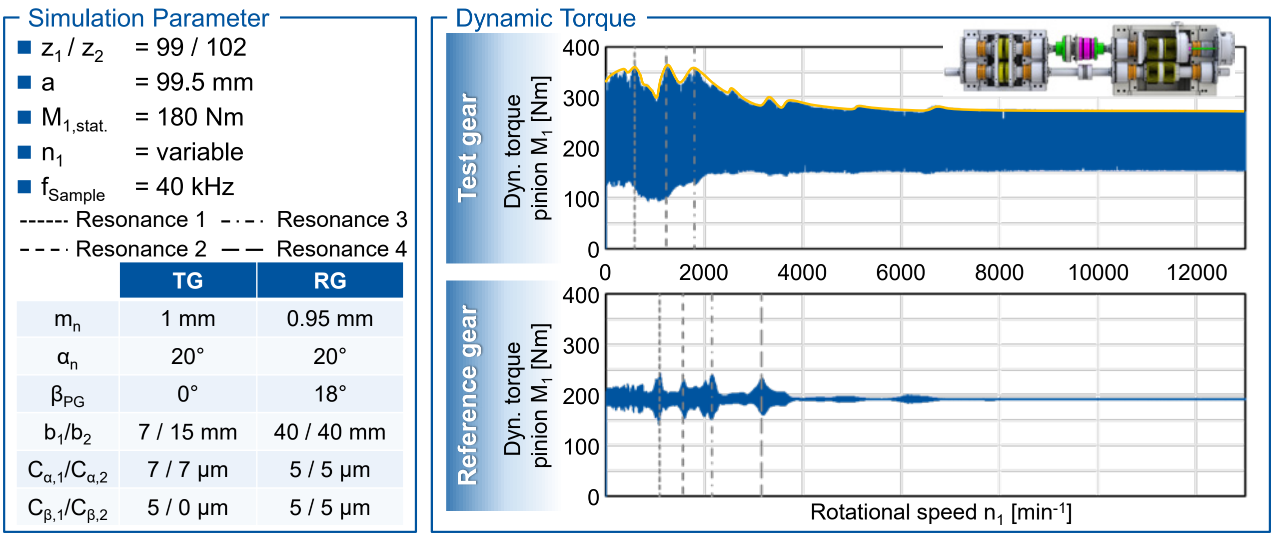


Figure 7 Analysis of the operational behavior over the rotational speed.

before the main resonance, the K_V factor according to ISO 6336 increases linearly up to a value of $K_{V,ISO} = 1.25$ at $n_1 = 2{,}500$ rpm. The dynamic factor according to Simpack is in the same range above $K_{V,Simpack} = 1.5$. In the transition range to the main resonance as well as in the main resonance, the dynamic factor from ISO 6336 increases to values of $K_{V,ISO} = 1.8–1.9$. In contrast, Simpack calculates a lower dynamic factor of $K_{V,Simpack} \approx 1.3$. In the supercritical speed range from $n_1 = 4{,}500$ rpm, the dynamic factor from ISO 6336 drops to a value of $K_{V,ISO} = 0.87$ while the dynamic factor from Simpack remains at a level of $K_{V,Simpack} \approx 1.2$. The dynamic additional load in the subcritical and supercritical operating region is underestimated by the simplified standard calculation according to method B, whereas the additional load in the main resonance is overestimated. This is because the additional loads in the test gear are determined by the excitation of the shaft natural frequencies in the subcritical speed range and not by the gear natural frequency. Furthermore, it is to be noted that the test gear represents a limiting case for the standard calculation due to the low total contact ratio of $\varepsilon_{\gamma,PG} = 1.26$. In the supercritical region, there is a reduction of the maximum load in the single engagement area critical for the standard calculation. Due to the low profile overlap, however, higher additional loads are to be expected in real operation. In the calculation of the dynamic factor in Simpack, only the maximum occurring torque is taken into account and not the time of occurrence related to the tooth mesh.

Figure 8 shows the comparison of the dynamic factors from the standard calculation and Simpack for the reference gear unit. Since the reference gear is a helical gear with $\beta = 18°$, the total contact ratio of the reference gear with $\varepsilon_{\gamma,VG} = 5.59$ is considerably larger. The large total contact ratio results in a low vibration excitation by the gears and correspondingly lower additional dynamic loads than in the test gear. The resonance speed of the reference gears according to ISO 6336 is $n_{E,VG} = 3{,}069.1$ rpm and agrees with the value from Simpack. In the subcritical speed range, the dynamic factors from ISO 6336 and Simpack are approximately the same. In the main resonance, the dynamic additional load is also reproduced sufficiently accurately by ISO 6336, compared with the values from Simpack. In the supercritical speed range, ISO 6336 overestimates the dynamic additional load with a constant value of $K_{V,ISO} = 1.32$ compared to Simpack ($K_{V,Simpack} \approx 1.008$). This is because the K_V factor in the standard calculation cannot fall below one for a total overlap of $\varepsilon_\gamma = 2.5$.

The comparison of the dynamic factors from ISO 6336 and Simpack shows that the standard calculation according to method B is not suitable for correctly representing the additional dynamic load in the operating range up to $n_1 = 12{,}000$ rpm. This can be explained on the one hand by the special geometry of the test gears and on the other hand by the influence of the shaft natural frequencies. For this reason, the dynamic factor calculated with Simpack is used to define the operating points of the test rig. For the final evaluation of the results from Simpack, it is necessary to convert the additional dynamic loads into the dynamic tooth root stress.

Definition of the Operating Points for the Investigation of the Tooth Root Load Capacity

The desired operating speed of the test rig is $n_{test} = 12{,}000$ rpm. The analysis of the vibration behavior has shown that no resonances occur above a speed of $n_1 = 4{,}000$ rpm. Accordingly, operation at the desired operating speed is feasible. The additional dynamic load at this speed must be taken into account when calculating the tooth root stress. To compare the tooth root stresses of different test rigs, it is necessary to calculate the additional dynamic loads using a simulation model and to transfer the additional dynamic loads to the dynamic tooth root stress. The simulation results also show that the definition of discrete torque levels is necessary for the speed ramp-up of the test rig up to the operating point, since otherwise pre-damage of the gears is likely, due to the high additional loads in the subcritical speed range, see Figure 9.

At standstill, the minimum torque $M_{min} = 10$ Nm is applied. This minimum tension serves to take out the backlash of the tooth flanks during speed ramp-up to the minimum bearing speed. In the second step, the speed is increased in a controlled manner to the transition speed of the journal bearings and kept constant at a speed level of $n = 750$ rpm. In the third step, the

Input Data

- z_1 / z_2 = 99 / 102
- $M_{1,\,stat}$ = 180 Nm

	Unit	TG	RG
m_n	[mm]	1	0.95
α_n	[°]	20	20
β_{PG}	[°]	0	18
b_1/b_2	[mm]	7 / 15	40 / 40
$c_{\gamma\alpha}$	[N/(mm·µm)]	15.7	17.1
ε_γ	[-]	1.26	5.59

Key

- ——— K_V Factor acc. to Simpack
- - - - - K_V Factor acc. to ISO 6336
- — · — Resonance speed n_E
- — · · — Operating speed

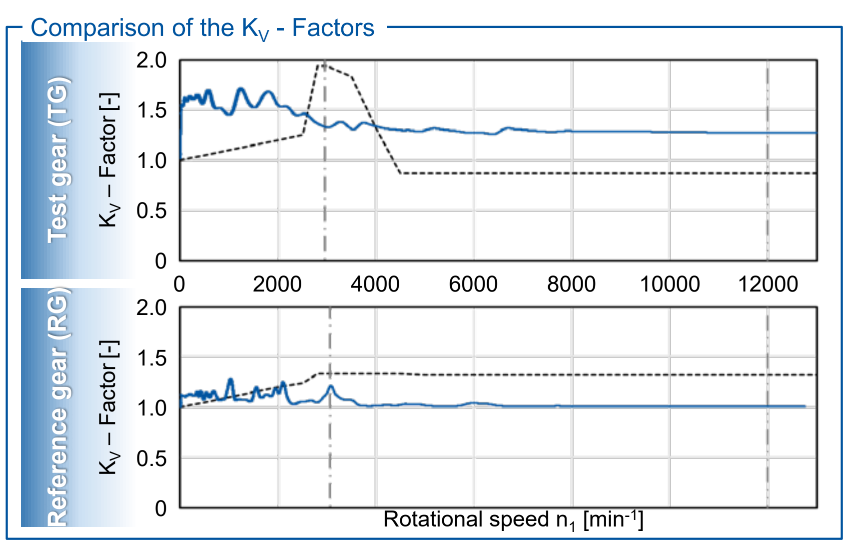


Figure 8 Comparison of the dynamic factors from Simpack and ISO 6336 (Ref. 27).

torque is increased to half the test torque $M = 0.5\ M_{test}$. In step 4, the speed is increased to the maximum operating speed of n = 12,000 rpm.

During the speed ramp-up from $n_{bearing,min} = 750$ rpm to $n_{test} = 12{,}000$ rpm, the main test rig resonances are run through, compare Figure 7 and Figure 8. Since the torque for ramp-up is significantly smaller than the test torque, no influence of the dynamic loads on the load capacity is to be expected. On the other hand, the increased torque compared to the first step ensures permanent contact between the tooth flanks. The last step is increasing the torque to the test torque.

Conclusion of the Dynamic Simulation

The main purpose of the prototype test rig is to investigate the influence of the speed on the tooth root load capacity. The results of the dynamic simulation in Simpack show clear differences to the estimation of the dynamic loads according to ISO 6336. After the test rig has been set up, the main resonances are investigated experimentally by means of impact tests. Additionally, during commissioning the main resonances are investigated, by vibration measurements directly on the rotating pinion shaft, and the dynamic mapping of the back-to-back test rig is validated in the multibody simulation. By validating the additional dynamic loads calculated in Simpack on the prototype, a separate multibody simulation model will be created for the main test rig so that, if necessary, design measures can be taken to change the natural frequencies of individual components before the main test rig is set up.

Thermal Simulation of the Test Rig

In addition to the dynamic behavior of the test rig, the focus is particularly on the thermal behavior. The temperature difference between the bearing injection temperature (T = 40°C) in the test gearbox and the gear injection temperature (T = 140°C) leads to a significant temperature gradient in the test gearbox housing. With regard to thermal expansion, the guarantee of function in the heated state must be ensured. In particular, the radial gaps at the seals must be considered. Another important point is the axial play of the mounted shafts, which is strongly dependent on the axial thermal expansion. For this purpose, the

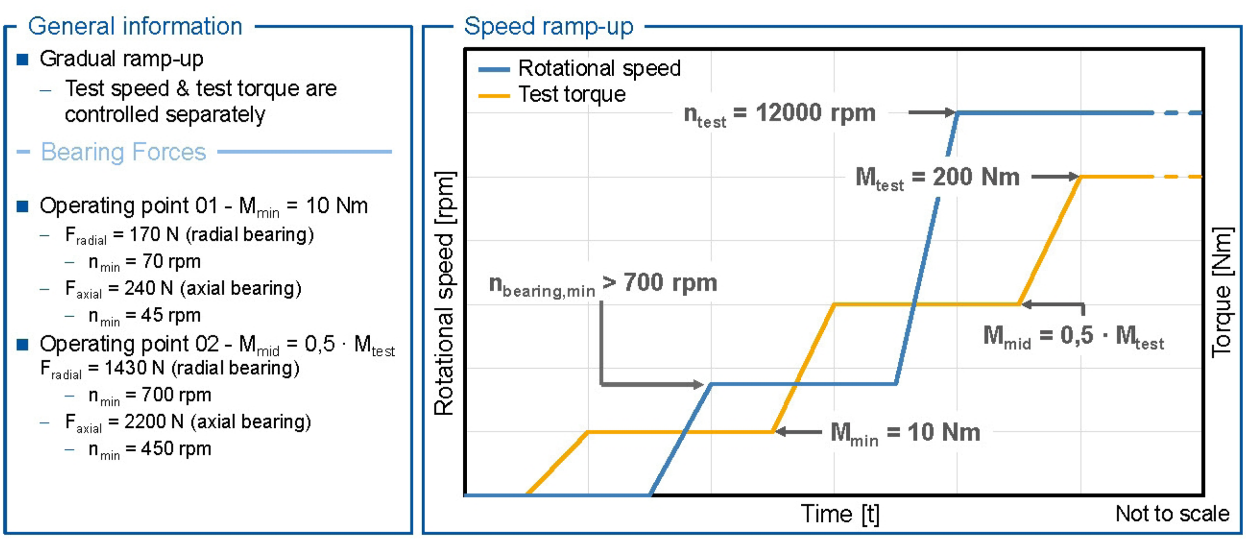


Figure 9 Speed and torque ramp-up for the prototype test rig.

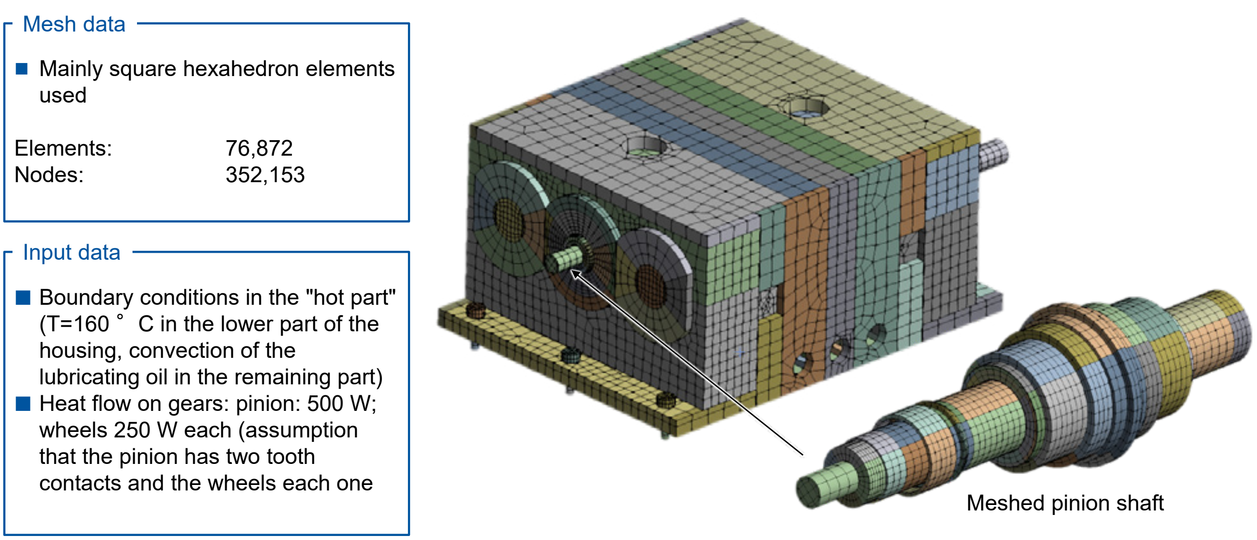


Figure 10 Meshed Ansys model of the test gearbox.

thermal behavior of the two gearboxes is simulated individually in Ansys. In this report, however, the focus is on the test gearbox, as this is where the highest temperature differences occur.

Structure and Basics of the Thermal Simulation Model

To simplify the model in Ansys, all chamfers, screws and small radii are removed to prevent refinement of the mesh and thus greater computational effort on these. Furthermore, the spline profiles of the gear shafts and gears are removed. Holes and small gaps, for example gaps to avoid double fits, are plugged to optimize the meshing. In addition, individual bodies are grouped together to optimize meshing and reduce contact conditions, which in turn reduces computation time. The admissibility of the assumptions made must be verified by means of measurements in subsequent operation, as these do not necessarily take place on the safe side. The meshed model of the test gearbox and the meshing parameters are shown in Figure 10. Mainly square hexahedron elements are used. The test gearbox exists of overall 76,872 elements with 352,153 nodes. As input data, the gear losses according to Niemann/Winter are assumed to be P = 500 W for the pinion and P = 250 W each for the wheels, which means that both tooth meshes have an individual loss of P = 500 W, which is split 50/50 between wheel and pinion (Ref. 23).

Boundary Conditions

The heat sources are the journal bearings and tooth meshes due to friction, compare power losses listed in "Lubrication and Cooling System (Thermal Conditions)." Heat dissipation takes place on the one hand via the dissipation of the oil volume flow and on the other hand via convection and radiation of the housing and clamping field. In addition, the heat transfer within the housing must be considered.

Convection of the Gearbox Housing to the Surrounding Air

The convection from the housing to the environment is approximated according to Niemann/Winter, compare Formula 2 (Ref. 23). For this purpose, the surrounding air is assumed to be at rest.

$$\alpha_a = f_k \times \left(10 + 0.07 \times (T_G - T_u)\right) \times \left(\frac{1}{h}\right)^{0.15} \quad W/(m^2K) \tag{2}$$

α_a	Heat transfer coefficient	W/m^2K	T_u	Temperature of surrounding medium	K
f_k			T_G	Temperature of the component	K
h					

The test gearbox housing is divided into a cold area (blue) with an estimated temperature delta of ΔT ~40 K and a warm area (red) with ΔT ~100 K, compare Figure 11. The estimation was made on the basis of the injection temperatures. The ambient temperature is assumed to be T = 20°C. Accordingly, the heat transfer coefficients for convection from housing to air result in $\alpha_{TG,\ warm} = 19.5\ W/m^2K$ for the warm part and $\alpha_{TG,\ cold} = 15.09\ W/m^2K$ for the cold part of the housing.

Convection of the Clamping Field to the Surrounding Air

The clamping field can be displayed as plane surface and therefore the relationship for natural convection of the plane surface according to Recknagel can be used (Ref. 29). With a temperature difference of ΔT = 100°K and a dimension of A = 1.6x4m (undisturbed area in front of the test gearbox housing), the heat transfer coefficient for convection of the clamping field to surrounding air is $\alpha_{CF,TG} = 6.3\ W/m^2K$.

Heat Radiation from Housing and Clamping Field to the Surrounding Air

According to Grossmann/Jungnickel, the emission coefficients $\varepsilon_{housing} = 0.6$ (for a mat steel) and $\varepsilon_{clamping\ field} = 0.94$ (for an unalloyed rough steel) can be assumed for the heat emission of the housing and clamping field via radiation to the surrounding air (Ref. 18).

Heat Transfers within the Test Gear Unit (Oil Convection and Radiation) from the Components to the Housing

In the warm area, the emission coefficient for a thick layer of oil on polished steel is given as $\varepsilon_{oill,\ thick} = 0.82$. Radiation at

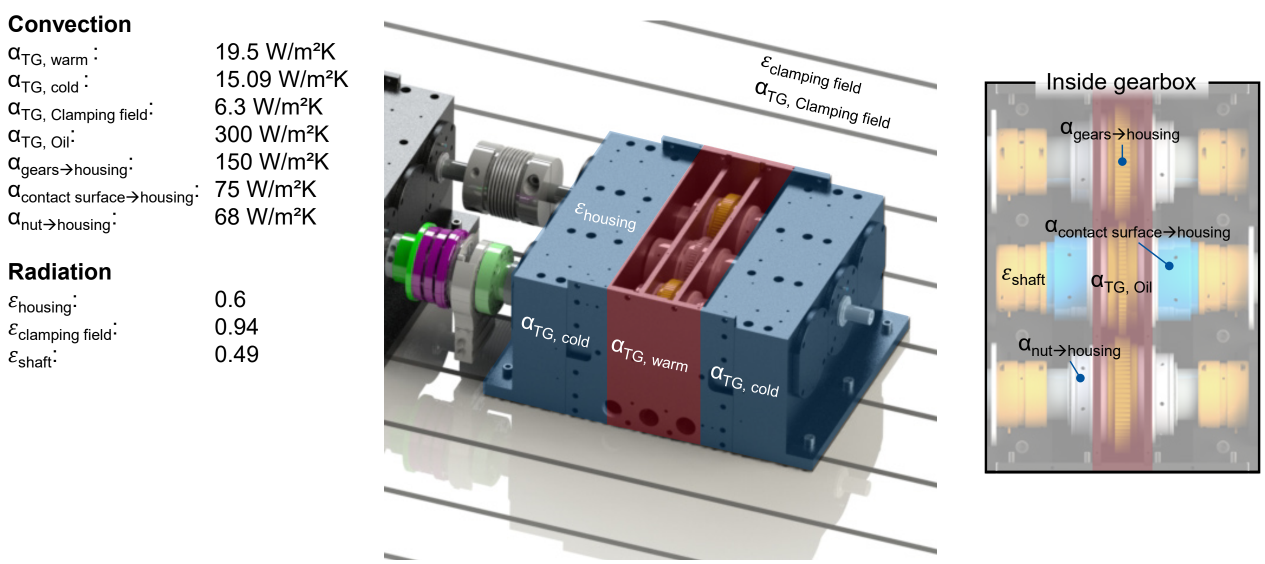


Figure 11 Overview of the boundary conditions of the test gearbox.

the bottom is neglected (assumption: lake of oil so chaotic that radiation is difficult to determine and also negligible). The housing there has the temperature of the oil (T = 160°C), according to Niemann/Winter with an oil injection temperature of $T_{oil,\,in} = 140°C$ and an $\Delta T = 20°C$ (Ref. 23). Also according to Niemann/Wint, a heat transfer coefficient of $\alpha_{oil,\,TG} = 300\,W/m^2K$ is assumed for the convection of the lubricant flow (Ref. 23). The convection at the gears is neglected since it is already included in the energy balance of the gears. The convection at the rotating shaft is calculated according to Grossmann/Jungnickel (Ref. 18). The shafts are represented by rotating smooth cylinders for heat transfer at the inner walls of the housing. According to Recknagel, the heat transfer coefficient from the individual components, like gears, shafts or nuts is calculated with Formula 3 (Ref. 29). The results are shown in Figure 11 on the left side.

$$\frac{1}{\alpha_{shaft,housing}} = \frac{1}{\alpha_{shaft,air}} + \frac{1}{\alpha_{air,housing}} \quad (3)$$

The heat transfer coefficient from the gears and chamber separations to the housing is $\alpha_{gears \rightarrow housing} = 150\,W/m^2K$, between contact surfaces and housing its $\alpha_{contact \rightarrow housing} = 75\,W/m^2K$ and between the nut and the housing its $\alpha_{nut \rightarrow housing} = 68\,W/m^2K$.

Contactless Seals

At the walls of the sealing, convection is assumed to be due to the oil volume flow with $\alpha_{oil,\,TG} = 300\,W/m^2K$ based on the same assumptions as in the "warm section". At the inner part of the sealing, convection is assumed to occur at the rotating cylinder, since here the shaft is presumably not completely or only slightly wetted by oil, and partial oil splashes back from the walls of the sealing. The cooling by inflowing air is neglected in the model. In the remaining areas, convection between shafts and air and convection in the narrow gap are assumed according to Jungnickel 2010 (Ref.18).

Journal Bearings

The heat transfer of the journal bearings into the housing is calculated according to Formula 4.

$$\alpha_{bearing} = \alpha_{housing} \times \frac{A_{housing}}{A_{bearing}} = 1{,}240.41\,W/m^2K \quad (4)$$

$\alpha_{bearing}$	Heat transfer coefficient bearings	[W/m²K]
$\alpha_{housing}$	Heat transfer coefficient housing	[W/m²K]
$A_{bearing}$	Bearing surface	[m²]
$A_{housing}$	Housing surface	[m²]

Simulated Temperature Distribution on the Test Gearbox

Under the assumptions and boundary conditions listed, the temperature distribution, heat flows and thermal expansion of the test gearbox were simulated with Ansys. Figure 12 shows the results of the thermal simulation. The test gearbox heats up to a maximum temperature of $T_{max} = 178.59°C$. The power loss, which is dissipated from the test gearbox to the environment accounts to P = 4.2 kW. The main part of the heat is dissipated via the oil flow (see "Lubrication and Cooling System (Thermal Conditions)"), which is defined as an input variable in the simulation. Via the tooth mesh in the test gearbox, P = 80 kW are dissipated. The total loss of the bearings in the test gearbox is P ~ 28 kW. The heat output, which is emitted to the environment, is divided as follows.

The clamping field emits $Q_{conv,\,clamping} = 1.18$ kW via convection and $Q_{rad,\,clamping} = 0.706$ kW via radiation to the ambient air. Via the warm part of housing $Q_{conv,\,housing,\,warm} = 0.800$ kW is transferred convectively and $Q_{conv,\,housing,\,cold} = 0.962$ kW via the cold housing part to the surrounding air. Additionally, $Q_{rad,\,housing} = 0.617$ kW is transferred from the housing to surrounding air by radiation.

The temperature at the sealing gaps results in negligible radial thermal expansion, so there is no solid-state contact and the sealing gap at maximum temperature still amounts s = 0.6 mm. The axial expansion of the gearbox shafts can also be classified as not critical according to the simulation, so that safe operation under full thermal load is guaranteed. The pinion shaft expands by $\Delta l_{pinion\ shaft} = 0.53$ mm and the wheel shafts by $\Delta l_{wheel\ shaft} = 0.43$ mm. This expansion has to be considered in the design of the axial moveable sealings to prevent solid-state contact, on the one hand, and to ensure sufficient sealing of the gearbox housing, on the other hand.

Symbol	Description	Value
$Q_{bearing,\,oil}$	Heat input through housing (bearing shells) and shaft into the lubricating oil in the journal bearings	345.23 W
$Q_{conv,\,clamping}$	Convection of the clamping field to the environment	1179 W
$Q_{rad,\,clamping}$	Heat dissipation by radiation at the clamping field to the environment	706 W
$Q_{conv,\,housing,\,warm}$	Convection of the housing to the environment	800 W
$Q_{conv,\,housing\ cold}$	Convection of the housing to the environment	962 W
$Q_{rad,\,housing}$	Radiation of the housing to the environment	617 W

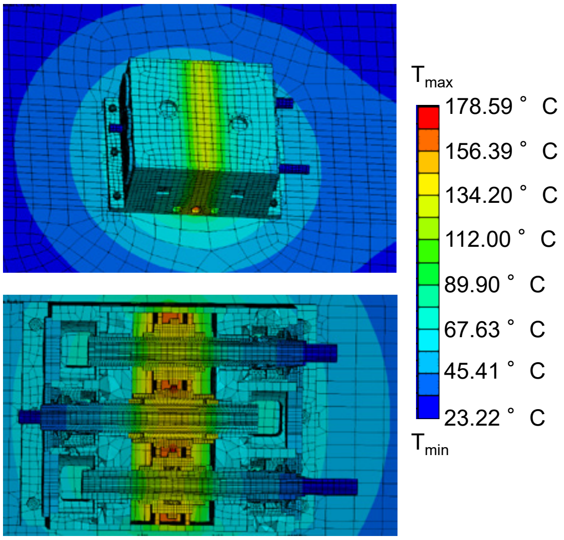


Figure 12 Results thermal simulation of the test gearbox with Ansys.

Summary and Conclusion

Aircraft engines can be made more efficient by integrating planetary gears. In such an application, the planetary gears experience very high load cycles under fully reversed bending loads. Pulsator test rigs, which nowadays offer the possibility to perform UHCF investigations, can only be used for purely pulsating loading of gears. Therefore, for the investigation of the UHCF tooth root load carrying capacity under fully reversed bending load, a back-to-back test rig is required. Back-to-back test rigs usually have speeds of n = 3,000 rpm, which makes investigations in the UHCF range take a very long time. Therefore, a high-speed back-to-back test rig was developed.

One aim of the report is to define the operating parameters of this high-speed back-to-back test rig. Additionally, the thermal behavior and thermal expansion of the components is checked by a thermal simulation with Ansys. The test rig design is transferred to the Simpack multibody simulation, and the dynamic operating behavior is analyzed. Additionally, the dynamic loads from Simpack are compared to the values of the K_V factor according to method B of ISO 6336, and the operating parameters of the high-speed back-to-back test rig are defined.

The test rig uses a shifting helical gear for load application, which realizes torque control during operation as well as a compact design. The analysis of the dynamic torque in the test and reference gearbox during run-up shows that the dynamic loads occurring in the test gearbox are greater compared to the reference gearbox. The local torque maxima are caused by the excitation of the torsional natural frequencies of the shafts by the tooth mesh frequency and its harmonics. The dynamic load in the test and reference gearbox from Simpack is compared with the dynamic factor according to method B of ISO 6336. The dynamic factor according to ISO 6336 underestimates the dynamic loads of the test gearbox at the operating speed $n_1 = 12{,}000$ rpm, whereas the additional loads in the reference gearbox are overestimated. Furthermore, the vibration behavior is decisively determined by the torsional natural frequencies of the gearbox shafts, which are not taken into account in the standard calculation according to method B. Since resonances do not occur above $n_1 = 4{,}000$ rpm, the operating point does not have to be restricted. Below $n_1 = 4000$ rpm and above the minimum speed of the journal bearings, it makes sense to load the gearing with half the test torque in order to avoid preliminary damage due to tooth flank lift off. According to the differences between standard calculation approach and multibody simulation, it is particularly useful for high-speed applications to use multibody simulations considering all degrees of freedom. Furthermore, the resonances at low rotational speeds require a gradual ramp up of torque and speed during the run-up of the test rig. Using this approach, preliminary damages of the test gears (e.g., caused by flank lift off) can be prevented.

The thermal simulation of the test rig estimates the power loss of the test rig to the surrounding room and is used to check the functionality of the test rig under thermal expansion at maximum operating temperature. In addition to the heat dissipated by the cooling oil volume flow (P =390 kW), a heat loss of P = 4.2 kW was obtained, which is dissipated by the test gearbox to the ambient air. For a safe and continuous operation, it has to be ensured that the room ventilation is powerful enough to remove the losses. The simulation also showed that the functionality of the test rig is fully given at maximum temperature and thermal expansion. The maximal thermal expansion of the shafts in axial length is $\Delta l_{pinion\ shaft} = 0.53$ mm and $\Delta l_{wheel\ shaft} = 0.43$ mm. The sealing gaps expand in radial direction, but at the maximum temperature there is still a gap of s = 0.6 mm. The amount of thermal expansion makes clear that a simulation of the thermal expansion for high testing temperatures, as well as for high rotational speeds, is necessary. Thereby, the fits of bearings and seals can be chosen as close as possible to ensure a safe operation.

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Moritz Trippe is a research assistant in the Gear Department of the Laboratory of Machine Tools and Production Engineering (WZL) of RWTH Aachen University and the group leader of the gear power density group. His research focus is the load carrying capacity of the tooth root and the influence of dynamics on the gear load capacity. Trippe studied mechanical engineering at RWTH Aachen University and has a master´s degree in automotive engineering and transport.

Oscar Malinowski, M.Sc. is currently a research associate at the Chair of Machine Tools at the Laboratory for Machine Tools and Production Engineering (WZL) of RWTH Aachen University in Germany. He started his scientific career with a bachelor's degree in mechanical engineering, followed by a master's degree in aerospace engineering from the RWTH Aachen University. He joined WZL as a research engineer in 2019. Since then, he focused on investigation and design of high-speed rolling bearings and spindle bearing systems. Furthermore, his research includes optimization methods for balancing of motor spindles to improve structural dynamic behavior of machine tools.

Dr.-Ing. Jens Brimmers is the head of the gear department at the Laboratory for Machine Tools and Production Engineering (WZL) of RWTH Aachen University since June 2019. He graduated from RWTH Aachen University with master's degrees in mechanical engineering and business administration. His Ph.D. thesis focused on beveloid gears and topological tooth flank modification.

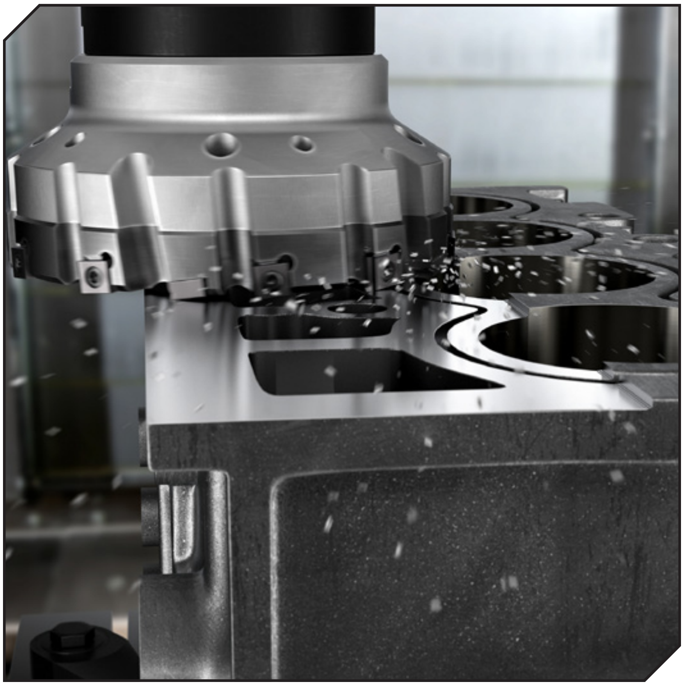
Dipl.-Ing. Stephan Neus, born in 1984, studied mechanical engineering at RWTH Aachen University before working as a research associate at the Laboratory for Machine Tools and Production Engineering (WZL) of RWTH Aachen University in the Machine Technology Department between 2014 and 2019. Since 2019, Mr. Neus has continued his work at the WZL as chief engineer of the Machine Technology Department.

Prof. Dr.-Ing. Christian Brecher has since January 2004 been Ordinary Professor for Machine Tools at the Laboratory for Machine Tools and Production Engineering (WZL) of the RWTH Aachen, as well as Director of the Department for Production Machines at the Fraunhofer Institute for Production Technology IPT. Upon finishing his academic studies in mechanical engineering, Brecher started his professional career first as a research assistant and later as team leader in the department for machine investigation and evaluation at the WZL. From 1999 to April 2001, he was responsible for the department of machine tools in his capacity as a Senior Engineer. After a short spell as a consultant in the aviation industry, Professor Brecher was appointed in August 2001 as the Director for Development at the DS Technologie Werkzeugmaschinenbau GmbH, Mönchengladbach, where he was responsible for construction and development until December 2003. Brecher has received numerous honors and awards, including the Springorum Commemorative Coin; the Borchers Medal of the RWTH Aachen; the Scholarship Award of the Association of German Tool Manufacturers (Verein Deutscher Werkzeugmaschinenfabriken VDW); and the Otto Kienzle Memorial Coin of the Scientific Society for Production Technology (Wissenschaftliche Gesellschaft für Produktionstechnik WGP).

Sandvik Coromant

EXAMINES SHIFT TO ELECTRIC VEHICLES

According to a report by the International Transport Forum, global transport activity is expected to double by 2050 compared to 2015 levels. With climate change a global concern, it's important that we produce more efficient vehicles that run on greener fuels. Material choice and innovations in metal cutting are supporting the transition to more environmentally friendly vehicles.



While COVID-19 caused a short-term reduction in transport activity, vehicle use is still set to rise over the coming decades as the global population increases and economic development continues. Producing more vehicles to meet increasing demand is inevitable—the real challenge is making them greener. The International Energy Agency (IEA) reports that transport already accounts for 24 percent of direct CO_2 emissions from fuel combustion, with road vehicles accounting for nearly three-quarters of this figure.

The United Nations Framework Convention on Climate Change (UNFCCC) recognizes the importance of lowering the transport sector's emissions and thus released its Climate Action Pathway for transport earlier in 2021. The UNFCCC's vision is that, by 2050, passenger and freight transport will be completely decarbonized following a shift to more sustainable vehicle technologies. This shift is broken down into transitioning to zero-emission modes of transport and increasing vehicle efficiency.

The EV evolution

Among zero-emission modes of transport are electric vehicles (EVs), which executive director at the IEA, Fatih Birol, says, "have an indispensable role to play in reaching net-zero emissions worldwide." By using electricity, specially from sustainable sources, from the grid to recharge batteries that power an electric motor, EVs produce zero tailpipe emissions and are thus a more environmentally friendly option than internal combustion engine (ICE) vehicles.

According to the IEA's Global EV Outlook 2021 report, there were ten million electric cars on the world's roads at the end of 2020, and EV registrations increased by 41 percent that year. While EVs are clearly on the rise, their adoption can be accelerated further by overcoming range anxiety, the fear that the vehicle has insufficient range to reach its destination.

And range is not just the responsibility of the battery. In addition to a more robust charging infrastructure and improvements to EV battery design, every element of a vehicle needs to be made lighter. An EV with a lower weight requires less energy to travel a given distance and can therefore go further on a single charge, increasing its range.

Aluminum plays a large role in the light-weighting of EVs, weighing just a fraction of the more traditional automotive materials of choice—steel or cast irons. In fact, aluminum is now commonly selected for a range of vehicle parts, such as the chassis, internal panels, motor housing and battery enclosures. According to the Aluminum Transport Group (ATG), using aluminum to reduce an EV's weight can result in range gains of approximately the same proportion. For instance, if the vehicle weight is reduced by 20 percent, the vehicle should be able to travel around 20 percent further on the same charge.

Mastering machining

However, components made from aluminum are notoriously more difficult to machine. Aluminum is softer than most metals, which can make it challenging to work with. What's more, the melting point of stainless steel is 1510°C, whereas for aluminum it's 660°C. When machining the metal, the lower melting temperature of aluminum means that chips can build up from the heat of friction at high speeds and adhere to the tool. This chip accumulation can dull the tool, making it difficult to cut through the billet. In addition, manufacturers can face issues such as time-consuming tool setups, inconsistent tool wear, burr formation and inferior surface finishes. The need for high speeds is also a struggle.

Fortunately, these challenges can be overcome by selecting a tool with an optimized design that's made from advanced materials. For instance, the Sandvik Coromant M5C90 face milling tool, part of its M5 cutter series, was designed for solid aluminum parts with long milling operations, as well as for roughing and finishing cylinder heads, blocks and electric car components. In just one efficient operation, the M5C90 can perform the entire machining process, from roughing to finishing. In many cases, this can be with a depth of cut of up to four millimeters. This tool can increase tool life by fivefold and reduce cycle time by up to 200 percent.

In addition, the M5 cutter series features step technology, where it's extremely hard-wearing polycrystalline diamond (PCD) inserts are arranged in a spiral and staggered vertically to remove material from the workpiece, both axially and radially. Furthermore, the last tooth has wiper geometry to further ensure a high-quality, flat-surface finish. The wiper cutting edge remains in a fixed position, which eliminates the need for time-consuming setups. Other tools in the M5 series include the M5B90 face milling cutter concept for fine finishing and the M5F90 combination milling cutter for roughing and finishing in smaller dimensions.

The transition to EVs will lower transport emissions, and adoption can be accelerated by increasing their efficiency. EVs that take advantage of aluminum components can travel further per charge, helping to overcome range anxiety. Automotive manufacturers who select machining tools optimized for aluminum will be able to produce high-quality aluminum EV components—helping to support the shift to greener travel.

sandvik.coromant.com

Forest City Gear

ANNOUNCES APPOINTMENT OF NEW PRESIDENT, KIKA YOUNG

Forest City Gear is proud to announce the appointment of **Kika Young** as president of the company.

Young looks forward to building upon the legacy of her parents, Fred and Wendy Young. Wendy Young was president of the company from 2002 until 2012, when she was named CEO. She served as CEO until her passing in February of 2022. Wendy was instrumental in establishing Forest City Gear's reputation as a world-class, family-owned gear manufacturer.

Under Kika Young's leadership, the company's senior management team, which includes Jared Lyford, director of manufacturing operations; Jeff Mains, director of technical operations; and Gary Strakeljahn, controller, will continue to drive Forest City Gear's commitment to excellence without exception, to delivering to the high standards of quality and service that the company is known for throughout the industry.

According to Fred Young, chairman of Forest City Gear, "To say it's been a difficult year is an understatement. At the same time, we are so thankful for the support we've received. It has been invaluable in helping us to make a plan as a family and for the company. Ultimately, we all agreed that the best way to honor Wendy's legacy and the legacy that every member of the Forest City Gear family has helped build, is by continuing to operate as a family-owned company. I look forward to watching Kika as president, with the support of her sisters Appy Mikel and Mindy Young, lead the company into the future, while continuing to honor the traditions that make Forest City Gear such a special place."

forestcitygear.com

Big Daishowa

ANNOUNCES VICE PRESIDENT SALES

Big Daishowa Inc. announces **Michael Herman** has been promoted to vice president sales. In this role Herman oversees the Big Daishowa sales team and supports the company's distribution network and machine tool builders throughout North America.

"Mike's dedication and extensive knowledge of the industry has been an invaluable asset in growing the company. The role of Vice President Sales is an expansion of the work Mike has already accomplished," says Big Daishowa President/COO, Jack Burley.

"In this role, I look forward to continue supporting manufacturing and sharing my knowledge to help companies not only reach, but surpass their productivity goals," says Herman.

Herman brings 40 years of experience to this role, including 10 years at Big Daishowa Inc. (formerly Big Kaiser Precision Tooling Inc.). During his tenure at the company, Herman has continued to build and develop relationships with the sales team, distribution and business partners, with a personal commitment of promoting a higher performance guarantee.

bigdaishowa.com

Velo3D

ANNOUNCES EUROPEAN ADDITIVE MANUFACTURING ROADSHOW

Velo3D, Inc. announced its Seeing is Believing Additive Manufacturing Tour for Europe, which will visit seven cities in 2022 across Germany, France, Italy, and the United Kingdom. The roadshow brings together innovators across key industries including space, aviation, oil and gas, and energy to share how additive manufacturing and the Velo3D end-to-end solution are transforming these businesses by helping engineers manufacture the parts they need without compromise.

"When we talk to prospective customers who have experience with additive manufacturing, the first thing they want is to see the parts for themselves to witness whether the geometries we offer can truly be achieved," said Renette Youssef, Velo3D CMO. "In our pilot of the Seeing is Believing roadshow we made amazing connections between engineers, technology influencers, and executives in the additive manufacturing industry to learn from and inspire one another. We look forward to achieving this same community building in Europe."

Velo3D initially launched its roadshow in 2021 across five cities. The events brought together hundreds of innovators to learn how the Velo3D end-to-end solution delivers unprecedented part quality, complex geometries, repeatability in distributed manufacturing, and high-volume production through its scale-up Sapphire XC printer.

"With 3D printing being a relatively new manufacturing technology, these community building events help innovators conceptualize how it can be used to transform their businesses, improve efficiency in key systems, and solve their biggest challenges," said Campbell MacPherson, Schoeller-Bleckmann Oilfield (SBO) EVP of Advanced Manufacturing. "As the first company to acquire a Velo3D Sapphire printer in Europe, we're thrilled to share how the technology has helped us better serve our customers."

The roadshow events focus on educating engineers of all types on the many benefits and capabilities of advanced additive manufacturing. The presentations highlight the process from start to finish, including pre-print, during printing, quality assurance, the underlying manufacturing process, and post processing. The 2022 show will make seven stops at European cities, including:

- Lyon, France — June 10
- Augsburg, Germany — June 21
- Toulouse, France — September 15

Future dates will also be announced for Milano, Italy; Birmingham, England; and Munich, Germany.

velo3d.com

JMP Solutions

OPENS AUTOMATION AND ROBOTICS FACILITY

JMP Solutions recently announced that construction is complete on its new state-of-the-art Automation and Robotics facility in Northwest Arkansas and that it is open for business. The 20,000-square-foot manufacturing and integration space will provide automation and robotics solutions to the region and create approximately 140 jobs in the next five years.

With the new facility, the automation systems integrator provides U.S. customers greater access to its intelligent automation and robotics services and can more easily serve its growing roster of core customers in the region. In addition to automation

and robotics integration, the company's offerings also include material handling, automated guided vehicles, and controls systems integration. The facility further positions JMP as a global leader in the growing $500 billion automation market.

"We're energized by the growth that we've experienced and the opportunity that it has afforded us to move forward with this new facility," said Ray English, commercial leader of JMP's Automation and Robotics Division in Arkansas. "This investment in Northwest Arkansas not only allows us to better serve our existing customers in this area, but it also gives other potential customers in the U.S. easier access to the automation services JMP provides."

Many manufacturing and process industries continue to experience staffing and labor related challenges, which have only been exacerbated by the pandemic. These challenges have introduced safety and sustainability concerns and presented opportunities for automated solutions to stabilize manufacturing operations. This announcement by JMP demonstrates its commitment to continued growth in addressing these challenges head-on through increased capacity and capabilities across a continually expanding footprint.

In addition to expanding its reach and providing greater access to its customers, the new facility is positioned to provide unique, ongoing skills development, technology training and continuing education to better recruit and maintain high-quality talent for the area. "JMP Solutions is a values-oriented and people-centric company," said Mark Fulmer, who was recently appointed director of operations for the new Arkansas facility. "We look forward to partnering with local universities and trade schools to attract the best new talent, and we are wholly committed to providing an enriching work experience for the entire team."

JMP's close relationships and partnerships with industry leaders in the region, along with the proximity of the University of Arkansas and other premiere vocational and technical schools, will make Northwest Arkansas a top training and development sphere for the next generation of automation innovators.

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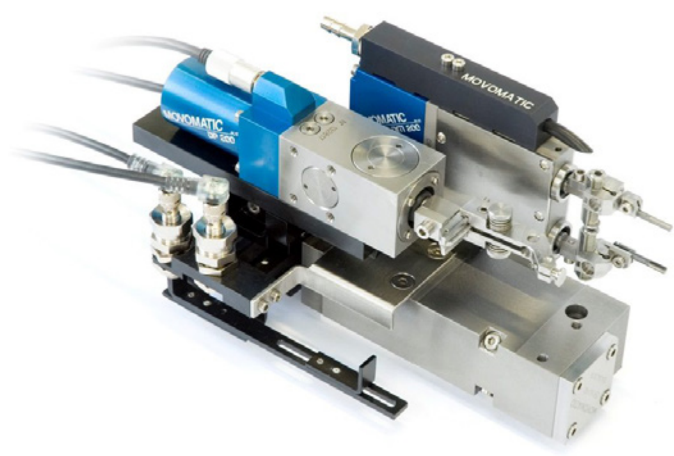
jmpsolutions.com

Marposs

ACQUIRES GRINDING MACHINE MEASURING TECHNOLOGY FROM JENOPTIK

Marposs has announced the acquisition of Jenoptik's (Jena, Germany) non-optical process measuring technology business for grinding machines, formerly known as Movomatic. With this acquisition, Marposs is taking over the facilities, management and employees at the main site in Peseux, Switzerland, as well as in Ratingen and Ludwigsburg, Germany, ensuring continuity of service and support levels.



Marposs is bringing the Movomatic brand back to the market, which had been abandoned following its integration into Jenoptik. For the time being, dedicated market strategies will be maintained with an autonomous approach to the respective customers. Possible integration between the product lines, such as combining Movomatic measuring solutions with Marposs wheel balancing systems and sensors, will be evaluated for possible future technical integration.

This acquisition, in combination with Marposs' expertise in the field of high precision measuring and control solutions for industrial production, expands Marposs market presence and portfolio of offerings for its customers.

marposs.com

LK Metrology and Wenzel Technologies

ENTER INTO A STRATEGIC PARTNERSHIP

LK Metrology and Wenzel Technologies have entered into a strategic partnership with immediate effect. In addition to a joint technological cooperation, Wenzel Technologies will sell LK's innovative, high-performance coordinate measuring machines (CMMs) and other metrological products in southern Germany.

For more than 50 years, the name Wenzel has stood for the highest standards in metrology. Frank Wenzel, who once built the measurement technology company founded by his father into a world market leader and since 2018 has been successfully concentrating on milling solutions for automotive design studios, is now opening the next chapter. With products from LK, he is expanding his portfolio again with powerful systems for measurement, inspection and quality control.

Frank Wenzel, CEO of Wenzel Technologies explains, "LK has an impressive equipment portfolio that surpasses even my previous products in terms of accuracy and customization. I am firmly convinced that we will be able to meet the high expectations of the German customers with these solutions."

Established nearly 60 years ago, LK Metrology is the oldest CMM manufacturer in the world. Many innovations in measurement technology, which are now recognized as standard in the industry, can be traced back to the English company. LK's products including CMMs, portable measuring arms and metrology software, are used worldwide to control and improve the quality of components, both in production and in the laboratory. Customers come from the aerospace, automotive, medical, energy and defense sectors, amongst others.

Wenzel Technologies operates two development centers in Germany as well as a software laboratory in California and employs numerous measurement technology experts, some of whom have decades of experience.

Angelo Muscarella, CEO of LK Metrology says, "With its immense technical know-how and excellent knowledge of the market, Wenzel Technologies is an ideal cooperation partner for us. Together, we will present highly attractive, innovative and holistic solutions at attractive prices."

lkmetrology.com
wenzel-technologies.com

Forest City Gear

WELCOMES VISITORS TO TOUR MANUFACTURING FACILITIES, VIRTUALLY

Forest City Gear, an industry-leading manufacturer of fine and medium pitch custom gears, welcomes visitors to virtually tour the Roscoe-based company's manufacturing facilities via the company's website.

This recently launched tool offers virtual visitors 360-degree views of 30 plus locations across Forest City Gear's primary and secondary facilities and highlights the company's state-of-the-art equipment and capabilities for gear shaping, grinding, hobbing, inspection, and engineering services.

According to Kika Young, president of Forest City Gear, "One of Forest City Gear's key differentiators has always been a commitment to maintaining our facilities to the highest standards and continuously reinvesting in the most technically advanced equipment available. As COVID-related safety measures and travel restrictions made in-person tours impossible, we were looking for a way to continue sharing that experience with our customers. And while COVID mitigations have relaxed and we're again able to welcome in-person customer visits, this virtual experience offers broader reach to share facilities updates, new equipment, expanded capabilities, etc. all in real time."

fcgvirtualtour.com

July 20–22—WZL Gear Conference—USA

The Eighth WZL Gear Conference—USA is being hosted by Reishauer Corp. and will provide the opportunity for North American companies to connect with WZL and learn about current research activities. For more than 50 years the annual WZL Gear Conference in Aachen, Germany, has been fostering technical collaboration and communication among the members of the WZL Gear Research Circle. The two-day conference is devoted exclusively to the presentation of the latest research in gear design, manufacturing and testing. Additionally, the software resources of the WZL Gear Research Circle are available for examination, including solutions for gear design and manufacturing process development.

geartechnology.com/events/5019-wzl-gear-conference-usa

August 10–11—AGMA Basic Gear Inspection for Operators

This AGMA course will provide a solid foundation for anyone going into gear inspection. Learn the common, current and basics of the tools and techniques used to measure and inspect gears. Understand the four main categories by which a gear is evaluated and classified. Gain proficiency in understanding gear quality by learning the numerical scale on which gear design, manufacture and inspection are based, and more. This course is taught at the AGMA National Training Center at Daley College by Mark McVea.

agma.org/education/advanced-courses/2022-basic-gear-inspection-for-operators/

September 12–17—IMTS 2022

The International Manufacturing Technology Show (IMTS) is the largest manufacturing technology show in the Western Hemisphere. The IMTS conference brings the industry together to discuss new opportunities and network with the manufacturing community. Other highlights include the Smartforce Student Summit, Exhibitor Workshops, the Emerging Technology Center and IMTSTV. Pavilions include additive, gear generation, machining, tooling, quality, controls and more. IMTS is co-located with Hannover Messe USA where cobots, digital twins, and smart factory solutions come together under one roof. See the latest industrial automation, motion and drive technologies in the East Building.

imts.com

September 12–14—International Conference on Gear Production 2022

Bildquelle: © WZL, RWTH Aachen / Ahmad

The requirements in gear manufacturing are increasing dramatically which delivers a dilemma between productivity (scale) and flexibility (scope) of today's technical solutions. Lead times need to be shortened in order to increase productivity while batch sizes are getting smaller and individual geometric features (topological modifications) push the need for intelligent support by manufacturing simulation and closed-loop approaches. The digitalization of the gear manufacturing processes promises high potential but also raises some challenges. In addition, new technologies evolve, that challenge the conventional manufacturing chain for gears. Topics in Garching, Germany include manufacturing of internal gears, manufacturing processes, gear soft machining, new concepts for machine and manufacturing processes, advances in special gearings, modeling in gear production, measurement technology and gear hard machining.

vdiconference.com/event/gear-production/

October 17–19—AGMA Fall Technical Meeting 2022

The gear industry is faced with emerging trends and innovations challenging engineers to stay course with the latest design, quality, materials, and analysis technology. It is imperative that researchers and gear engineers communicate ideas with fellow experts in the field. AGMA's annual Fall Technical Meeting (FTM) is the forum to share research and disperse knowledge for the benefit of the global gear industry. Each year, authors selected by AGMA write peer-reviewed technical papers on gear topics such as design and analysis; manufacturing and quality; materials, metallurgy, and heat treatment; operation, maintenance, and efficiency; and gear failure. The authors will present their work at the 2022 FTM in Rosemont, Ill., (outside of Chicago). All papers presented at FTM will be indexed in Scopus, the international database of peer-reviewed literature.

agma.org/events/fall-technical-meeting-ftm/

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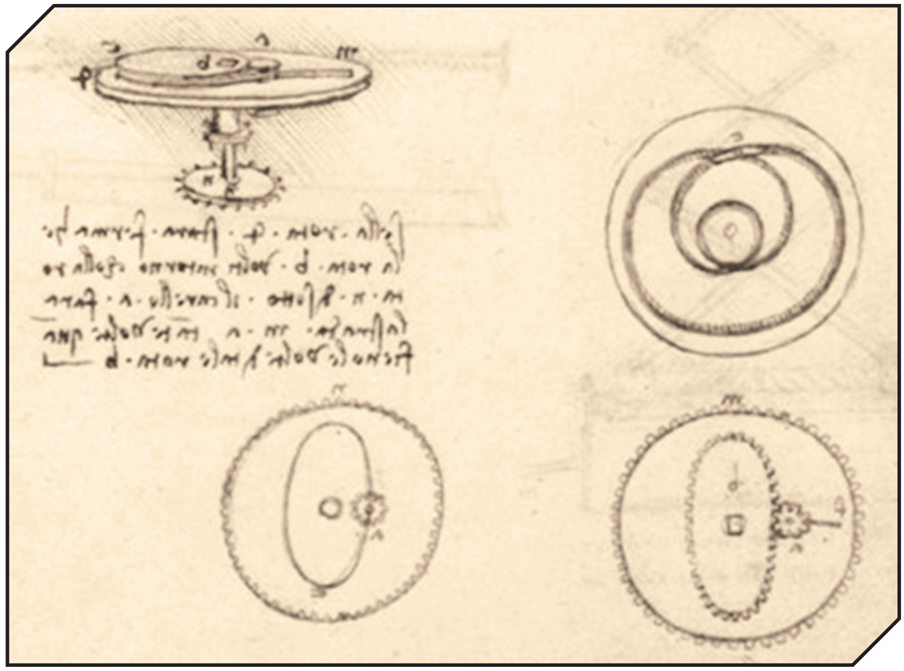
Noncircular Gears

The unicorn of machine technology

Aaron Fagan, Senior Editor

Noncircular gears are not a mere mathematical curiosity with limited practical utility. They were first sketched by Leonardo da Vinci around 1500 and have since found their way into a variety of useful applications. In the 18th century, noncircular gears were used in flow pumps, clocks, music boxes, toys, and other devices. Early publications on the gear type in the 19th century by Hamnet Holditch (1842), Henry T. Brown (1871), and Franz Reuleaux (1875) helped evolve the field of kinematics. First introduced by Uno Ollson in his book *Non-Circular Bevel Gear* in 1959, the noncircular bevel gear has remained obscure due to the complex geometry. Even though more and more publications are available on noncircular gears, the knowledge is, especially compared to cylindrical gears, still very limited. But in the last decade, there has been an increased interest in the field of noncircular gears due to certain advantages they have over circular gears.



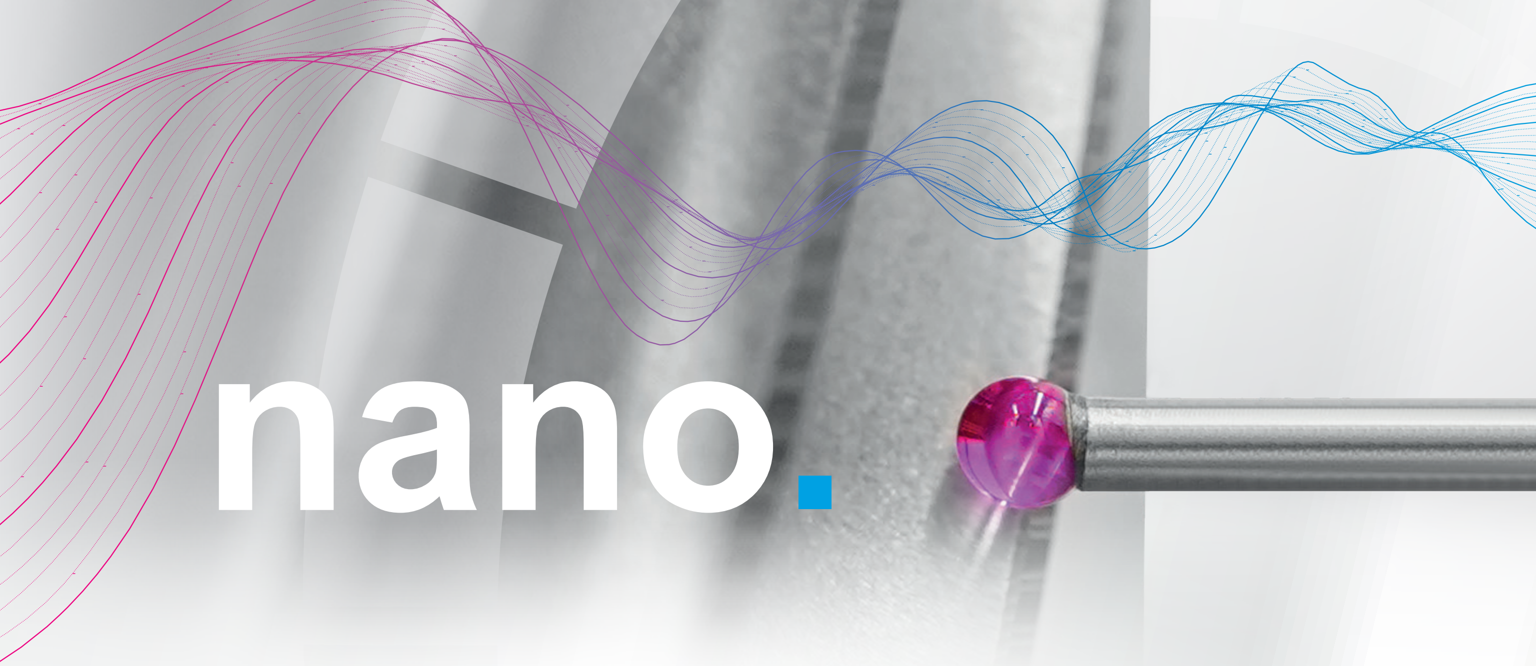
A noncircular gear design from Leonardo da Vinci's *Codex Atlanticus* (1478).

In 2009, Cambridge University Press published a comprehensive overview titled *Noncircular Gears: Design and Generation* by Faydor L. Litvin, Alfonso Fuentes-Aznar, Ignacio Gonzalez-Perez, and Kenichi Hayasaka. The book applies modern theories of gearing to the design and manufacture of the main types of noncircular gears: conventional and modified elliptical gears, eccentric gears, oval gears, gears with lobes, and twisted gears. It even goes so far as to speculate methods of generating noncircular gears by developing methods akin to those applied to the generation of circular gears and aims to extend the application of noncircular gear drives in mechanisms and industry.

As additive manufacturing, novel alloys, and other technologies mature they will allow greater manufacturability and exploit the unique mechanical features noncircular gears have to offer. Historical applications have included textile machines, nonlinear potentiometer drives, Geneva indexing devices, continuously variable transmissions (CVTs), window-shade panel drives, mechanical presses, flow meters, and high-torque hydraulic engines. A circular gear set is designed to roll together without slip and optimized to transmit torque to another engaged member with minimum noise and wear and maximum efficiency, whereas a noncircular gear's main objective might be ratio variations, axle displacement oscillations, and other modifications and configurations to achieve dynamic forms of utility. Because of the complex geometry, noncircular gears typically have a straight-toothed elliptical form and are manufactured by molding or sintering as opposed to conventional forms of gear generation.

The design and manufacture of noncircular gears are one of the unicorns of machine technology. Due to the limited technological capabilities of traditional machines and tools, past manufacturing methods were unable to calculate the assumed shape of the tooth line and its profile. Modern manufacturing methods using CNCs allow for the generation of noncircular gears. Development in the field has been slow due to the complicated nature of equations describing noncircular gears which have made their analysis virtually impossible without the aid of computer technology.

Recently, concurrent with the development of more sophisticated design and manufacturing technology, research on this gear type has been reinvigorated. Academics have synthesized the pitch cones for elliptical bevel gears; investigated a tooth profile generation method for noncircular straight bevel gear and high-order involute modified noncircular bevel gear; established a mathematic model for noncircular straight bevel gear based on the spatial meshing theory; and investigated noncircular bevel gear with concave pitch curve. Noncircular gears are increasingly used in practice and further research will eventually examine the geometric and kinematic accuracy, as well as the strength, of noncircular gears manufactured through different methods. Noncircular gears are designed to have variable output loads and speeds. It is also possible to design them with variable center distances. Compared to cylindrical gears, the main advantage of noncircular gears is their potential for EVs, robotics, automation, and other industries where the variable transmission function could be a revolutionary solution.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