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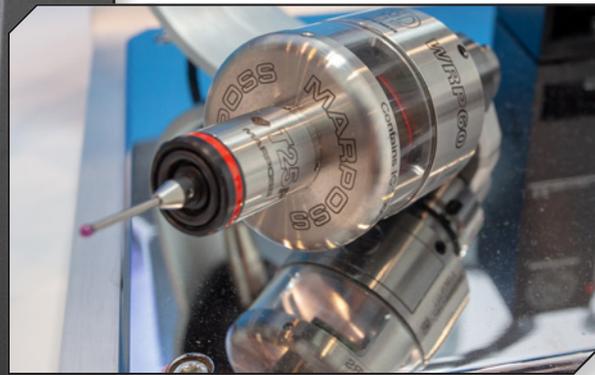
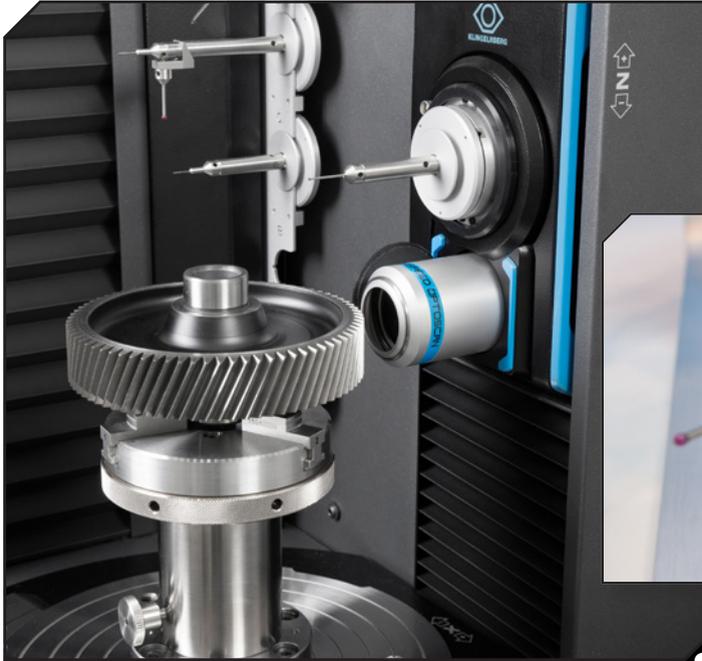
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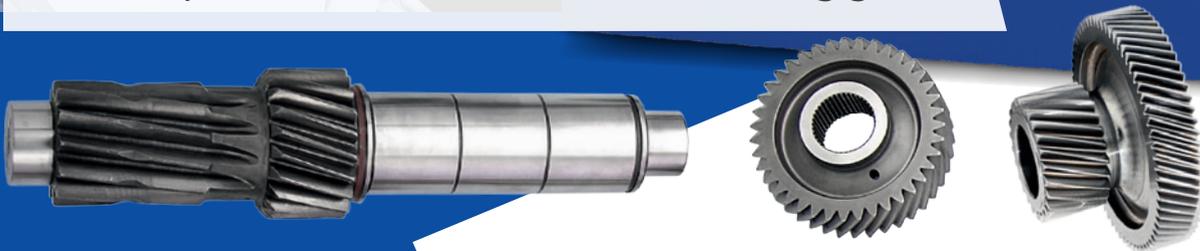
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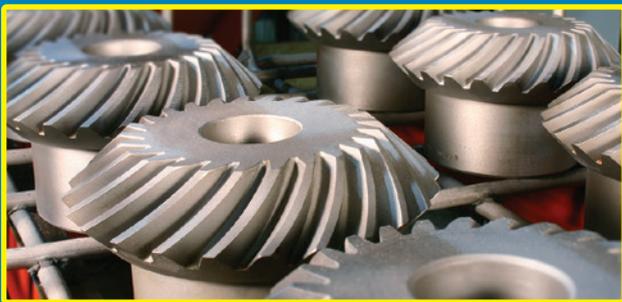
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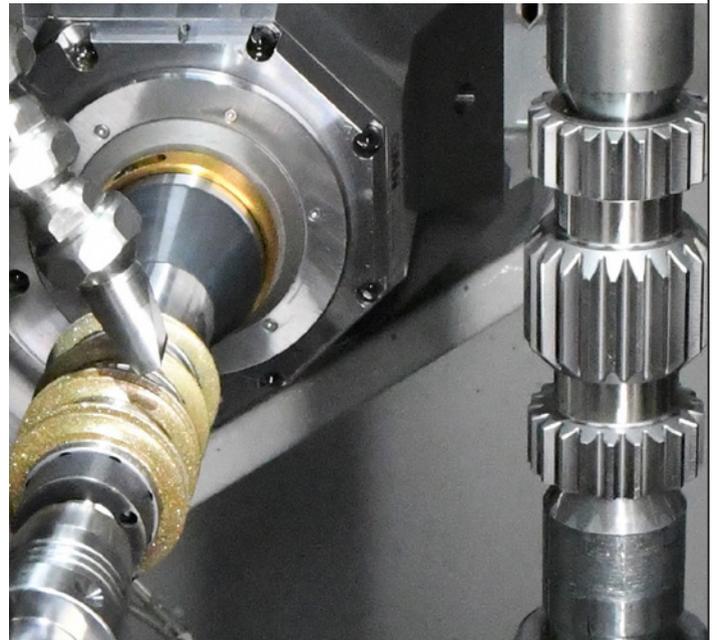
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**Michael Goldstein** founded Gear Technology in 1984 and served as Publisher and Editor-in-Chief from 1984 through 2019. Thanks to his efforts, the Michael Goldstein Gear Technology Library, the largest collection of gear knowledge available anywhere, 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](http://www.geartechnology.com). Michael continues working with the magazine in a consulting role and can be reached via e-mail at [michael@geartechnology.com](mailto:michael@geartechnology.com).



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# How Do You Make the Perfect Gear?



**Publisher & Editor-in-Chief**  
Randy Stott

**Okay, I'll admit it.** It's a trick question, because of course there's no such thing as a perfect gear. There will always be manufacturing errors, inconsistencies and human error. Tools and machines wear down, so that the first gear you produce will always be different from the one hundredth. Every time you change a tool or clamp a new workpiece, you introduce more opportunity for error. And those errors all stack up: machine, tooling, workholding, gear blank, etc., are all in a constant state of change throughout a manufacturing run, whether we're talking about a single piece or 10,000.

So you can't make a perfect gear. But even if you could, you would never know, because you can't measure one, either. Inspection equipment is made up of moving parts, so it's subject to the same problems as manufacturing equipment. When you introduce other variables – like the temperature of the room or the part being measured – it becomes clear that not only can you not measure a perfect gear, but neither can you measure an imperfect one the same way twice.

But it's not all bad news. The problem is really in the question. If I'd asked, "Can you make better gears, consistently and at an affordable cost?" the answer would be undeniably yes. And if you want to know how, this issue of *Gear Technology*, with its focus on quality, is certainly a good place to start. Whether you're looking for answers on the shop floor, in the inspection lab or for high-volume or low-volume production, we've assembled a batch of articles that can get you thinking in the right direction.

Duane Veit's article (p. 10) shows how a simple hand-held gage can improve and speed up the process of measuring gears over balls or pins. Traditionally performed with a micrometer, building the technique into a repeatable hand-held gage allows for the checking of parts even while they're still on the machine, allowing operators to catch problems in-process.

In "The Measure of Success," Senior Editor Matt Jaster interviews the experts in metrology software at Klingenberg to find out how gear inspection in the era of Industry 4.0 has kept pace through the use of advanced software and closed-loop processes (page 22).

Klingenberg has also contributed a short article on the company's hybrid inspection systems (page 26), whereby the repeatability and accuracy of physical probes is combined with the speed of optical metrology to provide highly accurate inspection capabilities at production volumes.

Gene Hancz of Mitutoyo explains the role of the CMM in modern gear inspection in his article, "Mitutoyo Examines Modern Gear Measurement," beginning on page 28. Hancz explains why the use of CMMs in gear manufacturing environments will likely increase due to the increased need for higher quality across many industries, as well as the increased use of nonstandard or modified gear profiles across those industries.

Measuring what you've made is surely an important part of gear quality and all of the above-mentioned articles go a long way toward helping you do just that. But equally important in terms of gear quality is designing your gears to meet the needs of your application in the first place. Because even a perfect gear doesn't remain perfect once it's out in the field. Its operation is greatly affected by lubrication, temperature and other factors.

That's why this issue's technical articles are equally important in terms of understanding gear quality. In his article (p. 54), Hanspeter Dinner of KISSsoft presents a comprehensive review of recent changes to ISO 6336, the leading global standard for the rating of cylindrical gears.

Also, the team from Mondragon University in Spain has written an in-depth research project about the effects of thermal distortion on small-size steel gears and how temperature affects quasi-static transmission error (page 34), crucial information for industries such as automotive and aerospace where pitch-line velocities of gears continue to increase.

Ultimately, every aspect in the life of a gear has an impact on its quality, from design to manufacturing, inspection and right on through to end use. We hope this issue has provided you with some insight into how you can keep pushing your gears closer and closer to perfection.

# Simplified Tooling Improves Gear Inspection Processes

DUANE VEIT, VEITTOOL & GAGE

## Introduction

Standard external gear sizing techniques include measurement “over balls” or “over pins.” For these measurements, a ball or a pin is inserted in teeth at opposite sides of the gear as shown in Figure 1. Using pins has the advantage of measuring the high spots of the gear surface, making the measurements more consistent. The figure on the left shows a measurement on a gear with an even number of teeth, on the right is shown gear measurement with an odd number of teeth.

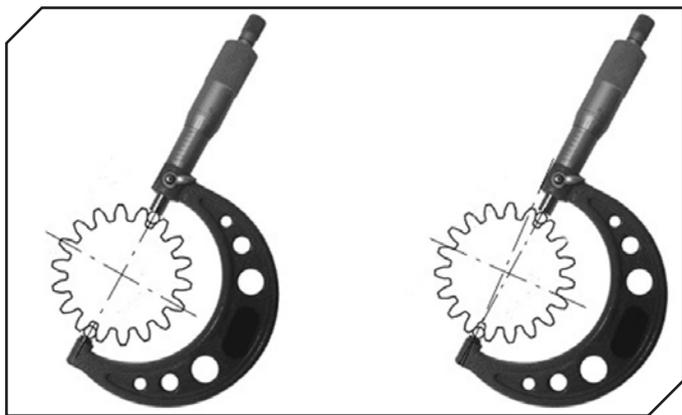


Figure 1 Tooth Thickness Measurement over Pins. Extracted from ANSI/AGMA 2002-D19 with AGMA permission.

Making over ball or over pins measurements using a micrometer requires careful test set up with a trained operator who has a good feel for the process to ensure proper measurement repeatability. Micrometers deliver absolute measurement

which require checking against a drawing, versus an absolute gage which directly delivers out-of-tolerance information. Testing with pins and a micrometer is difficult to do with a part that is in-process or still on a lathe or NC machine. Testing a gear while still on a machine can help validate processes and improve overall quality.

Inspecting gears with a CMM delivers high accuracy readings and helps ensure quality of multiple features. Using CMMs to measure every gear can be time consuming and costly and typically requires a trained operator. A CMM cannot make measurements with a part still on a production machine.

## Simplified Tooling Speeds Measurement

Integrating balls or pins into the tooling of a hand-held comparative gage offers many benefits to using a micrometer and loose balls/pins and testing with a CMM. Process checking is fast and can be done by an untrained operator even with a part on a machine. With simplified integrated tooling, a gage can be used to measure multiple gear characteristics with a simple tooling change, reducing the need to stock and calibrate multiple gages.

## Testing External Gear Pitch Diameter

Figure 2 shows a ONEGAGE measurement tooling with integral balls for testing external gear pitch diameter. The balls shown here are made of carbide or hardened steel with dimensional tolerances that meet the specifications called out in the gear manufacturing drawing. Tooling changeover is quickly made using two fasteners. Figure 3 shows a gear inserted into the tooling for measurement.



Figure 2 ONEGAGE tooling with integral balls for measuring external pitch diameter.

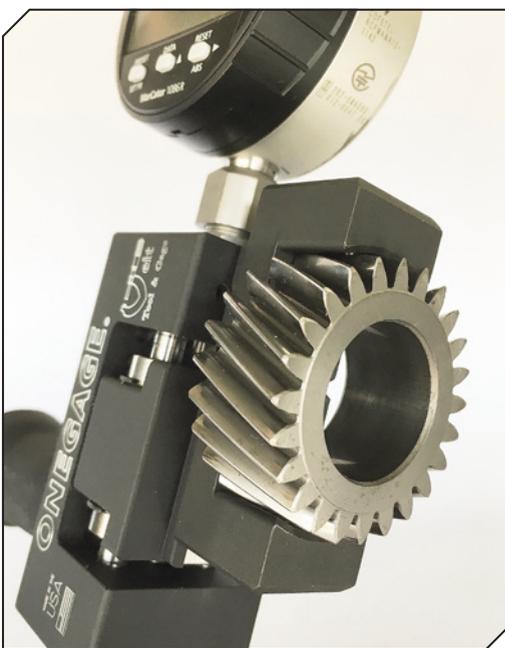
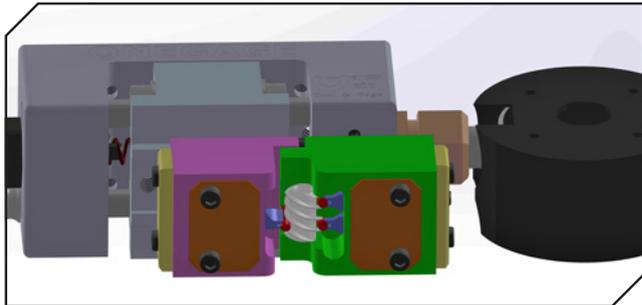


Figure 3 ONEGAGE measuring the external pitch diameter of a gear.

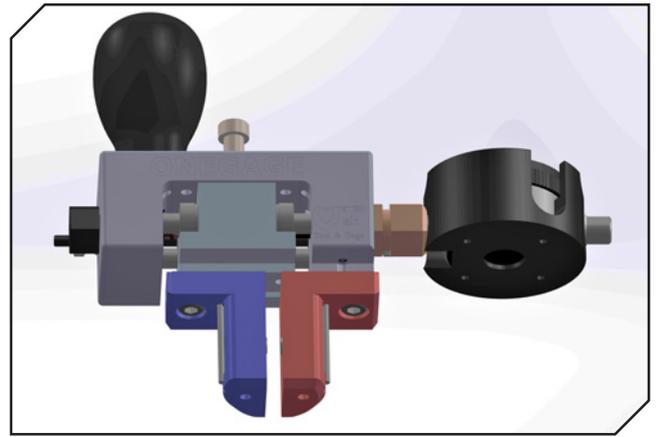
Figure 4 shows an engineering drawing of a dimension-over-pins test for a worm gear used on a snowmobile starter armature shaft. This test uses three pins to gage for proper pitch diameter. The pin diameters are called out on the specifications for measurement over pins on the worm gear drawing. The pitch diameter of the component gear will determine the placement location of the pins in the tooling. *SolidWorks* and customer part models were used to create the ONEGAGE tooling.



**Figure 4** Tooling and pin placement on worm gear. Pins are shown in red.

### Testing Internal Features

Testing internal features such as splines and ID gear pitch diameters can be accomplished in a similar fashion. Balls or pins are situated facing outward on the tooling. Figure 5 shows tooling designed for testing-over-pins of an internal spline. The pins in this application are trapped semi-floating against parallel surfaces, allowing them to find and comply to true position in relation to the spline pitch diameter.



**Figure 5** ID measurement using pins.

### In-Process Testing

Enabling engineers and machinists to make in-process measurements with parts on a machine helps validate processes and improve overall part quality. Having a comparative measurement gage which uses a master as a standard eliminates the need for the machinist to check dimensions against a drawing.

The gage shown in Figure 6 uses sealed and lubricated circuit bearings with 62Rc hardened and ground guide shafts to eliminate contamination from chips or lubricants that may affect gage performance. The spring actuation mechanism makes it easy to use one handed and with gloves. The part shown in Figure 6 is an internal 10mm threaded gear and is being inspected for internal thread pitch diameter. Electronic indicators with integrated

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wireless technology can also be used in place of the gage shown to collect data for trending or quality issues.



**Figure 6** Machinist gaging internal thread pitch of M10×1.5.

## Applications

### Aerospace/Aviation

A helicopter manufacturer needed to inspect an ultra-fine pitch (220 tooth) internal spline on a 50mm diameter. The part needed to be inspected while remaining captured in the machine to allow for adjustments to the pitch diameter to ensure the correct geometry. The tooling consisted of 0.4mm balls on a 3-point contact to accomplish a successful gaging solution with an acceptable R&R. The solution significantly reduced the overall scrap rate.

### Automotive Powertrain

An OEM powertrain manufacturer required tooling to inspect an internal 28 tooth spline over pins dimension. The solution used semi captured parallel floating carbide pin tooling on a standard ONEGAGE. The manufacturer was able to inspect parts on the floor, eliminating having to take them to an inspection lab to be ran on a CMM, saving both time and money.

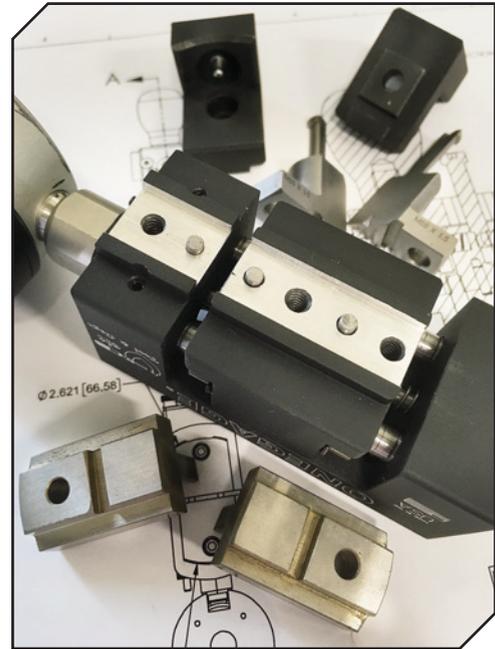
### Broach Manufacturing and Resharpener

A company that manufactures broaches needed a fast and easy way to validate the quality of their broaches. They used ONEGAGE to measure the diameter-over-pins measurement of their broaches. The same gage has also been used by a broach resharpener company.

### Simplifying the Process – Tool Changeover

To eliminate gage proliferation, the ONEGAGE was developed to work with a standardized mounting feature for common mounting of all tooling. Tooling consists of one fixed and one

sliding tool. Female keyways are ground at assembly and have a single centered dowel pin to ensure perfect alignment with the opposing tools. A single screw is used to secure each tool. Tooling can be changed out in under one minute.



**Figure 7** ONEGAGE Tooling Mounts.

### High Repeatability

In the development of any gage and gage tooling, it is critical to ensure high accuracy and repeatability. Using precision machining for tooling, ONEGAGE can achieve accuracies and repeatability of 1 to 2 microns. Using a “setting master”, a gage R&R of 10.17% was achieved when testing an internal spline pitch diameter.

### Conclusion

Comparative gaging with dedicated innovative tooling can simplify measurements, improve gear inspection processes, and reduce inspection costs and time. These types of gaging systems can enable inspection of parts while on a machine which helps improve the overall process and quality control. Having a gage with easy to replace tooling can reduce the overall number of gages that a manufacturer must own and maintain.

**veit-tool.com**

**Duane Veit** is the President of Veit Tool and Gage in Davison, MI which he started in 1988. Duane developed the ONEGAGE solution based on 30 years of gaging experience and input from industry leading gear experts. Veit Tool and Gage also manufactures gear roll test fixtures, dimension over ball/pin DOB/DOP gages, and gear burnishing systems. The company's website is [www.veit-tool.com](http://www.veit-tool.com), and Duane can be reached at [dveit@veitool.com](mailto:dveit@veitool.com).



# Affolter Group

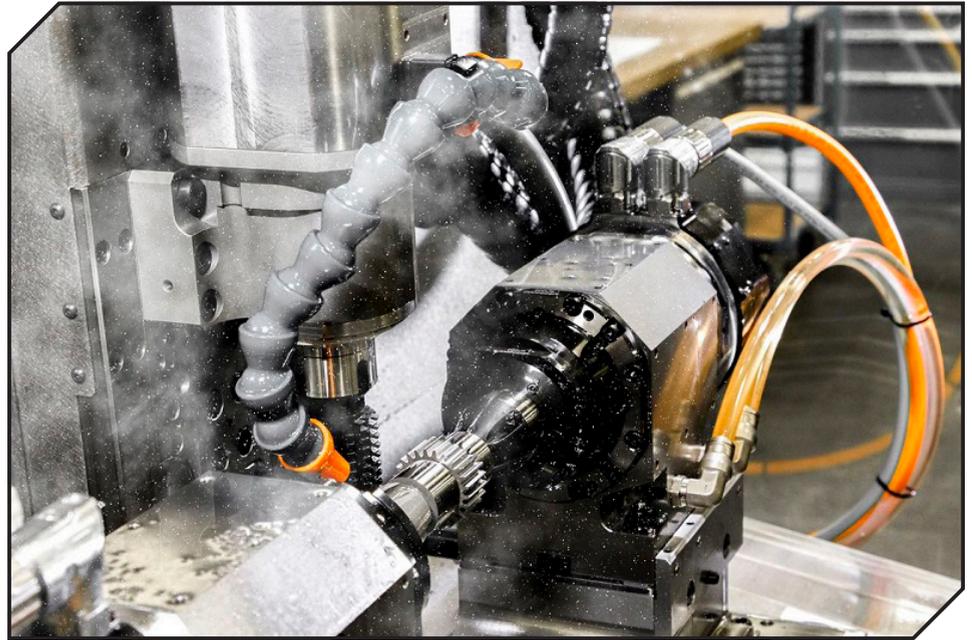
## OFFERS VERSATILE AND EFFICIENT GEAR HOBBING MACHINE

Affolter Group has introduced the AF160 is the most versatile Affolter gear hobbing machine to date. “The AF160 is designed for high precision manufacturers that need versatility and maximum efficiency,” explains Vincent Affolter, managing director of Affolter Group. “With eight axes, a state-of-the-art digital CNC control, a variety of automation solutions and a maximum module of 2 mm, it is ideal for manufacturers in industries such as automotive, aerospace, aircraft, gearbox, medical or robotics.”

The AF160 can process parts with an outside diameter of up to 60 mm and a length of 250 mm. The machining length is between 110 and 180 mm. The eight axes — all of them independent — make the AF160 the most flexible Affolter machine to date. It can produce straight gears, helical gears, straight bevel gears, face gears, straight or helical crowned gears, worm screws, worm wheels, cyclo gears, and internal gears. Power skiving, the milling of worms and shafts, as well as chamfering is possible, too. Affolter continued, “The AF160 enables manufacturers to produce an impressive variety of high precision gears, worm screws and worm wheels on the same, compact machine.” The footprint of the machine is only 4m<sup>2</sup>. Including the loader AF72 it is a little more than 6m<sup>2</sup>.”

“We think of it as a solution, not a machine — a solution that meets the needs of our customers. Thanks to the new CNC Control, various automation systems and peripherals, and the versatility of the AF160, we can offer maximum productivity for high-precision manufacturers in a very broad range of applications,” Affolter added.

The engineers of Affolter Group cooperated with Beckhoff Automation to launch the brand-new CNC Control Pegasus. The high processing power ensures extremely fast regulation. “It controls all machine axes as well as a multitude of peripherals for various options and automations,” said Affolter. Programming is simple,



intuitive, and user-friendly with a 19-inch touch screen. The new CNC Control integrates IoT. Data can be shared on the cloud, streamlining after-sales service and preventive maintenance, and therefore minimizing downtimes. Software updates can be done remotely.

Depending on the application and production processes, manufacturers need automation solutions to facilitate an autonomous operation for 12 to 24 hours. Affolter provides a range of such automation solutions: The universal multi-axes part loading and unloading system AF72 was especially designed for the AF160. It features a double gripper system for parallel loading and unloading and offers different methods of feeding based on the volume, product, and application. The AF160 can also be equipped with the deburring unit AF54, integrating the deburring process into the gear production. Different clamping systems provide for added versatility. Customizable coolant systems and chip extraction conveyors are available as well.

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# Liebherr

## OFFERS SOLUTION FOR ROBOTICS AND SPECIAL APPLICATIONS IN GEARING

Precision gears for industrial applications have to deliver a top-class performance. In order for a robotic arm to achieve precise gripping movements, for example, extremely small and lightweight components are required that must also provide enormous transmission ratios. Cycloidal drives or Harmonic Drive gearboxes are used in particular. These simply constructed gearboxes are characterized by their precise transmission of motion, zero backlash and high transmission ratios, and they are also free from wear and tear. Furthermore, they are very compact in size. In order to manufacture these demanding parts, Liebherr has developed solutions and made its range of

the quality even more,” said Dr. Andreas Mehr, who is responsible for gear grinding and gear shaping at Liebherr. Due to the faster grinding times, generating grinding is comparatively less expensive.

For internal profile grinding of the roller seats on the inner ring, a grinding wheel had to be developed that is capable of grinding a full radius. Liebherr succeeded in doing this by producing its own CBN grinding wheels with electroplated bonds that are dressing-free and wear-resistant. This ensures maximum process stability and process quality. The user is also more flexible when changing from external to internal grinding on one machine: changeover is possible in less than 30 minutes.

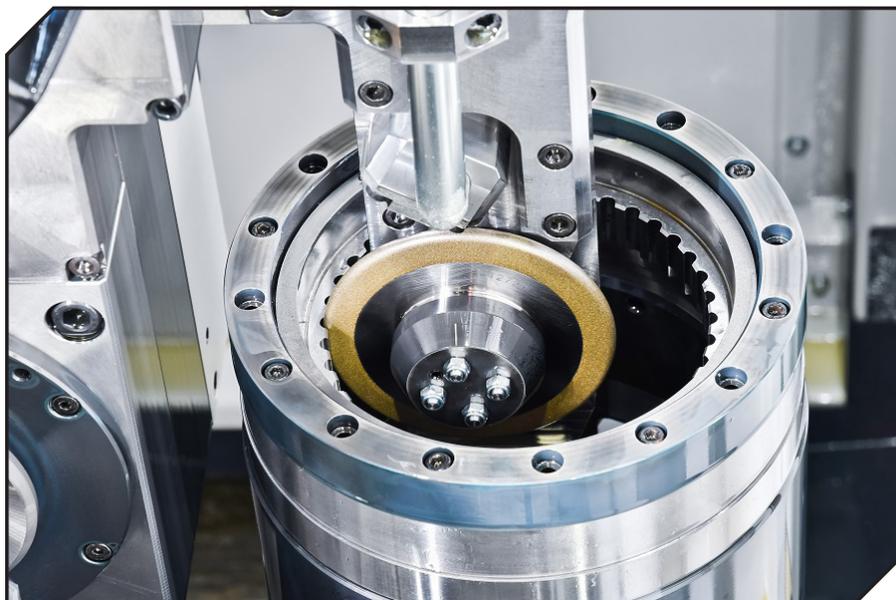
The gear teeth of the Harmonic Drive gearbox presented another manufacturing challenge. Here the load is distributed over a large number of tiny teeth, which, in extreme cases, are so tiny that they are barely visible to the naked eye. “When it comes to gear hobbing and gear shaping, we are sometimes at the limits of what is both technically feasible and still measurable” said Dr. Oliver Winkel, head of technology application at Liebherr. “But when it comes to extreme requirements in the production, handling and measurement of such small-modulus components, Liebherr is the perfect partner.”

Liebherr sees itself as a solution provider for the growing performance requirements resulting from the boom in automation, and is constantly working to expand its range of manufacturing processes. In the future, for example, the internal gear of the circular

spline for Harmonic Drive gearbox can also be produced by gear skiving, like on Liebherr’s LK 180 — another option for greater flexibility and efficiency. This also applies to other special cases, for which there may not yet be a solution already on the market but which will be developed in cooperation with the customer.

Liebherr also sees itself as a competent partner when it comes to meeting the growing demand for components for robotic applications and increasing productivity. Winkel explains: “Whether setting up a new production from scratch, supplying machines, defining processes, training employees or providing service and support,” Winkel said. “We have the expertise to advise and accompany our customers throughout the entire process.”

[www.liebherr.com](http://www.liebherr.com)



processes more flexible.

Dimensional stability, excellent surface quality, and high profile and pitch quality must all be ensured in cycloidal gears, and the rollers must fit perfectly with the inner ring. New from Liebherr and specially developed for cycloid gearing: externally toothed cams can now also be produced using single or precisely paired double clamping by means of generating grinding. Thus, depending on the number of pieces required, not only is profile grinding available to users but also a further grinding method for the cam discs.

For this application, generating grinding offers a number of advantages over profile grinding such as higher pitch accuracy, improved dimensional stability and an even profile over the whole cam disc, thanks to the improved wear distribution of the grinding worm. “By avoiding the undesirable ‘steps’ in the profile, we have been able to improve

# PTG Holroyd

## COLLABORATES WITH SIEMENS ON GEAR GRINDING CENTER

A new precision gear grinding center from PTG Holroyd hasn't simply been designed to bring greater levels of efficiency and accuracy to the production of specialized gears and tooth forms. The new machine, called the HG350-G, is also the first from PTG Holroyd - and believed to be the first in the UK - to use Siemens' new Sinumerik ONE future-proof CNC, the successor to the automation specialist's 840D CNC.

PTG Holroyd has committed in excess of £1.6 million to develop its brand new gear grinding center, a machine which has been designed to give the company a significant edge in the horizontal form grinding of high-quality gears. "We plan to build two HG350-G machines to begin with," comments Regional Sales Director, Mark Curran. "One will remain on site in Rochdale for R&D purposes. The other has been purchased by a long-standing PTG Holroyd customer."

In designing the HG350-G, PTG Holroyd's goal was to offer customers much more than a new generation machine for the one-off and batch grinding of high-accuracy precision spur and helical gears, as well as worms and screws up to 350 mm in diameter. "We wanted to

create a machine with class-leading integrated safety and failsafe features, rich, real-time reporting of machine health and performance data, as well as the highest levels of industrial security," adds Curran. "Other non-negotiables for the HG350-G included being exceptionally intuitive for operators,



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Reliability at work

easily able to accommodate each customer's Industry 4.0 strategy and being future proofed against legacy software issues. These were all attributes that the Sinumerik ONE CNC was able to offer. Additionally, the associated software suite's ability to create a digital 'working' twin of the machine on the desktop before build commenced was a considerable advantage."

"Typically, when creating a new machine tool, you begin with a vision — a concept of what you would

like your new machine to achieve," says Curran. "That said, software can't normally be written or mechanical components ordered until the design is complete and has been verified — and, even then, changes may be required. By working in close collaboration with Siemens, however, we were also able to embrace the 'Create my virtual machine' and 'Run my virtual machine' software capabilities of the Sinumerik ONE suite, in order to create and run a 'digital twin' of the proposed HG350-G."

Used in tandem with its own internal machine design packages, these capabilities enabled PTG Holroyd to build a virtual machine on the desktop, then grind virtual gears and threads, test safety and failsafe capabilities and eliminate potential problems before commencing the machine build. "At PTG Holroyd, we pride ourselves on a 'right first time' approach," adds Curran. "The virtual machine build and run capabilities offered in the Sinumerik ONE suite further helped us to fulfil this ethos, enabling us to input and observe entire manufacturing cycles before commencing a physical build. This will also make acceptance testing with future HG350-G customers simpler and straightforward. In short, they will be able to sign off on their gear grinder before it has even been built."

"It has been a privilege to work with the PTG Holroyd team to incorporate the capabilities of the new Sinumerik ONE CNC into the HG350-G gear grinder and to help utilize the benefits of the Sinumerik software suite to perfect the machine's design," comments Garry Mephram, application engineer at Siemens, "PTG Holroyd is an important customer of Siemens in the UK and I am confident their decision to use Sinumerik ONE will provide significant benefits — both in terms of machine design & development and by providing customers with future-proof capabilities and control."

Replacing PTG Holroyd's well-established GTG2 model, the HG350-G features the high power required for deep grinding operations. A specially developed extended machine bed allows screws and worm shafts of up to one meter in length to be accommodated. Dedicated software compensates for helical twist, and full topological capability comes as standard.

Embracing the Sinumerik ONE CNC's Profinet capabilities, IO-Link communication technology will be offered with all new HG350-G machines.

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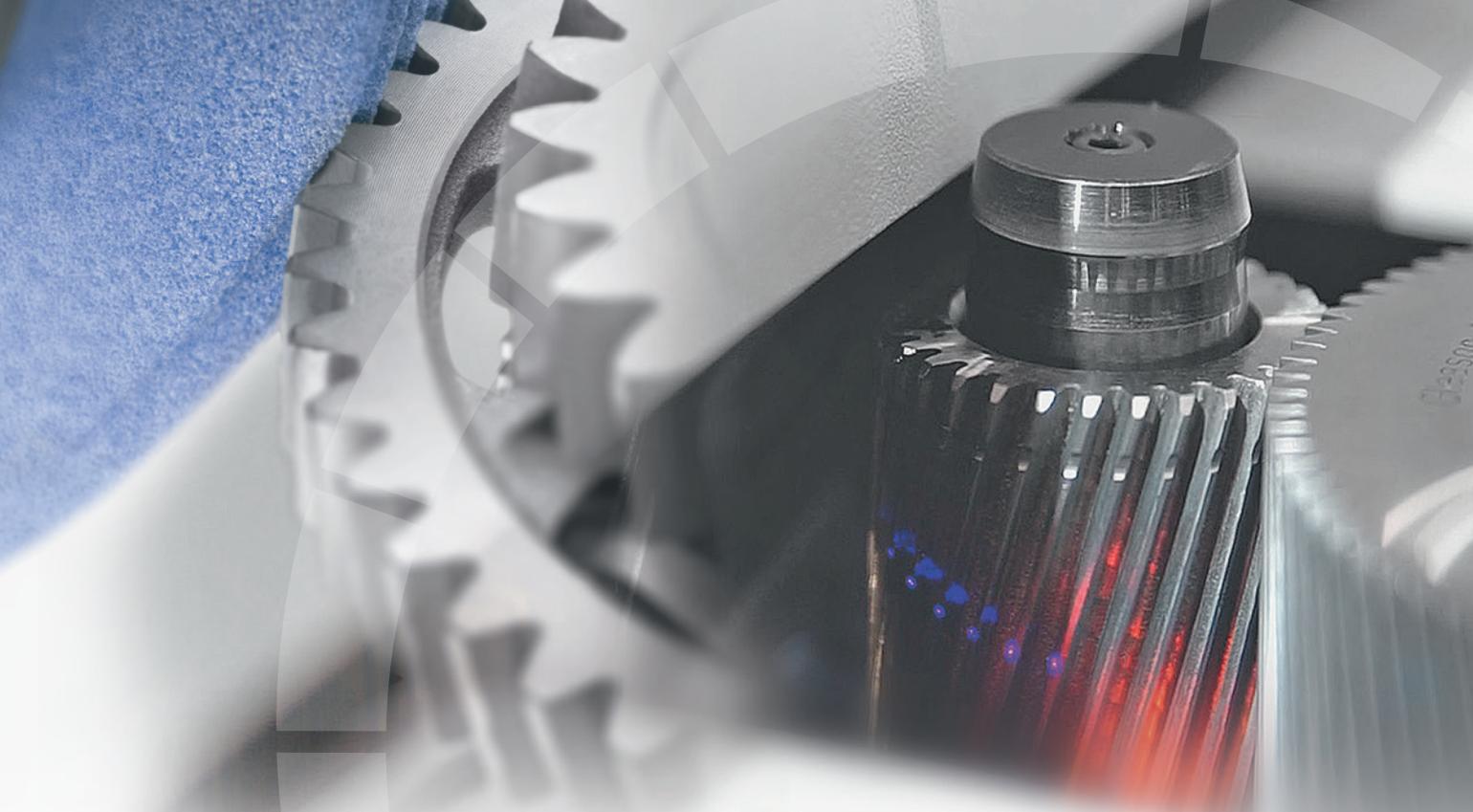
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A close-up photograph of a large industrial gear. A laser scanning probe is positioned against the gear's teeth, with a red laser line and blue light spots visible on the surface. The background is blurred, showing other gears and mechanical components.

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# BMW Motorrad

USES JUNKER JUCRANK FOR VARIETY OF GRINDING OPERATIONS

BMW Motorrad uses the JUCRANK non-cylindrical grinding machine for different grinding operations on boxer crankshafts. These are assembled into the latest BMW two-cylinder



boxer engines for efficient motorbikes. Impressed by the level of precision and extensive experience in developing CBN high-speed grinding machines, BMW Motorrad has been putting their trust in the Junker grinding experts for more than three decades, with 45 machines at five sites worldwide.

In-process measuring systems guarantee high process reliability and dimensional accuracy. The fully automated system measures the exact workpiece data and adjusts automatically during the grinding process if needed. This procedure reduces auxiliary process times and increases output.

Preventive Maintenance Assistance — the software-assisted solution from Junker. The online help system notifies of an upcoming maintenance or repair operation on the machine. The Junker machine notifies of upcoming repair operations and ensures reliable planning. Continuous monitoring of the results is possible with the new protocol manager. In addition, all data can be evaluated in freely selectable recording intervals. Thanks to digitalization possibilities with partner 4JU, the industry pioneers can adapt the digitalization of machines and systems precisely to the needs of BMW Motorrad.

When Service issues arise, the Junker experts can connect directly to the machine through a secure tunnel safely outside of the customers network. The Junker experts quickly analyze the data and reliably take all necessary measures.

It is possible to collect operating data efficiently and in a user-friendly way, with status determination (such as, for example, job start/end, ongoing updates and output. The advantage is that the entire production process can be

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BMW Motorrad and the Junker Group have been enjoying a positive business relationship for more than 30 years. BMW successfully uses Junker OD non-cylindrical grinding machines for machining crankshafts at their production sites in Germany, Europe and Asia.

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## Ransohoff

### MINI JET WASHER OFFERS EFFICIENT HIGH-VOLUME PARTS CLEANING

Ransohoff, a division of the Cleaning Technologies Group LLC, has introduced its Cell-U-Clean Mini Jet spray cabinet. This new small footprint platform provides an energy efficient, reliable solution for your high-volume cleaning needs, along with an innovated design for those smaller part cleaning applications.

The Mini Jet a member of the Cell-U-Clean line of parts washers, is an excellent cabinet washer for both small and large parts up to 500 Lbs. where floor space comes at a premium. This spray washer was designed as a

high-capacity solution with low energy as an alternative to our heavy-duty cabinet washer product line. The rotary hide away door eliminates the extra floor space required for the door swing while ensuring a dry floor when loading and unloading.

The new Cell-U-Clean Mini Jet is a wash and blow-off machine in a small footprint of 5 ft. wide x 8 ft. long x 7.6 ft. tall. The part production rate is 60 seconds cycles, with a variable speed table drive. The Cell-U-Clean Mini Jet is constructed of all stainless steel and comes with a robot loading option. The new Cell-U-Clean Mini Jet comes standard with the industry leading 3-2-1 warranty.

[www.ctgclean.com/cell-u-clean-mini-jet](http://www.ctgclean.com/cell-u-clean-mini-jet)

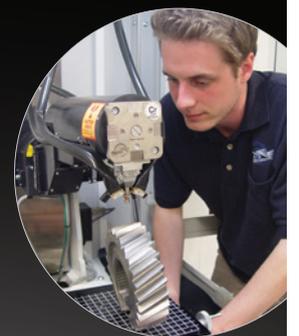


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# Helios Gear Products

## ANNOUNCES NEW CNC GEAR DEBURRING MACHINES

Helios CNC gear chamfer-deburring machines increase profitability for job shops and high-mix manufacturing.

“Gear manufacturers need a new level of gear chamfer-deburring that offers quick and easy setups with repeatable high quality,” said Adam Gimpert, president of Helios Gear Products. To meet this demand, Helios announces new CNC advances to its line of Tecnomacchine chamfer-deburring machine tools. These CNC improvements offer manufacturers a machine platform with maximum flexibility for a high mix of part sizes and types including spur and helical gears, bevel gears, bore- and shaft-type parts, and others. These abilities are driven by a new CNC system that programmatically controls all factors for the chamfer-deburring process. This equips manufacturers with a process that is easy to set-up and change-over for reliable and consistent quality.

Several factors impact successful chamfer-deburring with abrasive wheels

and carbide end mills. These include the tool position (radial, axial, tangential, and inclination), tool pressure, tool rotation speed, tool rotation direction, workpiece rotation direction, and workpiece rotation speed. Traditional machine tool deburring often requires manually

setting several if not all these items. Today, Helios offers its gear chamfer-deburring machine tools with friendly dialog programming of all these items. This allows a setup technician to store a complete application to later be recalled by the CNC with just a few software



### SMT

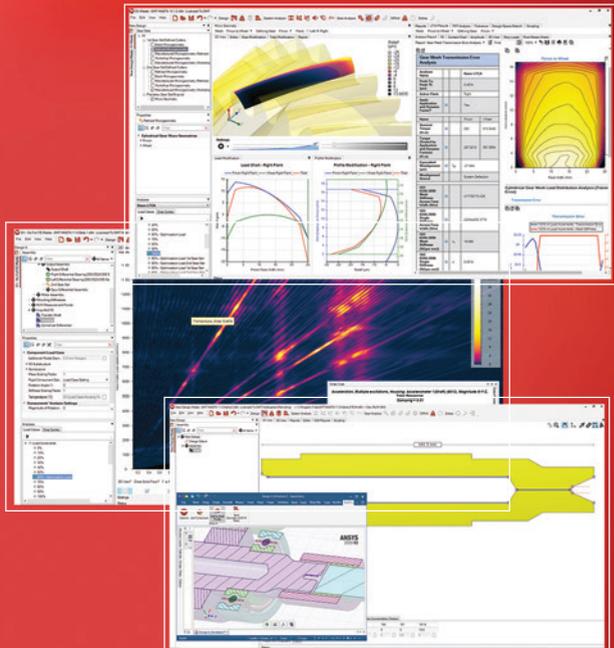
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# Mitutoyo

## RELEASES EJ COUNTERS AND LG100 SERIES LINEAR GAGES

steps. Additionally, tool spindles can be configured with brushing units for CNC brushing within the same chamfer-deburring cycle.

As one example, manufacturers use the Helios TM 250-CNC machine for profitable gear chamfer-deburring. This machine features a rotating table with two stations: one for loading/unloading parts by hand or integrated robot; the second for chamfer-deburring and brushing of the workpiece. This feature minimizes machine idle time and drastically improves productivity of the solution. Moreover, the machine implements a complete CNC system for the abovementioned abilities: quick and easy changeovers from saved setups of all tool and work positions and parameters. Changeovers can be accomplished in just a few minutes rather than the traditional 30-45 minutes.

For productive gear chamfer-deburring, the new series of Tecnomacchine equipment from Helios Gear Products offers manufacturers an ideal machining solution. With complete CNC functionality, these machines minimize change-over time, maximize repeatable quality, are easily programmed, and minimize idle time. Consequently, gear manufacturers can achieve top levels of productivity from their chamfer-deburring operations.

[Heliosgearproducts.com](http://Heliosgearproducts.com)

Mitutoyo America Corporation has released new EJ Counters and LG100 Series Linear Gages to its expansive sensor lineup of metrology products.

EJ Counters and LG100 Series Linear Gages are designed to be used inline or in measurement cells, enabling real-time measurement and data management in any type of work environment. Features of the EJ Counters and LG100 Series Linear

Gages include:

- Environmental Resistance: Sliding durability of more than 50 million cycles\*, with protection level of IP67G. A highly oil-resistant material is used in the rubber cap and cable sheath, so it can be used even in harsh environments (\*10 mm range models).
- Reference Point Detection: Reference point signal output functions are featured throughout this series. When incorporating into a device the master setting value is easy to set and is retained after shutdown.

- Combines compact size and ease-of-use: High-speed and compact, delivering production site visualization, improved productivity, and information feedback. Up to 8 compact EJ Counters can be linked providing the capacity to connect to 16 gages. On a DIN rail, each unit can be connected directly without using cables to maximize space. All linked units and gages can be driven by a single power source.
- CC-Link connection, USB connection: Data can be output through an industrial interface (CC-Link) by linking a compact EJ Counter with an interface unit. Constant data monitoring and positional management are performed. A USB interface is also provided for easy connection with a computer.
- Calculation Function: Enables sum difference operations between two gages connected to the same counter. Additionally, functions such as tolerance judgment and peak hold are included.

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# The Measure of Success

## Software Advancements in Gear Metrology/Inspection

Matthew Jaster, Senior Editor

**Thoughts on data-driven manufacturing?** What does it mean? What are the benefits? How will these solutions change the gear shop floor in the future? These are the questions many customers in our industry are asking right now. How can my organization become more productive via connectivity, flexible databases, or factory status reporting systems?

These are questions Klingelberg has been answering for more than 10 years, not only with the machine tools they produce, but all the products and technologies available to gear manufacturers to meet changing production demands. Metrology software is one area, in particular, that continues to evolve—the Closed Loop for cylindrical gears is one example of this.

“The focus is on the digital transfer of

measuring results to the manufacturing machine to enable automatic corrections,” said Dr. Christof Gorgels, director precision metrology at Klingelberg. “This procedure is well known and state of the art for bevel gears. For cylindrical gears we have chosen the open GDE standard to ensure compatibility of our measuring machines with machine tools from different sources.”

### High-Precision Gear Measurement

The basis for Klingelberg metrology has been high precision gear measurement for decades including parallel measurement software for different types of rotational symmetric parts, such as shafts and rings.

“In recent years, the importance of those measurements has increased. Today,

we offer our customers Done-in-One solutions for different types of parts such as geared shafts, crank shafts and even bearing rings,” said Gorgels. “The combination of coordinate- and form measurement plus surface roughness enables the Done-in-One principle—meaning all those measurements can be performed in one automated cycle. Thus, measuring time can be reduced up to 40 percent.”

According to Alexander Troska, head of software development, measuring devices at Klingelberg, the strength of Klingelberg metrology is the high number of different software options for measurement and evaluation. Klingelberg’s strategy is the close collaboration with customers to find tailored solutions to fit their needs best.

“There are two major fields for customization in our software. The in- and

Done-in-One metrology solutions allows Klingelberg to measure parts in one automated cycle.



output interfaces are highly individual from customer to customer,” Troska said. “The other big field is the evaluation. If you are looking into gears for example, it starts with different standards such as AGMA, ISO, JIS etc. But even looking into details many individual settings are needed like K-Charts. Another example can be the roundness measurement on bearing seats where different settings for filtering or FFT analysis are needed. Most of those settings are set by the operator.”

### Connectivity and Data-Driven Manufacturing

Industry 4.0 is all about data and communication. In fact, a measuring machine produces data only. Handling this data together with different customer’s systems is part of Klingelnberg’s daily business. Even communication with machine tools and servers providing nominal data has been part of Klingelnberg’s strategy for more than 20 years. A good example here is the Closed Loop for Bevel Gears.



By establishing a fast, continuous scanning cycle, measuring times could be reduced by nearly 50% without compromising on accuracy.

“The demand for data communication among our customers is highly increasing. Measuring programs are selected and started by scanning a DMC-Code and data needs to be collected from the production system. This is only one example where data connection makes

the machine handling easier,” added Jan Häger, head of new software development at Klingelnberg.

In E-mobility, for example, Häger said the requirements for gear design are increasing. This is related to load carrying capacity as well as noise behavior.

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This results in a need for tighter tolerances and better surface finish.

“For Klingelberg, this means putting a focus on measuring times since measuring frequency will increase as well as improving surface finish testing to cope with better surfaces. Also, the Gear Deviation Analysis software looking into gear noise will see higher demands.” Gorgels said.

Klingelberg’s latest development is the fully integrated optical system as part of its hybrid metrology. This enables significantly faster gear measurement while maintaining the high accuracy standard

Klingelberg is known for.

Last year Klingelberg introduced a new measuring software for cycloidal gears. The main challenge here is that all teeth have to be measured within a reasonable timing. By establishing a fast, continuous scanning cycle, measuring times could be reduced by nearly 50% without compromising on accuracy, Gorgels said.

“In the Industry 4.0 context, we have introduced the OPC-UA standard now also for measuring machines. It has been part of our gear machinery for some time but in the metrology area this is

new in the industry, but the demand is rising,” Häger said.

For bearing ring measurement, Klingelberg has introduced the G-Series. “Especially for thin rings a smart clamping system is available to avoid radial forces during measurement. Those radial forces cause deformations which influence the measuring results. Thus, we can ensure that deviations measured are really in the part and not caused by the clamping system,” Gorgels said. 

[www.klingelberg.com](http://www.klingelberg.com)

## Back to Basics: Closing the Loop

Gear cutting and gear grinding is faster and more efficient today thanks to the integration of gear design and evaluation software. The Closed Loop is the result of this gear manufacturing system which enables manufacturers to reduce gear development time.

The Closed Loop uses a modern software architecture to enable the exchange of data between design, production, quality assurance and statistical evaluation, and also to actively bring information to the consumer or, in another development stage, to initiate process steps automatically.

To meet the changing energy requirements in the automotive industry, many engines will be required to meet—and in some cases—exceed certain performance standards. Reproducible quality in series production with the fastest possible manufacturing times is the key requirement in this industry.

Spiral bevel gears are used to transmit torque “from the transmission to the road” in the all-wheel and rear-wheel drive systems in cars. Due to increasing performance requirements, some of the drives have to be able to transmit output of more than 300 kW. The bevel gears must also be efficient and

low-maintenance, and above all quiet.

The first step in modernizing a bevel gear production system is to implement a digital data transfer between the measuring center and the gear grinding machines. This means that information must be transferred digitally to the operator software of the gear grinding machine.

Now and in the future, gear manufacturers will be able to examine failure analysis if problems occur in the field. The data will provide a ‘digital roadmap’ for quality improvements and process optimization. Software products will be integrated into the closed loop system. As functions are added, this gear manufacturing platform will evolve as the technology does. 

For additional material on the Closed Loop System visit:

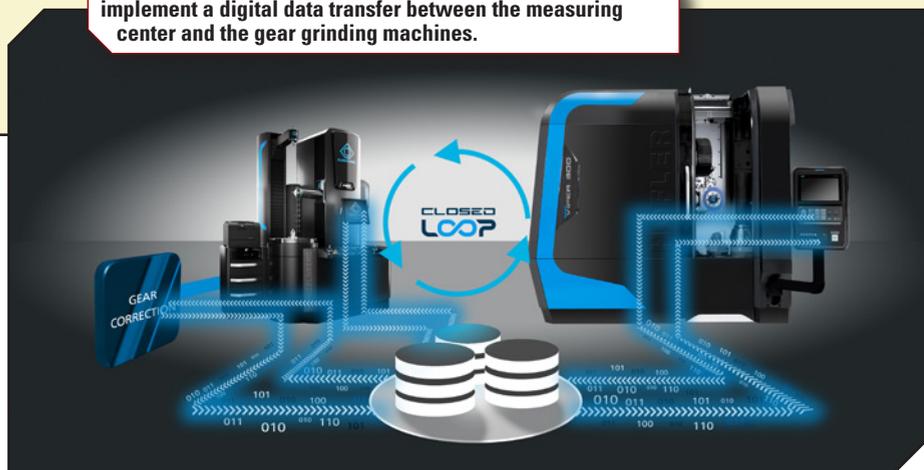
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The first step in modernizing a production system is to implement a digital data transfer between the measuring center and the gear grinding machines.



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# Hybrid Gear Metrology with Klingenberg

## An overview of tactile and optical gear metrology

Dr. Christof Gorgels and Markus Finkeldey

### Optical Gear Metrology

Gears are characterized by a rather complex geometry and tight tolerances, especially compared to other mechanical components. Looking into noise performance of gears, form tolerances are in the single micrometer range to ensure a quiet running behavior. Combined with the geometrical complexity, this is a major challenge for any new metrology standard to be established. For all features and their tolerances, the measuring system must be statistically capable to ensure appropriate accuracy and repeatability. With today's tactile measuring system, there are measuring devices available that meet nearly all the requirements. Tactile metrology is characterized by a high accuracy and repeatability as well as a high flexibility. However, for some applications measuring times are rather high, particularly if compared to production cycling times.

Non-contact optical systems offer the possibility to bring more speed into the measurement process. Faster measuring results will bring two major advantages. The first and obvious advantage is reducing the measuring costs. Nevertheless, the second and equally important advantage is the reduction of the setup time in production. Faster



Pitch measurement with HISPEED OPTOSCAN.

measuring results of the first part of a batch help to release the processes sooner, improving the output and overall efficiency of the production equipment.

The gear geometry, with its small pressure angles and therefore steep flanks, create a challenge for optical systems in combination with the tight tolerances. In the past optical systems were focused on bringing more information (such as flank topographies) in short time while compromising on statistical capability.

This can be an important feature for R&D but not for production.

With Klingenberg Hybrid Metrology a different approach is taken. It combines the best out of the two worlds. The tactile NANOSCAN is applied where the most flexible and robust system is needed. The HISPEED OPTOSCAN optical sensor is used where measuring speed can be gained without any disadvantages. The combination of both systems ensures fast and statistically capable measuring results.



Profile and lead trace measurement with tactile NANOSCAN sensor.

### Klingenberg Hybrid Metrology

In serial measurement of a cylindrical gear, the profile and lead are typically measured on three or four teeth, and pitch measurement is performed on all teeth. This tactile pitch measurement necessarily involves inserting the stylus into each gap to get in direct physical contact to both flanks of the teeth. With optical measurement, by contrast, only light enters contactless into the tooth gap. Accordingly, pitch measurement offers a great potential for reducing the measurement time. Through optical pitch measurement using one continuous, uninterrupted rotation of the component, the measurement time advantage

increases with large numbers of teeth to up to 80 percent while reducing the wear of the tactile stylus. This optical pitch measurement can be combined with the tactile measurement of profile and lead or other features for example using the tactile roughness system. Overall, the total measurement time can be decreased by up to 40 percent. Thus, in cases where there is a high utilization rate of the measuring machine, the costs for the optical metrology option are quickly recovered. The technology therefore makes a significant contribution to reducing quality costs in the first step. In addition, Klingelberg Hybrid Metrology will evolve in the near future, adding more optical measurement solutions.

**Decreased measurement time is not the only key factor**

Just as important is a high achievable accuracy of the measuring results, even in the case of extremely complex gears with ground surfaces and steep profile angles. This is the result of intensive optimization of Klingelberg's own sensor technology, the analysis algorithms, and the measurement strategy. The only difference in operation is that optical pitch measurement must be selected in the same cylindrical gear measurement software customers are already familiar with. The measuring run is automatically modified accordingly, and the pitch measurement is performed with the optical sensor. The changeover between the tactile 3D NANOSCAN probing system and the optical HISPEED OPTOSCAN sensor takes place automatically within approximately 1.5 seconds in conjunction with the entire measuring run.

In terms of achievable measurement time advantage, one thing is true: The more teeth there are, the bigger the advantage is. 

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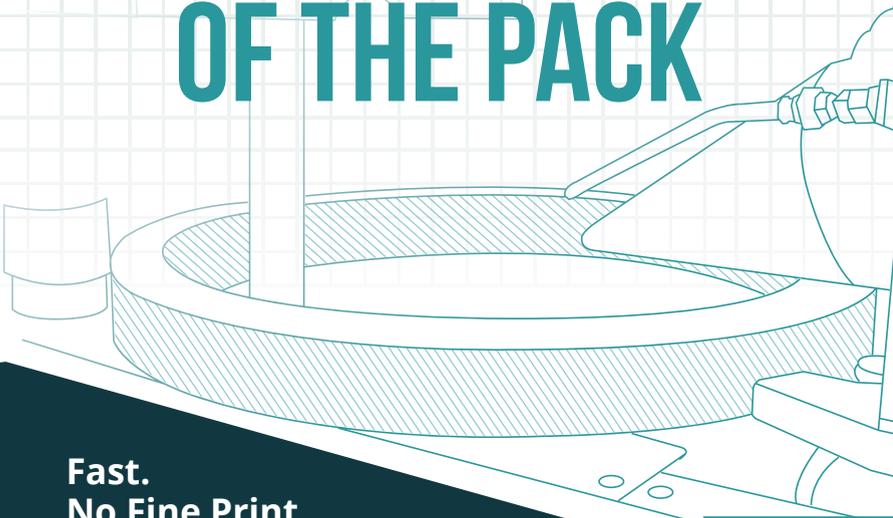


**Markus Finkeldey** is project manager of optical metrology at Klingelberg. Responsibilities include the operative and strategic controlling of the optical metrology section in Klingelberg's Hybrid Metrology approach.

**Dr. Christof Gorgels** is the head of the metrology business unit at Klingelberg. Responsibilities include R&D and after sales.



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# Mitutoyo Examines Modern Gear Measurement

Gene Hancz, CMM product specialist, Mitutoyo and  
Derek Sporleder, application engineer, Mitutoyo

**When it comes to modern gear measurement, there are still essentially two primary tools that shops use. The first is a machine dedicated solely to measuring gears, which had been the primary method for many years, until recently.** This type of dedicated machine can provide detailed measurement information that is solely specific to gears and gear teeth, including thickness, width, tooth thickness and pitch diameter. However, this type of machine does not measure characteristics of parts that gears get mounted on to. For this reason, a more flexible, automated option has become progressively more popular.

This second, and increasingly common, tool is a multiuse machine like a Coordinate Measuring Machine (CMM), where gear measuring technology can be added so that gears are one of the many things capable of being measured.

## What to Look for When Selecting a Gear Measuring Tool

You need to keep in mind that gears have their own standards of quality. In the U.S., the American Gear Manufacturers Association (AGMA) is responsible for setting those standards. To demonstrate accuracy of gear tooth geometry, up until 2015 AGMA used a rating system created in 2000 called the AGMA 2000-A88 standard, with accuracy grade numbers ranging from Q3 to Q15, where larger numbers indicated a tighter tolerance.

The new standard developed in 2015, the AGMA 2015-1-A01, reduced the number of accuracy grade numbers from 13 to 10, and reversed the accuracy related to such numbers to align better with other international standards. New accuracy grades are designated between A2 and A11, with lower-accuracy numbers signifying a tighter tolerance. In addition, this new standard is a pure metric standard.

DIN standards are another gear rating system that's accepted outside of the U.S., especially in Europe and Germany. In Japan, JIS standards are used. And there are also ISO standards for gears which generally apply outside the United States but do have acceptance in the states. Organizations must be able to adapt to the international standards present in the location where gears are being designed and manufactured.

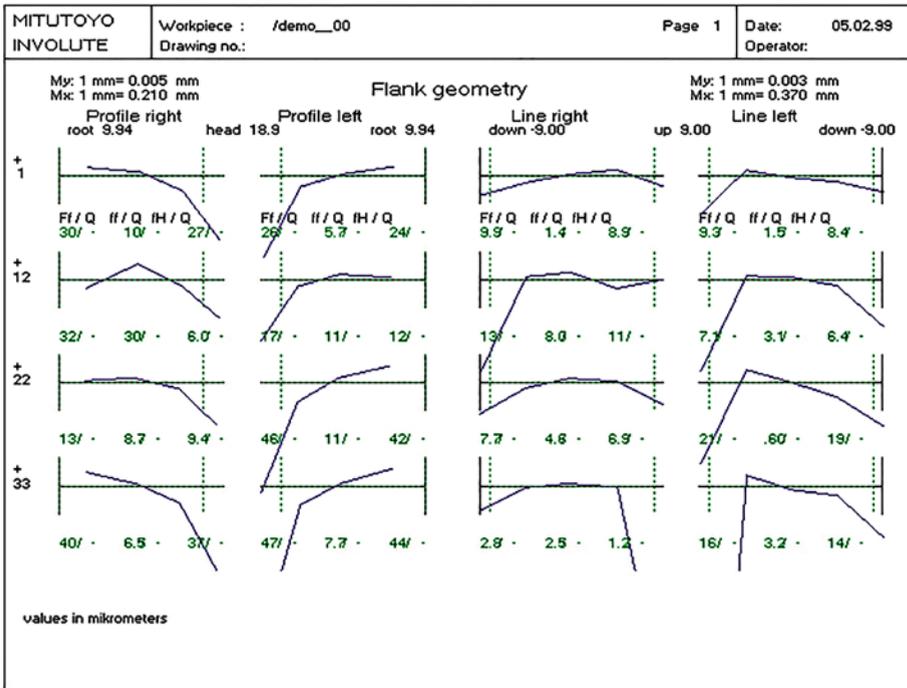
What all these standards have in common is that the grade or rating comes down to what kind of tolerance you'll have regarding the different gear parameters. And in most cases, it's not one tolerance. For example, for accuracies that are more open, you might end up with

multiples in a chart. This allows the gear manufacturer to have control of the geometric accuracy. It depends on whether the application for which the gear will be used calls for a gear that can run quietly or one that is more durable.

Speaking to AGMA standards, specifically, a gear manufacturer must pay close attention to the standard which the gear being produced was based upon. Translating that into machines, a standard CMM length measurement accuracy starts at about 1.7 microns. A CMM of this accuracy class is more than suitable for the inspection of "medium accuracy" and "low accuracy" gears ( $\approx$  A7-A11 for the AGMA 2015 standard and Q3-Q8 for AGMA 2000 standard



Depending on the probe-type, a CMM can provide a wide range of gear measurements, including gear-tooth profiles, faster than dedicated gear measurement machines while delivering needed accuracy.



**Better technology and software allow a CMM to measure tolerances and other parameters of a gear to determine its quality for specific uses where manufacturing accuracies are more important for different industries and applications.**

classes), but as you start to get into a Q9 accuracy grade or A2–A6 accuracy grades, it's time to look for even higher accuracy.

Here, a low-level machine CMM can have a baseline accuracy anywhere from 1.7 microns to 2.2 microns, depending on the probe system. A mid-range accuracy machine begins with a baseline 0.7 microns, which allows for the ability to measure a wider range of accuracy grades as you approach the upper limit of the AGMA threshold. The highest-end machine starts with a baseline accuracy 0.28 microns, and is a lab-grade machine that requires a temperature-, humidity-, and vibration-controlled environment in order to maintain this unparalleled high accuracy.

Which machine you select depends on how critical a gear component is. Is noise an issue? Will it be under a heavy load where teeth could potentially crack or fail? For instance, gears made in the powder metal industry typically only require an AGMA 2000 rating of Q6 or Q7, at most.

## Two Cases Where High Accuracy is Needed

We've been talking about standards and accuracy, and there are certain instances

when a high-accuracy gear measurement machine would be needed. First is the case of precision watches. These are very expensive, high-end timepieces with extremely small and intricate gears. Gear accuracy and precision are non-negotiable, while gear noise is unwanted.

Second is the case of high-end cars. In these vehicles, the transmission must be both precise and durable. And, as in

**“It's important that all data and graphs are reported in an easy-to-interpret, visual manner. Today's software options do this and more, generating a customizable table of contents around every parameter evaluated so that even someone who isn't a gear expert can interpret the results with relative ease.”**

the case of precision watches, noise is not wanted. In both examples, it's about reaching a balance between the application a gear is used for and the primary goals you want to achieve.

## Industries Where Gear Inspection Has Most Changed

Gear inspection has evolved in most industries, but most markedly in the aerospace and automotive industries.

In aerospace, planes now require high-efficiency engines and greater transmission-transfer capability. Gears within these mechanisms must function at all altitudes and all temperatures, from 110°F degrees in Arizona as low as -70°F at 35,000 feet. High gear accuracy and durability is required to withstand demanding use in all conditions, which has in turn affected and improved gear manufacturing methods, as well as the process of gear design itself.

In the automotive space, there was a sentiment that the rise in electric cars would make things simpler, gear-wise, without automatic transmissions and the gears involved in gas-powered cars. As it turns out, however, electric cars can be more complex from a gear perspective. In some cases, instead of one electric engine in a vehicle, there are four engines—one for each wheel. This is a dramatic change and requires a learning curve to figure out what gears produce the smoothest transmission of power to achieve the best driver experience while simultaneously requiring the least amount of maintenance.

## Why “Standard” Gears are Uncommon

When it comes to gears, gear measurement and gear technology, the number one thing you'll find is that you must always start off with a baseline. That baseline is usually derived from what would be considered a standard gear with standard parameters, but then those parameters are tweaked to make the gear optimal for its intended function, which

at a fundamental level is to transfer power from one point to another.

Given the parts advancements taking place in many industries, including automotive and aerospace, this means you'll very rarely find or use a standard gear anymore. Sure, standard gears are still made to adhere to AGMA grades and can be bought off the shelf. But because of the specialized nature of most parts today, those standard gears are mostly used as a jumping off point on the way to crafting a more unique and appropriate gear. It's no longer apples to apples. A gear that works on a jet engine to make it safer won't necessarily work on a jet turbo propeller.

### Reporting Functions to Look for in Gear Measurement Technology

There will always be different potential parameters and callouts that need to be evaluated based on the accuracy grade of the gear, as well as the intended use. For example, if a gear is going to be under heavy load, maybe it's the tip that gets shifted so teeth don't crack or break. There are modifications that can be applied for measurement, which you need the ability to evaluate.

Therefore, in terms of reporting functions, you may need to evaluate any of a multitude of parameters, including the total pitch or adjacent pitch between the teeth of the gear, profile and lead form error, tooth thickness and base tangent length, and tangential composite error, in addition to what's referred to as measurement over balls or wires and radial runout. The right technology allows you to do this virtually, based on data that's collected while measuring the gear.

You also want to be able to create graphs that show deviations of those measurements compared to what the measurement should be based upon the nominal values entered and the nominal shapes determined in the design phase.

It's important that all data and graphs are reported in an easy-to-interpret, visual manner. Today's software options do this and more, generating a customizable table of contents around every parameter evaluated so that even someone who isn't a gear expert can interpret the results with relative ease.

### The Future of Gear Measurement

The future of gear inspection and measurement will become even more automated.

The process of hand measuring will never completely disappear, especially in instances where accuracy isn't important or familiarity with "the way it's always been done" wins out.

However, momentum toward the use of multifunction measuring machines that can accommodate gear measurement tools and software will grow for two reasons: accuracy and cost. For the most part, those who measure gears across all industries require much



A CMM can measure many different types and sizes of gears such as the worm gear pictured.

greater accuracy and precision than hand tools allow. And, many companies are facing a labor shortage combined with tight budgets. They no longer have the time or the capital to devote to workers sitting and taking manual measurements. Workers need to be able to multi-task throughout the day.

In the end, it is most likely that the coming decades will be a time when the industry will push for ever-higher measurement quality, and it will be exciting to see what technology advancements develop as a result. 

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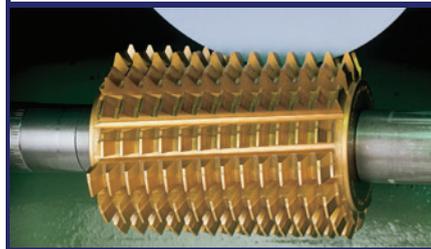


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# Quasi-Static Transmission Error Behavior Under the Composite Effect of Temperature and Load

Aitor Arana, Jon Larrañaga, Ibai Ulacia, Mikel Izquierdo, & Miren Larrañaga

## Introduction

Technological trends in automotive and aeronautical industries are pushing geared powertrains towards high rotational speeds with pitch line velocities approximating 100 m/s (Ref. 1), large gear ratios with low number of stages (Ref. 2) and minimum oil immersion depths (Ref. 3). At such conditions, thermo-mechanical issues are expected as bearings and gears are close to their thermal capacities.

Field experience in the turbomachinery industry has long proved that high-speed gearing is prone to thermal distortion effects. However, much of the knowledge remains restricted to the field of application and it is not available in the open literature. Seminal experimental work by Welch and Boron (Ref. 4) showed that in helical gear drives with relatively large face widths, the temperature of the teeth rises non-uniformly across the face width due to axial oil pumping. Longitudinal thermal gradients result in an uneven load distribution along the tooth trace and the authors pointed out that this behavior could be regenerative until tooth failure occurs. Subsequent works by Martinaglia (Ref. 5), Akazawa (Ref. 6), Matsumoto et al. (Ref. 7) and Amendola et al. (Ref. 8) prove such behavior and even numerically correlate thermal gradients to tooth root stress increase (Ref. 9).

Meanwhile, the number of evidences in small size gears is scarce and it is mostly concentrated on plastic gearing because thermal expansion coefficients are large and mechanical properties are temperature dependent. Wang (Ref. 10) concluded by finite element analysis of Nylon PA6 gears that because of temperature increase, the length of the contact path is expanded

with increased premature contact effects and multiple teeth in mesh. Later, Kashyap et al. (Ref. 11) experimentally analyzed the thermal expansion of Acetal spur gears and found that geometry change is mainly due to local pressure angle deviations that affect peak-to-peak transmission error at elevated temperatures. Even though references on thermally induced geometry distortion of steel gears are rare, recently, Hensel et al. (Ref. 12) have numerically shown that temperature increase can affect transmission error harmonic behavior when design contact ratios are close to an integer value. In the same direction, an experimental study by Luo and Li (Ref. 13) points out that temperature affects vibration amplitude of the system through thermally induced profile deviations, with temperature increases yielding larger vibration amplitudes.

In view of the lack of information on thermal distortion effects in small size steel gears, an experimental study of quasi-static transmission error behavior under thermomechanical conditions is carried out in this work. Composite effect of temperature and load on backlash, mean level of transmission error and its peak-to-peak value are experimentally studied and results are compared to analytical predictions.

## Methodology

In order to analyze the thermo-mechanical behavior of external cylindrical gears, a specific back-to-back test rig has been built based on the standard FZG machine architecture (Ref. 14). The main differences between the designed test bench and those available in the market are the speed and torque limitations, which have been overcome to study high power density

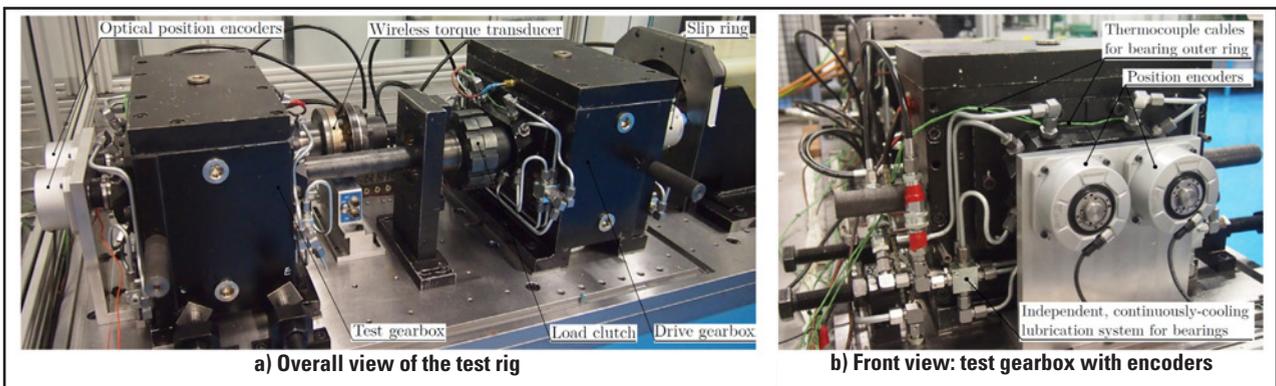


Figure 1 Designed test rig for gear thermomechanical behavior analysis.

transmissions. Furthermore, test and drive gear sets have been thermally isolated from their bearings, so that the thermal behavior of the gear pair can be studied independently from that of the bearings.

**Test rig description.** The designed test rig is shown (Fig. 1). It is composed of two equal gearboxes with inverse gear ratios (known as “test” and “drive”) following standard back-to-back configuration. Gearboxes are connected by two shafts; one of them being split in two parts with a load clutch in between for manual torque application with a lever arm. The traction motor rotates the main shaft so that the torque necessary to rotate the system is equal to the torque loss inside the loop. Two wireless, dual-range torque transducers located inside and outside the mechanical loop, measure the applied load and torque loss respectively.

In order to reach high circumferential speeds a liquid-cooled squirrel-cage induction AC motor for electric vehicles (EV) has been selected, reaching a maximum rotational speed of 10.000 rpm. Built-in sensors and position encoders allow accurate speed and torque control in the whole range. Moreover, in order to maximize tangential speeds, center distance has been increased up to 110 mm, so that the maximum attainable tangential velocity is 60 m/s, as larger diameters and gear ratios are allowed with respect to the standard FZG machine. Shafts and bearings have also been adapted and the maximum torque inside the loop now reaches 1000 Nm, which results in 1 MW maximum recirculating power.

When the test rig is subject to high speeds, considerable heating of the bearings arises which affects temperature distribution of the gear pair due to their proximity. To solve this issue, an independent lubrication circuit has been designed such that the oil from the bearings is physically separated from the gear oil sump by means of a specially designed housing. An ISO VG 46 oil is continuously pumped into each bearing housing through the lubrication pipes in Figure 1b at a minimum oil flow rate of 1 l/min. In order to avoid overflowing them, an additional suction pump has been installed which is driven by a servomotor and balances the oil level inside each housing by controlling the oil level in the tank with a digital level and a programmable logic controller (PLC). Meanwhile, gears inside the gearbox may be jet or dip-lubricated, and even oil-out conditions can be simulated without affecting bearing supports. In case of oil bath lubrication, the relative immersion depth is adjusted easily, and if necessary, oil is heated by means of high-power density thermoelectric resistances located inside a sealed aluminum plate submerged in the oil sump. A thermocouple immersed in the oil sump measures the temperature of the lubricant which is then heated by means of thermal resistances connected to a PID temperature controller.

The test rig includes several sensors to measure temperature, torque, speed, transmission error and oil condition. T-type thermocouples measure the temperature of different parts (bearings, housing, etc.) and a common data acquisition chassis

	Symbol	Set A	Set B
Normal module	$m_n$ [mm]	3	
Normal pressure angle	$\alpha_n$ [°]	20	
Number of teeth	$z_{1,2}$ [-]	37, 37	25, 50
Profile shift coefficient	$x_{1,2}$ [-]	-0.1608, -0.1608	-0.0234, -0.7337
Face width	$b$ [mm]	25	
Tip rounding	$h_k$ [mm]	0.6	
Tolerance field acc. DIN 3967	$E_s$ [-]	cd25	
Quality acc. ISO 1328	$Q$ [-]	5	
Reference tool acc. ISO 53	[-]	A (1.25/1.00/0.38)	
Material and treatment	[-]	17NiCrMo6, case-hardened	

	Symbol	Value
Input speed	$N_i$ [rpm]	60
Input torque	$T_i$ [Nm]	50, 100, 200, 400, 600
Immersion depth	$H/R$ [-]	0.5
Oil temperature	$\phi_o$ [°C]	50, 75, 100, 125, 150

synchronizes all signals. Temperatures of pinion and gear at several radial locations are measured and the rotating thermocouple signals are transmitted by means of a pair of slip rings. Moreover, each gearbox includes an oil condition sensor monitoring not only sump temperature, but water contamination and debris concentration as well — allowing for failure detection.

Finally, transmission error is measured with a pair of high-resolution optical angle encoders with  $\pm 2.5''$  accuracy, equivalent to  $\pm 0.6 \mu\text{m}$  for a 100 mm base diameter gear.

**Test specimens.** In this work, two spur gear sets are analyzed. Both are characterized by having a common 3 mm module, 25 mm face width and  $20^\circ$  pressure angle; while gear ratios are different: test set A is a 1:1 transmission while set B is 2:1. Additional information can be found in Table 1.

Note that the number of teeth is non-hunting and therefore each tooth will contact the same mate in the gear every time so that the composite manufacturing pitch and profile deviations will be constant for each mesh cycle. This allows to clearly identify thermal distortion effects; as variable composite tooth errors are not expected. Finally, both geometries have been manufactured with the same reference profile, material, quality and tooth thickness tolerance; the only difference being the profile shift coefficient which has been selected to balance specific sliding on each gear set. As a consequence, pinion and gear teeth geometry in set A are identical and those of set B are different due to dissimilar addendum modifications.

**Operating conditions.** The test program considers two steps: mechanical tests are carried out first and thermomechanical behavior is analyzed next. The former is used as a reference to analyze the effects of temperature increase on quasi-static transmission error. Table 2 summarizes working conditions for both sets.

Tests will be completed at 1 Hz constant rotational speed, which is considered sufficiently low to avoid introducing dynamic effects. Nevertheless, in order to guarantee quasi-static behavior, a preliminary dynamic study has been carried out and TE repeatability measurements have been completed. Besides, torque levels have been selected such that low and high unit loads are considered in combination with thermal effects. Nominal torque is 200 Nm and a total amount of five torque

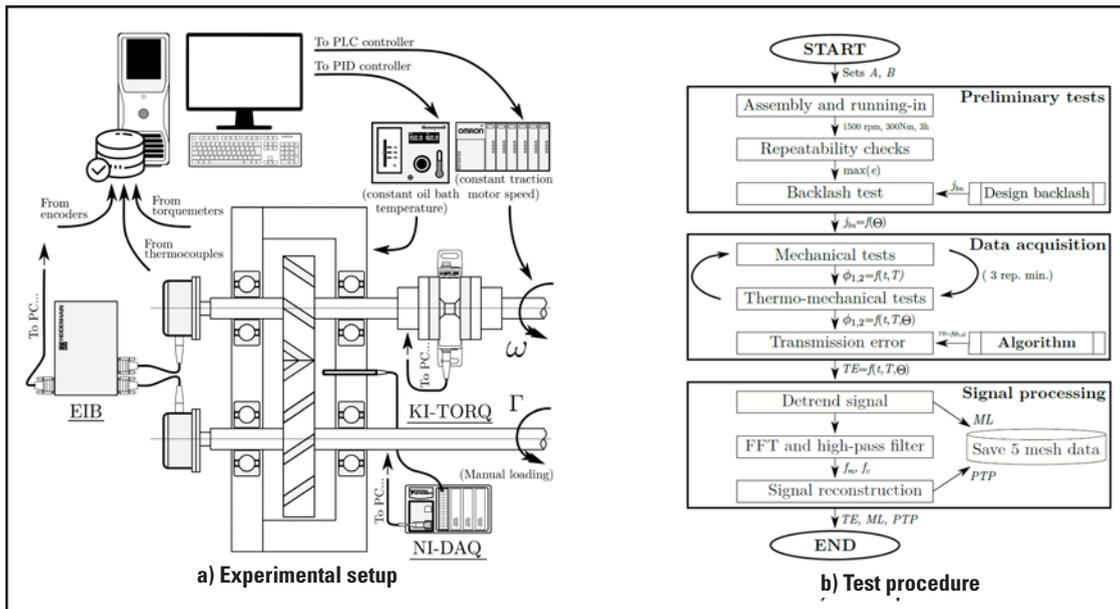


Figure 2 Experimental setup and procedure for thermo-mechanical quasi-static TE measurement.

levels is considered.

Finally, thermo-mechanical tests are completed by progressively heating the oil sump up to 150°C while rotating the gear pair at constant speed and prescribed torque, until the steady-state temperature is reached. A paraffinic mineral oil ISO VG 100 (Ref. 15) is used in the tests and the relative immersion depth is kept constant, just below the gear hub to ensure that radial temperature gradients are negligible.

**Experimental procedure.** The experimental setup and test procedure for the thermo-mechanical tests are graphically summarized in Figures 2a and 2b respectively. First, gears are mounted on the test and drive gearboxes and contact pattern is checked. If both gears are correctly assembled, lubricating oil is poured into the sump until the desired immersion depth is reached and torque level is adjusted with the lever arm. Once the exact value of torque is measured in the loop's torque transducer, bolts are tightened in the load clutch and the lever arm is removed. Then, motor speed is preset in the control software and prescribed value is sent to the PLC controller which turns on the traction motor. Simultaneously, the bearing-independent lubrication pumps are turned on along with the motor cooling system. Gears are run-in for at least three hours at 1,500 rpm and 300 Nm torque. Afterwards, encoders are mounted on the input and output shafts of the test gearbox and both are connected to the evaluation unit. Finally, gear thermocouples and slip ring are mounted on the drive gearbox which is connected to the data acquisition chassis. Both gearboxes being identical, temperature measurements in the drive gearbox can be extrapolated to test one.

Mechanical tests are carried out first at room temperature (20 ±1°C). Torque is preset to the first load stage of the test program and motor is rotated at constant 60 rpm rotational speed. Then, angular position measurements are conducted at steady-state rotational speed comprising a minimum of 20 full rotations. Then, the motor is stopped and next load stage is prescribed with the lever arm. The measurement procedure is repeated with each of the torque levels until the maximum load

is reached. Once the first load cycle is finished, the next cycle begins and the same steps are repeated from the lowest load stage to the highest one. A minimum of three repetitions are carried out in mechanical tests, each of them comprising several torque levels. Although no significant temperature increase is expected in these tests due to their short duration, component temperatures are monitored to ensure thermal effects are minimal.

The test program continues with the thermo-mechanical tests. The general procedure is kept but this time oil sump temperature is increased progressively in each load stage. Once torque value is preset, traction motor is rotated at constant speed and oil sump is heated with thermal resistances. Oil temperature is measured with a thermocouple immersed in the oil sump which sends instantaneous values to the PID controller and the acquisition system. The temperature of the oil bath, gear and housing is monitored and when steady-state condition is reached, angular position measurements are conducted following the procedure of the mechanical tests. Temperature is increased afterwards and when the next steady-state thermal stage is reached, measurements are completed in the same way. When the maximum temperature level for the considered load stage is attained, measurements are stopped until the whole system is cooled down. Then, next load stage is prescribed and the process is repeated. When all load-temperature combinations are finished, the process starts again until three full repetitions are completed.

Finally, when all tests in set A are finished, set B is tested following the same procedure. In between, additional tests such as no-load transmission error tests and backlash measurements are performed.

**TE measurement.** Transmission error (TE) is defined as the variation of the output rotational motion of the driven gear for constant rotational speed in the driver one due to elastic effects and clearances in the mesh. Therefore, TE is a relative magnitude relating angular positions of pinion and gear, which can be measured by the optical encoders. In terms of the length of the

line of action,  $TE$  can be computed following Equation 1 and the order of magnitude of its fluctuation at a specific torque is several microns for common gear sets.

$$TE = r_{b2} \cdot \Delta\theta_2 - r_{b1} \cdot \Delta\theta_1 \quad (1)$$

where  $r_{b1}$  and  $r_{b2}$  are the base diameters of pinion and gear respectively and  $\Delta\theta_1$  and  $\Delta\theta_2$  are the (1)

incremental angular positions relative to the starting position of the measurement. From Equation 1 one realizes that  $TE$  is always a negative parameter as the driven gear lags behind its theoretical conjugate position due to elastic deflections and backlash.

When transmission error is directly computed following Equation 1 the resulting curve presents a sinusoidal shape (Fig. 3a), characterized by a low frequency amplitude related to gear eccentricity and assembly errors and a high frequency cyclic variation due to time varying stiffness of the meshing teeth (Fig. 3d). In this work, thermomechanical effects are analyzed in the high frequency term which is related to tooth geometry and backlash. The influence of gear eccentricity is filtered out while maintaining the shape of the cyclic variation at mesh frequency with characteristic mean levels and peak to peak values. To this aim, a Fast Fourier Transform (FFT) is carried out followed by a high-pass filtering of the signal as shown (Fig. 3).

First, the original signal is detrended by an amount equal to the mean level of the initial measurement, so that the sinusoidal

curve is located on the abscissa as in Figure 3a. If the FFT is performed with the original signal, a big amplitude arises at 0 Hz masking small amplitudes of interest; hence the offset must be removed beforehand. However, the mean value of  $TE$  is preserved for signal reconstruction as it depends on the initial position of the gear pair, the applied load, temperature and available backlash.

Then, Fast Fourier Transform is computed in Figure 3b. Mesh frequency  $f_m$  in these tests corresponds to the number of teeth  $z$  as the shaft rotation frequency is 1Hz. Subsequent harmonics are located and  $N$  integer times the mesh frequency. Gear eccentricities to be filtered out are long-term errors (below mesh frequency) therefore, the high-pass filter must keep frequencies above the cut-off value  $f_c$  (in these tests  $f_c \cong \frac{1}{2} \cdot f_m$ ). Once the original signal has been filtered, it can be reconstructed by computing the inverse transform (see Figure 3c) and finally, a set of 5 mesh cycles is selected for analysis, mean peak-to-peak transmission error is measured and initial offset is added to keep the mean level as shown (Fig. 3d).

## Results

In the following section experimental results are summarized. Three types of measurements are shown: i) unloaded backlash tests at different temperatures, ii) loaded  $TE$  tests at room temperature and iii) full thermo-mechanical tests. In all cases, mean level of  $TE$  and peak-to-peak values are analyzed.

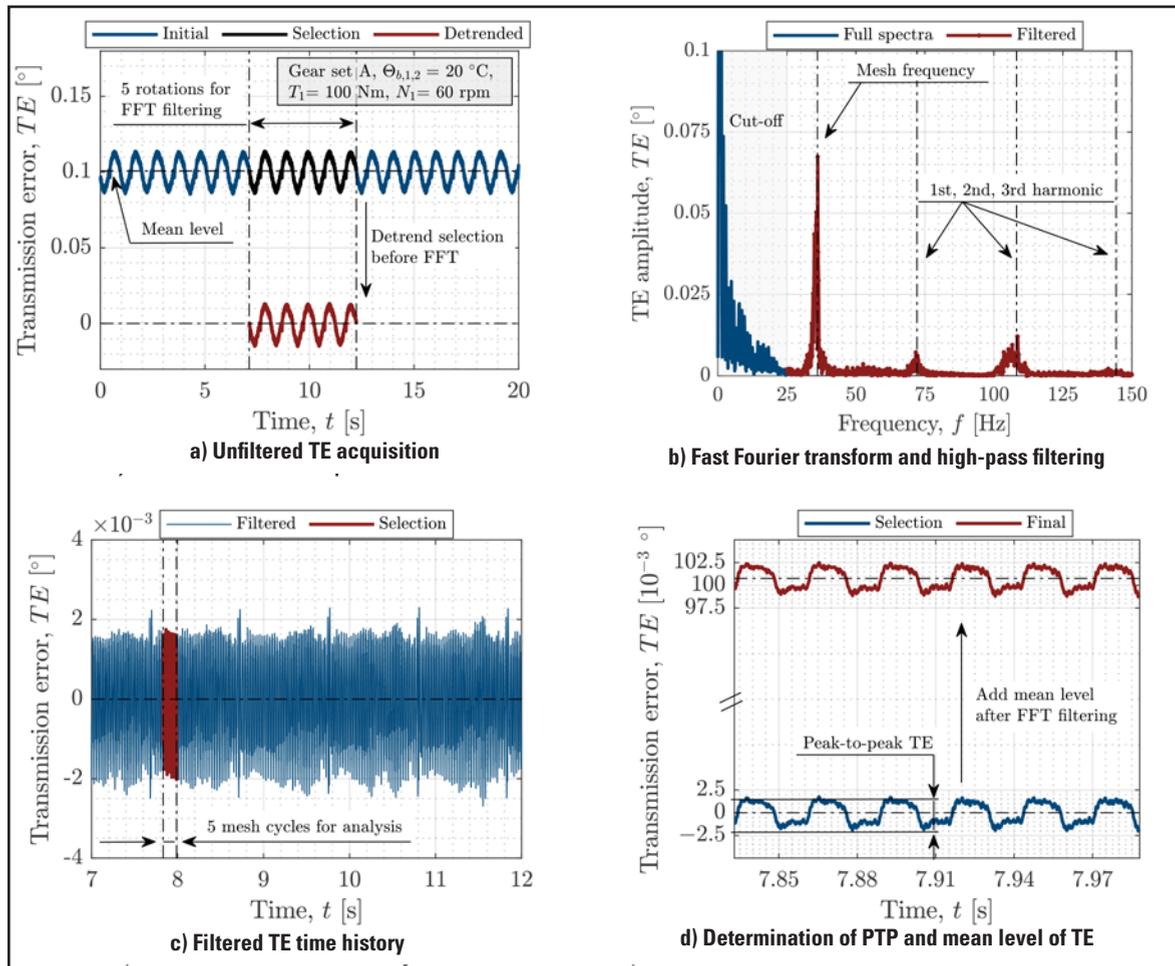


Figure 3 Sample TE measurement procedure in gear set A at 60 rpm and 100 Nm torque.

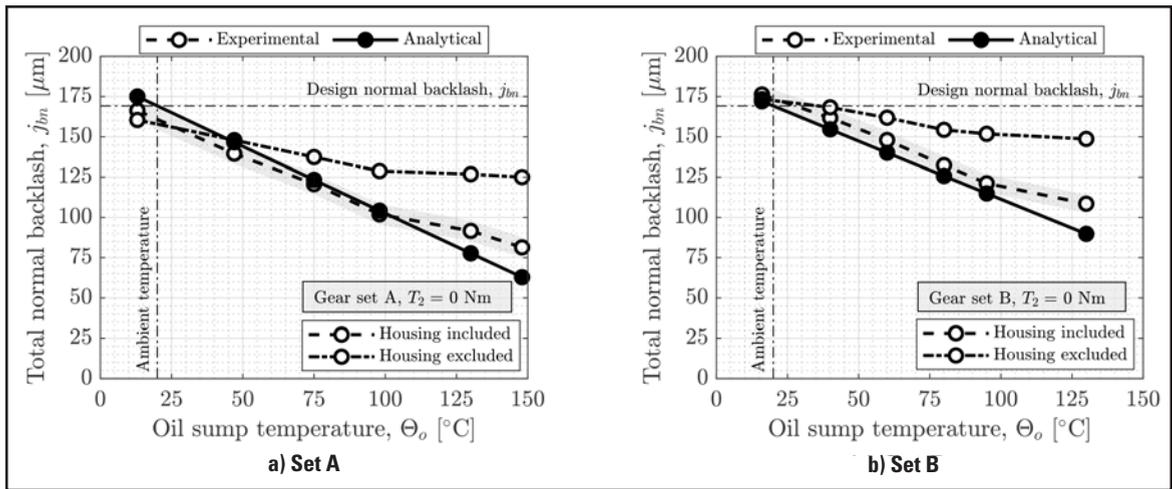


Figure 4 Temperature effect on total normal backlash reduction.

**Backlash tests.** Backlash tests are preliminary measurements to characterize the amount of available clearance at increasing oil sump temperatures. These tests are carried out before thermomechanical ones and they are used to validate the influence of temperature on backlash so that a correlation between this parameter and the mean level behavior can be established in future experiments. The driven gear position is fixed with the load clutch support while the pinion is rotated in clockwise and counter-clockwise directions until contact of the flanks occurs and loop torque signal is increased in the torque transducer. Several repetitions are completed in three different angular positions and constant oil sump temperature. The graphical representation of the measurements of both encoders shows the amount of available backlash and the mean value of the different measurements is used for comparison with the design backlash. Figure 4 depicts the influence of temperature on total normal backlash,  $j_{bn}$ , relative to the design value. Analytical predictions meet the design value at ambient temperature while increasing the oil temperature reduces the amount of available backlash linearly; which is the expected behavior if pinion and gear temperatures are assumed to be constant and equal to that of the oil sump. Meanwhile, experimental measurements follow the analytical prediction provided that the influence of the housing expansion is suppressed as shown by DIN 3967 standard (Ref. 16). Note that gear dilatation tends to reduce backlash while center distance expansion increases it. Hence, if the effect of the latter is not suppressed from the raw experimental measurement, it is not possible to analyze the influence of the gear expansion term.

Both gear sets show the same trend with temperature with analytical predictions following closely at least up to 100°C temperature. However, it is to be noted that the experimental

measurement slightly deviates from the analytical predictions due to several reasons. On the one hand, housing and gear manufacturing and assembly tolerances affect this correlation and on the other hand, theoretical linear thermal expansion coefficient for steel may deviate up to  $\pm 5 \cdot 10^{-7} \text{ K}^{-1}$  from its real value. Moreover, temperature differences may exist between the preset oil sump temperature and that of pinion and gear. Although manufacturing tolerances of the housing and thermal expansion coefficient deviations have been considered in the shaded error bar, the temperature differences are difficult to control, as it will be shown later in the discussion, especially at the highest temperatures where the largest deviations arise.

**Loaded TE tests.** Figure 5 summarizes loaded transmission error measurements for both gear sets. Analytical predictions according to the thin slice model presented in (Ref. 17) have been included to highlight the expected behavior.

If attention is paid to the TE diagrams in Figs. 5a and 5b, it is observed that load tends to increase both, transmission error mean level and peak-to-peak values; therefore, the gear is increasingly delayed with respect to its theoretical position. Reference position corresponding to half normal backlash and coincident with the no-load transmission error term is highlighted in all diagrams such that increasing separation from this position indicates that the backlash gap increases with load. Moreover, it is also remarked that peak-to-peak values increase with torque while the premature contact effect tends to contract the region of single tooth contact at higher loads. This behavior is consistent with scientific literature and it is also predicted by the analytical model.

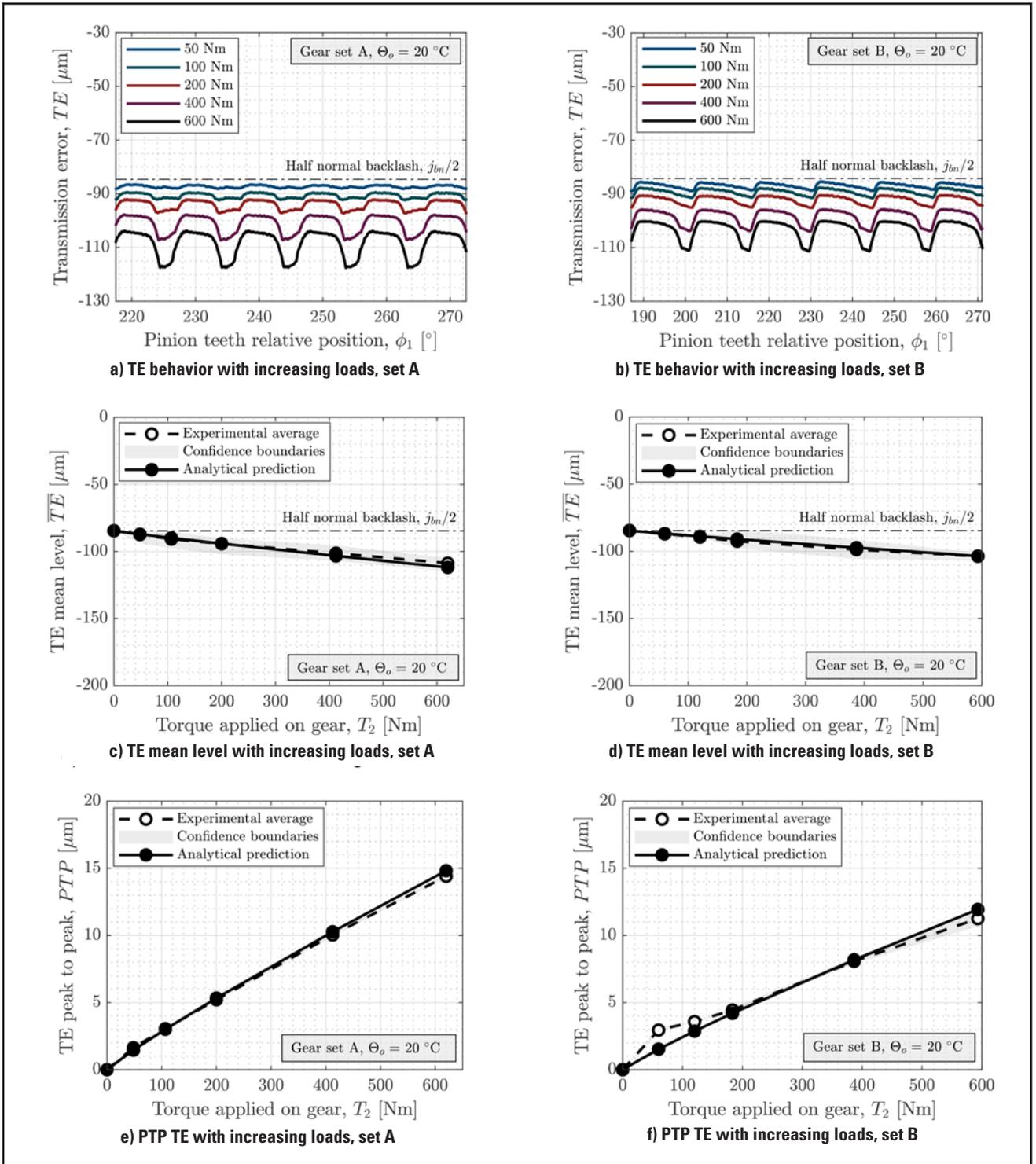


Figure 5 Experimental loaded TE measurements for gear sets A and B at 20°C temperature.

**Loaded and thermally affected TE tests.** Figure 6 shows the influence of increasing temperature at constant torque. The initial TE curve at ambient temperature is shifted because the backlash gap is reduced while the overall shape of TE remains unchanged. No apparent peak-to-peak variation is noticed at first sight and no premature contact effect is clearly visible from subfigures 6a and 6b. If the latter are compared to the loaded results, it is observed that the effect of temperature on mean level is greater than that of load which is confirmed by the slope

of the curve in subfigures 6c and 6d. Moreover, if the mean level behavior in these figures is compared to that of backlash in Figure 4 it is confirmed that the shift in mean level correlates to backlash change. Besides, it is interesting to remark that if pinion and gear temperatures are constant and equal to that of the oil sump, the analytical trend does not predict any significant change in peak-to-peak TE behavior. However, experimental results in subfigures 6e and 6f show increasing peak-to-peak values with temperature.

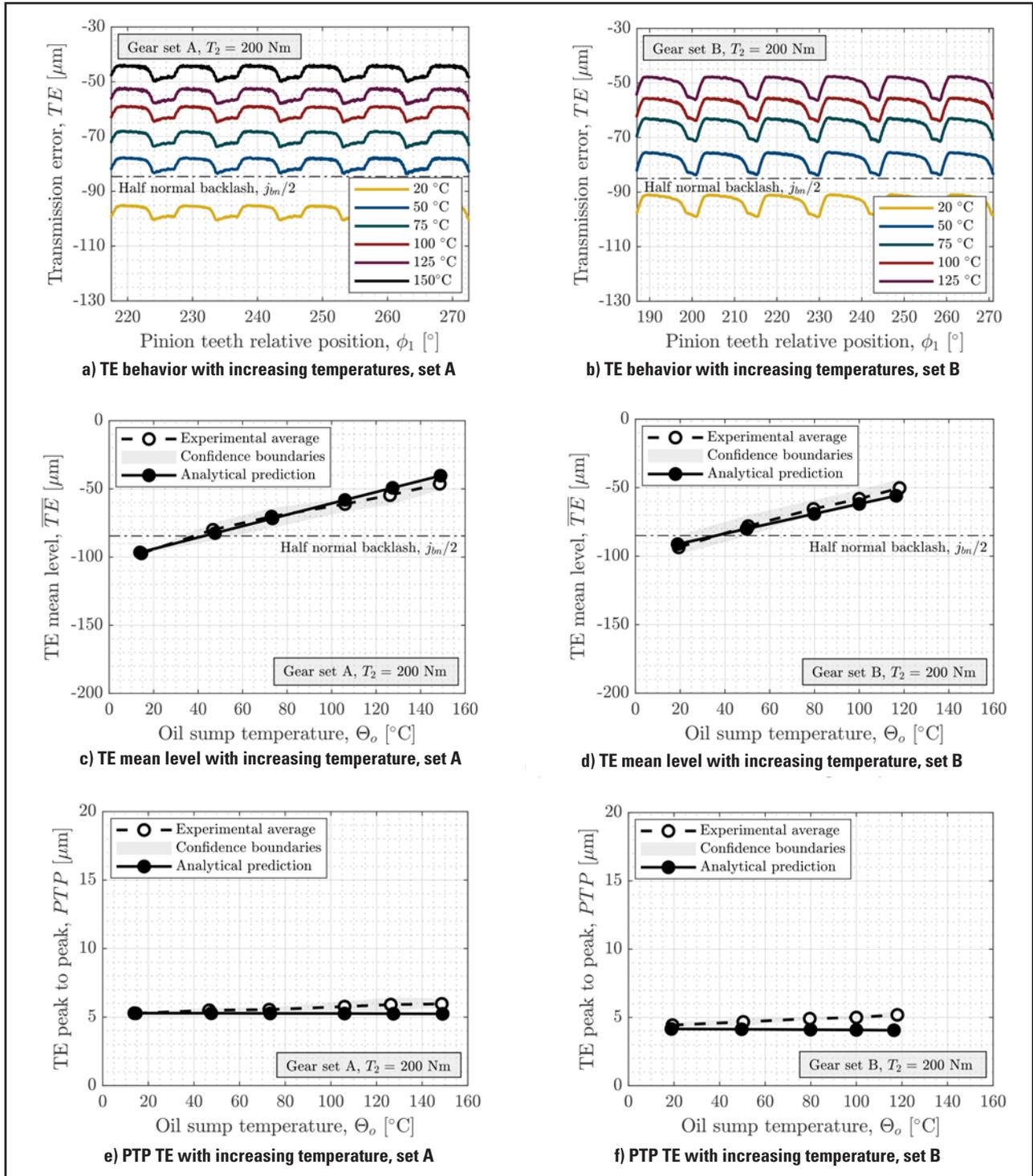


Figure 6 Experimental thermal TE results for gear sets A and B at 200 Nm torque.

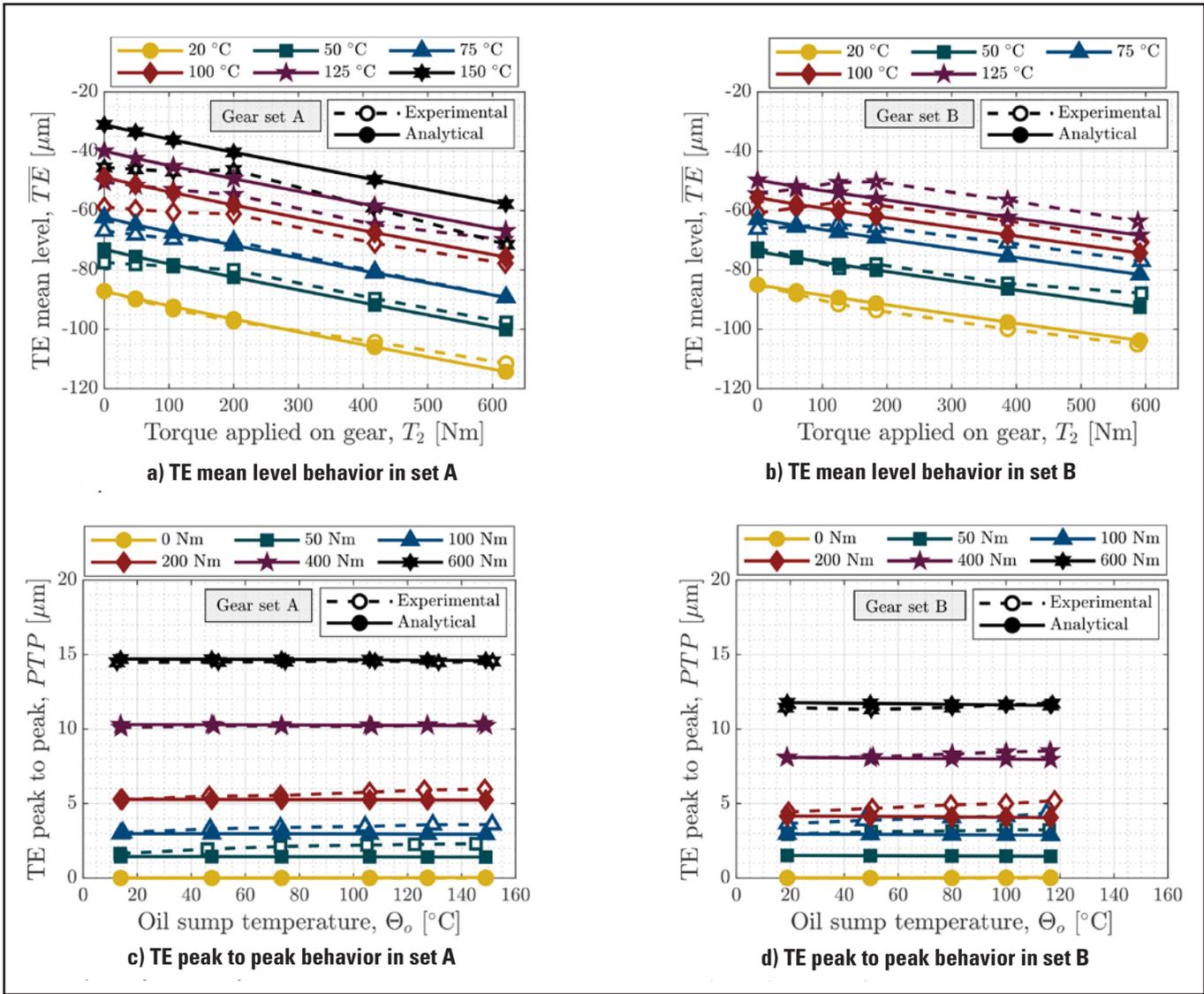


Figure 7 Experimental and analytical TE mean level and PTP comparison for variable temperatures and torques in both gear sets.

### Summary of Experimental Results

Finally, Figure 7 gathers all thermomechanical results in gear sets A and B. All torque and temperature combinations repeat the mean level and peak-to-peak patterns described in preceding figures. Increasing torque decreases TE mean level (stretches available backlash gap) while increasing temperature increases it (contracts available backlash gap); the influence of temperature on the latter being more prominent.

Meanwhile, peak-to-peak value is mostly influenced by torque and experimental results show that it slightly increases with temperature, which is not predicted by the analytical model where values remain almost constant. Furthermore, it is interesting to note that the effect of temperature on experimentally measured peak-to-peak seems to be more pronounced at low torque. Such differences specially arise at the highest temperatures and they may be explained by existing thermal gradients between components as shown later.

**Discussion**

**Constant temperature increase.** Figure 8 depicts the effect of diameter growth on spur gear teeth profile and mesh behavior. A radial expansion results in a pitch increase, local pressure angle deviation and normal backlash decrease.

If pinion and gear temperature increase are equal,  $\Delta\theta_{b1} = \Delta\theta_{b2}$ , no relative pitch deviation exists and profile inclination errors are equal in pinion and gear as predicted by Equations 2 and 3, respectively. Therefore, mesh behavior should not be affected by constant temperature increases and peak- to-peak TE is expected to be exclusively ruled by torque.

$$f_{pt} = p_{yt}' - p_{yt} = \tau(r_y' - r_y) 2 \cdot \pi z^{-1} u(r_y) = m_t \cdot \pi \cdot \alpha_L \cdot \Delta\theta_b \quad (2)$$

$$f_\alpha \approx -\Delta d_b \cdot [d_b \cdot \tan(\alpha_{yt})]^{-1} = -\alpha_L \cdot \Delta\theta_b [\tan(\alpha_{yt})]^{-1} \quad (3)$$

where  $m_t$  is the module,  $\alpha_L$  is the linear thermal expansion coefficient and  $\alpha_{yt}$  is the local pressure angle.

However, it can be analytically proved that a constant temperature increase does lead to a constant backlash reduction in the line of action and therefore TE curves are shifted as experimentally observed in Figures 6a and 6b. If Equation 1 is broken down in loaded and unloaded terms, we have:

$$TE = r_{b2} \cdot [\Delta\theta_{2,L} + \Delta\theta_{2,NL}] - r_{b1} \cdot [\Delta\theta_{1,L} + \Delta\theta_{1,NL}] = r_{b2} \cdot \Delta\theta_{2,L} - r_{b1} \cdot \Delta\theta_{1,L} + NLTE \quad (4)$$

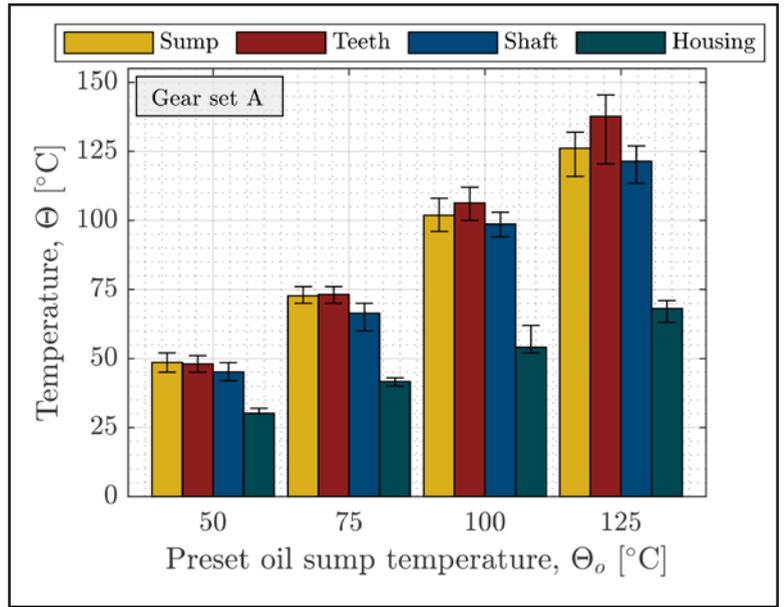
where  $NLTE$  is the no-load transmission error term representing composite geometry deviations,  $\Delta\epsilon_1 + \Delta\epsilon_2$ . Now, consider the following relation for profile thermal distortion in the base tangent plane:

$$\Delta\epsilon(\xi_y) \approx u(\xi_y) \cdot \sin(\alpha_{yt}) = [r_b \cdot \sqrt{1 + \xi_y^2} \cdot \Delta\theta_2 + \alpha_L] \cdot \sin[\arctan(\xi_y)] \quad (5)$$

where  $r_b$  is the base radius and  $\xi_y$  is the roll angle at any profile position. Applying the inverse trigonometric relation  $\sin[\arctan(\xi_y)] = \xi_y (1 + \xi_y^2)^{-1/2}$ , the preceding Equation 5 can be rewritten as:

$$\Delta\epsilon(\xi_y) \approx \Delta\theta_b \cdot \alpha_L \cdot r_b \cdot \xi_y = \alpha_L \cdot \Delta\theta_b \cdot \rho_y = \alpha_L \cdot \Delta\theta_b \overline{T_n P_y} \quad (6)$$

where  $T_n P_y$  is the distance between any point in the profile and the tangent to the base circle. Equation 6 describes a linear



**Figure 9 Gearbox temperature distribution relative to preset temperature in PID.**

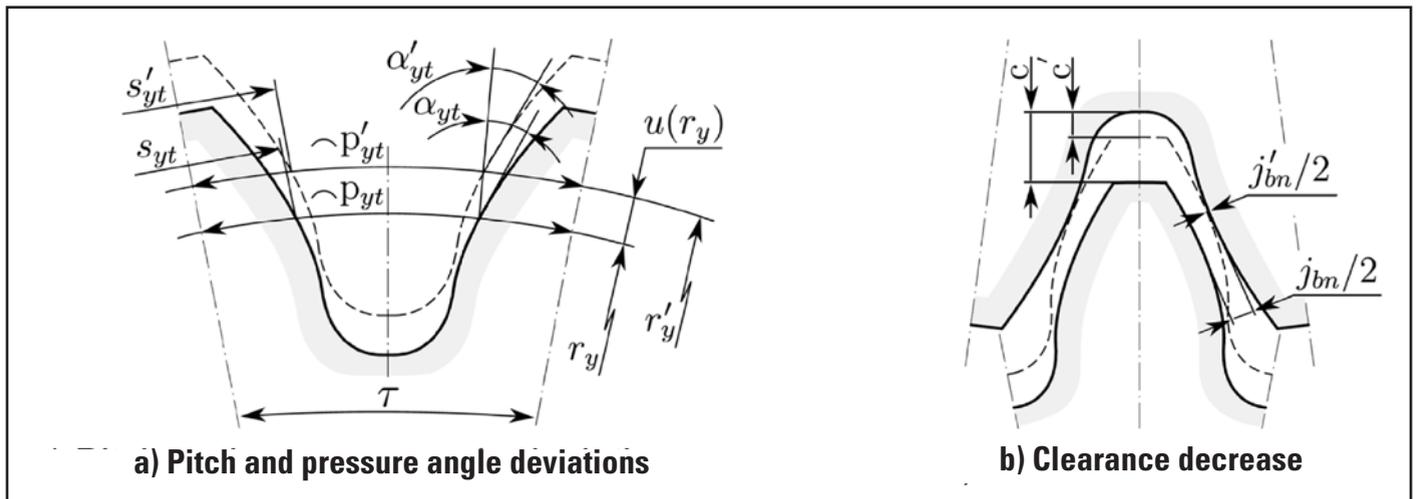
behavior of profile thermal distortion  $\Delta\epsilon$  with roll angle  $\xi_y$ , which is consistent with Equation 3. Computing backlash decrease for equal pinion and gear temperatures we have:

$$\Delta j_{bn}' = \Delta j_{bn} - \Delta\epsilon_H = \Delta j_{bn} - [\alpha_w \cdot \alpha_{L,H} \cdot \Delta\theta_H \cdot \sin(\alpha_{wt})] \quad (7)$$

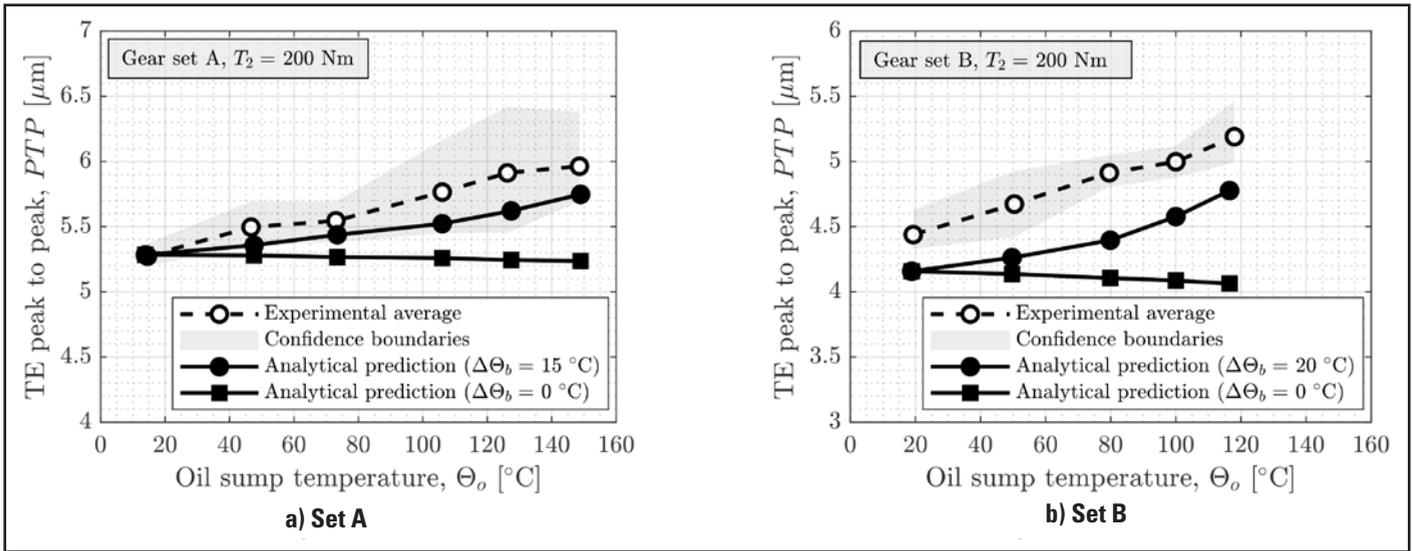
which is constant for gear pairs with the same material. This proves that under these conditions the  $NLTE$  term in Equation 4 is constant as well and the TE diagram is shifted with respect to the purely loaded behavior. If housing expansion  $\Delta\epsilon_H$  is considered together with Equation 7, the amount of total backlash change  $\Delta j_{bn}'$  is reduced but maintains the constant trend along the line of action observed experimentally.

$$\Delta j_{bn}' = \Delta j_{bn} - \Delta\epsilon_H = \Delta j_{bn} - [\alpha_w \cdot \alpha_{L,H} \cdot \Delta\theta_H \cdot \sin(\alpha_{wt})] \quad (8)$$

**Thermal gradients.** Equations 4 to 8 prove that an equal temperature increase in pinion and gear does not affect the TE curve shape because relative profile deviations are constant. Only backlash (i.e. mean level of TE) is reduced. However, it has been observed that peak to peak TE slightly increases in almost



**Figure 8 Effects of temperature increase on profile geometry and mesh behavior.**



**Figure 10** Influence of thermal gradient on peak-to-peak transmission error.

all test cases at the highest temperatures, which is not followed by analytical predictions.

Figure 9 summarizes measured mean temperatures and corresponding deviations of different parts of the gearbox for each temperature stage. At oil bath temperatures below 100°C, the steady state sump temperature is almost equal to the preset value and no significant temperature difference exists between the oil sump and the gear teeth. However, at higher temperatures, the standard deviation increases significantly indicating that the PID control hardly maintains oil sump temperature. Moreover, gear teeth temperature is higher than that of the oil bath, probably due to friction coefficient increase which progressively heats gear teeth above the oil bath temperature due to surface sliding.

In cases where the tooth temperature is higher than that of the oil, the available backlash should be reduced more than expected according to Equation 7. Therefore, experimentally measured mean level of transmission error in Figure 7a should overcome the predicted value, but this is not the case. It is to be noted in Figure 9 that at high temperatures not only thermal gradients arise between the oil sump and the gear teeth but also between the teeth and the shaft. Consequently, radial thermal gradients prevent maximum tooth deformations and corresponding backlash reduction. The final mean level depends on the exact temperature distribution for each case. Moreover, the existence of temperature differences between pinion and gear also explains the increasing peak-to-peak TE with temperature. Figure 10 shows the influence of thermal gradients between pinion and gear on both gear sets. As expected, peak-to-peak transmission error computed including temperature differences approaches the experimental result.

## Conclusions

In this work an experimental study of thermo-mechanical quasi-static transmission error behavior has been developed. Scientific literature review presented in the introductory paragraphs has shown that no experimental evidence on the composite effect of temperature and torque on transmission error exists up to date. Then, a novel back-to-back test rig for high-speed gears has been described, experimental methodology has been presented and finally, measurement results have been summarized. Tests have been conducted at low rotational speed, constant torque and constant oil sump temperatures such that pinion and gear teeth temperature is assumed to be equal to that of the oil bath. Overall results show that the effect of temperature and torque coexist in TE diagrams. Both parameters have a significant role in the mean level of transmission error while the influence of torque on peak-to-peak is prominent relative to that of temperature. Furthermore, it has been shown that temperature increase reduces the amount of available backlash and therefore mean level of transmission error is affected. Both parameters have been shown to be correlated and an analytical proof has been provided. However, the analytical model has not predicted the experimentally observed variation of peak-to-peak TE with increasing temperature and discrepancies have been attributed to thermal gradients between components. ⚙️

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# Drive Concepts using Super Reduction Hypoids Combined with Cylindrical Gear Reductions

Dr. Hermann J. Stadtfeld

## Why Transmissions in Electric Vehicles?

Compact electric vehicles require a cost-effective and compact solution for the location of the electric motor and the transmission. Yes, even today electric vehicles require a transmission, if the maximal possible motor efficiency has to be available in the majority of drive conditions. Transmissions also allow higher motor rpm, resulting in smaller size electric motors with improved dynamic properties. The torque and efficiency optimal rpm of an 80 kw electric motor for a compact vehicle is between 6,000 and 10,000 rpm. For example if the nominal driving speed is 80 km/h and the optimal motor speed is 10,000 rpm, the optimal ratio between motor and wheels (using a wheel diameter of 410 mm) is 9.66:

$$i = (n_{Motor} \cdot D \cdot \pi) / v$$

Whereas:

- $i$  Transmission ratio
- $n_{Motor}$  Optimal motor rpm [1/min]
- $D$  Outer tire diameter [m]
- $v$  Average vehicle speed [m/min]

A variety of eDrive concepts have been developed during the past years. One example which has already been introduced in Chapter 1 is the design shown in Figure 1 (Ref. 1). It is very compact, but like most concepts, it does not solve the three major obstacles of the “inline design,” which are:

- Large width between the front wheels used for drive unit
- Asymmetric weight distribution
- Higher heat radiation to the wheel and tire on the side of the electric motor

The large width between the wheels requires short drive shafts. Each of the drive shafts has two CV-joints which wear fast in the case of short drive shafts due to the steering inclination and control arm swings. This will also result in a reduced efficiency and front axle noise.

The asymmetric weight distribution

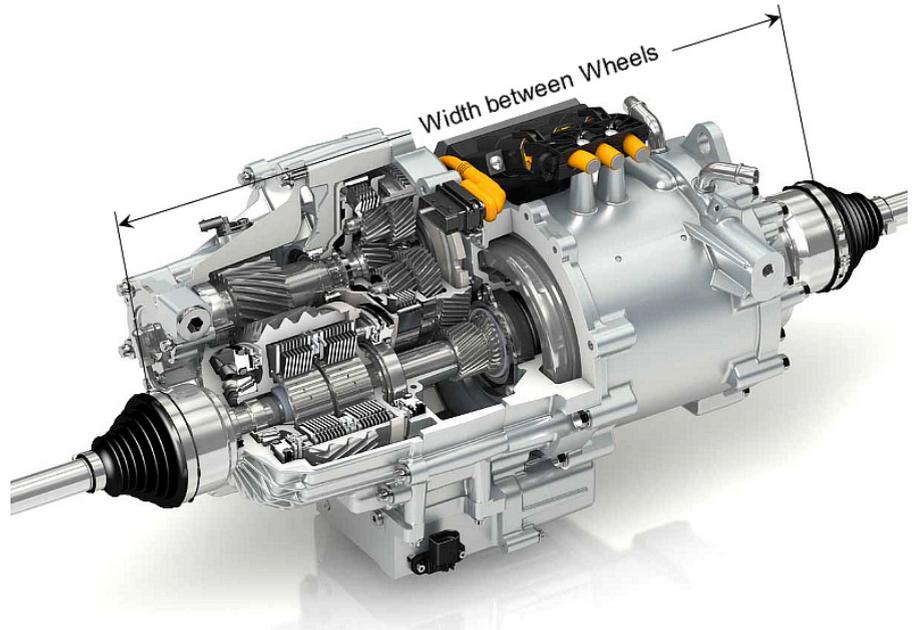


Figure 1 Electric motor in line with transmission and front wheels (Source [1]).



Figure 2 Hypoid gearset with a ratio of 7 × 55.

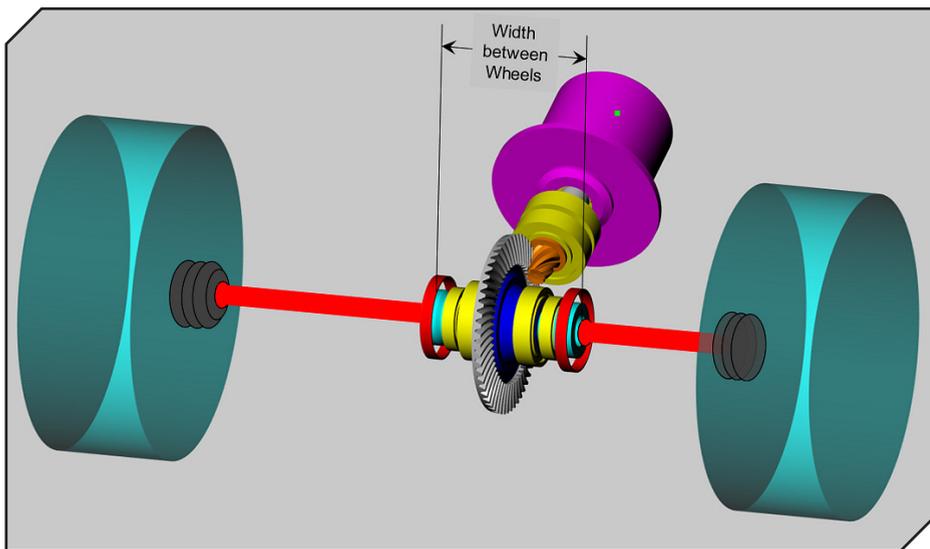
has to be offset with other asymmetric vehicle components such as the battery. However, there will still be an influence on the dynamic behavior of the vehicle.

The permanent heat radiation of the electric motor in Figure 1 might increase the temperature of the adjacent tire by 10 to 20°C. Thermal insulation and an additional cooling fan can reduce the temperature of the hot tire to the e-motor but the consumption of electrical energy for the evacuation of motor heat is not something EV-manufacturers like to see.

### Transmissions Establishing Symmetry in the Drivetrain

Gleason suggested the possibility of a high reduction hypoid gearset as shown in Figure 2. The ratio of the gearset in Figure 2 is 7.85 ( $7 \times 55$ ), which is high for an automotive transmission, but appears rather low for a one stage eDrive reduction. The objective of this application is to rotate the electric motor by 90° which would establish a symmetric eDrive unit by accomplishing the required ratio between motor and wheels solely with this one single reduction stage. The gearset in Figure 2 is designed as a regular hypoid with the *Gleason Gear Engineering and Manufacturing System GEMS*. It is not a high reduction hypoid (HRH) or a Super Reduction Hypoid (SRH). The pinion has seven teeth and the ring gear has 55 teeth. This first design was the attempt to do a moderate and predictable step to achieve the major objectives of a single stage eDrive reduction.

Any eDrive reductions require high efficiency as well as a good back driving ability. The back driving is important in two ways. The first reason is the regeneration of electrical energy in case the vehicle driver takes the foot off the accelerator pedal. The electric motor is switched to generator operation and the kinetic energy of the vehicle is used to re-charge the battery rather than being wasted by simply using the brakes. The second reason for the back driving ability is to avoid wheel locking in case of an abrupt release of the accelerator pedal. The gearset in Figure 2 fulfills both requirements very well. The ring gear is phosphatized in order to increase the efficiency of the gearset before break-in and avoid costly polishing. As the phosphor layer breaks down, the break-in of the gearset is



**Figure 3** eDrive unit with single stage hypoid reduction, shown without housing between the front wheels of a small sedan [2].

finished and the initial efficiency will be maintained. Both members have been ground after heat treatment.

The gearbox design, which accommodates the hypoid reduction and the new orientation of the electric motor was designed and optimized with the Gleason *KISSsoft* development and optimization system. The result of this development is shown in Figure 3 (Ref. 2).

The eDrive unit in Figure 3 has the highest degree of symmetry and moves the heat radiating electric motor away from the tire it was exposed to with the inline design of Figure 1. The distance between the drive shaft flanges presents a very small “width between wheels,” which allows for rather long drive shafts. With the possibility to face the motor either towards the front or towards the back, the ideal weight distribution and optimal packaging for a particular vehicle can be achieved. This very compact design with only two gears and two shafts can be manufactured very cost-effectively and presents a very good eDrive solution for a small compact electric vehicle. The remaining question is the possibility to realize even higher ratios with a single stage hypoid gearset. Also the question of whether the concept presented in Figure 3 could be extended to a combination of a hypoid and a cylindrical reduction and still preserve the basic advantages mentioned above, has to be analyzed.

### The Super Reduction Hypoid Solution

In order to increase the ratio of the hypoid gearset in Figure 3, the conventional hypoid calculation appears to be unsuitable. Several sample calculations, using the SRH design applet, delivered very good results (Ref. 3). In the sample calculations, the pinions had 4, 5 and 6 teeth. SRH creates a face milling duplex design which can be optimized regarding tooth depth, pinion diameter and face angles specifically to the requirements of an eDrive.

One point of attention is the maximal sliding velocity generated by the hypoid offset of the pinion. During the SRH analysis, the sliding velocity is calculated. Hypoid gearsets with an offset as used in automotive and truck applications have about 125 m/min relative sliding velocity when the vehicle is driving at a speed of 100km/h (62.5 mph). An eDrive hypoid with a ratio of 9.66 generates at a motor speed of 10,000 rpm a relative sliding velocity of 333 m/min.

This is more than twice the relative surface sliding of the conventional hypoid gearset. It will be required to use high-pressure synthetic hypoid oil. It also was discussed whether surface coatings are required in order to achieve the necessary gear life with respect to surface damages.

The number of teeth is not the only indicator of the back driving ability of a hypoid gearset; it is also the pinion spiral angle. The larger the spiral angle the

lower the back driving ability becomes. The following categories of spiral angles are defined:

- Small spiral angle: 0° to 20°
- Medium spiral angle: 20° to 35°
- Large spiral angle: 35° to 65°
- Very large spiral angle: 65° to 90°

Two examples — one with a large spiral angle and one with a very large spiral angle are shown in Figure 4. A certain number of teeth on a small diameter results in a lower spiral angle than the same number of teeth on a large diameter. In order to take the opposing effects into account, the UNICAL dimension sheet program (Ref. 3) calculated the back driving ability by considering the correct geometry and an assumed coefficient of friction of 0.08. The program calculates the back driving opposing force  $T_{br}$  and divides it by the back driving force  $T_{dr}$ . The fraction  $T_{br}/T_{dr}$  called the Back Driving Coefficient  $C_{BD}$  (Ref. 2). A value of  $C_{BD} = 1.0$  and above indicates a self-locking condition. A value of  $C_{BD} = 0$  would be ideal but cannot be achieved because it would require the absence of any friction losses. In Table 1 the results of  $C_{BD}$  for five different real hypoid gearsets are listed.

While the 17-tooth automotive hypoid pinion has a  $C_{BD}$  of 0.091, which is excellent, the 1-tooth pinion example is self-locking with a  $C_{BD}$  of 1.295. The 2- to 5-tooth pinions have very similar coefficients, with the unexpected low coefficient of the 3- tooth pinion, which

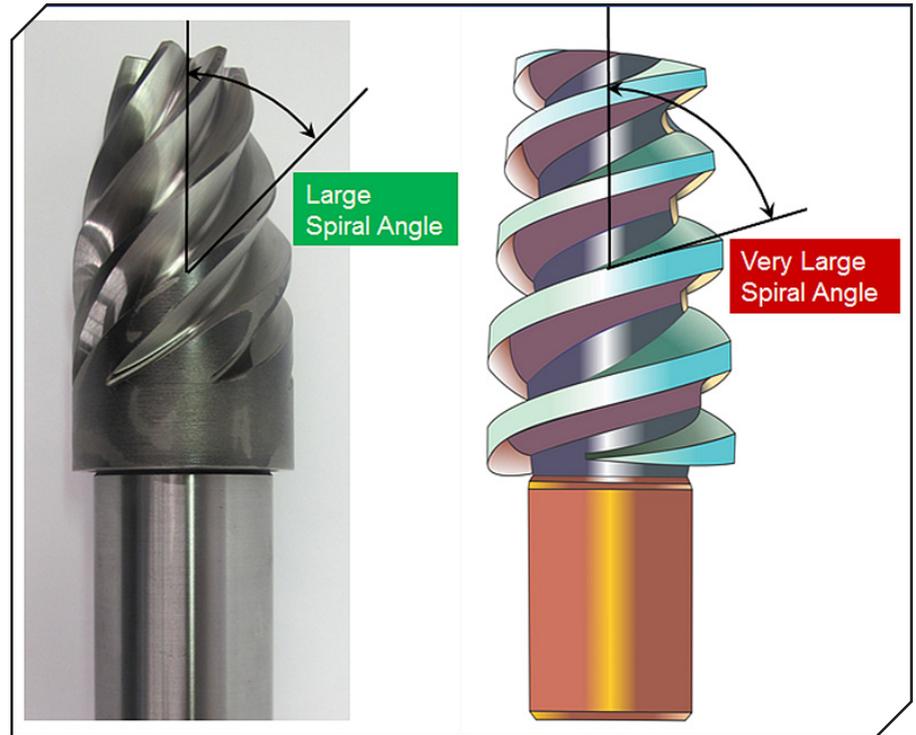


Figure 4 Large pinion spiral angle (left) and very large spiral angle (right).

is lower than the 5-tooth pinion. This shows that optimizing the right parameters will allow reducing the pinion tooth count to 3 in order to still achieve a reasonably good back driving ability. The values in Table 1 are part of the SRH Dimension Sheet output (Ref. 3).

If an ideal ratio for an eDrive hypoid reduction is in the range of 15, then it seems realistic to select 4 pinion teeth and 60 ring gear teeth (better 59 or 61 because of hunting tooth advantage). Such a SRH gearset should be optimized with the goal to achieve a back driving

coefficient  $C_{BD} = 0.3$  or below.

The SRH system has several advantages versus HRH or regular hypoid gears. It is possible for a given ring gear spiral angle to search for the lowest spiral angle difference between pinion and gear. This is possible by changing the pinion diameter and the offset for given gear diameter and spiral angle. The pitch angle of the gear can be adjusted in order to achieve optimal roll conditions. It also proved to be advantageous for eDrive SRHs to increase the depth factor to 30% above the standard tooth depth. Pressure angles can be entered asymmetrically into the SRH calculation to assure the desired contact ratio balance between drive and coast side.

Table 1 Back driving analysis results for 5 different hypoid gearsets.

Number of Pinion Teeth	Coefficient of Friction	Back Driving Coefficient $C_{BD}$	Condition of Back Driving
17	$\mu = 0.08$	$T_{br}/T_{dr} = 0.091$	Not Self Locking
5	$\mu = 0.08$	$T_{br}/T_{dr} = 0.324$	Not Self Locking
3	$\mu = 0.08$	$T_{br}/T_{dr} = 0.305$	Not self Locking
2	$\mu = 0.08$	$T_{br}/T_{dr} = 0.336$	Not Self Locking
1	$\mu = 0.08$	$T_{br}/T_{dr} = 1.295$	Self Locking

### Different Hypoid Transmission Types

Compact and low-priced electric vehicles require a simple, compact and cost-effective transmission between electric motor and front or rear wheels. Small compact vehicles with eDrive mostly do not require top speeds above 90km/h. Their major application is inner city driving for commuting or shopping. All important objectives for such a vehicle can be fulfilled with a single-stage hypoid transmission, as shown in Figure 5.

The small width between the wheels makes for a slick compact vehicle

solution which allows optimal packaging and an ideal vehicle weight distribution. The ratio can be below 12 and so a second gear reduction which can serve to adjust the vehicle speed more optimally to the eMotor rpm is not really required. The transmission in Figure 5 has a ratio of 7.85 ( $7 \times 55$ ) and a back driving coefficient of about 0.3, which is acceptable for energy recuperation during coast operation.

A second eDrive transmission concept is shown in Figure 6. This concept is a dual-stage reduction with a first cylindrical reduction of 2.33 ( $21 \times 49$ ) and a second hypoid reduction of 4.4 ( $11 \times 51$ ), which results in an overall ratio of 10.27. The driving efficiency of the dual-stage transmission will be higher than the transmission in Figure 5 and the back driving coefficient of about 0.15 is also better than the transmission in Figure 5.

Midsize or premium electric vehicles would benefit from the transmission concept in Figure 7. An electromagnetically actuated clutch unit can activate either a ratio of 1.46 ( $26 \times 38$ ) or a ratio of 3 ( $16 \times 48$ ). The hypoid reduction has a ratio of 3.85 ( $13 \times 50$ ). In forward driving with lower speeds (e.g., up to 50 km/h (31.3 mph)) the overall ratio can be switched to  $3 \cdot 3.85 = 11.55$ . As the speed increases above 50 km/h (31.3 mph), the eMotor will operate with less efficiency and the second cylindrical gear pair can be activated. Now the overall ratio changes to  $1.46 \cdot 3.85 = 5.62$ , which enables the motor to reduce its rpm to a more efficient operation.

The transmission in Figure 7 becomes very interesting in coast operation when the motor is switched to generator mode to utilize the kinetic energy of the vehicle to recharge the battery. For example, if the speed is 80 km/h (50 mph) with a transmission ratio of 5.62, the transmission will switch within some milliseconds to the higher ratio of 11.55 in order to give the generator (motor) more rpm for a more efficient electricity generation. Although this concept does not benefit from the originally discussed single-stage transmission, its flexibility makes it a very attractive solution. The advantages like packaging, good weight distribution as well as heat radiation away from one of the wheels are all maintained.

In order to complete the possibilities of

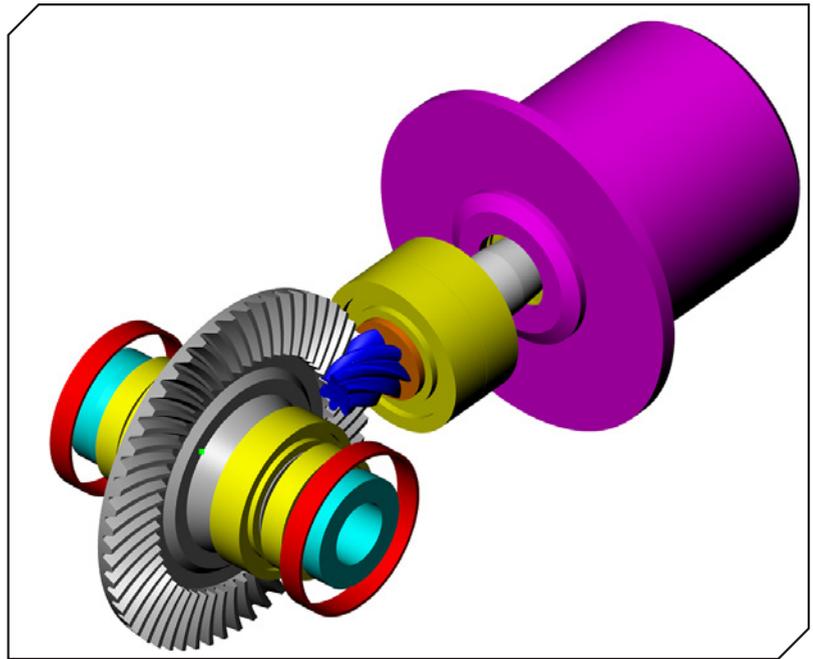


Figure 5 Single stage hypoid transmission – Design 1.

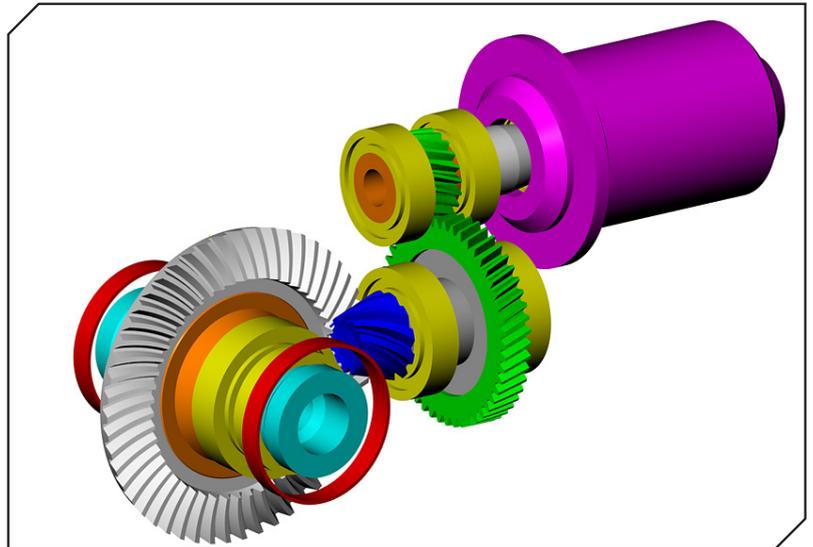


Figure 6 Dual stage cylindrical-Hypoid transmission – Design 2.

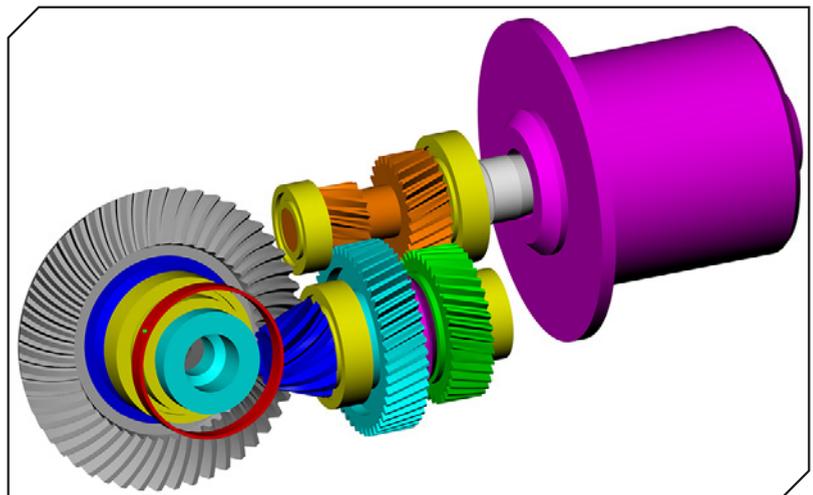


Figure 7 Dual speed cylindrical-hypoid transmission – Design 3.

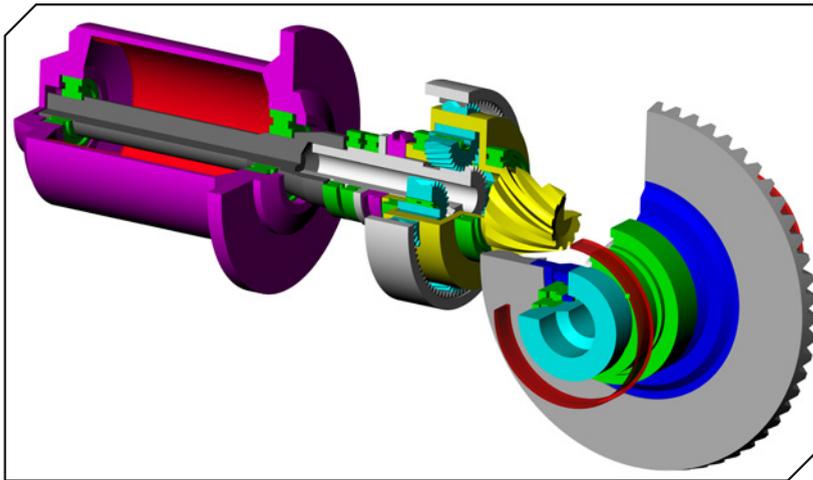


Figure 8 Planetary-hypoid reduction – Design 4.

combined transmission solutions, a planetary concept design was also developed, which is shown in Figure 8. The motor shaft is connected to the sun gear and the hypoid pinion is connected to the cage as planetary output. With the planets having the same number of teeth as the sun gear (29) and the internal ring having three times the number of teeth of the sun gear (87), two ratios are possible. The planetary transmission requires a clutch which can connect the internal ring gear to either the sun gear (ratio 1.0, 1<sup>st</sup> gear) or to the transmission housing (ratio 4.0, 2<sup>nd</sup> gear). Because one of the two possible ratios is always 1.0, the flexibility of the planetary transmission is lower than the dual-reduction cylindrical-hypoid version. In Figure 8, the hypoid ratio is 5.18 (11×57). The overall ratio in 1<sup>st</sup> gear is 1.0×5.18=5.18 and in 2<sup>nd</sup> gear 4.0×5.18=20.72.

### Motor Speed Versus Vehicle Speed

In the case of the single-speed transmission the relationship between motor rpm and vehicle speed is of course proportional, as shown in Figure 9. This applies if the electrical prime mover is in either motor or generator mode. The efficiency optimal operating rpm of a motor which is rated for 15,000 rpm maximum is in the vicinity of 6,000 to 10,000 rpm. This is the speed range with high efficiency in case of average load. If the load is small, then the efficiency optimal rpm is lower, and vice versa if the load is higher. The ratio of a single-stage transmission has to be defined such that the majority of the driving falls into the optimal efficiency range (see blue range in Figure 9). Figure 9 also indicates that the red speed increasing graph is only for a short period within the optimal efficiency range. In case of taking the foot off the accelerator pedal, the motor-generator control can disconnect the motor from the current flow and the vehicle coasts down naturally. If the brake pedal is applied, then the control can switch the motor to generator mode while the disk brakes are not yet engaged. Only when the brake pedal is pressed hard will the disk brakes kick in and support the electric generator brake.

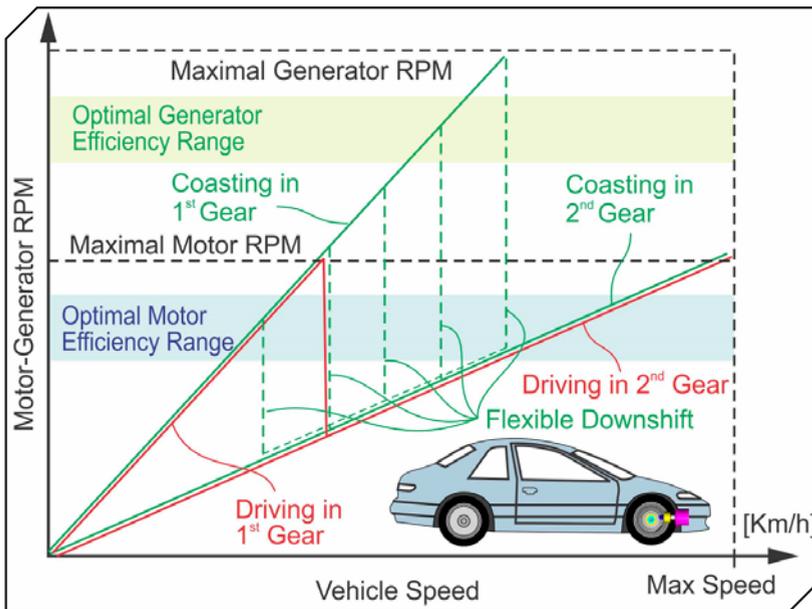


Figure 9 Speed diagram for single speed transmission.

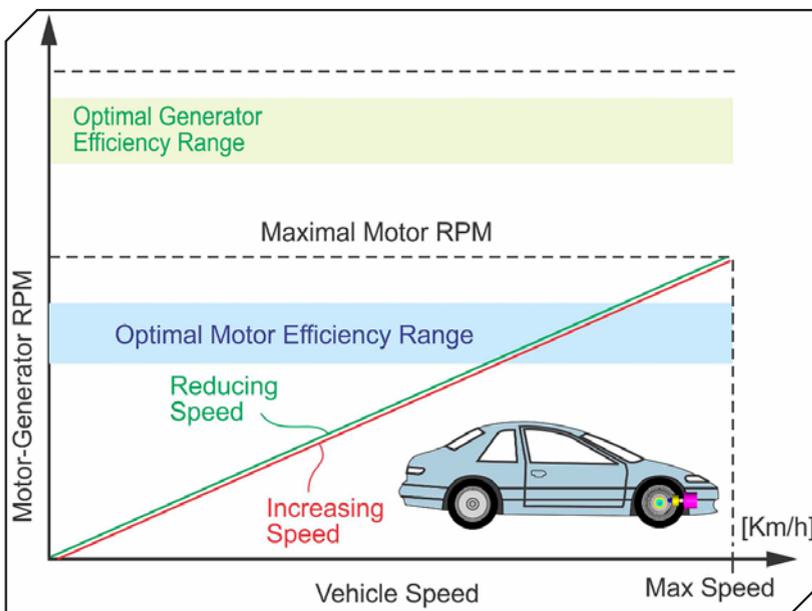


Figure 10 Speed diagram for dual speed transmission.

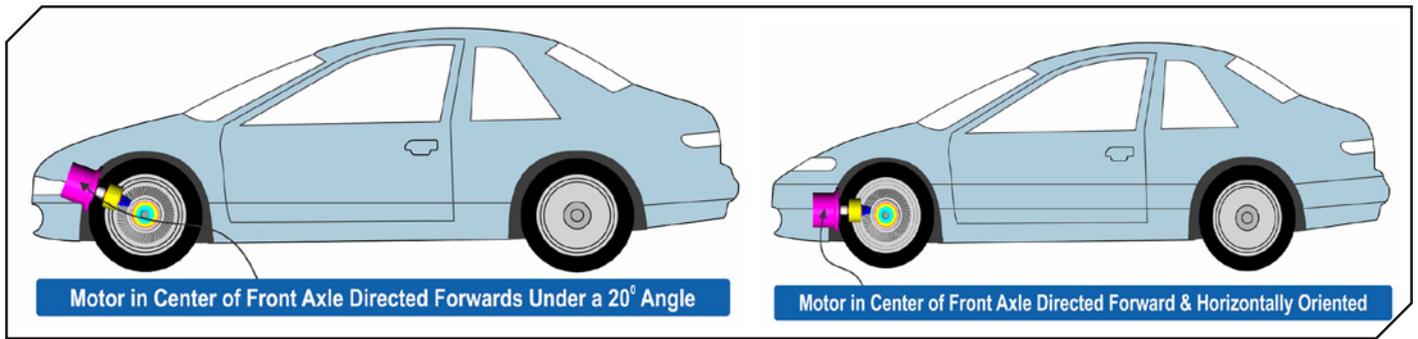


Figure 11 Possible orientations of hypoid transmission in front of axle.

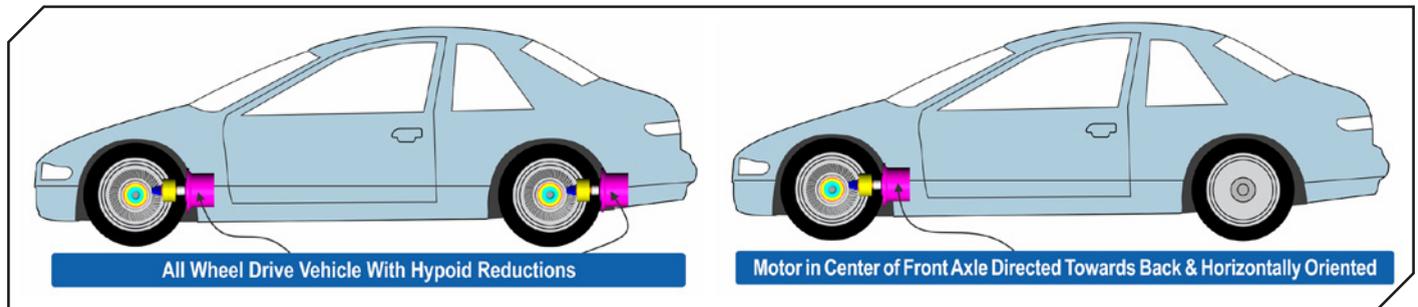


Figure 12 Possible orientations of hypoid transmission for front or all-wheel drive.

In case of a dual-speed transmission, the electronic control module can decide which of the two ratios with respect to the load will provide better motor efficiency. Figure 10 shows a typical speed diagram with the first ratio larger than the second. The first gear is active until the maximal motor rpm is reached. Then the clutch switches to the second gear, which stays active until maximal motor speed is reached again. The speed graph in Figure 10 has two sections which pass the optimal efficiency range. Depending on the duty cycle of the vehicle, the two-speed transmission can double the operating time within the optimum efficiency speed range and reduce the vehicle's energy consumption significantly.

Depending on gentle coasting to a full stop (leads off motor) or breaking light or hard, the electronic control module can regulate the downshift in order to optimize the brake force and maximize the battery re-charging. The flexible downshift is shown in Figure 10 as green dashed lines. The following breaking conditions are proposed:

- *Foot off the accelerator:*  
Coasting with leads off the motor/ generator
- *Pressing the brake pedal up to 30% of its travel:*

Braking force proportional to the pedal force by controlled downshift and a controlled generator charging conditions (ABS function still active by generator pulse charging)

- *Pressing the brake pedal beyond 30% travel:*  
Mechanical brake applied in addition to the generator brake

Some electric vehicles, even larger premium models, realize braking by releasing the accelerator pedal. This technology reduces the driving comfort, is counterintuitive and can lead to unsafe driving conditions. It requires a steady and unnatural foot position which also fatigues the driver—not merely the foot pressing the accelerator.

### Possible Orientations of Hypoid Transmission in Vehicles

As mentioned earlier, the hypoid reduction allows placing the eMotor in the center of the front or rear axle between the wheels. The images of a small-size compact sedan in Figure 11 show the eMotor in front of the front axle. With a battery location below the passenger cabin floor (between front and rear wheels), the motor orientation as shown in Figure 11 presents a good weight distribution and

could become part of the passive passenger impact safety concept. It is possible to reverse the direction of the hypoid offset (motor higher or lower) and find a packaging optimal motor orientation angle as shown in the left example in Figure 11.

In case of an all-wheel drive passenger car, the same transmission unit which is propelling the front wheels can be used to propel the rear wheels as well. The images in Figure 12 show a front wheel drive with the motor pointing to the rear (left graphic) and an all-wheel drive arrangement with both motors pointing backwards.

The concepts in Figures 11 & 12 provide a feeling of how compact the hypoid e-drive is and how well it can adjust to the given packaging constraints in a given overall vehicle concept. Many more e-transmission orientations are possible which accomplished tailored solutions for all already existing electric vehicle designs.

### Application Examples

For the application in premium mid-size electric vehicles as shown in Figure 13, a multiple-speed transmission as discussed with Figures 7 and 8 would be most suitable. Optimal motor and generator operating speeds are important if the mass of a vehicle is high. A multiple-speed transmission can assure that also larger vehicles operate with a good efficiency.

Some compact and micro electric cars are shown in Figure 14. Those cars can benefit from a single-stage SRH reduction without any further transmission elements. Especially the two shown micro cars in the center and to the right in Figure

14 have to be attractively priced in order to be appealing people who mostly do inner city driving with rather low speed.

Driving in congested inner cities does not allow much time for shifting between different ratios. The energy gained by optimal motor rpms does not make up for the lost energy due to constant up and down shifting. The inner city delivery micro truck to the right in Figure 14 has many applications, where the driving distance per day is below 100km (63 miles) and the start-stop duty cycle makes it the ideal candidate for a single-stage Super Reduction Hypoid application. The simplicity of a micro car often promotes a

rear-wheel drive. A good weight distribution is given by the location of the battery, which makes those small rear-wheel drive vehicles safe and good handling cars with a stunningly small turning radius.

### Summary

An eDrive concept, employing a hypoid reduction, was discussed in this chapter. The hypoid reduction brings a variety of advantages. Symmetric weight distribution and heat radiation away from one of the driving wheels, but also away from the batteries, can be accomplished very well. The speed drop which is possible with SRH hypoids is a multiple of what is realistic for cylindrical gearsets (ratios between 6 and 15 have been realized for eDrive developments). This makes it possible for small-size compact vehicles to limit the transmission to one fixed but large ratio. The result is a simple and low-cost transmission with pinion shaft equal motor shaft and only one additional shaft for the ring gear. Of course, just like in all axle drives with hypoid gears, the differential cage fits conveniently inside of the pinion-ring gear silhouette without additional space consumption.

This Chapter also discusses several transmission examples as combinations of the hypoid reduction and a cylindrical gear reduction. This solution is ideal for mid-size and premium electric vehicles and can be extended to dual-reduction for increased motor efficiency and higher efficiency in energy recuperation while the vehicle reduces speed in coast condition.

A variety of possible hypoid eDrive orientations and placements in an electric vehicle were proposed in Figures 11 and 12. Those examples suggest that there are numerous possibilities and the potential of the new solution is tempting for electric vehicle designers. ⚙️



Figure 13 Premium midsize electric cars.



Figure 14 Compact and micro electric cars.

Table 2 Comparison of some key parameters for the four presented designs.

	Design 1	Design 2	Design 3	Design 4	
Ratio	7.86	10.82	13.91	18.55	[-]
Power	60	60	60	60	[kW]
Mass	33	35	35	35	[kg]
Power Density	1.85	1.73	1.72	1.73	[kW/kg]
Gear Life	5,130	30,748	7,058	7,058	[h]
Gear Efficiency	95.2	95.4	94.3	94.5	[%]
Motor Speed	6,252	8,609	11,068	14,758	[RPM]
Motor Torque	92	67	52	39	[Nm]

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**Dr. Hermann J. Stadtfeld** is the Vice President of Bevel Gear Technology and R&D at the Gleason Corporation and Professor of the Technical University of Ilmenau, Germany. As one of the world's most respected experts in bevel gear technology, he has published more than 300 technical papers and 10 books in this field. Likewise, he has filed international patent applications for more than 60 inventions based upon new gearing systems and gear manufacturing methods, as well as cutting tools and gear manufacturing machines.



Under his leadership the world of bevel gear cutting has converted to environmentally friendly, dry machining of gears with significantly increased power density due to non-linear machine motions and new processes. Those developments also lower noise emission level and reduce energy consumption.

For 35 years, Dr. Stadtfeld has had a remarkable career within the field of bevel gear technology. Having received his Ph.D. with summa cum laude in 1987 at the Technical University in Aachen, Germany, he became the Head of Development & Engineering at Oerlikon-Bührle in Switzerland. He held a professor position at the Rochester Institute of Technology in Rochester, New York from 1992 to 1994. In 2000 as Vice President R&D he received in the name of The Gleason Works two Automotive Pace Awards—one for his high-speed dry cutting development and one for the successful development and implementation of the Universal Motion Concept (UMC). The UMC brought the conventional bevel gear geometry and its physical properties to a new level. In 2015, the Rochester Intellectual Property Law Association elected Dr. Stadtfeld the “Distinguished Inventor of the Year.” Between 2015–2016 CNN featured him as “Tech Hero” on a Website dedicated to technical innovators for his accomplishments regarding environmentally friendly gear manufacturing and technical advancements in gear efficiency.

Stadtfeld continues, along with his senior management position at Gleason Corporation, to mentor and advise graduate level Gleason employees, and he supervises Gleason-sponsored Master Thesis programs as professor of the Technical University of Ilmenau—thus helping to shape and ensure the future of gear technology.

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# Changes in ISO 6336:2019 — Parts 1, 2, 3, 5 and 6

Hanspeter Dinner

## Introduction

ISO 6336 is the most widely used and technically advanced document for cylindrical gear strength rating. It falls under the responsibility of ISO technical committee TC 60, subcommittee 2, work group WG 6 “Gear Calculations”. Convenor of WG6 is Prof. Dr. Ing. Karsten Stahl of the FZG Munich, Germany.

The third edition of ISO 6336 has been extended and now includes the now replaced ISO/TR 13989-1 and -2 on scuffing, the now replaced ISO/TR 15144-1 and -2 on micropitting, and a new part 4 on tooth flank fracture. Calculation examples form parts 30 and 31. The different parts and classification as standard, technical specification or technical report are listed in the below table.

In ISO/TR 6336-30:2017, (Ref.11), calculation examples based on ISO 6336-1:2006, (Ref. 1), ISO 6336-2:2006, (Ref. 3), ISO 6336-3:2006, (Ref. 5), ISO 6336-5:2003, (Ref. 7), are given. These documents are already replaced by the currently valid editions, but as of today, ISO/TR 6336-30:2017 is based on the prior edition. This technical report (TR) is currently being updated to represent the changes in ISO 6336-1:2019, (Ref. 1), ISO 6336-2:2019, (Ref. 4), ISO 6336-3:2019, (Ref. 6) and ISO 6336-5:2016, (Ref. 10).

The purpose of this paper is to point the reader to changes in the third edition of parts 1, 2, 3, 5 and 6 compared to the previous edition (from the year 2006 for parts 1, 2, 3, 6 and from the year 2003 for part 5). The paper is limited to a description of the changes and does not comment on their merit or their consequences. Some comments on the limitations of this paper are due:

- In the 2006 edition, a new section often started with some introductory text. In the 2019 edition, this text is put into a subsection “General” so that no body of text is directly beneath a top-level section title. This change is not mentioned

**Table 1 Parts of the ISO 6336 series**

Calculation of load capacity of spur and helical gears	International Standard	Technical Specification	Technical Report
Part 1: Basic principles, introduction and general influence factors	X		
Part 2: Calculation of surface durability (pitting)	X		
Part 3: Calculation of tooth bending strength	X		
Part 4: Calculation of tooth flank fracture load capacity		X	
Part 5: Strength and quality of materials	X		
Part 6: Calculation of service life under variable load	X		
Part 20: Calculation of scuffing load capacity—Flash temperature method		X	
Part 21: Calculation of scuffing load capacity—Integral temperature method		X	
Part 22: Calculation of micropitting load capacity		X	
Part 30: Calculation examples for the application of ISO 6336 parts 1, 2, 3, 5			X
Part 31: Calculation examples of micropitting load capacity			X

for each section.

- Only normative annexes are addressed here. Informative annexes are not documented in this paper.
- In the 2019 edition, some errors in the 2006 edition were corrected (e.g., formula for basic rack factor CB, changed from «...1.25 -  $h_{fp}$ ...» to «...1.20 -  $h_{fp}$ ...»), but such corrections are not documented.
- New parts of ISO 6336, e.g., part 4, are not documented here. They deserve a more detailed introduction outside the scope of this paper.
- Additional comments on the formulas and calculation examples for ring gears can be found in (Ref. 17).
- In all parts, a table with the abbreviated terms and symbols used in the part, is added to section 3 “Terms, definitions, symbols and abbreviated terms.”
- Bibliography has changed in all parts.

## Part 1 of ISO 6336:2019

### Overview

The scope of ISO 6336-1:2019, (Ref. 2) is defined as follows:

“This document presents the basic principles of, an introduction to, and the general influence factors for the calculation of the load capacity of spur and helical gears. Together with the other documents in the ISO 6336 series, it provides a method by which different gear designs can be compared. It is not intended to assure the performance of assembled drive gear systems. It is not intended for

use by the general engineering public. Instead, it is intended for use by the experienced gear designer who is capable of selecting reasonable values for the factors in these formulae based on the knowledge of similar designs and the awareness of the effects of the items discussed.

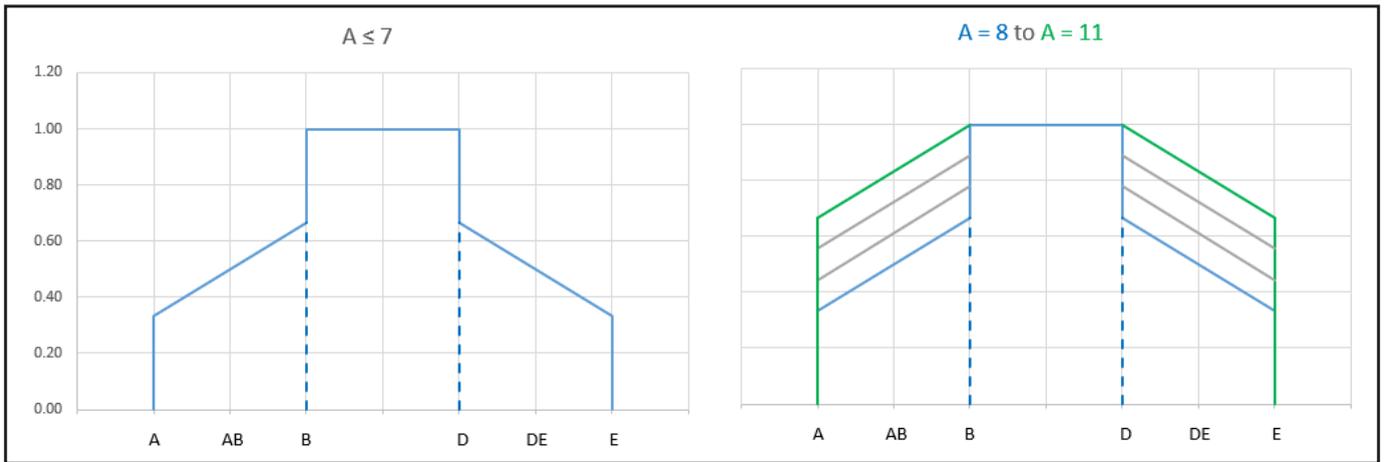
The formulae in the ISO 6336 series are intended to establish a uniformly acceptable method for calculating the load capacity of cylindrical gears with straight or helical involute teeth.”

The introduction section now includes above overview of the ISO 6336 documents, Table 1, as parts 4, 20, 21, 22, 30 and 31 have been added to ISO 6336 series.

This part now includes as normative reference ISO 21771:2007, (Ref. 13), for cylindrical involute gear geometry. This standard was published in 2007 and in ISO 6336:2006, no reference to such a standard was available. ISO 6336:2019 now states that terms and definitions of ISO 21771:2007 apply, along with those from ISO 1122-1:1998 (Ref. 12). With this, the gear strength rating standards are linked to gear geometry calculation standards.

## Structure of ISO 6336-1:2019, compared to previous edition

Foreword, Introduction, Sections 1, 2 and 3 maintain their structure. The



**Figure 1** Left: Load sharing factor, spur gear, unmodified profile,  $A \leq 7$ . Right: Load sharing factor, spur gear, unmodified profiles, for  $A = 8$  to  $A = 11$ .

introduction now includes an overview of all the documents belonging to ISO 6336 series, see Table 1.

Section 4, specifically section 4.1, now includes those failure modes addressed in the different parts of the standard in the same sequence as they are covered in the different part numbers. These are “Surface durability (pitting),” “Tooth bending strength,” “Tooth flank fracture,” “Scuffing” and “Micropitting.” Wear and plastic yielding are still mentioned as possible failure modes but are not covered in ISO 6336. Comments on specific areas of gearing, e.g., “Vehicle final drive gears,” “Main drive for aircraft and space vehicles” or “Industrial high-speed gears” remain included.

The section on safety factors remains as well; it is now more detailed and explains differences between different safety factors, e.g., that the safety factor for bending is calculated from two stresses while e.g., the safety factor for scuffing is calculated from two temperature levels. Both 2006 and 2019 version include the most important statement that “... Recommendations concerning these minimum values (for safety factors) are made... but values are not proposed...”

Section 5 on application factor  $K_A$  retains its structure but a subsection with guide values for the application factor, method B is added, see Table 3 below. In section 6 on the dynamic factor  $K_v$ , a section on  $K_v$  for low loaded gears,  $(F_t \times K_A \times K_y) / b < 100 \text{ N/mm}$ , has been added. Otherwise, the structure of the section is the same as in 2006 edition.

The structure of section 7 on the face

load factor  $K_{H\beta}$ , section 8 for the transverse load factors  $K_{H\alpha}$  and  $K_{F\alpha}$ , section 9 for the tooth stiffness parameters,  $c'$  and  $c_y$  and Annex A (normative), additional methods for determination of  $f_{sh}$  and  $f_{ma}$ , is the same as in 2006 edition.

A whole new section 10, Parameter of Hertzian contact, has been added in 2019 edition of ISO 6336. It originates from ISO/TR 15144-1:2014 (there, as section 10) and ISO/TR 13989-1:2000 (there, as section 9). In section 10.1, the formulas on how to calculate the normal and transverse radius of relative curvature in the contact point CP,  $\rho_{red,CP}$  and  $\rho_{red,t,CP}$  and from the transverse radius of curvature of the pinion and wheel are given (division of the product of the radii of curvature of pinion and wheel by sum of the radii of curvature of pinion and wheel). Note that the radii here are for the unmodified involute shape, not considering modifications like tip relief or similar. In a similar manner, the reduced modulus of elasticity  $E_r$  is calculated from the modulus of elasticity of the pinion and the wheel as well as from the Poisson’s ratio of pinion and wheel.

The local Hertzian contact stress  $p_{dyn,CP}$  calculated as method A through a loaded tooth contact analysis LTCA, typically using a 3D load distribution program, is defined as the local nominal Hertzian contact stress  $p_{H,CP,A}$  multiplied by the square root of  $K_A$ ,  $K_y$  and  $K_v$  (comments on these can be found above and below). For method B, no loaded tooth contact analysis is used, the load distribution is considered as additional factors (again the square root thereof)  $K_{H\alpha}$  (transverse load factor) and  $K_{H\beta}$  (face load factor).

The formula for the local contact stress in the contact point CP as per method B,  $p_{H,CP,B}$  then uses the relative radius of curvature (above), the reduced modulus of elasticity (above), the transverse tangential load at the reference cylinder  $F_t$ , the face width  $b$ , and the load sharing factor  $X_{CP}$  and is:

$$p_{H,CP,B} = \sqrt{\frac{E_r}{2 \times \pi}} \times \sqrt{\frac{F_t \times X_{CP}}{b \times \rho_{red,CP} \times \cos(\alpha_t)}}$$

In section 10.4, another very helpful formula to calculate the half of the Hertzian contact width,  $b_H$  is given as follows:

$$b_{H,CP} = 4 \times \rho_{red,CP} \times \frac{\rho_{dyn,CP}}{E_r}$$

Section 10.5 deals with the load sharing factor  $X_{CP}$  and is complex. The factor accounts for the load sharing between succeeding pairs of meshing teeth. It is defined along the path of contact, using parameter  $g_{CP}$  to describe the position of the contact. The value of  $X_{CP}$  does not exceed 1.00, where 1.00 means full transverse single tooth contact. The load sharing factor depends on the profile modifications and—for helical gears—is combined with the buttressing factor  $X_{but,CP}$ . The buttressing factor accounts for a stress increase at the start and end of the oblique contact lines on the flank in case of unmodified helical gears. Its minimum value is 1.00 and reaches a maximum of 1.30.

Section 10.5 starts with a definition of points A, AB, C, D, DE and E defined as special contact points CP, all on the path of action. If the pinion (the gear in mesh with the lower number of teeth) is driving, then, point A is the start of mesh. If the gear is driving, start of mesh is point E. It also defines the diameter of the

circles with center equal to pinion and gear center going through the contact point,  $d_{CP1}$  and  $d_{CP2}$ , both as a function of the parameter  $g_{CP}$  describing the position of the contact point CP on the path of action.

In section 10.5.2, the load sharing factor is given along the path of action (from points A to E) for spur gears with unmodified profiles, spur gears with profile modifications, helical gears with  $\epsilon_\beta \leq 0.80$  and unmodified profiles, helical gears with  $\epsilon_\beta \leq 0.80$  and modified profiles, helical gears with  $\epsilon_\beta \geq 1.20$  and unmodified profiles, helical gears with  $\epsilon_\beta \geq 1.20$  and modified profiles and helical gears with  $0.8 < \epsilon_\beta < 1.20$ . Only in case of a spur gear with unmodified profile, the load sharing factor is given for different quality classes A=8 to A=11.

Section 10.6 gives formulas on how to calculate the tangential velocities of the contact point CP on the flank of the pinion and the wheel. The two speed vectors in the contact point have the same direction (normal to the path of contact or tangential to both flanks), the velocity of the resulting speed vector is hence

equal to the sum of the velocities of the two vectors.

Section 11, Lubricant parameters at given temperature, has been added in edition 2019 of ISO 6336-1. It is the same content as in ISO/TR 15144-1:2010 (or its second edition of 2014, both withdrawn and replaced by ISO/TS 6336-22), section 7.2. Note that the identical text can again be found in ISO/TS 6336-22:2018 on micropitting. In this new section, the procedure to calculate the dynamic viscosity at a given temperature from the kinematic viscosity is given. Also, the calculation formulae for calculation of the kinematic viscosity at a given temperature from the kinematic viscosity at 40°C and 100°C are given. Finally, the formula for the calculation of the lubricant density at a given temperature from the lubricant temperature at 15°C is given in this section.

**Application**

The method for calculating the load capacity of cylindrical gears are in good agreement or validated for the below boundary conditions. Note the slight

changes between 2006 and 2019 edition for pressure and helix angle Table 2:

**Changes in formulas and factors**  
**Mesh load factor  $K_y$**

Edition 2019 now specifically includes the mesh load factor  $K_y$  in the formulas where it is—like e.g., the application factor  $K_A$ —multiplied with the nominal tangential load. This is clearer than in edition 2006 where a statement was given that  $K_A$  needs to be replaced by  $K_A \times K_y$  if a gear drives two or more mating gears. Some additional comments concerning the use of  $K_y$  are given.

Note that  $K_y$  is also used in the other parts of ISO 6336 in the formulas where in previous edition, only  $K_A$  was present. This change is not further mentioned in below sections.

**Application factor  $K_A$**

The system for describing application factors has been extended. In 2006 edition, the application factor  $K_A$  could be derived along method A or method B, becoming  $K_{A-A}$  and  $K_{A-B}$  respectively. In 2019 edition, separate application factors  $K_{HA}$  for pitting,  $K_{FA}$  for root breakage,  $K_{FFA}$  for tooth flank fracture,  $K_{\theta A}$  for scuffing,  $K_{\lambda A}$  for micropitting are introduced. Again, they can be determined along method A or B and the respective suffix is added.  $K_{FFA-B}$  for example means application factor along method B, for flank fracture calculation.

“Table 4—Application factor,  $K_A$ ” with guide values for  $K_{A-B}$  is added. Additional tables 6 and 7, with application examples, to be used in conjunction with this table, explaining the meaning of e.g., “Light shocks,” are also added in 2019 edition. With this, a practical tool to select a suitable value for  $K_{A-B}$  is available now:

**Dynamic factor  $K_v$**

In 2006 edition, a value for  $K_v > 2.00$  was possible. This means that gear flanks could separate and in 2019 edition, the dynamic factor  $K_v$ , for gears operating outside their resonance condition, shall be set  $K_{v-B}$  or  $K_{v-C} = 2.00$  if its calculated value is higher than 2.00.

The dynamic factor in the subcritical range ( $N \leq N_s$ ) and for main resonance range ( $N_s < N \leq 1.15$ ) is calculated using a parameter  $B_k$ .  $B_k$  was calculated in 2006 edition considering the tip relief  $C_a$  only.

**Table 2 Range of validity and limitations**

Property	2006 edition, limits	2019 edition, limits
Gear type	Cylindrical, external, internal	Cylindrical, external, internal
Profile geometry	Involute	Involute
Normal working pressure angle	Up to 25°	15° to 25°
Reference helix angle	Up to 25°	Up to 30°
Transverse contact ratio	1.00 to 2.50	1.00 to 2.50
Gears with $\epsilon_\alpha < 1.00$	Formulas not applicable	Formulas not applicable
ooth tips and root fillets interfere	Formulas not applicable	Formulas not applicable
Teeth are pointed	Formulas not applicable	Formulas not applicable
Backlash is zero	Formulas not applicable	Formulas not applicable

**Table 3 Guide values for application factor,  $K_{A-B}$ , as added in 2019 edition.**

Working characteristic of driving machine	Working characteristic of driven machine			
	Uniform	Light shocks	Moderate shocks	Heavy shocks
Uniform	1.00	1.25	1.50	1.75
Light shocks	1.10	1.35	1.60	1.85
Moderate shocks	1.25	1.50	1.75	2.00
Heavy shocks	1.50	1.75	2.00	≥2.25

**Table 4 Range of validity and limitations**

Property	2006 edition, limits	2019 edition, limits
Lubrication	Oil primarily, grease approximatively	Cylindrical, external, internal
Profile geometry	Basic rack as standardized in ISO 53, [15]	Basic rack as standardized in ISO 53, [15]
	Other basic racks where actual transverse contact ratio is less than $\epsilon_{\alpha n} = 2.50$	Other basic racks where actual transverse contact ratio is less than $\epsilon_{\alpha n} = 2.50$

**Table 5 Factor  $f_{Cza}$**

Condition	Value for $f_{Cza}$
Helical gear sets with suitable profile and longitudinal modifications based on the 3D load distribution program, and with the maximum contact stress near mid-height and essentially uniform stress distribution	1.00
Helical gear sets with suitable flank modifications acc. to manufacturers experience	1.07
Helical gear sets without flank modifications	1.20

In 2019 edition,  $B_k$  is calculated using  $\min(C_{a1}+C_{f2}, C_{a2}+C_{f1})$ , considering also root relief and being clear that the minimum value per gear shall be used.

Equations (25) and (26), for the calculation of the moment of inertia of a stationary and rotating ring gear in planetary gearboxes have changed. The correct index for the ring gear is now used and second instead of fourth power of the ring gear diameter is used in the denominator.

Edition 2019 of ISO 6336 references to ISO 1328-1:2013 (where lowest quality is grade 11), while edition 2006 references to ISO 1328-1:1995 (where lowest quality grade is 12). Quality grade 12 is no longer used for  $K_v$  calculation (see section 6.6.2 in ISO 6336-1:2019), lowest quality grade used is now 11.

### Face load factors $K_{H\beta}$ and $K_{F\beta}$

For calculation of  $K_{H\beta-C}$ , in case that favorable position of contact pattern is verified, the tolerance on helix slope deviation for ISO tolerance class 5 (along ISO 1328-1:2013),  $f_{H\beta 5}$  is used in edition 2019 instead of tolerance class 6 (along ISO 1328-1:1995),  $f_{H\beta 6}$  in edition 2006.

### Tooth stiffness parameters, $c'$ and $c_y$

In section 9.3.2.6, clarification is added that  $c_{y\beta}$  and  $c_{y\alpha}$  are defined in the transverse direction in the plane of action. This clarification was missing in 2006 edition.

## Part 2 of ISO 6336:2019

### Overview

The scope of ISO 6336-2:2019, (Ref. 4), is defined as follows:

“This document specifies the fundamental formulae for use in the determination of the surface load capacity of cylindrical gears with involute external or internal teeth. It includes formulae for all influences on surface durability for which quantitative assessments can be made. It applies primarily to oil lubricated transmissions, but can also be used to obtain approximate values for (slow running) grease lubricated transmissions, as long as sufficient lubricant is present in the mesh at all times.

The given formulae are valid for cylindrical gears with tooth profiles in accordance with the basic rack standardized in ISO 53. They can also be used for teeth conjugate to other basic racks where the

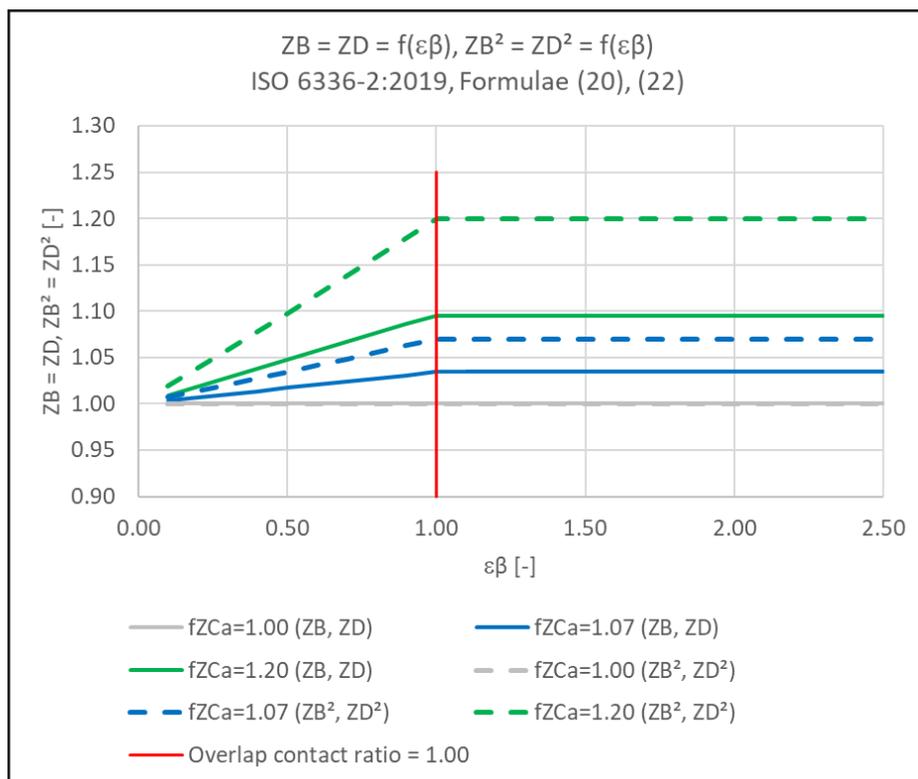


Figure 2 Zone factors  $Z_{K_{H\beta-C}} B$  and  $Z_D$  and their square as a function of  $f_{ZCa}$  and overlap contact ratio  $\epsilon_\beta$ .

Table 6 Material combinations considered for calculation of work hardening factor  $Z_W$

Material combinations, cases	2006 edition, material combinations considered	2006 edition, material combinations considered
	Case 1	Surface-hardened pinion with through-hardened gear
Case 2	Through-hardened pinion and gear	Through-hardened steel pinion with through-hardened steel gear
Case 3	NA	Surface-hardened steel pinion with ductile iron gear

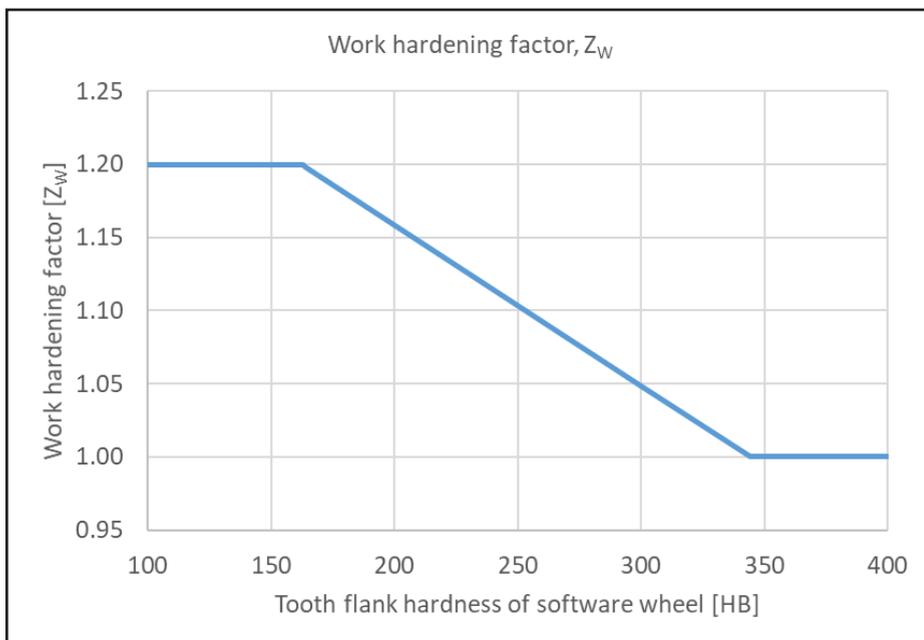


Table 7 Work hardening factor  $Z_W$ , ISO 6336-2:2019, for combination of surface-hardened steel pinion with ductile iron gear.

actual transverse contact ratio is less than  $\epsilon_{an} = 2,5$ . The results are in good agreement with other methods.”

**Structure of ISO 6336-2:2019, compared to previous edition**

The structure of the foreword, introduction and sections 1 to 12 and 14 remain the same.

In section 13, Work hardening factor,  $Z_W$ , a new subsection 13.3.3 Surface-hardened steel pinion with ductile iron gear, is added. It describes graphical values and determination by calculation of work hardening factor  $Z_W$  for the mentioned material pairing.

Annex A (informative), Start of involute, present in 2006 edition has been removed in 2019 edition.

**Application**

Further to the conditions for the application of ISO 6336 as found in part 1, see Table 2. These conditions apply, Table 4.

**Changes in formulas and factors**

Calculation of permissible contact stress for through hardened wrought steel, nitrided, nitro carburized steel

Formula (15) (it is the same formula in 2006 and in 2019 edition), for the calculation of the permissible contact stress of through hardened wrought steel, nitrided, nitro carburized, for the limited life stress range, has changed and now includes a different factor (0.7686 in 2019 edition vs. 0.7098 in 2006 edition).

**Calculation of zone factors  $Z_B$  and  $Z_D$ , diameters**

In the formulas, the active tip diameter  $d_{Na}$  is used (in edition 2019) instead of the tip diameter  $d_N$  (in 2006) edition.

**Calculation of zone factors  $Z_B$  and  $Z_D$ , auxiliary factor  $f_{ZCa}$**

For helical gears with  $\epsilon_\alpha > 1.00$  and  $\epsilon_\beta \geq 1.00$  (section 6.3, clause b) as well as for helical gears with  $\epsilon_\alpha > 1.00$  and  $\epsilon_\beta < 1.00$  (section 6.3, clause c), a new auxiliary factor  $f_{ZCa}$  is introduced and used for the calculation of the zone factors  $Z_B$  and  $Z_D$ :

For helical gears with  $\epsilon_\alpha > 1.00$  and  $\epsilon_\beta \geq 1.00$ , zone factors  $Z_B$  and  $Z_D$  are not  $Z_B = Z_D = 1.00$  anymore (as in 2006 edition) but  $Z_B = Z_D = f_{ZCa}^{0.5}$ . Contact stress is hence increased by  $f_{ZCa}^{0.5}$  or

transmittable torque is reduced by  $1/f_{ZCa}$ .

**Transverse and overlap contact ratio  $\epsilon_\alpha$  and  $\epsilon_\beta$**

Instead of root form and tip diameters (in edition 2006), active root and active tip diameters (in edition 2019) are used for determination of contact ratios.

**Lubricant factor  $Z_L$**

The table listing viscosity parameters (nominal viscosity  $\nu_{40}$ ,  $\nu_{50}$ , viscosity parameter  $\nu_j$ ) now includes the parameters for additional viscosity grades, VG10, VG15 and VG22. These values are used for determination of the lubricant factor  $Z_L$  by calculation.

**Factor  $Z_W$**

In 2006 edition, two cases were considered for the gears in mesh material combinations whereas in 2019, a third case has been added as listed in Table 6:

**Part 3**

**Overview**

The scope of ISO 6336-3:2016, [6], is defined as follows:

“This document specifies the fundamental formulae for use in tooth bending stress calculations for involute external or internal spur and helical gears with a rim thickness  $s_R > 0,5$  ht for external gears and  $s_R > 1,75$  mn for internal gears. In service, internal gears can experience failure modes other than tooth bending fatigue, i.e. fractures starting at the root diameter and progressing radially outward. This document does not provide adequate safety against failure modes other than tooth bending fatigue. All load influences on the tooth root stress are included in so far as they are the result of loads transmitted by the gears and in so far as they can be evaluated quantitatively.

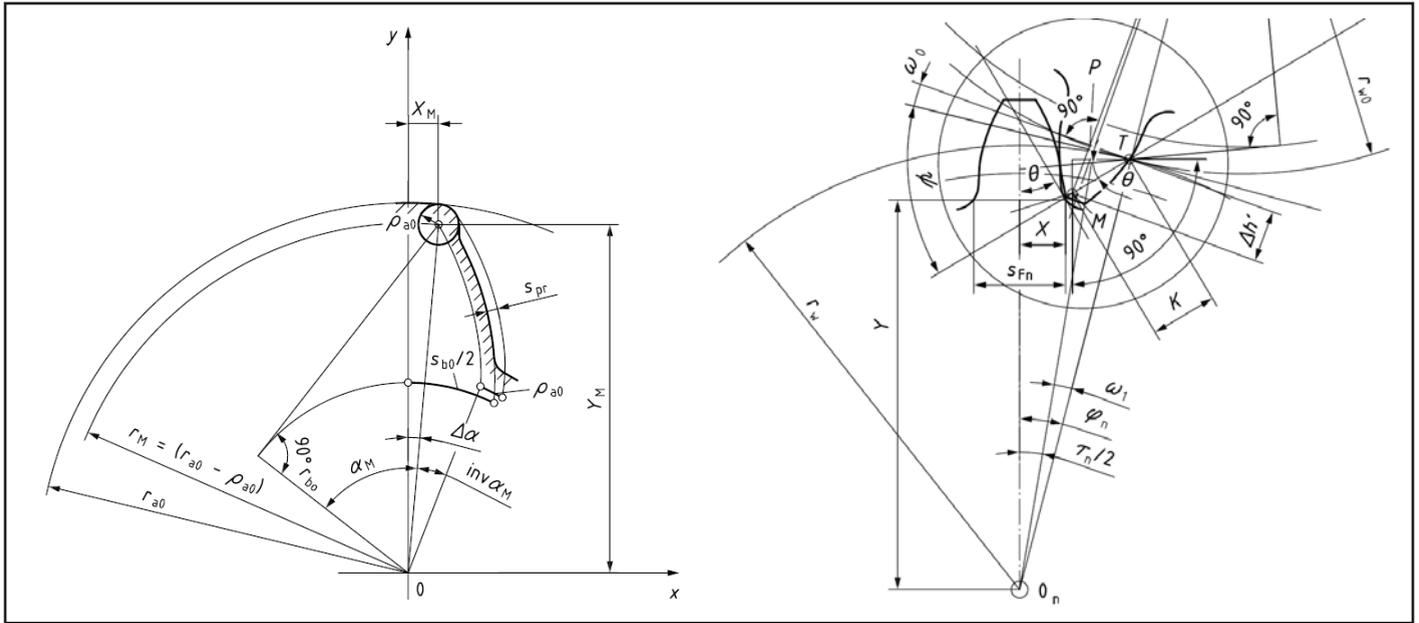
This document includes procedures based on testing and theoretical studies such as those of Hirt, Strasser and Brossmann. The results are in good agreement with other methods. The given formulae are valid for spur and helical gears with tooth profiles in accordance with the basic rack standardized in ISO 53. They can also be used for teeth conjugate to other basic racks if the virtual contact ratio  $\epsilon_{an}$  is less than 2,5.

The load capacity determined on the basis of permissible bending stress is

Table 8 Range of validity and limitations		
Property	2006 edition, limits	2019 edition, limits
Lowest applicable number of load cycles	10e3 cycles	10e3 cycles

		Virtual contact ratio of the virtual spur gear, $\epsilon_{an}$												
		1	1.1	1.2	1.3	1.4	1.5	1.6	1.7	1.8	1.9	2	2.1	2.2
Overlap ratio, $\epsilon_\beta$	0	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000	0.700	0.700	0.700
	0.1	1.000	0.995	0.992	0.988	0.986	0.983	0.981	0.979	0.978	0.976	0.707	0.705	0.704
	0.2	1.000	0.991	0.983	0.977	0.971	0.966	0.962	0.958	0.955	0.951	0.707	0.704	0.701
	0.3	1.000	0.986	0.975	0.965	0.956	0.949	0.942	0.936	0.931	0.926	0.707	0.702	0.697
	0.4	1.000	0.982	0.966	0.953	0.941	0.931	0.922	0.914	0.907	0.900	0.707	0.700	0.694
	0.5	1.000	0.977	0.957	0.941	0.926	0.913	0.901	0.891	0.882	0.874	0.707	0.699	0.691
	0.6	1.000	0.972	0.949	0.928	0.910	0.894	0.880	0.868	0.856	0.846	0.707	0.697	0.688
	0.7	1.000	0.968	0.940	0.916	0.894	0.876	0.859	0.844	0.830	0.818	0.707	0.695	0.684
	0.8	1.000	0.963	0.931	0.903	0.878	0.856	0.837	0.819	0.803	0.788	0.707	0.694	0.681
	0.9	1.000	0.958	0.922	0.890	0.862	0.837	0.814	0.793	0.775	0.757	0.707	0.692	0.678
	1	1.000	0.953	0.913	0.877	0.845	0.816	0.791	0.767	0.745	0.725	0.707	0.690	0.674
	1.1	1.000	0.953	0.913	0.877	0.845	0.816	0.791	0.767	0.745	0.725	0.707	0.690	0.674
	1.2	1.000	0.953	0.913	0.877	0.845	0.816	0.791	0.767	0.745	0.725	0.707	0.690	0.674
	1.3	1.000	0.953	0.913	0.877	0.845	0.816	0.791	0.767	0.745	0.725	0.707	0.690	0.674
	1.4	1.000	0.953	0.913	0.877	0.845	0.816	0.791	0.767	0.745	0.725	0.707	0.690	0.674
	1.5	1.000	0.953	0.913	0.877	0.845	0.816	0.791	0.767	0.745	0.725	0.707	0.690	0.674
	1.6	1.000	0.953	0.913	0.877	0.845	0.816	0.791	0.767	0.745	0.725	0.707	0.690	0.674
	1.7	1.000	0.953	0.913	0.877	0.845	0.816	0.791	0.767	0.745	0.725	0.707	0.690	0.674
	1.8	1.000	0.953	0.913	0.877	0.845	0.816	0.791	0.767	0.745	0.725	0.707	0.690	0.674
	1.9	1.000	0.953	0.913	0.877	0.845	0.816	0.791	0.767	0.745	0.725	0.707	0.690	0.674
	2	1.000	0.953	0.913	0.877	0.845	0.816	0.791	0.767	0.745	0.725	0.707	0.690	0.674
2.1	1.000	0.953	0.913	0.877	0.845	0.816	0.791	0.767	0.745	0.725	0.707	0.690	0.674	

Figure 3 Load distribution factor as a function of the  $\epsilon_\beta$  and  $\epsilon_{an}$ .



**Figure 4** Left: Quantities at the shaper cutter, (Ref. 6). Right: External gear in mesh with shaper cutter, coordinates  $X$  and  $Y$  of tangent point, tangent angle  $\theta$  and tooth thickness  $s_{Fn}$ , (Ref. 6). Tooth root fillet radius  $\rho_F$  and the bending moment arm  $h_{Fe}$  not shown explicitly.

termed “tooth bending strength.” The results are in good agreement with other methods for the range, as indicated in the scope of ISO 6336-1.”

### Structure of ISO 6336-3:2019, compared to previous edition

The structure of the foreword, introduction and sections 1 to 5 remain the same.

In section 6, the 2006 edition section 6.2.1 Tooth root normal chord,  $s_{Fn}$ , radius of root fillet,  $\rho_F$ , bending moment arm,  $h_{Fe}$  is replaced by sections 6.2.3 Tooth root normal chord,  $s_{Fn}$ , radius of root fillet,  $\rho_F$ , bending moment arm,  $h_{Fe}$  for external gears generated with a hob, 6.2.4 Tooth root normal chord,  $s_{Fn}$ , radius of root fillet,  $\rho_F$ , bending moment arm,  $h_{Fe}$  for external gears generated with a shaper cutter and 6.2.5 Tooth root normal chord,  $s_{Fn}$ , radius of root fillet,  $\rho_F$ , bending moment arm,  $h_{Fe}$  for internal gears generated with a shaper cutter.

Section 6.3 Derivations of determinant normal tooth load for spur gears has been moved to Annex C (informative) in 2019 edition.

The structure of sections 7 to 15 and Annex A (normative) again remain the same.

### Application

Further to the conditions for the application of ISO 6336 as found in part 1, see Table 2.

### Changes in formulas and factors

Load distribution influence factor  $f_{\epsilon}$  for the calculation of the form factor  $Y_F$  Compared to 2006 edition, in 2019 edition, the formula for the calculation of  $Y_F$  now contains a new factor  $f_{\epsilon}$ , the load distribution influence factor.  $Y_F$  as calculated in 2019 is equal to  $Y_F$  as calculated in 2006 edition, multiplied by  $f_{\epsilon}$ , both for external and internal gears. The factor  $f_{\epsilon}$  considers the influence of load distribution between the teeth in the mesh. It improves the results accuracy for gears with contact ratios  $\epsilon_{an} \geq 2.00$  (note that index  $n$  refers to the virtual spur gear). Contact ratios  $\epsilon_{an} \geq 2.00$  are reached for gears with high helix angles and or high transverse contact ratios  $\epsilon_{\alpha}$ . For gears with  $\epsilon_{an} < 2.00$ ,  $f_{\epsilon}$  becomes  $f_{\epsilon} = 1.00$  and the same values for  $Y_F$  result in 2019 edition compared to 2006 edition (except for the influence of the tooth thickness tolerance, see there). Note that the deep tooth factor  $Y_{DT}$ , for gears of high precision (ISO tolerance class  $\leq 4$ ) and  $2.00 \leq \epsilon_{an} \leq 2.50$  and with profile modifications to obtain a trapezoidal load distribution along the path of contact, does not change in 2019 edition compared to 2006 edition.

### Calculation of tooth root geometry for external gears generated with a shaper cutter

The calculation of tooth root geometry for external gears generated with a hob, section 6.2.3. in 2019 edition corresponds

to section 6.2.1 in 2006 edition (this section in 2006 edition is used for external gears generated with a hob or a shaper cutter and for internal gears generated with a virtual basic rack), with values and formulas “for external gears.”

For the calculation of tooth root geometry for external gears generated with a shaper cutter (and for internal gears, see below) however, a new section 6.2.4 (and 6.2.5 for internal gears, see below) with a new set of formulas is introduced in 2019 edition. These formulas originate from VDI 2737, Calculation of the load capacity of the tooth root in internal toothings with influence of the gear rim, 2016, (Ref. 16).

The formulas introduced in section 6.2.4 of 2019 edition are formulas No. (33) to (61) resulting in the coordinates  $X$  and  $Y$  of the tangent point (with tangent angle  $\theta = 30^\circ$  for external gears and  $\theta = 60^\circ$  for internal gears, see below), the tooth root thickness  $s_{Fn}$ , the tooth root fillet radius  $\rho_F$  and the bending moment arm  $h_{Fe}$ , as a function of the quantities at the shaper cutter.

### Calculation of tooth root geometry for internal gears

In 2006 edition, for internal gears, a virtual basic rack profile is used which differs from the basic rack profile in the tooth root radius  $\rho_{FP}$ . Formulas in section 6.2.1. “for internal gears” apply. In 2019 edition, for internal gears, only the shaper

cutter data is used. The calculation of the tooth root geometry (tooth root normal chord  $s_{Fn}$ , radius of root fillet  $\delta_F$  and bending moment arm  $h_{Fe}$ ) to derive form factor  $Y_F$  and stress correction factor  $Y_S$ , follows the same formula as used for external gears generated with a shaper cutter. However, negative signs are used for all diameters, and the manufacturing center distance  $a_0$  is used. Furthermore, the tangential angle  $\theta$  is  $\theta = 60^\circ$ .

### Influence of tooth thickness tolerances

In 2006 edition, form factor  $Y_F$  and stress correction factor  $Y_S$  are calculated from the nominal tooth form with the theoretical profile shift coefficient  $x$ . If the tooth thickness deviation near the root results in a thickness reduction of more than  $0.05 \times m_n$ , this shall be considered, by taking the generated profile,  $x_B$ , relative to rack shift amount  $m_n$  instead of the nominal profile. In 2019 edition, when the manufactured geometry is measured, it should be used. If not, then, based on the tooth thickness tolerance, the smallest generating profile shift,  $x_F$  min should be used to determine  $Y_F$  and  $Y_S$ .

### Helix angle factor $Y_\beta$

The tooth root stress of a virtual spur gear, calculated as a preliminary value, is converted by means of the helix factor,  $Y_\beta$ , to that of the corresponding helical gear. In 2019 edition, the formula to calculate  $Y_\beta$  has changed compared to 2006 edition, resulting in the values shown in Figure 5. Note that the value 1.00 is

Condition	2006 edition	2016 edition
Use with	ISO 6336-2, ISO 6336-3	ISO 6336-2, ISO 6336-3, ISO 6336-6
Use with	Application standards for industrial, high-speed and marine gears	Application standards for industrial, high-speed and marine gears
Applicable also	For rating of bevel gears along ISO 10300 series	For rating of bevel gears along ISO 10300 series
Limitations	Applicable to all gearing, basic rack profiles, profile dimensions, design, etc.	Applicable to all gearing, basic rack profiles, profile dimensions, design, etc.
Scope	Range indicated for scope of ISO 6336-1 and ISO 10300-1	Range indicated for scope of ISO 6336-1 and ISO 10300-1

substituted for  $\varepsilon_\beta$  when  $\varepsilon_\beta > 1.00$  and  $30^\circ$  is substituted for  $\beta$  when  $\beta > 30^\circ$ . Helix factors  $Y_\beta$  for  $\beta > 25^\circ$  shall be confirmed by experience.

### Relative notch sensitivity factor $Y_{\delta_{relT}}$ for static stress

The relative notch sensitivity factor  $Y_{\delta_{relT}}$  was defined in 2006 edition for normalized base steel (St), case-hardened wrought steel (Eh), flame or induction hardened wrought special steel (IF), nitrided wrought steel and nitriding steel (NT), through-hardened wrought steel, nitrided, nitrocarburized (NV), grey cast iron (GG) and modular cast iron (GGG). In 2019 edition, formulas to calculate the relative notch sensitivity factor for black malleable cast iron (GTS) are added in section 13.3.2.2, clause e).

### Part 5 Overview

The scope of ISO 6336-5:2016, (Ref. 8), is defined as follows:

“This document describes contact and tooth-root stresses and gives numerical values for both limit stress numbers. It specifies requirements for material

quality and heat treatment and comments on their influences on both limit stress numbers.” Furthermore, in the description of the scope of part 5, the application and limitation as listed in Table 8 are described.

### Structure of ISO 6336-5:2019, compared to previous edition

The structure of ISO 6336-5:2016, (Ref.8) is — with below exceptions — the same as in ISO 6336-5:2003, (Ref.7). Additional subsections have been introduced in 2016 edition:

- Section 4.4 on method Br for determination of allowable stress numbers.
- Annex A was normative in 2003 edition whereas it is informative in 2016 edition.

### Application

The conditions and limitations for application of ISO 6336-5 remain basically the same and are summarized in the table for the two editions, Table 8:

Minor changes in the normative references include:

- Some additional references are added (e.g., ISO 642, ISO 683-2, EN 10204 Metallic products — Types of inspection documents).

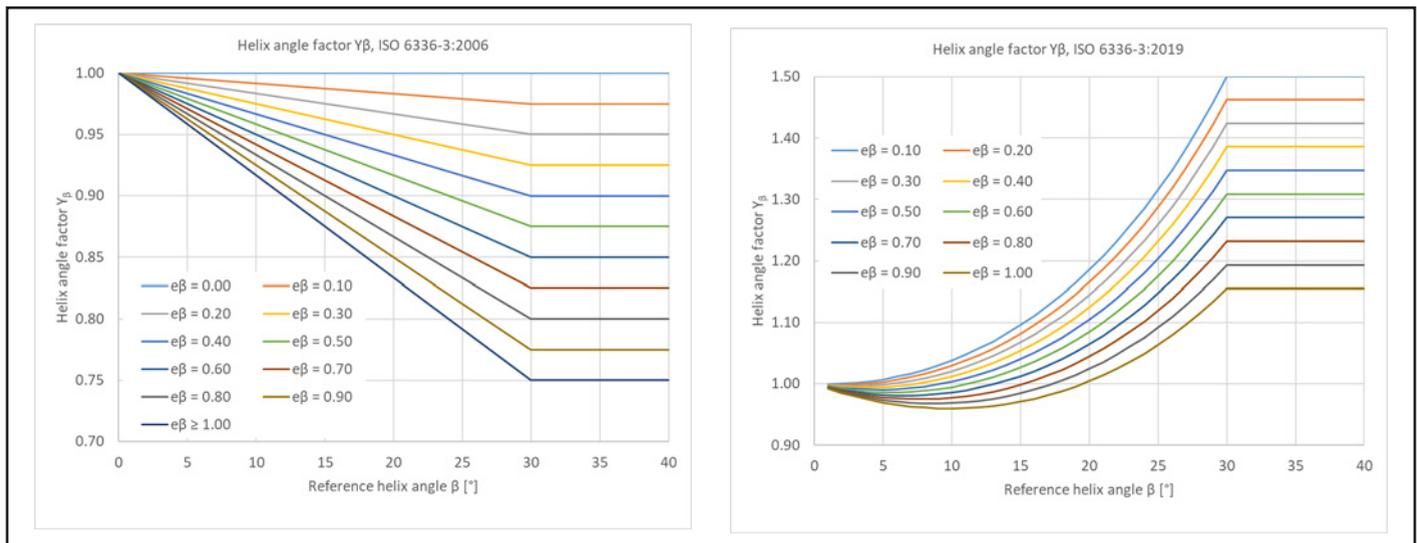


Figure 5 Helix angle factor  $Y_\beta$ . Left: along 2006 edition, Right: along 2019 edition.

- Year indicating revision of standards has been removed.
- Section 5.2 and 5.3.2: reference is made to ISO 1328-1. As ISO 1328-1 has been updated recently, the reference conditions are not necessarily the same anymore compared to ISO 6336-5: 2003.

### Changes in formulas and factors

In the new section 4.4, Method Br for methods for the determination of allowable stress numbers, the user is cautioned that "...contact stress numbers derived from rolling contact fatigue testing have to be used with caution since they tend to overestimate allowable contact stress numbers ..."

There are several minor changes and additions, but the permissible values typically used in gear rating are not changed. Changes include:

- Figure 5: Warning added for ME material grade for alloy steels "...relies heavily on the experience of the manufacturer..."
- Figure 14: Notes added with additional information on embrittlement due to white layer and aluminum containing nitriding steels.
- Figure 17+18, term "CHD" is used instead of "Eht" and "NHD" instead of "Nht".
- Figure 18: Maximum recommended NHD of 0.8mm is introduced. Values to determine hardness coefficient are moved to appendix B.2. A note is added regarding the use of heavier case depths for general designs.
- Table 3, item 3.1: some changes in the required cleanliness, conditions about calcium and oxygen content is moved to new items 3.2 and 3.3, item 4: requirements on grain size are now more specific, item 5.1: requirements on UT are now more specific, item 6: requirements for area reduction ratio now more specific, item 3: requirements on grain size are now more specific.
- Table 5, item 3: requirement that H2 content shall not exceed 2.5ppm, item 3.1: some changes in the required cleanliness, conditions about calcium and oxygen content is moved to new items 3.2 and 3.3, item 4: requirements for area reduction ratio now more specific, item 5: requirements on grain size are now more specific, item 10.4: permissible retained austenite level increased from 25% to 30%, magnetic particle inspection information removed.
- Table 8, item 7: "dwell time" removed, information on NHD added.
- Section 6.5: representative test bar size

changed.

- Section 6.7.1: standards for control of shot peening process added.
- Section 6.7.3; use of shot peening as a salvage operation, section added.

## Part 6

### Overview

The scope of ISO 6336-6:2016, (Ref. 10), is defined as follows:

"This document specifies the information and standardized conditions necessary for the calculation of the service life (or safety factors for a required life) of gears subject to variable loading for only pitting and tooth root bending strength."

It is noteworthy that this part 6 does not apply for other rating methods covered in ISO 6336 series, namely tooth flank fracture, scuffing and micropitting. Refer to the respective documents to find guideline on how to consider variable loads for these rating procedures.

### Structure of ISO 6336-6:2019, compared to previous edition

Section 4.1 and Annex B (informative) with guide values for the application factor  $K_A$  in ISO 6336-6:2006 has been removed in 2019 edition. They are now given in part 1, see Table 3 above. Otherwise, the structure of the document remains the same with edition 2019 as it was in edition 2006.

### Application

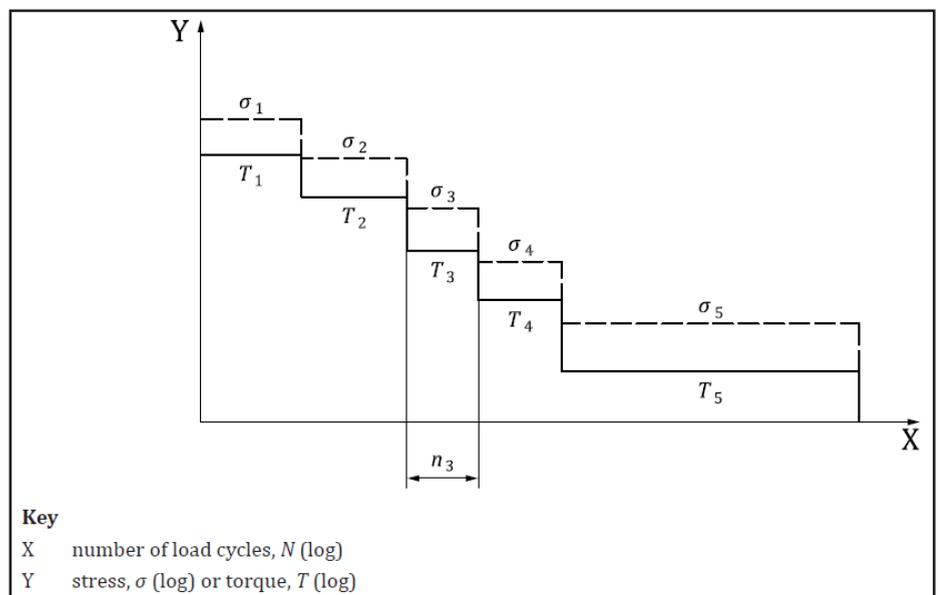
No specific information on application is given and those in parts 1, 2 and 3 apply.

### Changes in formulas and factors

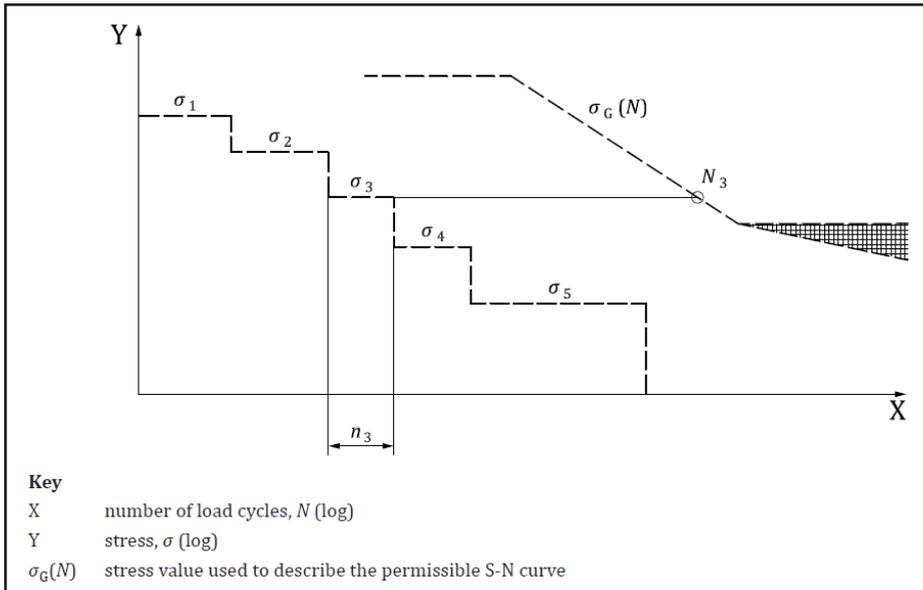
In 2019 edition of this part 6, at the end of section 4.3 on the Palmgren-Miner rule, a sentence is added as follows: "... Other damage accumulation (including non-linear) hypotheses in addition to the herein described method and permissible damage sums other than one may be used upon agreement of the purchaser and the gear box manufacturer..." This now allows for the use of a permissible total damage different (typically smaller) than unity, or modified Miner's rule, e.g., using Haibach modification and total permissible damage  $D=0.50$  as shown in (Ref. 16) to match experiments.

Formulas and the calculation process in section 5 of edition 2019 have not changed compared to 2006 edition. However, the graphics have been extended and improved. For each step in the calculation process, a separate graphic is now available, explaining the calculation step by step.

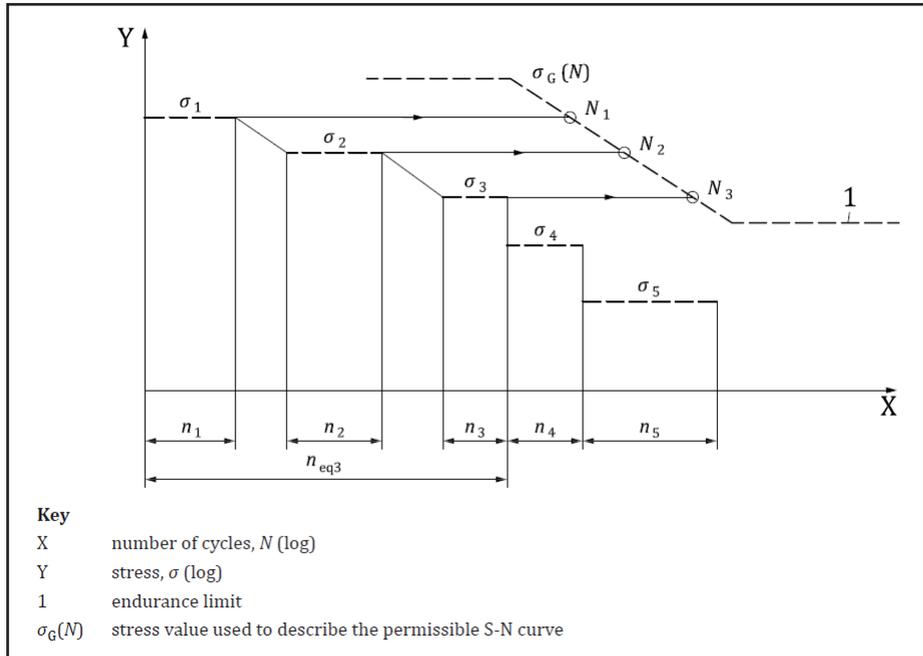
An additional figure for the case where the life factor  $Z_{NT}$  or  $Y_{NT}$  is less than unity in the range of long life is also included as Figure 5 of the 2019 edition of this part, not shown here.



**Figure 6** Load and stress spectrum, Figure 2 in ISO 6336-6:2019. The figure explains how the load (torque) spectrum, represented by the torque levels  $T_i$ , associated with a speed level  $n_i$  each, is converted to a stress spectrum with stress levels  $\sigma_i$  (for speed levels  $n_i$ ), using the methods given in ISO 6336-2 and -3. Above and below figures were combined in one figure in 2006 edition, lacking some clarity.



**Figure 7** Stress spectrum and S-N curve, Figure 3 in ISO 6336-6:2019. The figure explains how for a given stress level  $\sigma_i$ , an allowable number of load cycles  $N_i$  can be determined using the S-N curve(s) determined along ISO 6336-2 and -3. Note that the value  $\sigma_G$  is either  $\sigma_{HG}$  or  $\sigma_{FG}$ . Above two figures were combined in one figure in 2006 edition, lacking some clarity.



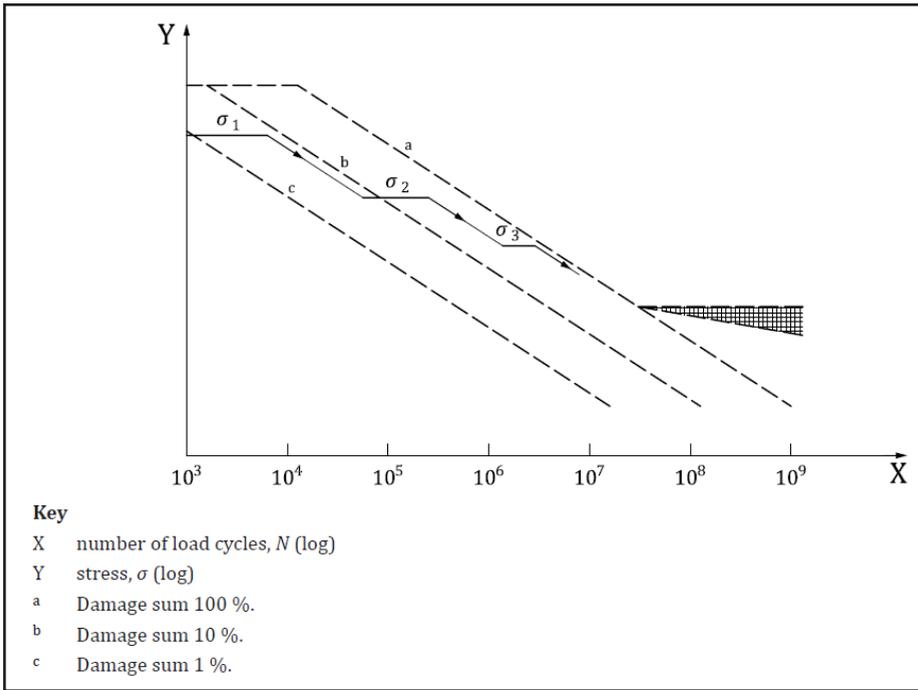
**Figure 8** Cumulated stress spectrum and fatigue curve limit when there is an endurance limit, Figure 4 in ISO 6336-6:2019. The figure explains how to represent the total accumulated damage  $U$  for the given load spectrum as the ratio  $n_{eqi}/N_i$  ( $i=3$  in this example) where  $\sigma_i$  is the lowest stress level still higher than the endurance limit. Note that the bins are shifted by drawing a line of the same slope than the S-N curve from the extremity of the stress bin  $\sigma_i$  to the stress level  $\sigma_{i+1}$ . This figure is new in edition 2016, adding clarity.

### Conclusion

The members of ISO TC 60/SC 2/WG 6 have contributed their expertise and time to advance ISO 6336. Industry experts and researchers have contributed new methods, run calculation examples with newly developed software and discussed how to improve the wording and structure of the document. The result is an even broader, more complex, practical and accurate tool aiming at increased gearbox power density, lowered risk of failures and deepened understanding of mechanisms governing the load capacity of cylindrical gears. Work does not stop here, under the guidance of the convenor, WG 6 is already working on the next edition of the documents.

The above changes will result in different load ratings, some of the differences are substantial, e.g., for internal gears or for helical gears without modifications. Users of ISO 6336 are advised to run extensive comparisons and gain experience using the third edition of ISO 6336. Further papers discussing the pros and cons of the changes or the implications on gear design are desirable. The influence of the changes on application standards and guidelines in specific industries is currently being discussed. This paper may help the engineers responsible for application standards and guidelines in adjusting e.g., K factors or required safety factors listed there.

While care has been taken to list all changes, the complexity of the document will have resulted in some changes having been overlooked. If so, readers' comments will be appreciated, thank you. 



**Figure 9** Accumulation of damage, shown here for the first three bins (following bins are below endurance limit in this example). Accumulated damage  $U$  is  $U = n_{eq3}/N_3$ . Lines b and c are corresponding to line a (S-N curve determined along ISO 6336-2 or -3) but for higher probability of failure. This figure is the same in both editions.

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**Hanspeter Dinner** studied mechanical engineering at the Swiss Federal Institute of Technology, Zürich, Switzerland and the National University of Singapore. He first worked as FEM engineer with a Swiss consultancy and as lead stress engineer with a roller coaster developer. He joined KISSsoft AG as software support and project engineer. In 2008, he started the consultancy company EES KISSsoft GmbH, representing the KISSsoft company in China, Japan, Korea, Taiwan and India. He has conducted about a hundred FEM, gear, bearings and transmission projects serving the wind, tractor, industrial gearbox and fine pitch gear industry. Since August 2019, he has been working in the function of Director Global Sales with KISSsoft – A Gleason Company.



# Nordex

## RECEIVES AS9100D AND ISO 9001:2015 ACCREDITATION

Nordex, Inc. recently announced the establishment of a quality management system for AS9100D & ISO 9001:2015. The company is involved in the manufacture, distribution, assembly and refurbishing of catalog and non-standard high precision mechanical/electro-mechanical components and assemblies such as bearings, couplings, gears, ball slides, enclosed gear-trains and related instrument grade power transmission components and assemblies for medical laboratory and industrial analytical instruments, factory automation, semiconductor and aerospace OEMs.



Nordex growing strong over three generations.

“Nordex has been in business for more than 60 years. We have accomplished this through continuously striving to improve our business so we can better serve our customers. AS9100D certification helps Nordex to be a better company, through improving our quality processes and reducing our costs—this helps us to provide more value to our customers and ensures we always deliver the high-quality products they have come to expect,” said Dan Agius Jr., CFO and part-owner of Nordex, Inc.

It’s not just the quality department that has been improved by AS9100 certification, but Nordex’s entire manufacturing operations, according to Agius.

“Through our heightened quality procedures, we are running more efficiently and finding and correcting potential problems very early on in our manufacturing process. Obtaining AS9100D certification has helped provide us with the tools to do this by developing improved risk mitigation strategies. This helps us to meet and exceed our customer’s needs when it comes to quality and on-time delivery,” he added.

With the evolution of the power transmission industry, Nordex has been able to thrive thanks in part to quality control procedures that have helped the organization to better adopt

and utilize cutting-edge power transmission technologies.

Agius believes the company will continue this approach in the future.

“We need to consistently update our quality requirements so we can keep meeting our customers’ needs as their businesses are ever changing. AS9100D is a quality framework that gives us the tools to keep building out our quality systems and standards, however necessary, well into the future.

[www.nordex.com](http://www.nordex.com)

# Gleason

## STRENGTHENS NORTH AMERICAN SALES TEAM

Gleason has added to its North American sales team in order to support increased customer activity in several market sectors growing in importance. These personnel additions include:

The addition of **Nick Deaville**, who has joined Gleason as regional sales manager. Deaville is located in the Houston, TX area and will support Gleason’s customers in the Western US and Canada. He brings 10+ years of machine tool, cutting tool/abrasive and automation experience to the market. Deaville is a hands-on technical service and support provider with a strong mechanical background which fits well with the technical expertise of all the Gleason sales team.



**Mike Gessler** has transitioned to Detroit, MI as regional sales manager for the North-central sales region. Gessler will continue to bring his depth of industry experience to provide Gleason’s customers with world class support. In addition, he will expand his role to take responsibility as Gleason’s key market manager - eDrive technologies, a fast-growing and vitally important market for Gleason.



Gleason also recognizes the long and distinguished sales career of **Jamie Washburn**, who is retiring from his role of regional sales manager after a 49-year career with Gleason. Gleason honors the accomplishments Washburn has amassed through his dedicated service to the company and the gear industry.



[www.gleason.com](http://www.gleason.com)

# Solar Atmospheres

## SOUTH CAROLINA AWARDED AEROJET ROCKETDYNE APPROVAL

Solar Atmospheres Greenville, SC facility is pleased to announce it has been awarded Aerojet Rocketdyne approval, their second aerospace prime approval of 2021.



Steve Prout, president of Solar Atmospheres' Greenville facility states: "We are proud to once again provide our customers in the Southeastern U.S. with another regional option for aerospace vacuum thermal processes, saving them time and money while continuing to deliver the high level of quality required."

With the ability to support vacuum thermal processing needs ranging from development cycles to 50,000 pound loads at temperatures of up to 2400°F, Solar Atmospheres provides AS9100 and Nadcap quality accredited heat treatments, providing our customers with the confidence their product is being processed as specified.

[www.solaratm.com](http://www.solaratm.com)

# API and Mitutoyo

## ANNOUNCE WORLDWIDE RESELLER AGREEMENT

API and Mitutoyo Corporation have announced a partnership to distribute API's portable dimensional metrology equipment throughout Mitutoyo's worldwide network.

Production times and tolerances continue to grow tighter for manufacturers around the world, and their need for precise, portable measurement hardware and software has increased to meet this demand. Adding API's complete portfolio of portable equipment to Mitutoyo's already diverse portfolio to meet this demand was a natural fit for both companies.

"API really invented portable dimensional metrology, especially in the form of precise, laser measurements," says Yoshiaki Numata, president of Mitutoyo Corporation. "So, for Mitutoyo, this was a great opportunity to add the most complete line of high-quality portable metrology equipment to what we can offer to our customers."

"Mitutoyo has developed the most diverse, comprehensive

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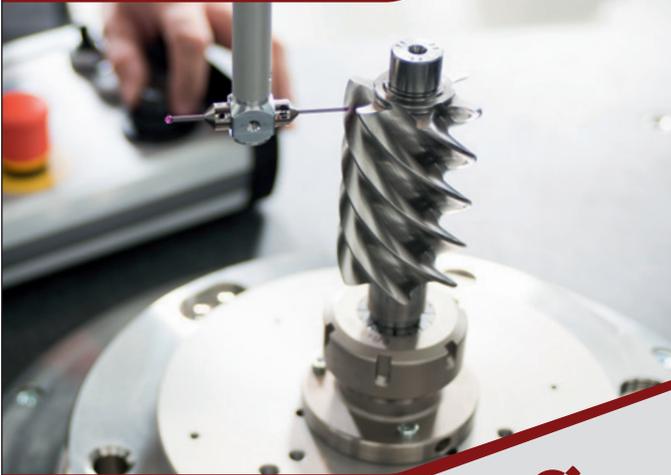


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portfolio of metrology equipment, hardware, and software, and they have built a truly impressive global distribution network,” says Joe Bioty, president of API. “By adding API’s portable equipment to that offering, Mitutoyo really becomes a one-stop shop for metrology needs in manufacturing worldwide, which provides tremendous value to the customer.”

[www.apimetrology.com](http://www.apimetrology.com)

## LK Metrology CELEBRATES 58<sup>TH</sup> ANNIVERSARY

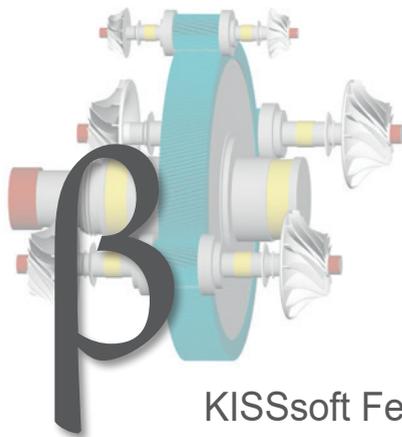
LK Metrology, the Derbyshire, UK-based coordinate measuring machine (CMM) manufacturer, is celebrating its 58th anniversary this year as a fully independent company again, following a change of ownership in 2018. The company pioneered CMM technology and its name is linked to numerous innovations such as ceramic bridge design, air bearing and granite guideways, carbon fiber composite spindles and horizontal-arm coordinate measuring.

The new proprietor, Angelo Muscarella, who also runs Italian firm ASF Metrology, is keen to share the progress that LK has made in the short time he has been in charge. He said, “We are delighted to find ourselves entering the fourth year of operation in April 2021 with a healthy order book. Revenue met budget in the last financial year and we even made a profit, despite incredibly difficult trading conditions due to the pandemic. Machine service and upgrade activities actually performed ahead of budget every month since July 2020.”

LK has made great strides growing its product portfolio in



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the last three years. In 2020 it entered the articulating-arm 3D metrology system sector with the launch of a range of six different FREEDOM arms with six size options. Earlier introductions included the SCANtek 5 version of the ALTERAM CMM with Renishaw REVO-2 probe as well as a new product, AlteraC 10.7.5, in three sizes. The latter range lowers the price of entry to LK CMM multi-sensor inspection technology and has recently been further improved in terms of accuracy and speed of measurement.

Two notable introductions have been made on the software side as well. The company's DMIS-based CMM programming software, now called CAMIO 2021, has been given enhanced features in five important areas, helping to increase inspection productivity, improve the quality of data collected and gain better insight into the components being measured.

Prior to that, LK introduced the Metrology Gate portal, a web-based platform consistent with the aims of Industry 4.0 that enables inspection activities performed by connected metrology devices to be monitored and controlled around the clock from anywhere in the world. A new MTConnect adapter has just been made available to allow users to add their LK CMMS to existing dashboard applications in their factories that previously were only able to display information on machine tools. Learn more here:

[www.lkmetrology.com](http://www.lkmetrology.com)

## Index Corporation WELCOMES REGIONAL SALES MANAGER FOR MIDWEST

Index has announced the hiring of industry veteran **Bob Pellegrini** for the role of regional sales manager for Northern Illinois, Michigan's Upper Peninsula and Eastern Wisconsin. The Hartland, WI native brings more than 20 years of successful manufacturing technology sales experience to Index, including 11 years with DMG MORI North America.



[www.index-usa.com](http://www.index-usa.com)

## Verisurf

### PARTNERS WITH SMARTTECH3D FOR METROLOGY SOLUTIONS

Verisurf Software has partnered with SmartTech3D, Lomianki/Warsaw, Poland, to offer metrology customers, worldwide, comprehensive end-to-end solutions for 3D scan-to-print, quality inspection and reporting, and reverse engineering to intelligent CAD. Under the partnership agreement SmartTech3D will offer complete 3D scanning solutions including SmartTech3D scanners, Verisurf software, installation, training, and technical support services.



SmartTech3D color scanners are portable and can be used in the quality lab, on the shop floor, or in the field. The high-resolution precision scanners collect information about complex structures that were difficult to scan until now, like highly reflective surfaces or dark objects, ceramic ornaments, clothes, or oil paintings. When combined with optional shadowless lighting, the system provides the possibility to obtain actual color information in high resolution.

“Our high resolution structural light scanners are focused on collecting precise geometry and color data. When it comes to processing and analyzing that data, we feel Verisurf provides most comprehensive set of applications, all of which are based on a powerful CAD platform. This is both unique and extremely customer focused,” said Anna Gebarska, managing director of SmartTech3D.

SmartTech3D dealers are now able to provide customers with complete end-to-end scanning solutions and provide added customer value through software training, maintenance support and metrology integration across the manufacturing enterprise. With the Verisurf Software Development Kit (SDK) dealers, systems integrators and customers alike have access to a flexible programming environment specifically designed for creating customized dimensional metrology applications that enable manufacturing automation.

“Capturing good data is paramount to all application processing that follows. At Verisurf we pride ourselves on being able to apply measurement data effectively to inspection and reporting, reverse engineering, and assembly guidance applications. Our Solution Partnership with SmartTech3D provides customers,

worldwide, with a best-in-class scanning solutions that are easy to use and based on an open CAD platform,” said Ernie Husted, president and CEO of Verisurf Software, Inc.

Verisurf Software SCAN DATA SUITE for SmartTech3D includes Verisurf CAD, ANALYSIS and REVERSE modules:

- CAD is at the foundation of all Verisurf applications and supports 3D wireframe modeling, 3D surface modeling, 3D solid modeling and 2D drafting. Model-Based Definition (MBD) — includes ISO 10303 & ASME Y14.5 compliant 3D model associated GD&T creation and editing. TOOLS for model-based metrology including auto flip surface normals, auto hole axis finder, screen shot grabber, 3D-PDF and more. Import files in ACIS, AutoCAD, Inventor, IGES, KeyCreator, Parasolid, Rhino, SolidEdge, SolidWorks, Spaceclaim, STEP, STL, VDA formats. Export files in ACIS, AutoCAD, IGES, Parasolid, STEP, STL, VDA file formats
- ANALYSIS module includes fast best fitting of measurements to 3D CAD model nominal. Analyze points-to-points, points-to-curves, points-to-surfaces or points-to-meshes and communicate results with color deviation maps, whisker plots, custom ballooning, and reporting
- REVERSE is set of powerful tools to reverse engineer scan data into fully closed mesh surface models ready for 3D Printing, Additive Manufacturing and Solid Modeling. Imports, edits, and exports massive point clouds and STL meshes. Extensive mesh alignment, filtering, smoothing, mesh bridging, hole filling, edge extending, and trimming, and slicing. Create NURB surfaces by sketching or projecting curves onto mesh or with intelligent prismatic feature fitting and auto surfacing

The SCAN DATA SUITE is also available with QUICK SURFACE, a powerful option for Verisurf Reverse that efficiently creates smooth, high-quality surfaces from meshes derived from scan data or STL files. Quick Surface maintains curvature continuity between adjacent surfaces and is ideal for digital surface modeling and creating smooth, high-speed toolpaths.

[www.verisurf.com](http://www.verisurf.com)

## Klingelberg

### WINS 2021 GERMAN INNOVATION AWARD

Machine manufacturing firm Klingelberg is the proud recipient of yet another award. This time, the company won over the expert jury for the 2021 German Innovation Awards with its “Done-in-One — Complete Measurement in a Single Stage” solution. The German Innovation Awards honor products and solutions that distinguish themselves from earlier solutions primarily by their user centricity and added value. The German Innovation Awards are granted by the German Design Council, which was enacted into law by the German Parliament in 1953 and is funded by the Federation of German Industries (BDI).

### A novel approach

With its innovative “Done-in-One — Complete Measurement in a Single Stage” solution, Klingelberg entered the “Machine and Engineering” category in the “Excellence in Business to Business” competition class. Klingelberg’s approach is to perform various measurement processes in a single stage as one complete measurement (“Done-in-One”), all in the immediate production environment. A Klingelberg Precision Measuring



Center (G variant) has rapid measurement capability for dimensions, shape, contour, and surface roughness in one automated cycle. By combining measurement tasks traditionally performed on up to four different devices, it is possible not only to lower investment costs, but also to decrease setup times and reduce quality costs.

The integration of measuring technology into the immediate production environment, in particular, helps to increase the productivity and output of the production plants. In sum, a fully automatic measuring run can save approximately 40% of the measurement and setup time.

“The German Innovation Awards honor projects that are pioneering in their field. So we are thrilled that our Done-in-One solution is among the winners of the 2021 German Innovation Awards,” said Jan Klingelberg, CEO of the Klingelberg Group. “True to the expert jury’s ‘Making innovations visible’ motto, the award represents our hard work in recent years.”

“This success is ultimately the result of many small- and large-scale developments from recent years, some of which are patented, including the further development of the tactile measurement system 3D NANOSCAN,” added Dr. Christof Gorgels, division head of the precision measuring centers product line. “The Klingelberg Done-in-One solution can only be implemented when a wide range of innovations and detailed technical solutions come together. We are therefore extremely pleased to have received the award and gained widespread recognition in the market.”

### Assessment criteria

The jury is made up of independent interdisciplinary experts from industry, science, institutions, and finance. The assessment was based on the criteria of innovation, benefit to users, and economic efficiency. The innovation strategy had to take aspects such as social, ecological, and economic sustainability and the use of energy and resources into account. Factors such as the potential of the location and employability, durability, market maturity, technical quality and function, materiality, and synergy effects also played a decisive role in the judging process.

[www.klingelberg.com](http://www.klingelberg.com)

# Open Mind

CELEBRATES 25 YEARS OF CAM SOFTWARE INNOVATION

Open Mind Technologies USA, Inc., the North American subsidiary of Open Mind Technologies AG, announced it is celebrating its 25th anniversary of CAM software achievements this year. Open Mind is the developer of the hyperMILL software suite, and is one of the world's leading CAD/CAM solutions.



“We are excited to reach this important milestone,” said Alan Levine, managing director of Open Mind Technologies USA, Inc. “We attribute our sustained growth to the strong product provided by our global headquarters, together with our local sales, engineering and service delivery. Our team is dedicated to providing customers with an optimal CAD/CAM experience including outstanding technical consultation and service.”

Levine continued, “Our team is strategically located across the nation as well as in Canada and Mexico to closely collaborate with customers on local and regional levels, and stay up-to-date on market requirements.”

Volker Nesenhöner, CEO, Open Mind Technologies AG said, “We congratulate Alan and his entire team in North America. Open Mind continues to expand market share in the United States, Canada and Mexico as the hyperMILL software suite is a superior choice for highly productive machining.”

Leading the way in 5-Axis processing, Open Mind developed the world's first CAD/CAM system utilizing an automatic multi-axis simultaneous machining strategy for mold and die, and also introduced many CAM software techniques including strategies for conical barrel cutters, auto-indexing, and 5-axis helical drilling. In addition to providing a high-performance and innovative CAM software solution, Open Mind's hyperMILL software offers connectivity to CAD systems, feature and macro technology, tool management systems, automation, simulation, probing, and more. The company continues to be at the forefront of key and emerging technologies, such as providing integration with additive manufacturing processes and supporting Industry 4.0 initiatives. hyperMILL is successful in a wide range of industries including mold and die, aerospace, automotive, turbomachinery, energy, medical and more.

[www.openmind-tech.com](http://www.openmind-tech.com)

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# Not Your Average Drone Delivery

Matthew Jaster, Senior Editor

**It's getting complicated.** First it was the Amazon packages dropping into the backyard via an unmanned vehicle. This was followed by numerous companies working on autonomous vehicles for buses, taxis and even two-seater air shuttles. Now, a prototype unmanned aerial tanker refueled a F/A-18F Super Hornet fighter jet in early June. That's right—a drone just refueled a fighter jet in mid-air.

The drone—a T1 prototype of the U.S. Navy's MQ-25A Stingray unmanned aerial vehicle—was controlled by operators from the MidAmerica Airport in Illinois. This successful flight demonstrated that the MQ-25 Stingray can fulfill its tanker mission using the Navy's standard probe-and-drogue aerial refueling method, according to a U.S. Navy press release.

"This flight lays the foundation for integration into the carrier environment, allowing for greater capability toward manned-unmanned teaming concepts," said Rear Adm. Brian Corey who oversees the Program Executive Office for Unmanned Aviation and Strike Weapons. "MQ-25 will greatly increase the range and endurance of the future carrier air wing – equipping our aircraft carriers with additional assets well into the future."

During the flight, the receiver Navy F/A-18 Super Hornet approached the Boeing-owned MQ-25 T1 test asset, conducted a formation evaluation, wake survey, drogue tracking and then plugged with the unmanned aircraft. T1 then successfully transferred fuel from its Aerial Refueling Store (ARS) to the F/A-18.



The MQ-25 T1 test asset refuels a Navy F/A-18 during a flight June 4 at MidAmerica Airport in Illinois.

"This is our mission, an unmanned aircraft that frees our strike fighters from the tanker role, and provides the Carrier Air Wing with greater range, flexibility and capability," said Capt. Chad Reed, program manager for the Navy's Unmanned Carrier Aviation program office (PMA-268). "Seeing the MQ-25 fulfilling its primary tasking today, fueling an F/A-18, is a significant and exciting moment for the Navy and shows concrete progress toward realizing MQ-25's capabilities for the fleet."

The test flight will provide important early data on airwake interactions, as well as guidance and control, Reed said. The team will analyze that data to determine if any adjustments are needed and make software updates early, with no impact to the program's test schedule.

Testing with T1 will continue over the next several months to include flight envelope expansion, engine testing, and deck handling demonstrations aboard an aircraft carrier later this year.

The MQ-25A Stingray will be the world's first operational carrier-based unmanned aircraft and provide critical aerial refueling and intelligence, surveillance and reconnaissance (ISR) capabilities that greatly expand the global reach, operational flexibility and lethality of the carrier air wing and carrier strike group. The MQ-25 is foundational to the Navy's Unmanned Campaign Framework and is the first step toward a future fleet augmented by unmanned systems to pace the evolving challenges of the 21<sup>st</sup> century. Learn more here: 

[www.navair.navy.mil/](http://www.navair.navy.mil/)



The test flight will provide important early data on airwake interactions, as well as guidance and control.

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