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A Publication of
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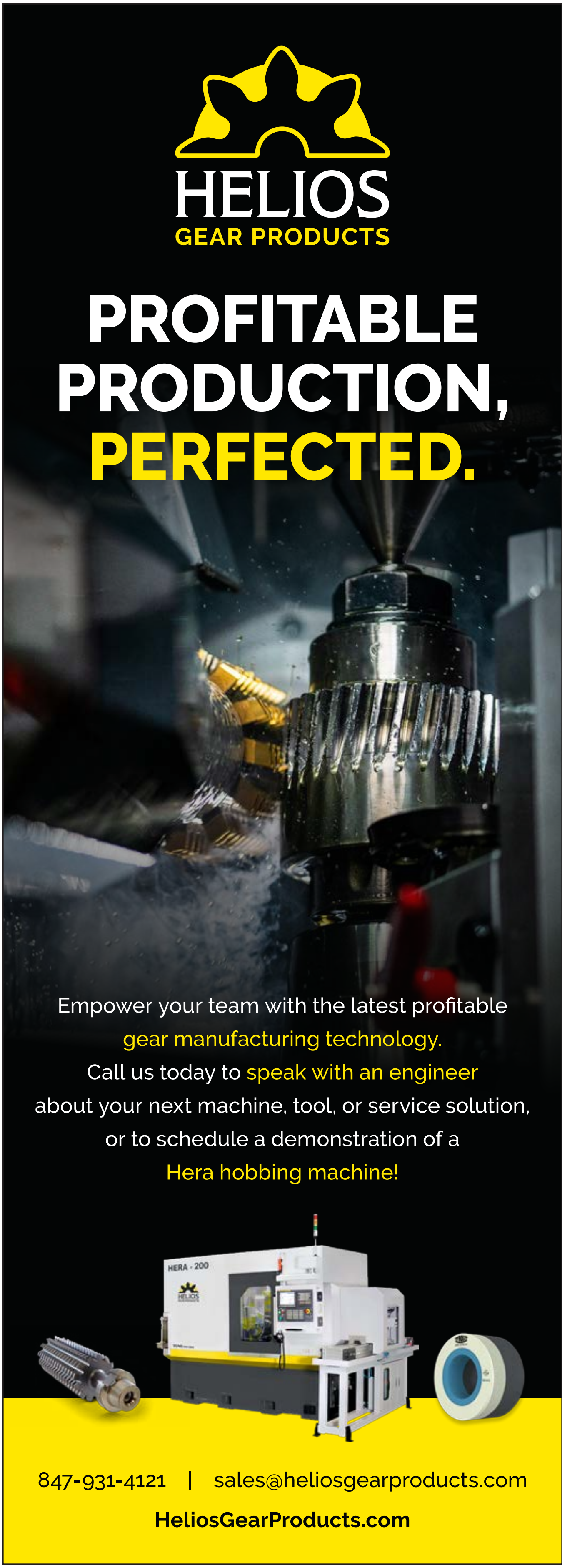

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Mitsubishi ZE26C

An inside look at Mitsubishi's ZE26C machine that finishes gears with outer diameters up to 260 mm. Superior performance has been achieved by reducing non-grinding time and maximizing grinding speeds. Learn more here:

www.geartechnology.com/videos/Mitsubishi-ZE26C-Gear-Grinding-Machine-/

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KX 260 TWIN

Kapp Niles KX 260 Twin HS

Machines of the KX TWIN series have been designed for continuous generating grinding with dressable and non-dressable tools. Profile grinding of small shafts can be achieved with the KX 260 Twin HS. Learn more here:

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Gear Talk with Chuck

In his recent blog entry "Learning the Ropes," Charles Schultz examines some potential suggestions for new hires as they begin their new career in gear manufacturing.

www.geartechnology.com/blog/learning-the-ropes/

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

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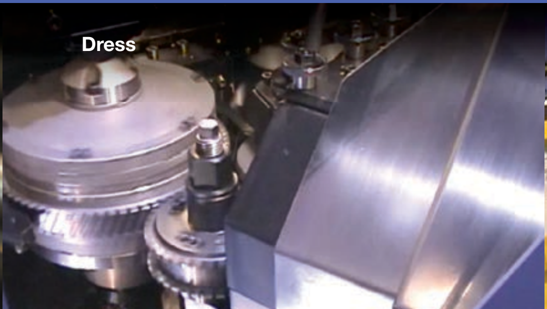
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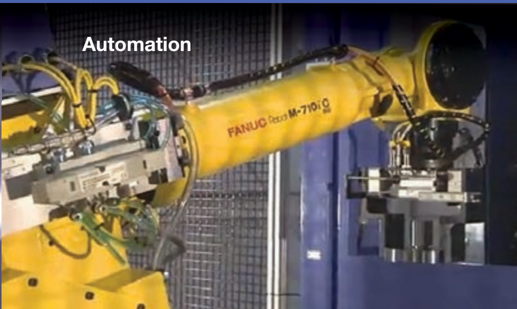
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Keeping in Touch. From a Distance.

Publisher & Editor-in-Chief
Randy Stott

I've been tuning in to a lot of information in new ways, lately. Chances are, you have, too. The pandemic has restructured all of our lives in ways we couldn't have imagined just a year ago. From work to school to family, video chat and teleconference have replaced warm hugs and handshakes. There's no kibitzing around the water cooler.

There's been a big learning curve, and we're still on that path. Although we now know how to Zoom or Webex or meet on Microsoft Teams (some days, it's all three!), most of us are still figuring out how to adjust to the new highly scheduled way we've had to structure our lives.

These changes were coming anyway. The need for social distancing and working from home has just made them happen a lot sooner than anyone thought.

Gear Technology is definitely here to help you stay connected. For starters, now is a great time to renew your subscription. If you're working from home, update your address so that the printed copy arrives where you need it. If you haven't done so already, sign up for the digital version of the magazine, so you can stay up to date on the latest in gear manufacturing technology and technical information whether your office is in the basement, garage or back yard.

At the same time, you can subscribe to our various e-mail products, including our twice monthly e-mail newsletter, product alerts and targeted messages from our advertisers. Each subscription is separate, so you can choose which e-mails you want to receive, and if you feel inundated by too much information, you can change your e-mail preferences or unsubscribe at any time.

To take advantage of these great information tools, just go to *www.geartechnology.com/subscribe.htm.*

Subscribing to our e-mail newsletters and advertiser e-mails is a great way to learn about some of the other digital opportunities available to you. Over the past several months, we've helped thousands of readers like you find and take advantage of numerous white papers, webinars and educational events that were targeted specifically to the gear industry. If you missed any of them, you can visit *www.geartechnology.com/sc/* to see some of those digital content offerings.

The newsletters are also a great way to follow the *Gear Technology* blog. Often, we post content there that can't be found anywhere else, including product information, case studies demonstrating the application of the latest gear related technology and "Gear Talk with Chuck," featuring the advice, musings and commentary of our resident blogger and longtime technical editor, Charles Schultz. (You can also check out the blog without subscribing to the newsletters. Just visit *www.geartechnology.com/blog/*)

We've also been trying to keep you updated about all the digital offerings of the American Gear Manufacturers Association. AGMA has shifted many of its highly sought-after training classes to digital formats, and the association also offers many webinars and networking opportunities online. Learn more at *www.agma.org.*

Of course, we're all looking forward to some return to normalcy. Sometimes it might not feel like it, but it's going to happen. In-person, face-to-face meetings will return in full force, and 2021 will be a much different year than 2020.

That's why the AGMA is declaring itself "ALL IN" for 2021. Please take a moment to read the *Voices* column in this issue (page 10). Written by John Cross, AGMA's chairman of the board and President of ASI Drives, Cross explains how the entire industry is coming together to support MPT Expo in October 2021. We're very much looking forward to getting together with as many of you as possible, and MPT Expo looks like it's going to be the best time and place to do so.

In the meantime, we'll help you keep in touch. Just go to *www.geartechnology.com/subscribe.htm.*

Randy Stott

All In for 2021

John Cross, Chairman of the Board, AGMA

As Knute Rockne once said, "When the going gets tough, the tough get going."

And so it is with AGMA members.

COVID-19 has been as tough an experience as many of the longtime AGMA members I have spoken to have ever seen. I know one 50+ year industry pro who said it plainly: "I have never seen it worse, ever."

That's the definition of "when the going gets tough." Forget the books, articles, and prognosticators, and absolutely ignore the media — I go with good old fashioned experience-defined wisdom to position where we currently are, economically. It's TOUGH.

And now the tough get going, period.

As AGMA Chair during this period of volatility, I have made it a priority to pivot the organization, assess where we are, and where we need to be, and ensure that our adjustments are 100% supportive of members today, and into the very near future.

I have an important message to deliver to you all — and I need your support to make it happen.

First — AGMA is ALL IN for 2021. We are making Motion + Power Technology 2021 THE power transmission event of the year. Frankly, after the cancellation of almost every single tradeshow in 2020, and the expected continuation of this through at least the first quarter of 2021 — we think AGMA's largest power transmission event will be the first time that we can get together as an industry.

> ***"AGMA is ALL IN for 2021. We are making Motion + Power Technology 2021 THE power transmission event of the year."***

So, WE ARE ALL IN for 2021 — and we want your leadership when it comes to focusing attention on this event as an exhibitor or as an attendee. Bring your colleagues and customers, and work with the AGMA team to make this a strong, positive event that truly brings us together.

Second — AGMA is ALL IN for DIGITAL CONNECTIONS in 2021.

AGMA now has three divisions — a Technical Division that connects the industry via standards and information sheets via 23 active committees and more than 400 participants; a Business Division that delivers meetings, education and market intelligence that no one else delivers in the industry; and, since January 2020, a Media Division that connects all of the pieces of what AGMA does via print and online media.

In 2021, AGMA will be directing resources to create synergy between the incredible customer-oriented resources we now have under one umbrella. From printed publications and online newsletters, to blogs, video and curated technical content — AGMA Media is a critical hub of information.

AGMA is ALL IN

- to help you position your company and its products to the right audience
- to ensure your company has the standards it needs to deliver to customers.
- to connect you to education, resources and market intelligence to grow your knowledge and your business.

And now — AGMA is ALL IN to do this digitally.

AGMA believes in its members — and believes in the core mission every single member rallies around: delivering power transmission innovation.

You are going to see AGMA celebrate our industry's efforts to be "ALL IN" via a series of advertisements, blogs and downstream marketing beginning in the fourth quarter of 2020, and running through 2021.

Frankly, it's time we all rally together and "get going" as we handle the challenging times we are facing. AGMA is here for you to guide you, support you and provide affordable opportunities.

WE'RE ALL IN — and you should be too.

John Cross
Chairman of the Board, AGMA
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COMBI HONING EDRIVE GEARS

DR. ANTOINE TURICH, DIRECTOR, PRODUCT MANAGEMENT OF HARD FINISHING SOLUTIONS, GLEASON CORPORATION

As the automotive industry moves toward e-mobility, transmission manufacturers are faced with new challenges. Larger gear ratios are necessary to reduce the high input speeds of electric motors to the required speed of the drive wheels. At the same time, gear noise that was concealed by the sound of a combustion engine is now evident, presenting completely new challenges for acceptable transmission noise levels. Finally, there are the special requirements to consider for the various new transmissions developed specifically for eDrive application. A common solution for eDrive transmissions are planetary transmissions using "stepped pinions", as shown in Fig. 1.

In specific planetary gear applications (Fig. 1), the two gears on the stepped pinion are synchronized to fulfill an exact timing within very tight tolerances.

Due to the noise sensitivity of such components, hard finishing by grinding or honing is indispensable. Gear honing proves to be particularly advantageous, since honed components have a proven lower noise behavior than ground components due to their specific, curved surface structure. Gear honing is also a requirement for machining gears with interfering contours, as is the case with stepped pinions. This is due to the small cross axis angle between the honing tool and the component and the fact that, unlike grinding, no tool overrun paths are required.

Combi Honing, New Possibilities

With the acquisition of the Faessler gear honing business, Gleason has added a unique process to its gear hard finishing portfolio that makes it possible to hone synchronized stepped pinions in one clamping with extremely tight tolerances and the highest quality. This so-called Combi Honing system uses two honing rings. The honing head of a Gleason 260HMS Honing Machine (Fig. 2), for example, can clamp two honing rings in parallel. The resulting eccentric offset of the honing rings is compensated for with a B-axis (swivel axis). In addition, flank line modifications such as crowning can be realized with the B-axis during the honing process.

The Combi Honing process starts with honing ring 1 honing the larger gear, and then honing ring 2 honing the smaller gear, all in the same clamping. Although this may sound trivial, this process has decisive and unique advantages, especially with regard to finished quality. While this specific component could also be machined in two separate set-ups, e.g. grinding the larger gear and honing the smaller one, the quality of the resulting gear would not be the same, particularly the angular synchronization of both gears. When finishing both gears in one clamping, non-productive time for loading/unloading as well as indexing (centering tools and gears) occurs only once and not twice per component.

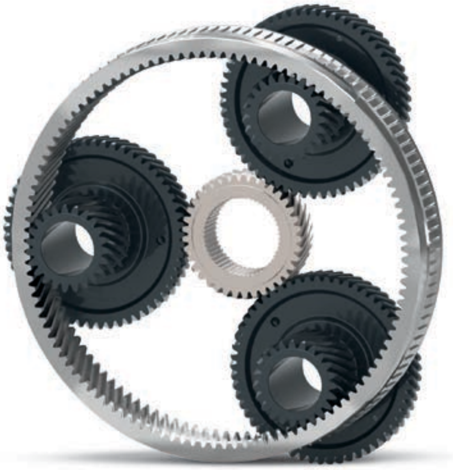

Figure 1 **Planetary transmission with stepped pinions.**

The Combi Honing Process on the 260HMS was specially developed for synchronized stepped pinion applications. A particular challenge was achieving the reliable and accurate positioning of the synchronized gears in relation to the honing rings. When indexing, i.e. centering gear teeth and tools, both teeth of the large and the small gear must be detected while corresponding exactly to the required angular offset and the tolerances of the index hole on the face side of the gear. The latter guarantees the final correct installation position of the stepped pinion in the planetary transmission. Three indexing sensors (Fig. 2, right hand side of the picture) are used to measure the position of all teeth of the large and small gear as well as the position of the index hole on the face side. A corresponding algorithm calculates the correct position of the gear teeth in relation to the honing rings. Parts with

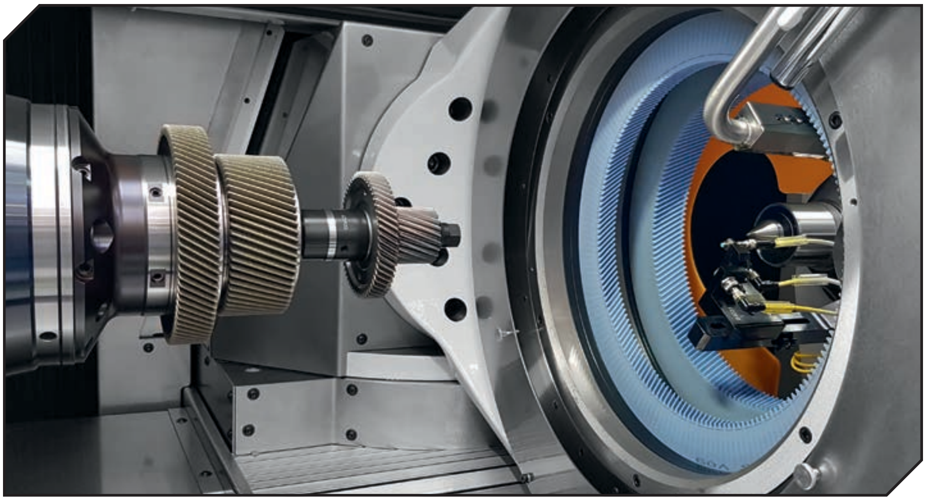

Figure 2 **260HMS can apply the Combi Honing process to produce stepped pinions reliably and at the quality levels needed to ensure the correct installation position in the planetary transmission. Integrated sensors measure the position of all teeth on both gear as well as the position of the index hole on the face side.**

excessive hardening distortions, which can't be honed in exact tolerances to the index bore, are automatically ejected.

Another important feature determining quality is the fixed position of the two diamond dressing tools on the work spindle (Fig. 2). The location of the dressing tools ensures that the position of the teeth on the honing rings does not change either absolutely or relatively — even after dressing of the honing rings. Loading/unloading of dressing tools to the work spindle, as is often the case in other honing applications, cannot reliably achieve this important quality aspect.

Polish Honing for Better Performance

Another advantage of the Combi Honing process is the possibility of super finishing gears with "Polish Honing." The requirements for increased transmission efficiency and reduced noise levels demand a superior surface quality of hard-finished components. While Polish Grinding using a two-zone polish grinding worm is a proven approach, a similar process has not, until now, been possible with gear honing.

With Combi Honing, however, it is now possible to use two honing rings in one clamping and thus use two completely different tool specifications for rough finishing and polishing of a gear (see Fig. 3). This makes it possible to achieve the surface qualities of Rz ≤ 1 µm typically required for polish grinding by means of gear honing — but with the added benefit of achieving the surface structures typical for the gear honing process.

For more information:
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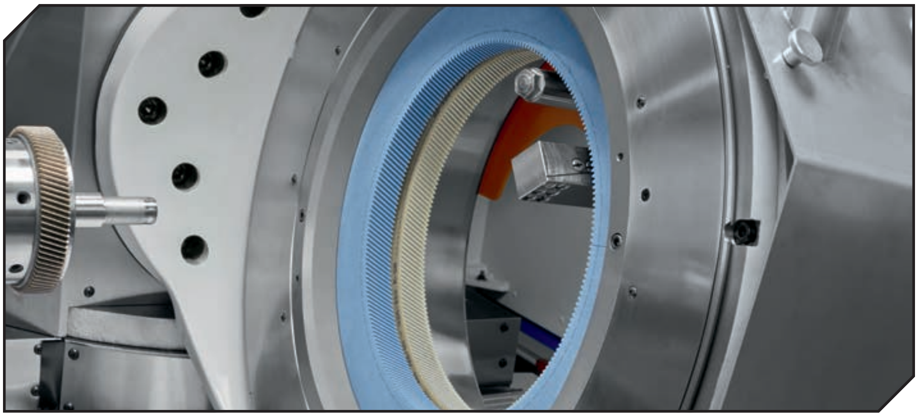

Figure 3 Super finishing by Polish Honing can now be done with Combi Honing, and using two honing rings for rough finishing and polishing of a gear.

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OFFERS EFFICIENT, QUICK-CHANGE WORKHOLDING

GREG MCLENNAN, SALES MANAGER-PRECISION WORKHOLDING, EMUGE CORP.

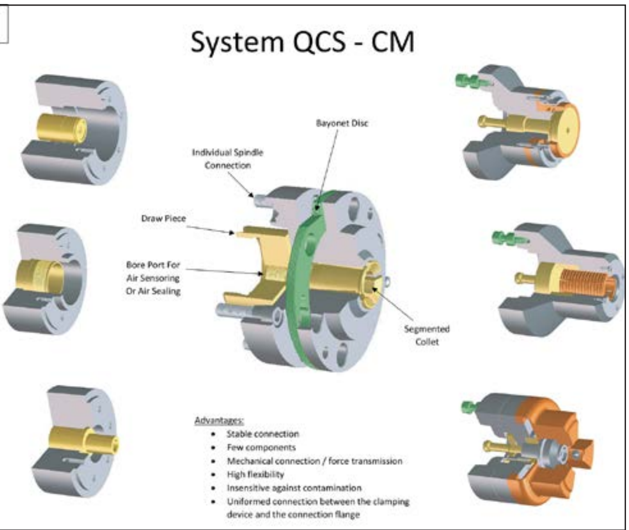
Controlling production costs is critical for today's gear manufacturers, especially when factoring in global economic pressures. One key effective cost savings measure is using a quality quick-change workholding system (QCS). A quality quick-change workholding system can increase machine uptime, resulting in greater production on existing machines. Also, by minimizing the changeover time from one workpiece to another, investing in quick-change workholding for new machines can often reduce the number of machines needing to be purchased.

Controlling or reducing costs is paramount in gear manufacturing facilities due to the considerable investments to produce today's high-quality gears. In gear manufacturing, being flexible with the ability to adapt to challenging workholding requirements is more important than ever. Utilizing a quick-change system across multiple operations is a great way to achieve this. For instance, the same arbor can be used for hobbing a green blank, as well as both end face grinding and finish tooth grinding operations after heat treat. This saves costs by not requiring an investment in complete dedicated arbor assemblies for multiple machines. Where production is high enough that machines run concurrently, having the same arbors saves on the amount of spare assemblies, spare parts and requires less operator and maintenance personnel training.

Emuge offers three quick-change designs, including the QCS-CM, QCS-CA and the QCS-HM, to suit a wide range of requirements. With all three quick-change systems, each machine spindle is fitted with an Emuge quick-change adapter. This allows for precision clamping devices to be interchangeably mounted with minimal changeover time. In most cases, after the clamping devices have been mounted to the adapter, there is no need to adjust the runout of the clamping devices. It is only necessary to check runout with the master to verify proper mounting to the adapter. And while the primary functionality is constant, the machine spindle and workpiece requirements will factor into the final design of the quick-change system.

The system QCS-CM (Fig. 1) utilizes a bayonet disc to uniformly secure the clamping devices to the adapter. Each clamping device has multiple studs that protrude through the bayonet disc. The bayonet disc is then rotated and locked into position, pre-mounting the clamping device to the adapter. The studs are uniformly tightened with a wrench, firmly mounting the clamping device to the adapter against the tapered connection flange. A draw piece attaches the drawbar of the machine spindle to a segmented collet within the EMUGE quick-change adapter. The segmented collet enables the machine spindle drawbar to operate various clamping methods, dependent upon the design of the clamping devices. If a customer requires air sensing or air pressure within the clamping device, a channel can be bored through the quick-change adapter. Air sensing is used to facilitate proper workpiece seating to the location points during loading and prior to centering the workpiece. This ensures the clamping device only operates when the workpiece is in the correct position. Air pressure is used for pressurizing the clamping devices to aid in preventing coolant from penetrating the device.

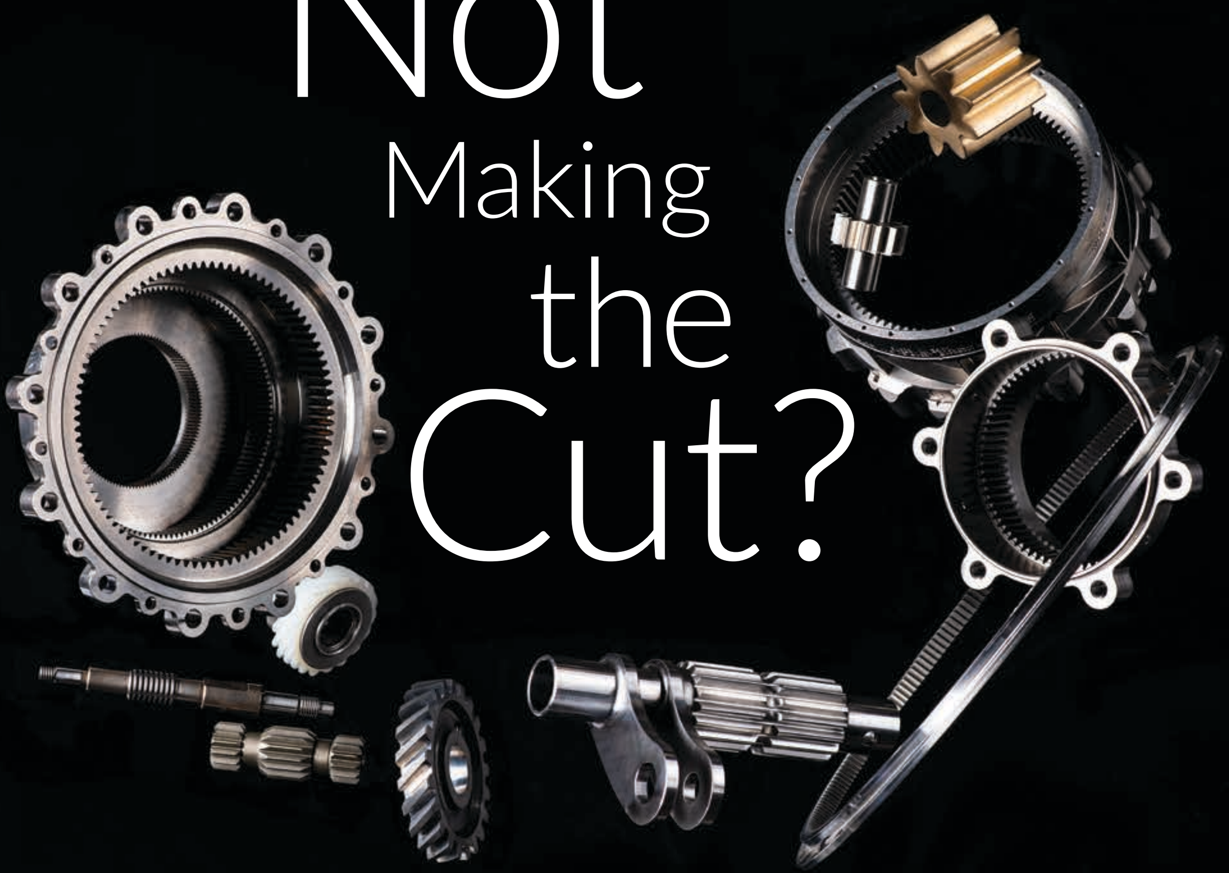
FIG. 1


System QCS - CM
Bayonet Disc
Individual Spindle Connection
Draw Piece
Bore Port For Air Sensoring Or Air Sealing
Segmented Collet
Advantages:
Stable connection
Few components
Mechanical connection / force transmission
High flexibility
Insensitive against contamination
Uniformed connection between the clamping device and the connection flange

The system QCS-CM utilizes a bayonet disc to uniformly secure the clamping devices to the adapter.

Advantages of the QCS-CM system include providing an extremely stable mechanical connection with few components, maintaining a highly flexible system with maximum force transmission from the machine spindle to the customer's workpiece, and excellent resistance to chips and coolant penetrating the connection during use.

The system QCS-CA (Fig. 2) uses a bayonet to hold the clamping devices onto the adapter, like the QCS-CM. The bayonet is actuated from an internal disc spring package and released from the machine spindle drawbar via the thrust bolt. When changing the clamping device, the spindle drawbar is moved to the full forward position, compressing the disc spring



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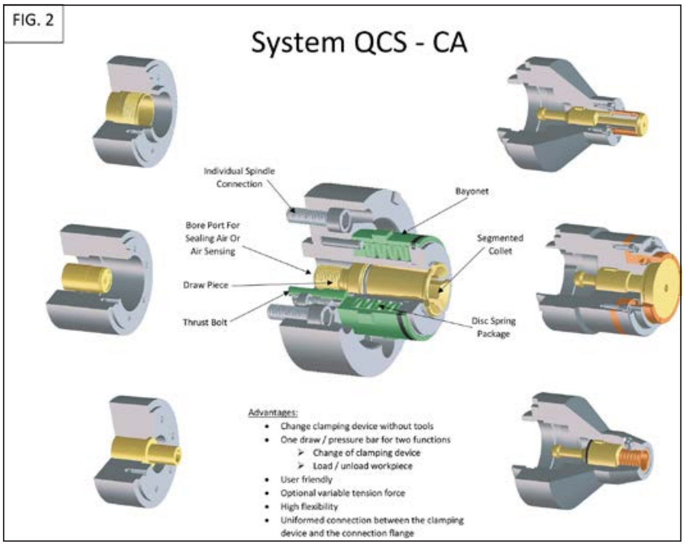
package and releasing the clamping device. The clamping device is then rotated until the bayonet lugs are aligned to the load/ unload slots and can be removed from the adapter. After the adapter and new clamping device are properly cleaned, the clamping device is inserted onto the adapter in the reverse order of which it was removed. Once the spindle drawbar moves to the operating position, the disc spring package holds the clamping device firmly against the connection flange. Also, similar to the QCS-CM system, there is a segmented collet and draw piece, which allows the machine spindle drawbar to actuate the clamping device and enables air sensing or sealing air to be supplied through the adapter.

Advantages of the QCS-CA system include the highly flexible, user-friendly changeover without the need for tools. Also, the drawbar can be used for two functions: changing the clamping device and loading/ unloading of the workpieces.

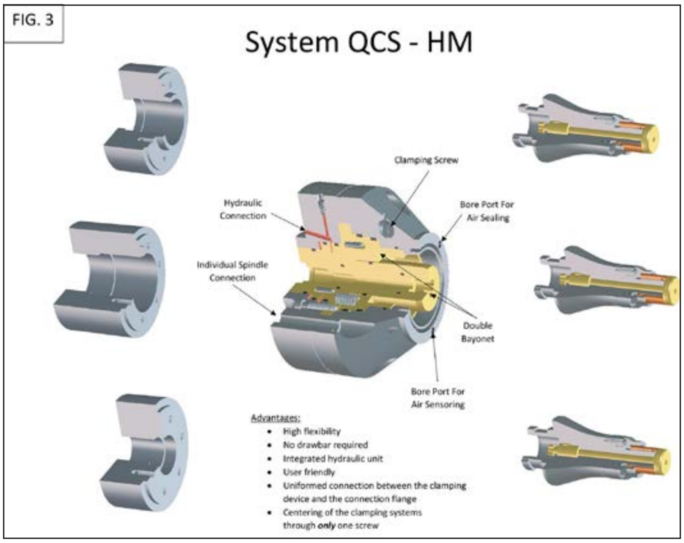
The system QCS-HM (Fig. 3) uses a double bayonet system without the requirement of a drawbar from the machine

FIG. 2


System QCS - CA
Individual Spindle Connection
Bayonet
Bore Port For Sealing Air Or Air Sensing
Segmented Collet
Draw Piece
Thrust Bolt
Disc Spring Package
Advantages:
Change clamping device without tools
One draw / pressure bar for two functions
Change of clamping device
Load / unload workpiece
User friendly
Optional variable tension force
High flexibility
Uniformed connection between the clamping device and the connection flange

The system QCS-CA uses a bayonet to hold the clamping devices onto the adapter, like the QCS-CM.

FIG. 3


System QCS - HM
Clamping Screw
Hydraulic Connection
Bore Port For Air Sealing
Individual Spindle Connection
Double Bayonet
Bore Port For Air Sensoring
Advantages:
High flexibility
No drawbar required
Integrated hydraulic unit
User friendly
Uniformed connection between the clamping device and the connection flange
Centering of the clamping systems through only one screw

The system QCS-HM uses a double bayonet system without the requirement of a drawbar from the machine spindle.

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spindle. The connection bayonet mounts the clamping device to the quick-change adapter. The second bayonet, the clamping bayonet, actuates the clamping device.

The connection bayonet uses springs and a hydraulic unit contained within the quick-change adapter to attach the clamping device to the quick-change adapter. To remove the clamping device from the adapter, the operator rotates one clamping screw charging the internal hydraulic system. This pressure compresses the spring force allowing the release of the clamping device. Next the clamping device is rotated, aligning the bayonet lugs from the clamping device with the loading/unloading slots in the bayonet adapter. The clamping device is now easily removed from the quick-change adapter. After proper cleaning of both the quick-change adapter and the next clamping device, assembly takes place in reverse order of the removal process.

The clamping bayonet is actuated from the machine spindle hydraulic pressure connection. As the hydraulic pressure is applied to the quick-change adapter, the unclamping spring force is overcome, enabling the bayonet to pull on the clamping device drawbar and achieve workpiece clamping. After the machining operation is finished, the hydraulic pressure is removed for the unclamping springs to ensure the return of the drawbar and removal of the workpiece from the clamping device has been completed.

Advantages of the highly flexible QCS-HM system include the user-friendly centering of the clamping device by using only one screw, and fitting to machines where no draw bar is required.

Low Mix, High Mix—No Problem

All three QCS systems achieve more machine up time with smaller lot groups of workpieces, as well as facilitate easier preventative maintenance of the clamping devices. In smaller lot groups of workpieces, the Emuge clamping device can be mounted to multiple machine operations, reducing the number of clamping devices required. The clamping device can be moved from one machining operation to another, with minimal to no components needing to be exchanged. Where a machine is dedicated to one workpiece, having a quality quick-change workholding system installed will allow for the removal of one clamping device for preventative maintenance, while another identical clamping device can be installed to keep the machine running.

Emuge QCS systems are also suitable for a family of workpieces similar in size and geometry. Gear manufacturers can use the quick-change system to keep the machine running, while they are preparing for the next workpiece to run.

Whether production ranges from high volume using machine dedicated clamping devices, or smaller lot sizes that are changed over frequently, Emuge quick-change systems can be customized to meet the requirements of keeping production costs down and machine run-time up.

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Mitutoyo America

LAUNCHES CRYSTA-APEX V SERIES

Mitutoyo America Corporation is pleased to announce the release of the CRYSTA-Apex V Series to its Coordinate Measuring Machine product line. The new generation of the CRYSTA-Apex Series will provide the highest speed, accuracy, precision, and versatility in the current Mitutoyo CNC coordinate measuring machine lineup. The launch of the CRYSTA-Apex V also introduces the Mitutoyo Gold Care PLUS Productivity Program.

Key features:

High-Speed/High Acceleration/Drive: Features a maximum measuring speed significantly faster than competitive CNCs. The combination of high speed and high acceleration dramatically reduces measuring time and cost.

Shorter Measurement Times: Users can set measuring paths for high speed scanning, 3D and active scanning of complex workpieces.

Temperature Compensation System: Guarantees measurement accuracy for temperature conditions between 16 to 26° C. Independent sensors work in tandem to measure scale, workpiece and ambient temperature. Results are superior to comparable systems that only compensate for scale temperature.

Smart Factory Functionality: The CRYSTA-Apex V series utilizes three smart factory applications, consolidating the information management manufacturing process within a network: Status Monitor allows remote monitoring of the operational status of measuring instruments; Condition Monitor enables remote monitoring of the current condition of measuring instruments; MeasurLink reduces the production of defective parts through "visualizing quality" via complete data management.

Smart Measuring System (SMS): Allows online monitoring of the measuring status and visualization of measurement data, enabling product quality improvement and internet of things capabilities.

Flexibility: CRYSTA-Apex V Series can adapt to work on a range of sizes for small- to medium-sized work pieces. It also features multi-sensor capabilities and compatible vision and scanning probe technologies.

Gold Care PLUS is an updated version of the Mitutoyo legacy productivity program, Gold Care. Gold Care PLUS continues many of the Gold Care features while adding MeasurLink V9, Status Monitor with integrated MT Connect, Quick Launcher, Fixture builder, a 12A power conditioner and an air dryer.

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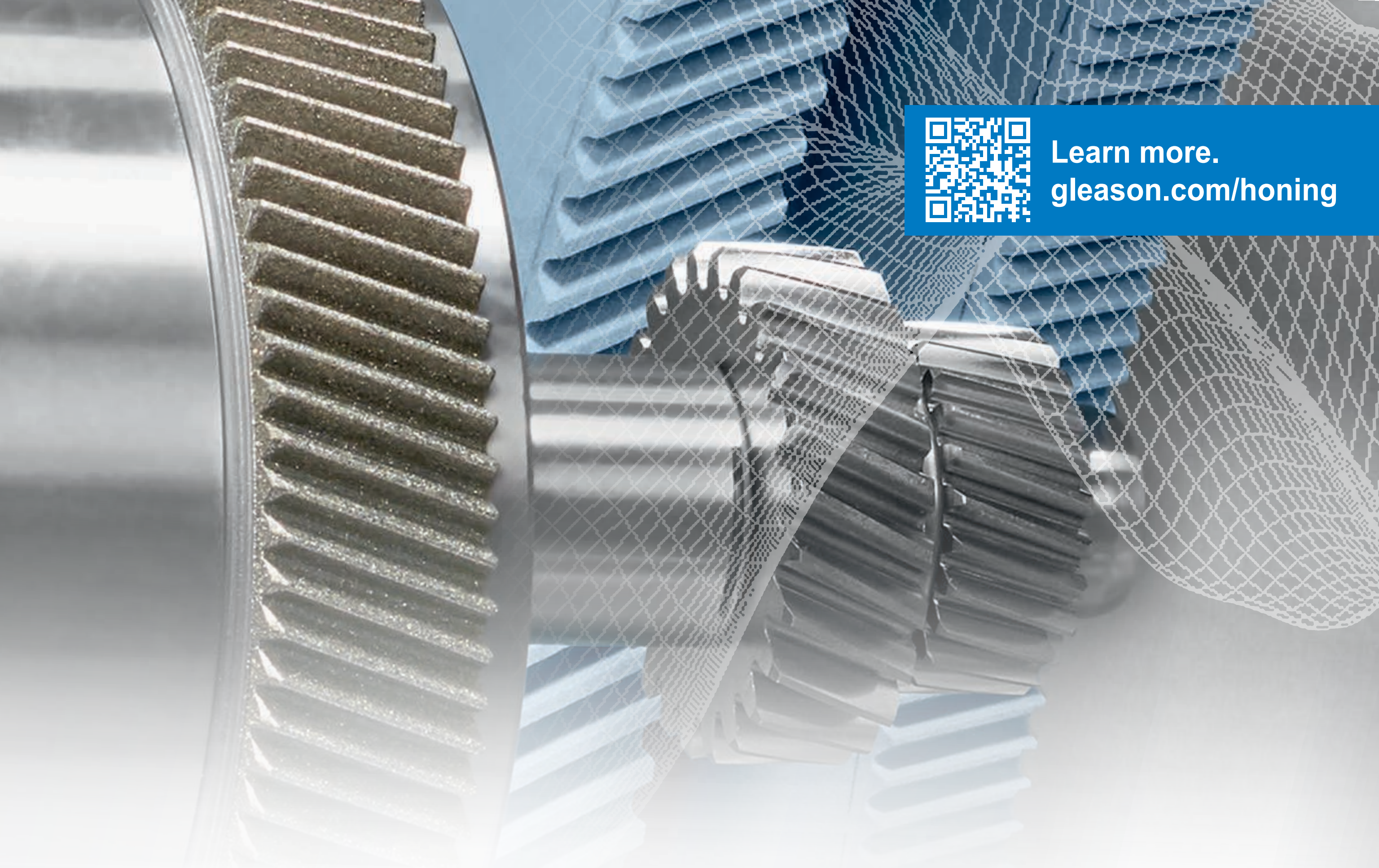

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Schunk

INTRODUCES ITENDO SENSORY TOOLHOLDER

Schunk is putting the power of data in your hands with the new iTENDO sensory toolholder. For the first time, it will be possible to monitor machining processes at high resolution at the tool and to control cutting parameters in real time. The required acceleration sensor and electronics are integrated into the toolholder without affecting its interfering contour and other characteristics.

The iTENDO seamlessly records the metal cutting process, monitors previously defined exact limit values and, in the event of irregularities, enables real-time adaptive control of the speed of rotation and feed rate, among other measures. Equipped with a sensor, battery, and transmitting unit, the intelligent toolholder records the data at the tool and transmits it wirelessly via Bluetooth to a receiver unit in the machine room, where it is forwarded by cable to a control and evaluation unit. This makes the system fundamentally different to other solutions for process monitoring by providing precise process data. In pilot applications, the intelligent mounting has proven performance for milling, drilling, countersinking and even deburring.

This information collection closes the loop on industry 4.0 machine processes and takes the guesswork out of machine adjustments.

Starter set for simple commissioning

In a first step, Schunk is standardizing the iTENDO for the common interface HSK-A 63 with clamping diameters from 6 mm to 32 mm and a length of 130 mm. The sensory toolholder is suitable for the use of coolant and is designed for speeds of up to 10,000 rpm. The commissioning and data analysis is carried out via a browser-based dashboard on standard PCs, tablet computers or smartphones. In the simplest configuration, which can be implemented completely without machine-side adjustments, the live data from the sensor can be displayed on the Schunk dashboard via a local connection. For this purpose, Schunk provides a special case system with integrated display, enabling toolholder commissioning within two hours and with minimal effort. In a second configuration, the real-time controller is ideally connected to the machine control system by a service technician via digital or analog I/O so that, for example, alarms can be triggered or processes can be adaptively controlled. The third and most sophisticated configuration enables additional information exchange with the machine (e.g. in the case of the latest Siemens control system via OPC UA). All variants can also be operated and centrally controlled via a cloud solution.

For more information:
Schunk
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NUM

LAUNCHES DIGITAL TWIN TECHNOLOGY

CNC specialist NUM has launched digital twin technology that enables machine tool manufacturers to reduce their time to market dramatically, by using powerful Industry 4.0 simulation techniques.

Originally known as pairing technology, and first used by NASA in the early days of space exploration, digital twin technology is now rapidly gaining industry acceptance as one of the most cost-effective means of accelerating the development of products, processes and services.

For automation products such as machine tools, a digital twin is a virtual model that uses simulation, real-time data acquisition/analysis and machine learning techniques to allow full evaluation of a machine's dynamic performance before constructing a physical prototype. The same technology can also be employed for customer presentations, virtual commissioning, and operator training purposes — and all well before the actual machine itself has even been built.

NUM offers two versions of digital twin technology, to best suit customers' needs.

Both versions are designed for use with NUM's powerful, open-architecture Flexium+ CNC platform. One version

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uses a naked Flexium+ controller and resident virtualization software running on the system's industrial PC to simulate the twinned machine automation. The other version uses the actual Flexium+ controller that will eventually be incorporated in the machine, linked via EtherCAT to a standalone PC running specialist high speed hardware simulation software to represent the mechatronics of the twinned machine.

The virtual controller version includes a software development kit for creating the software model of the machine. The model is a standalone PLC program that uses predefined components to simulate individual machine elements, such as sensors, spindles, pneumatic cylinders, etc. It is loaded into the integrated PLC of the Flexium+ controller.

The Flexium NCK in the controller executes the NC programs and simulates the changing position values of the machine's axes. To help users visualize the process, NUM's package includes the CODESYS Depictor software tool produced by CODESYS GmbH, which is used to produce 3D visualizations from the IEC 61131-3 code created by the simulation.

The other version of NUM's digital twin technology package accommodates real-time data acquisition and analysis. It is based on the ISG-Virtuos hardware simulation software produced by Industrielle Steuerungstechnik GmbH (ISG). The Flexium+ controller that is intended to be used in the physical machine is connected via an EtherCAT network to a standard PC and interacts with the simulation software in real-time. The PC acts as the twinned virtual machine — with all simulated, virtual components behaving like real components in terms of their interfaces, parameters, and operating modes — to accurately replicate the structure and dynamic performance of the real machine. The movements of the machine are displayed realistically on the PC, using the supplied 3D simulation software.

NUM's new digital twin technology provides machine tool manufacturers with a very powerful and cost-effective means of reducing their developments costs and accelerating their time to market. The virtual controller version is especially useful for the early development stage of a project, before the CNC system has been finalized, while the real-time hardware simulation version has the advantage that all sequencing (PLC) and motion control (CNC) programs that are created during development can simply be transferred to the real machine as soon as it becomes available.

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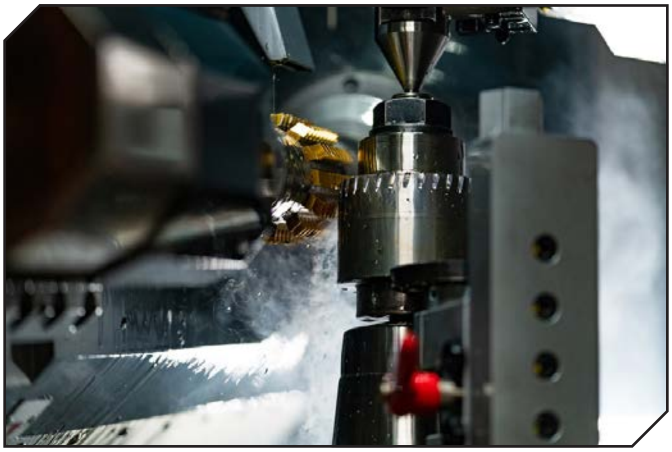
Helios

INTRODUCES HERA 200 CNC GEAR HOBBING MACHINE



Helios Gear Products exclusively offers the Helios Hera 200 CNC gear hobbing machine. This machine, manufactured by YG Tech, provides gear manufacturers a vertical hobbing solution with its siblings, the models 150, 350, and 500. The model 200 completes the fine- to medium-pitch range of the Hera series with the same standard of proven technology. Adam Gimpert, president of Helios, remarked, "We are excited to bring the new Helios Hera 200 to North American gear manufacturers who need an updated, cost-effective, profitable hobbing platform for gears up to about 8 inches — all with 30+ years of domestic support expertise."

The Hera 200 offers 6 axes (7 with automation) of Fanuc CNC, a 4 module (6.35 DP) pitch rating, and 1,200 rpm maximum hob speed. This combination comes on a base that handles parts up to 200 mm (7.874 in) diameter and 530 mm (20.866 in) length with up to 250 mm (9.840 in) axial travel. With hob shifting up to 180 mm (7.087 in), this machine offers versatility for short- or long-run jobs. Said David Harroun, vice president of Helios, "The Hera 200 offers many of the great features found on a Hera 350, but in a more compact size with a smaller financial footprint." The machine offers safety features such as electro-mechanical interlock and a splashguard door. The machine's cast iron base provides superior dampening and stability for extreme cutting conditions.

Software on the Hera 200 provides gear manufacturers simple dialog programming with visual examples to help guide operators and accelerate training. Cycles include cutting of one or two gears on a single workpiece using single- or two-cut cycles with radial, axial, climb, or conventional hobbing (or any combination thereof). Additionally, crowning (lead modification) and automatic shifting over a damaged hob section are included in the machine's base package.

With Helios and YG Tech, the Hera series comes with over a half-century of gear machine tool expertise.

For more information:
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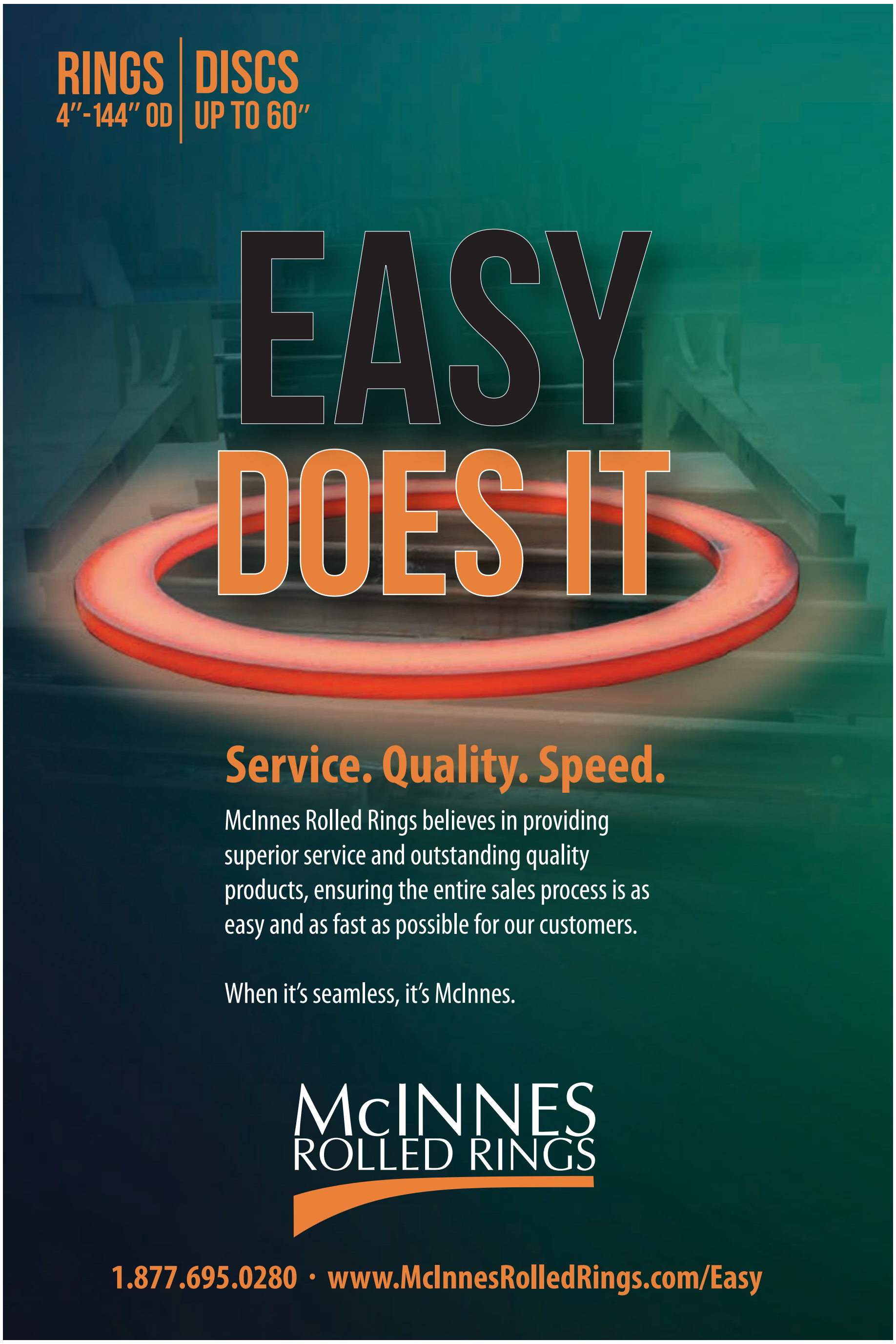

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Best Practices to Improve Results in Gear Deburring

Rick Sawyer, OEM & technical business development manager, Weiler Abrasives

All cutting and machining operations produce some type of burr or leave sharp edges on metal components.

These unwanted by-products are especially troublesome when producing precision components such as gears. The burrs can loosen from the gear — either during assembly or later when the gear is in operation — and damage components or lead to critical part failure.

Burr removal and edge prep are important processes performed during the manufacture of gears. These processes improve the quality, performance and reliability of the gears.

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Burr removal and edge prep are important processes performed during the manufacture of gears. These processes improve the quality, performance and reliability of the gears. In this article, we'll learn more about some best practices for gear deburring.

Why is gear deburring important?

Gear deburring is very prevalent in the automotive industry — especially for large trucks — as well as in the production of agricultural equipment, construction equipment and aerospace components.

Under operating conditions, sharp edges become areas where internal stresses concentrate and failures become more likely. If corners are left very sharp or there's any sort of crack or nick, it will promote failure.

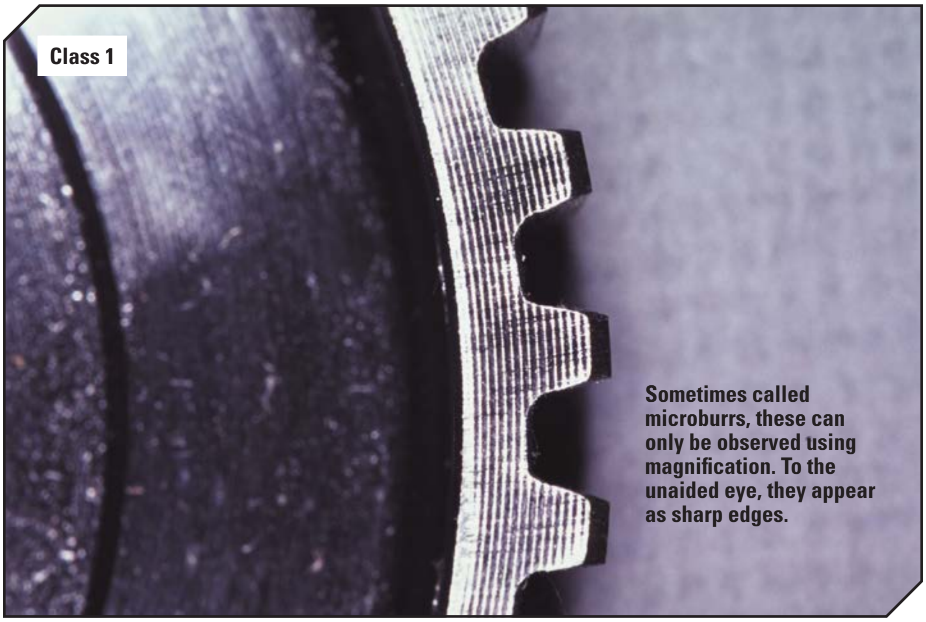
Deburring and radiusing eliminate edge defects, minimize stress risers and contribute to better mesh and lower operating noise between individual components.

Failure to remove burrs can result in quality issues or potential breakdowns later — often at great expense.

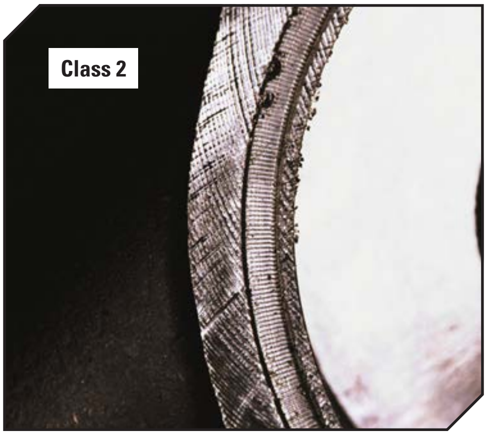
5 burr classifications

Understanding the size, shape and orientation of burrs to be removed can help in choosing the right equipment and media for the process. Burr classifications produced by common metalworking processes are:

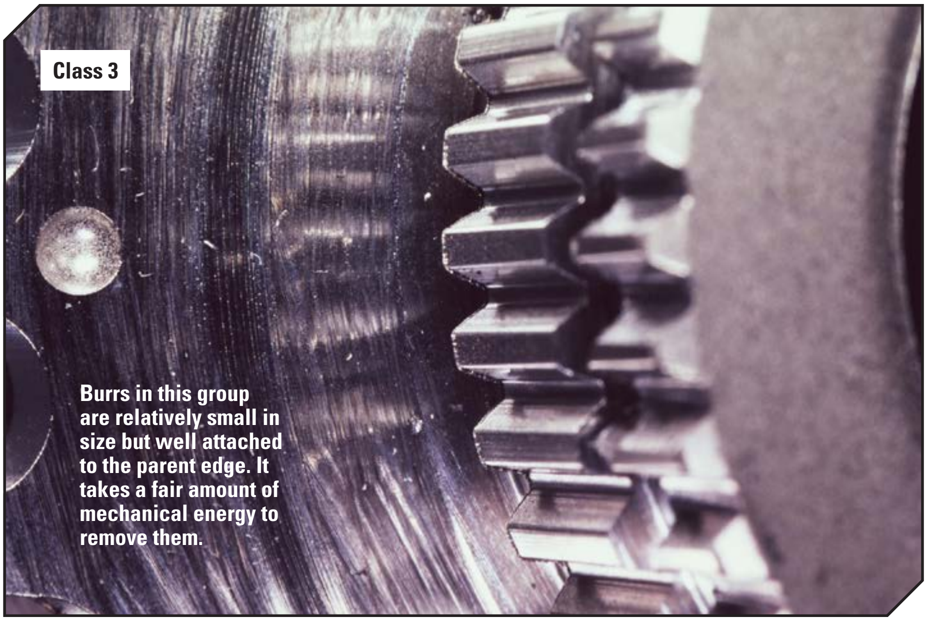
- **Class 1:** Sometimes called microburrs, these can only be observed using magnification. To the unaided eye, they appear as sharp edges. Grinding operations are a common source of this type of burr. Grinding operations are capable of holding tight tolerances while providing very good surface finishes.
- **Class 2:** These feather burrs are readily visible without magnification and characterized by extremely thin roots. They can be easily removed.
- **Class 3:** Burrs in this group are relatively small in size but well attached to the parent edge. It takes a fair amount


Class 1

Sometimes called microburrs, these can only be observed using magnification. To the unaided eye, they appear as sharp edges.


Class 2

These feather burrs are readily visible without magnification and characterized by extremely thin roots. They can be easily removed.


Class 3

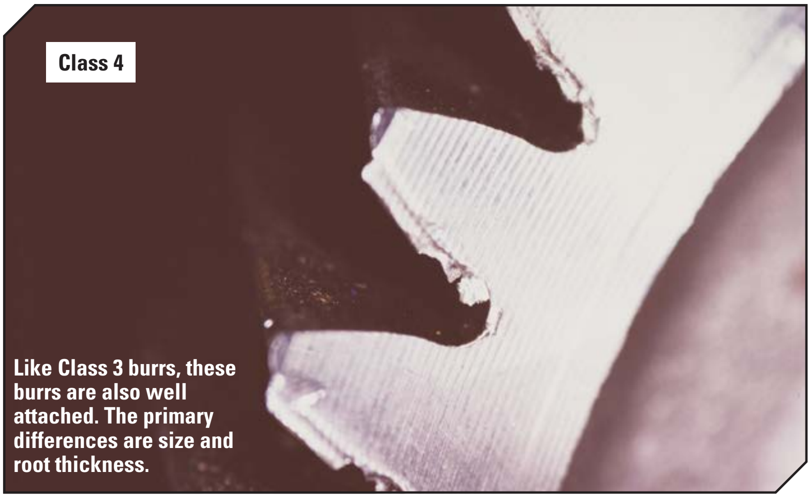
Burrs in this group are relatively small in size but well attached to the parent edge. It takes a fair amount of mechanical energy to remove them.

of mechanical energy to remove them.

- **Class 4:** Like Class 3 burrs, these burrs are also well attached. The primary differences are size and root thickness. Because of this, a significant amount of mechanical energy must be applied to remove them.
- **Class 5:** These burrs are very large with thick, rigid roots. Burrs in this class are different than conventional burrs because they are comprised of displaced base material that is still fully-attached to the parent part. Stock removal operations such as grinding or machining are required for complete removal.

The burr's root thickness and how well attached it is to the part are key factors in determining what type of media can be used for removal. The thicker the root (attachment to the part), the more energy it will take to remove the burr.

If an operation is consistently seeing larger Class 5 burrs, it's likely a sign that something should be adjusted upstream in the process.



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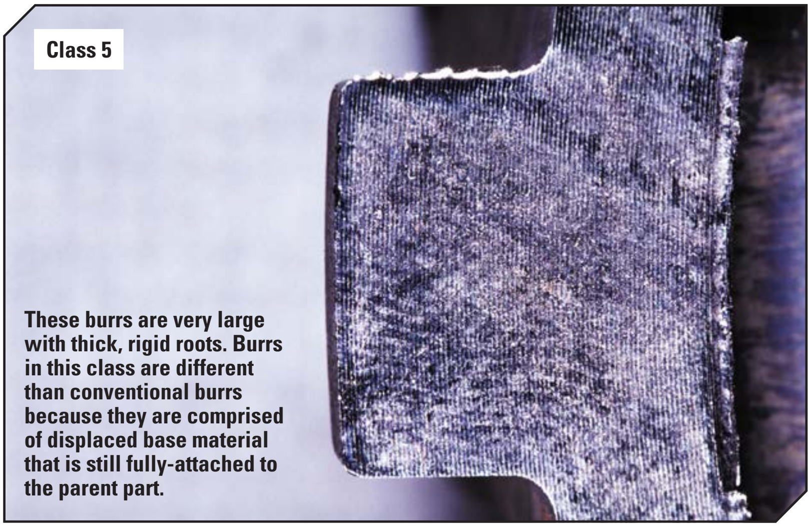

These burrs are very large with thick, rigid roots. Burrs in this class are different than conventional burrs because they are comprised of displaced base material that is still fully-attached to the parent part.

Filament selection for deburring

Brushes are well suited for removing many types of burrs, as they can be used in semi-automated and fully automated deburring processes.

Brushes offer greater conformability than most media types and don't alter part geometry. The moving filaments of a brush concentrate their collective kinetic energy on the edges of the part. They can remove the burr and unwanted edge defects while blending the edge and maintaining the shape of the gear. Compare that to using a cutoff wheel or grinding wheel, both of which will generate a secondary burr.

To optimize the process, the most important factor is to use the right media for the spindle rpm range and horsepower of the system. In order from least aggressive to most aggressive, the three brush configurations commonly used for gear deburring are:

- **Nylon abrasive filament brushes:** Think of these brushes as a collection of flexible files. Each nylon filament contains thousands of abrasive grains that act as little cutters. Although these filaments contain the same grain types as other abrasive products, the flexible nylon carrier does not forcefully apply them to flat surfaces. However, when they encounter an edge, these abrasive grains act like the teeth on a file, removing burrs and generating small edge radii.
  Due to the flexibility of the nylon bristles, the aggression of abrasive filament brushes can be somewhat limited. Different abrasive grain types and grit sizes, as well as brush media featuring different filament sizes, fill densities and trim lengths, are all variables that can be used to tailor the brushing action to the requirements of the application.
  Nylon abrasive filament brushes can be used to remove Class 1 and 2 burrs, and sometimes Class 3.
- **Crimped wire brushes:** These brushes use an impact-driven process similar to grit blasting. The focused velocity of millions of sharp wire tips striking an edge within a short period of time strips away burrs and peens sharp edges. Brushes with crimped wire offer a more compliant brushing action than knotted wire. Because wire brushes can focus a great deal of mechanical energy on an edge, they are best suited for removing large Class 3 and some Class 4 burrs. The crimp design is more capable of absorbing geometry changes than the knotted design. One limitation of these brushes is they will likely change surface finish.
- **Knot wire brushes:** These brushes work similarly to crimped wire brushes, but knot wire brushes are not as compliant. The twisted wire strands maximize the impact of the wire tips on the work and can handle larger burrs with a greater root thickness.
  In higher volume Class 3 and 4 burrs, knotted brushes will typically provide a lower cost per part due to parts per brush and shorter cycle times.

Wire brushes leave a peened or chamfered edge configuration, while nylon abrasive brushes provide more of a radiused edge.

When removing larger burrs with nylon abrasive brushes, it may work best for the system to use coolant. This allows the brushes to be run with a greater depth of interference (DOI) and at higher rpms, both of which increase aggression. Coolant prevents nylon transfer, which can occur when the abrasive filament generates too much heat and melts the nylon to the part's surface.

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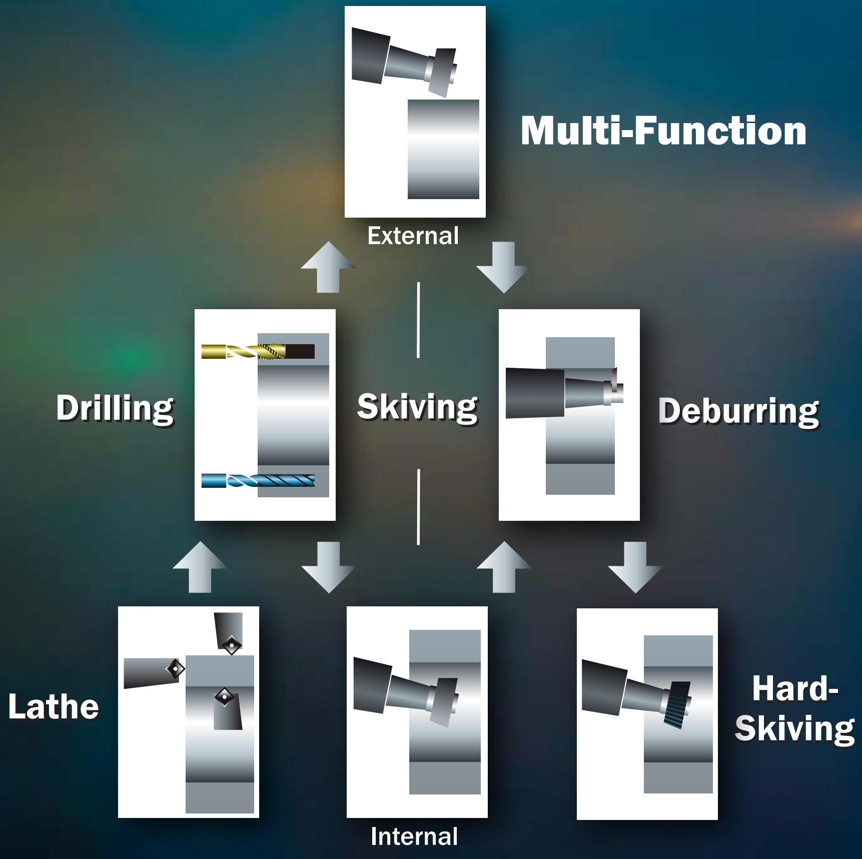
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Best practices for gear deburring

Choosing the proper media for the system and application is one of the most critical factors, but there are several other tips that can help improve results in gear deburring.

1. **Proper orientation.** Burr location determines the proper orientation of the brush to the part. The brush filaments must have direct access to the burr, and the edge must be oriented properly, relative to the direction of filament movement. The biggest factor is to have the brush be as perpendicular to the edge that is being deburred as possible — within 15 degrees plus or minus to 90 degrees for efficient performance. Brush orientation also affects product life and wear. Product life is directly dependent on the amount of penetration into the face (DOI). Minimizing brushing pressure reduces filament fatigue in wire-filled brushes and filament wear in nylon abrasive brushes.
2. **Operating parameters.** Before selecting the operating parameters for a deburring application, determine brush size, since the velocity at which the filaments strike the work is a function of spindle speed and product diameter. Typically, larger diameter brushes are preferred even when dealing with small parts, because they lower the consumable costs per part and increase production stability. A smaller brush will take longer to produce the desired results and require more frequent changeovers. Using a 3-inch brush may produce 100 parts, but operations looking to deburr thousands of parts will find a 10- or 12-inch brush more cost effective. One exception to this is when a small diameter brush is needed to access an edge due to the part geometry. Look for a chart from the brush manufacturer that shows recommended operating parameters for wheel brushes based on diameter and fill material. Wire filled brushes should be used at higher surface speeds with a minimum amount of penetration. Nylon abrasive brushes require lower surface speeds and greater amounts of penetration to allow the filaments to smoothly file across the target edges.
3. **Troubleshoot aggression.** If the recommended speeds and parameters don't work in a particular application, it may be necessary to troubleshoot and increase or decrease aggression. There are several steps that can be taken when the brush is too aggressive or not aggressive enough — both for wire brushes and for nylon abrasive brushes. These options include increasing or decreasing operating speed, changing the type of brush being used, and changing the brush trim length or fill density.
4. **Match equipment to the process.** Don't buy deburring equipment without knowing what the media can do. If the machine is the wrong speed or horsepower, it won't provide full efficiency with the tools. Make sure the process parameters are known before the equipment is purchased to ensure the proper parameters are in place from the start.

Improve performance in gear deburring

Better understanding burr classifications and what products are best suited to remove each type can help improve results and performance in gear deburring operations. Using the right media and parameters for the deburring application will extend product life and improve quality while saving time and money.

For more information:

Weiler Abrasives
Phone: (800) 835-9999
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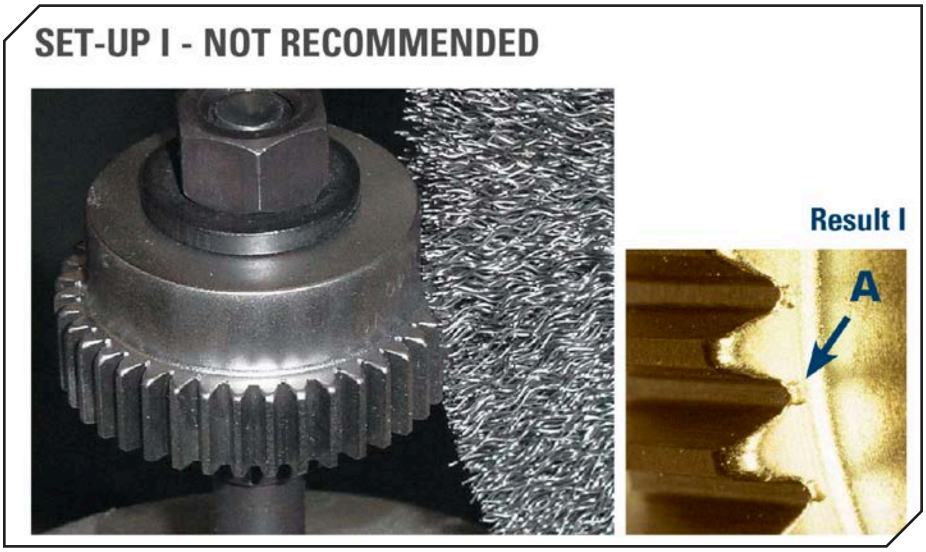
Rick Sawyer is an OEM & Technical Business Development Manager at Weiler Abrasives. He is a manufacturing engineer with an extensive background in deburr automation, especially with robotics, and has worked for three media manufacturers during his career. He has been employed with Weiler Abrasives for over 30 years.



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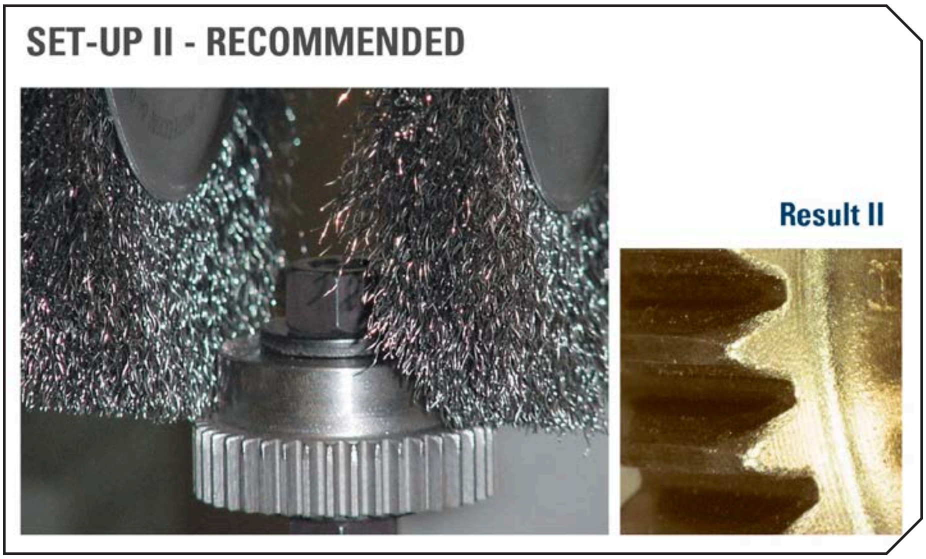

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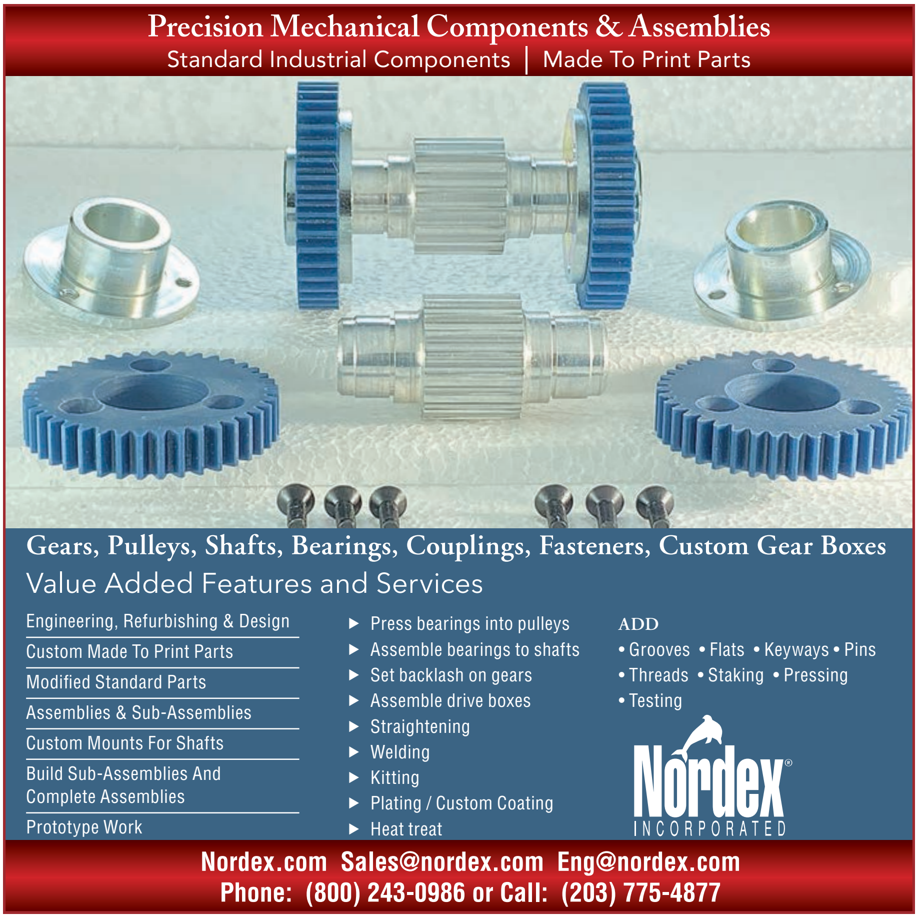

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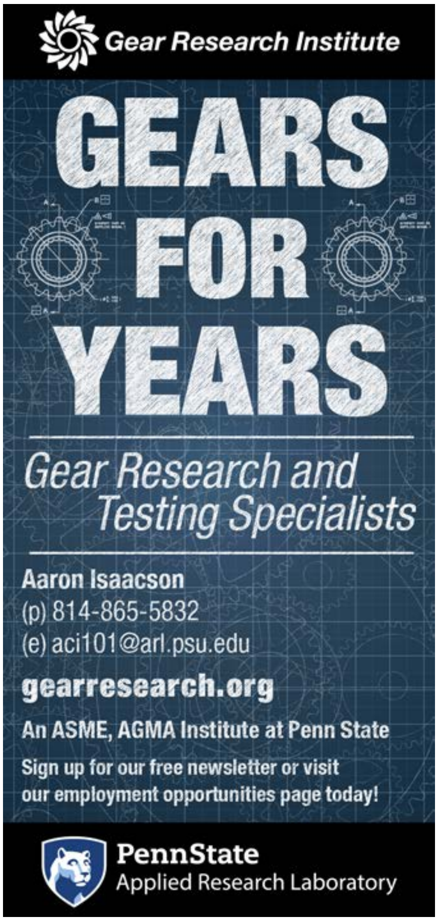

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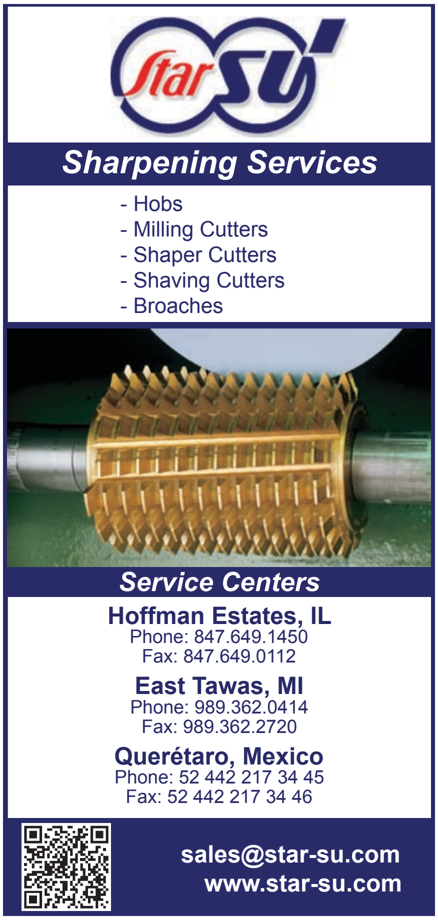

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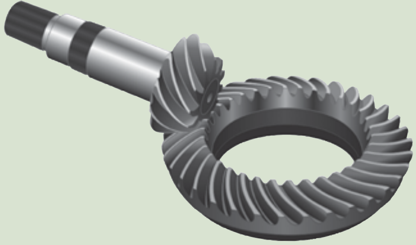

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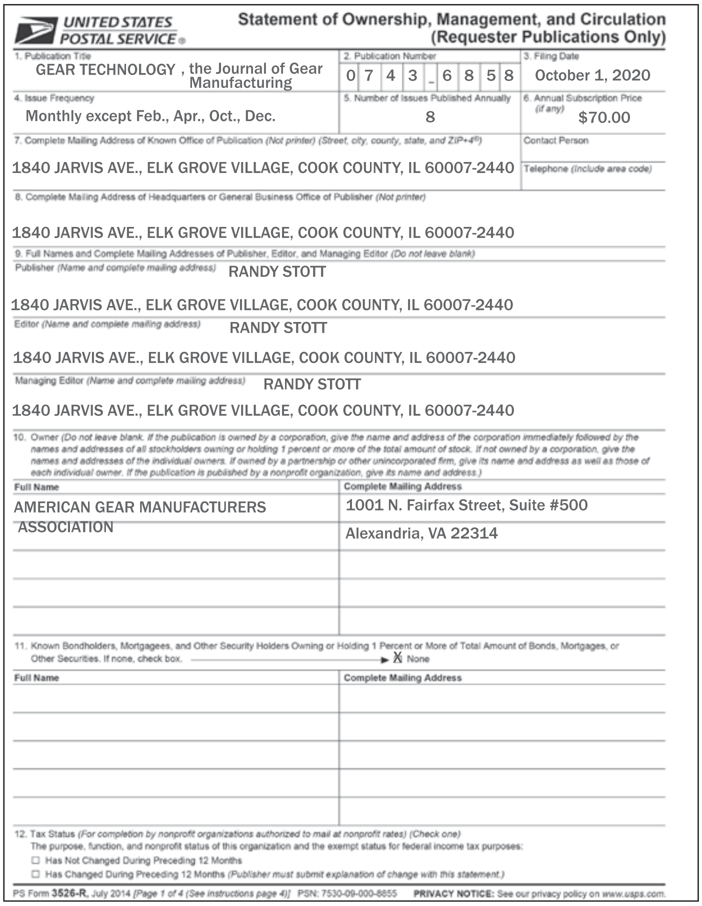

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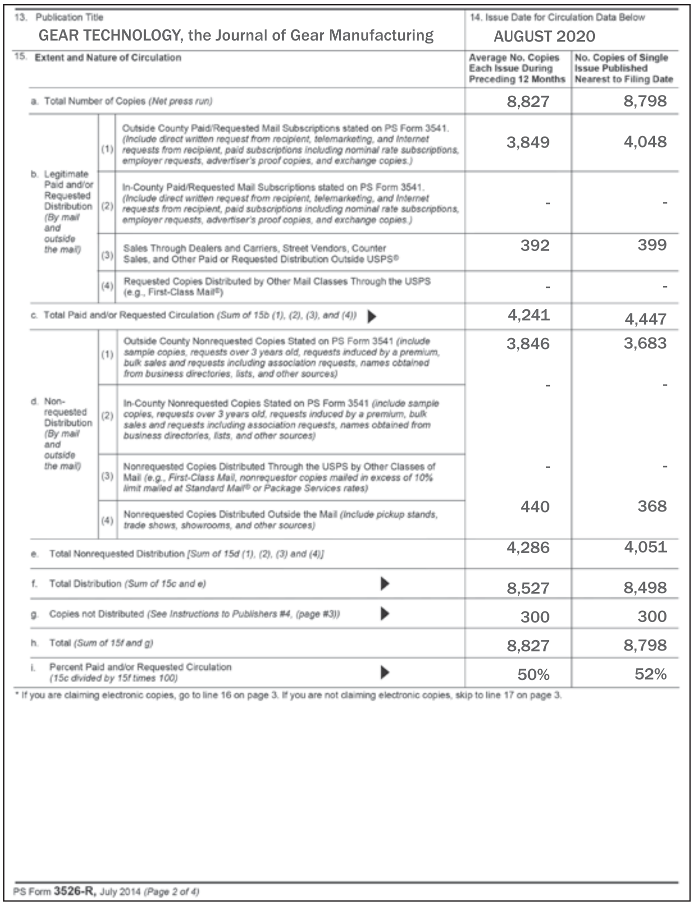

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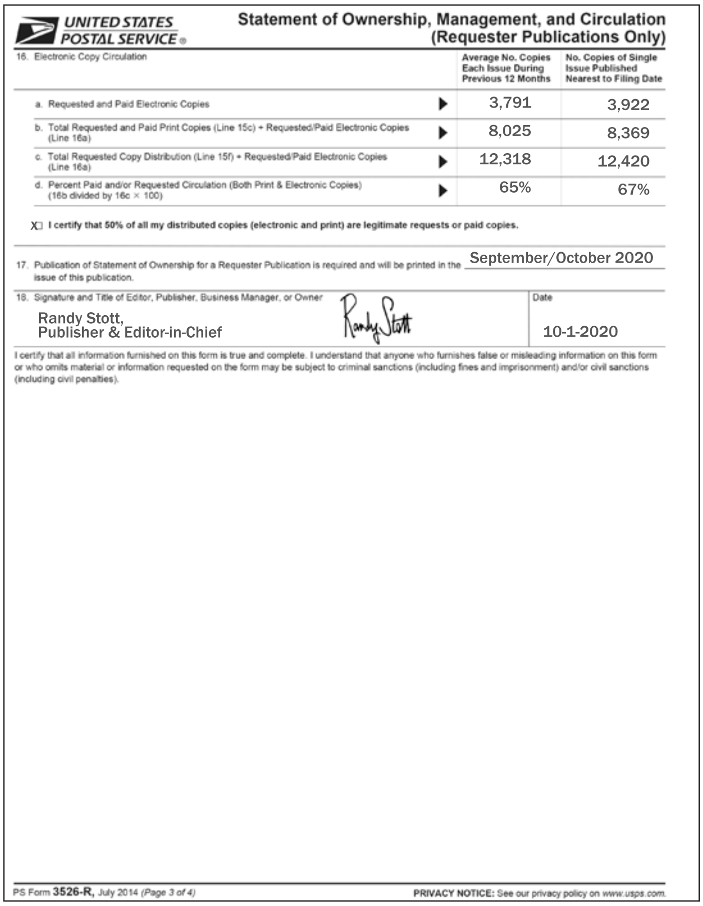

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That Industry 4.0 Digital Factory Thing — Part 2

Joe Arvin

Welcome to the second installment of an article which is presented as a guide for navigating the topic of the Industry 4.0 Digital Factory. In the first part, featured in the August 2020 Issue of *Gear Technology Magazine*, I presented a fictional account of a Zoom meeting between me, a gear company president named Phil (a fictional character) and Chuck Gates - a very real person and one of our AGS consultants who is very knowledgeable on Industry 4.0.

In the first installment, Phil learned not to panic, for there is a methodical approach for evaluating what should or shouldn't be done. He also learned about the multi-phased approach for assessing and implementing the Industry 4.0 Digital Factory, and the specifics involved in Phase 1.

In this episode, we continue the conversation as Chuck outlines the steps involved in Phase 2.

Chuck: So Phil, it's good to talk to you again. Why don't you tell Joe and me about how you did with the READINESS Phase — or Phase 1?

Phil: It's good to talk with both of you again as well. Here's what we've done so far with Phase 1.

In Step One, we looked into the advanced manufacturing and smart factory enabling technologies that are available to us. Specifically, we focused on the technologies that can be used to make us more effective and efficient. Among these were the Industrial Internet of Things, artificial intelligence, augmented reality, robotics, additive manufacturing, and so on.

In Step Two, we detailed the current state of our business — which helped us to understand our demand forecast, our performance, and our process flow.

In Step Three, we determined where we wanted to be in the future — or in other words, we defined the future state of our business.

And finally, in Step Four, we compared our current state and our future state, and we looked carefully for any gaps between the two. Let me tell you, I think we've learned quite a bit about ourselves by going through this process.

Chuck: Great Phil. Now let's talk about the next phase in this process which is Phase 2 or the ACTION plan.

The goal of the action plan is fairly straightforward. By looking at the current state of your business, and then comparing that to the future state of your business, this comparison will allow you to identify gaps. Then you develop the action plan in order to use the advanced technologies you identified which can be used to fill these gaps as a way to achieve the future state of your business.

Phil: That makes perfect sense. I think we have identified a future state that has many gaps and opportunities for improvement over our current state.

Chuck: That's really important to know. You see, companies that do not develop a more competitive approach, and remain reactive to the changes in customer demand, they will likely have problems down the road.

Phil: Yes, that makes sense. But here's a problem that I'm having. As our Digital Factory Committee learned about these technologies, they're all fired up about us adding them to our operation. In fact, they're already talking about a CAPEX investment in the area of $4.5M to address our gaps. We just don't have that kind of money right now.

Chuck: Instead of trying to address all of the enabling technologies that you've identified at once, pick two to get started with. In other words, use the *Pick-Two* approach. Here is how to do that.

First, determine the two areas of performance that you feel are your top priorities. In other words, which two business results that are being achieved with your current processes require immediate improvement? Some potential examples could be quality, revenue, cost reduction, delivery time, throughput or cycle time, profit, customer service, demand forecast accuracy, asset utilization, return on sales, and growth.

This focus can lead to a more disciplined approach to creating the plan of action. Trying to integrate all of the enabling technologies at once can be extremely daunting and lead to being overwhelmed.

This is really important. The keys to success in manufacturing are to invest wisely and do so in a timely manner. The Industry 4.0 Digital Factory is not about chasing a fad; it is about making advanced technology business decisions that have an attractive return on investment and a short payback period. Do not implement a digital transformation out of fear. Only innovate with a solid business purpose in mind. Change for the sake of change can be dangerous if there is no proof of the need.

Joe: If I can interject at this point, perhaps I can offer some ideas from my perspective as a gear guy. It's always important to identify your gaps, but it's also important to realize that not all of these gaps will require an Industry 4.0 technology to be resolved.

Phil: That's interesting Joe. I think I see where you're going.

Joe: I've always said that the key to success is investing in the latest technology. But that often comes with a high price. While it's critical to invest as much as you can, there are many things that you can do to improve the gaps without the large price tag.

Let me ask you a question Phil. When you identified the gaps, what was one of your most significant problems?

Phil: That would have to be scrap and rework.

Joe: What did your quality assurance people attribute those problems to?

Phil: The main problem was rooted in our machines that continually require maintenance to maintain tight tolerances. There was also a lot of operator error.

Joe: That's a good example of what I'm talking about. To address the machine problem, you could consider retrofitting or rebuilding those machines. Or you might want to look for used machine tools that will increase your capabilities. There is a lot of late model equipment on the market in as-new condition, often available from companies that have gone out of business. This equipment can be purchased at a dramatic cost savings when compared to new. As for operator error, you might consider a focused training solution to address this problem.

Phil: We also have the need to increase our productivity.

Joe: In this case, you might want to really consider the purchase of a machine that is capable of performing multiple operations in one chucking.

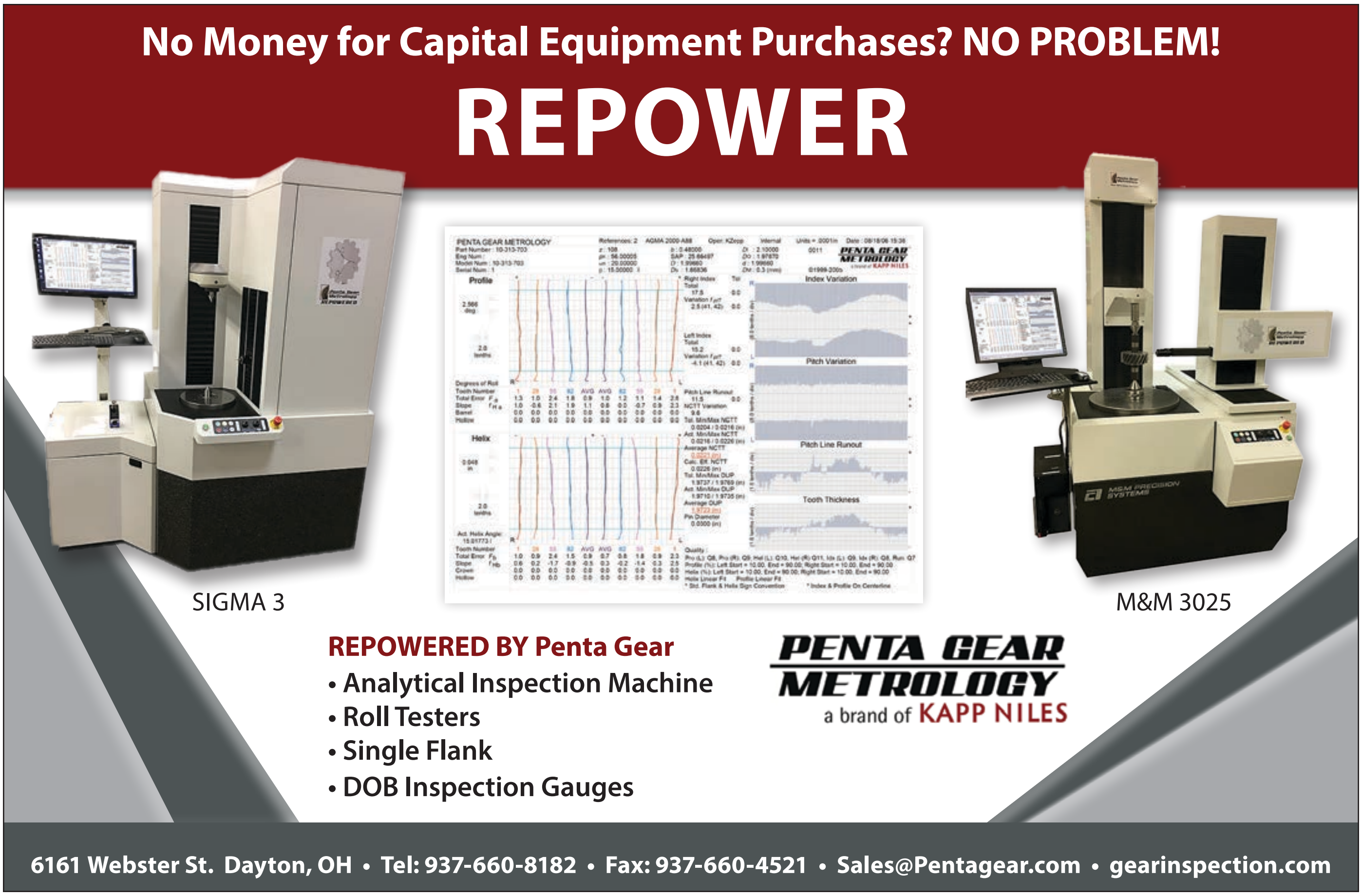
On the other hand, robotic load and unload with integral inspection and auto adjustment to the machine if necessary would be a great way to integrate Industry 4.0 technology to improve your productivity. Furthermore, 3D printing for prototyping and wear resistant tooling is now a viable option and something to seriously consider.

These are just a few ideas, but my point is that it's best to analyze these other approaches before rushing into an expensive digital factory technology solution.

Chuck: Joe's exactly right. When you identify the gaps, you need the most cost-effective business case to make this decision on integrating the enabling technologies. Many deficiencies can be fixed without digital factory components. You need to be sure that the investment in Industry 4.0 technology is really your best solution.

Phil: Yes, that's a good perspective. So as we plan to implement improvements that really do need Industry 4.0 specific components, what do we do next?

Chuck: First, I would recommend you take a look at this book. The title is *4.0: The Use of Emergent Technologies in Manufacturing*, which is published by Palibrio Publishing. This book lists some key enabling technologies which include the Industrial Internet of Things (IIoT), Manufacturing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Execution Systems (MES), Artificial Intelligence (AI), Digital Twins, Augmented Reality (AR), Robotics, and Additive Manufacturing. Some additional key enabling technologies that can be used to move to a smart, digital factory are Big Data/Advanced Analytics, Automatic Guided Vehicles (AGV), Virtual Reality (VR), Machine Learning, Cloud Computing, Radio Frequency Identification (RFID), and Real Time Location Systems (RTLS). This resource can help you to focus on what technologies are best suited to address your gaps that can't be resolved adequately with other methods.

With this information, you're ready to proceed with Phase 2. Here are some helpful tips on how to get started in developing the plan of action.

Phil: I have a question. What if we are unable to select the two enabling technologies to include in the plan of action?

Chuck: If that is the case, I would recommend looking at Big Data/Advanced Analytics and Manufacturing Execution Systems (MES).

As you develop the action plan, you'll need to determine the specific projects to implement along with the details including timelines and investment dollars needed. These projects also need to include an estimated cost-versus-benefit analysis.

Think of it this way. The plan of action will serve as the roadmap to identify the specific, new projects that will be deployed to implement the next following phase which is Phase 3.

Here is another important point. Your plan of action will need to include two major measurable objectives, and two outlined projects for each of the two major measurable objectives. It should have two investment amounts for each project — a minimum and a maximum. There needs to be two deadlines for each step in each project — a rapid finish and a slow finish. And finally, your plan will need two contingencies for each project, and two calculated return-on-investments for each project.

Also keep in mind that effective and efficient projects will include a project charter, project scope, activities, estimated activity durations or timelines, estimated activity costs, quality metrics, and resources needed. It is essential that a project manager is assigned. To manage the projects and control the execution of the activities, there should be a monthly status review of the value proposition. The value proposition chart shows the estimated activity completion versus the actual activity completion, the estimated costs-to-date versus the actual costs expended, and the estimated benefits to date versus the actual benefits received. During these monthly project reviews, upper management needs to provide leadership in order to control the projects and to stay within the budget and deadlines.

Phil: What kind of timeline should we be looking at for completing Phase 2?

Chuck: Well, that depends on how aggressive the company is in pursuing excellence in the future. The willingness to develop a plan of action with detailed projects identified is actually a critical motivating factor. On the other hand, if company leaders are not sure about activating this systematic approach, and prefer to remain on their current course without pursuing a future of the digital transformation to a smart factory, it's tough to move forward. This may be referred to as relying on luck, fate, chance,

Previous Arvin's Angle Articles

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January/February 2018 — Page 48

Training — A Top Priority for Investment
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That Industry 4.0 Digital Factory Thing - Part I
August 2020 - Page 24

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and magic to compete in the future. It is difficult to pursue excellence on a part-time, half-hearted basis.

So, good luck with Phase 2 as you develop your action plan. Certainly let us know if you have any questions. And when this phase is completed, we can talk about putting that plan into place with Phase 3.

Phil: Thanks. This has really been helpful. I'll let you guys know how we're doing.

Final Words

In the next part of this article series, we will hear what Phil uncovers in Phase 2 and then we will hear from Chuck Gates again on the steps involved in Phase 3. I hope this story provides some valuable insights into your evaluation of the digital transformation of your operation.

Please look for the continuation of the story in the next installment of *Arvin's Angle* in *Gear Technology Magazine*.

Finally, I would like to thank Chuck Gates for his valuable assistance in the development of this article. Of course, if you have any questions or comments, please contact me at ArvinGlobal@Gmail.com.

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

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History of the AGMA JFACTOR

Robert Errichello, P.E.

I felt a tap on my shoulder. Turning, I saw the chief draftsman who said, "You're in charge of gears."

And he walked away.

Dumbfounded, I stared at the back of his head, and sat down at my drafting board. It was November, 1963, shortly after JFK was assassinated, and after I was discharged from the U.S. Army in October, 1963. I rejoined the company that I had worked for before being drafted into the Army (in those days companies were required to rehire veterans). Safety Electrical, in Hamden, Connecticut, was so named because their main product was electrical generators for passenger trains that were first introduced to replace the gas lamps that illuminated passenger cars. The generator was installed below the passenger car and was driven by an axle-mounted gearbox that was manufactured by a division of the Dana Corporation. In the 1960s, JFK inspired resurgence in mass transportation by funding expansion of the nation's subway systems. Dana upgraded the gearbox used for driving the Safety Electrical generator and used it to drive subway rail cars. It became a booming business because many of the major subway systems in the U.S. purchased Dana gearboxes. Subsequently, however, Dana decided to offer Safety Electrical a gearbox business at a price that couldn't be refused. And so, overnight, I was indeed "in charge of gears."

Fortunately for me, the deal included an agreement whereby Safety Electrical acquired Dana's chief engineer of the rail car gearbox division who was assigned to train me in all aspects of the design and manufacture of rail car gearboxes. This was a very exciting time for me — and a wonderful learning experience. However, the Dana engineer was on loan for only three years, and he returned to Dana in 1966.

By then, I had determined to specialize in gear design, but there were only electrical engineers at Safety Electrical, and no mechanical engineers to continue my training. Therefore, I joined Sier Bath Gear Company in North Bergen, New Jersey in 1966. Sier Bath was a very innovative company who specialized in high-accuracy carburized and ground gears. But most importantly for me, they regularly used consultants for gear design, including then-industry living legends Eliot Buckingham, Darle Dudley, and Ernest Wildhaber. What's more, my gear knowledge grew exponentially with mentoring from Sier Bath's chief engineer Jack Pearson and working with Sier Bath's consultants. Dudley showed me how to determine the AGMA gear strength geometry factor (JFACTOR) by graphical layout, which is a very tedious and not especially accurate method to determine JFACTORs. However, the JFACTOR determines a gear's bending strength, and an accurate value is needed for the design of every gear. Realizing that the graphical layout was not practical, I searched the literature for a computer method. I discovered Wadhwa's (Ref. 1) analytic method and programmed it in the *BASIC* ("*Beginner's All-purpose Symbolic Instruction Code*") language using Sier Bath's connection to a GE time share computer. I found that Wadhwa's method worked well for most gears, but failed to solve certain gear geometries. Wadhwa was an IBM programmer who chose a difficult gear geometry issue to demonstrate the capability of computers; however, he was not a gear engineer. Therefore, he turned to Buckingham's classic book (Ref. 2) for the equations of a trochoid. Unfortunately, as will be shown, Wadhwa's choice of using trochoid equations and selecting θ_T as the independent parameter introduced a numerical problem. (Note: There is no closed-form solution for the JFACTOR, and numerical iteration is required to find the critical stress point defined by the Lewis parabola (Ref. 3). Wadhwa's analytic method fails to converge to the correct solution for certain gear geometry.)

Figure 1 shows Wadhwa's analytic method where the independent parameter is a coordinate of the primary trochoid, θ_T. After solving for point S on the primary trochoid, point F on the secondary trochoid (on the root fillet) is found by matching the slope, ψ_T at point S.

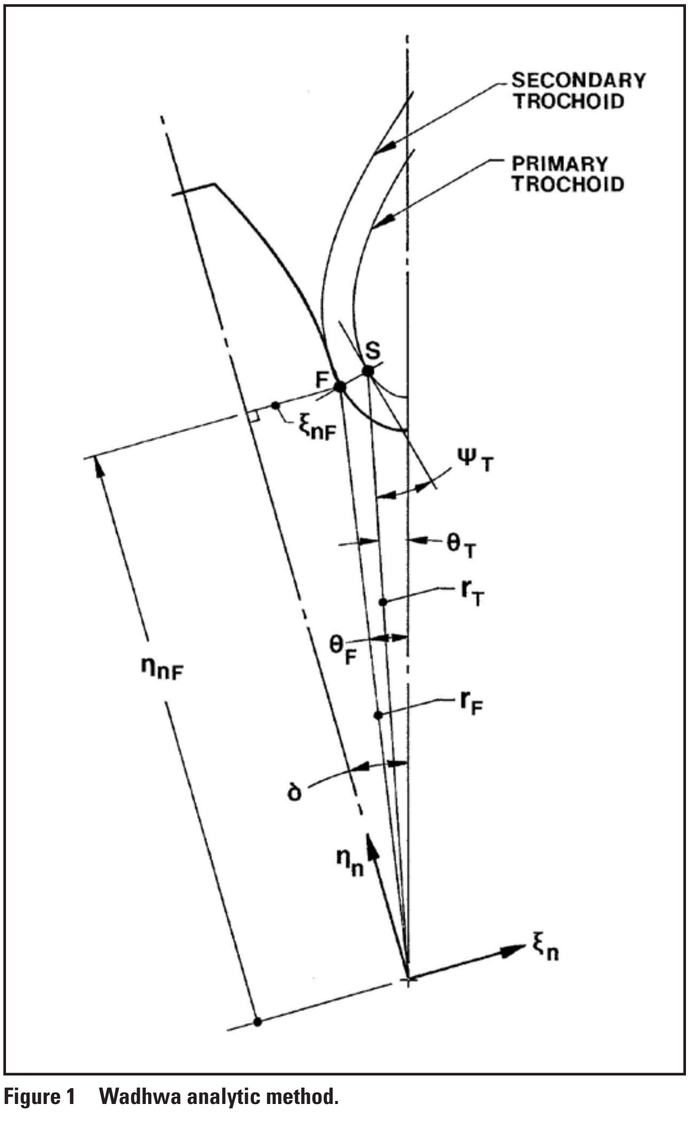

Figure 1 **Wadhwa analytic method.**

Knowing that Wadhwa's analytic method had a numerical problem, I asked Wildhaber (A brilliant geometrician with over 300 patents (Ref. 4) with applications to gears. In 1924, he began a long career at the Gleason Works where he invented the hypoid gear and the Revacycle cutting process that enabled a high production rate for straight bevel gears. As a consultant to Sier Bath, he helped to develop the Vari-Crown gear coupling.) to derive the equations for the AGMA JFACTOR (I didn't mention Wadhwa's paper because I wanted an independent analysis). Wildhaber developed what he termed a "kinematic" method, whereby the gear tooth geometry was generated from the kinematics of the generating tool and gear workpiece.

Fundamentally, it was based on the law of gearing, which requires that the normal to the cutting point, F, must pass through the pitch point, P, at all points of contact. Figure 2 shows Wildhaber's kinematic method where the independent parameter is the inclination of the contact normal, α_n.

Figures 3 and 4 show graphical depictions of the numerical problem that plagued Wadhwa's analytic method. Figure 3 shows there are two roots to the solution of the equations for the JFACTOR. The correct root is marked "ROOT" (Fig. 3). The incorrect root lies closer to $\alpha_n = 0$ (Fig. 3).

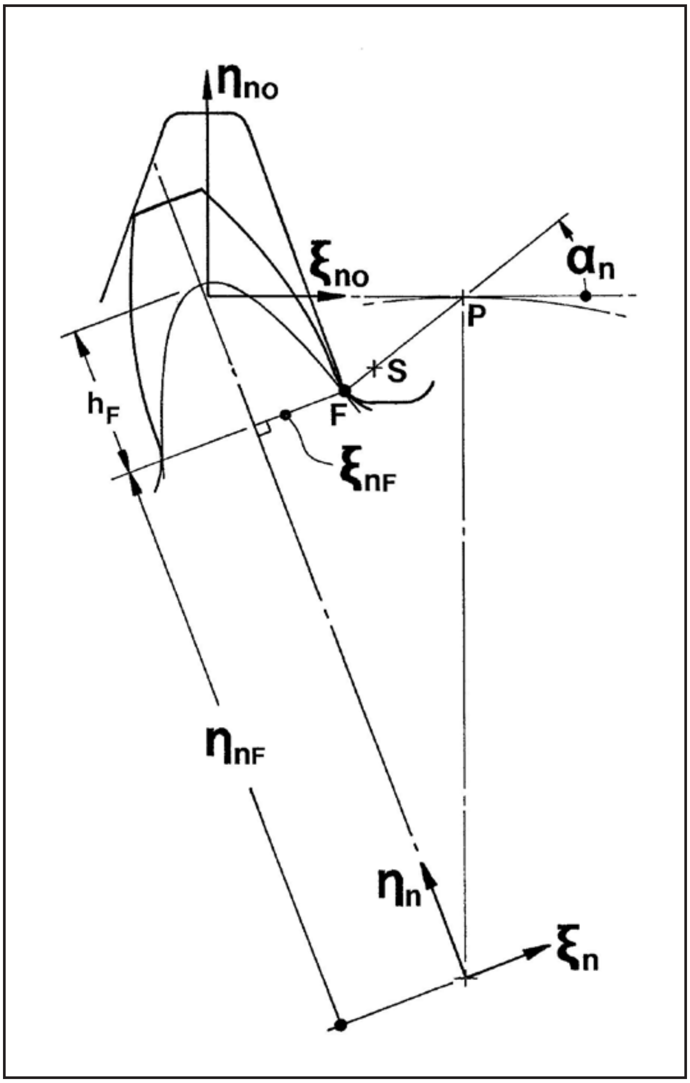

Figure 2 Wildhaber kinematic method.

Figure 4 shows the incorrect root corresponds to an inverted Lewis parabola that opens upward, rather than downward.

Hence, Wadhwa's "analytic" method converged to the incorrect root for gears with a large number of teeth or large profile shift. Happily, I found that Wildhaber's "kinematic" method resolved the numerical problem that exists in Wadhwa's "analytic" method.

In 1971, I finally tired of going to night school (I had spent nine years trying to complete a BSME degree) and decided to leave Sier Bath to attend full-time at the University of California in Berkeley (UCB). I graduated from Berkeley's Mechanical Engineering Department with BSME and MSME degrees in

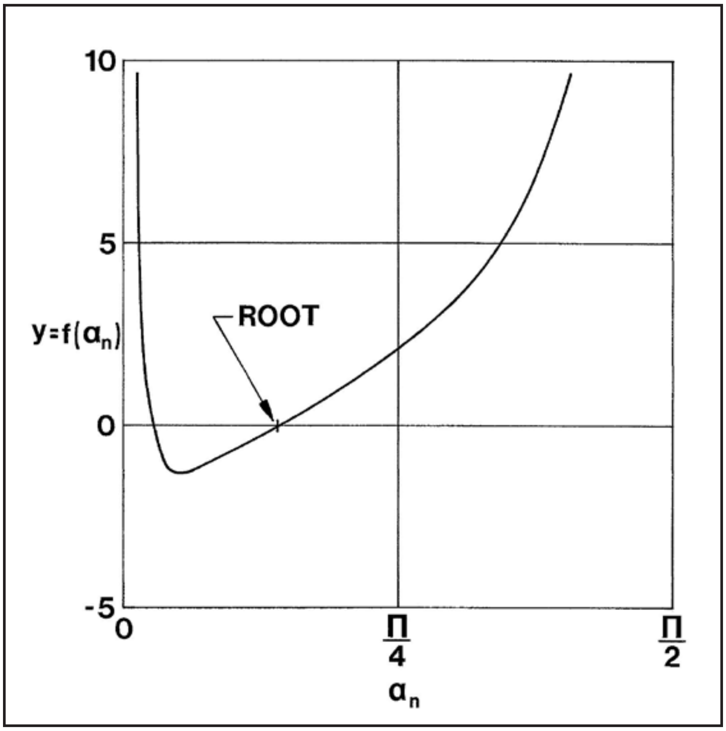

Figure 3 Graph showing there are two roots to the solution.

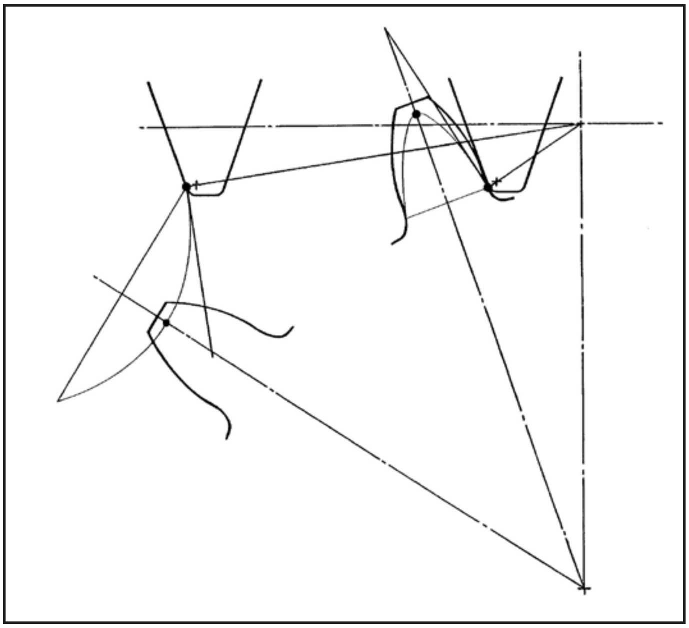

Figure 4 Showing correct and incorrect roots.

1975. I then — from 1975 through 1976 — worked briefly for Western Gear in Lynwood, California. During my brief time there I became interested in structural dynamics. I therefore decided to go back to UCB to study earthquake engineering and graduated with a Master of Engineering in Structural Dynamics in 1978.

In 1978, I founded GEARTECH as a consulting firm. At the time, the only personal computer available was the Apple computer — but it was expensive and required some assembly. So I decided to program the JFACTOR algorithm on a Texas Instruments TI-59 calculator. I soon found that the iteration method I had used for Wildhaber's kinematic method was much too slow using the TI-59 calculator; it took hours to solve for one JFACTOR. I knew Newton's method of iteration was very efficient, but it required calculation of the derivative from a complex set of equations. Nevertheless, I bit the bullet, and slogged through the algebra. Months and reams of paper later, I arrived at a simple equation for the derivative — so simple, in fact, I didn't believe that it could be correct. However, when I programmed the algorithm on the TI-59 calculator, it converged in three iterations within seconds (Fig. 5). Figure 5 shows that Newton's method of iteration converges to the correct root in as little as three iterations.

I published the algorithm in a paper (Ref. 5) that was presented at the 1981 International Symposium on Gearing & Power Transmissions in Tokyo.

Sier Bath encouraged me to participate in AGMA technical meetings, and in 1967 I started to attend the face-to-face meetings of the Helical Gear Rating Committee (HGRC). The HGRC began the first draft of AGMA 218.01 (Ref. 6) in 1973. During one of the meetings, committee members expressed the desire that the current graphical layout method for determining the JFACTOR be replaced with a computer algorithm, and asked for a volunteer to contribute their computer program. Although there were representatives from all the major gear manufacturers on the HGRC, no member was willing to contribute their program because they knew their programs were not reliable.

I then immediately recognized their problem, i.e. — they had programmed Wadhwa's method (Ref. 1). I explained this to the committee and offered my algorithm — and stated that I had resolved the numerical problem. However, the committee decided that the best approach would be to publish my algorithm as an AGMA technical paper, and then ask the AGMA membership to test the algorithm to validate that it was reliable. I agreed with the committee's recommendation, and I published the algorithm in a paper (Ref. 7) and presented it at the 1981 AGMA Fall Technical Meeting in Toronto. During the Q&A following my presentation, several attendees expressed an interest in extending the algorithm to encompass external spur and helical gears generated by pinion-type shaper cutters. Thus I began applying Wildhaber's kinematic method to shaper-cut gears. I recalled that Wells Coleman — a member of the `HGRC — mentioned that it was possible to write an algorithm for pinion-type shaper cutters that would also be applicable to rack-type cutters (hobs, rack cutters, and generating grinding wheels) by simply inputting a pinion-type cutter with a large number of teeth. When Wells first mentioned this notion, I didn't follow his recommendation because I was overwhelmed with the task of developing an algorithm for rack-type cutters. However, after I developed the algorithm for pinion-type cutters, I tried inputting a pinion-type cutter with 10,000 teeth in the algorithm and it successfully duplicated the JFACTOR that was generated by a rack-type cutter. No question — Coleman's recommendation was correct. I published the algorithm for pinion-type cutters in a paper (Ref. 8) that was presented at the 1983 AGMA Fall Technical Meeting in Montreal.

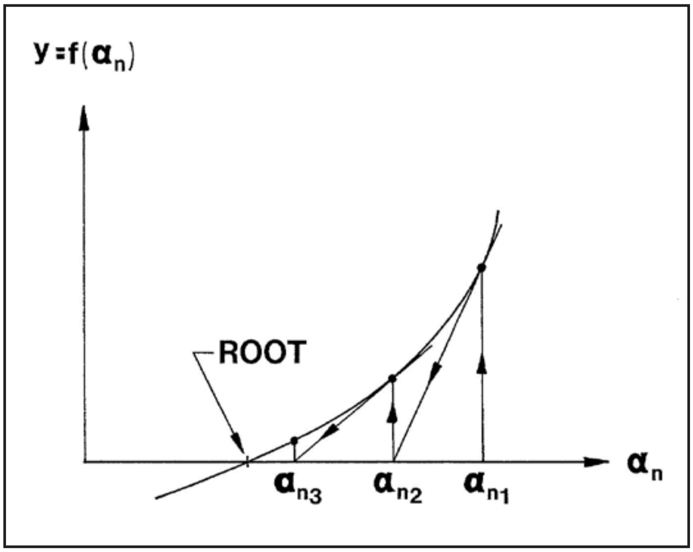

Figure 5 **Iteration by Newton's method.**

The algorithm (Ref. 8) was tested for several years by the AGMA membership and was finally accepted as the official AGMA method for calculating JFACTORS and was published in AGMA 908-B89 (Ref. 9).

Next, the HGRC developed the information sheet — AGMA 918-A93 (Ref. 10) — to assist designers in the proper use and interpretation of AGMA 908-B89 and to assist in the development of computer programs for calculating geometry factors for pitting resistance I and bending strength J. AGMA 918-A93 includes a flow chart and several numerical examples.

For more information.

Questions or comments regarding this paper? Contact Robert Errichello at *rlegears@mt.net.*

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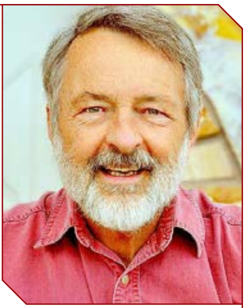

Robert Errichello is founder of GEARTECH. He has over 50 years of industrial experience and has taught courses in material science, fracture mechanics, vibration, and machine design at San Francisco State University and the University of California at Berkeley. He presented seminars on design, analysis, lubrication, and failure analysis of gears and bearings to professional societies, technical schools, and gear, bearing, and lubrication industries. He is a graduate of the University of California at Berkeley and holds BS and MS degrees in Mechanical Engineering and a Master of Engineering degree in structural dynamics. Bob is a member of the AGMA Gear Rating Committee, AGMA/AWEA Wind Turbine Committee, ASM International, ASME Power Transmission and Gearing Committee, STLE, NREL and GRC. Bob has published over 100 articles on design, analysis, and application of gears. He is technical editor for Gear Technology magazine and STLE Tribology Transactions. He is recipient of the AGMA TDEC Award, AGMA E.P. Connell Award, AGMA Lifetime Achievement Award, STLE Wilbur Deutch Memorial Award, STLE Edmond E. Bisson Award, and AWEA Technical Achievement Award.

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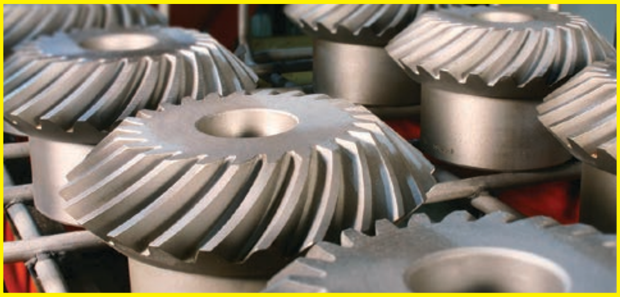

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Introduction to Electric Vehicle Transmissions

Dr. Hermann J. Stadtfeld

Transmissions in Automobiles with Internal Combustion Engines

Traditional automotive transmissions are designed to adjust the engine speed to the speed of the driving wheels, required in order to achieve the desired driving speed. The engine speed of a modern internal combustion engine has a range for optimal efficiency between 1,000 and 2,500 rpm.

A midsize sedan with an outer tire diameter of 600mm has to rotate with a speed of 778 rpm in order to achieve a vehicle speed of 88km/h or 55mph,

$$n = 10^6 \cdot v / (D \cdot \pi \cdot 60)$$

whereas:

n rotational wheel speed [rpm]
v vehicle speed [km/h]
D outer tire roll diameter [mm]

If the engine idle-speed is 600 rpm, and if the engine crank shaft output was directly connected to the wheels, then the vehicle speed would be:

$$v = n \cdot D \cdot \pi \cdot 60/106 = 600 \cdot 600 \cdot \pi \cdot 60/10^6 = 67.9\text{km/h (42.44 mph)}$$

One problem is that the engine torque in idle would not be sufficient to keep a vehicle moving at 67.9 km/h (42.44 mph) on a level pavement. A second problem is encountered when the engine is instantly connected with the wheels at idle speed. The vehicle would first jerk and then the engine would die. The torque characteristics of a combustion engine and an electric motor (Fig. 1) show the low-torque availability of a combustion engine at idle speed.

Even if a compliant element like a torque converter between engine and wheels is used, it would not be possible to control acceleration, speed and deceleration the way it is expected for safe driving. Besides all of these obstacles, the fuel consumption of a vehicle without a transmission would be several times that of a vehicle today that is equipped with a multi-speed transmission.

The study of a simple driving sequence can already reveal all basic requirements for an adaptive transmission element between engine and wheels. When the vehicle starts from a full stop, the engine has to increase its speed from 600 rpm idle to 1,500 rpm in order to develop enough torque for the acceleration of the standing vehicle. At the beginning, a hydraulic torque converter or a slip clutch will connect the rotating crankshaft of the engine with the not-yet-rotating gears in the transmission that are connected to the wheels — which also do not yet rotate. At this instance, the transmission has to provide a sufficient reduction, such that the torque converter output torque is amplified enough to accelerate the vehicle from zero speed to a moving condition. Shortly after that, when the vehicle is driving between 10 and 20 km/h (6.25 and 12.5 mph), the transmission shifts into a higher gear because the engine rpm would have to double when the vehicle speed is 30 km/h (18.75 mph) and be about 6 times higher (=9,000 rpm) when the vehicle reaches the desired 88 km/h (55 mph). Such a high engine speed would be undesirable in many ways. The fuel consumption of the vehicle would become extremely high and the exhaust and noise emission would also reach unacceptable levels. A high-revving engine would also be subject to high wear and to many possible mechanical failures.

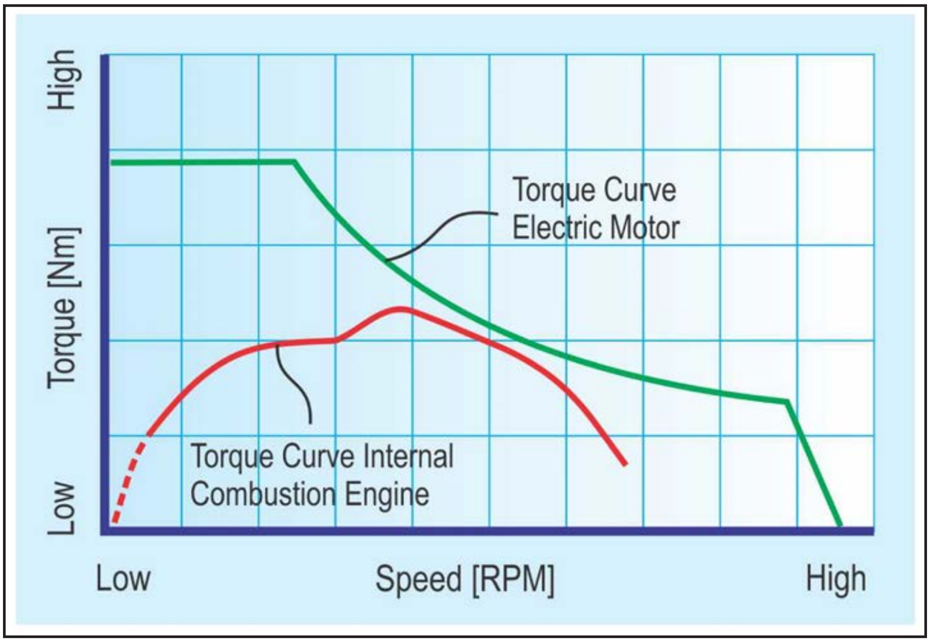

Figure 1 Torque versus speed, combustion engine and electric motor.

In order to keep the engine running in a desirable range between 1,000 and 2,500 rpm, the transmission will shift up about 7 times until the vehicle reaches 88 km/h (55 mph). After the transmission shifts into a higher gear, the engine rpm drops, for example, down to 1,000 rpm, while the gas pedal is kept at a steady position. The higher gear (lower ratio) requires more load from the engine that initiates the rpm drop. As the vehicle continues to accelerate, the engine rpm increases proportionally with the vehicle speed, and loses torque until the next shift occurs at, for example, 2,500 rpm. Now the engine speed drops to 1,000 rpm and the acceleration torque increases again. The load hysteresis is the highest at the low engine rpm and the lowest at the high rpm. The gas pedal position creates this hysteresis while the driver signals to the engine that either a faster or a slower speed is desired.

A cross-sectional cut through a modern, electronically controlled eight-speed automatic transmission is shown (Fig. 2). The input from the engine and the torque converter is on the right side of the transmission. The input shaft passes through three planetary stages that have two multiple-disk clutches on the right side and two multiple-disk clutches to their left that actuate the eight transmission ratios for forward driving. At the left side of the transmission is one additional disk clutch that actuates the planetary stage to its left for reverse driving. The output shaft is exposed on the left side of the transmission.

Conventional automotive drive trains. A view of all transmission components in an all-wheel drive passenger car with a longitudinally oriented combustion engine is shown (Fig. 3). A transmission, similar to the one shown (Fig. 2, center-left), is used to adapt the engine speed to the wheels. One long propeller shaft connects the transmission output with the rear axle unit (right side). The rear axle reduces the transmission output speed by a constant factor (usually around three) and additionally re-directs torque and rotation from the input direction by 90° — which matches the wheel rotation direction. The rear axle unit output flanges are connected to the two rear wheels with drive shafts. Each drive shaft uses two constant-velocity joints in order to disconnect the mass inertia of all drive components from the wheels. The wheels are connected to space control arms that ensure a minimum of un-sprung weight on each wheel; low un-sprung weight enhances vehicle stability and driving comfort.

In order to also propel the front wheels, a transfer case is added to the output of the transmission. A second, shorter propeller shaft connects the front axle with the transfer case. The front axle and wheel suspension also follow the principle of minimizing the un-sprung weight of the individual wheel.

The concept in Figure 3 clearly demonstrates that typically, only one engine is used as a prime mover and only one transmission adapts the engine speed to the desired speed of the wheels. This central speed and power are then transferred to the driving wheels via propeller shafts and drive shafts. Equipping a vehicle with two combustion engines appears impractical. Internal combustion engines are rather large and require an infrastructure of connections for fresh air intake, gasoline lines, electrical, electronical and mechanical control, and actuation signals — as well as a complex exhaust system. Experiments in the past also showed that synchronizing two combustion engines is nearly impossible and poses many safety concerns.

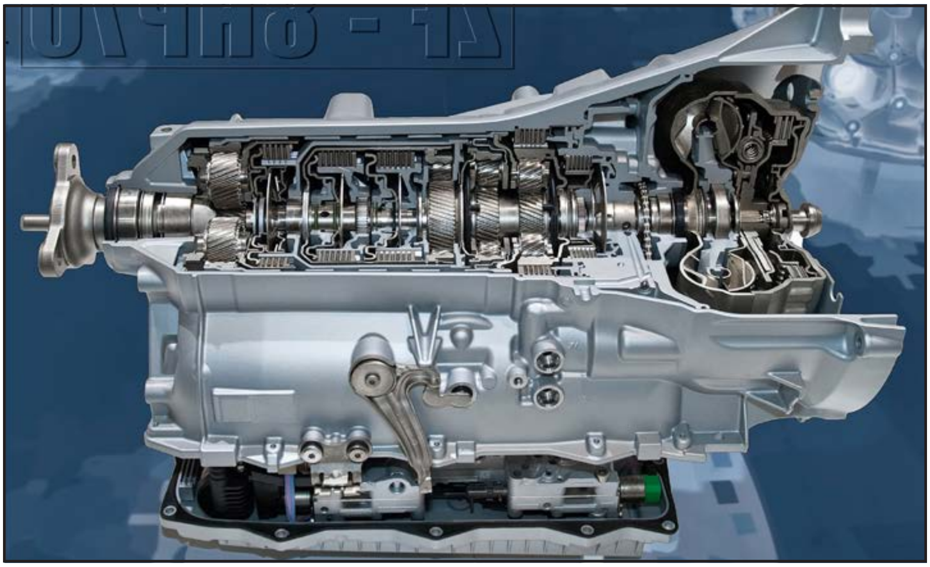

Figure 2 **Eight-speed automatic transmission (Refs. 1–2).**

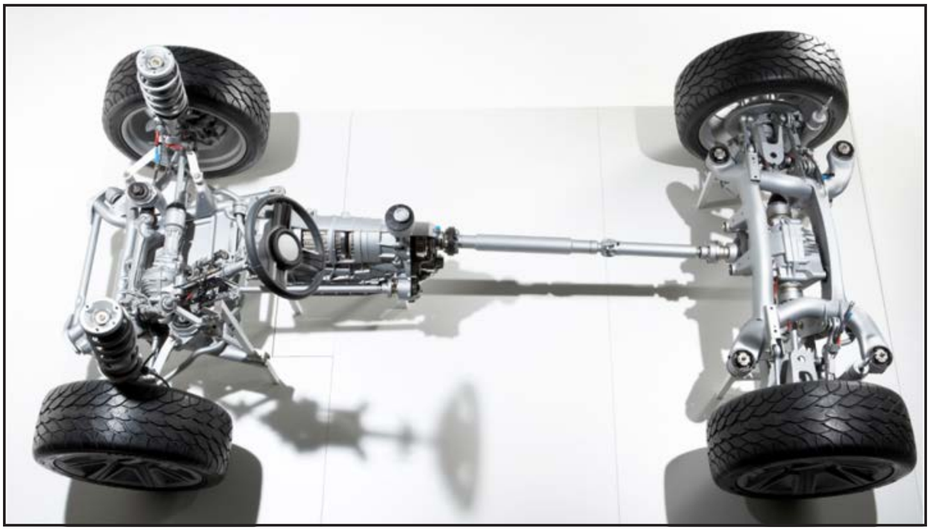

Figure 3 **Powertrain in an all-wheel drive sedan (courtesy ZF Friedrichshafen AG).**

The strength of electric motors is their small size and their nearly non-existing infrastructure. The following sections will discuss these aspects and new possibilities presented by e-Drives.

Transmissions in electrical vehicles. Electric motors have a number of advantages versus internal combustion engines. The size of the latest high-performance motors that use rare earth magnets with many poles is very small compared to their HP or kW rating. Their peak torque is higher than that of combustion engines. Electric motors start with zero rpm and can develop high torques at low speeds. However, their speed for optimal performance regarding available energy and consumption of electricity is rather high. At a cruising speed of 88 km/h (55 mph), today's electric vehicles operate, for example, at motor speeds of 10,000 rpm. The rotational wheel speed, at 88 km/h (55 mph), was given above with 778 rpm, which results in a 12.85 ratio between electric motor and wheels; the ratio at the same speed for a car with a combustion engine is 1.93 (engine speed equal to 1,500 rpm). This comparison shows that electric vehicles require more than six times the transmission ratio of a conventional car in order to deliver good performance and high efficiency.

If electric motors are built even smaller than today, this would reduce the cost for rare earth magnets and make the motors lighter and easier to integrate between the wheels of a vehicle. Electric vehicle manufacturers have already announced that electric vehicle motor development will increase rpms to 20,000 within the coming four years, and further increase to 30,000 rpm before the year 2030. These high-speed motors require new bearing solutions and their copper windings have to be tighter and need to be wound with the highest accuracy in order to reduce vibration from unbalance and prevent the coils to take a "set" due to the high centrifugal forces. For the transmission solutions, this means higher ratios; the above mentioned ratio of 12.85 will have to increase up to 38.55 — and ever higher.

Electric vehicle transmission design and manufacturing requirements. Ratios are not the only different requirement between conventional and electric vehicle transmissions; the requirement portfolio also covers of course the criteria "power density," "noise" and "efficiency." Electric motors can deliver short bursts of peak torques which are several times as high as the nominal power rating. This provides the electric vehicle a sporty touch and makes it attractive to certain groups of consumers. The transmissions have to be able to handle these high peak torques during the vehicle's entire lifecycle. Although, from a practical point of view, the efficiency should have the highest priority right after the strength of the gears, in reality the noise emission has been found to be of much higher priority for customers. Due to the high rpms of motor and gears, some vehicle owners notice strange high-pitch humming sounds they never experienced in a vehicle before. Some vehicle owners just complain that it is uncomfortable, while others claim that it puts a permanent ringing in their ears, which does not go away after they leave their electric cars.

This means, for electric vehicle transmissions, that advanced manufacturing and gear mating technologies have to be applied. Gears have to be ground or hard skived and honed. Combinations of a honed and a ground gear, or a ground gear with a hard skived gear have proven to deliver the lowest noise emission and are also less likely to emit high pitch frequencies. Electric vehicle cylindrical gears will also require sophisticated topological flank surface optimizations that provide conjugate flank centers for optimal transmission characteristics, as well as high load carrying capabilities. Only the tooth boundaries in path-of-contact direction are relieved to prevent load concentration peaks under highest loads. Although hard skiving is not a common hard finishing process for cylindrical gears, it is about to have a breakthrough for internal transmission rings. These rings are not hard finished at present because grinding would require a miniature-sized grinding wheel. Today the internal teeth are finish-shaped or broached, and then either heat treated — with the goal of low distortions — or ion-nitrited. The nitrite only creates a 0.01 mm hard skin on the surface, but it guarantees very low distortions. It is also possible today, with the power skiving process, to perform a hard finishing operation after heat treatment by applying carbide hard skiving cutters.

Noise emission and high loads also put difficult requirements on the bearings and on the transmission housing design. Even the smallest vibrations can become noise problems when the vibration finds a resonance in the surrounding vehicle components.

Practically realized electric transmission examples. All transmissions shown and discussed in this section are placed between the wheels of an axle — front and/or rear. Their output flanges are connected to the wheels with drive shafts that use constant velocity joints on both ends. In comparison to in-wheel motors, the un-sprung weight of the wheels and wheel suspension units is kept as low as in a modern, conventional car. High un-sprung weight will reduce the traction contact between tire and road and will also contribute the wheel to trample while driving on uneven or bumpy surfaces. The trample reduces driving comfort and the vehicle handling properties and therefore presents a safety risk.

A two-stage and single-speed electric vehicle transmission is shown without the electric motor (Fig. 4). The transmission ratio is 12.5 and cannot be changed in order to adjust to the driving speed or to traffic conditions. Single-speed transmissions are very well-suited for small-sized electric vehicles due to their small size and low weight, as well as the possibility to manufacture them cost effectively.

The transmission in Figure 4 requires only three shafts and six bearings. Due to the helix angle of all applied cylindrical gears, the bearings are either tapered roller bearings or angular ball bearings that are axially shimmed in order to achieve a light pre-load. This transmission is suited for driving one single axle of a two-wheel or both axles of an all-wheel drive vehicle.

The transmission (Fig. 5) presents a very interesting three-stage design that can accommodate a maximal ratio of 18. This transmission has a second, smaller-size motor that realizes, in connection with the planetary stage, a variation of the output speed of one wheel versus the other. This functionality not only replaces the conventional differential; it is also utilized to realize a high-efficiency torque-vectoring function.

The arrangement of the two motors facing each other, and the low width of the central transmission, accommodate a small distance between the output shafts

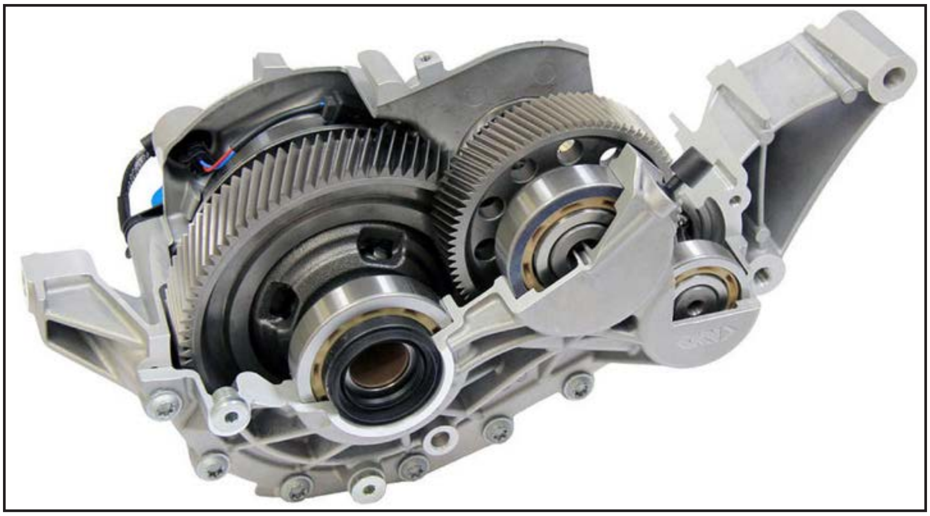

Figure 4 Two-stage and single-speed electric vehicle transmission — ratio 12.5 (Ref. 3).

allowing for long drive shafts, which is a desirable condition.

The transmission (Fig. 6) is two-stage, two-speed — with a maximal ratio of 16. This transmission is very compact and requires very little extra space next to the electric motor; the differential with its four straight bevel gears is integrated in the final drive gear.

One of the differential outputs is visible at the right side of Figure 6. Because of the concentric orientation of the electric motor relative to the final drive gear of the transmission and the differential, the problem of transmitting the rotation from the second differential outputs to the left side of the motor is solved by using a hollow motor shaft where an extension shaft of the left differential output is placed. This puts the left output flange at the backside of the electric motor. The distance between the output flanges is larger compared to the transmissions shown (Figs. 4 & 5), but still allows for a reasonable length of the drive shafts.

A rather high reduction transmission, with the motor integrated within the same housing, is shown (Fig. 7); the maximal ratio of the transmission in Figure 7 is 20. The transmission has four reduction stages and can switch between two different ratios. The two multiple-disk clutches assume the differential function and can realize a torque-vectoring of the driven wheels. Each of the disk clutches is connected to one output shaft — one of which exits at the left side of the transmission directly with a drive shaft flange. In this transmission concept, the second output shaft is guided through a hollow motor shaft to the right-side drive shaft flange.

This transmission looks slick and clean, and is very well designed. The high ratio with the four-cylindrical gear-planetary stages — including the final drive gear set — requires the same amount of space as the electrical motor, which results in a significant width increase of this transmission. It may also be questioned, i.e. — does realizing the differential function with the multi-disk clutches present an adverse aspect regarding the concept of low energy consumption? Torque-vectoring should be done in certain driving conditions in order to improve traction and reduce or eliminate lateral sliding of the tires on the pavement (Ref. 7).

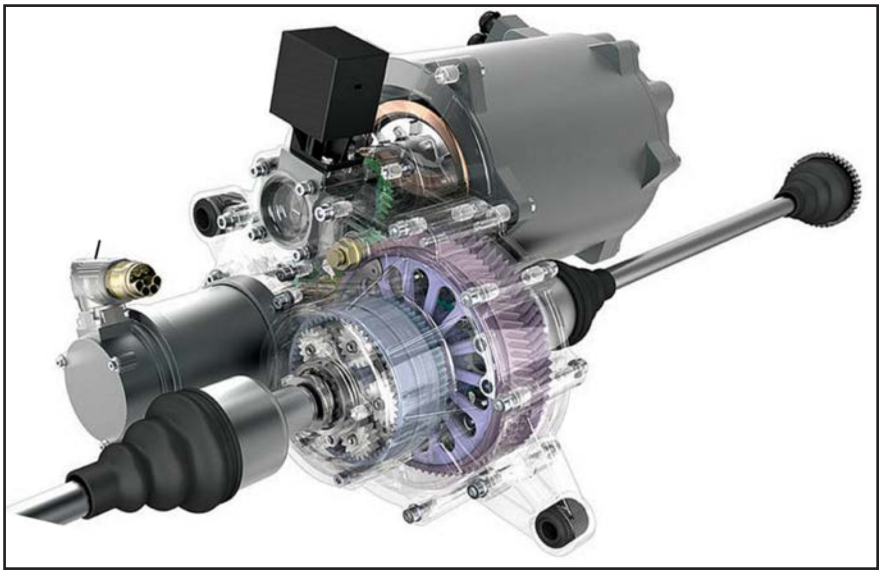

Figure 5 Three-stage variable ratio electric vehicle transmission — max ratio 18 (Ref. 4).

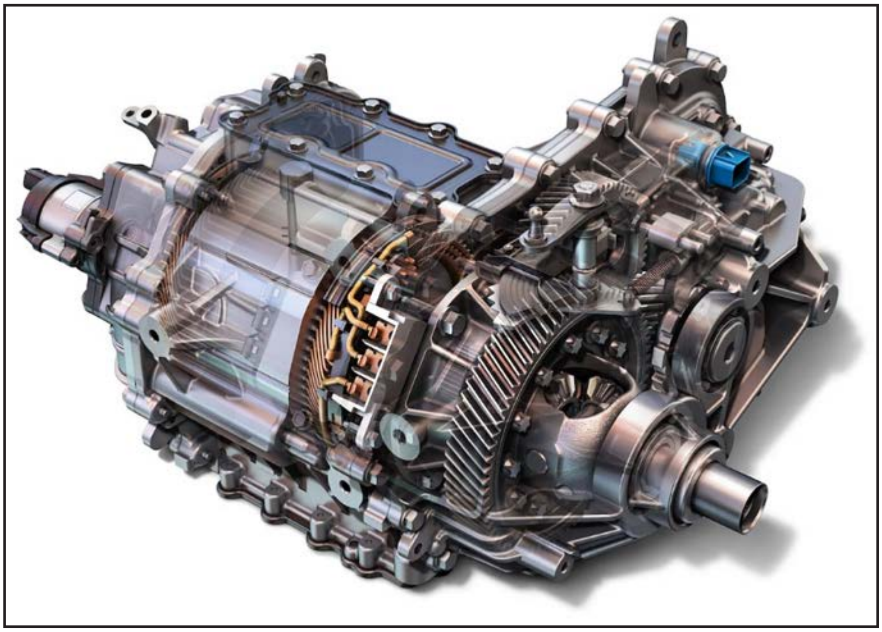

Figure 6 Two-stage and two-speed electric vehicle transmission — max ratio 7.05 (Ref. 5).

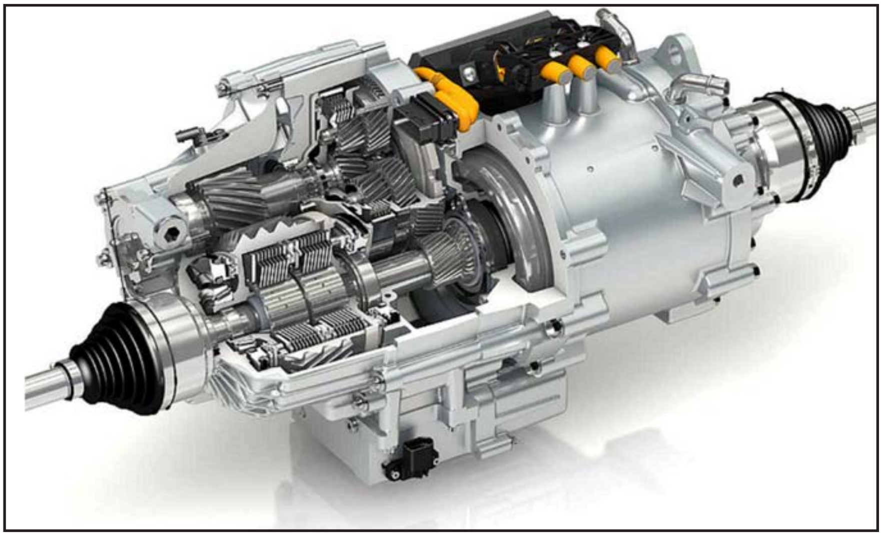

Figure 7 Four-stage and two-speed electric vehicle transmission — max ratio 20 (Ref. 6).

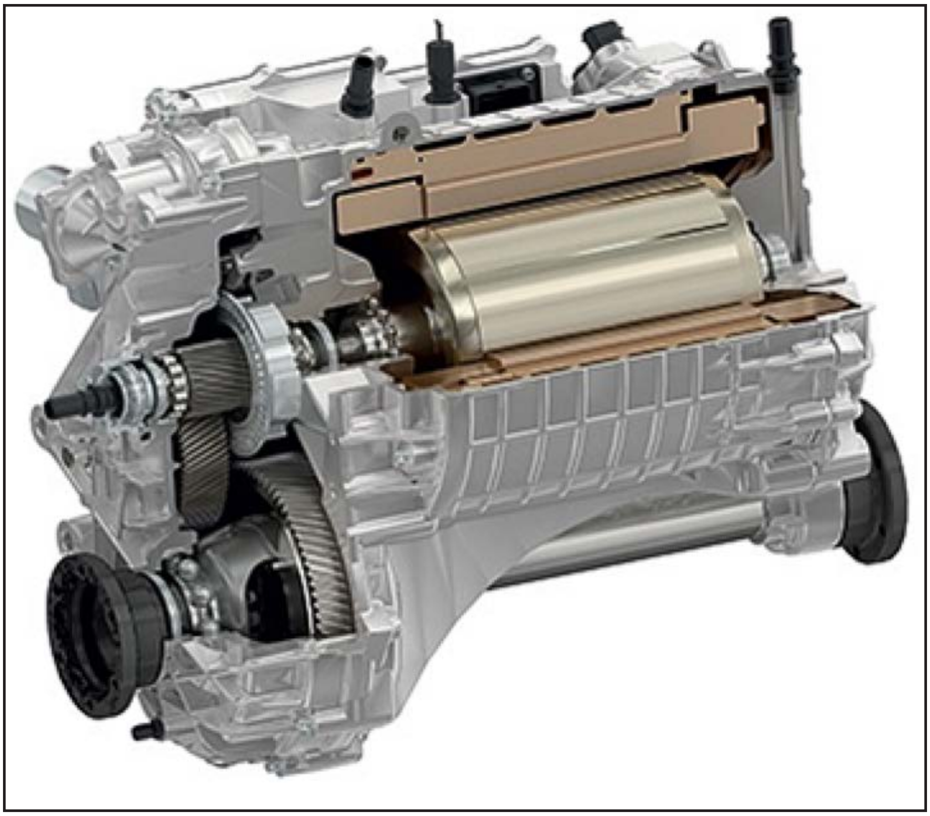

Figure 8 Two-stage and two-speed electric vehicle transmission — max ratio 12 (Ref. 8).

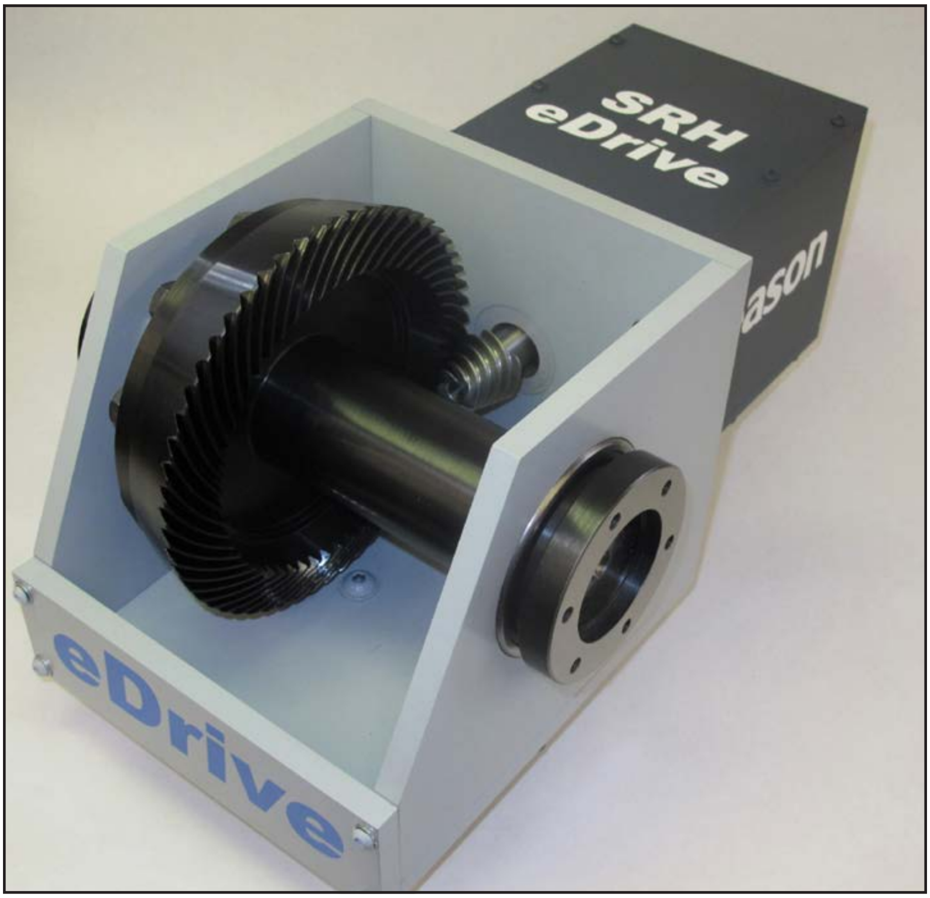

Figure 9 One-stage and single-speed electric vehicle transmission — max ratio 15 (Ref. 9).

Each torque-vectoring approach will burn energy from the motor between the sliding clutch disks. When the torque vectoring function is used to re-establish driving safety in an unsafe condition, then there is no alternative, which means the extra energy is well-spent. However, when the same function is used to emulate a differential — for example, when the vehicle is driving through a bend — then it contrasts with the traditional differential with virtually no friction due to the speed compensation with a very slow rotation of four straight bevel gears that have an efficiency rating of about 97%.

A small-diameter high-torque motor with a long rotor but a very small two-stage transmission that allows combining between two speeds is shown (Fig. 8). The differential unit is shown on the left, inside the final drive gear. The left drive shaft flange is directly connected to the left-differential side gear. Because the center of the differential unit is below the motor, a long shaft below the motor connects the right differential side gear with the output flange at the right side of the motor unit. This particular design represents an alternative to a more complex design where the connecting shaft of one side would have to pass through a hollow motor shaft as shown in the examples in Figures 6 and 7.

The stud above the left-side output flange is used to shift between the two gear ratios. The stud is part of a shaft that passes through a hollow gear shaft to actuate a synchronizing dog clutch. This principle has the advantage of minimal energy losses during gear shifting actions.

A very simplistic single-stage reduction utilizing a super reduction hypoid gearset is shown (Fig. 9); this transmission has a ratio of 12. Similar designs can realize ratios up to 15 without an increase of the gearbox size. The main advantage of this transmission is the small distance between the drive shaft flanges and the perfect symmetric weight distribution and heat radiation. Two shafts and four bearings make this transmission easy to build and cost-effective to manufacture.

Although hypoid gears are considered more difficult to manufacture, compared to cylindrical gears, there are certain advantages to the application of hypoid gears in electric vehicle transmissions.

The hypoid gears in Figure 9 are ground by using selective crowning. The selective crowning applies a so-called "Universal Motion Concept" that allows combining conjugate flank centers and relieved entrance and exit sections of the tooth mesh. This technology has been available in state of the art bevel gear manufacturing machines for many years and has proven to reduce load concentration while maintaining low meshing impacts under all load conditions.

In addition, advanced bevel gear grinding machines come with a high-frequency noise reduction feature, i.e. — "MicroPulse." A three-axis pulsing strategy creates a certain favorable surface structure which is then phase-shifted from tooth to tooth according to a calculated mathematical function. A surface structure that is shifted from tooth to tooth ensures that no two equal tooth meshes can occur throughout an entire hunting tooth sequence, which is a number of pinion revolutions equal to the number of gear teeth. For the example in Figure 9 (60 ring gear teeth) this means 60 pinion revolutions have to pass until the structure-related sequence repeats. A MicroPulse pinion rolls with a Formate-ground ring gear without pulse structure. This combination compares directly to the requirements of electric vehicle transmission gears discussed previously.

In-wheel electric motors. The solutions presented previously place an electric motor connected to a transmission between the wheels of one axle. It appears that an attractive solution is to use a direct-drive motor on each driving wheel. There are no drive shafts required, nor is a differential for driving through curves or disk clutches for torque vectoring necessary.

An example for a direct-drive wheel hub motor is shown (Fig. 10) (Ref. 5). The large outer-diameter of the blue motor rotor helps to achieve high torques at low speed — without the need for a transmission. The higher speeds are also not critical, because they are equal to the moderate wheel rpms at a certain vehicle speed. Earlier, it is mentioned that the wheels will only rotate with about 600 rpm while the vehicle is driving a speed of 67.9 km/h (42.44 mph). The "pancake style" wheel hub motor therefore is a high-torque and low-speed motor.

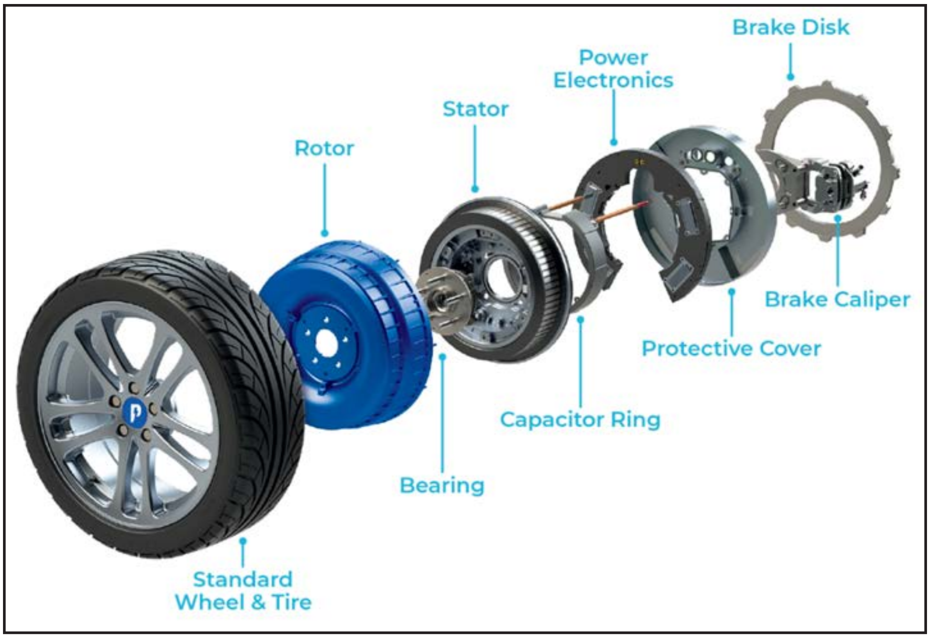

Figure 10 **In-wheel direct drive motor (Ref. 10).**

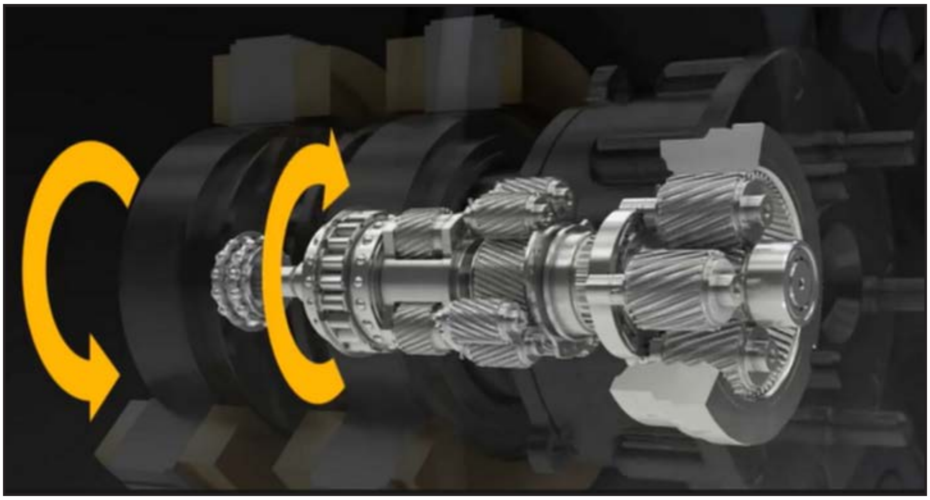

Figure 11 **Double-wheel hub motor with transmission (Ref. 11).**

As mentioned, most electric vehicle manufacturers tend to go to higher-speed and lower-torque motors with the goal to maximize system efficiency and reduce weight, even if this is only possible in connection with a transmission. Obviously, the developer of in-wheel motors discovered certain phenomena that help to overcome the physical restrictions other electric vehicle designers encountered.

The un-sprung weight connected to each individual wheel is definitely higher than in the case of the central drive units discussed earlier, but the developer of this system created very compact motor units that might even be able to replace the brake disk and associated weight entirely.

A wheel hub drive unit that features two motors that are connected to three planetary transmissions is shown (Fig. 11). The two motors can rotate in opposite directions in order to achieve an output rpm of zero. If both motors rotate in positive direction then, depending on the two motor speeds, each output speed between zero and the maximum driving speed can be achieved. If the first motor is a less-dynamic, high-torque motor and the second motor is a higher-dynamic, low-torque motor, then every driving condition can be efficiency optimized by the vehicle's electronic control module (ECM), which can adjust the optimal speed combination between the two motors.

Problems occur due to the length and the weight of the drive unit in Figure 11. If this unit is assembled to a front axle, then accommodating the steering will be difficult due to the length of the unit towards the center of the vehicle. Also, the un-sprung mass of this unit is rather high compared to the compact solution in Figure 10.

Improved electric vehicle from in-wheel to central drive. An example of a popular electric microcar is the Mitsubishi Miev (Fig.12). In 2006 the Miev was introduced with two in-wheel electric motors (Ref. 1). Even for this small-size vehicle, the in-wheel concept presented a number of problems. In order to make the Miev microcar attractive to buyers, Mitsubishi strived for a vehicle reach of 160km (100miles) per battery charge and an available output power of about 50 kW by achieving a maximum wheel torque of around 1400 Nm.

Besides the handling compromises due to the weight of the in-wheel motors causing trampling and lateral sliding of the driving wheels, the excessive energy consumption and the heat peaks of the in-wheel motors have been found to be difficult to eliminate. The heat peaks are initiated by short acceleration sprints that frequently occur in city driving. In slow-speed driving cycles the un-cooled wheel motors would require forced air or water cooling in order to stay below the temperatures that distort the wheels and changes the properties of the tire rubber.

Today, electric vehicle motor manufacturers promote rather high-speed motors in connection with multi-speed transmissions. Most of these motors can be operated without forced external cooling; their weight is low and their efficiency with the required transmissions is excellent.

The latest Mitsubishi Miev edition for the model year 2020 has a central electric motor with a two-speed transmission and a maximum ratio of 7.065. The e-Drive unit is connected to the rear wheels with drive shafts with each having two constant velocity joints. The rear axle is a modern, independent three-link de Dion design, as is rather expected in luxury premium vehicles or in sports cars. The maximum motor output torque of 196 Nm feels high for such a small size car. However, the battery weight added to the vehicle components results in a curb weight for the Miev of 1090 kg, which gives a reasonable but not overly high torque-versus-weight ratio of 0.19 Nm/kg.

The departure from in-wheel e-Drives can also be seen more and more for mid-size sedans as well as for SUVs and electric sports cars. This shows a clear trend in transmission solutions with multiple speeds that can adapt the rpm of high-speed motors within their optimal efficiency and torque range, resulting in better electric vehicles — which not only avoids the use of gasoline but also minimizes the consumption of electrical energy.

The Following Chapters of this Book

The motivation for writing this book is based upon the many discussions and requests from electric vehicle builders regarding specific solutions which Gleason may be able to offer. These discussions made the requirement portfolio of such transmissions more clear and Gleason scientists began to think in the direction of transmissions with high input speed and low output speed, using a minimum of transmission stages and allowing multiple reduction ratios.

Chapter 2 presents a revisiting of the basics in propelling vehicles with internal combustion engines, hybrids and electric vehicles; a major part of Chapter 2 is the comparison of overall efficiencies. Standard efficiency factors are used for each transmission element, and then the factors are multiplied to gain an overall efficiency. By using this technique, it becomes nearly secondary if the assumed efficiency factors are correct or slightly deviating from the latest effective values. The task of comparing different concepts to propel a vehicle can be accomplished with this strategy very well, because the same efficiency factors are used for the same transmission elements — independent from the vehicle concept. One example of this study is the hybrid without any transmission and wheel hub electric motors (Fig. 13).

The goal of Chapter 2 is to find motor and transmission concepts that result in high, overall efficiency. As alternative ways of propelling future vehicles are investigated, the concepts with the highest overall efficiency should be considered with a higher priority because there will be already a reduction of consumed energy before the details of hybrid, hydrogen or electric propulsion are even decided.

While Gleason Corp. began actively thinking about good electric vehicle transmission solutions, the idea was borne to present a first solution at the 2018 Japanese International Machine Tool Fair, i.e. — JIMTOF in Tokyo, Japan. Figure 14 shows one of the solutions presented at the JIMTOF.

Chapter 3 reports about this first solution, which was the application of high-reduction hypoids and super reduction hypoids (SRHs). At the JIMTOF Show, a closed loop development of transmission housing, shaft and bearings, as well as the SRH reduction and additional cylindrical gear shift ratios, was shown. The transmission design began in *KISSsys* and the hypoid design was conducted with the Gleason *GEMS* software. A closed loop optimization between *KISSsys* and *GEMS*

Figure 12 Mitsubishi Miev (Ref. 12).

for the optimization of deflections and contact forces uses a specially written dynamic *XML* interface that was demonstrated in life workshops during the show.

The first technical paper about SRH e-Drives was presented at the 2019 *Fall Technical Meeting* of the **American Gear Manufacturers Association** (**AGMA**) in October of 2019. Figure 14 shows one of the solutions presented in this paper. The high interest in technologically new and advanced transmission solutions with extremely high ratios and high efficiency led to the development of the reversed pericyclic transmission.

An example of a reversed pericyclic transmission is shown (Fig. 15). Significant advantages of pericyclic transmissions are the low relative sliding between the engaged tooth surfaces and the high contact ratio between the meshing gears. The low sliding velocity enables pericyclic transmissions to handle high speeds regarding surface damages and wear rather well. In common cases like the one shown in Figure 15, more than 10 pairs of teeth are engaged and contribute to the transmission of the output torque, which makes these transmissions insensitive to torque peaks and shock loads. A disadvantage of pericyclic transmissions is the nutating "wobble motion." The inertia forces due to this motion always require, in high-speed applications, a pair of two nutating gears back to back with a phase shift of 180°. Such a timed relationship between the two nutating gears accomplishes a complete cancellation of the inertia forces.

The fascination of planetary gears and differentials led to a third development presented in Chapter 6 of this book. They are three-dimensional planetary transmissions. Their function becomes more complex if a second differential is placed around the first differential. Depending on the way some of the differential gears are connected to each other or constraint to the housing, very low or very high ratios can be realized. An example of a double-differential with a ratio of 78 is shown (Fig. 16). Next to the easy realization of high ratios, there are tangible advantages of double-differentials versus conventional gear reductions. It begins with the natural reduction of the relative speed between the gears in mesh due to the carrier rotation. Higher input

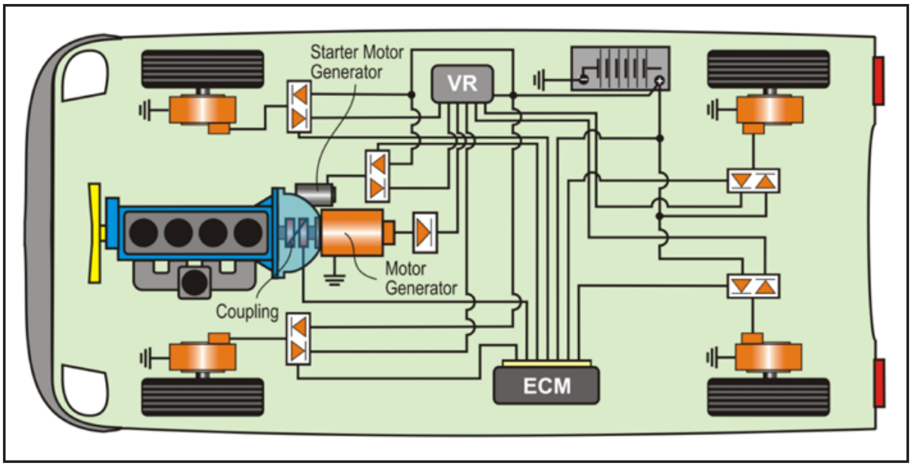

Figure 13 **Hybrid vehicle with in-wheel electric motors (Ref. 13).**

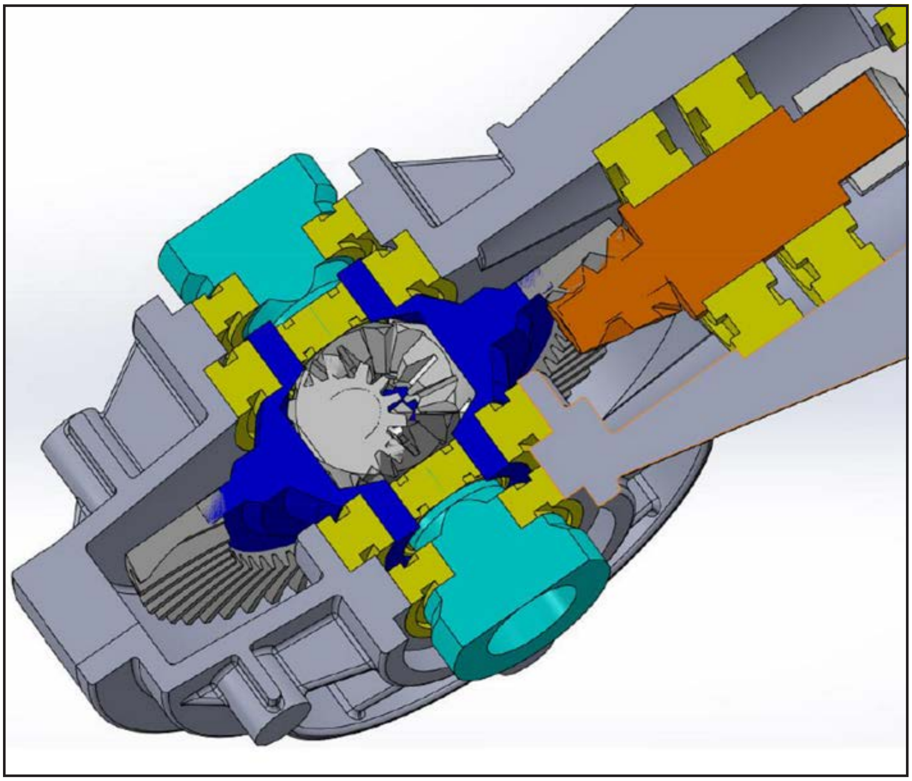

Figure 14 **Single-stage electric vehicle hypoid transmission (Ref. 9).**

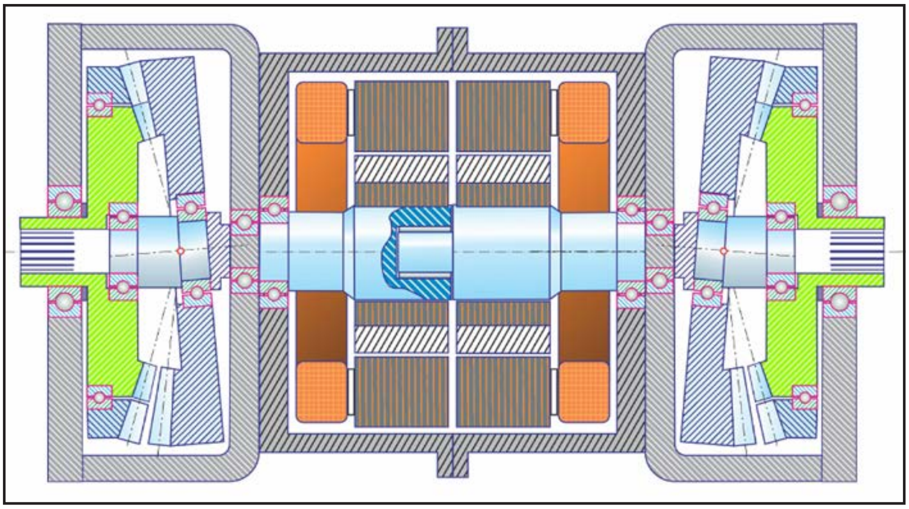

Figure 15 **Reversed-pericyclic transmission.**

speeds are permissible because the relative rotational speed between the meshing gears is about 50% of the input speed. Also, the contact forces are reduced to 50% because the torque is always transmitted by two equal opposite gears. Transmission housing deflections are symmetric and will only increase the size of the transmission housing without any warping.

For more information.

Questions or comments regarding this paper? Contact Dr. Hermann Stadtfeld — *hstadtfeld@gleason.com.*

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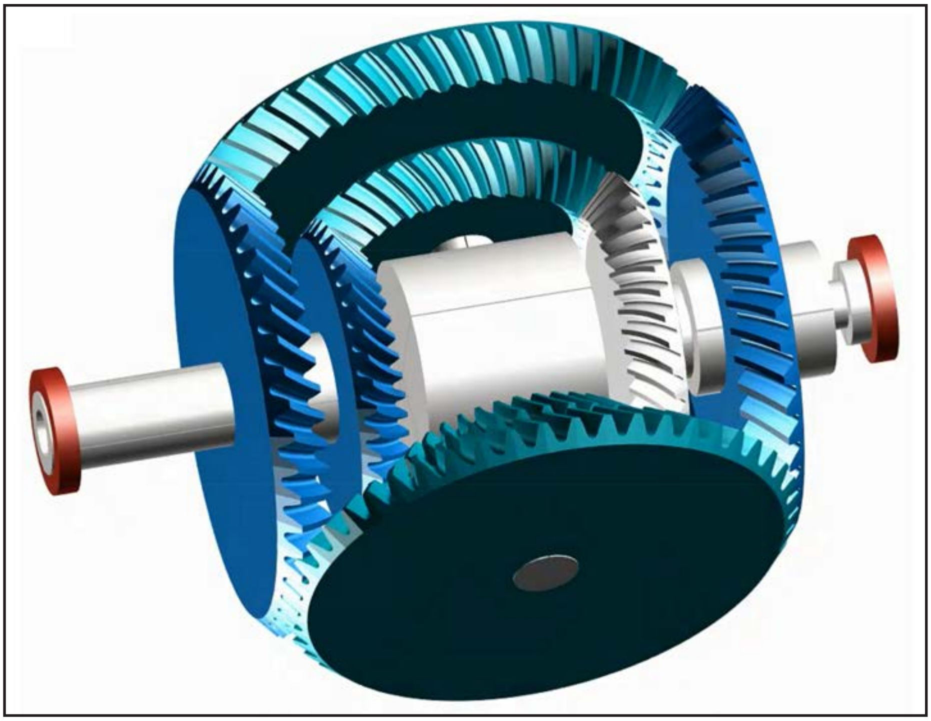
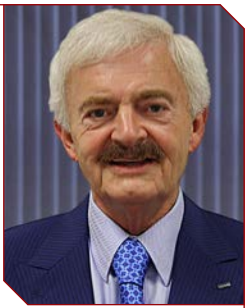


Figure 16 **Double-differential transmission with ratio 78.**



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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Spiral-Bevel Gear Noise: An Approach to Solving In-Field Issues

Claude Gosselin, Ph.D., P.Eng., Involute Simulation Softwares Inc., Canada
Bastian Leitz, B.Eng., Neugart GmbH, Germany

Introduction

Gear noise is a common evil any gear manufacturer must live with. It is often low enough not to be a major problem but, at times, gear whining may appear and then, tracking the source and, especially, curing the ill can be tricky at best.

A large number of publications, too numerous to be listed here, have been published on gear noise; the work of Endo and Sawalhi is a typical example (Ref. 1). However, once the gearbox is in production, such models are of little use.

The Gleason Works also markets advanced software and machinery systems (Ref. 2) to run gear sets and determine at which positions the best behavior is found. Such systems are well-adapted to large-batch manufacturing.

This paper presents an approach that allows identifying the source of the noise and offers avenues in correcting the issue using more limited means, which is typical of small-to-medium- size gear manufacturers.

Problem Statement

A reversing gearbox comprising a spiral-bevel gear set has been in production for several years. Noise had not been mentioned by the customer until recently, when different testing procedures were introduced. Gear stall torque is ~80 Nm, but testing rather replicates nominal conditions.

The pinion convex flank can be operated at different torques and rpms without undue noise, but the pinion concave flank exhibits annoying noise levels around 1,400 and 3,200 rpm.

Table I lists the main specifications of the gear set. Both members are hard finished using CBN grinding wheels of different point widths.

Table I Spiral bevel gearset specifications

	Pinion	Gear
Z	9	59
Module [mm]	2.5	
Spiral angle [0]	35	
Cutting Process	Duplex Helicall	Non generated
Finishing	Grind—CBN GW	Grind—CBN GW
Max RPM	3,500	
Nominal Torque [Nm]	3	19.67

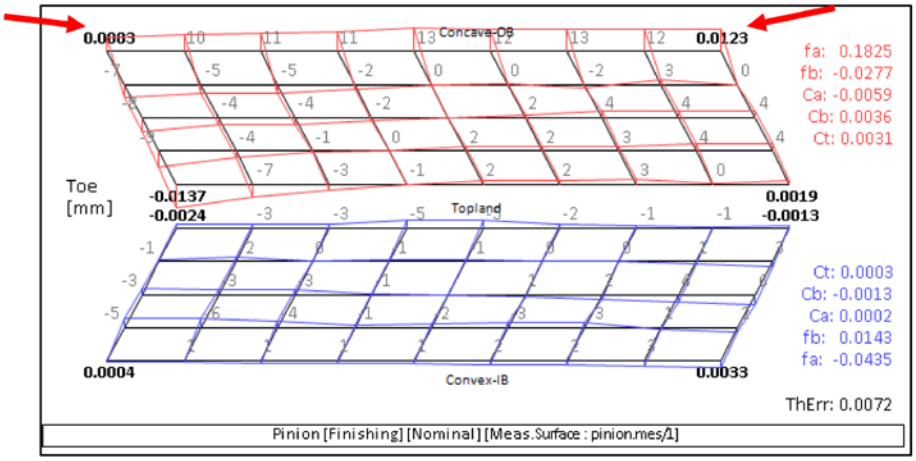


Figure 1 Pinion CMM results.

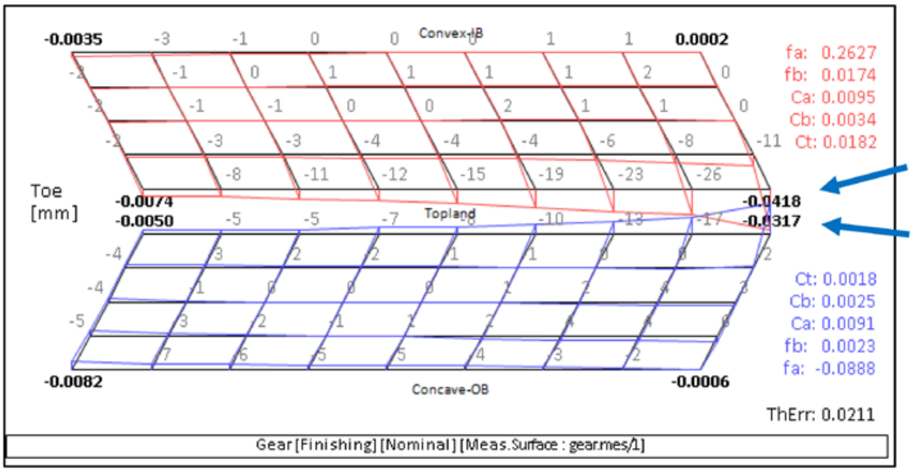


Figure 2 Gear CMM results.

Preliminary Assessment

The pinion and gear were measured on a Wenzel CMM, giving the results shown in Figures 1 and 2. Both the pinion and gear show overall minor deviations relative to the nominal surface, but we note that on the concave flank, (Fig. 1), the pinion exhibits some positive profile curvature near the root (red arrows), which is expected to cause premature contact entry. The gear, Figure 2, exhibits tip relief towards the heel (blue arrows), which is usually beneficial.

Because of manufacturing tolerances on both the gearbox housing and the gear teeth, the pinion and gear were shimmed in order to provide the visually, and therefore subjectively, best contact patterns.

Actual contact positions in the gearbox are (definitions shown in Fig. 3):

- E = -0.007 mm, caused by gearbox manufacturing tolerances
- P = -0.019 mm, to obtain an acceptable contact pattern position
- G = +0.005 mm, to obtain the correct backlash

The contact patterns measured in the above positions are shown (Fig. 4). Contact patterns on both gear flanks are fairly well centered lengthwise on the tooth, and appear to cover most of tooth depth.

Using the *HyGEARS* (Ref. 3) software and accounting for the measured errors shown in Figures1 and 2, the contact patterns shown in Figure 5 are obtained.

Although the calculated contact pattern on the gear concave flank is quite similar to that of Figure 4, the contact pattern calculated on the gear convex flank differs substantially from that shown (Fig. 4).

The contact patterns were therefore re-measured — this time using a much thinner marking compound to better reveal the boundaries. The results are shown in Figure 6 where, clearly, the calculated contact patterns now correlate very well with the measured ones shown in Figure 6.

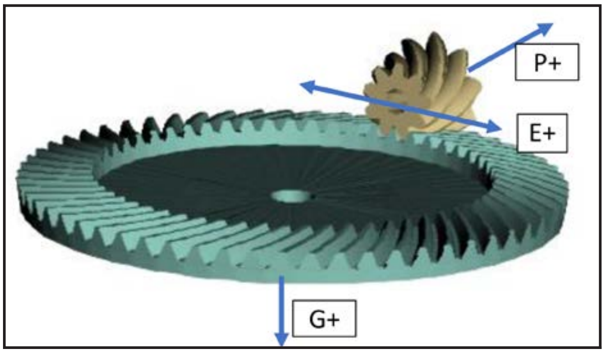
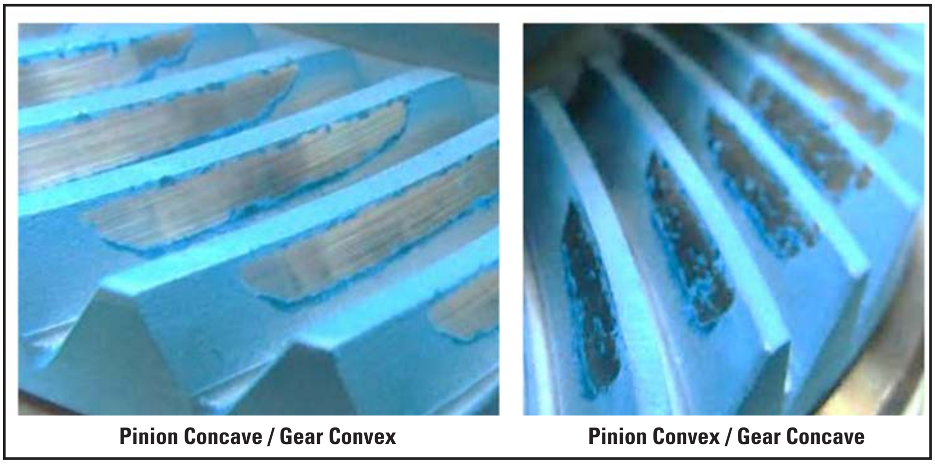
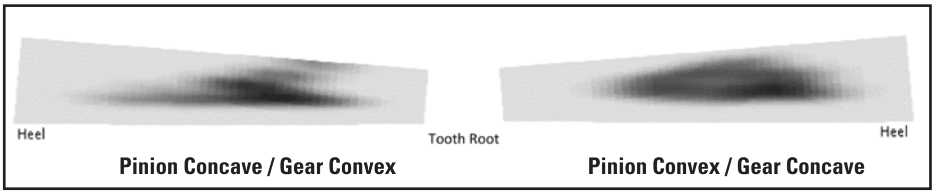


Figure 3 E-P-G definitions.


Pinion Concave / Gear Convex

Pinion Convex / Gear Concave

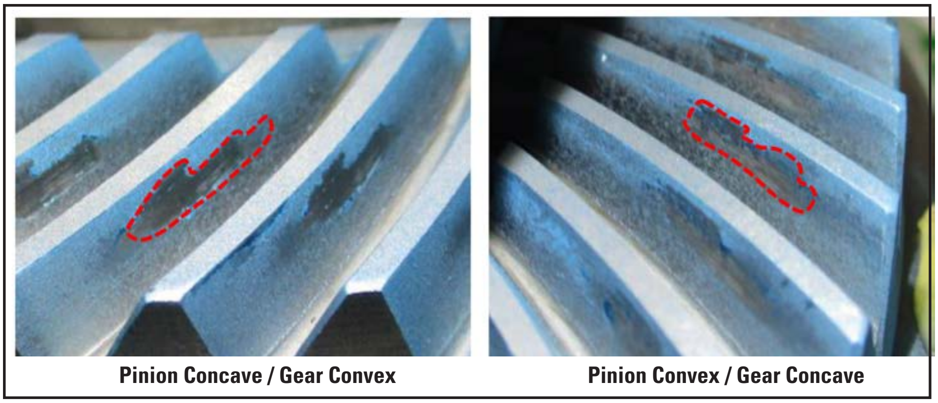
Figure 4 Measured contact patterns: thick marking compound.



Pinion Concave / Gear Convex

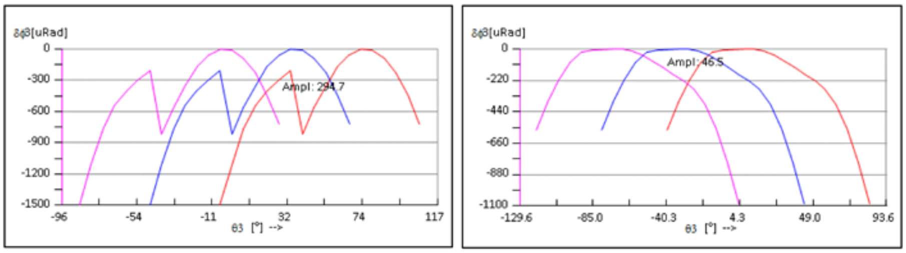
Pinion Convex / Gear Concave

Figure 5 Calculated contact patterns.


Pinion Concave / Gear Convex

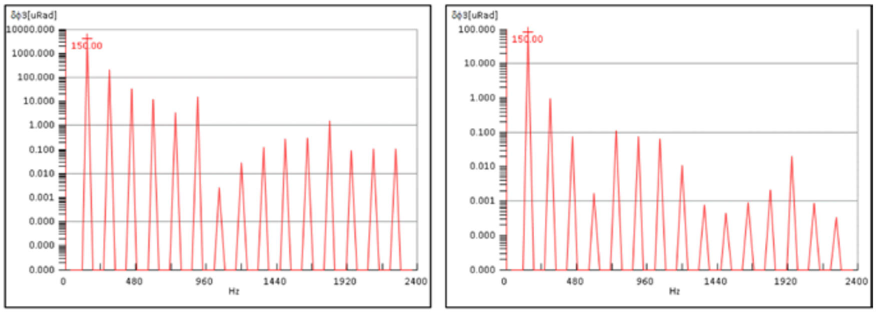
Pinion Convex / Gear Concave

Figure 6 Measured contact patterns: thin marking compound.



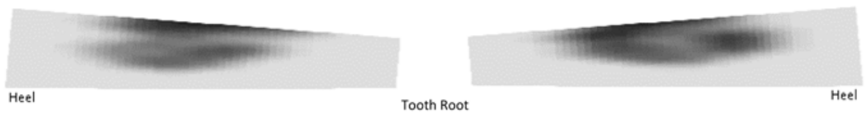
Pinion Concave / Gear Convex **Pinion Convex / Gear Concave**

Figure 7 No load transmission error — calculated from the measured surfaces.



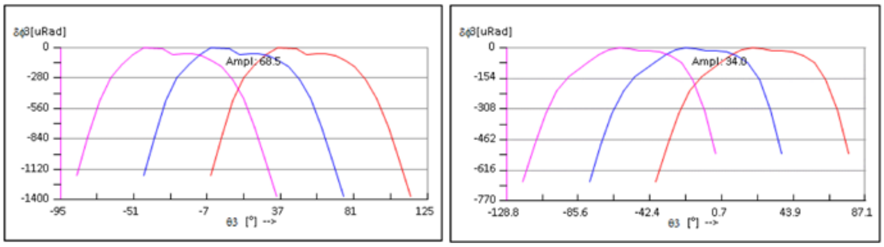
Pinion Concave / Gear Convex **Pinion Convex / Gear Concave**

Figure 8 No-load TE FFT — calculated from the measured surfaces.



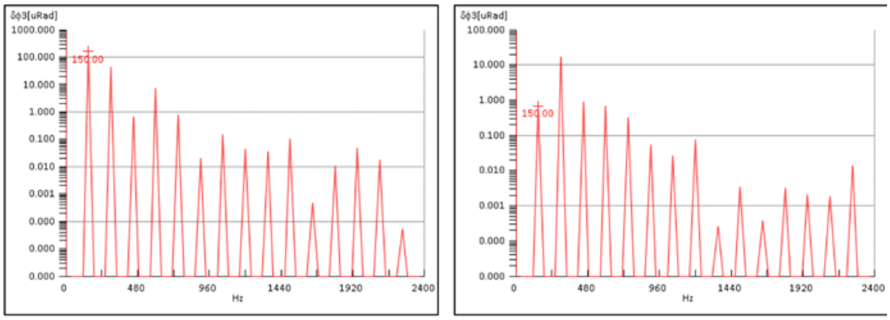
Pinion Concave / Gear Convex **Pinion Convex / Gear Concave**

Figure 9 Improved contact patterns — calculated using the measured surfaces.



Pinion Concave / Gear Convex **Pinion Convex / Gear Concave**

Figure 10 Improved TE — no-load — calculated using the measured surfaces.



Pinion Concave / Gear Convex **Pinion Convex / Gear Concave**

Figure 11 Improved no-load TE FFT — calculated using the measured surfaces.

Transmission Error and Gear Noise

The good correlation between the measured and calculated contact patterns shown in Figures 5 and 6 indicates that the tooth contact analysis (TCA) model evaluates correctly the no-load kinematics (Ref. 4). The resulting transmission error curves (TE) for 3 consecutive meshing tooth pairs — pink, blue, orange — are shown (Fig. 7).

The TE curves on the pinion convex flank (right figure, below) are of convex shape, continuous, and with transfer points (TP) at a depth of ~46 µ*Rad* — a value frequently found in gearsets of this module; rotation proceeds from left to right in the graph.

By contrast, the TE curves on the pinion concave flank, although also of generally convex shape, show a much deeper TP at 295 µRad, which is likely to cause a sharp acceleration when motion is transferred from one tooth pair to the next. Rotation proceeds from left to right in the graph.

Figure 8 shows the FFT of the TE curves displayed in Figure7. Clearly, based on the amplitude of the 3–4 first harmonics, the pinion concave flank (left below) is expected to be noisier than the convex flank (right below) which, again, correlates with what was noted.

Of course, the graphs shown in Figure 8 do not indicate at which rpm, or torque, noise is to be more prevalent since the actual gearbox, bearings, shafts, etc. are not modeled. However, they indicate that there is a potential noise issue — which is validated in practice.

Improving Contact Pattern and Gear Noise

There are basically three solutions to the situation depicted above, which derives from the measured manufacturing errors:

a. Apply closed loop to eliminate the profile curvature at root noted on the pinion concave flank — Figure 1.; of course, this means new pinions and that the gearboxes already in use are to be recalled — a very expensive operation at best.
b. Measure and correct the CBN grinding wheel used for pinion finishing, concave flank; the same gearbox recalls and cost issues as in a) are expected.
c. Modify the pinion and gear installation in the gearbox to try and improve the contact patterns and TE.

Of course, solution c), if a practical combination of pinion and gear shimming can be obtained, is by far the easiest and least expensive as it involves a simple operation that can easily be performed in the field.

Again, using the gear simulation model, the following combination of EPG was found; note that only P is changed.

- E = –0.007 mm (imposed by the manufacturing tolerances on the gearbox housing)
- P = +0.173 mm
- G = +0.005 mm (no change)

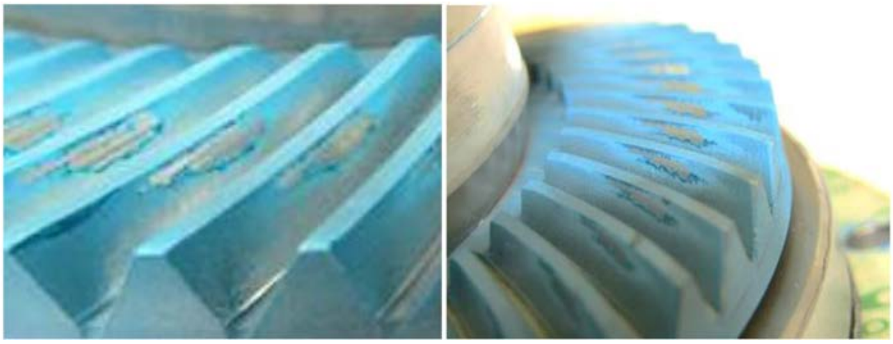
The resulting no-load contact patterns, TE and FFT, are shown in Figures 9–11. In particular, for the pinion concave flank, Figures 10–11 show a dramatic change when compared to Figures 7–8.

Of course, other P and G combinations could be used, but the above-selected combination involving pinion shimming only results in lower efforts and costs.

Assessment of Improved Contact Pattern and Vibrations

Figure 12 below shows the contact patterns measured in the modified EPG positions while Figure 13 shows the calculated contact patterns in the same gear tooth orientation to ease comparison. Clearly, the predicted behavior is obtained.

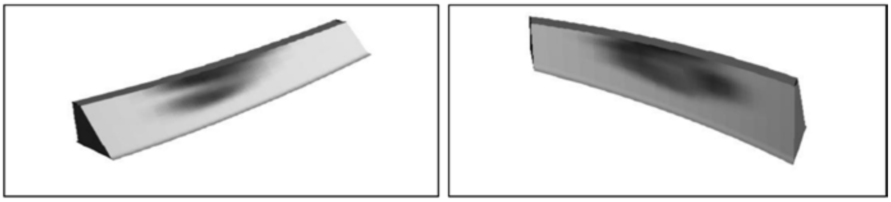
The gearset was run at different torque levels and rpms in the original, and improved EPG positions and vibrations levels were recorded. The results, appearing in Figures 14–15, show a significant improvement in vibration levels when shimming the pinion position from P = –0.019 mm to P = +0.173 mm; not only on the pinion concave flank, which was originally the problem tooth flank, but also on the pinion convex flank, which was originally found as acceptable.



Pinion Concave / Gear Convex

Pinion Convex / Gear Concave

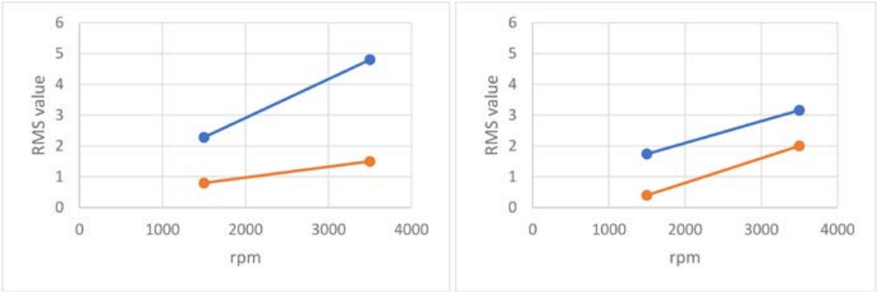
Figure 12 Measured contact patterns P = +0.173 mm.



Pinion Concave / Gear Convex

Pinion Convex / Gear Concave

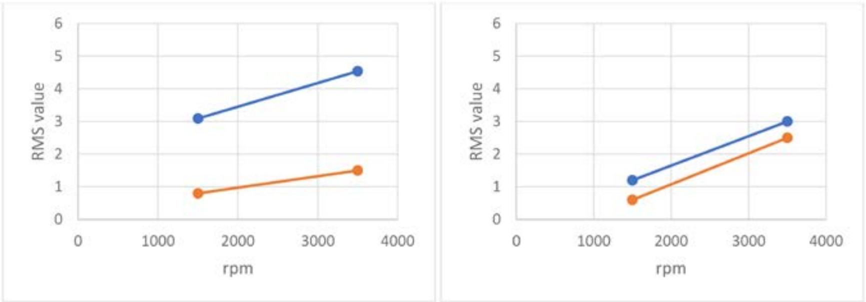
Figure 13 Calculated contact patterns P = +0.173 mm.



Pinion Concave / Gear Convex

Pinion Convex / Gear Concave

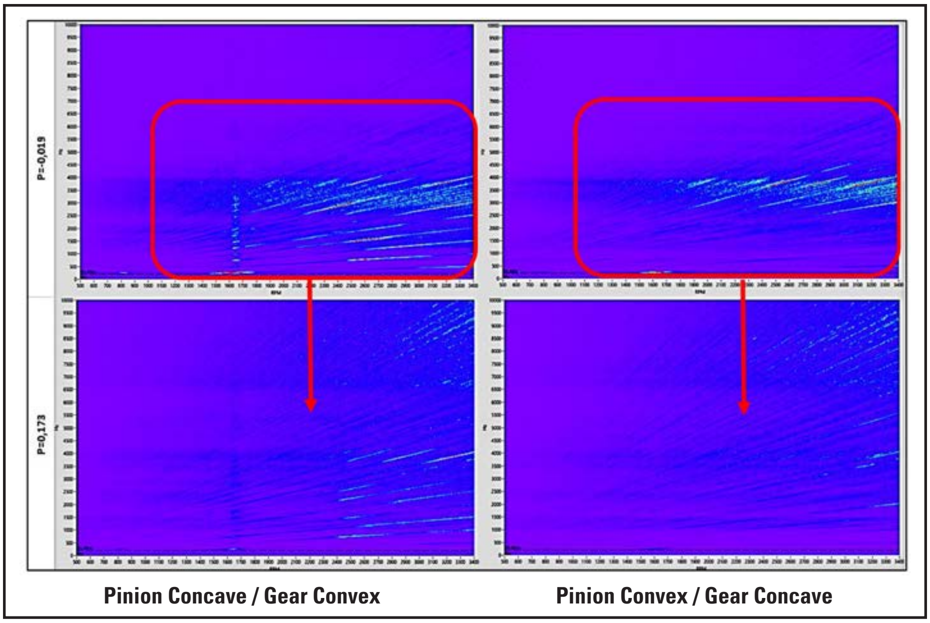
Figure 14 Mechanical vibrations: 0 Nm pinion torque P = –0.019 mm P = +0.173 mm.



Pinion Concave / Gear Convex

Pinion Convex / Gear Concave

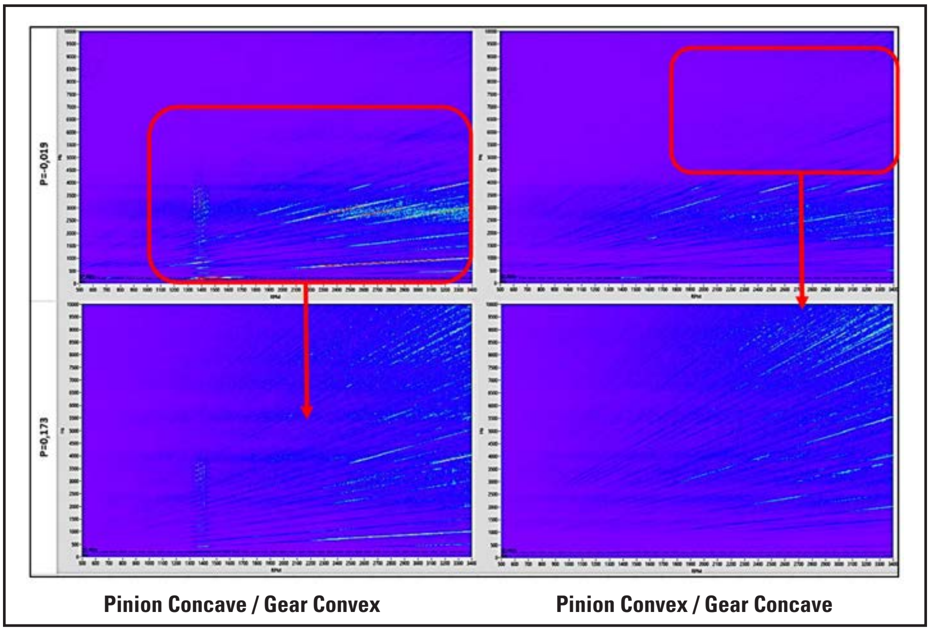
Figure 15 Mechanical vibrations: 3 Nm pinion torque P = –0,019 mm P = +0,173 mm.



Pinion Concave / Gear Convex

Pinion Convex / Gear Concave

Figure 16 Waterfall plots: 0 Nm pinion torque.



Pinion Concave / Gear Convex

Pinion Convex / Gear Concave

Figure 17 Waterfall plots: 3 Nm pinion torque.

Figures 16–17 show the waterfall plots for the original and modified operating positions for, respectively, 0 Nm and 3 Nm pinion torque. Of course, 0 Nm pinion torque means that no braking other than friction is applied at the output. rpm on the horizontal axis ranges from 500 to 3,400, whereas frequency in Hz, on the vertical axis, ranges from 0 to 10,000.

Both graphs show the same dramatic reduction in noise for the pinion concave flank operating in the modified position. On the pinion convex flank, Figure 17 shows a slight increment at frequencies above 5,500 Hz for 3 Nm pinion torque.

Overall, noise improvement on the gearbox renders it *more than acceptable* to the customer, which was the aim with this analysis.

Conclusion

Gear noise, which results from mechanical vibrations transmitted to the gearbox housing and environment, is a problem that is often difficult to cure. While highly evolved mathematical models of gear trains, including shafts, bearings and gearboxes, are available, they are of little use to the manufacturer once the gearbox and components are in production and noise is experienced at certain rpm and torque levels.

This paper presents an approach focusing on the use of tooth flank topography measurement (CMM) data coupled to spiral bevel gear modeling software to analyze how a problem gearset operates in a given installation. It is shown that the software calculated contact patterns closely match those actually measured on the gearset, which leads to the conclusion that the calculated transmission errors are also close to what is actually occurring.

This leads to the use of the software to try and improve the contact patterns by shifting the pinion and gear positions, an operation easily achieved in practice through shimming, thereby improving transmission error and allowing in-field correction of the noise problem.

A sample 9x59 duplex helical pinion / non-generated gear spiral-bevel gearset is used to validate the approach. While the pinion convex tooth flank could mesh at different torque levels without undue noise up to 3,400 rpm, the pinion concave flank exhibited significant noise at around 1,400 and 3,200 rpm.

The spiral bevel gear modeling software was used to calculate new pinion and gear installation positions, based on the CMM data, in order to improve on both the contact patterns and transmission error.

Results confirm that the improved contact patterns actually resulted in significantly lower noise levels on *both* tooth flanks, thereby allowing in-field correction of the noise problem at a very low cost.

While the demonstrated methodology is applied to a spiral bevel gearset, it is applicable to any type of gear to allow the identification of the noise source, and is easily used with all types of bevel gears since their respective mounting distances directly affect contact pattern location and transmission error.

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Sizing of Profile Modifications for Asymmetric Gears

Ulrich Kissling

Introduction

Lately, the use of asymmetric gears in automotive and other applications is an upcoming trend, though few applications are known to have asymmetric teeth. However, an increased interest in asymmetric gears can be seen (Refs. 2, 12–13). Many companies have started to design and test such applications.

The pressure angle in the normal section of an asymmetric gear is different for left and right flank. This can be used as an advantage compared to symmetric gears, since higher pressure angle increases the pitting capacity (Ref. 4). For gears rotating mostly in one direction, the loaded flank of the tooth can be designed with a high pressure angle and the non- (or seldomly) loaded flank with a lower pressure angle. For a given gear pitch, a low pressure angle on one flank permits the loaded flank to have a higher pressure angle compared to a symmetric gear (while avoiding a pointed tip, for instance). The main benefit of asymmetric gears over symmetric is higher load capacity for a given design (Ref. 4). The main drawback is a more complex manufacturing process (for steel gears) and related costs.

Currently there is no standardized method available for the geometry and strength calculation of asymmetric gears. For the calculation of bending strength, a method proposed by Langheinrich (Ref. 3) can be used, which is a combination of an analytical (based on ISO 6336) and an FEM approach. For the calculation of flank strength, the ISO 6336 calculation can be applied. Other gearing characteristics such as efficiency, micropitting, and scuffing can be checked based on the same calculation methods as for symmetric gears.

When an asymmetric gear design is evaluated, the following must be carefully analyzed:

- The potential gain in power capacity
- The eventual increase of the manufacturing costs
- The difference in noise/vibration behavior

For asymmetric gears, just like symmetric gears, profile and lead (flank line) modifications are important sources of improvement for torque capacity and noise/vibration behavior, as well as other characteristics. Due to the different profile of the left and the right flank, different modifications must be applied on both flanks. This again makes the manufacturing process more complicated. It's often possible to apply modifications to only the higher-loaded flank, but then noise performance of the lower loaded flank may be unsatisfactory.

Profile and Lead Modifications

The application of tooth modifications for asymmetric gears is like the technique used for symmetric gears. Modifications are mainly used to compensate for tooth and shaft deflection and manufacturing errors. Lead modifications are used to compensate for deflection of shafts, bearings and housing deformation. Profile modifications compensate primarily for tooth bending to avoid contact shock (premature start of meshing contact) and to reduce the transmission error (PPTE). Additionally, modifications may be used for other purposes, such as the reduction of Hertzian pressure in areas of the tooth flank where high pressure may cause issues.

Lead modifications are applied to compensate for shaft bending and torsion, misalignments due to manufacturing errors, bearing clearance, deformation and housing influence. Optimal lead modifications will normally increase the torque capacity of the gearbox due to a more even load distribution along the flank, thus reducing the face load factor $K_{H\beta}$. Typically, a helix angle modification is applied to compensate for shaft misalignments and crowning to compensate for general manufacturing errors and torsional effects.

An efficient layout technique to design modifications is to subdivide the task in three steps:

The first step is to apply the lead modifications for optimal load distribution over the face. Once the optimum modification is defined, the second step is the profile modifications. Profile modifications are more difficult to define and optimize, due to the different, and sometimes contradictory, requirements that must be fulfilled.

Various effects such as lower contact temperature, higher efficiency, smooth normal force distribution, and higher

Table 1 Symbols

Symbol	Description	Units
b	Face width	mm
c'	Maximum tooth stiffness per unit face width (single stiffness) of a tooth pair	N/mm/μm
$C_{\gamma\beta}$	Mean value of mesh stiffness per unit face width, secant value used for KHp	N/mm/μm
$f_{P(i)}$	Single pitch deviation of the paired gear	μm
$C_{\alpha a}$	Tip relief	μm
F_{bt}	Nominal transverse load in plane of action	N
$K_{H\beta}$	Face load factor (1S06336)	-
L	Length on involute	mm
L_{ca}	Length of the tip relief	mm
α_n	Normal pressure angle.	°
ε_α	Contact ratio under load	-

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micro-pitting resistance may be achieved. The third step is to recheck the lead modification. But normally the load distribution over the face is not strongly changed by profile modifications, and so no or small adaptations are sufficient.

Load distribution over face calculation according to ISO6336-1, Annex E. For the application of the lead modification, a procedure, as described in ISO 6336-1, Annex E (Ref. 7), can be used for a fast and straightforward design of an optimum lead modification (Ref. 8). The procedure is relatively simple, compared with a full LTCA (see next section), and considers shaft misalignment due to bending, torsional deformation and manufacturing errors. Bearing stiffness/offset and housing deformation are also included (Fig. 1). Basically, this method is a one-dimensional contact analysis, providing the load distribution along the operating pitch diameter.

The method in ISO 6336 is not designed for asymmetric gears. But the only value, influenced by the asymmetric tooth form, is the meshing stiffness $[c_{\gamma\beta}]$. This value can be obtained with a LTCA calculation for the asymmetric gear pair or with the slightly adapted (Ref. 3) formulas of ISO 6336-1 for single stiffness c' and meshing stiffness $c_{\gamma\beta}$ (Table 2).

Loaded tooth contact analysis. The detailed effects of modifications must be checked with a complex calculation procedure, the loaded tooth contact analysis (LTCA). The aim of LTCA is to evaluate the gear mesh under load. For a consistent number of steps over one pitch during the rotation of the gears, the contact between all teeth under load is calculated. For the calculation of tooth deformation, a tooth stiffness model is required. An analytical model for tooth deformation was presented by Weber & Banaschek (Ref. 1), where gear deformation is divided into three main components:

- Gear body deformation
- Tooth bending deformation
- Hertzian flattening

Based on this method, an analytical stiffness model can be created. A loaded tooth contact analysis can then be performed based on the tooth deformation, shaft misalignments, manufacturing errors (e.g. pitch error) and a defined partial load for the calculation (Fig. 2). The results of LTCA provide important

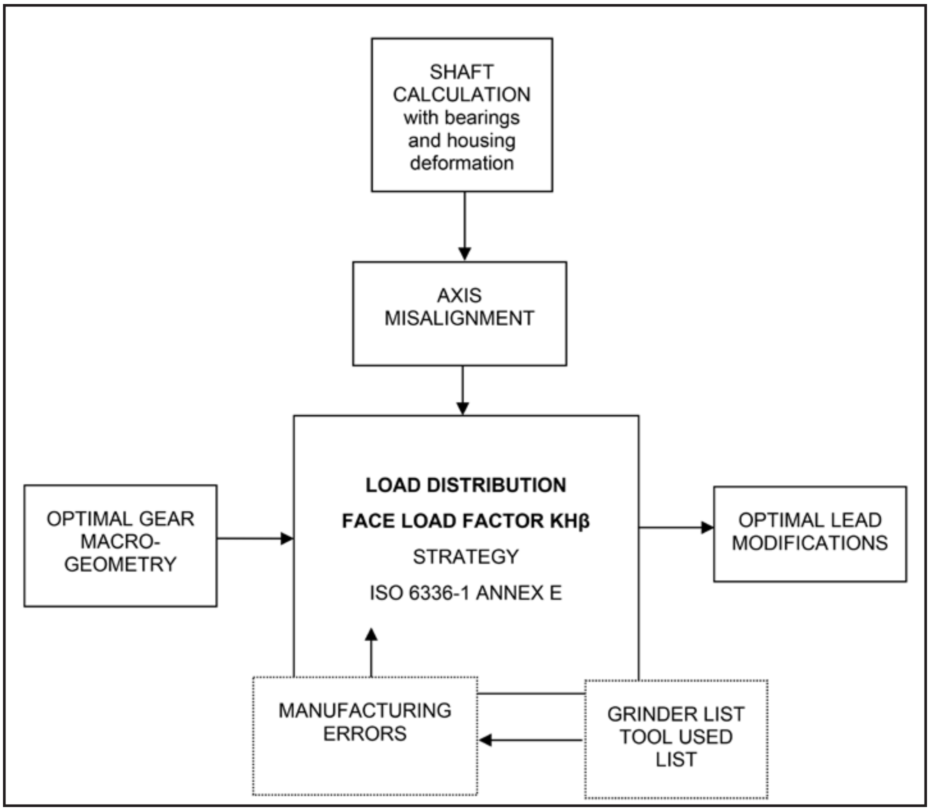

SHAFT CALCULATION with bearings and housing deformation
AXIS MISALIGNMENT
OPTIMAL GEAR MACRO-GEOMETRY
LOAD DISTRIBUTION
FACE LOAD FACTOR KHβ
STRATEGY
ISO 6336-1 ANNEX E
OPTIMAL LEAD MODIFICATIONS
MANUFACTURING ERRORS
GRINDER LIST TOOL USED LIST

Figure 1 Optimization of face load distribution with lead modifications.

Table 2 Mean stiffness according ISO6336 and LTCA

	Mean value for single stiffness c′		Mean value for mesh stiffness cep	
All values in N/mm/µm	ISO 6336	LTCA	ISO 6336	LTCA
Symmetric gear, α_n 24°	13.6	10.8	15.5	19.4
Asymmetric gear, right flank, α_n 34°	14.8	12.6	15.1	19.8
Asymmetric gear, left flank, α_n 14°	12.3	9.9	17.5	21.3

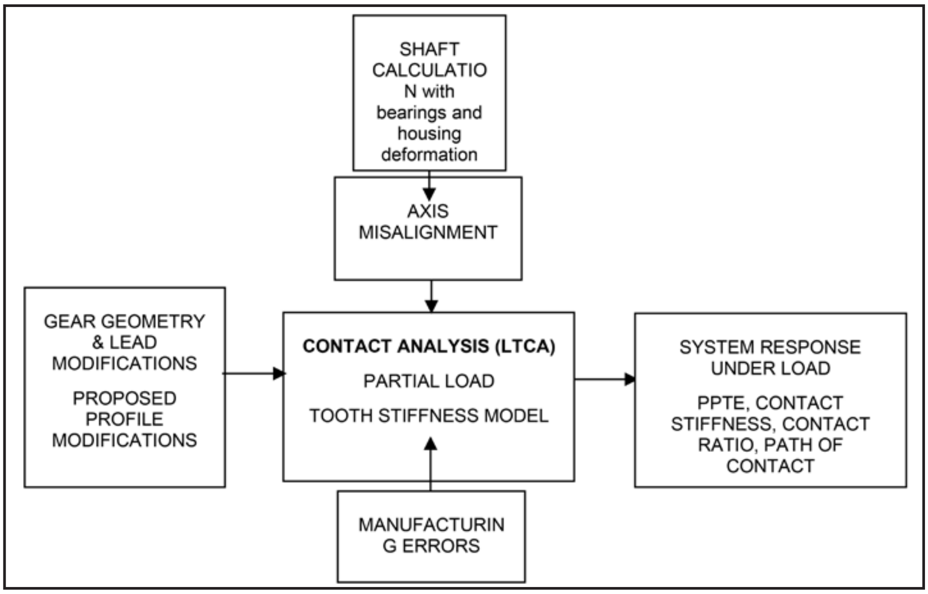
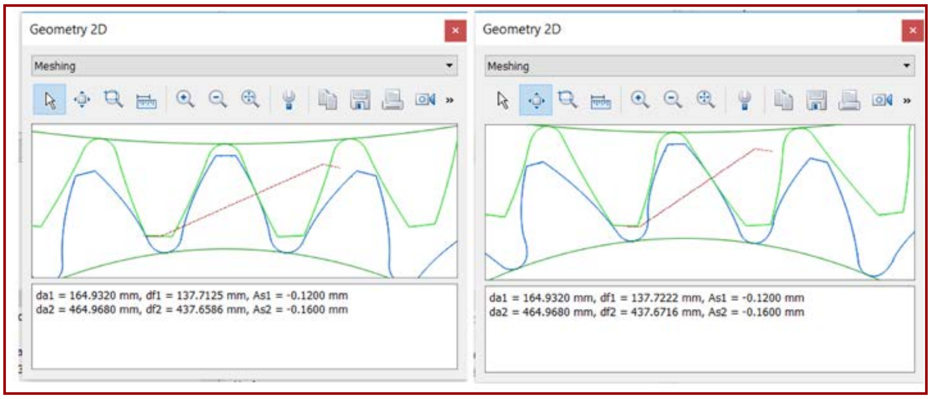

SHAFT CALCULATIO N with bearings and housing deformation
AXIS MISALIGNMENT
GEAR GEOMETRY & LEAD MODIFICATIONS
PROPOSED PROFILE MODIFICATIONS
CONTACT ANALYSIS (LTCA)
PARTIAL LOAD
TOOTH STIFFNESS MODEL
SYSTEM RESPONSE UNDER LOAD
PPTE, CONTACT STIFFNESS, CONTACT RATIO, PATH OF CONTACT
MANUFACTURIN G ERRORS

Figure 2 Loaded tooth contact analysis.



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da1 = 164.9320 mm, df1 = 137.7125 mm, As1 = -0.1200 mm
da2 = 464.9680 mm, df2 = 437.6586 mm, As2 = -0.1600 mm
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```
da1 = 164.9320 mm, df1 = 137.7222 mm, As1 = -0.1200 mm
da2 = 464.9680 mm, df2 = 437.6716 mm, As2 = -0.1600 mm
```

Figure 3 **Tooth form of a symmetric gear pair (pressure angle $\alpha_n = 24°$) and an asymmetric pair (pressure angle left/right $\alpha_n = 14°/34°$).**

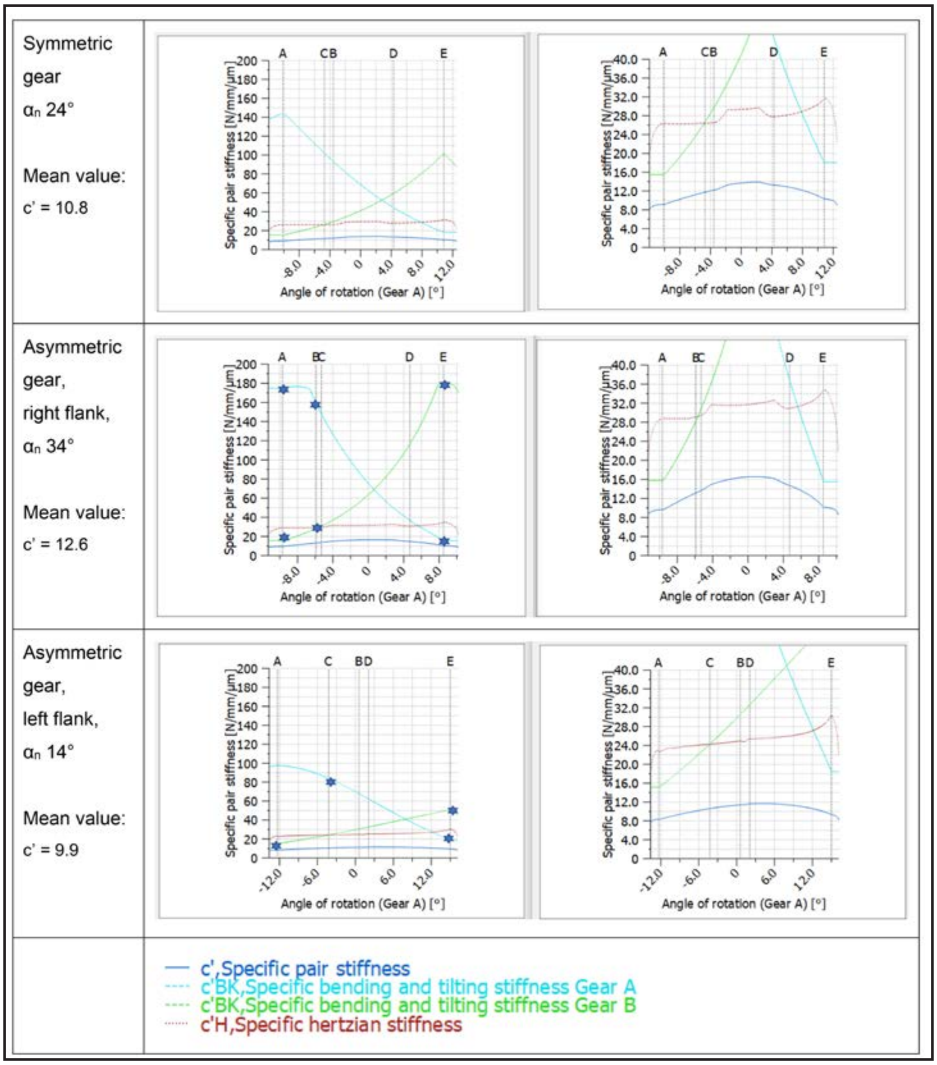


Figure 4 **Mesh stiffness c' of a single tooth pair over a meshing cycle, bending stiffness for a symmetric and an asymmetric gear pair; graphs on the left and rig, 0 to 40 (right). Indicates stiffness value calculated by FEM.**

parameters for noise characterization and optimization:

- Transmission error;
- Amplitude spectrum of the transmission error;
- Force excitation;
- Path of contact under load, etc.

The semi-analytical process according Weber & Banaschek (W&B) (Ref. 1) is very efficient and reliable, used by some of the best-known software programs (*as RIKOR;* Ref. 11) and others) for symmetric gears. The method describes the deformation of a tooth as a combination of four phenomena: The bending and the shear deformation of a one-sided clamped beam, the Hertzian flattening in the meshing contact and the rotation of tooth in the gear body. An LTCA calculation can also be performed with an FEM tool, but every LTCA result requires a large number of individual FEM calculations (to calculate different contact positions during a gear mesh) and is very time consuming.

As it is not easy to directly find a modification set providing a good solution, many variants must be checked. Therefore, a consistent amount of LTCA calculations is needed. Doing this with a FEM-based method is not recommended. Kapelevich (Ref. 2) proposes a partially FEM based method to speed up calculation time, but only for spur gears.

As the W&B method is accepted, quick, and well documented, the author, together with Mahr & Lang-Heinrich (Ref. 5), decided to adapt the method for asymmetric gears. The method is implemented in the contact analysis of *KISSsoft* (Ref. 9) and compared with FEM results. This now permits the study of the behavior of asymmetric gears with applied modifications.

Contact Analysis for Asymmetric Gears

Single tooth pair stiffness and meshing stiffness. The tooth stiffness of an asymmetric gear is quite different from symmetric gears. As previously mentioned, tooth stiffness is composed of tooth bending, gear body deformation and Hertzian flattening. The primary difference originates from the tooth bending. Hertzian flattening is different for the left and the right side of the asymmetric tooth due to the different flank curvatures, but identical to a symmetric tooth having the same pressure angle. Gear body deformation is similar to that of a symmetric tooth, with the exception of the load application angle.

Figure 3 shows a symmetric and an asymmetric gear pair with tooth numbers 25:76 (case carburized spur gear with module 4.0 mm, face width 44 mm, torque on pinion 1,600 Nm, speed 441 rpm). For these gears Figure 4 shows the single stiffness *c'*, which is the stiffness of a single tooth pair over the contact path. Also, the components of *c'* are displayed, the stiffness due to Hertzian flattening and due to the tooth bending and gear body deformation (tilting) of both gears. The stiffness for bending of the right flank with the higher pressure angle is *two to three times* higher than the stiffness on the left flank! Still, the single tooth pair stiffness c' is not so different: *c'* is just 20% lower on the lower pressure angle side. To verify the results, at some meshing positions the bending stiffness was also calculated by FEM. These points are indicated by a blue star in Figure 4.

The mesh stiffness, $c_{\gamma\beta}$, is the mean value of the stiffness of all the teeth in a mesh. For the determination of the face load factors $K_{H\beta}$ and $K_{F\beta}$, a line is constructed in the load-deflection graph between the origin and the pertinent load point used for evaluation of $c_{\gamma\beta}$. The mesh stiffness $c_{\gamma\beta}$ is displayed (Fig. 5). Although the stiffness for tooth bending is quite different for the left flank versus the right flank, the mesh stiffness is only 7% higher on the lower pressure angle side. It should be noted that *c' is lower, but is* $c_{\gamma\beta}$ higher, on the lower *pressure angle side*! This is due to the increase in profile contact ratio, ε_α, which is greater than two on the lower pressure angle side; therefore, at least two teeth are always in contact under load, increasing the total stiffness (Fig. 5).

As discussed earlier, the mean mesh stiffness $c_{\gamma\beta}$ is needed for the calculation of the load distribution over the gear face. The formulas provided in ISO 6336-1 (Ref. 7) for symmetric gears, adapted for asymmetric gears by Langheinrich (Ref. 3, Chapter 7), give relatively good results, compared with results obtained with the more sophisticated LTCA calculation. Therefore, for a first approach, the ISO 6336 stiffness values are practical.

Contact analysis. The stiffness model is an important component of the LTCA method. For this analysis, the gear is cut into several transverse sections and the stiffness is calculated for these slices. For a spur gear with lead modifications, or a helical gear, the beginning and end of contact of the slices is dependent upon the position of the slices along the tooth width. The total stiffness is calculated by integrating the stiffness functions for the slices over the width, with increasing delay at the begin of contact. In the simulation of the meshing the deflection of the teeth is introduced by the normal force applied divided by the stiffness.

Since the point where the force is applied varies in the height direction, the stiffness depends on the meshing position. Further, if an additional pair of teeth comes into contact the stiffness increases sharply, so the deflection of the first pair of teeth is reduced. To find the correct point of contact, an iteration must be performed.

Basically, the stiffness is the only part where a LTCA method for symmetric gears must be adapted for asymmetric gears. Clearly the sense of rotation is now significant, as this decides which flank is in contact.

Sizing of a Tip Relief Modification

The final step in any gear design is the definition of the profile modifications. As mentioned before, different features such as noise, contact temperature, efficiency, micro-pitting or scuffing can be improved with well-sized profile modifications. The reduction of noise/vibration generation is based on the following strategy:

- Eliminate contact shocks at the beginning and at the end of the mesh
- Reduce the amplitude of the transmission error (PPTE)
- Reduce the second and higher order of harmonics of PPTE to become as close to zero as possible

In ISO 21771 (Ref. 10) various modification types are defined. Typically, a tip relief (Fig. 6) on both gears is applied to reduce gear noise. The amount of tip relief $C_{\alpha a}$ is adjusted to eliminate contact shocks and the tip relief roll length L_{ca} is chosen to minimize PPTE.

In Niemann's book (Ref. 6), a useful suggestion for a reasonable tip relief is documented (Table 3). The values for $C_{\alpha a}$ depend on the gear stiffness numbers (*c'* for spur and $c_{\gamma\beta}$ for helical gears) and the single pitch deviation f_p. Based on a long

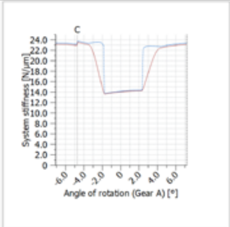
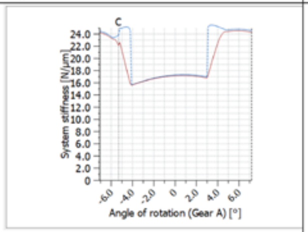
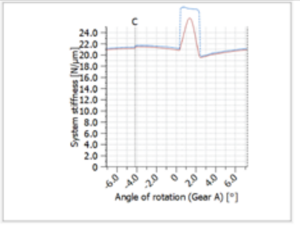
Symmetric gear, α_n 24°	Asymmetric gear, right flank, α_n 34°	Asymmetric gear, left flank, α_n 14°
		
Mean value for single stiffness: c' = 10.8 N/mm/µm	c' = 12.6 N/mm/µm	c' = 9.9 N/mm/µm
Mean value for mesh stiffness: $c_{\gamma\beta}$ = 19.4 N/mm/µm	$c_{\gamma\beta}$ = 19.8 N/mm/µm	$c_{\gamma\beta}$ = 21.3 N/mm/µm
Contact ratio under load ε_α = 1.70	ε_α = 1.49	ε_α = 2.07

Figure 5 Stiffness (of all the teeth in a mesh) (blue dashed line: tangent stiffness $c_{\gamma\alpha}$; red line: secant stiffness $c_{\gamma\beta}$).

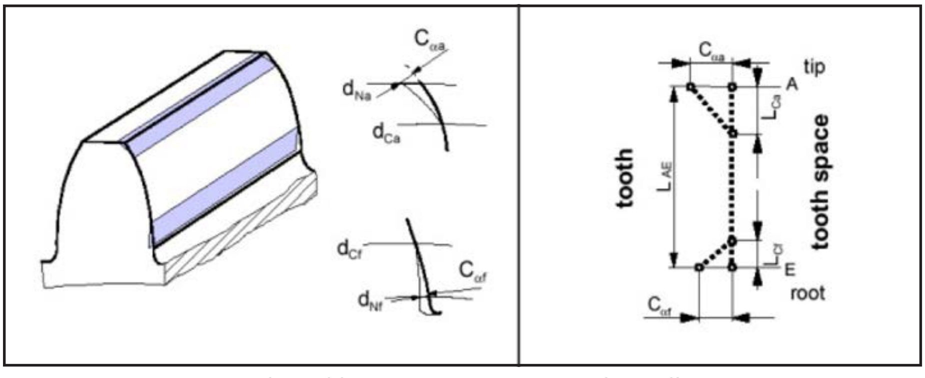


Figure 6 Tip and root relief (linear) (Nomenclature acc. ISO6336 (Ref. 10)).

experience with these propositions, it can be confirmed that for a first guess the values are appropriate; this applies also for asymmetric gears. With the correct amount of tip relief $C_{\alpha a}$, the pressure at the tip diameter should reach zero — and thus eliminate contact shock.

The length of the tip relief Lca has a great effect on the reduction of the peak-to-peak transmission error; PPTE is a valuable parameter for noise optimization. The Fourier transformation provides the orders of harmonics and allows the evaluation of excitation frequencies. From the transmission error and the contact stiffness, it is possible to derive the excitation force (EF), which allows the comparison of different geometric solutions in terms of vibration excitation.

In literature (Ref. 6), short and long modifications are proposed. The contact of the gears on the path of contact starts at point A and ends at E, going over meshing point C and the single contact points B and D as defined in ISO 21771 (Ref. 10). The short modification is applied from A to AB, respectively from DE to E. The long modification is applied from A to B, respectively from D to E.

As a general rule, which gives good results in many cases, the short modification increases the torque capacity of a gear set because high pressure and scoring risk in the tip area is reduced, but no noise improvement will be achieved (PPTE is not reduced). The long modification in most cases reduces PPTE significantly, but also reduces torque capacity.

Profile modifications can be defined by many different parameters. Tip relief is the easiest modification to apply in manufacturing, so in this paper for the sizing of the modification, tip relief is used. The main parameters are the amount of tip relief $C_{\alpha a}$ and the length L_{ca} of the modification. The form of the relief, if linear, arc-like or crowned, has also a significant, but less dominant, effect. So, for the sizing of an optimum profile modification, mostly $C_{\alpha a}$ and L_{ca} are crossed-varied.

General Procedure for the Sizing of Profile Modifications

Optimization of profile modifications in a case-by-case manner is extremely time-consuming and demanding. Results of an LTCA are not easy to evaluate. Comparing results of different LTCA calculations with slightly changed modifications is even more challenging.

Knowing this problem, a concept was developed, in partnership with a German gear company, the "modification sizing" tool. The basic idea is to systematically vary properties of an unlimited number of modifications. The possibility to cross-vary properties of individual modifications (such as tip relief to length of modification) is also available. With this, a certain number of variants with different modifications are defined. Then for every variant a full LTCA is performed and all relevant data is stored. This can be time-consuming if hundreds of variants are analyzed, but the process is fully automatic. To be able to provide such a process and still get a solution with hundreds of variants in a reasonable time (such as one hour), it is so important to use the modified Weber & Banaschek approach for the LTCA of asymmetric gears.

To explain the procedure, an example is discussed which was made for a US/Italy-based company. The main issue of a gear transmission was noise/vibration, but the Hertzian pressure was just at the limit at higher torques. The agreement made was that the noise had to be minimized at mean torques, but the pressure at high torques should not be increased.

Such problems are typical. Often noise problems must be optimal at the most typical load conditions, usually close to the mean load; but stress parameters must be optimal at peak loads. It is not possible to get optimum performance with modifications for every torque level, because tooth deformation is torque-dependent. But it also must be verified, that a modification with good results at mean torque does not perform poorly at the lower and higher torques.

The asymmetric gear pair discussed here has tooth number 30:43, module 4 mm, helix angle 17° and a pressure angle on the right flank of 31° (face width 40 mm, torque on pinion 1,682 Nm, speed 2,180 rpm). The actual lead modifications were checked with good results, and therefore not changed. For the profile modification it was decided, due to manufacturing restrictions, to apply only a tip relief. Then the modification variation process was used. Tip relief $C_{\alpha a}$ is varied in 6 steps, from 18 to 48 µm, and the modification length factor L_{ca}^* (in module) also in 6 steps, from 0.5 to 5.5. All the values are cross-varied, therefore 36 variants are generated. Additionally, as a reference, the variant with no tip relief (declared number 0) is calculated. Both gears were synchronized, so in every variant both gears have the same profile modifications. This is a reasonable decision, otherwise 36*36 variants would have been generated (Fig. 8). More importantly, every variant is run at three torque levels (declared as min,

Table 3 Proposal for tip relief according Niemann (Ref. 6)

	Start of contact (for the gear, if pinion is driving)		End of contact (for the pinion, if pinion is driving)	
Tip relief	$C_{\alpha a}$ (min)	$C_{\alpha a}$ (max)	$C_{\alpha a}$ (min)	$C_{\alpha a}$ (max)
Spur gear	$\frac{F_{bt}}{c'b}+f_{p(i)}$	$\frac{F_{bt}}{c'b}+2f_{p(i)}$	$\frac{F_{bt}}{c'b}$	$\frac{F_{bt}}{c'b}+f_{p(i)}$
Helical gear	$\frac{F_{bt}}{c_{\gamma\beta}b}+\frac{f_{p(i)}}{2}$	$\frac{F_{bt}}{c_{\gamma\beta}b}+\frac{3f_{p(i)}}{2}$	$\frac{F_{bt}}{c_{\gamma\beta}b}$	$\frac{F_{bt}}{c_{\gamma\beta}b}+f_{p(i)}$

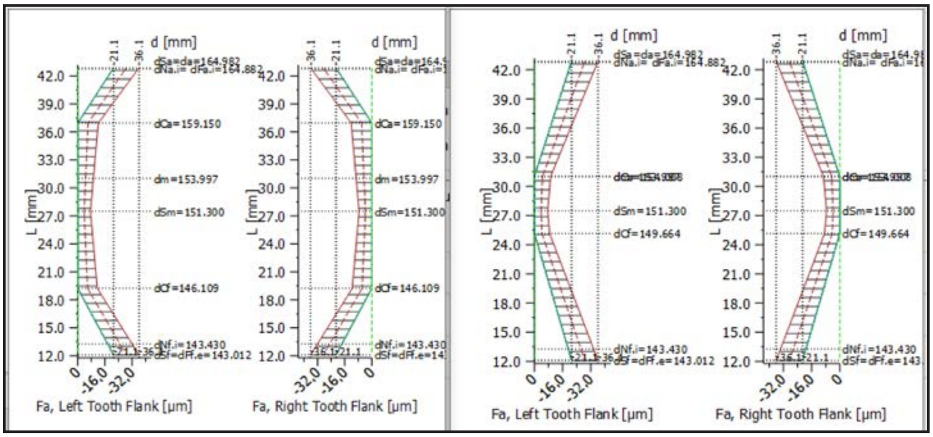


Figure 7 Profile diagram of a short (right) tip and root relief and a long relief (left).

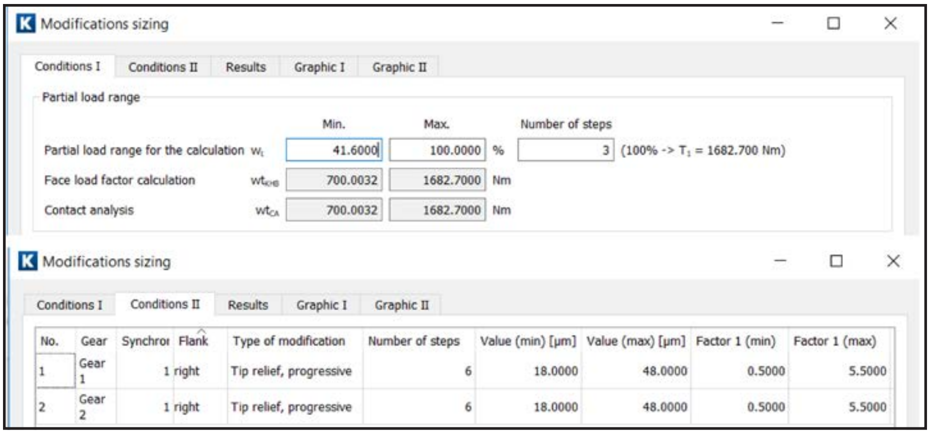
mean and max). This allows the analysis of the results depending on the torque level. With 3 torque levels and 36+1 variants, a total of 111 LTCA calculations are performed.

The main LTCA results of every variant are collected in a list (in csv-format) and presented in different graphics. It is not easy to extract the optimum solution from such a large amount of data! In Figure 9 the spider diagram technique is used to give an overview of the most pertinent data. The solution 0 considers only the lead modifications (no profile modification). The solutions from 1–6 have equal $C_{\alpha a}$, the smallest value (18 µm), while the length of the modification L_{ca}* vary from 0.5 to 5.5. The solutions from 7 to 13 have a $C_{\alpha a}$ increased by 5 µm, corresponding to the profile slope deviation $f_{H\alpha}$ tolerance, and the same behavior for the length factor.

The use of the profile slope deviation $f_{H\alpha}$ tolerance as a step between different $C_{\alpha a}$ values tested is a clever strategy. If, for example, variant 11 is selected as optimal, then variant 5 and variant 17 have the same modification length, but 5 µm more material (variant 5) or 5 µm less material (variant 17) on the profile, therefore showing what happens if the manufactured tooth is at the min or max limits of the profile slope deviation.

The result overview in Figure 9 shows the profile contact ratio ε_α under load, together with the theoretical ε_α (orange circle). If the profile contact ratio under load is bigger than the theoretical value, than noise due to a contact shock will be generated. So, solutions for mean load (green line) outside of the orange circle should be avoided. The PPTE graph shows clearly which solutions have low values for all torque levels, therefore the choice for a variant with good performance can be made. The Hertzian pressure display shows as a red line the Hertzian pressure at the maximum torque. Solution 0 indicates the actual value (approx. 1,200 N/mm^2), which should not be increased.

Which solution is best? The first step is to check the contact ratio ε_α at the medium load. Solutions 13, 19, 25 and 31 are close to the theoretical contact ratio (orange curve) and have a good PPTE, but the Hertzian pressure is too high; solutions 2, 8, 14, 20 are better.



Partial load range

		Min.	Max.		Number of steps	
Partial load range for the calculation	w_L	41.6000	100.0000	%	3	(100% -> T_1 = 1682.700 Nm)
Face load factor calculation	wt_{KHB}	700.0032	1682.7000	Nm		
Contact analysis	wt_{CA}	700.0032	1682.7000	Nm		

No.	Gear	Synchror	Flank	Type of modification	Number of steps	Value (min) [µm]	Value (max) [µm]	Factor 1 (min)	Factor 1 (max)
1	Gear 1	1	right	Tip relief, progressive	6	18.0000	48.0000	0.5000	5.5000
2	Gear 2	1	right	Tip relief, progressive	6	18.0000	48.0000	0.5000	5.5000

Figure 8 Process input data for the modification variant generator (Ref. 9).

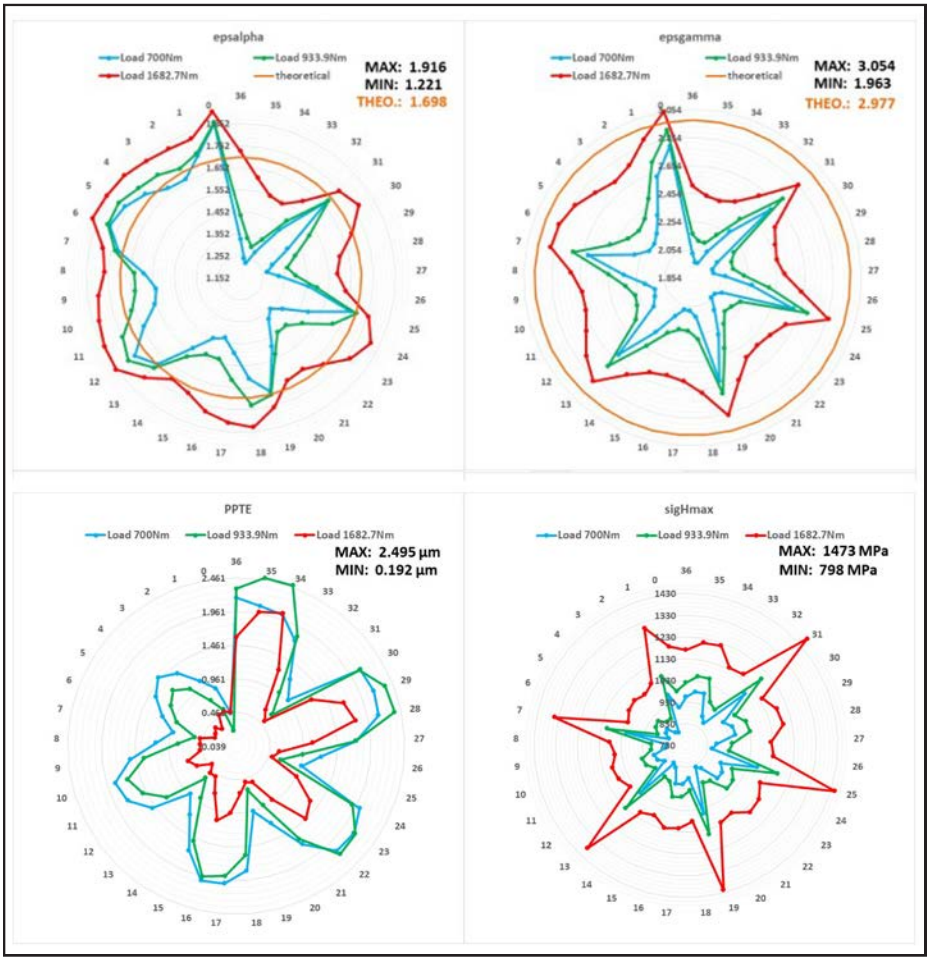


Figure 9 Overview of the main results for 37 profile modification variants at 3 torque levels; profile contact ratio under load ε_α, total contact ratio ε_γ, PPTE and maximum Hertzian pressure.

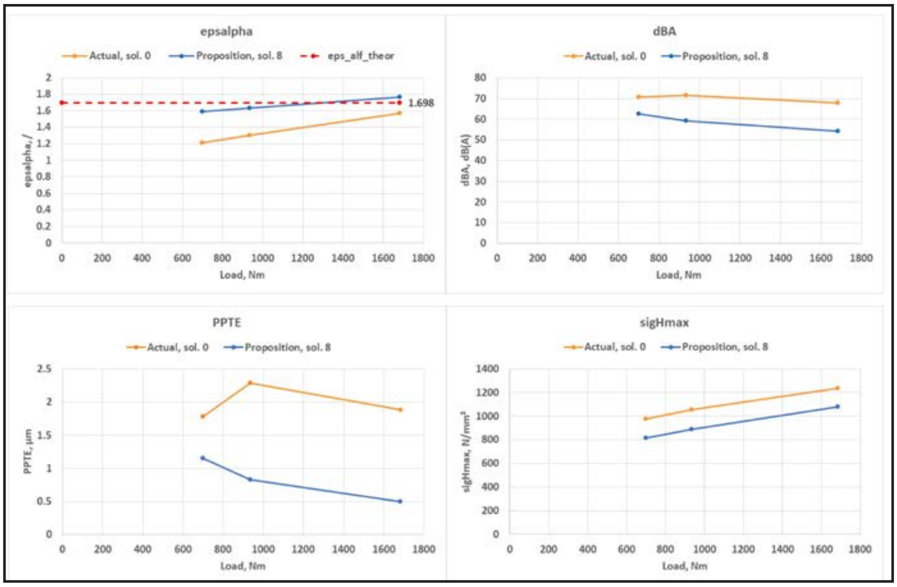
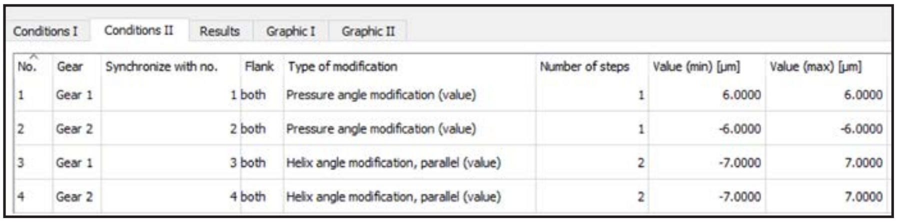


Figure 10 Difference between the gear pair with the actual modifications and the new solution.



No.	Gear	Synchronize with no.	Flank	Type of modification	Number of steps	Value (min) [µm]	Value (max) [µm]
1	Gear 1	1	both	Pressure angle modification (value)	1	6.0000	6.0000
2	Gear 2	2	both	Pressure angle modification (value)	1	-6.0000	-6.0000
3	Gear 1	3	both	Helix angle modification, parallel (value)	2	-7.0000	7.0000
4	Gear 2	4	both	Helix angle modification, parallel (value)	2	-7.0000	7.0000

Figure 11 Inputs for the simulation of manufacturing errors to check the 'stability' of a proposed solution.

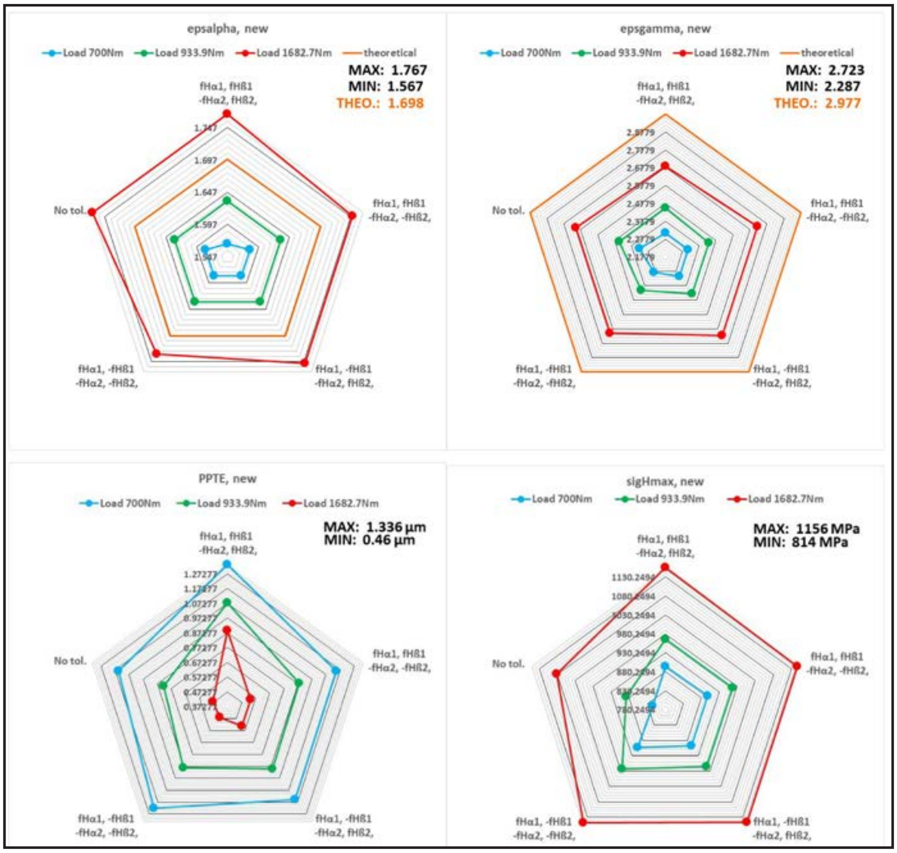


Figure 12 Simulation of manufacturing errors to check the 'stability' of a proposed solution. 'No tol' on the right side is the theoretical solution without any error. Displayed is the profile contact ratio under load ε_α, total contact ratio ε_γ, PPTE and maximum Hertzian pressure.

Comparing these 4 solutions, solution 8 can be a "good compromise" for min, mean and max load. The solutions 2 = (8–6) and 14 = (8+6) have a similar PPTE and an ε_α a bit higher (solution 2) or a bit smaller (solution 14). For this gear pair, solution 8 was chosen.

Then the procedure must be repeated for the left flank. The left flank, in the discussed project, is less loaded and seldom used, so no improved modifications were needed. In Figure 10 the main parameters of the actual gear set are compared with the optimized new solution. The difference is significant. The same procedure was then repeated for all gear pairs in the transmission. Meanwhile, the modified gears could be tested with a prototype on the test fixture. The first results are very satisfactory. We are looking forward to getting permission to publish these results in our next paper.

Influence of Manufacturing Tolerances

Grinding of asymmetric gears is challenging (Ref. 4). It is tougher to achieve the same quality that can be achieved for symmetric gears. Therefore, it is very important to consider manufacturing tolerances in the selection process of modifications.

As explained in the previous section, a specific modification should be as 'stable' as possible for torque variation. This is already a challenge, but the influence of manufacturing tolerances makes this more difficult. Tolerances, depending on the required gear quality, may be substantial compared with the amount of the proposed modification. What if we find a good modification with a tip relief of 5 µm, but profile deviation according the quality of ±10 µm?

Clearly, for good success a high gear quality is needed. Nevertheless, a check if the tolerances may cancel any benefit coming from the selected modifications is recommended. So the solution, as found in the previous section, must be checked for stability of the main parameters when profile and lead errors are added.

To consider the manufacturing tolerances, again the "modification sizing" tool can be used. This time the modifications of the previous solution 8 are kept constant, but additionally profile and helix angle modifications are varied to simulate manufacturing errors (Fig. 11).

Figure 12 shows the results. The solution without errors "No tol" is compared to variants with positive and negative errors. The contact ratio under load ε_α has a minimum of 1.57 and a maximum of 1.77. The largest PPTE increases from 1.15 to 1.34 μm. The Hertzian pressure increases from 1082 N/mm^2 to 1,156 N/mm^2. Overall, the increase in the range of 10% of the critical parameters is acceptable. The proposed modifications can be considered appropriate for the manufacturing process.

Conclusion

The layout of modifications, particularly profile modifications, is difficult. It is challenging to find a good solution also with high professional knowhow. Therefore, the parameter variation technique as described in this paper is a useful procedure.

The verification of the effects of a modification must be made by LTCA, which is a time-consuming calculation process. To permit a parameter study, which demands a large number of variants, a reliable but fast method is preferred. The Weber-Banaschek approach was recently adapted for asymmetric gears to allow such efficient analysis.

The procedure to obtain optimized modifications of a gear set is applied on a gear drive with noise issues. The procedure is discussed step by step on one of the gear sets. The results are very satisfactory compared to the original design. Low-, mean- and high-torque PPTE was reduced by 40% or more, contact shock was eliminated and the $db(A)$ value was reduced by nearly 10 $dB(A)$. The Hertzian stress was also consistently reduced. Finally, the proposed solution with manufacturing tolerances is verified to ensure that the selected modifications are appropriate for the manufacturing process.

For more information.
Questions or comments regarding this paper? Contact Ulrich Kissling at *ulrich.kissling@kisssoft.ag.*

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Dr. Ulrich Kissling studied machine engineering (1976–1980) at the Swiss Technical University (ETH), where he also completed his doctoral thesis — "Pneumatic Weft Insertion on Weaving Machines. In 1981 he started his professional career as calculation engineer for a gearbox manufacturing company in Zurich, progressing there to technical manager and ultimately managing director. As a calculation engineer for gearbox design, he began developing software for gear, bearing and shaft layout. In 1985 he branded this software 'KISSsoft' and started to market it, selling its first license in 1986. In 1998 he founded his own company — KISSsoft AG — concentrating on software development and growing staff from three people in 1998 to workers in 2017. Today, aided by the contributions of partner and managing director Dr. Stefan Beermann, KISSsoft is the leading drivetrain design software, used by more than 3,000 companies on all continents. An internationally respected gear expert, Dr. Kissling is chairman of the TK25 committee (gears) of the Swiss Standards Association (SNV) and a voting member for Switzerland in the ISO TC 60 committee. He actively participates in different work groups of ISO for the development of international standards.

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Forst

ESTABLISHES U.S. SUBSIDIARY

Effective January 1, 2020 Forst Technologie GmbH & Co. KG of Solingen, Germany had established their own wholly owned U.S. subsidiary Forst Technology Americas.

Forst is a global technology leader in the design and manufacturing of broaching machines and associated tools, i.e. to produce ring gears in automatic transmissions and disks in jet engines or power generation turbines. All major automobile and disk manufacturers and their Tier-1 suppliers are among the customers Forst has established over the past 100 years.

With the new U.S. organization, Forst has complete management control over their broaching machine and tool business, service, and sharpening activities in what Forst has identified as the most important market for Forst globally.

Ulrich Salwender, who has been the global sales director and vice president at the Forst headquarters in Solingen Germany for 33 years, will retire on August 31, 2020. He will be succeeded by **Jan Eberhard** (*eberhard@forst-online.de*) who has been the global sales director at the Forst headquarter since 2018. Before joining Forst, Eberhard held various senior management positions with global companies. **Peter Hoelzel** (*hoelzel@forst-online.de*) has been appointed general manager of Forst Technology Americas, LLC. He has held various senior management positions with U.S. subsidiaries of global companies.

Simon Brauns (*brauns@forst-online.de*) is in the process of transitioning from the Forst headquarters in Germany to the U.S. where he will be the major liaison for the Forst customers. He has been appointed senior account manager and customer service North America. Previously, he held various positions as application engineer and senior manager business development in global US and German companies. (*www.forst-online.de*)

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Mitutoyo America Corporation is pleased to announce the expansion of its Custom Solutions offerings. Mitutoyo will now offer “end-to-end solutions” for all its customers’ needs and have the ability to customize products and advanced technology for specific applications.

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Customers interested in an evaluation will participate in a five-step process with a Mitutoyo Custom Solutions Engineer to determine how to best integrate the right approach:

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Currently, Mitutoyo has nine Custom Solutions Centers located throughout the United States. (*www.mitutoyo.com*)

Nixon Gear

CELEBRATES 100 YEARS IN BUSINESS

Nixon Gear, a division of Gear Motions Inc. located in Syracuse, NY, is celebrating 100 years in business this year.

The company was founded by George Nixon in 1920 as Nixon Broach and Tool Company with the intention of selling general machining and special tooling. The company drifted into gear manufacturing and by 1924 it had changed its name to Nixon Gear and Machine Company.

GEAR MOTIONS

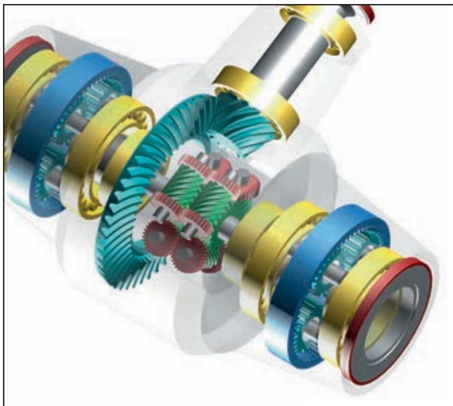
Since then, Nixon Gear has remained successful throughout its century-long history by continuing to grow and adapt through changing times. This has included several building expansions and relocations, shifts in business focus as industry needs evolved, and even changes in ownership.

In 1977, Nixon Gear was purchased by Gear Motions Inc. and now operates as a division of the corporation. In 2010, Gear Motions became a 100% Employee Owned Company, putting ownership in the hands of its employees and assuring that the company would remain viable for many years to come.

In 1992, Nixon Gear moved into its current location on Milton Avenue in Syracuse. The relocation allowed the company to improve its operations and efficiency while also allowing for additional investment in precision gear grinding.

Today, the company continues to invest in the future, adding industry leading robotic automation and super grinding capabilities with the purchase of new gear grinding equipment earlier this year.

Over the past 100 years, Nixon Gear has grown into one of the premier precision gear manufacturing companies in the United States. The employee owners at Nixon Gear and Gear Motions will continue to grow and adapt into the next century as they strive to provide the highest quality products and service in precision manufacturing. (*gearmotions.com*)



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Insights, outlooks, and current news about all companies of the DVS Technology Group can be found centrally on the newly created platform DVS NOW. “With DVS NOW, we would like to show our customers which solutions we can provide them with from the power of the competence network of the Group - so DVS NOW is a little bit like a permanent mini trade fair,” Bernd Rothenberger, member of the board of directors, describes the idea of the platform in an interview on the site. In the future, the site will report on projects and developments on an ongoing basis and increasingly serve as a dialogue platform with customers and industry insiders can be expanded. Learn more at *www.dvs-now.com/en*.

Auf ein Wort,
Bernd Rothenberger
Der Ort, an dem die Zukunft ins Rollen kommt

Platinum Tooling

HIRES SALES MANAGER

Platinum Tooling Technologies, Inc., the exclusive North American importer of Heimatec Live Tools and Angle Heads, Tecnicrafts Swiss Collets and Guide Bushings, Henninger Speeders, and AMF Cleaning and Marking tools, is excited to announce that they have hired a new regional sales manager.

Effective immediately, **Chris Blaine** will oversee all sales activities related to four Platinum Tooling rep groups across the country. This includes Machine Tool Accessory Sales which covers the states of Alabama, Georgia, North Carolina, South Carolina, Tennessee, Virginia, and West Virginia. Additionally, the territory includes Evergreen Tool Group Tool in Michigan, McGill Sales in Ohio and Western Pennsylvania, and lastly, CMI covering the states of Indiana and Kentucky.

Blaine has strong technical knowledge of machine tools and machine tool accessories. Before joining Platinum Tooling, he worked for three years at Morris Group, Inc./Velocity Products headquartered in Windsor, Connecticut. As sales manager, he was responsible for managing their Live Tool sales for many states in the Midwest. In addition to his vast knowledge of Live Tools, Blaine also held the position of project manager for machining center products including rotary tables. He also has experience with new product development and all levels of product and sales management.

Blaine has gained experience through working with complex parts used in the aerospace, medical, defense, diesel fuel systems, automotive, oil and gas, and down hole industries. He has a vast understanding of metals which allows him to make recommendation for the appropriate tooling given a specific application.

Blaine studied Manufacturing Technologies at Ivy Tech Community College in Fort Wayne, Indiana. He looks forward to his work at Platinum Tooling saying, “I am excited to apply and expand my technical expertise and sales ability as a member of Platinum Tooling.”

Platinum Tooling Technologies, Inc., is located in Prospect Heights, Illinois (just outside of Chicago). They recently expanded their facility to provide increased inventory to better serve the marketplace. (*www.platinumtooling.com*)

Chiron Group

ACQUIRES MECATIS SA

On August 1, 2020, the Chiron Group, Tuttlingen, Germany, acquired Mecatis SA with headquarters in Isérables, Switzerland. Mecatis specializes in small, high-precision, high-speed 4- and 5-axis machining centers and complementary automation, used mainly in the watch and jewelry industry.

Mecatis, founded in 2007, designs, builds and maintains machining centers for the high-speed cutting of microtechnical components. The highlight of the product range is the Micro5, launched in 2017, a compact machining center with 4 or 5 axes, which customers use for the rapid, high precision production of components for the Swiss watch and jewelry industry. The Micro5 offers very high static stiffness 2x10^6 N/m, repeatability of 0.5 µm and target accuracy of 2 µm. With excellent thermal stability, very high dynamic rigidity, and acceleration of 2g,the compact machine is made for HSM.

Mecatis machining centers will be sold worldwide by the Chiron Group. The entire service network of the corporate group will also be available to Mecatis customers. On the other hand, the Chiron Group is expanding its technology base with the acquisition.

"The high-precision machining centers from Mecatis perfectly complement the Chiron product range below the 08 series, which handles a work cube of 450-270-360 mm. With them, we are strengthening our offering to the precision and medical technology industries. This means that we can now optimally meet the demands of many customers for compact, highly dynamic machines for extremely small, high precision components," explains Dr. Claus Eppler, managing director R&D at Chiron.

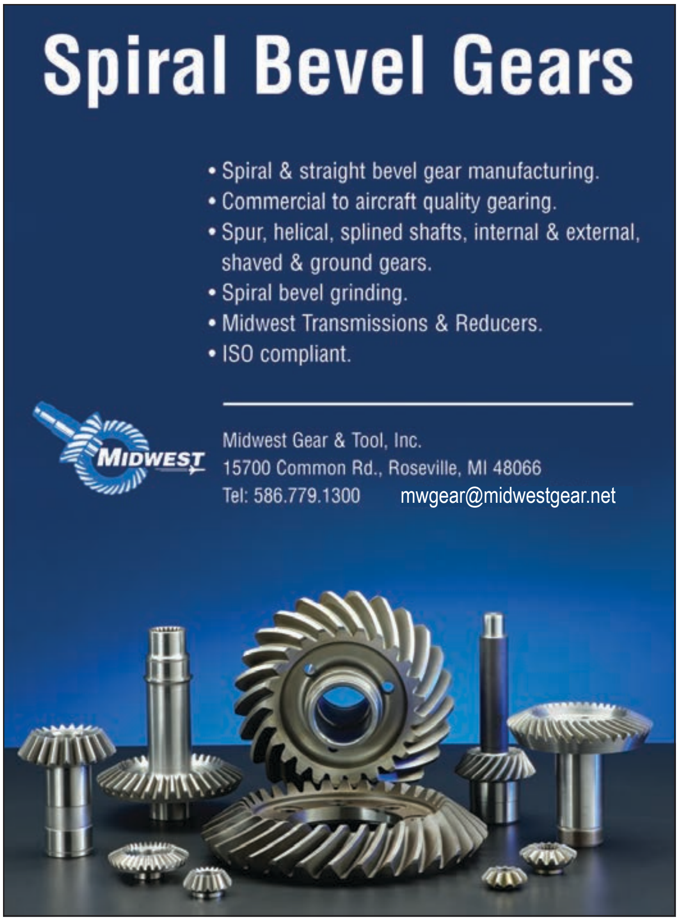
With the acquisition, the Chiron Group is now present with its own companies in ten countries outside of Germany. All Mecatis employees will be transferred to the future Chiron Swiss SA. (*www.chironamerica.com*)

Poggi

ENHANCES CAPABILITIES WITH PHOTOVOLTAIC SYSTEM

A network connected photovoltaic system, with a unit capacity of 394.52 kWp was recently installed at Poggi Trasmissioni Meccaniche S.p.a. The system, connected to the network and installed on the roof of the company, consists of 1,409 modules of 280 W and, according to the appraisal made by SIAT technicians, will have a production capacity of 459,853 kWh/year. This will allow Poggi Trasmissioni Meccaniche S.p.a. to reach a significant energy saving and to consequently devote larger resources for research in the field of transmission parts. This activity has always distinguished the company through the design of leading edge solutions, capable of bringing innovation to the sector. Current examples of this capability are P-drive, a range of low-noise and high-performance synchronous pulleys and belts and the P-gear concept, motion transmission system with non-contact gears, applied in the field of bevel gearboxes and reducers.

"We are very proud of the project," said Andrea Poggi, president of the company. "The investment in photovoltaics reflects a radical commitment to sustainability, a responsibility towards the environment that will continue to characterize our history also in the coming years through choices and strategic decisions aimed at maximizing the use of renewable resources and energy." (*www.poggispa.com/?lang=en*)



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Before the Bookmobile, there was the Bookwheel

Jack McGuinn, Senior Editor

Agostino Ramelli was a 16th-century Italian military engineer of some note who designed many machines and other contributions used in the go-go Renaissance period, including cranes, grain mills, and water pumps. But his most compelling apparatus was a real mindbender — a revolving wooden wheel with angled shelves that allowed users to read multiple books at one time.

Ramelli boasted, "This is a beautiful and ingenious machine, very useful and convenient for anyone who takes pleasure in study, especially those who are indisposed and tormented by gout."

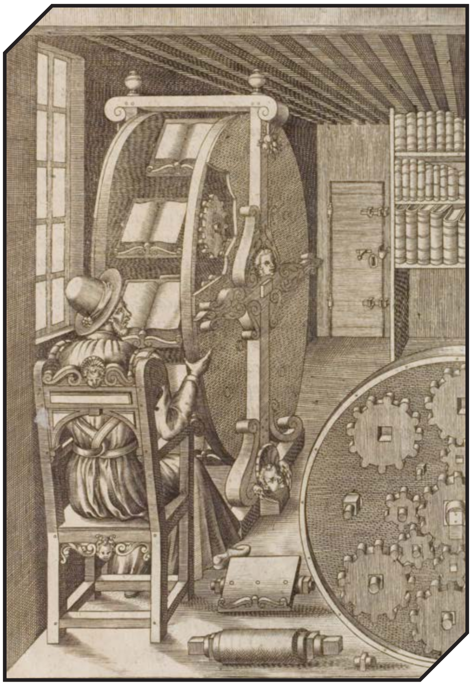
Huh?

Ramelli wrote this for Le Diverse et Artificiose Machine, his "catalog of creations," one might say. And convenient, too! As Ramelli further explained, "Moreover, it has another fine convenience in that it occupies very little space in the place where it is set."

One problem: Ramelli never got around to actually building the thing.

Indeed, the whole eccentric epic is laid out in a 2020 *Facebook* article by Claire Voon. The fact is, Ramelli's bookwheel was eventually built — just not by Ramelli.

And, of primary importance to readers of this page, the bookwheel had gears — lots of gears. In fact according to Ian Kurtz, a Rochester Institute of Technology (RIT) grad and a member of the then undergraduate team that actually brought the bookwheel to fruition, "The actual construction may not have been worth the time with 16th-century techniques. Cutting the gears by hand would have taken a considerate amount of time. I think Agostino was more so showing his understanding of how gear systems worked."



But for bibliophiles, the bookwheel has attained almost legendary status, which is why, in 2018, Kurtz and some fellow RIT students decided they were not only going to build Ramelli's bookwheel — they would build *two*.

They studied closely Ramelli's illustration and built it with wood stock popular in Renaissance times, i.e. — European beech and white oak. Then, utilizing 20th-century power tools, including a CNC machine along with computer modeling — the bookwheel was born.

According to Voon, Ramelli's design probably led to wheels that were built in the 17th and 18th centuries, several of which still exist. But it was probably more complicated than it needed to be, a la Rube Goldberg's comical machines back in the day. "There are simpler objects you could build that would accomplish mostly the same goals," Matt Nygren another RIT team member, explains. "This is more extravagant than it is entirely practical." A more efficient bookwheel, he adds, would be one structured like a Ferris wheel, with hanging, weighted cradles rather than shelves that move along a gear system.

Less complex bookwheels did in fact precede Ramelli's rotating system. Readers in the late Medieval Period could sit by a book carousel, which rotated open books along a horizontal plane, and didn't require side supports. Steven Galbraith, curator of the Cary Collection, suspects that the Italian engineer was trying to improve this design and cater to an increasing need to cross-reference books, which were often large and heavy. "Through the 16th century, books are beginning to talk to each other a lot more — one might reference another — so a bookwheel could have been convenient," he says. "Some scholars say it's the beginning of the idea of hypertext, the idea that a reader can sit in one spot and have access to multiple texts at once."

The Cary Collection's device is used for individual reading research, but it might have even more value as a teaching aid. At RIT, Juilee Decker, an associate professor of museum studies, has had her classes design visitor experiences around the bookwheel. Indeed, museums have displayed interest in the wheel, e.g. — in Russia, the Museum of Languages of the World built its own version using the RIT team's plans, which are published online; and the University of the Pacific in California is also considering acquiring one.

But in the final analysis, Kurtz and Nygren agree that the apparatus, while historically significant, is more engineering oddity than useful machinery. "I don't think it's something you should buy and try and keep in your living room — nowadays there are better tools for the job," Nygren says. "But it's certainly an eye-catching thing, and one of the fanciest ways I can think of for storing books."

And there later came to be something much more practical and efficient — the library table.

A final nugget gleaned from Voon's article that bears repeating. Do you know what the Japanese word tsundoku means? It's a term describing the habit of acquiring books without reading them.

(Source: *www.atlasobscura.com/articles/behold-the-renaissance-bookwheel*.)

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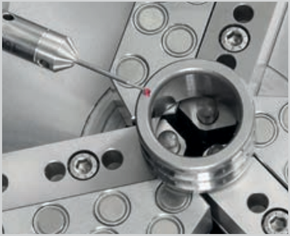
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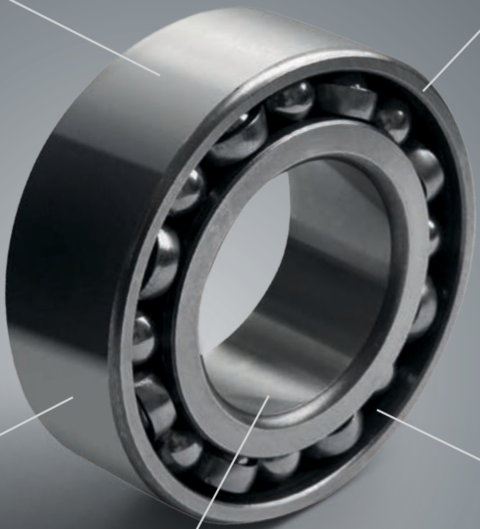

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