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JUN/JUL 2013

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THE JOURNAL OF GEAR MANUFACTURING


GEAR EXPO
The Worldwide Gear Industry Event
2013
September 17-19, 2013
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gear TECHNOLOGY

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departments

Cover Photo by David Ropinski

THE GEAR INDUSTRY'S INFORMATION SOURCE

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GT VIDEOS

The *GT* website currently features a video on Hexagon Metrology's latest product offerings in portable metrology, software, automated inspection, metrology assisted assembly and multi-sensor inspection. Several new products were featured for the first time at HxGN Live from June 3-6 in Las Vegas, Nevada.

Back to Basics

Looking for good articles on gear fundamentals? Look no further than the *Gear Technology* Articles Archive. Just go to the home page and type "basics" in the search box.

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basics

at **www.geartechnology.com**

E-Newsletter

Upcoming E-News topics for *Gear Technology* include the following:

August — EMO pre-show coverage

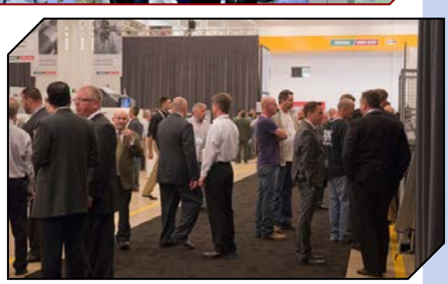
September—Gear Expo

October—Heat Treating

Contact *wrs@geartechnology.com* with editorial ideas.

LinkedIn

September is right around the corner. Start planning your Gear Expo visit in Indianapolis on our LinkedIn page with links to the Educational Programs as well as a brief Gear Expo Schedule. The August and September issues of Gear Technology will feature the most comprehensive and in-depth coverage for AGMA's biennial gear event.



Innovation Days Recap

Check out our Facebook page for a photo album from DMG/Mori Seiki's highly successful Innovation Days event that took place in May 2013 in Hoffman Estates, Illinois.

(*https://www.facebook.com/pages/Gear-Technology-and-Power-Transmission-Engineering/109042075794176*).

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Do you have a question about gear design, manufacturing, heat treating, inspection or assembly? Submit your questions to our panel of experts at: *www.geartechnology.com/asktheexpert.php*

gear TECHNOLOGY

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BPA WORLDWIDE

4TH OF JULY
FOREST CITY GEAR

Boom or Bust— Are You in the Right Markets?

Publisher & Editor-in-Chief
Michael Goldstein

Over the past few months we've talked with a lot of gear manufacturers. Many of them tell us business is strong, while others are struggling with reduced demand. The difference between them isn't so much in the quality of their manufacturing operations, but rather trends in the end markets they serve.

For example, those serving the aerospace and automotive industries seem to be doing quite well. With the major automakers feeling the pressures of government-mandated higher fuel efficiency standards, work continues on building transmissions with more and more speeds. Land Rover introduced the first 9-speed transmission in a passenger car at the Geneva Motor Show in March. GM and Ford recently announced a joint venture to develop both 9- and 10-speed transmissions for cars and light trucks. So there appears to be plenty of gear-related work ahead in the automotive industry.

Despite decreased military spending in both the United States and Europe, the aerospace industry also seems like a bright spot, particularly with regard to commercial aircraft, which saw record levels of production in 2012. With order books full at both Boeing and Airbus, gear manufacturers serving this industry can expect continued business in 2013 and 2014.

But not all industries are such success stories. The diverse but related companies that comprise the energy industry represent more of a mixed bag to gear manufacturers. Because demand for different types of energy changes on an almost monthly basis, serving those industries as a gear manufacturer can be extremely challenging.

For example, in the March/April editorial, I mentioned that the natural gas industry seemed to be in a holding pattern, due to a lack of infrastructure—namely, pipelines—that needed to be built in order to take advantage of the huge energy reserves captured through fracking and other methods. At the time, this was causing a delay in orders for gears used in the heavy-duty compressors used by this industry. Lately, however, I've been reading that the industry has begun relying heavily on rail transport to deliver at least some of the product, and that fracking production is picking up again. Hopefully this will translate into greater demand for geared compressors as well.

Unfortunately, the energy markets don't represent a long-term win-win situation for all manufacturers. Increases in one resource (e.g. natural gas) may reduce dependence on another resource, such as coal. This will have a decidedly negative impact on the gear manufacturers whose parts go into coal pulverizing gearboxes and mining equipment.

And then there's wind energy, which is an almost entirely now-you-see-it, now-you-don't business, dependent on extremely long development cycles and huge investments, coupled with the uncertainty of fickle government policy. Although 2012 was a record year for wind energy installations in the U.S., 2013 so far has not kept pace with those levels. At the recent American Wind Energy Conference & Exhibition in Chicago, major gear industry were focused more on service and rebuild than on building new gearboxes.

What all of this means to me is that it pays for gear manufacturers to be diverse in their capabilities and nimble in their operations. Serving more than one market offsets the risk of a downturn or stagnation in just one. You probably learned from your mother not to put all your eggs in one basket, but the old adage holds true. Building your shop's capabilities is a lot like building an investment portfolio. Diversifying minimizes risk. On the other hand, building the capabilities that allow you to compete in multiple industries takes more than just deciding to do so. It may require additional capital equipment, training, sales effort and industry networking.

Speaking of which, I feel compelled to remind you that Gear Expo is coming up (Sept. 17-19), and if you're inclined to look for ways to diversify your gear manufacturing portfolio, there may be no better time or place. In addition to the tremendous networking and educational opportunities available, Gear Expo offers you the chance to see the latest gear manufacturing technology all in one place.

We'll be there (booth 1123), and we hope to see you, too.

Gleason

EXPANDS GMS SERIES INSPECTION CAPABILITIES

Dennis Traynor, Sales Manager, Gleason Metrology Systems

The world's leading gear manufacturers and their top suppliers are demanding greater system capability and utilization from their inspection systems. Just measuring gear geometry is no longer sufficient, now that accuracy requirements are at sub-micron levels, and other inspection criteria such as surface finish measurement and grind burn detection are becoming increasingly commonplace.

For Gleason, the solution is to simply expand the capabilities of the GMS Series of Analytical Gear Inspection Systems. With the GMS Series, Gleason already provides what high accuracy measurement systems require most: a stable, robust platform, able to position and control movement in both linear and rotational axes. As a result, new functions such as surface finish measurement, form measurement, grind burn detection and general prismatic (CMM) measurements are a natural extension of the GMS' inspection capabilities.

The reality is that our ability to control motion, combined with our robust system for measuring the complex geometries of gear tooth forms, offers an elegant solution: a multi-purpose system that eliminates the expense of multiple machines that take up valuable space and add redundant fixturing, programming calibration and maintenance cost.

Integration of Multi-Sensory Capabilities on the GMS.

When a gear (or any rotationally symmetrical workpiece) is set up for measurement on a Gleason GMS inspection machine, it is convenient to test for other quality characteristics called out on most part prints today, such as surface roughness. Though our machines had this feature in the late 1990s, it was limited. Digital probing systems were still in their infancy, not as versatile and reliable as they are today, and while we sold quite a few systems, the integration was not as seamless as most had hoped.

Fast forward to today's motion control technology, systems integration techniques and capabilities, and market demand, and the picture of supporting the manufacturer's needs becomes much clearer. Today's systems utilize devices kinematically coupled to our motion control system, which allows the stylus to automatically:

- Rotate to the helix angle of the gear
- Move normal to the surface of the workpiece
- Deflect the probe at a constant force
- Gather data
- Determine roughness to whatever parameter required

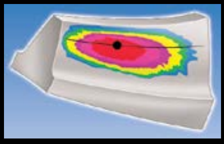
All while not requiring manual intervention with the measurement or manipulation of the part. No special fixturing or material handling, outside of loading the part, is needed for comprehensive measurement. By the same token, because we understand it is not just surface finish, but also waviness that can affect gear behaviors, we have added the ability to analyze waviness parameters to the GMS series as well. All analysis is shown after measurement on screen, and is printed in additional pages in the inspection report.

In adding these capabilities to our machines, we realized that careful consideration must be given to a number of factors. Actual capacity of the gear metrology machine must be taken into account, as well as the design and integration of accurate sensors for surface roughness measurement and grind burn detection (Barkhausen noise analysis), in order to properly accommodate these added applications. In addition, a thorough review of parts requiring this testing must be conducted and factored into the design phase.

In 2011, Gleason Metrology Systems partnered with American Stress Technologies, a global supplier for Barkhausen noise analysis (BNA). Together, we devised a way to mount the Stresstech RollScan products on our machines and apply our motion control to follow the lead profile at various radial positions in order to create a

The F35 represents advanced fifth generation aerospace technology. And pilots flying these systems rely on exacting quality from every component - especially the gears which are produced by Arrow Gear Company.

With over 65 years experience, Arrow Gear's products have been used in the most advanced aerospace systems produced by our civilization. With a state-of-the-art production facility and dedicated personnel who are among the best in the business, Arrow Gear offers the expertise and precision for the most demanding quality requirements.



Arrow Gear also offers leading edge design and development capabilities ensuring new designs meet performance specifications without costly delays.

When you need quality, expertise, and precision, you can rely on Arrow Gear!

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full flank analysis for residual and compressive stresses induced by hard finishing processes. This type of testing is non-destructive, so that when the part tests as "accepted," it can be used as a production workpiece, as compared to the more traditional nital etching process, which often renders the part unusable.

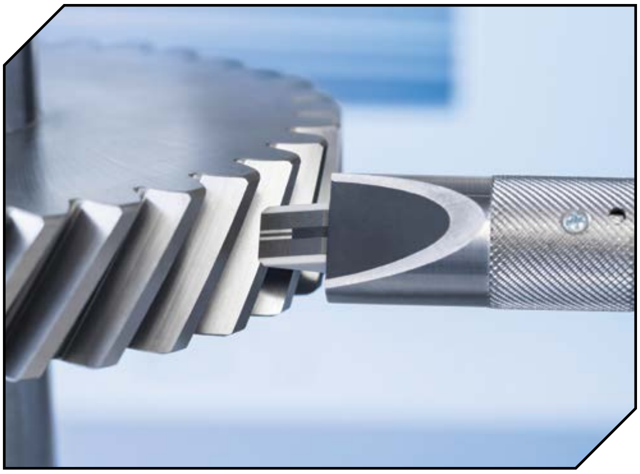
In large-gear production environments where the lot size is typically small (1–5 pieces), and materials and process times are expensive, cost savings are swiftly realized. Having this type of materials testing technology built into your gear inspection system really pays off when the parts are large and material handling, fixturing, and warranty costs all add up quickly. BNA also has the significant advantage that it is a great predictor of sub-surface defects in the microstructure of the material. It also is a valuable tool for assessing hardness, especially when compared to the older, destructive testing.

Powerful and User-Friendly.

GMS has devised a charting technique that is easy for the average operator to interpret. The illustrations below display typical Rollscan measurement results. If all the trace outputs are uniform, as in the first graph, it's quite easy to see that tooth form is consistent and the material is uniform in respect to residual and compressive stresses. In contrast, the second graph shows measurement of a tooth form where certain traces are "spiked" in their signature, and some surface anomalies are present.

But on a closer look, you ask yourself, "Where?"

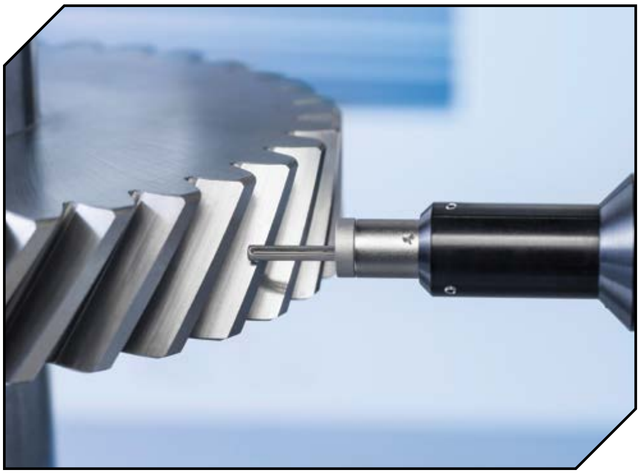
With GMS' new software, we have extrapolated those output signals to create an easily understandable graph that shows, at a glance, where the material defects occur. The rectangle represents the tooth surface area. The graph's horizontal or X axis shows the measureable tooth surface in the lead direction of the part. (Not all of the tooth surface can be measured, as the sensor must stay fully on the tooth surface.) The sensor, in this case, covers a surface area equivalent to the small blue square shown on the right. The vertical or Y axis depicts the radial depth or number of scans that were taken along the lead direction. (Again, this measurement does not go into the root or to the very edge of the tooth tip.)



As with all metrology systems, certain limitations may exist and consulting with our applications department is the best way to determine which machine and software combination will best fit your particular applications. An important consideration is the fact that GMS isn't just for gears: all rotationally symmetrical workpieces are also candidates for these technologies.

CAD-Based Inspection is Now a Reality

A third addition to GMS inspection machine functionalities is a CAD-based inspection package. Geometric analysis is no longer confined to the tooth form, though admittedly this is still quite complex, given the number of profile and lead modifications used today to make gear sets perform to design intentions. More and more gear sets are designed in CAD, and as part of assemblies that require measurement at various stages of assembly.



Recognizing this, Gleason now offers CAD-based inspection to complement our other inspection packages. We've integrated the capability to measure non-gear features into our measurement platform. Programming in conjunction with popular CAD models (e.g., *Step*, *IGES*, *Catia V4*, (.mod .model, .exp, .dlv, .cat CatiaV5) *CATPart*, *CATProduct*, *Parasolid*, *ProE*, *Unigraphics*, *VDA*, *SoldWorks*, *STL*, *Cadds Fils VRML*) are supported.

This opens the doors to exciting new opportunities for GMS to support manufacturers of non-gear parts who require high-accuracy inspection on any rotationally symmetrical workpiece where the rotation of the part is integral to the inspection and not just handled as an indexing axis to present the part to the measurement probe.

The modular structure of our CAD inspection system allows users to purchase basic and advanced versions, with upgrade costs well below industry standards. Users can easily enhance system capabilities as they learn, or as their needs change.

Gleason is planning the formal release of this new feature at EMO and Gear Expo in September, 2013. Both exhibitions will have live demonstrations of all the aforementioned inspections on actual workpieces.

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Ingersoll

SELECTS LIEBHERR LC1200 FOR TECH CENTER

Ingersoll Cutting Tools has selected a Liebherr LC1200 gear hobbing machine for installation in its Rockford, IL Technical Center (ICTC). The LC1200 will demonstrate the productivity of Ingersoll's indexable carbide tools for gear machining. Indexable carbide insert (ICI) cutting tools, including Ingersoll's innovative two-start hobs and single-index gashing tools, offer more cutting edges, permit higher cutting speeds, and therefore greater cutting capacity for large module gears. The Liebherr LC1200 has special design features for ICI applications. Its rigid structure, 900 mm table diameter, and 45 kW (60 hp) spindle drive provide a suitable platform for large cutting tools, permitting high metal removal rates. Liebherr's new FK 3.3 hob head can handle tools up to 450 mm diameter, 540 mm length, and gears up to module 27 mm. Liebherr also offers a form-milling head for internal gear applications.

In the Spring of 2012, Ingersoll Cutting Tools completed construction of its new Technology Center on the campus of its Rockford, IL headquarters. The new Tech Center is equipped with state-of-the-art machines from select machine tool partners for turning, milling, hole-making, and now gear machining. The Tech Center will be used primarily for training seminars, testing, and product demonstrations.

"A key goal of the new Technical Center is to create a unique environment where customers can not only see the latest tooling and machining technology, but also participate in discussions with Ingersoll engineers and our machine tool partners to find solutions for their specific manufacturing needs," said Chuck Elder, ICTC president.

Peter Wiedemann, president of Liebherr Gear Technology, Inc., adds; "We have a history of providing our customers with turnkey solutions in gear production and inspection. Having the LC1200 in Rockford allows Ingersoll to demonstrate and highlight the most productive cutting tool technology available in the gear industry, and gives Liebherr the opportunity to demonstrate the machine's capabilities to our North American customer base."

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Wenzel GearTec

WGT 280 EXTENDS SERIES

Wenzel GearTec presented the new WGT 280, the first model of a new generation of gear measuring machines in a new industrial design at Control 2013. This extends the approved WGT series by a measuring machine especially for small gears and rotationally symmetrical parts.

With the new WGT 280 Wenzel offers a gear measuring machine for the fast and efficient analysis of small gears up to a maximum diameter of 280 mm. Because of its compact construction and small footprint it can be easily integrated into existing processes. The easy to access measuring volume allows a simple loading and operation of the measuring system. Therefore the WGT 280 is ideal for the use of automated loading systems.

The high precise rotary table can be loaded with parts up to 50 kg weight. The WGT 280 is equipped with the Renishaw scanning probe SP 600 and allows the measurement of gears starting with a module of 0.5 as standard. For the accurate measurement of shafts, the gear measuring machine can be equipped additionally with a tail stock. The WGT 280 allows measurements in a Z range of 500 mm.

"The developments in drive technology make precise and reliable measurements of gear components and especially gears mandatory. Drives of any kind have to be as efficient as possible. In future gears will still be of great importance," explains Heinrich Brüderle, responsible for the product series gear measurement at Wenzel Group. With the new gear measuring machine Wenzel combines innovative technology and modern industrial design.

During the development of the WGT 280, Wenzel focused on one of its core competencies. The base plate and linear guides are made of massive hard stone and finished by hand. The combination of granite technology and precise air suspension makes a gear measuring machine a durable and highly accurate measuring system according to VDI/VDE 2612/13 group 1. Furthermore the specially developed controller for gear metrology provides optimal 4-axes-control and high measuring performance. The WGT gear measuring machines are equipped with a modular software package. The user is led intuitively through different input masks to enter parameters for part, measurement and analysis. Additionally the user is supported by different graphics. The basic package *TGear* can be individually extended by different modules for the measurement and analysis of different types of gears and shafts.

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Sulzer

COATING TECHNOLOGY FOR COMPLEX SURFACES

Sulzer offers a variety of coating equipment, material and services for the most complex surface applications. The company's custom solutions based on PVD and diamond coating technologies reduce production costs, increase tool cutting rates, reduce use of lubricants, prolong tool life and improve wear resistance. Significant for gear cutting is the company's new M.Power coating technology. "M.Power is a micro-alloyed TiSiXN-based coating that offers smooth surfaces; high hardness; high oxidation resistance; high wear resistance; low coefficient of friction; high tool quality; prevention of cold welding; and formation of built up edges," says Annette Norin at Sulzer Metaplas GmbH.

The key strengths of the coating are its dramatic reduction of the sticking material on the cutting edge due to reduction of the surface roughness. It also boasts extremely high heat resistance. According to Norin, Sulzer provides custom solutions through the combination of pre and post treatment of the tool surface coating material, layer architecture and system/equipment technology with which the coating will be applied. "We focus on advanced plasma-assisted/arc evaporator technology for innovative coatings," Norin says. "APA is based on the cathodic vacuum arc and offers diverse development possibilities for new layer architectures in terms of morphology, stoichiometry, doping, multiple layers and nano layers."

The benefits of APA include excellent coating adhesion and smooth coatings through the reduction of macro-particles. APA is also the basis for new hybrid technology. "Customers are willing to test new coatings to determine the benefits and also to come up with specific problems and we develop solutions together," Norin adds. "We develop coatings on our own R&D equipment and test new coating/layer designs together with our customers."

Norin believes the hybrids mentioned earlier will play a significant role in coating technology in the future. "This technology will include hybrids based on HI3 technology, a combination of AEGD (arc-enhanced glow discharge), a plasma etching process for layer adhesion; HIPAC (high-ionization plasma assisted coating), a highly ionized sputter process; and APA Arc (advanced plasma-assisted arc), a highly ionized arc process."

For more information:
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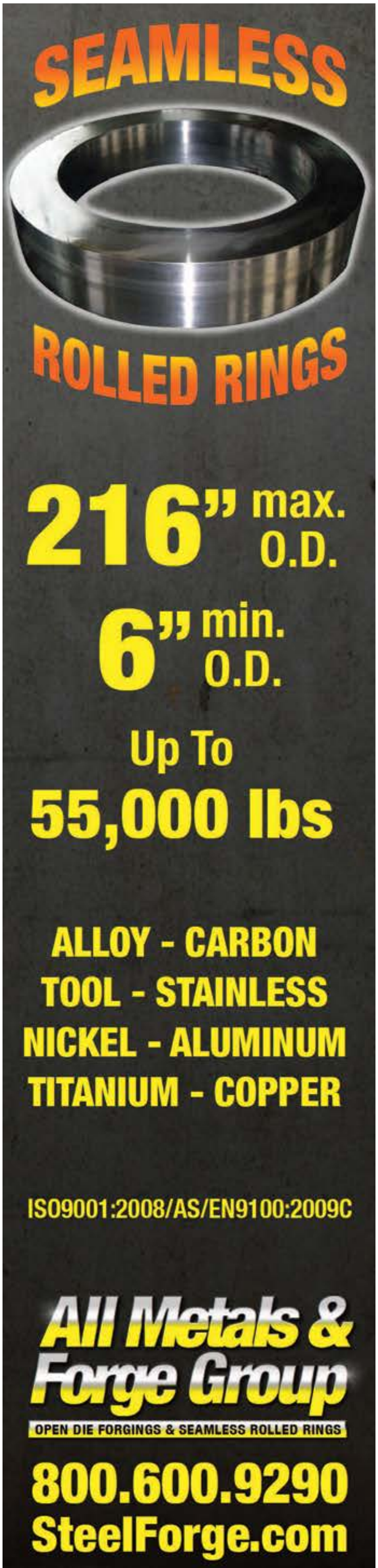

Hexagon Metrology

LAUNCHES ONLINE TECH SUPPORT

Hexagon Metrology recently launched the new Hexagon Metrology Support Center in conjunction with World Metrology Day. The online portal is a centralized resource for support of any Hexagon Metrology product. Customers will have 24/7 access to an FAQ (frequently asked questions) knowledgebase, and receive lifetime diagnostic phone support of any Hexagon metrology hardware. Clients with active software maintenance agreements (SMAs) can utilize the helpdesk for technical support inquiries and software downloads. The Hexagon Metrology Support Center establishes a one-stop location for all technical support needs, and provides three ways for customers to contact technical support staff — the online portal, email or via phone. All three methods provide closed-loop support, as all inquiries are automatically logged into the dedicated technical support call center.

"The Support Center's innovative blend of self-service tools and professional advice delivers a customer service unmatched in scope and functionality," states Al Chiasson, general manager of Hexagon Metrology Services. "This new resource is unique to the metrology industry, and reflects our ongoing commitment to enhance the overall Hexagon customer experience. The Support Center provides convenient, self-directed solutions designed to achieve best-in-class response times to technical support inquiries."

To use the system, Hexagon Metrology North America customers create a free account. Video tutorials and multimedia answers to common questions are posted throughout the site to enhance the user experience. Within the online portal, users can review support tickets and their history at any stage, even after a ticket is closed. Additionally, customers receive real-time e-mails when the status of their ticket has changed. Users also can view a listing of their Hexagon product inventory at any time.

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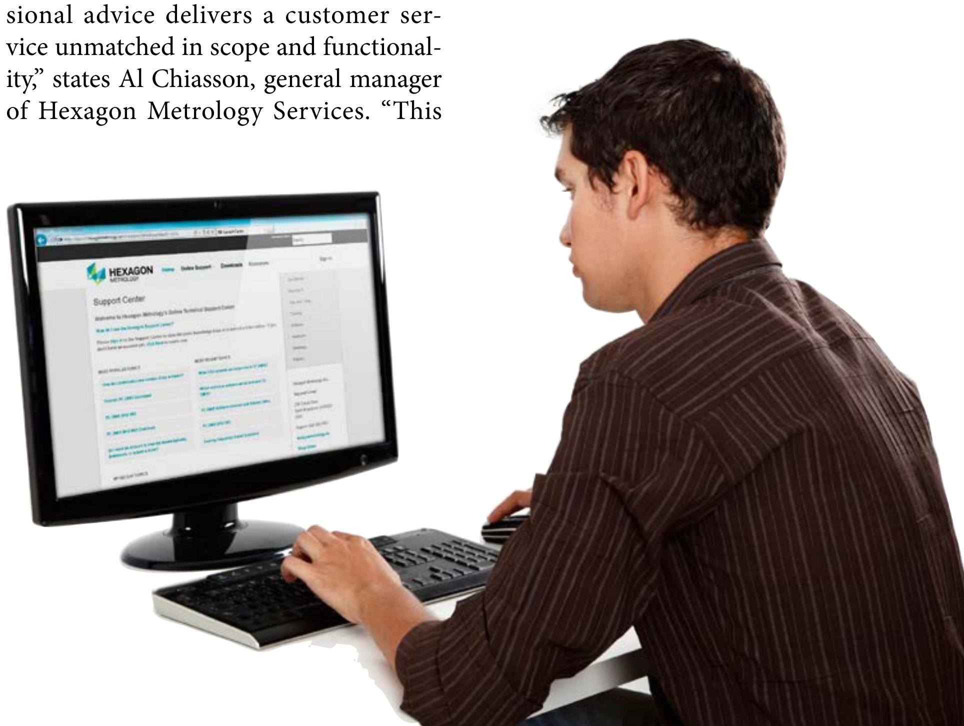

Leistritz

KEYSEATING TECHNOLOGY IMPROVES

Keyseating is a fundamental and traditional method to cut keyways in bores that has been stagnant and without development for many decades. Yet the quality and sophistication of the components with keyways have greatly improved through smaller tolerances, more exotic materials and greater load carrying abilities. Keyways in

high dollar value hubs or gears which have been processed to near completion using CNC turning and possibly grinding operations are often trusted with an antique keyseater for one of the final steps of production. Or worse yet, manually broached by hand in a press using shims to control the final depth.

Leistritz, a German machine tool builder, has a line of keyseaters to match the control and flexibility needed by today's manufacturers to produce a straight and accurate keyway. Their CNC keyseaters are very well suited for cutting small keys yet heavy duty enough for huge keyways, up to 7" wide, as well. The CNC control is a valuable time saving device and can optimize the cutting parameters through fully flexible programming of feed rates and cutting speeds, as well as variable depths, if required. The CNC also allows the addition of a rotary table to produce helical grooves for special applications or multiple grooves within a single bore.

Leistritz has also developed keyseaters with linear motors to make full use of modern cutting tool materials, including carbide to cut keyways in already hardened workpieces. The linear motor systems produce the high cutting speeds required to match that needed by the tooling and to shorten the cycle times for a production environment. With machine strokes up to 60" these machines can cut a wide variety of grooves in steels previously un-thought of with little investment in both tooling and capital.

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Bryant Grinder

MINIATURE GRINDER BOASTS HIGH RANGE CAPACITY

The new Bryant Miniature precision grinder (Model RU1) with standard 4.0 inch (100 mm) travel in both “X” and “Z” axis, brings a whole new dimension to small precision part processing. This machine is suitable for small parts manufacture, including optical and medical applications as well as bearings, fuel management and automotive requirements. Within the 64 cubic inches (1048 cubic centimeters) work zone, there are unlimited processing opportunities including conventional chucking for ID and OD grinding, between center workholding for multi-surface OD shaft work, roller blade fixture for ultra-small and thin walled parts, shoe fixture for ID and OD, bores, ball tracks and special applications, tri-roll fixture for long thin walled parts, dressing options including: single point, high frequency (cup or disk), radius (ball tracks), formed roll and CNC profile dress, loading options including double/single arm plug, bucket, single plane and robotic, acoustic emissions sensing for gap, dressing and crash detect and vector oscillation grind.

The model designation of this miniature machine is the Bryant RU1, which stands for *Revelations* operating system, “universal” capabilities and “1” for the machine size. This designation makes the newest machine an integral part of the new Bryant Grind Systems family that offers a range of sizes for totally universal, multi-surface machine tools to process a wide range of complex, precision workpieces.

Eugene P. Supernor, vice president of operations, says “There are hundreds of old Bryant Model B’s still working long hours in American and foreign bearing companies. The older Bryant’s have extremely small effective work zone due to the short stroke of both axes. The new Bryant Miniature is ideal for medical, optical, fuel management and automotive applications due its range capacity. And, Bryant’s patented *Revelations* CNC operating system offers customers total flexibility, the simplicity of single screen operation and programming, and a guaranteed reduction in cycle times.”

The Bryant Miniature, as with all the larger Bryant models, features round hydrostatic guide ways with Bryant’s

Hydra-Truc 360° bearing system which controls the slides within all degrees of freedom, Fanuc linear motors and glass scales with 0.00004" (1 μm) accuracy in the work zone. The configurable slides with the versatile Hydra-Truc allow total flexibility in the number and positioning of slides, with thermal compensation for slides and spindles. All Bryant Grind Systems machines are equipped with proven Bryant work heads, wheel heads, dressers and the patented *Revelations* CNC operating system software.

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Kennametal

SHOWS BENEFITS OF MTCONNECT

The recent 2013 MTConnect Connecting Manufacturing Conference was instrumental in further preparing the foundation of a monumental change in how manufacturing firms can obtain, manage, and share their machine tool and tooling data. Cutting tools can be an extremely valuable data source for shops to monitor, collect, and study, says Tom Muller, senior manager, Innovation Ventures Group, at Kennametal Inc. Noting that MTConnect is a protocol and that ISO 13399 is an international standard, Muller demonstrated that MTConnect does not have to go into the standards development business. Since ISO 13399 already defines and standardizes such cutting tool attributes as cutting diameter, edge angle, body diameter, overall length, functional length, and functional width, among many others, MTConnect can simply adopt the ISO 13399 definitions and achieve a consistent language for exchanging data between machine tools, tool data management systems, presetters, and even CAD/CAM systems. "The more we can drive the standardization of cutting tool data, the simpler and more efficient we can make a shop's business," Muller said. Imagine not only staying on top of the volumes of data a shop produces, but having machine tools "know" what the presetter and the cutters "know" through the efficient exchange of data. Hours of setup, touch-offs, and test cutting could become a thing of the disorganized past. Tribal knowledge or "how Joe does it" can make way for organized and systematized ways of manufacturing parts. MTConnect is in place, and the applications from the machine tool, tooling technology, and software leaders are coming.

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JOB SHOP LEAN

Ed's Note: This is the fourth article in an eight-part "reality" series on implementing Continuous Improvement at Hoerbiger Corporation. Throughout 2013, Dr. Shahrukh Irani will report on his progress applying the job shop lean strategies he developed during his time at The Ohio State University. These lean methods focus on high-mix, low-volume, small-to-medium enterprises and can easily be applied to most gear manufacturing operations.

Dr. Shahrukh Irani, Director IE Research, Hoerbiger Corporation of America

Design of a Flexible and Lean Machining Cell (Part 1.)

Background

Group technology (GT) has been practiced around the world for many years since the 1960s as part of sound engineering practice and scientific management. S.P. Mitrofanov (1966) defined GT as "a method of manufacturing piece parts by the classification of these parts into groups and subsequently applying to each group similar technological operations." Cellular manufacturing (CM) is an application of GT to factory reconfiguration and shop floor layout design. I. Ham (1985) defined:

- a *part family* as "a collection of parts which are similar either because of geometric shape and size or because similar processing steps are required to manufacture them."
- a *manufacturing cell* as "an independent group of functionally dissimilar machines, located together on the shop floor, dedicated to the manufacture of a family of similar parts."

Although a cell is dedicated to produce a single part family, it must have the requisite equipment capabilities, routing flexibility, cross-trained employees and, to the extent possible, minimal external process dependencies. Cells are often implemented in job shops since they provide the operational benefits of flowline production.

Cells: The Foundation for Job Shop Lean

Starting in 1959, Serck Audco Valves, a U.K. manufacturer of industrial stop valves and actuators, began to implement GT and CM as a foundation for reorganizing their complete manufacturing enterprise, *even though they started with their machine shops*. In his book *Group Technology: A Foundation for Better Total Company Operation*, G.M. Ranson wrote "As a practitioner with some twelve years (of) experience of this technique (group technology), the definition which I think most clearly describes it is as follows: *The logical arrangement and sequence of all facets of company operation in order to bring the benefits of mass production to high variety, mixed quantity production.*"

Following in the footsteps of Serck Audco Valves and many other similar HMLV (high-mix low-volume) manufacturers, the starting point for implementing Job Shop Lean in any high-mix low-volume manufacturing facility is to convert their existing facility layout, usually a functional layout, into a cellular layout. The functional layout has advantages such as high machine utilization and high flexibility in allocating parts to alternative machines in any department (aka process village or workcenter) whenever any batch of parts arrives for processing. However, it has disadvantages such as high stock-to-dock order flow times, high WIP levels, poor quality control and difficulty in locating orders. In direct contrast, the cellular layout has advantages such as short stock-to-dock order flow times, lower WIP levels and effective quality control.

Table 1 Routings and Production Data for a Sample of Parts

			Routing (Sequence of Operations)						
Part No.	**Qty**	**Revenue**	**Op 1**	**Op 2**	**Op 3**	**Op 4**	**Op 5**	**Op 6**	**Op 7**
Part 1	10642	31336	1	4	8	9			
Part 2	4270	21300	1	4	7	4	8	7	
Part 3	1471	10901	1	2	4	7	8	9	
Part 4	4364	25774	1	4	7	9			
Part 5	5013	1580	1	6	10	7	9		
Part 6	4679	36069	6	10	7	8	9		
Part 7	5448	47776	6	4	8	9			
Part 8	5339	50339	3	5	2	6	4	8	9
Part 9	9117	48784	3	5	6	4	8	9	
Part 10	8935	37774	4	7	4	8			
Part 11	7100	68153	6						
Part 12	8611	60272	11	7	12				
Part 13	9933	39903	11	12					
Part 14	3824	19258	11	7	10				
Part 15	1359	7800	1	7	11	10	11	12	
Part 16	1235	8562	1	7	11	10	11	12	
Part 17	8581	44074	11	7	12				
Part 18	3963	23137	6	7	10				
Part 19	2309	3012	12						

Table 2	Information about Machines in the Different Departments			
Machine No.	**Area Requirements**	**No. available**	**Purchase Price**	**Mobility**
1	2000	2	N/A	N/A
2	1000	1	N/A	N/A
3	1000	1	N/A	N/A
4	2000	2	N/A	N/A
5	1000	1	N/A	N/A
6	2000	2	N/A	N/A
7	4000	4	N/A	N/A
8	1000	1	N/A	N/A
9	2000	2	N/A	N/A
10	4000	4	N/A	N/A
11	3000	3	N/A	N/A
12	1000	1	N/A	N/A

In this column, we will illustrate how to form the part families and machine groups that will constitute the cells which are the foundation for Job Shop Lean. Table 1 contains the P-Q-R-$ information for each of 19 parts that are produced in a hypothetical machine shop that consists of 12 machines:

1. Product Number and Name
2. Annual Production Quantity
3. Manufacturing Routing
4. Annual $ales

In addition, information about each piece of equipment used in the machine shop is needed (Table 2). The last two attributes of each machine — purchase price and mobility — are important because (1) if additional copies of that piece of equipment need to be purchased and placed in several cells, the capital expense should be affordable and (2) it could be exorbitantly expensive to relocate that piece of equipment. Figure 1 shows the existing functional layout for the shop. Figure 2 shows a cellular layout with three cells that was designed for the same shop. Each cell was designed to produce a subset of the 19 parts listed in Table 1. With reference to the cells shown in Figure 2, the existing machines in the departments 1, 6, 7, 9 and 10 in the Functional Layout have been distributed among the cells in the Cellular Layout. Finally, an up-to-date layout of the existing facility must be available along with Tables 1 and 2.

A Comprehensive Approach for Implementing Job Shop Lean

Figure 3 presents a flowchart for a comprehensive approach for implementing lean in job shops. At the core of this iterative process is the expectation that a job shop (i) will identify the stable part families in its product mix, and (ii) will implement a FLEAN cell to produce each part family. What is a FLEAN (flexible and lean) cell in a job shop? It is essentially a mini-job shop which is (a) ***flexible*** because it is designed to produce all parts in a part family, and (b) ***lean*** because its design has incorporated all the lean tools that are essential for job shops to use. In theory, each iteration of the design process shown in Figure 3 will result in the implementation of a stand-alone cell that is dedicated to producing a part family. In reality, numerous constraints will arise that could prevent implementation of any cell. Some constraints could be broken (Example: Operators could be cross-trained to operate multiple machines in a cell). Whereas, some constraints may remain unbreakable (Example:

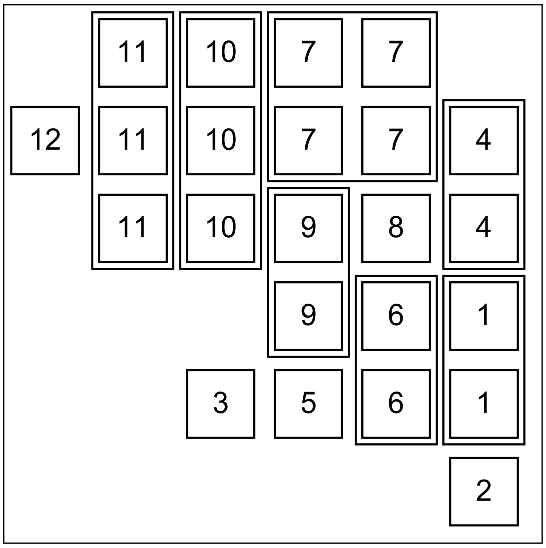

Figure 1 Functional Layout

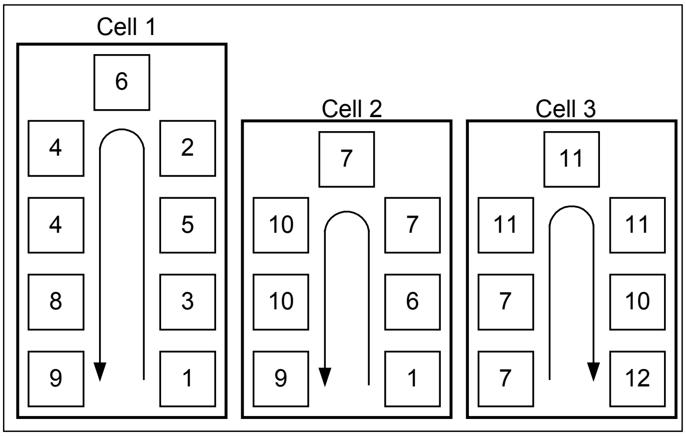

Figure 2 Cellular Layout

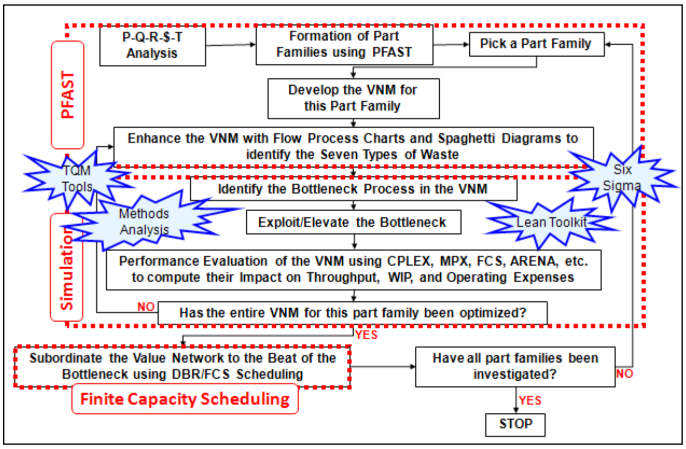

Figure 3 Comprehensive Approach for Implementing Job Shop Lean

Heat treatment furnaces cannot be placed inside a cell next to a CNC grinder).

Upon completion of this process, a job shop would typically end up being divided into at least two sections:

- Section 1: This area of the facility will consist of FLEAN cells with each cell dedicated to a product family.
- Section 2: This area of the facility is a "remainder shop" where the non-production orders (spare parts, prototypes and one-offs for new customers, rush orders) are produced.

By dividing the job shop into these two sections, two benefits are gained: (1) The cells provide unquestionable quick response, high quality, team work and order traceability and (2) A smaller portion of the entire business now needs to be managed as a complex job shop where flexible automation co-exists with firefighting and overtime to fulfill orders.

An Illustrative Example to Explain the Analytics Underlying the Methodology

This section briefly describes how we analyze the data captured in Tables 1 and 2 to implement the approach shown in Figure 3 in a high-mix low-volume facility like HCA-TX. We used the *PFAST* (*Production Flow Analysis and Simplification Toolkit*) software that helps to implement Prof. John L. Burbidge's method of production flow analysis (PFA) for implementing cellular manufacturing in any complex multi-product facility.

From-To Chart: Given the routings of a sample of parts in Table 1 and the Quantity (or Revenue) for each of those parts, the From-To Chart (Table 3) captures the cumulative volume of material flow between every pair of consecutive machines that occurs in one or more of the routings listed in Table 1. Each entry in the chart represents the aggregate material flow "from" the machine listed in any row of the table "to" any machine listed in any column of the table. For example, the total number of parts flowing From Machine #1 to Machine #2 is 1,471 because only the routing of Part #3 contains that pair of consecutive machines (1→2) and Q = 1,471 pieces for this part.

Table 3 From-To Chart

		To											
	Machine	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**
From	**1**		1471		19276		5013	2594					
	2				1471		5339						
	3					14456							
	4							19040	43751				
	5		5339				9117						
	6				19904			3963			9692		
	7				13205				6150	9377	7787	2594	17192
	8							4270		36696			
	9												
	10							9692				2594	
	11							21016			2594		12527
	12												

If you input the From-To Chart in Table 3 to a standard facility layout software like *Storm*, *Plantopt* or *Factoryflow*, it will produce a functional layout for the entire shop like the one shown in Figure 4 (similar to the one shown in Figure 1). The algorithm internal to any of these software tools will place departments with the highest traffic volume adjacent to each other. The same software tools could be used to design the layout of an individual cell or a shadowboard for tools used to assemble a variety of products.

Product-Process Matrix Analysis**:** This is the method that is widely touted in the lean literature for the formation of product families and manufacturing cells. However, history will show that it was first utilized by Burbidge for identification of part families and machine groups in the fabrication shop of a crane manufacturing facility. The Initial 0-1 matrix (Table 4) converts the original part routings in Table 1 into a matrix with part numbers in the rows and machine numbers in the columns. Each cell in the matrix which contains a "1" links a part number to a particular machine number that occurs one or more times in the routing of that part. For example, the routing for Part #1 is 1→4→8→9. Therefore, in Table 4, in the row for Part #1, a "1" occurs in the columns for "m1", "m4", "m8" and "m9."

The Initial 0-1 matrix does not show potential groups of machines and parts that are the basis for implementing FLEAN cells. However, when the same matrix is manipulated by reordering the rows and columns, it produces the final 0-1 matrix in Table 5. This new matrix reveals the potential for implementing two cells provided that some machines can be duplicated in both cells. Each block in the final matrix, which is defined by a set of consecutive rows and consecutive columns, represents a family of parts that could potentially be produced by putting the appropriate group of machines into a manufacturing cell. For example, in Table 5, we can see that two cells can be formed: Cell 1 consists of machines 2, 3, 5, 4, 8, 9, 6, 1, 7 and 10 and Cell 2 consists of machines 6, 1, 7, 10, 11 and 12.

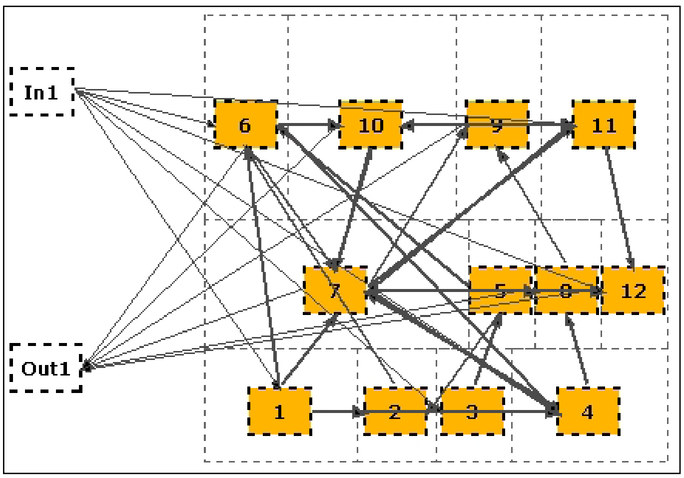

Figure 4 Functional Layout

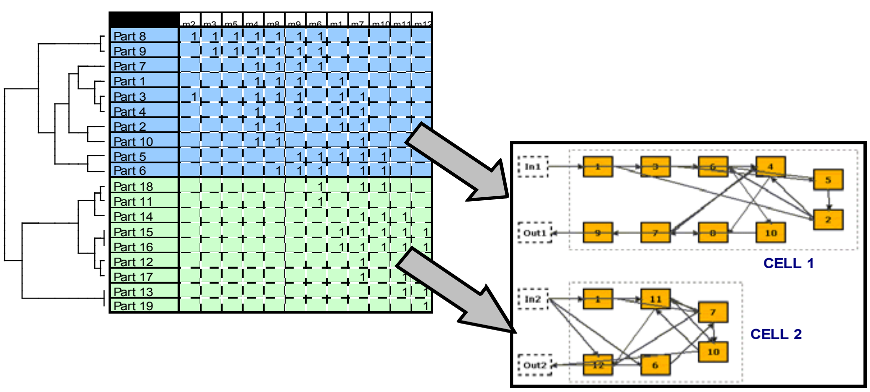

Figure 5 Cellular Layout with No Inter-cell Flows

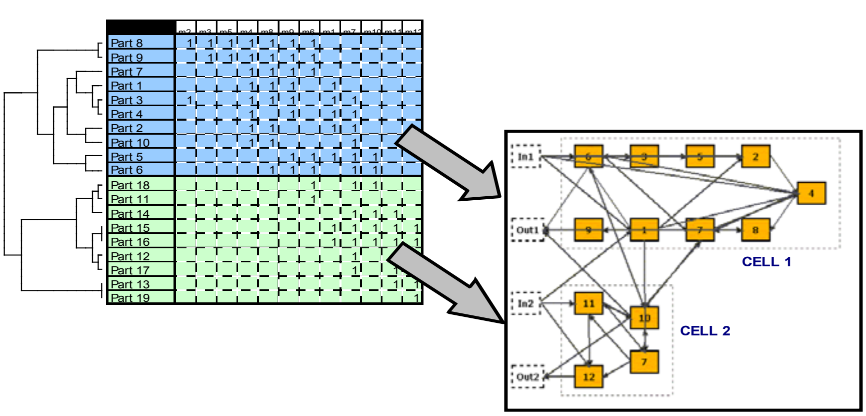

Figure 6 Cellular Layout with Inter-cell Flows

If we desire to implement two cells, then machines 1 and 6 must also be placed in Cell #2, and machines 7 and 10 must also be placed in Cell #1. Unless these machines are duplicated and assigned to both cells, inter-cell flows will occur that are not easy to coordinate and will be disruptive to operations in both cells. This is where the "Purchase Price" column in Table 2 plays a key role. Alternatively, instead of drawing the cut-off line between Part #6 and #18 in Table 5, we could have drawn it between Part #11 and #14. That would have eliminated the need to duplicate Machine #6 but it would make one part family (and its cell) much larger than the other part family. Many other strategies exist to eliminate the inter-cell flows, besides distributing the existing machines of each type among several cells or acquiring extra machines. They would be evaluated by a cross-functional team during one or more kaizens authorized by management. Interested readers who wish to obtain the two strategy maps that summarize all the strategies to eliminate or manage inter-cell flows in HMLV facilities are welcome to e-mail Dr. Shahrukh Irani at *shahrukhirani1023@yahoo.com*.

Based on Table 5, if it is desired to implement two independent cells with no inter-cell flows of parts, machines 1, 6, 7 and 10 must be duplicated in both cells. Figure 5 shows the layout that was developed for each cell. Instead, if it is desired to allow inter-cell flows between the two cells because machines 1, 6, 7 and 10 will not be duplicated, then the shop layout would be as shown in Figure 6. Both layouts were developed using the *Storm* software.

Sequence Similarity Analysis of Routings: If we use the 0-1 matrix to represent the routings in Table 1, we fail to capture the exact sequence in which machines are visited by a part. Table 6 shows results from an alternative method — sequence similarity analysis – that overcomes this major shortcoming of product-process matrix analysis. How would you put Table 6 to work? Imagine that, for each part produced in the facility, you implement a flowline cell simply by placing the machines that occur in its routing in sequence. You could not afford to buy that many machines if you had to do the same for all the parts being produced, right? Instead, what if you placed side-by-side those flowlines that produce parts with identical, or at least similar, routings? This would allow an entire part family with identical, even similar, routings to be produced on a single flexible flowline whose layout conforms with the routings of the parts in that family. For example, with reference to Table 6, you would start with just machine

Table 4	Initial 0-1 Matrix										
	m1	m2	m3	m4	m5	m6	m7	m8	m9	m10	m11	m12
Part 1	1			1				1	1			
Part 2	1			1			1	1				
Part 3	1	1		1			1	1	1			
Part 4	1			1			1		1			
Part 5	1					1	1		1	1		
Part 6						1	1	1	1	1		
Part 7				1		1		1	1			
Part 8		1	1	1	1	1		1	1			
Part 9			1	1	1	1		1	1			
Part 10				1			1	1				
Part 11						1						
Part 12							1				1	1
Part 13											1	1
Part 14							1			1	1	
Part 15	1						1			1	1	1
Part 16	1						1			1	1	1
Part 17							1				1	1
Part 18						1	1			1		
Part 19												1

Table 5	Final 0-1 Matrix											
	m2	m3	m5	m4	m8	m9	m6	m1	m7	m10	m11	m12
Part 8	1	1	1	1	1	1	1					
Part 9		1	1	1	1	1	1					
Part 7				1	1	1	1					
Part 1				1	1	1		1				
Part 3	1			1	1	1		1	1			
Part 4				1		1		1	1			
Part 2				1	1			1	1			
Part 10				1	1				1			
Part 5						1	1	1	1	1		
Part 6					1	1	1		1	1		
Part 18							1		1	1		
Part 11							1					
Part 14									1	1	1	
Part 15								1	1	1	1	1
Part 16								1	1	1	1	1
Part 12									1		1	1
Part 17									1		1	1
Part 13											1	1
Part 19												1

Cell 1

Cell 2

These machines will need to be duplicated in both cells

Table 6	Sequence Similarity Analysis of Routings													
Part No.	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Part 11				6										
Part 18				6					7	10				
Part 14								11	7	10				
Part 12								11	7			12		
Part 17								11	7			12		
Part 13								11				12		
Part 19												12		
Part 15	1					7	11	10	11			12		
Part 16	1					7	11	10	11			12		
Part 5	1			6				10	7				9	
Part 6				6				10	7			8	9	
Part 10					4				7		4	8		
Part 2	1				4				7		4	8		7
Part 4	1				4				7				9	
Part 3	1		2		4				7			8	9	
Part 1	1				4							8	9	
Part 7				6	4							8	9	
Part 8	3	5	2	6	4							8	9	
Part 9	3	5		6	4							8	9	

#6 in a machining flowline on the shop floor. But then you could only make part #11. Next, you would expand this flowline with three machines sequenced as follows: 6→7→10. This flowline could now produce part #11 *and* part #18. Next, if you placed machine 11 side-by-side with machine #6 and machine #12 side-by-side with machine #10, this new flowline could also make part #s 12,17, 13 and 19. Next, add Machine #1 at the front of the flexible flowline and we could make part #s 15 and 16 too. Stop! Beyond this point, this current flowline's part family ought *not* to include any more parts (See Figure 7).

Starting with the routing for Part #5, begin building a second flexible flowline for the second part family. Actual projects have involved up to about 1,500 routings. Therefore, we have preferred to compare the part families suggested by product-process matrix analysis with those suggested by sequence similarity analysis to determine the part families and compositions of their corresponding cells.

Part No.	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Part 11				6										
Part 18				6					7	10				
Part 14								11	7	10				
Part 12								11	7			12		
Part 17								11	7			12		
Part 13								11				12		
Part 19												12		
Part 15	1					7	11	10	11			12		
Part 16	1					7	11	10	11			12		
Part 5	1			6				10	7				9	
Part 6				6				10	7			8	9	
Part 10					4				7		4	8		
Part 2	1				4				7		4	8		7
Part 4	1				4				7				9	
Part 3	1		2		4				7			8	9	
Part 1	1				4							8	9	
Part 7				6	4							8	9	
Part 8	3	5	2	6	4							8	9	
Part 9	3	5		6	4							8	9	

If a standard 2-cell Cellular Layout were designed, this is where the two part families would be separated.

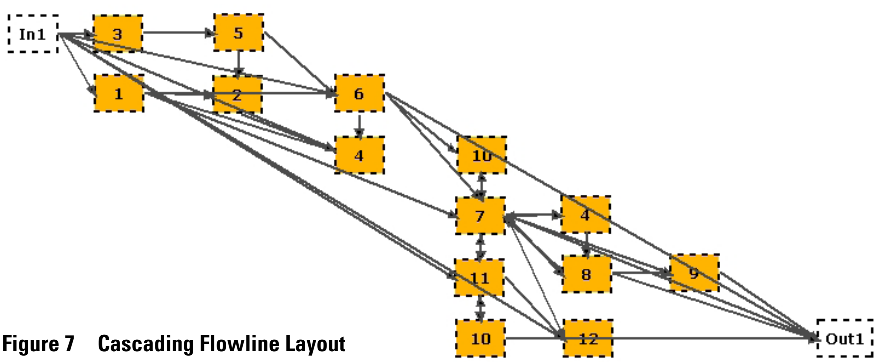

Figure 7 Cascading Flowline Layout

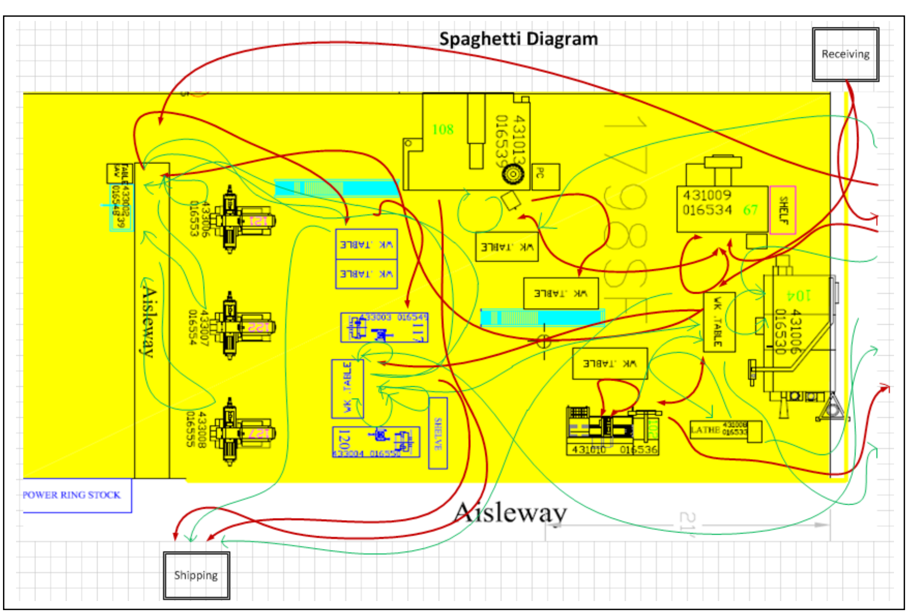

Figure 8 Material Flows in the Current Layout for the MP Cell

Application of the Theory at HCA-TX

At HCA-TX, we are already organized loosely into seven FLEAN cells — five in the machine shop and two in the molding department. While this gives us an excellent foundation for implementing Job Shop Lean, the word "loosely" describes our current state very well. Except for the QRC, the other cells are not self-contained, hence unable to function as ABUs (autonomous business units). Of the five existing machining cells in our facility, the MP Cell (MPC) was the best candidate for demonstrating the use of our computer-aided methodology for implementing Job Shop Lean, as described in Figure 3. Unlike the MPC, the other four machining cells are currently in flux for a variety of reasons, such as changes in their product mix, technology upgrades, reduction or replacement of vendors, etc.

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Dr. Shahrukh Irani is the Director of Industrial Engineering (IE) Research at Hoerbiger. In his current job, he has two concurrent responsibilities: (1) To undertake continuous improvement projects in partnership with employees as well as provide them OJT training relevant to those projects and (2) to facilitate the implementation of Job Shop Lean in HCA's U.S. plants.

Bearings Education: A Lot to Learn

Jack McGuinn, Senior Editor

Bearings ain't beanbag. They are complicated. They are big-business. They are often counterfeited. They are used in virtually anything that moves.

But it is the "complicated" part that challenges OEMs, job shops and other operations, and, most of all—their employees. Add to that the countless other entities around the world that are intimately involved with bearings and you can arrive at a semblance of an idea of just how important these precious orbs can be to a successful operation.

So is it any wonder that the bearing companies of the world—particularly the big boys—have made it a part of their doing business to provide quality, comprehensive education courses addressing bearings and just about anything else that bearings come in contact with? Yes, they educate "students" regarding each and every kind of bearing that exists and their attendant applications. But were you aware they also typically present training/education modules addressing tough topics such as condition monitoring, continuous improvement, phase analysis, bearing fault detection and analysis?

But this knowledge does not come cheaply. Depending on the provider, courses—especially on-site instruction—can run several thousand dollars per person—more than $10,000 for groups. Whether the cost is worth it has two answers. The training is undoubtedly something that a customer's employees cannot easily find anywhere else. As for the training providers, chalk it up to the cost of doing business. Think of it as hotel room service in that it is not a money-maker for the bearing companies, but it is something that customers have come to expect.

(Timken policy does not provide online course fees information, requiring interested parties to contact them directly.)

As David A. Novak Jr., Timken director of service engineering & strategic projects, puts it, "While some nominal fees can apply, our efforts are largely designed to help improve the operating outcomes for customers and end user operators. (Customers) tell us that they find the training to be very useful, often allowing them to recoup their investment in training many times over in improved maintenance practices or faster set-up and operational efficiencies."

Most companies no longer provide internal apprenticeship (training) programs and have chosen to outsource training for maintenance skills (courtesy SKF USA Inc.).

At SKF, "Our program is expected to cover our own costs, including enough to reinvest in our training business for continuous improvement," says Joseph Bruno, director, training & development for SKF USA Inc. "We make sure to maintain the integrity of our training programs by not turning them into sales events. We've been successful by providing high-quality training and focusing on competencies and skills growth. We determine our tuition rates based on our own costs and we are not influenced by other training organizations."

When you consider the cost outlay, time and expert personnel (when expert personnel are in short supply) needed to conduct these programs, their very existence is somewhat surprising.

"Most companies no longer provide internal apprenticeship programs and have chosen to outsource training for maintenance skills," Bruno points out. "SKF has decided to participate in providing skill training by bringing over 100 years of experience into the classroom. SKF expertise includes knowledge in bearing technology, condition monitoring and precision maintenance skills."

But a trained customer is typically a happy customer, as Timken's Novak points out.

"The Timken Company trains thousands of customers each year in formal training at our own facilities and onsite at their locations. Our knowledge of metallurgy, friction management and mechanical power transmission means we are well-qualified to provide both theoretical and practical customer education in the field or the classroom. Our customers demand longer life from their equipment, while using it at higher utilization levels, so our training is designed to help them solve problems more quickly and apply their new knowledge to make even more effective business and operating decisions.

"For example, Timken training programs teach customers about the merits of bearing remanufacturing and repair, which can result in significant cost and time savings compared to purchasing new bearings. Another focuses on detecting modes of damage so that costly downtime can be proactively addressed."

As for the instructors themselves, Timken and SKF take different approaches to reach the same result. The former's trainers hold other positions in the company while SKF chooses to have dedicated trainers.

SKF's Bruno puts it this way: "We have decided to take the right people and make them dedicated trainers. There are a lot of people with subject matter knowledge, but that doesn't mean they are good trainers. Full-time positions mean we can identify subject matter experts with good training skills; then we can focus on honing those training skills and helping them become great teachers."

While over at Timken, "Timken associates deliver training based on their expertise in a functional area, and their roles in educating are in addition to their other job responsibilities," says Novak. "With this kind of model, Timken experts can offer training based on their most current and relevant working knowledge."

Given the different approaches, a natural next question is what kind of experience does each of these two companies expect from their instructors?

"Across our company, more than 40 percent of our salaried force is engineers by trade or training," Novak explains. "Beyond their formal training, Timken associates gain further knowledge from their years of experience in engineering, service, research and product development. They then share their knowledge and experiences as part of our comprehensive customer and end-user training."

"Our trainers have several years' experience in their field and a broad range of experience in a variety of industries and applications," says Bruno.

Another unavoidable thought regarding instructors' levels of expertise—and their availability—is that it must be difficult to identify, hire and retain people who can present the coursework capably.

"Certainly knowledge sharing is a concern for our industry, as we experience attrition and deal with fewer and fewer candidates coming into the labor force who have practical hands-on experience," Novak allows. "You just don't find as many who have rebuilt a transmission, torn apart a tractor or even replaced bearings in their trailer. And these days electronics plays a larger role as well. So at Timken, we help them further develop those applied skills. We employ those who have a passion for their craft and natural curiosity in the mechanical world, and find them eager to learn and share what they know."

Also in play is the steady advance of technology in the bearings and related industries, meaning that training personnel must work to keep pace in order to remain relevant and at the top of their game.

Indeed, "Our program has grown because of the needs in the marketplace," Bruno says. "We listen to our customers' needs and when we feel there is a match with our knowledge and capabilities, we will provide the appropriate competency training."

Novak agrees in that "Technical advances and customized solutions create a need for our customers to increase their understanding of a product or sys-

> "This (training) all adds up to a bigger picture of effectively managing production assets. The skills and competency training we provide is designed around our philosophy of 'asset efficiency optimization.'"
>
> Joseph Bruno, SKF USA

tem. Timken expertise supports the growth in training services."

And with all this technical expansion and customer need, SKF, for example, has continued to expand its training force.

"We started with one person and have grown to several trainers in over 70 countries with a globally standard course portfolio," Bruno says. "We have a team that coordinates the program globally and each country manages their training staff locally."

As stated at the outset, bearings can be extremely complicated. Is there a particular area that is most problematic for instructors?

Bruno maintains that "One of the more challenging areas is analyzing bearings that have failed or become damaged in operation. Determining the root cause of these conditions can provide solutions to keeping machines running longer and increasing productivity.

Timken customers find bearings training to be very useful, often allowing them to recoup their investment in training many times over in improved maintenance practices or faster set-up and operational efficiencies (courtesy The Timken Company).

"But analyzing a damaged bearing is a complex and complicated process. Teaching these skills requires a highly competent and experienced instructor. We've dedicated a tremendous amount of time, resources and development in a course that can guide the student through the process of successfully using visual distress and standard terminology to find the true root cause."

And then of course there is the training provided to bearing companies partners—bearing distributorships. As you might expect, the training approach is somewhat different.

"It's a matter of their needs and focus," Bruno points out. "Our distributors need to understand the products' capabilities so they can match their customers' needs to the best solution. Then our customers' needs become how to install, maintain, monitor and prolong the life of the product."

At Timken the distinction is narrower.

"Although distributor partners want more commercial training, we offer similar technical training to both groups," says Novak. "Our distributor partners desire the same technical knowledge to best serve their customers."

At the end of the day, these training programs are just a part—albeit an important part—of what is required to be a valued supplier. If the proper training is provided, the customer personnel being trained—and the company they work for—are all the better for it.

"As our customers fine-tune their operations and increase their need to understand bearing care and maintenance, the demand for our training also increases. We provide value to our customers through focused training that is designed to help them perform their jobs better and solve problems quicker," says Novak.

SKF's Bruno adds that "This (training) all adds up to a bigger picture of effectively managing production assets. The skills and competency training we provide is designed around our philosophy of 'asset efficiency optimization.' This is a structured process where we start with a focus on maintenance strategy, then identifying what work needs to be done, putting controls around that work and finally executing the work. Then we begin a follow up process to bring things full-circle for continuous improvement. Providing training to our customers and distributors is an intense but rewarding career."

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Off-Highway Gears

Market needs push in 2013, but will it get one?

Matthew Jaster, Senior Editor

The construction/off-highway industries have been here before. New equipment, technologies and innovations during an economic standstill that some have been dealing with since 2007. Business is bad in Europe, lukewarm in the United States and companies are still waiting for the gigantic construction boom promised in areas like India, China and South America. Companies like Boston Gear, Durst Drives, ZF, Oerlikon and Forest City Gear supply gears for the off-highway industry. Some simply provide the gears themselves while other companies provide a total gearbox and/or assembly.

While slow growth will be the norm in the United States for 2013, encouraging signs in Europe (Bauma 2013 in Munich, Germany) suggest the economy will look better toward the end of the year. For the rest of the world, significant economic growth probably won't make news until 2014. And even then, how much growth are we talking and what part of the world will take the most advantage of it?

Anticipated Growth

"Major market segments are experiencing growth, and Durst expects continued business expansion over the next five-year period as a result," says John Locarno, global sales and marketing manager for the Durst division of Regal Beloit Corporation. "The increase in global population and personal incomes over the next several years, particularly in developing economies, will lead to investment into water, sewer, transportation and development programs and will support demand for infrastructure construction. Durst anticipates greater demand for the off-highway equipment into which Durst products are integrated."

Boston Gear expects moderate growth in this area, according to Mike Stegmann, product manager. "North America and Asia should provide the largest opportunities."

Wendy Young, president of Forest City Gear, expects the off-highway market to recover in the near future. "Last year we lost 15 percent of gross sales in heavy construction and off-road production. We expect some return toward the end of this year with some possible growth for 2014, but not returning to the levels of 2012."

Young continues, "Our gears are going into major off-highway applications. When our economy improves, I believe our orders will increase and return to above former levels due to pent up demand. We do not have orders for parts such as these offshore. I expect we will resume levels in our off-road and recreational applications, as well, from the business we receive in the United States and Canada."

Oerlikon Drives Systems Segment is currently strengthening its precision gearing, transmission and driveline solutions in Russia. Recently, Oerlikon attended the CTT 2013 in Moscow to meet current and new customers and reinforce the brand presence of both Oerlikon Fairfield and Oerlikon Graziano.

Dana Holding Corporation is bolstering its ongoing support of the growing construction market in Russia and the

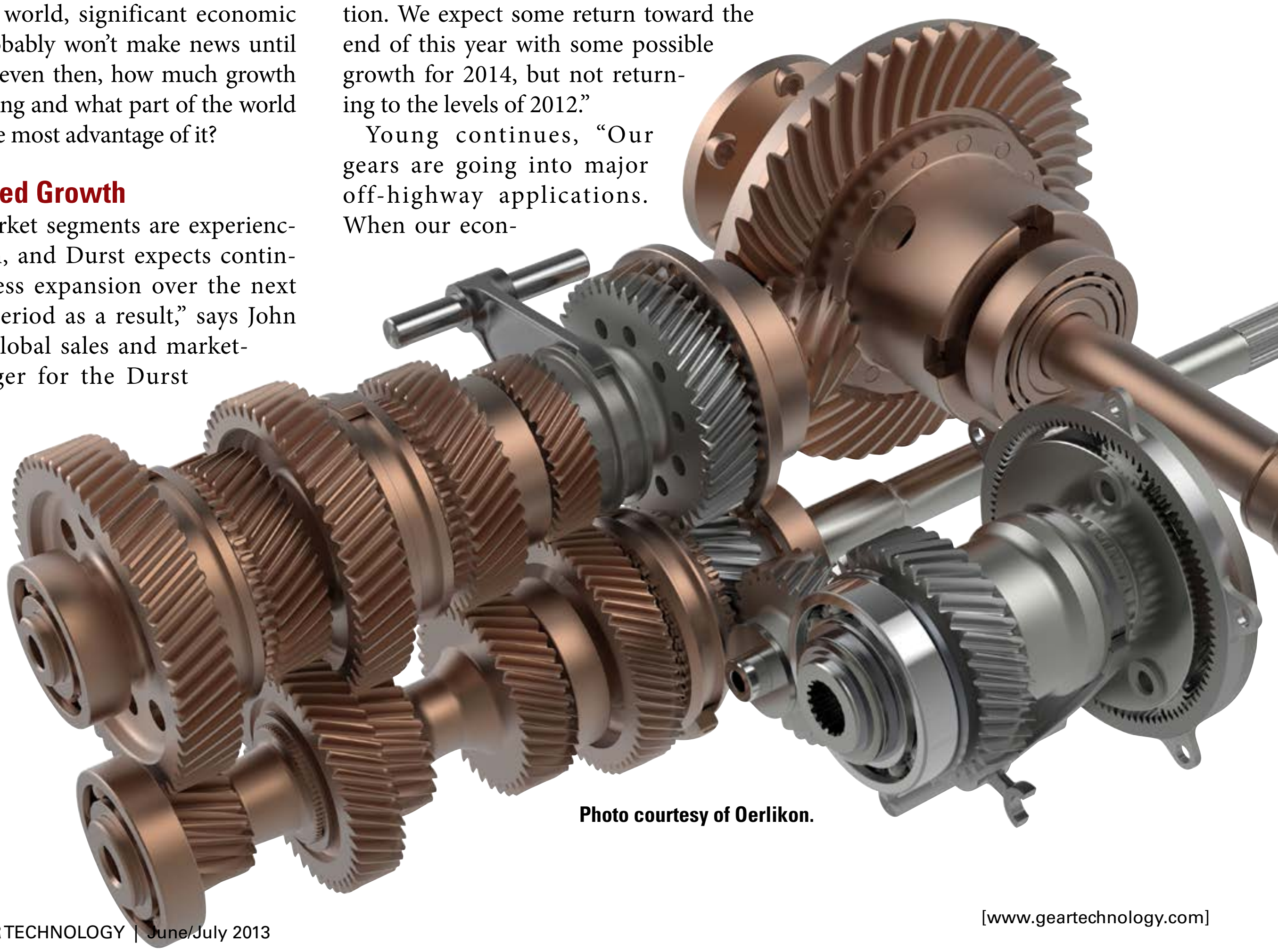


Photo courtesy of Oerlikon.

Commonwealth of Independent States by expanding its aftermarket network and offering customized drivetrain solutions for the region. Construction activity in Russia is expected to continue accelerating through 2015, spurred by state-funded civil engineering projects for transport and infrastructure expansion. Dana is increasing its local resources by actively expanding its network of Authorized Spicer Off-Highway Service Centers and strengthening the capabilities of the Dana distribution center in Gyor, Hungary, to improve the delivery of genuine original-equipment service parts.

Dana is also increasing the availability of driveline solutions produced in China for equipment buyers in the region. These solutions include the Spicer TZL Series of four-speed powershift transmissions as well as the Spicer Rui Ma brand of transmissions and axles, which provide an optimized blend of product features, performance, dependability and cost.

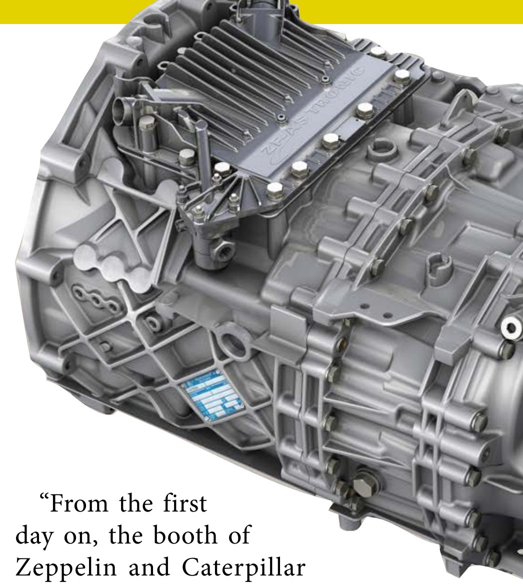
"At Dana, we recognize the increasing importance of this vibrant, growing market and the crucial role the construction industry will play here," says Aziz Aghili, president of Dana Off-Highway Driveline Technologies. "We will continue to expand our support with complete drivetrain solutions and world-class aftermarket capabilities for construction equipment and other off-highway applications."

Is a trade show a good market indicator of future business? If so, Europe has been in need of good news on the construction/off-highway segment for some time. Normal growth is expected with some areas even poised to make larger impacts by the end of the year. The 31st edition of Bauma 2013 amassed more than 530,000 visitors in April, breaking not only attendance records, but also exhibitor numbers and exhibition space.

"This is a very good for our industry in these turbulent times and it will certainly give it a boost," says Johann Sailer, chairman of the Construction Equipment and Building Material Machinery Association of VDMA and the president of the committee for the European Construction Equipment Industry (CECE).

"This has been an absolutely positive trade show for us," says Stefan Heissler of the Liebherr International AG board of directors. "We welcomed customers from all around the world at our booths and we signed up lots of new orders. In some product sections, we exceeded our expectations."

> "Major market segments are experiencing growth, and Durst expects continued business expansion over the next five-year period as a result."
>
> John Locarno



"From the first day on, the booth of Zeppelin and Caterpillar was almost overrun with visitors," says Michael Heidemann, chairman of the management board at Zeppelin Baumaschinen GmbH. "We have sold more machines at Bauma 2013 than ever before and have set a new record."

Putting the Gears to Good Use

Whether standalone products or part of an assembly, off-highway companies have made the most of their downtime by producing new transmission and driveline products and technologies.

ZF, a worldwide automotive supplier for driveline and chassis technology, supplies automated transmission systems for mobile cranes and special vehicles. The 12 or 16-speed ZF AS Tronic is available for mobile cranes in direct speed or overdrive transmissions. The main components, including dry clutch, are fully integrated into a light-metal alloy housing. There is no need for an additional transmission cooler in the standard vehicle configuration under normal usage when the outside temperature is at 40°C or below. The ZF AS Tronic is able to do more than driving and braking. The transmission can be combined with up to two clutch dependent or drive dependent PTOs that can be engaged independently of one another.

The technology of AS Tronic mid is based on the large AS Tronic; the 12 gear steps are shifted pneumatically. In order to cover a wide range of applications, ZF offers AS Tronic mid with low power-to-weight ratio for torque ranges from 800 to 1,600 N-m. In harbors, the medium AS Tronic version is used for in-terminal tractors of well-known manufacturers of these special vehicles.

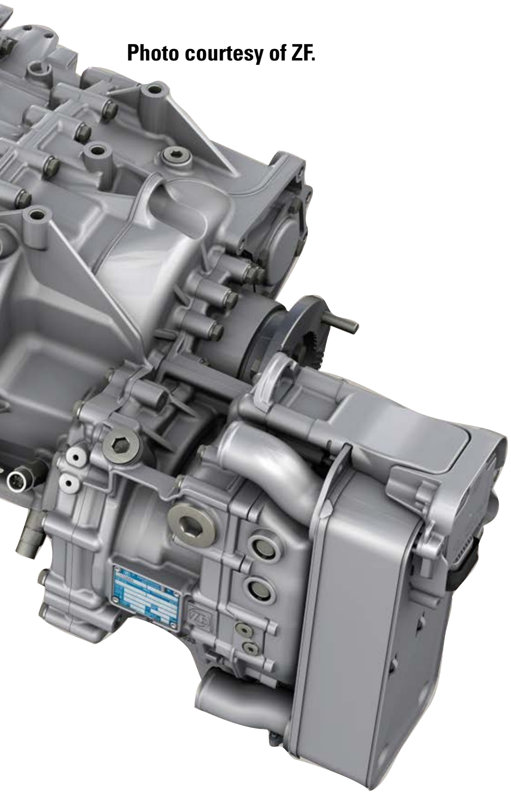


Photo courtesy of ZF.

At 840 or 900 mm in length and 50 or 65 kg in weight, the two ZF AS Tronic mid variants are both shorter and lighter than the "original" model. The 12-speed transmission works with a direct speed or overdrive ratio. Clutch dependent or drive dependent PTOs can be engaged independently of one another via two PTO locations. The TC Tronic HD (Heavy Duty) was developed by ZF to meet the requirements of engines with continuously increasing horsepower. It also aims at applications in crane trucks and heavy trucks starting from 6 axes (72 tons) and has been designed for torques of up to 3,000 N-m.

For several years Oerlikon has been studying, developing and producing new solutions for a market that is increasingly demanding a more efficient energy use and emission reduction. In the last 20 years Oerlikon Graziano has acquired experience in electric and hybrid driveline assemblies. The products go from the very first golf/utility transaxle to single- and multi-speed transmission systems, developed for vehicles ranging from the ever more popular and fancy "full electric zero emissions" city cars, to light commercial electric vehicles, and exotic full electric and hybrid sports cars. All these transmissions share some advantages, such as the compact size, the light weight and a high operational efficiency.

Additionally, these products were recently displayed at the VDI Wissensforum in Friedrichshafen, Germany including a four-speed seamless-shift transaxle, a new eDCT multispeed transmission that provides EVs with greater range while reducing vehicle weight and battery pack size. This inno-

> "Our gears are going into major off-highway applications. When our economy improves, I believe our orders will increase and return to above former levels due to pent up demand."
>
> Wendy Young

vative transaxle uses the principles of dual clutch transmissions (DCTs) to provide seamless shifting and up to 15 percent improvement in vehicle efficiency.

For passenger cars and light commercial vehicles the company will exhibit a dual-speed seamless-shifting transaxle that can be coupled with a transversal electric motor, for front or rear full electric axle. It has been designed for inner city transport, not only for the reduction of the CO_2 emissions and environment noise but also for significant cost savings. The transaxle has been developed together with VOCIS Driveline Controls, contributing with its control software and electronic hardware design skills to the transmission design.

"Electric drive is a field where integrated system-level optimization is the only way to offer significantly improved customer benefits, and our efforts are aimed at increasing as much as possible the integration level of motor and transmission within the powertrain," said Paolo Mantelli, head of performance automotive at Oerlikon Graziano. "Another crucial element for us is the innovation level of transmissions conceived for the ultimate electric and hybrid vehicles: Our multi-speed concepts are the most suitable for a modern full electric or even hybrid vehicle, allowing the best sizing of the electric motor and usage of batteries' power."

Dana recently showcased the Spicer Model 114 planetary rigid axle for medium-sized front-end loaders during CTT 2013 in Moscow. The Spicer Model 114 axle includes a limited-slip differential lock with a 45-percent locking capacity, delivering excellent tractive force and nimble vehicle steering for front-end loader applications. The axle can be configured with a variety of other hydraulic-locking differential options, including dog clutch and multi-disc clutch. Inboard wet disc brakes offer proven fail-safe stopping performance and energy absorption, while a fail-safe park brake featuring internal negative SAHR is available as an option.

Durst is a complete solutions provider, focusing on challenging power transmission applications across a number of industries. "Durst manufactures all products to customer specifications, including clutches, brakes and yokes. As a solutions provider with extensive industry experience, Durst fills the gaps left by most vendors. Durst provides single-source capabilities that translate into reduced project costs for the client. Our engineering team works with system integrators to evaluate needs and quickly isolate the critical factors needed for the success of complete power transmission solutions," said Locarno.

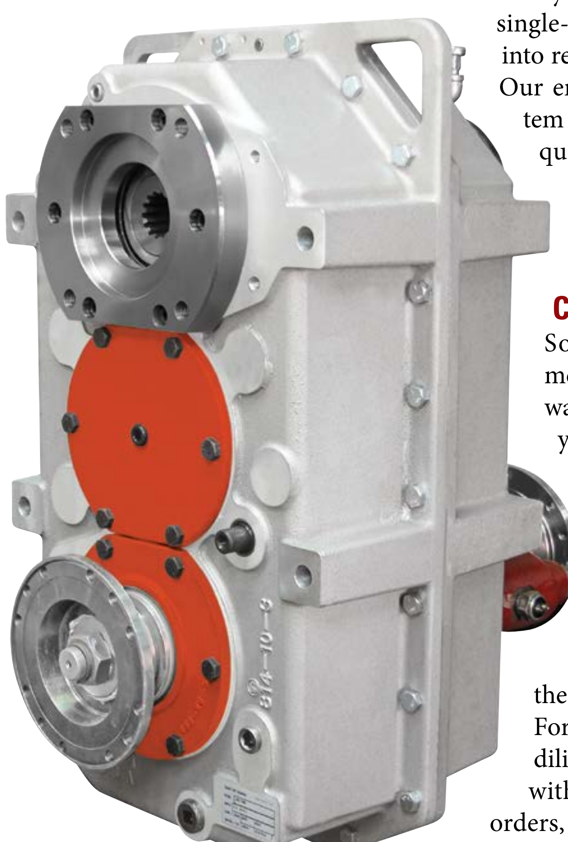


Photo courtesy of Durst.

Challenges and Trends

So what does this new business mean to those working in off-highway applications? Staying on top of your inventory, reducing costs and getting the right people trained (or rehired) to do the job.

"In the past when business such as this returned, our experience is that it is usually with immediate vigor, and we cannot fulfill those needs with the same immediacy," said Young at Forest City Gear. "We have worked diligently to fill our empty machines with orders (perhaps much smaller orders, but many) and do not have the capacity. We had reduced our workforce, and now have to address the growing needs of former customers. We increase second shift requirements, add overtime, and if we are extraordinarily lucky, can find a few unemployed and talented individuals who were laid off by companies such as ours during the slowdown."

At Boston Gear, significant challenges include offshore competition and tooling costs. "We manufacture mostly in-house with 70 percent stock and 30 percent customized gears. Higher value-add from engineered solutions distinguishes our company from a pure commodity supplier."

"Thousands of new products hit the market each year. Many quietly disappear without ever registering on the radar," said Locarno. "What sets Durst products apart from the also-rans is product excellence and cost-effective manufacturing. Product development is an ongoing process, and the Durst organization is always working toward new product opportunities. Development procedures streamline the steps from the creation of the concept through development and design to the final production and service."

Gear manufacturers serving the off-highway industry have had plenty of time to make products since 2007. Here's hoping that 2013-2014 is when customers get back to purchasing them.


Photo courtesy of Forest City Gear.

> "In the past when business such as this returned, our experience is that it is usually with immediate vigor, and we cannot fulfill those needs with the same immediacy."
>
> Wendy Young

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Off-Highway Global Update

Asia

The Asia Off-Highway Vehicle Summit took place in Singapore from June 27-28. Despite a turbulent economy and an imposing wall of challenges facing the OHV market in recent years there are still formidable advances and developments occurring in the Asian and global markets. With the internationalization of many leading OHV companies, competition is also increasingly fierce. Topics discussed included independent analysis of leading markets, technical development trends as well as new profit points of the aftermarket, including remanufacturing and leasing. The event gathered more than 540 delegates from 438 businesses and was sponsored by Duxes of Shanghai.

Europe

Strict emission legislation is forcing OEMs of construction machinery to implement not only wide-ranging but also expensive measures in their diesel engines. Over the last decade, many engine suppliers invested heavily on research related to developing emission control technologies. Understanding and controlling the combustion process is the first step in reducing engine-out emissions and reducing the burden on the emission control after-treatment systems. Thus the engine design is an important part of controlling and facilitating the combustion process. In diesel engines, particulate emissions are controlled by optimizing the mixing between air and fuel. The turbulence, which helps the better mixing of air and fuel, is achieved by modification. Additionally, the topic of high efficiency of non-road vehicles has become an important issue on the market. All of these issues were recently discussed at the 3rd International Conference of Next Generation Off-Highway Engines 2013. European regulations, advanced engine concepts and efficiency improvements were debated during the event in Hamburg, Germany.

India

According to the organizers of The Big 5 Construct India, the country is expected to emerge as the world's 3rd largest construction market by 2020. The Big 5 trade show, taking place in Mumbai from September 2-4, will give insight into the trends and developments taking place in the off-highway and construction markets. The largest construction event in the Middle East, Big 5 attracted more than 60,000 audited buyers at last year's show. This event will feature free seminars, panel discussions and live interactive demonstrations including trend analysis and case studies on these growing markets.

Worn Gear Contact Analysis

Email your question—along with your name, job title and company name (if you wish to remain anonymous, no problem) to: *jmcguinn@geartechnology.com*; or submit your question by visiting *geartechnology.com*.

All graphics courtesy KISSsoft AG.

QUESTION

How does one perform a contact analysis for worn gears?

Response provided by Dr. Stefan Beermann:

Wear is an ongoing process that decreases the quality of a gear with increasing number of load cycles. So typical issues addressed by a contact analysis of a worn gear is the transmission error caused by the deviations from the involute, and maybe the effect on the contact stress due to nicks and edges on the theoretically smooth profile. Also, the change in load distribution is of interest.

The information about the wear can come from two sides: it might be from simulation or from a real gear measured on an appropriate measuring machine. In the first case there is usually no problem to get the worn profile into the contact analysis. In the second case you have to be careful in the preparation of the profile data; scan the gear on a measuring machine. Use a high resolution, preferably more than 100 points-per-flank. Then the best next step would be to read the points into a CAD package and convert the points into a curve. For instance—a spline curve or circular arcs. Here you have to pay extra attention, since you might analyze approximation artifacts afterwards instead of the effect of the wear.

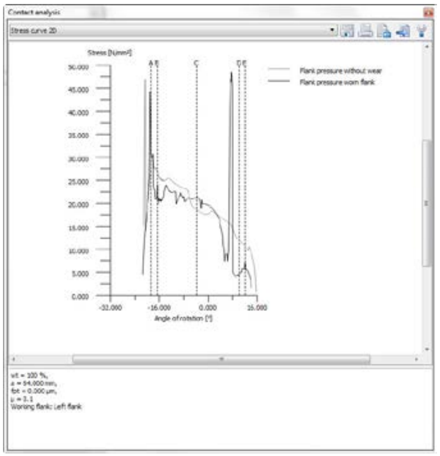

Figure 3 **Hertzian pressure with and without wear.**

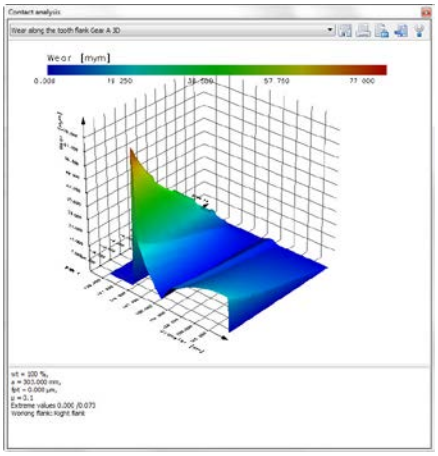

Figure 4 **Wear on a gear set with misaligned axes.**

The next decision is if you want to look at three-dimensional wear data or two- dimensional in a transverse plane. If the wear is more or less constant over the width, meaning along a constant diameter, the two-dimensional analysis is usually clearer in the results. This would be typical for plastic gears that often show significant wear due to sliding in the profile direction. However, metal gears also can show this type of wear, especially at slow speed or if micropitting occurs.

Should the wear significantly differ over the width—for example, if it is stronger in the middle or on one side of the gear—the three-dimensional approach is necessary. One would expect this if the gears are misaligned due to shaft deformations or tolerances. Sometimes wear can even help in these situations; i.e., an initially high $K_{H\beta}$ could be reduced due to wear in a running-in phase.

For a three-dimensional analysis the easiest way is to read the wear in as topological modifications. This way you can define the wear as a two-dimensional function over the whole flank.

Figure 1 shows the result of a simulated wear inside *KISSsoft* after performing contact analysis. You can see how the path of contact is completely torn apart. The respective transmission error shown in Figure 2 and the Hertzian pressure in Figure 3 show the bad effect of the wear: grey is the curve without wear; blue and black with wear. Of special interest in Figure 3 is the shift of the first contact to a later point in time, and the second peak in pressure towards the end of the contact.

For the three-dimensional case you find the simulation of the wear over the flank for a gear set with misaligned axis in Figure 4. This would be the topological modifications that one would feed into the contact analysis to determine the transmission error, and so on.

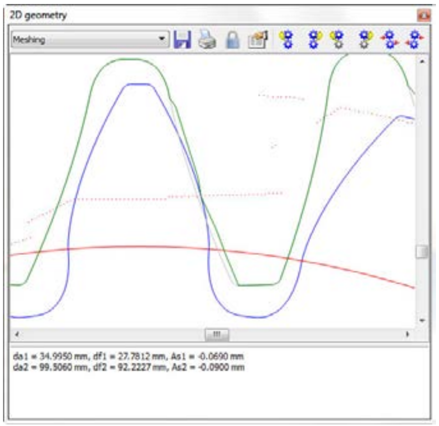

Figure 1 **Contact points of a gear meshing with one (plastic) gear showing significant wear.**

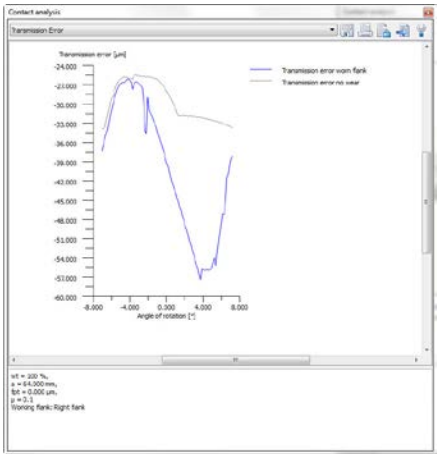

Figure 2 **Transmission error with and without wear.**

Dr. Stefan Beermann is CEO of KISSsoft AG

New Methods for the Calculation of the Load Capacity of Bevel and Hypoid Gears

Christian Wirth, Bernd-Robert Höhn and Christo Braykoff

Flank breakage is common in a number of cylindrical and bevel gear applications. This paper introduces a relevant, physically based calculation method to evaluate flank breakage risk vs. pitting risk. Verification of this new method through testing is demonstrably shown.

Introduction

Flank breakage in cylindrical and bevel gear applications typically initiates in the active flank, approximately in the middle of the active tooth height, and subsequently propagating to the tooth root of the unloaded flank side.

Crack initiation can be localized below the surface in the region between the case and core of surface-hardened gears. This failure mode cannot be explained by known causes such as tooth root breakage or pitting. Even bevel gears in truck and bus applications are at risk of damage from sub-surface fatigue if an optimum utilization of material is not achieved. In such cases a balance between the flank breakage and pitting risk must be struck. This paper describes a new "material-physical-based calculation" method to evaluate risk of flank breakage vs. pitting damage. This method was used to improve the design of a gear set that failed in several cases due to flank breakage on the wheel in test vehicles of MAN Truck & Bus (MTB). Figure 1 shows a typically damaged tooth on the wheel. In some cases pitting occurred on the coast flanks of the wheel (Fig. 1 — right). The following demonstrates how it is possible to increase the load capacity of the wheel regarding pitting and flank breakage by means of this new method, as proven in successful test runs.

Flank Breakage in Bevel Gears

Flank breakage often appears without any of the other commonly seen surface failures such as scuffing, pitting or micropitting. In some cases only one tooth is affected, but usually more than one tooth fails (Fig. 2). Inadequate material properties and heat treatment are expected to increase the risk of flank breakage, especially insufficient core strength and toughness, or insufficient—or too high—case depths (Refs. 4–5). In many U.S. publications flank breakage is also called "sub-surface fatigue" or "sub-case fatigue." In these papers the flank pressure is also regarded as the decisive parameter (Refs. 2, 6, 13, 15, 16).

By virtue of systematic tests, Annast (Ref. 1) investigated the influence parameters of flank breakage in bevel gears. He identifies — beyond the known influence of load conditions — case depth and core hardness as important parameters. Analysis of damage patterns of test and practical gears showed that the initiating crack always started below the surface; i.e., in the region of the transition from case to core. For unidirectional loading, the crack propagates to the active flank on one side and to the tooth root on the other. Annast analyzed the subsurface stresses with *ROSLCOR* (*Rolling and SLinding Contact according to OsteR*) (Refs. 9 and 17) by using the shear stress intensity hypothesis according to Tobie (Ref. 18). Oster (Ref. 17) defined, on the basis of (Ref. 21), the potential for considering compressive stresses in the case for the evaluation of the material expo-

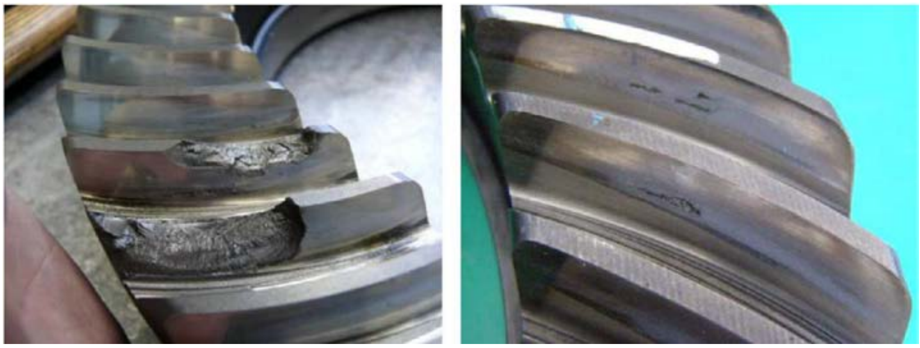

Figure 1 Flank breakage and pitting on wheel.

Figure 2 Flank breakage — two different wheels.

sure. Tobie expanded this method with the possibility of including tensile, residual stresses (in the core) as well. However, the overriding influence of tensile-residual stresses simulated by the Tobie model could not be validated in any other investigation (e.g., ANSI/AGMA 2003–B97). Annast argued that it is sufficient to analyze only the section between the flank surface and the transient region of case and core. Indeed, tensile residual stresses were therefore not examined by Annast. In his proposed standard-capable method, only the region-of-transition is regarded. A critical ratio between the acting maximum shear stress and the core hardness was derived from the test gears and practical applications; if the ratio exceeds the limit, the risk of flank breakage is considered high.

Because of the uncertainties in the methods described above, a new calculation method was proposed by Wirth (Ref. 20) for the rating of bevel and hypoid gears. The method is based on Oster's and Hertter's work (Ref. 8). Hertter proposed for cylindrical gears an enhanced shear stress intensity hypothesis to evaluate the subsurface stresses — even under consideration of compressive and tensile residual stresses; a separate examination of dynamic and static exposure allows for consideration of fatigue failures, as well as failures from yielding. Because the local allowable strength values in the considered material element are derived from the hardness by means of material-physical relations, the new method is termed the "material-physical-based calculation method." Wirth adapted Hertter's method to bevel and hypoid gears, considering their specifics as, for example, sliding conditions.

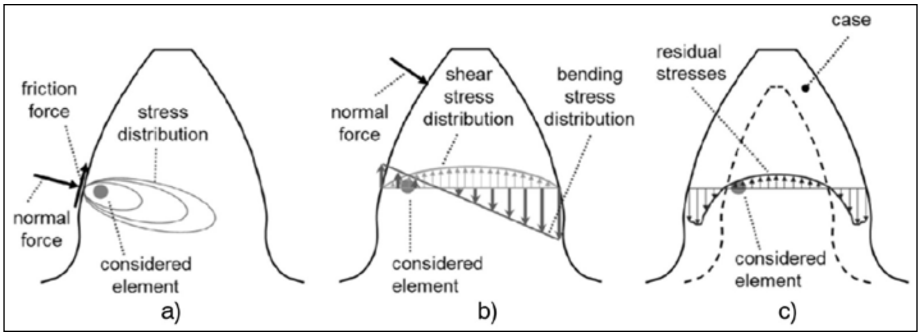

Figure 3 **Stress conditions inside the tooth.**

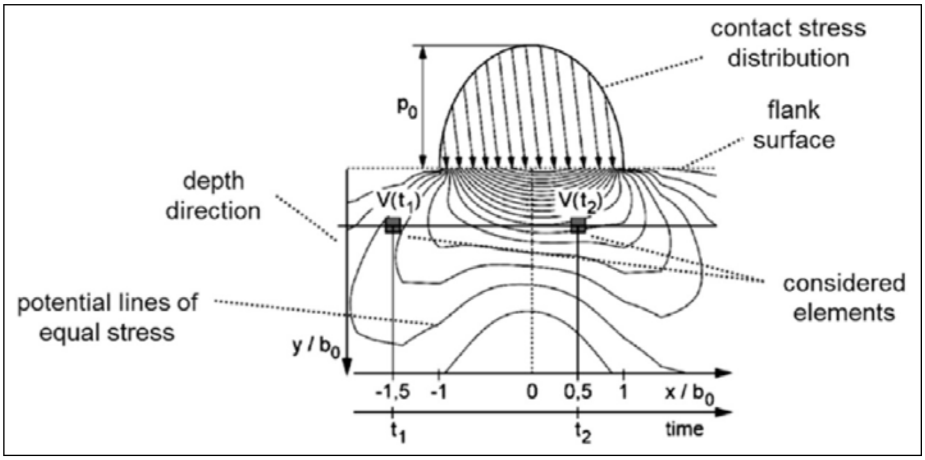

Figure 4 **Time–dependent stress components in a rolling contact.**

Material-Physically Based Calculation Method for Bevel and Hypoid Gears

The material-physical-based calculation method allows the consideration of complex stress conditions beneath the flank surface that are caused by the load and heat treatment process. The permissible stresses are derived from the hardness values and material-physical parameters. The occurring stresses are compared with the permissible stresses in discrete sections in the material. On the basis of a shear stress intensity hypothesis (SIH), the material exposure is determined.

General stress conditions in the tooth. Inside the tooth, beneath an ideal smooth flank surface, the total stress conditions are composed of:

- Stresses due to normal contact load (Hertzian theory)
- Shear stresses on the surface caused by friction
- Thermal stresses caused by the thermal gradient
- Stresses caused by bending mechanism
- Residual stresses

Figure 3 illustrates the stress components that influence the material exposure in a considered (infinitely small) element. In Figure 3a the stress components that result from the normal load on the flank are shown; the stresses according to Hertzian theory originate at the normal force. Due to the sliding components, the friction force that is tangential to the flank surface induces shear stresses. Figure 3b demonstrates the effect of bending by a normal force that acts above (in profile direction) the considered element. The components of the normal force cause normal stresses, with an approximately linear distribution over the tooth thickness and shear stresses, and with an approximately parabolic distribution and a maximum (distribution) in the middle of the tooth. Residual stresses result from the hardening and finishing process; as an example, Reference 20 shows that compressive stresses are occurring in the case and are balanced by tensile stresses in the core. Unlike the stresses in Figure 3a and Figure 3b, the residual stresses in Figure 3c are load-independent.

Oster and Hertter developed the *STRORHR* program for the calculation of all mentioned stress components on cylindrical gears. With this program it is possible to examine the material exposure in the sub-surface, below any contact point on the flank surface. Thus the examination direction is perpendicular to the flank surface.

Stress Conditions in the Rolling Contact

In any contact point on the flank, the rolling direction x can also be seen as the time axis. Figure 4 shows in principle the stress components under the surface. All volume elements in the same depth are exposed to equal stresses — but at different times. To evaluate the material exposure in a certain depth beneath the flank surface, the corresponding stresses have to be considered over the entire time axis (x axis).

However, in rolling contacts a turning principal coordinate system complicates the evaluation of the material utilization. A possible alternative is analyzing the dynamic stresses in rolling contacts; i.e., the shear stress courses in a sectional plane that are defined on the surface of the base sphere (Fig. 5).

Figure 6 shows for a rolling contact an example of shear stress courses in a certain sectional plane $\gamma\alpha$ and a material depth of $y/b_0 = 0.3$; $\tau\gamma\alpha$ (t_i) is the time-dependent graph for the projection of shear stresses in the directions n_2 and n_3 (Fig. 5). In Figure 6a no residual stresses are considered; as a consequence the point (0/0) is part of the course. At a certain time t_i — when the contact is still unloaded (e.g., the contact point of the flank surfaces is still far away from the regarded volume element) — $\tau_2 = \tau_3 = 0$ in the examined sectional plane. During the movement of the contact over the flank surface, the stress components τ_2 and τ_3 can be marked in the diagram. Of course if the influence of the moving contact point on the stresses at the examined plane is diminishing, the course will again reach point 0/0. As Figure 6a shows, the instantaneous stress vector (τ_2/τ_3) is completely turning during one load cycle, which means that it acts as an alternating load. Its maximum length is defined as the "maximum shear stress" $\tau_{max,a}$. The diameter of the circumscribed circle is $\Delta\tau_{max,a}$.

Figure 6b shows for the same examined sectional plane an equal load cycle, but in consideration of the residual stresses. Unlike before, the shear stresses τ_2 and τ_3 have discrete values — even if the contact is unloaded. However, the course of the pair of values τ_2/τ_3 is similar, meaning that $\Delta_{\tau max,a} = \Delta_{\tau max,b}$. An important fact is that the maximum shear stress $\tau_{max,b}$ is decreasing under the influence of (compressive) residual stresses ($\tau_{max,b} < \tau_{max,a}$). In other words, the maximum shear stress with high compressive residual stresses is smaller than the maximum shear stress without residual stresses. This is also valid for other sectional planes of the base sphere and corresponds with the accepted fact that compressive residual stresses reduce the maximum material exposure (e.g., Ref. 22).

The aim of a strength hypothesis is the evaluation of stresses occurring in the examined base sphere (or base element) to determine a number for the material exposure that correlates well with the failure mechanism. Many common criteria are not applicable for alternating stresses, as described before; various important hypotheses are discussed regarding their applicability for the rating of the material exposure in rolling contact. As a result of the investigation, a modified SIH was established (Ref. 8). This hypothesis can principally be used for:

- Rating the maximum exposure of the material (analysis regarding yielding)
- Rating the dynamic exposure of the material (analysis regarding fatigue)

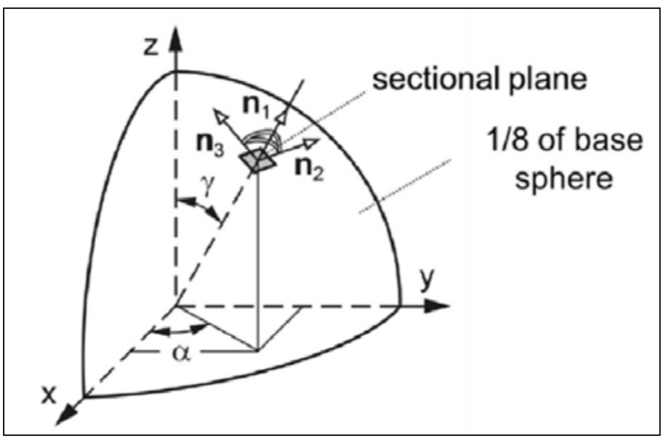

Figure 5 Base sphere with sectional plane (Ref. 8).

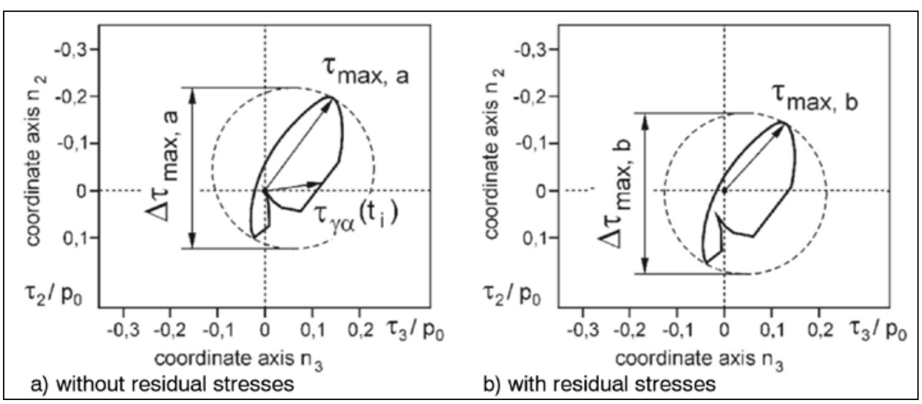

Figure 6 Example of shear stress courses in a sectional plane αγ in the depth y/b_0 = 0.3 for rolling conditions (Ref. 8).

Modified Shear Stress Hypothesis by Hertter (Ref. 8)

The bases for calculation of the decisive exposure in the examined base sphere are the stress courses in all of its sectional planes. According to (Ref. 12), in every sectional plane the normal stresses (orthogonal to sectional plane) can be calculated according to Equations 3 and 4. The shear stresses $\tau\gamma\alpha_a$ and $\tau\gamma\alpha_m$ are defined (Fig. 7) for the considered sectional plane. The radius of the smallest circumcircle of the stress course is the decisive amplitude of the shear stress. (*Ed.'s Note: The circumscribed circle—or circumcircle—of a polygon is a circle which passes through all the vertices of the polygon.*) The vector of its center point represents the mean shear stress.

Amplitude of shear stress (Fig. 7):

$$\tau\gamma\alpha_a = \tau_a(\gamma,\alpha) \tag{1}$$

Amplitude of shear stress (Fig. 7):

$$\tau\gamma\alpha_m = \tau_m(\gamma,\alpha) \tag{2}$$

Amplitude of normal stress: $\sigma\gamma\alpha_a = \sigma_a(\gamma,\alpha)$

$$\sigma_a(\gamma,\alpha) = \frac{\sigma_{max}(\gamma,\alpha) - \sigma_{min}(\gamma,\alpha)}{2} \tag{3}$$

Mean value of normal stress: $\sigma\gamma\alpha_m$ $\sigma_m(\gamma,\alpha)$

$$\sigma_m(\gamma,\alpha) = \frac{\sigma_{max}(\gamma,\alpha) - \sigma_{min}(\gamma,\alpha)}{2} \tag{4}$$

Dynamic exposure in sectional plane: $A(\gamma,\alpha)$

$$A(\gamma,\alpha) = \sqrt[\mu]{\frac{\alpha\tau_a^\mu(1 + m\tau_m^\mu) + b\sigma_a^\mu}{\sigma_A^\mu}} \tag{5}$$

Total dynamic exposure, $A_{int\,a}$

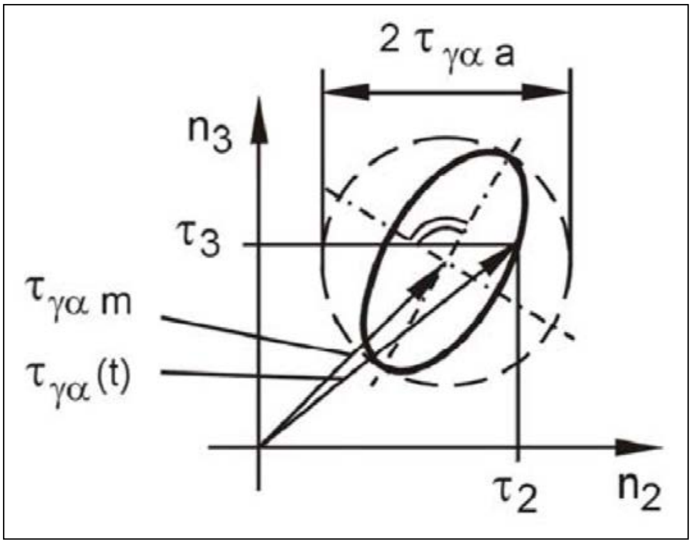

Figure 7 **Definition of amplitude and mean value of any shear stress curve plotted over time in a discrete sectional plane (Ref. 8).**

$$A_{int\,a} = \sqrt[\mu]{\frac{15}{8\pi}\int_{\gamma=0}^{\pi}\int_{\alpha=0}^{2\pi}\left([A(\gamma,\alpha)]^2 \sin\alpha\right) d\alpha d\gamma} \tag{6}$$

Maximum exposure in sectional plane, $A_{max}(\gamma,\alpha)$

$$A_{max}(\gamma,\alpha) = \sqrt[\mu]{\frac{\alpha\,\tau_{max}^{\mu} + b\,\sigma_{max}^{\mu}}{R_{p0,2}^{\mu}}} \tag{7}$$

Total maximum exposure, $A_{int\,a}$

$$A_{int\,a} = \sqrt[\mu]{\frac{15}{8\pi}\int_{\gamma=0}^{\pi}\int_{\alpha=0}^{2\pi}\left([A_{max}(\gamma,\alpha)]^2 \sin\alpha\right) d\alpha d\gamma} \tag{8}$$

Constant a

$$a = \frac{1}{5}\left[3\left(\frac{\sigma_W}{\tau_W}\right)^2 - 4\right] \tag{9}$$

Constant b

$$b = \frac{1}{5}\left[6 - 2\left(\frac{\sigma_W}{\tau_W}\right)^2\right] \tag{10}$$

Constant m

$$m = \frac{1}{\alpha}\left[\frac{\sigma_W^2 - \left(\frac{\sigma_W}{\tau_W}\right)^2\left(\frac{\tau_{Sch}}{2}\right)^2}{\frac{12}{7}\left(\frac{\tau_{Sch}}{2}\right)^4}\right] \tag{11}$$

where:

$\tau_a(\gamma,\alpha)$ is amplitude of shear stress in sectional plane, N/mm^2
γ is angle for sectional plane (Fig. 5), degrees
α is angle for sectional plane (Fig. 5), degrees
$\tau_m(\gamma,\alpha)$ is mean value of shear stress in sectional plane, N/mm^2
$\sigma_a(\gamma,\alpha)$ is amplitude of normal stress in sectional plane, N/mm^2
$\sigma_{max}(\gamma,\alpha)$ is maximum of normal stress in sectional plane, N/mm^2
$\sigma_{min}(\gamma,\alpha)$ is minimum of normal stress in sectional plane
$\sigma_m(\gamma,\alpha)$ is mean value of normal stress in sectional plane, N/mm^2
$A(\gamma,\alpha)$ is dynamic exposure in sectional plane
μ is constant exponent $\mu = 2$
b is constant (material-dependent)
σ_A is material amplitude strength according to normal stress, N/mm^2 (Ref. 8)
$A_{int\,a}$ is total dynamic exposure of base sphere
$A_{max}(\gamma,\alpha)$ is maximum exposure in sectional plane
$R_{p0,2}$ is yield strength, N/mm^2
A_{int} is total maximum exposure of base sphere
a is constant (material-dependent)
σ_W is material alternate strength according to normal stress, N/mm^2 (Ref. 8)
τ_W is material alternate strength according to shear stress, N/mm^2 (Ref. 8)
m is constant (material-dependent)
τ_{Sch} is material pulsating fatigue strength, N/mm^2 (Ref. 8)

With the stress values according to Equations 1–4, the material exposure $A(\gamma,\alpha)$ in the considered sectional plane can be calculated (Ref. 8). The local amplitude strength σ_A is dependent on the mean value of normal stress $\sigma_m(\gamma,\alpha)$. The constants a, b and m are a function of the strength ratio σ_W/τ_W and the local torsional pulsating fatigue strength τ_{Sch} (Ref. 12).

The total dynamic exposure $A_{int\,a}$ is defined as the integral value of the exposure values $A(\gamma,\alpha)$ in all sectional planes and determined by Equation 6. The endurance limit of the considered element regarding fatigue is, per definition, reached when the total dynamic exposure becomes $A_{int\,a} = 1$. Values below stand for infinite life. In an analogous way, the maximum exposure regarding yielding is calculated (Eq. 7). For all sectional planes in the base sphere, the total maximum exposure for yielding is considered according to Equation 8. Again, for an infinite life the total maximum exposure has to fall below $A_{int} < 1$.

It is remarkable that all local strength values can be determined out of the Vickers hardness values by means of material-physical-based relations (Ref. 8).

Calculation Process for Bevel and Hypoid Gears: Overview

As described above, Hertter (Ref. 8) developed for cylindrical gears the material-physical-based method to evaluate the material exposure in the sub-surface of the tooth. The strength values are, accordingly, derived from the Vickers hardness test relative to volume element (base sphere).

The comparison of the local stress values with the local strength values provides a 3-D evaluation of the material exposure — not only close to the flank surface but also in an area close to the core. Whereas the allowable stress numbers according to ISO, DIN or AGMA are only valid for an optimally designed case depth, the material-physical-based method allows investigation of the influence of different hardness profiles on load capacity. Moreover, due to the local consideration of the material exposure, the failure mode becomes apparent.

Hertter expanded the computer program *ROSLCOR* (*Rolling and SLinding Contact according to OsteR* (Refs. 9 and 17)) that was developed at the FZG (Gear Research Center of the Technical University of Munich) with his material-physical-based method; Wirth (Ref. 20) adapted Hertter's method to bevel and hypoid gears. Wirth developed the computer tool *LokAna* (*Local Analysis*) that handles not only the sub-programs *BECAL (Bevel Gear Calculation)* (Ref. 10) and *ROSLCOR*, but also further calculations for bevel gears.

Calculation of Material Exposure

The Hertzian stresses on the flank surface of bevel and hypoid gears are determined with the FVA program *BECAL* (*Bevel Gear Calculation*) (Ref. 10). With the machine settings for the gear set, *BECAL* is able to generate the geometry of flank surface and tooth root. Based on this information a loaded tooth contact analysis (TCA) leads to the tooth root stresses and to the local-occurring Hertzian stresses; deflections of housing, bearings and shafts can be also considered.

For any calculated point on the flank — the partial line of contact, its relative curvature, the acting normal force, and Hertzian stress — are determined. For *ROSLCOR*, the considered contact point can be simplified to a contact of an infinite long cylinder with a half-plane (Fig. 8).

In the stress calculation with *ROSLCORHR* (Ref. 8) — an enhancement of *ROSLCOR* — the normal stresses and shear stresses resulting from bending (Fig. 3b) are addressed. The acting normal force that moves over the flank surface causes — at any later stage — normal stresses and shear stresses in the considered volume element (base sphere). For the material exposure, the maximum values over one load cycle have to be considered.

The stresses caused by the normal load in the sub-surface are calculated with the model shown (Fig. 8 — right). The basis for the stress mechanics is the plain strain state, meaning that deformations are only allowed in a plane vertical to the contact line. *ROSLCOR* considers the normal pressure on the surface as well as the influence of a loss-of-friction contact. The influences of thermal stresses and shear stresses on the material exposure are also addressed.

The material-physical-based calculation model evaluates for a certain volume element the risk for an initial crack. Whether the crack will result in damage of the flank or not is dependent on its potential for crack growth; especially near the flank surface, this potential is affected by the slip conditions in the contact point. As is known (Refs. 14 and 19), pitting occurs mainly in the flank area with negative slip, which is below the pitch point for pinion and wheel at bevel gears without offset. Deeper below the flank surface the influence of slip in the contact point seems, as is known thus far, negligible.

Wirth (Ref. 20) introduced a so-called "slip factor" that accounts for the difference in strength between negative and positive slip conditions. The influence of this factor is restricted to the material close to the surface. Because the material strength values are derived from the material hardness, it is allowable to (virtually) increase the hardness values appropriately. Consequently, for the same load conditions the material exposure in flank areas with positive slip is lower than in areas with negative slip. The hardness values are modified by Equations 12–14. Wirth demonstrated that reasonable results are calculated if $a/b_0 = 0.5$ and $b/b_0 = 1.0$ are chosen (b_0: half of the Hertzian contact width, Fig. 8).

Depth range $0 < y < a$:

$$HV(y) = HV_0(y)\ Z_{S1,2} \tag{12}$$

Depth range $y > b$:

$$HV(y) = HV_0(y) \tag{13}$$

Depth range $a \leq y \leq b$.

$$\text{Linear interpolation of } Z_S \tag{14}$$

where:

y is material depth below the contact point, mm
a is certain material depth, mm
$HV_{(y)}$ is modified local hardness in consideration of the slip influence, HV
$HV_0(y)$ is local hardness, HV
b is certain material depth, mm
Z_S is factor according to (Ref. 20)

Residual stresses in the tooth. Hertter demonstrated that the influence of residual stresses has to be considered in the material exposure (Fig. 6) for the evaluation of tooth failures. In particular, the maximum material exposure A_{int} is influenced by the residual stresses. Whereas compressive stresses typically have a positive effect on the material exposure, tensile stresses increase the material stresses (Ref. 8). The total dynamic exposure $A_{int\,a}$ is only influenced by means of the mean stress sensitivity. As Hertter proved, the material exposure in the range of the transition zone from case to core accounts for failure modes like flank breakage that are usually characterized by an initial crack in this region.

Wirth (Ref. 20) proposes adoption of the (compressive) residual stresses according to Lang (Ref. 11) for the case. Due to the balance of forces in the core, tensile residual stresses have to exist. For the estimation of the residual stress distribution in the core, Wirth made investigations based upon finite element (FE) methods. Using a parabola of the fourth degree, the tensile stresses can be closely approximated by the balance of forces. Figure 9 shows qualitatively in a normal section of the tooth the residual stress distribution. It is a sufficient correlation that the residual stresses in tooth height direction are equal to the residual

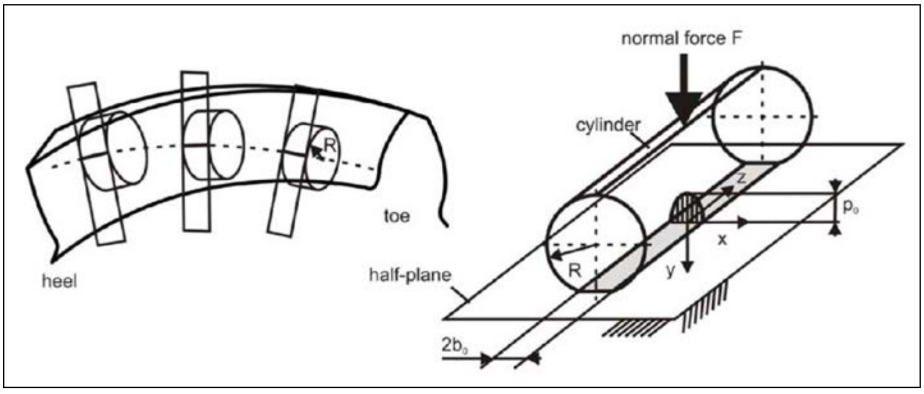

Figure 8 Simplification of complex contact conditions — cylinder model.

stresses in lengthwise direction. Residual stresses in the orthogonal direction to the flank surface may be ignored.

Improvement of a Gear Set with Flank Breakage

Wheel flank breakage. A decisive number of hypoid gear sets used in axle gear drives in test vehicles failed due to flank breakage; only the wheels were affected by this failure mode. Figure 1 (left side) shows a flank breakage on one tooth of a wheel; in Figure 1 (right side) pitting on the coast-side could be detected. Figure 2 shows on another wheel a characteristic flank breakage; as can be seen, the failure plane runs on both flank sides through the active tooth height.

To learn more about the conditions where and when flank breakages occur, a new type of test for the stationary test rig has been developed and comprehensive test runs conducted. The gear sets have been tested for a defined load spectrum where the highest load stage was the torque that has been considered in the following calculations. The gear sets failed—either by pitting or flank breakage. Pitting occurred on the pinion as well as on the wheel; flank breakage was only observed on the wheel.

Figure 10 shows for a damaged wheel the investigation of the fracture surface in the scanning electron micrograph (SEM). In this case a small inclusion was detected from where the crack propagates to the flank surfaces. Inclusions can be regarded as a catalyst for the crack initiation because of the notching effect of different elasticity moduli. Investigations by Annast (Ref. 1) showed that an Al_2O_3 inclusion causes a stress increase (von Mises criterion) of approximately 30%–40%; the size and the depth beyond the surface have a relatively small influence. Therefore, the lower the material exposure in the core, the smaller the risk of flank breakage with initial cracks in this region will be (Fig. 10).

Design of an improved gear set. The aim of the re-design was to develop a new gear design with a smaller material exposure to avoid flank breakage on one hand and pitting as far as possible on the other. In a first step the old design was analyzed with the newly introduced material-physical calculation method. In the second step a new gear design with same ratio and diameters but lower material exposure was searched by an iterative process. Table 1 contains the main geometry data of the old and new gear design.

For the calculation, discrete contact points on the flank are chosen for the evaluation. Figure 11 shows that the selected contact points are positioned in a section with considerably high load and the suspected crack origin. To evaluate not only the risk of an initial crack at one single point, but also the potential of crack growth, four different positions were examined.

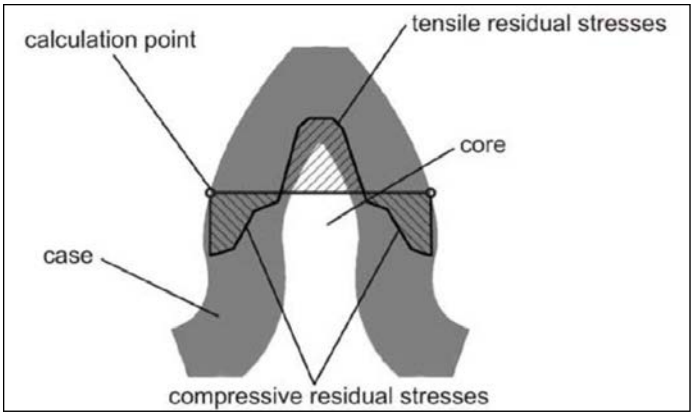

Figure 9 **Residual stress distribution in the tooth.**

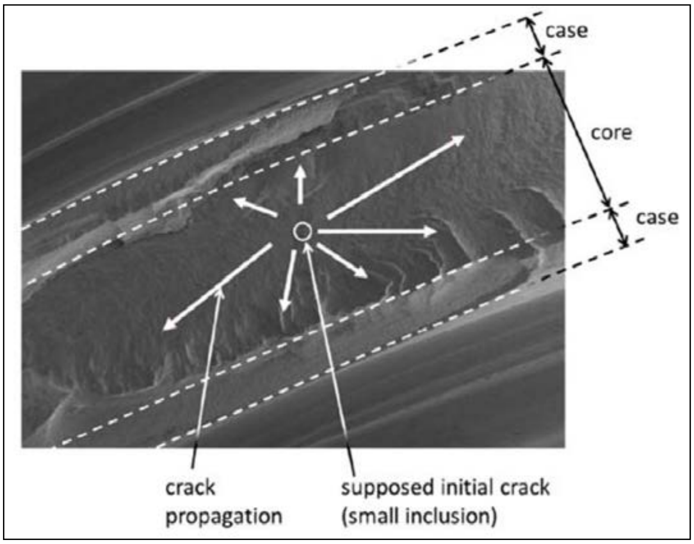

Figure 10 **Flank breakage—two different wheels.**

Table 1 **Geometry of the examined gear sets**

Nomenclature	Symbol	Unit	Old design		New design	
			Pinion	Wheel	Pinion	Wheel
Number of teeth	z		8	45	8	45
Pinion offset	a	mm	34		34	
Normal module	m_{mn}	mm	6.134		6.122	
Mean pitch diameter	d_m	mm	69.6	331.2	69.9	332.3
Face width	b	mm	60.7	58.3	62.2	59.0
Spiral angle	β	°	45.5	34	45.5	34
Material			25MoCr4E		25MoCr4E	
Roughness Rz flank/tooth root	Rz	mm	3/16 (after run in)			
Total overlap ratio (under load) drive/coast			3.0/2.7		2.92/2.7	
Lubricant			Shell Spirax ASX 75W 90			
Temperature of lubricant	θ	°C	90			

NOTE: Because only the wheel was affected by flank breakage, all calculations have been made for the wheel only!

Table 2 contains the Hertzian stresses that were determined by means of the loaded tooth contact analysis with *BECAL* (Ref. 10). Deformation and deflections of housing, shafts and bearings have been considered. Under the same load conditions it was possible to reduce the stresses on both flank sides in the critical area of the flank by approximately 15%. This was possible with an optimized crowning ("ease-off") in combination with a different gear design (duplex instead of semi-completing) and changed pressure angles.

Of course the reduction of contact stresses leads in most cases to an increase in load capacity—especially when the failure mode pitting is addressed. But in the case of flank breakage the failure mechanism is influenced by not only the contact stresses, but also by the material exposure deep inside the tooth. Because of the requirement to keep the amount of transferred torque by retaining the gear dimensions (and module), the total flank load cannot be significantly reduced. To avoid flank breakage, the goal must be to reduce material exposure—mainly in the core—where, in this case, the crack initiation could be detected in several cases (Fig. 10).

As mentioned earlier in the evaluation of the material exposure in the sub-surface section, especially in the middle of the tooth thickness, the following stress components must not be ignored:

- Shear stresses due to shearing forces (flank normal forces)
- Tensile residual stresses

The shear stress distribution reaches its maximum in the middle of the tooth thickness (Fig. 3b). Also, the maximum values of the tensile residual stresses are supposed to be in this region. Whereas the determination of the shear stresses is strictly a mechanical problem, the residual stresses are caused mainly by the heat treatment process. Only in the area directly beneath the surface are residual stresses influenced by the finishing process of the gear. As such, the residual stresses are derived by the hardness profile described earlier.

Figure 12 shows that, for the following calculations results, the assumed hardness profiles are based on detailed measurements, yet smoothed for calculation. Because of the slightly different cooling conditions during the hardening process in profile direction of the tooth, the hardness gradients and core hardness are slightly different. The derived residual stress distributions are shown as well (Fig. 12). It can be seen that the compressive stresses in the case are up to $\sigma_{res} \approx 400\,\mathrm{N/mm^2}$ and are decreasing up to the transient region of case and core. Because the case thickness in profile direction is more or less constant, the compressive residual stress profiles are similar. In contrast to that, the tensile residual stresses in the middle of the tooth thickness are increasing from P4 to P1. The reason is the mechanical balance of forces; i.e., the separating forces that are caused by the compressive stresses in the case are approximately constant for P1 through P4. The attracting force is represented by the tensile compressive stresses and requires having an equal amount. Because the core section becomes smaller—P4 through P1—the corresponding tensile stresses must increase.

Figure 13 reveals the calculated material exposure for P1 through P4; the black lines represent the total dynamic exposure $A_{int\,a}$; the grey lines represent the total maximum exposure A_{int}.

$A_{int\,a}$ can be seen as a value to describe the material fatigue; it is based on an endurance strength (derived from the Vickers hardness) for a failure probability of 50%. Pitting is a typical fatigue failure that correlates with the total dynamic exposure $A_{int\,a}$. (Ref. 8); (Refs. 18

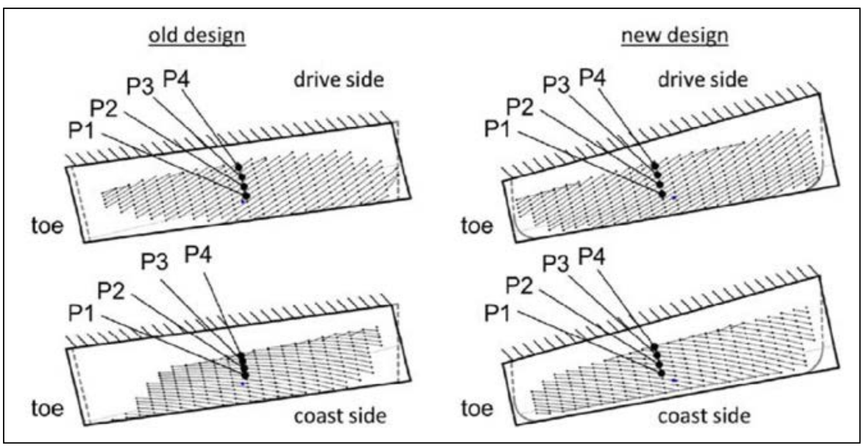

Figure 11 Contact pattern and calculated contact points on the wheel flanks.

Table 2 Hertzian stress under the considered load calculation

Calculation point	Old design		New design	
	Drive side	Coast side	Drive side	Coast side
P1	1,865	1,951	1,629 (-13%)	1,692 (-13%)
P2	1,891	1,979	1,571 (-17%)	1,698 (-14%)
P3	1,841	1,961	1,593 (-13%)	1,651 (-16%)
P4	1,612	1,901	1,553 (-4%)	1,585 (-17%)

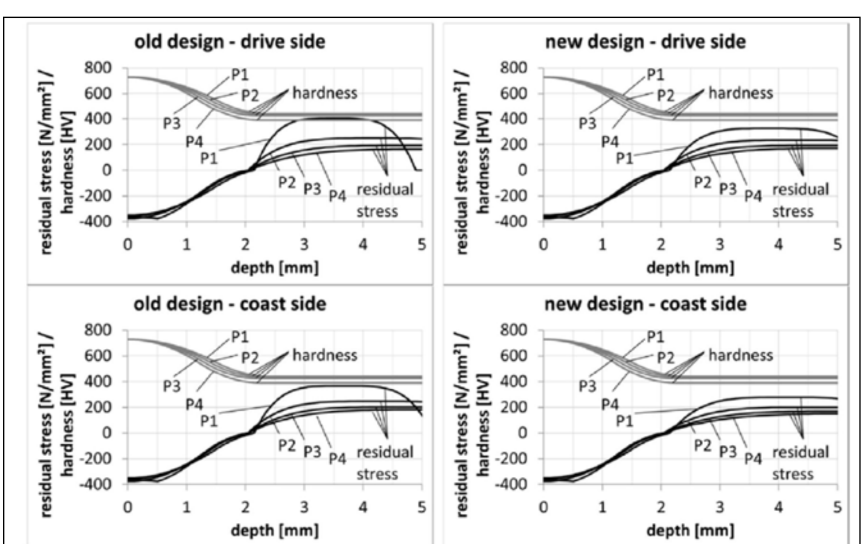

Figure 12 Profiles for hardness and residual stress.

and 20) show that if material exposure values exceed a certain limit in the subsurface — up to a depth of $y/b_0 \approx 1$ (b_0: half of Hertzian contact width) — pitting failure occurs with a high probability. For the examined gear this decisive range is up to approximately $y \approx 1$ mm. As Figure 13 shows, $A_{int\,a}$ exceeds the limit of 1. Due to the load spectrum of the test vehicles, which had only a few time slices with this considered load, pitting failures on the drive-side where not detected. The fatigue strength may be the reason for that.

A_{int} represents the material exposure concerning yielding. According to the theory of the calculation method — $A_{int} > 1$ — local redistribution of stress or initial cracks occurs. This situation is exacerbated if the notching effect of inclusions or incongruities increases the material exposure. At this time there is no possibility provided by the material-physical-based method to address this fact in the calculation process. Therefore the practical limit for A_{int} values that are determined for a homogenous material should be reduced to values smaller than 1 in order to be on the safe side. Because of the specifics of the hardening process, incongruities occur more typically in the core than in the case; this is why the total maximum exposure A_{int} should be limited — especially in the core.

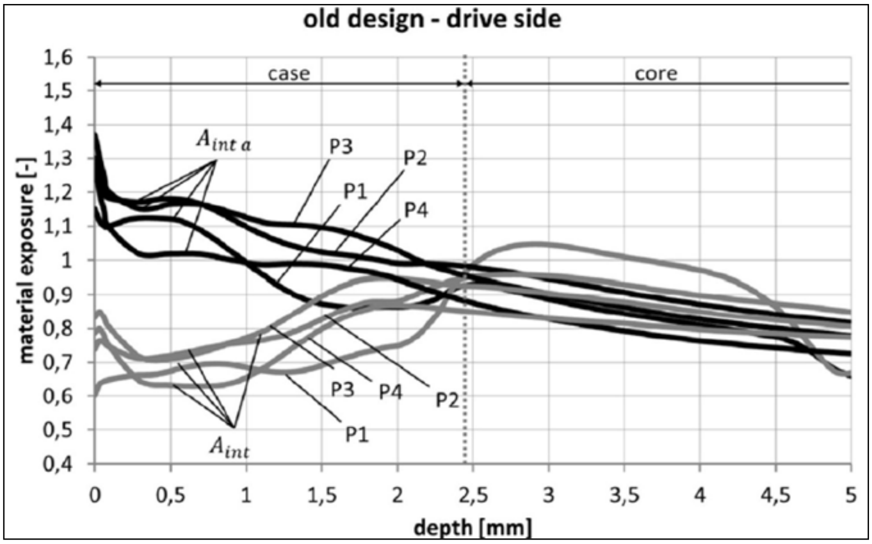

Figure 13 **Material exposure for calculated points: old design — drive-side of wheel.**

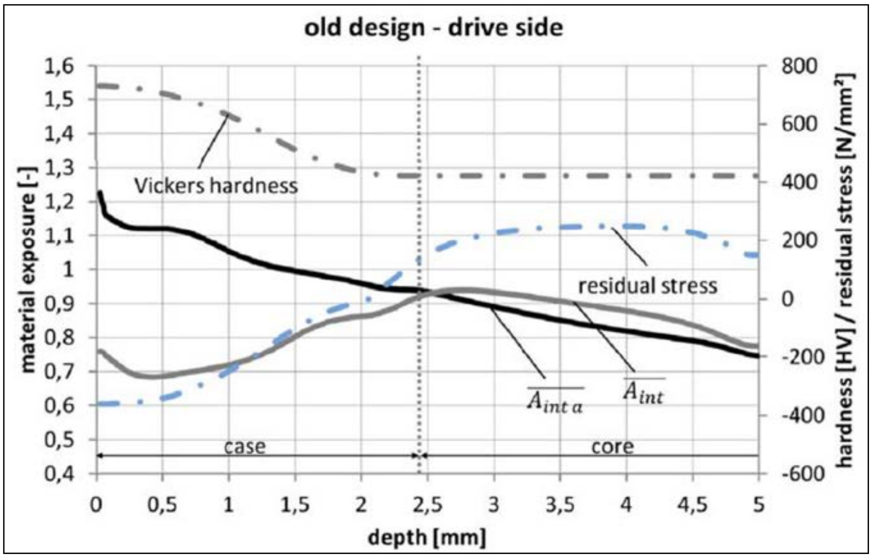

Figure 14 **Mean values of material exposure (crack growth potential): old design — drive-side of wheel.**

Hertter (Ref. 8) and Wirth (Ref. 20) found good correlation between the total maximum material exposure A_{int} and the failure mode flank breakage. Especially high values in the material depth between the transition of case and core — as well as in the core — seem to cause flank breakage. It must be pointed out that crack initiations caused by yielding have no endurance limit or fatigue strength for finite life. According to theory, only a very few single-load cycles are enough for stress redistribution or crack initiation. These cracks may also have the ability to grow at lower loads. Unlike for the calculation against fatigue where there is a high strength for finite life, it is already critical if an initial crack occurs during a momentary high load. In other words, it is more important to reduce high A_{int}- values (concerning yielding) than the $A_{int\,a}$ values (concerning fatigue) if the gear set is stressed by a load spectrum with only momentary high loads.

Figure 13 shows that A_{int} of P1 exceeds the limit 1 in the depth between $y = 2.5$–3.7 mm. P2 causes values $A_{int} > 0.9$ in a range between $y = 2.2$–4 mm and P3 for $y = 1.5$–3.2 mm. Only the A_{int} graph for P4 is constantly under 0.9. These high values of A_{int} over a very large section of the tooth correlate well with the witnessed flank breakages.

The first evidence of flank breakage does not necessarily appear in a single volume element. Local peaks of material exposure may be reduced after a yielding process and no growing crack is initiated, meaning that the failure mode of flank breakage will not occur. Therefore it is not only decisive for flank breakage if in one contact point the limit for yielding is reached; in fact, high values of A_{int} in adjacent contact points support crack growth. In order to evaluate this potential a mean value of the total maximum exposure is defined (Eq. 15). For the four considered contact points the average value of each A_{int} graph is determined in a certain depth. Of course, based on this consideration, determination of the $A_{int\,a}$ mean value is also useful in determining an idea of the pitting danger over the considered flank area.

$$\overline{A}(y) = \frac{1}{i}\sum_{1}^{i} A_i(y) \tag{15}$$

where:

$\overline{A}$ is mean value of total dynamic exposure $A_{int\,a}$ or mean value of the total maximum exposure A_{int}
y is material depth, mm
i is amount of considered calculation points

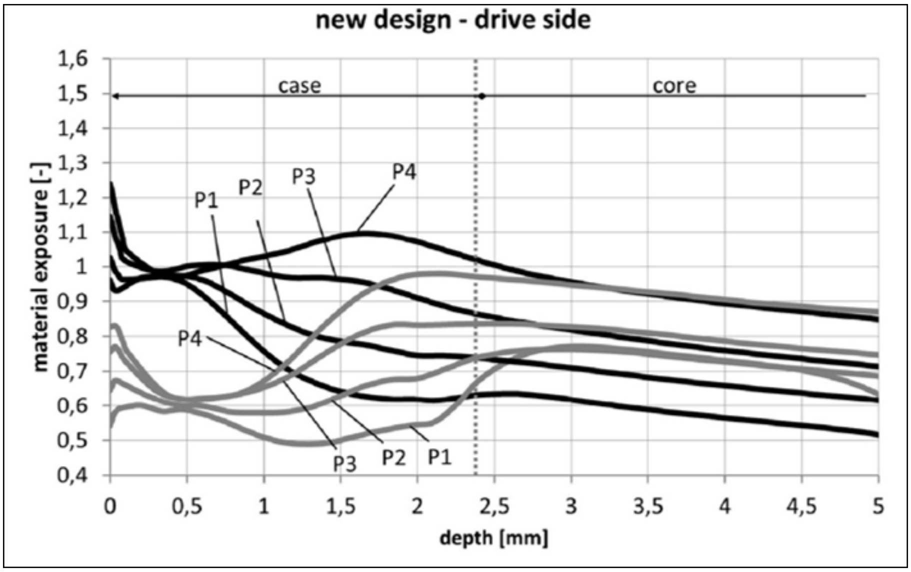

Figure 15 Material exposure for calculated contact points: new design — drive-side of wheel.

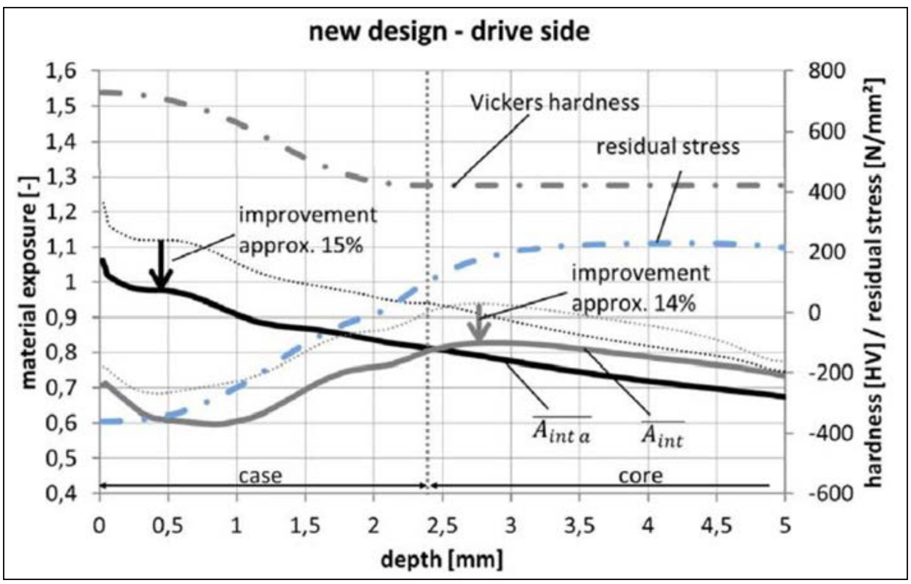

Figure 16 Mean values of material exposure (crack growth potential): new design — drive-side of wheel.

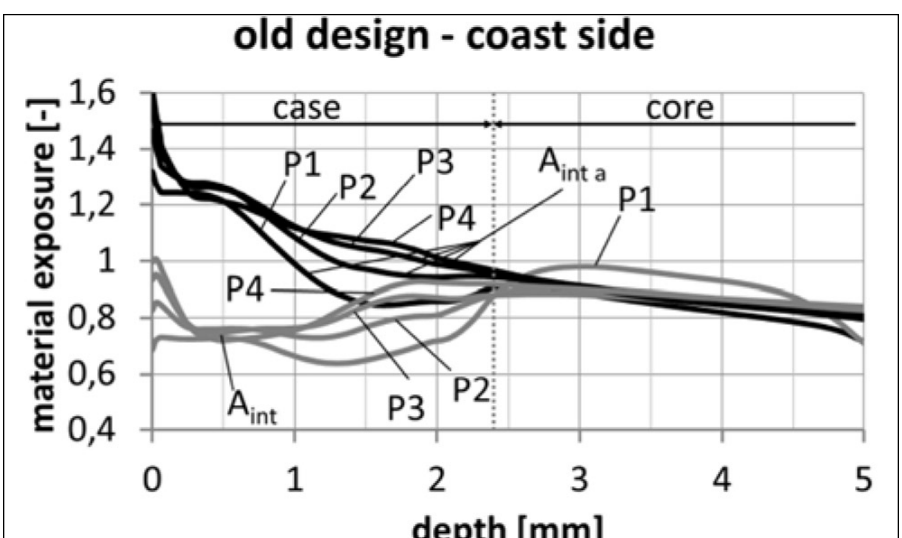

Figure 17 Material exposure for calculated contact points: old design — coast-side of wheel.

A is total dynamic exposure $A_{int\,a}$ or total maximum exposure A_{int}.

In Figure 14 the graphs of the mean values $A_{int\,a}$ and A_{int} are shown for the drive-side of the wheel flank. It is obvious that in the depth $y < 1$ mm, $A_{int\,a}$ has considerably high values. As mentioned, this is an explanation for the observed pitting in the field. But more important for flank breakage is, as described before, the profile of A_{int}. It can be seen (Fig. 14) that in the depth $y = 2.3–3.7$ mm, the exposure is $A_{int} \geq 0.9$. Together with the mentioned influence of inclusions like those detected (Fig. 10), it is obvious that there is a high risk of flank breakage under these load conditions. The profiles for hardness and residual stress shown (Fig. 14) are also principally derived by Equation 15 and so can be interpreted as the mean values.

Figure 15 shows for the new design the corresponding calculations for points P1–P4. Compared to Figure 13 it is obvious that in the close region to the surface ($y < 1$ mm) the total dynamic exposure $A_{int\,a}$ can be reduced. But more critical to the failure mode flank breakage is the reduction of the total maximum exposure A_{int} in the sub-surface of the tooth; the exposure profiles of all considered points do not exceed the theoretical limit of 1. Further, the maxima of A_{int} for the points P1 and P2 are significantly lower.

In Figure 16 the mean values according to Equation 15 are shown for the drive-side of the new design; as mentioned, their values represent the potential for crack growth. It can be seen that in the relevant depth for pitting ($y < 1.0$ mm), there is a significant reduction of the total dynamic exposure $A_{int\,a}$. For better illustration, the profile of the old design is shown by a dotted line; improvement for pitting can be estimated at approximately 15%. Again, the mean values show also the reduced crack growth potential in the inner tooth. In the transition from case to core the values for the total maximum exposure A_{int} are lowered by approximately 14%.

Finally, it can be said that, for the drive-side flanks of the new design, the exposure profiles $A_{int\,a}$ and A_{int} representing the risk of crack initiation are significantly lower, as are the mean values $A_{int\,a}$ and A_{int} that can be regarded as the potential for crack growth.

Indeed, it is not sufficient to optimize only the drive-side flanks. The specific load spectra in the practical field show that the coast-side flanks are considerably high-loaded; thus the coast-side is analyzed using the same load as the drive-side.

Figures 17 and 18 show a comparison of the exposure profiles for the old and new gear designs' coast-side flanks. As on the drive-side, the values of $A_{int\,a}$ in the decisive depth for pitting could be significantly reduced. But also in the decisive depth for flank breakage, where the initial crack is suspected (Fig. 10), the maximum exposure A_{int} could be lowered in every considered calculation point.

The old-design, mean values of the exposure profiles are shown (Fig. 19). As can be seen, the $A_{int\,a}$ level regarding pitting ($y < 1.0$mm) is even higher than for the drive-side. This correlates well with the pitting that was observed on several gears of the test vehicles (Fig. 1). Regarding the crack growth potential for flank breakage, the mean values for the exposure A_{int} are similar to those on the drive-side. This means that initial cracks close to the middle of the tooth thickness can grow for drive-side as well as for coast-side loading.

Figure 18 proves that the decisive $A_{int\,a}$ values for pitting could be clearly decreased, and is confirmed by the mean values $A_{int\,a}$ (Fig. 20). Further, the risk of crack initiation in the subsurface $y > 1.5$ mm is lowered for every considered point (Fig. 18); this obviously leads to an improved situation for crack growth potential. As shown (Fig. 20), the mean values A_{int} could be reduced by approximately 11% with the new design.

Because of the positive prediction in load capacity — especially concerning flank breakage and pitting — prototype gear sets based on this new design have been produced and analyzed in the test rig. The gears were loaded by the same load spectrum. The main goal — avoiding flank breakage — was attained. What's more, pitting on wheel flanks did not occur during the tests. Though the pinion is now failing via pitting, the run time could be stretched by approximately a factor of four.

Summary

- Flank breakage occurred on the wheel of several bevel gear sets used in test vehicles of MTB. In some cases the damage was accompanied by pitting on several wheel flanks. Tests on a MTB test rig confirm that this type of failure is reproducible for a certain load spectrum.

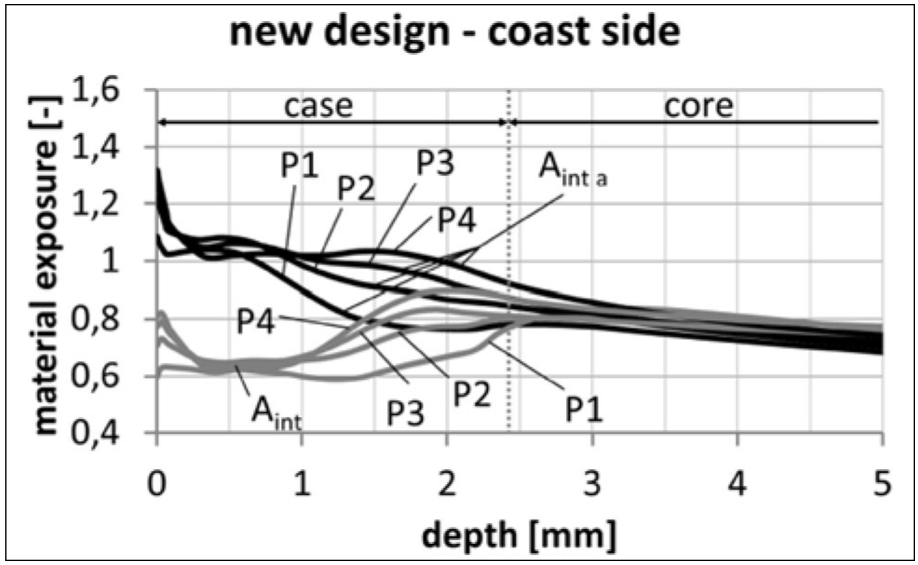

Figure 18 Material exposure for calculated contact points: new design — coast-side of wheel.

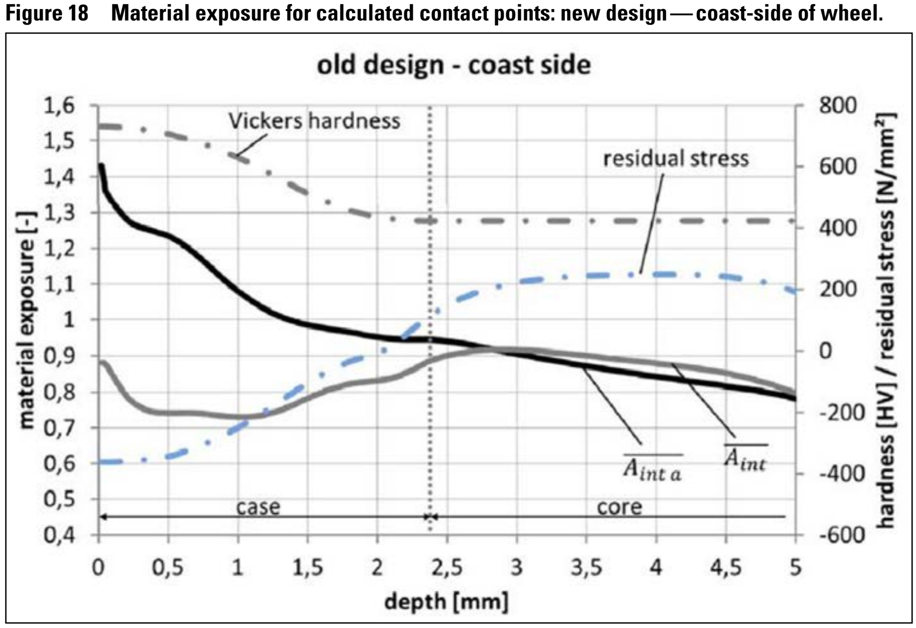

Figure 19 Mean values of material exposure (crack growth potential): old design — coast-side of wheel.

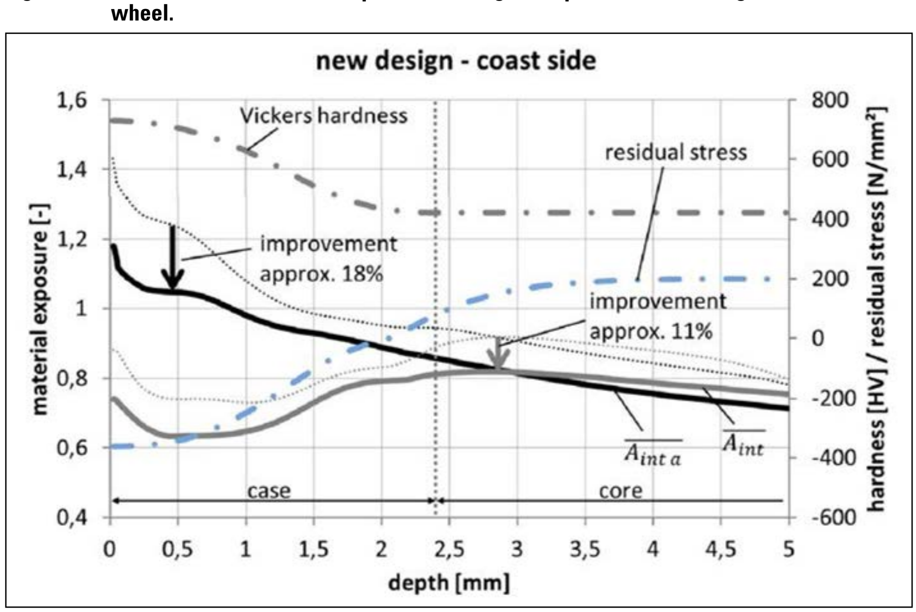

Figure 20 Mean values of material exposure (crack growth potential): new design — coast-side of wheel.

- The goal of a new gear design was to avoid flank breakage in all circumstances and to increase pitting-related load capacity. Because it was essential to retain the outer wheel diameter, the pinion offset and gear ratio — mostly the micro-geometry of the corresponding flanks (ease-off) — were modified. To reduce the amount of test gears needed during the re-design process, a new calculation model was used to evaluate the risk for flank breakage and pitting on the basis of a loaded TCA.
- The new calculation model was introduced by Hertter (Ref. 8) for cylindrical gears — and expanded by Wirth (Ref. 20) — taking into account the specifics of bevel and hypoid gears. The new method is based on material-physical relationships and allows for the evaluation of material exposure in the sub-surface. The influences of local hardness (via hardness profile) and residual stresses can be explored. As demonstrated — (Refs. 18, 8 and 20) — the material exposure in the depth closest to the surface is decisive for pitting failure. On the other hand, extensive material exposure in the transient region from case to core, or in the middle of the tooth thickness, seems to affect flank breakage.
- With this new model, the risk of pitting and flank breakage failures can be better predicted. But while pitting resistance can be predicted with good accuracy, a consistently successful calculation of the load capacity regarding flank breakage is virtually impossible. Uncertainties in the calculation model include the influences of inclusions and/or material discontinuities on the material exposure. Furthermore, the tensile residual stresses that have a decisive influence are estimated by reliance on a simple model. However, FE analysis could improve the prediction quality of the model. Ultimately, the comparison of the load capacity of two gear designs produced in a similar way is reliably doable.
- In an iterative process, new gear designs have been analyzed with the material physical calculation method. The result is a new gear set with significantly lower risk of failing due to pitting or flank breakage; testing with first-prototype gear sets confirmed the improvement. Neither pitting nor flank breakage occurred during the previous test runs.

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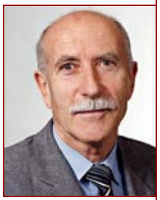
Dr. Christian Wirth has since 2011 served as managing director of Zahnräder und Getriebe GmbH. Previously, beginning in 2008, he was research group leader at the Gear Research Laboratory of the Technical University of Munich (FZG) before being named technical manager at Zahnräder und Getriebe. From 2002 until 2008 Wirth was a scientific assistant at Technical University of Munich (FZG), where he received his mechanical engineering and PhD degrees (thesis: the load capacity of bevel and hypoid gears).



Bernd-Robert Höhn studied mechanical engineering at the Technical University Darmstadt (1965-1970) and served as an assistant lecturer (1970-1973) at the Institute for Machine Elements and Gears at the Technical University Darmstadt prior to becoming an assistant professor at the university (1973-1979); in 1978, he received his PhD (Dr. Ing.) in mechanical engineering. In early April, 1979 Höhn worked as a technical designer in the department for gear development of the Audi, and by 1982 was head of the department for gear research and design for the automaker. In 1986 Audi named Höhn department head for both gear research and testing of automotive transmissions, until his departure in 1989 to become head of both the Institute of Machine Elements at the Technical University and of the Gear Research Centre (FZG). Höhn has served since 2004 as vice president for VDI for research and development and since 1996 has led the working group 6 and 15 for ISO TC 60—calculation of gears.

Dr. Christo Braykoff received his degree in mechanical engineering from the Technical University of Sofia and the Technical University of Karlsruhe in 2001. From 2001–2007, he worked at the Gear Research Centre (FZG) and obtained his PhD on the topic, Load-Carrying Capacity of Fine-Module Gears. He has worked since 2007 at MAN Truck & Bus AG as a design engineer and is responsible for the development of driven rear axles and transfer cases of trucks and buses. Dr. Braykoff is an active member of the FVA working groups "Bevel Gears" and "Spur Gears" and has since 2008 been a member of the FVA scientific advisory board as MAN representative.

Tooth Root Optimization of Powder Metal Gears: Reducing Stress from Bending and Transient Loads

Dr. Anders Flodin and Dr. Michael Andersson

This paper will provide examples of stress levels from conventional root design using a hob and stress levels using an optimized root design that is now possible with PM manufacturing. The paper will also investigate how PM can reduce stresses in the root from transient loads generated by abusive driving.

Introduction

By 2020, some 100 million cars with geared transmissions are estimated to be manufactured yearly. If the amount of transmission gears is estimated at 6 kg per vehicle, the total mass for gears will be 600 million kilograms. This amount represents a huge opportunity for powder metal (PM) as a replacement technology with improved system performance. The manufacturing technology inherent to PM offers design freedom and performance advantages not available with wrought machined steel. But introducing the PM-specific geometrical features using traditional machining is very costly, and sometimes impossible on a mass-production scale.

The automotive gear industry is very conservative and often unaware of what can be achieved with PM technology, so this paper is aimed at just one part of the gear — the root — where PM gear design and manufacture can reduce stress levels by introducing design features difficult to obtain using traditional gear manufacturing technologies.

Traditional Gear Hobbing and its Limitations of Root Geometry

The root of the gear — when hobbed — often goes unspecified in the gear drawing; it is indirectly given in the tool drawing and data. The root is a function of the trochoid movements of the hob flutes, gear rotation, and the geometry of the tip of the hob. There are also limitations as to what hob radius might possibly be used, that is given by Equation 1 (Ref. 1):

$$r_{c(max)} = \frac{0.785398\cos\Phi - b\sin\Phi}{1-\sin\Phi} \tag{1}$$

where:

r_c is the hob tip radius
b is the dedendum constant
Φ is the pressure angle

In an effort to quiet the gear mesh, the teeth tend to become more slender and the pressure angle decreases. In order to hob such a gear, short-pitched hobs are used. Figure 1 shows two gear spaces cut with different pressure angles on hobs. The root on the left gear is more favorable from a stress point of view than the root cut on the right gear, where a short-pitched hob has been used. The short-pitched hob has longer service life and is sometimes necessary for cutting gears with smaller pressure angles.

Figure 2 displays the change in root stress for a gear cut with 20° (left) and 11° (right) pressure angle hobs. The peak stress for the gear cut with a 20° pressure angle is 352 MPa versus 405 MPa for the gear cut with an 11° hob.

PM gear technology does not suffer from these limitations in root shape, and the root can therefore be actively designed in coordination with the tool manufacturer.

Root Geometry of Powder Metal Gears

Since no hobbing action is required when making powder metal gears, some of the limitations as well as a number of ISO recommendations regarding root shape can be ignored, and a more

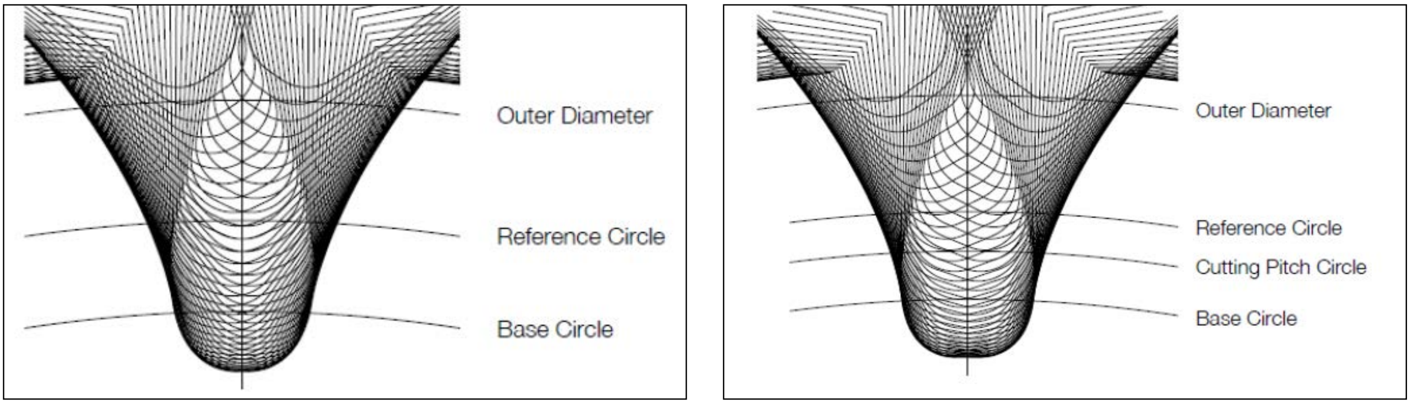


Figure 1 **Left: Tooth space cut with 20° pressure angle on hob; Right: Tooth space cut with 11° pressure angle on hob.**

Presented at World PM 2012 in Yokohama, Japan, October 18, 2012

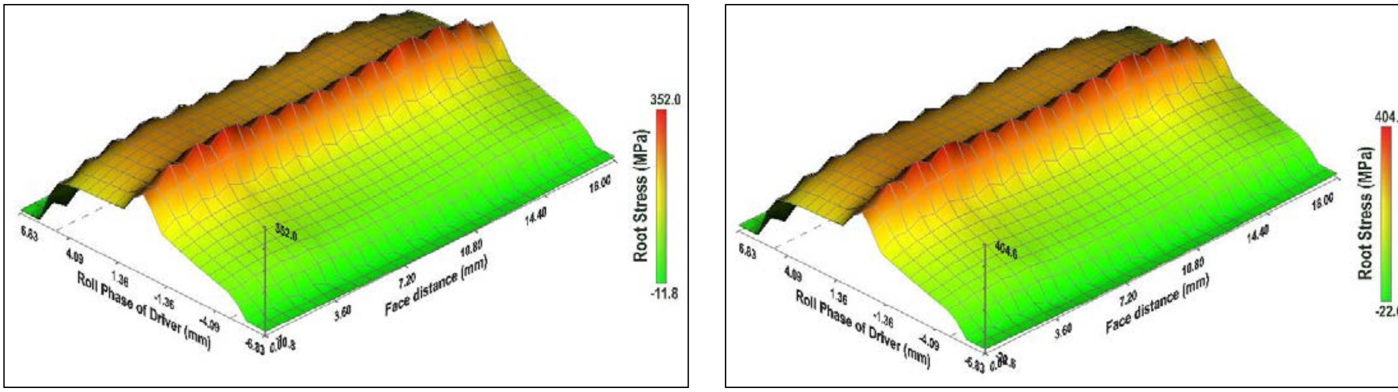


Figure 2 Right: Root stress for gear cut with 20° pressure angle hob; Left: Root stress for gear cut with 11° pressure angle hob (Image courtesy Dontyne Systems).

active design philosophy adopted. In Figure 3 there are 4 different roots depicted — all based on the FZG gear geometry. They are:

- Original 1.99 mm (given by hob tip radius $\rho a_0 = 0.8$ mm)
- Full radius
- Optimized curve shape
- Asymmetric gear tooth

The original 1.99 mm root design is one adopted for tooth root bending fatigue testing internally at Höganäs AB. The first modification is to introduce a large root radius to decrease tooth root stresses. This is done using *KISSsoft's* suggestion for the largest possible root radius that may be manufactured with a hob. The final optimization step is done by hand, applying a progressive transition using a spline in *Pro Engineer*. The result is similar to what is presented by Kapelevich (Ref. 2), with the difference that the latter has an analytical optimization routine. Sanders (Ref. 3) and several others have also investigated this, but since the favorable geometry calculated has been impossible to hob in mass production, it has not gained widespread use. What is of interest here is the manufacturability inherent in PM that allows for efficient mass production of an optimized root.

To investigate the influences of the modifications, both static and dynamic finite element analyses are performed using *Calculix*. For tooth roots the critical stress is very often found to be caused by impacts and abuse, which begs the question of whether the root — when modified for lower static stress — will show a reduction in dynamic stress and how it scales.

Finite Element Models

Figure 4 shows the model for the calculations in which a section of the gear — consisting of five teeth — is cut. On the middle tooth a force is applied on the tip, tangent to the base circle. On the inner hub and cut boundaries, displacements are locked. The calculations are performed under plane strain conditions, with second-order triangular elements (Fig. 5).

Calculations are done with an elastic material model; corresponding material properties are given in Table 1.

Table 1 Material properties for the simulations

Property
Young's modulus, E=160 GPa
Poisson's ratio, ν=0.28
Density, ρ=7.30 g/cm3

In the static calculations, a force is applied (Fig. 4) and stresses are calculated. The force is chosen, arbitrarily, to give a peak stress of 500 MPa for the original gear design. This corresponds to $F = 7.128$ kN; the same force is used for the dynamic calculations. Since the models are linear, all results will scale proportionally with the force. In the dynamic calculation the force is applied as a step corresponding to a sudden impact. The speed of the wave propagation in the gear compared to a typical rotational speed justifies the constant load applied to the tip. This will be discussed further below. The time step in the dynamic calculations is $\Delta t = 0.1\ \mu s$. The asymmetric gear uses a different base radius, so for that gear the loading force has been increased so that the transmitted torque is equivalent to the other gears.

Results

The results from the calculations (Table 2; Fig. 4) demonstrate that an optimized root can reduce even the most optimized machined root by another 5 percent. The authors have inves-

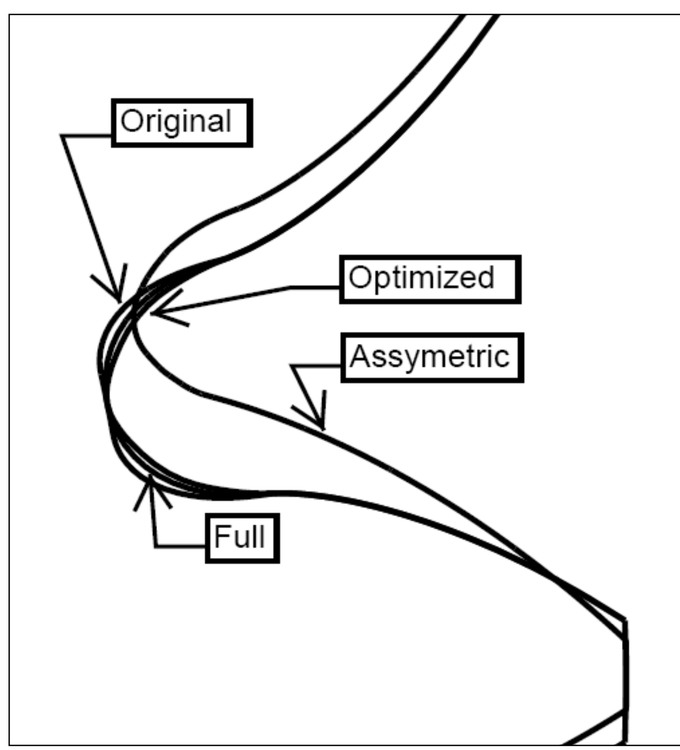


Figure 3 Root geometries compared.

Table 2 Peak stress in tooth root

Geometry	Static stress [MPa]	Difference	Dynamic stress [MPa]	Difference	Dynamic amplification
Original	500	-	1010	-	2.02
Max radius	434	-13%	865	-14%	1.99
Optimized	408	-18%	792	-22%	1.94
Asymmetric	405	-19%	719	-29%	1.78

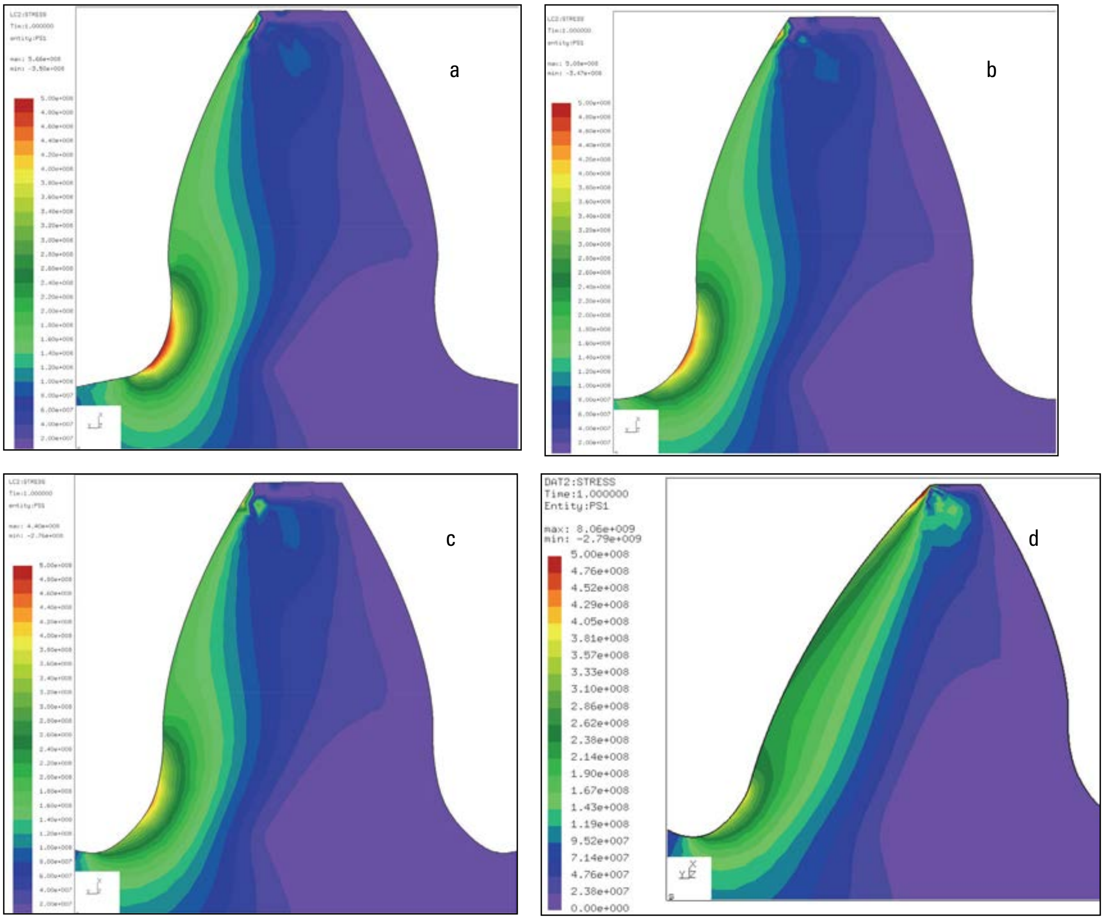


Figure 4 Tooth root stresses/static analysis: a) original; b) maximum radius; c) optimized; and d) asymmetric. (Note: same stress scale for all geometries.)

tigated several other in-production root geometries and found 5–25 percent improvements in root stress reduction. Also, the compression side of the symmetric gears is improved to the same extent as the tension side.

The asymmetric root shows the highest stress reduction; this is not typical for this type of gear teeth but may occasionally be the result. There is always an accuracy error and simplifications made in the model — such as omitting residual stresses and surface roughness — that have to be taken into account when drawing conclusions from the results.

The true benefit of the asymmetric tooth is the *contact stress reduction*, but one has to be careful and balance root stress and contact stress on the coast side if the gear is being used in, for example, an automotive transmission.

The reason that the dynamic amplification factor is improved with reduced stress

Table 3 Gear data

GEAR		OPTIMIZED	MATING
NUMBER OF TEETH		16	24
MODULE		4.50	4.50
PRESSURE ANGLE		20°	20°
PITCH DIAMETER (PD)		72	108
BASE DIAMETER		67.6579	101.4868
OUTER DIAMETER		82.46	118.36
ROOT DIAMETER	Current	61.34	98.98
	Optimized	61.11	-
TOOTH TIP RADIUS		0.06	0.06
TOOTH THICKNESS AT PD		7.664	7.630
FACE WIDTH		14	14
MATING GEAR NUMBER OF TEETH		24	16
CENTER DISTANCE		91.500	

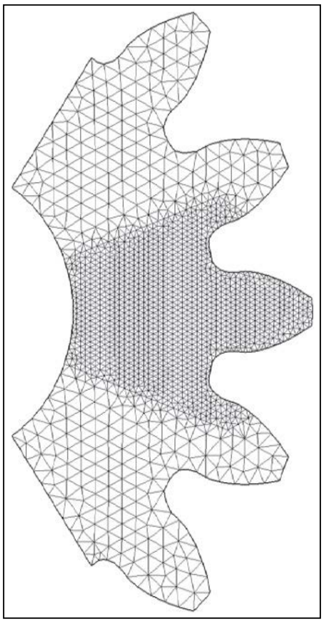


Figure 5 Mesh in FE model.

levels at the root can be found in the dynamics of the gear tooth itself. The Eigen frequency of the gear tooth is elevated when bending stiffness increases, resulting in less sensitivity towards sudden loads.

Conclusions

It has been demonstrated that it is possible to reduce root stress in a gear tooth by replacing the cut trochoid root shape with a curve shape defined by a spline that is designed, iteratively, to reduce root stress. It is also possible to manufacture a gear wheel with this root shape in mass production using PM manufacturing technology. It may not be possible to manufacture this root shape using conventional cutting technology at the same speed as is possible with PM technology.

It was also demonstrated that the dynamic stress levels benefit significantly from a non-trochoid root, and that the amplification factor is reduced as a result of increased tooth stiffness.

Asymmetric gear teeth designed for reduced contact pressure may be designed in such a way that their root stress is reduced to levels where they can operate at similar stress levels as cut gears. Their increased stiffness will improve their dynamic properties.

Discussion

By employing a more active design of the root in particular, and the gear shape in general, a PM transmission gear can be subjected to lower stress than a conventionally cut gear. This can be achieved without sacrificing productivity, provided that powder metal manufacturing technology is used. As a matter of fact, PM is normally the more cost-efficient and significantly less resource-demanding alternative to conventional gear manufacturing.

The next step will be to utilize the PM design advantage in the rebuild of an automotive six-speed manual transmission. The possibility to reduce the stresses in the root will enable a different modulus to be used in the design; i.e., reducing contact pressure by increasing contact ratio — and thus expanding the range of gears that PM can replace.

Where contact stress cannot be kept within the allowable stress levels, asymmetric gearing will be introduced. The challenge will be to balance stresses and NVH when driving on the coast-side of the gear.

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Also, the effects tooth stiffness have on dynamic stress amplification can be utilized in the design of other non-involute gear types such as Convoloid gear teeth, asymmetric gear teeth or internal gear teeth.

With these techniques, a PM substitution of the solid steel gears in this particular automotive gearbox will be possible. This work will be presented in another article.

Acknowledgement. *The authors wish to thank* Dr. Alex Kapelevich *for his valuable input on the root optimization and asymmetric gear tooth design. His contributions include his asymmetric gear tooth by Direct Gear Design methodology and corresponding stress calculations. He has also analytically optimized the gear root using this method. The authors used these results to create a similar root design that produced the same stress results that are presented in this paper (Ed.'s Note: Kapelevich's new book,* Direct Gear Design, published by CRC Press, *is now available for online purchase at both CRC Press ($129.95) and Amazon.com ($117.58).*

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Correction

The technical paper in the March/April issue, "Optimization of a Process Chain for Gear Shaft Manufacturing," by Fritz Klocke, Markus Brumm, Bastian Nau and Arne Stuckenberg, was incorrectly credited as being an American Gear Manufacturers Association (AGMA) paper. The work is in fact owned solely by the authors.

How to Design and Install Bevel Gears for Optimum Performance: Lessons Learned

Stephen Marsh

Introduction

Bevel gears must be assembled in a specific way to ensure smooth running and optimum load distribution between gears. While it is certainly true that the "setting" or "laying out" of a pair of bevel gears is more complicated than laying out a pair of spur gears, it is also true that following the correct procedure can make the task much easier. You cannot install bevel gears in the same manner as spur and helical gears and expect them to behave and perform as well; to optimize the performance of any two bevel gears, the gears must be positioned together so that they run smoothly without binding and/or excessive backlash.

Bevel gears can include straight, spiral, Zerol, hypoid and Spiroid (to address the differences between each one is beyond the scope of this guide). Because these types of bevel gears are basically conical in shape, they all have an optimum position for best performance. Usually the manufacturer of the bevel gear determines this optimum position by running tests of individual gear sets. However, the *gearbox designer* and *assembly technician* share responsibility for incorporating this optimum positioning in the gearbox. The designer must provide shimming dimensions that are easy to measure in order to aid the fitter/technician in assembling the overall gearbox. The aim of this guide is to provide instruction on how best to accomplish this.

Part I: Key Factors

Various parameters contribute to proper gearbox assembly that help ensure smooth and efficient operation; the two most important criteria are:
1. Mounting distance
2. Backlash

Mounting distance. The distance from a locating surface on the back of one gear (most commonly a bearing seat) to the center-line of a mating gear is the mounting distance (Fig. 1). In some cases, for convenience, a front surface may be used for the assembly. This is the most important parameter for ensuring correct and optimal operation.

Typically, the pitch cone apex-to-crown-value is a fixed, nominal value determined by a formula. The dimension between the mounting surface (in Figure 2 it is the bearing seat) and the crown point will have a tolerance; it is common to see a tolerance of ±0.05 mm up to ±0.1 mm. Obviously the mounting distance is the addition of the two values and will therefore also have a tolerance.

For straight bevel gears, the pitch cone apex is the intersection point of the pinion and gear axes (Fig. 2).

The pitch cone apex-to-crown-value is provided in engineering handbooks and in publications provided by Gleason Corp., for example. Gleason publications include most of the information required by the gear designer to select a satisfactory pair of bevel gears. The pitch cone apex-to-crown value is determined by a formula that is a function of the pitch diameter, pitch angle and addendum; therefore, the pitch cone apex-to-crown-value cannot be chosen arbitrarily. However, the mounting distance is determined by the designer as long as the pitch cone apex-to-crown-value is maintained. (Of course, in reality this mounting distance will have to have some degree of tolerance.) It may be possible to manufacture all gears to the nominal mounting distance specified on the drawing, but the additional cost to do so is usually not warranted.

If two gears are being machined as a set, the manufacturer can establish the optimum value for this distance by running the gear set and adjusting its position to obtain a tooth contact pattern that is consistent with smooth running and optimum load distribution between mating gear teeth. The optimum mounting distance can then be recorded.

Because of dimensional variations between parts, each gear will have a unique value for the mounting distance and, in most cases, the manufacturer permanently marks this value onto each gear; the mounting distance will be within the tolerance

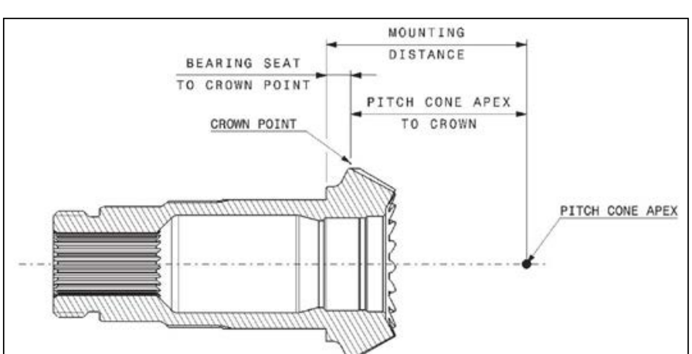

Figure 1 Gear showing make-up of the mounting distance.

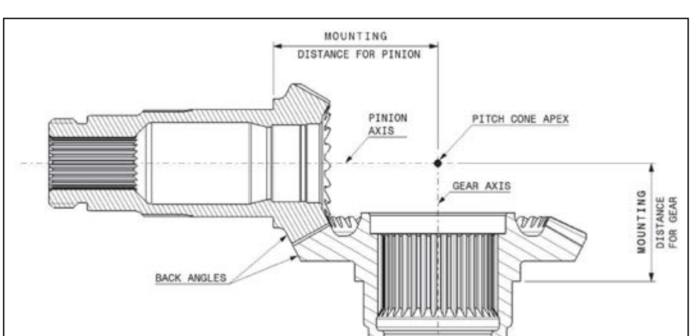

Figure 2 Assembly of a pair of straight bevel gears (pinion and gear).

specified on the drawing. Manufacturers aim for a tolerance of ± 0.05 mm from the nominal value. If, at assembly, the mounting distance of one or both gears is made at less than the dimension specified, the teeth may bind, and excessive wear or breakage can result. If the mounting distance of either gear is made longer than the dimension specified, the gears will not be in full mesh on a common pitch line and may have excessive backlash.

Backlash. The second most important variable for a pair of bevel gears is backlash—i.e., the space between mating gear teeth or the difference in width of the gear tooth and pinion tooth of the mating gear. Unless otherwise specified, it is measured at the tightest point of the mesh.

Backlash is necessary to achieve correct operation of the gears and varies with the size of the tooth and operating conditions. Bevel gears are cut to have a definite amount of backlash when correctly assembled together. But excessive backlash or play, if great enough, can cause a sudden impulse or shock load in starting or reversing that may cause serious tooth damage. Excessive or insufficient backlash can also result in noise, excessive wear and damage. Backlash can be changed by changing the position of one or both gears.

General Notes for the Design Engineer

Gearbox housing/mounting. It is very important that the designer consider the mountings for the gear and pinion so that they are rigidly supported to handle all loads to which the gears will be subjected in service. It is equally important that the designer detail the housing or mounting parts (e.g., bearing cap) with adequate tolerance for alignment, squareness, fits, and runout, etc. While bevel and hypoid gears can accommodate reasonable displacements and misalignments without detriment to tooth action, excessive misalignment of the gears reduces their load capability, with the consequent danger of surface failure and breakage while in service.

Provision should be made by the designer in the design of the housing and mounting parts so that both the pinion and gear mounting distances can be adjusted. This is usually done using shims. Determining the range of shim thicknesses can be calculated by the designer so that the mounting distances of the gears are always maintained. An example of setting up gears and determining the range of shims is explained in more depth later in this guide.

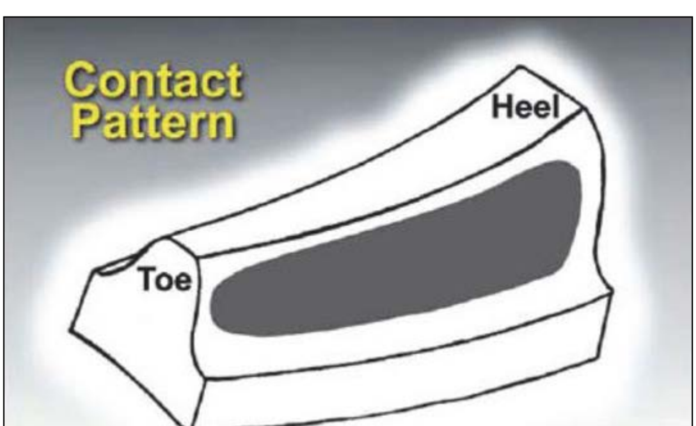

Figure 3 Ideal contact pattern under load (Ref. 2).

It is good practice for the designer to make provision so that the teeth can be inspected and that assembly personnel can also check for backlash, etc. This feature can also be used for inspection during service to check the condition of the teeth without disassembling the gearbox. Once the components are machined, they must be fully inspected to check the housing and mounting components for errors. While the shaft angles, offset and mounting distances are the usual items to check, the alignment of the bores and the squareness of the bearing seats are also important.

Bearings. While bearings are not a focus of this guide, it must be noted that the bearings of a gear system are responsible for holding the components in position, absorbing and distributing forces and moments, and assuring the rotation of shafts and gears. Any clearance in the radial and axial directions inside the bearing can be adjusted by axial pre-loading of the inner or outer ring or race. This pre-loads the rolling bodies and bushings and achieves a stiff, play-free radial and axial support bearing.

Ideally, the bearing manufacturer should be consulted regarding the selection of bearings suitable for the radial and thrust loading required to maintain the alignment and position of the bearing and pinion. The axial and radial movement of the bearings should not allow the load to substantially shift, and the bearings should be stiff enough to prevent deflection in order to maintain proper contact. Deflection of the bearing shaft and the housing are all influenced by bearing contact. It is highly recommended that the bearings be pre-loaded; this should be mandatory—even in cases of small bevel gear sets with low torque transmission.

Solid pre-load is obtained by mechanically locking the races of the bearing into position with the aid of shims. The shims and, hence, bearing races, spacers, etc., are subjected to an axial load; this axial load is commonly provided by the end-caps. By adjusting the thickness of the shims, the amount of pre-load can be increased or decreased. This must be taken into account when determining the range of shims during the design stage and determining the actual size of the required shim at assembly. The disadvantage of the solid pre-load method, however, is the high variation in pre-load as temperature changes, and its reduction as the bearing wears. Pre-loads can be applied by a spring to counteract this and provide a consistent pre-load as temperature varies, but generally this solution is more complex and is beyond the scope of this guide.

It must be remembered that excessive pre-load will reduce the life of the bearing and increase the amount of noise produced and heat generated, as well as the bearing starting and running torque. The amount of pre-load required is determined by virtue of design and stress analysis.

Contact pattern. A critical attribute of any bevel gear design is its contact pattern (Fig. 3).

Simply stated, the contact pattern is the area in which the gear teeth come in contact as they engage and disengage during their rotation. When a gear is installed in a gearbox and is powering the designated application, there are varying degrees of pressure—or load—on the gear teeth. These pressures are influenced by box deflections, bearing movement and temperature changes. When the gear teeth are subjected to these variables, the contact pattern will change. There is a general rule of thumb stating that the heavier the load, the larger the contact pattern.

For a gear to perform properly under load, the contact pattern must be a certain shape and at a certain location. Typically, an ideal tooth contact pattern under load should encompass the bulk of the tooth surface while avoiding any contact with the edges of the tooth surface.

When assessing how the contact pattern will perform in an operating gearbox, another critical issue to consider is gear displacement. In the operation of many gearboxes, the gears and their shafts do not remain in a fixed orientation. Thermal forces and stress from being under load can cause significant movement of the gearbox components from their original positions.

There are typically four different types of movement that can take place (Fig. 4).

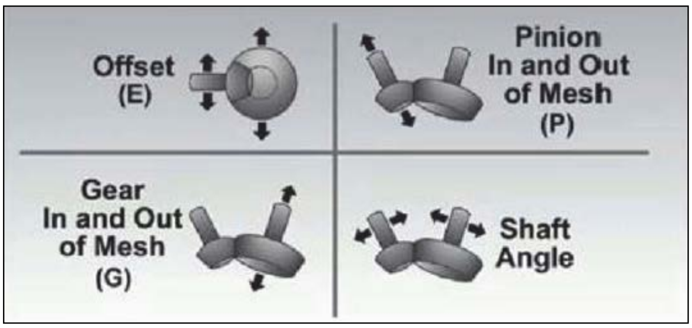

Figure 4 Gear displacement conditions (Ref. 2).

It is this movement that is referred to as "gear displacement," and it can occur in any combination of the four types. In aerospace gearboxes, where keeping weight to a minimum is a high priority, the mass of the gearing used is usually smaller and these displacements can be significant. On the other hand, in commercial applications—where the gearbox components are typically more rigid—there is not the same degree of displacement.

A question could be asked by the designer: "What is the best way to maintain the correct mounting distance and corresponding tooth contact pattern?" And that very question is the "fly in the ointment" in determining the setting position.

Bearings play a significant part in setting proper position. The real issue is not so much where and how one sets it; but more in how the gears run. If the bearings are assembled with preload, then there is usually not much variation in measurement. However, it must be remembered that one is essentially working with some elasticity in the components and the dimensional changes are small. There is always some variation in measurement and shim sizes in the nature of one- or two-thousands of an inch, but bevels are not normally that sensitive where it could be considered a problem.

End-play, however, can cause quite a problem. The traditional response would be to set the gears in the position they would run. That is, if the gears will move in a particular direction under load, force them that way at assembly, taking out end-play in that direction. But if the gears run in both directions, or the operating conditions greatly affect the end-play or position, the technique is not so simple. Bearings typically like to run at a pre-load; if operating conditions are stable and the housing and shafts do not move or grow, the pre-load can be built in at assembly. Ideally, that is the case. However, if there is growth or movement (especially with an aluminum gearbox housing), it would be ideal for the bearings to go to light pre-load so all the rotating elements share the load.

If an advanced analysis of the system design in order to predict its growth and movement have been done, the results could be used to compensate for position at assembly. (Note here that these results should be considered in the bevel design to manufacture the teeth with the proper loaded tooth contact pattern.) Most gear designers/manufacturers cannot predict how the components will be affected in service and, unfortunately, there is only one solution—research and development.

In an ideal world, the first build could be assembled using all the advanced analysis and techniques discussed in this guide. Then the gearbox could be run at operating conditions and checked to see what happened at the contact areas. The gearbox could be stripped and re-shimmed to compensate. What everybody is *really* looking for is, "What is the *loaded* contact pattern?"

Regardless of how it looked at assembly, it is how it runs that matters most.

Part II: Determining Thickness Range of Shims at Design Stage

When laying out a pair of bevel gears, the designer should always begin with the pitch cone apex-to-crown dimensions. These will have been determined by formula, and their values depend on the size of the gears. Once the pitch cone apex-to-crown-value has been established, the designer can determine suitable locating surfaces on the back of the gears. It is recommended that this mounting surface be the seat/shoulder for a bearing; otherwise, it is okay as long as the designer makes the mounting surface a flat surface perpendicular with the axis of the gear/pinion.

The designer must lay out the gears, bearings and spacers, etc. inside the housing so that shims may be used to position the pinion/gear in its correct mounting distance. The designer must arrange the components and assign the appropriate tolerances so that the shim thickness does not fall below 1 mm or exceed 5 mm. If this tolerance analysis has not been properly done, the designer could end up with a negative-value shim (which, of course, cannot physically happen). On assembly, the fitter would have to modify the actual components in order to situate the gear into the correct mounting distance. When the fitter/technician is assembling the gears and is adjusting their position along their axes to achieve smooth running, it may be necessary to adjust the shims. It is far better to adjust shims than to make changes to the housings, end-caps or the gears. This is why the range of shims should be between 1 mm and 5 mm. A shim below 1 mm would be extremely hard to manufacture and handle; it's basically dealing with a sliver of metal. A shim over 5 mm indicates the designer should probably tighten some tolerances or change the nominal size of some components in attempting to reduce this range.

If a worst-case scenario ever arose, it would be more desirable to re-machine the housing rather than re-work the gears to accommodate errors in the mountings. Re-working the gears to accommodate angle or offset errors in the housing is a critical process and should be undertaken only as a last resort; the results are usually not satisfactory.

It is important that the designer think about shimming and setting up the correct mounting distance, as bevel gears are conical in shape and so can be assembled in an almost infinite number of positions (most of which will cause poor performance) and still obtain the required backlash. An inexperienced fitter could in theory assemble bevel gears to obtain a specific amount of backlash without regard to the mounting distance; this is especially true for low-quality or lightly loaded bevel gears. Although this method occasionally works, it is a risky approach where gears are loaded to maximum capacity. It usually causes shorter life and poor performance. Only at the proper mounting distance will a gear set run smoothly and still have the right amount of backlash.

This guide has been written to show the designer a method of determining the size range of shims in a typical gearbox. An example has been included with an in-depth methodology so that a designer can apply the method to other similar gearboxes.

This example gearbox incorporates bevel gears at right angles and bevel gears at an acute angle; this method can also be used for bevel gears at an obtuse angle (Fig. 5).

To begin, the designer can choose any bevel gear in determining the range of shims for that specific gear. For assembling by the fitter, the center gear would be the logical starting point (Fig. 6).

First, the designer positions the pitch-cone apex of the gear on the intersection point of the two bores in the housing. It can be seen that the bores are at an acute angle with one another.

The size range of the first shim can be determined so that the pitch-cone apex of the gear always remains approximately on the intersection point of the two bores. However, it has already been established that the exact mounting distance is not marked onto the gears and, in reality, it would be nearly impossible to measure the intersection of the two bores (as the point exists in space and the bores are at an acute angle) without use of highly specialized and expensive measuring equipment. In practice, the fitter sets the pitch-cone apex of one gear as close as possible to the theoretical intersection of the bores (using the nominal value of the mounting distance and the nominal value of the intersection point) with the use of shims. The fitter then positions the mating gear so that it comes into contact with its counterpart, after which the fitter would physically turn the gears to check for smooth running, adjusting the shims as needed. Once the fitter is satisfied that the gears are running smoothly, he will then measure the backlash to determine if its value is within the backlash limits specified on the drawings.

To make sure that the fitter uses a shim in a range of 1 to 5 mm, the designer has to allow for this fitting process and imagine that the bevel gear is being positioned so that the apex point sits on top of the intersection point. As is known, this will never be exact, so the designer has to assign a ±0.15 (a conservative figure) to the nominal value of the theoretical intersection point. This is the setting distance for the gear, so that the apex point is theoretically "floating" in this range. This should allow for any tolerance build-up of the mating gear's mounting distance, the tolerance of the actual mating gear's teeth, the tolerance of the true intersection point, etc. Figure 7 diagrams a method for determining the maximum and minimum thickness of the first shim.

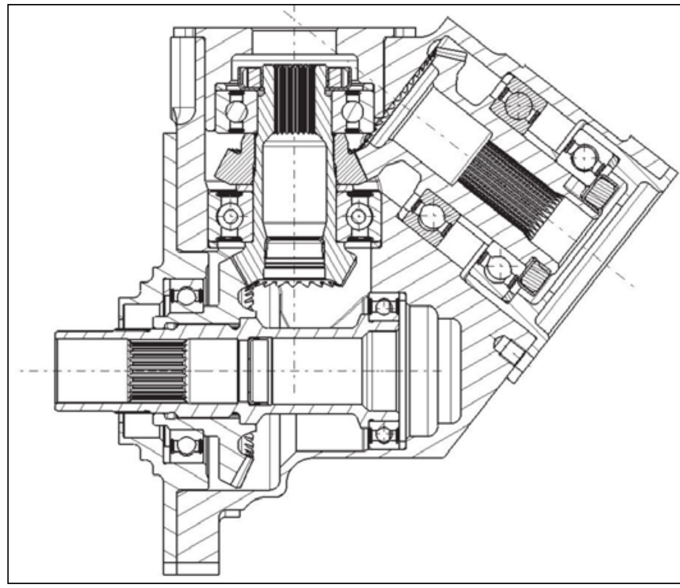

Figure 5 Example gearbox incorporating bevel gears at right angles and bevel gears at an acute angle; this method can also be used for bevel gears at an obtuse angle.

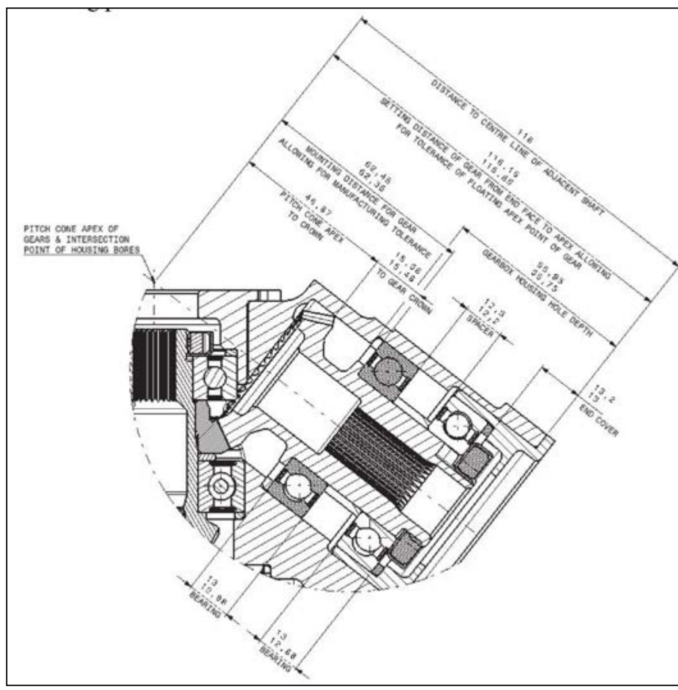

Figure 6 Designers can choose any bevel gear in determining the range of shims for that specific gear. For assembling by the fitter, the center gear would be the logical starting point.

The 0.025 value is shown (Fig. 7) to account for the shift of the inner and outer races due to the axial pre-load.

Determining Maximum First Shim

Maximum Mounting Distance + Maximum Race of Bearing + Axial Play = A

$62.45 + 13 + 0.025 = 75.475$

Minimum Setting Distance of Gear: A = B

115.85: 75.475 = 40.375

Maximum Depth of Bore: (B + Minimum Race of Bearing) = Maximum Shim

55.95: (40.75 + 12.88) = 2.695

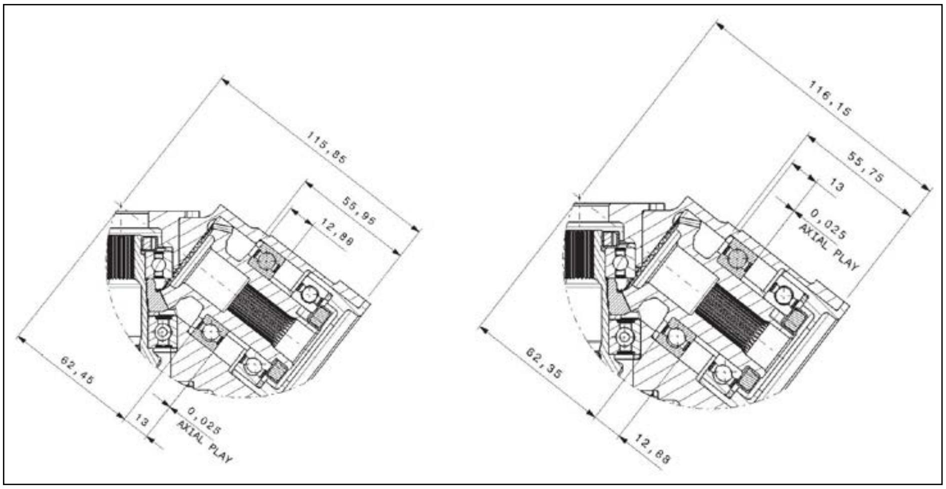

Figure 7 **Diagram of a method for determining the maximum and minimum thickness of the first shim.**

Knowing the range of the first shim, the designer can calculate the size range for the second one. The second shim is important here as it will set up the mounting distance for the second bevel gear that shares the same shaft as the first.

Determining the Maximum Second Shim

Maximum Bore Depth + Maximum Setting Distance Second Gear = A

63 + 5.435 = 68.435

Minimum First Shim + Minimum Race of Bearing + Minimum Mounting Distance Second Gear = B

1.232 + 14.88 + 49.285 = 65.397

A - B = Maximum Shim

68.435: 65.397 = 3.038

Determining the Minimum Second Shim

Minimum Bore Depth + Minimum Setting Distance Second Gear = A

62.8 + 5.135 = 67.935

Maximum First Shim + Maximum Race of Bearing + Axial Play + Maximum Mounting Distance Second Gear = B

2.122 + 15 + 0.025 + 49.385 = 66.532

A - B = Minimum Shim

Determining the Minimum First Shim

Minimum Mounting Distance + Minimum Race of Bearing = A

.62/.35 + 12.88 = 75.23

Maximum Setting Distance of Gear: A = B

116.15: 75.23 = 40.92

Minimum Depth of Bore: (B + Maximum Race of Bearing + Axial Play) = Minimum Shim

55.75: (40.92 + 13 + 0.025) = 1.805

Knowing the range of the first shim, the designer can calculate the size range for the second one.

Determining the Maximum Second Shim

Maximum Depth of Bore: (Minimum First Shim + Minimum Thickness of all Components) = Maximum Shim

55.95: (1.805 + 12.88 + 12.2 + 12.88 + 13) = 3.185

Determining the Minimum Second Shim

Minimum Depth of Bore: (Maximum First Shim + Maximum Thickness of all Components) = Minimum Shim

55.75: (2.695 + 13 + 0.025 + 12.3 + 13 + 0.025 + 13.2) = 1.505

If the minimum thickness of the shims fell below 1 mm, the designer would have to tighten the tolerances or change the nominal size of a component.

The designer can now calculate the size range of shims for the center gear (Fig. 8).

Figure 9 shows the detail for the second bevel gear on the shaft.

Determining the Maximum First Shim

Maximum Mounting Distance + Maximum Race of Bearing + Axial Play = A

38.547 + 15 + 0.025 = 53.572

Minimum Setting Distance of Gear: A = B

99.57-53.572 = 45.998

Maximum Depth of Bore: (B + Minimum Race of Bearing) = Maximum Shim

63-(45.998 + 14.88) = 2.122

Determining the Minimum First Shim

Minimum Mounting Distance + Minimum Race of Bearing = A

38.447 + 14.88 = 53.327

Maximum Setting Distance of Gear: A = B

99.87-53.327 = 46.543

Minimum Depth of Bore: (B + Maximum Race of Bearing + Axial Play) = Minimum Shim

62.8-(46.543 + 15 + 0.025) = 1.232

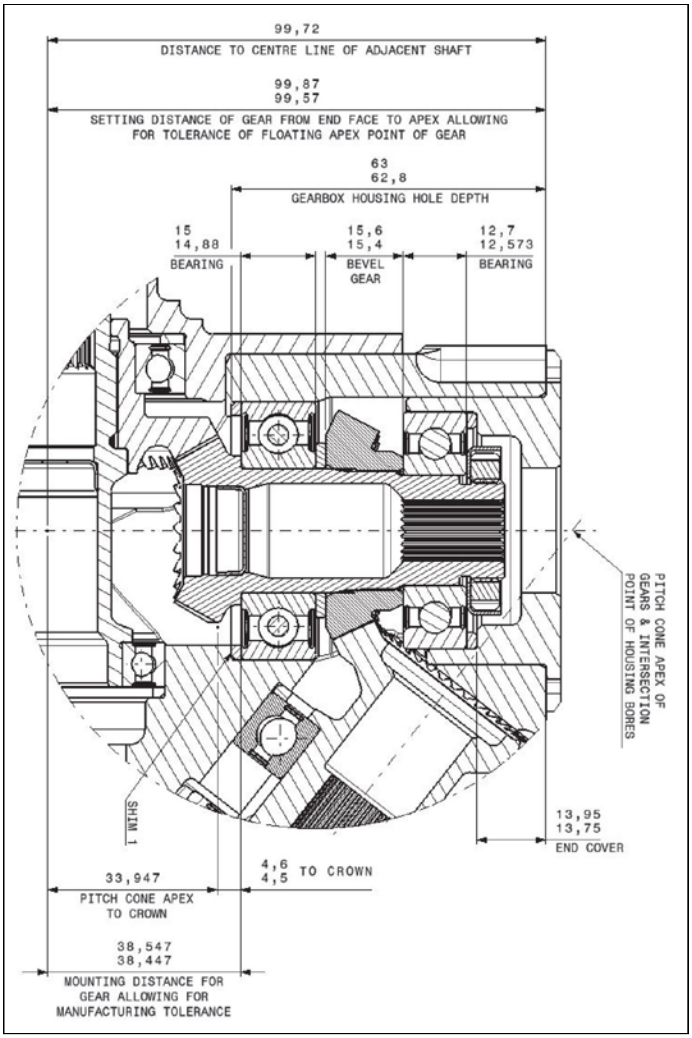

Figure 8 **Designers can now calculate the size range of shims for the center gear.**

67.935-66.532 = 1.403

Determining the Maximum Third Shim

Maximum First Shim + Minimum Second Shim + Maximum Thickness of all Components + Axial Play = A

2.122 + 15 + 0.025 + 1.403 + 15.6 + 12.7 + 0.025 + 13.95 = 60.825

Maximum Depth of Bore: A = Maximum Shim

63 – 60.825 = 2.175

Determining the Minimum Third Shim

Minimum First Shim + Maximum Second Shim + Minimum Thickness of all Components = A

1.232 + 14.88 + 3.038 + 15.4 + 12.573 + 13.75 = 60.873

Minimum Depth of Bore: A = Minimum Shim

62.8-60.873 = 1.927

The designer can now calculate the size range of shims for the third and final gear (Fig. 10).

Determining the Maximum First Shim

Minimum Setting Distance of Gear: Maximum Mounting Distance = A

35.35-27.675 = 7.675

A + Minimum Thickness of Components = B

7.675 + 10.95 + 38.875 + 8.88 = 66.38

Maximum Depth of Bore: B = Maximum Shim

69.1-66.38 = 2.72

Determining the Minimum First Shim

Maximum Setting Distance of Gear: Minimum Mounting Distance = A

35.65-27.575 = 8.075

A + Minimum Thickness of Components = B

8.075 + 11.05 + 38.925 + 9 + 0.025 = 67.075

Minimum Depth of Bore: B = Minimum Shim

68.9-67.075 = 1.825

Determining the Maximum Second Shim

Maximum Bore Depth + Maximum End-Cap Mounting Face to Shim Seat = A

69.1 + 4.1 = 73.2

Minimum First Shim + Minimum Thickness of all Components = B

1.825 + 8.88 + 38.875 + 10.95 + 9.88 = 70.41

A - B = Maximum Shim

73.2-70.41 = 2.79

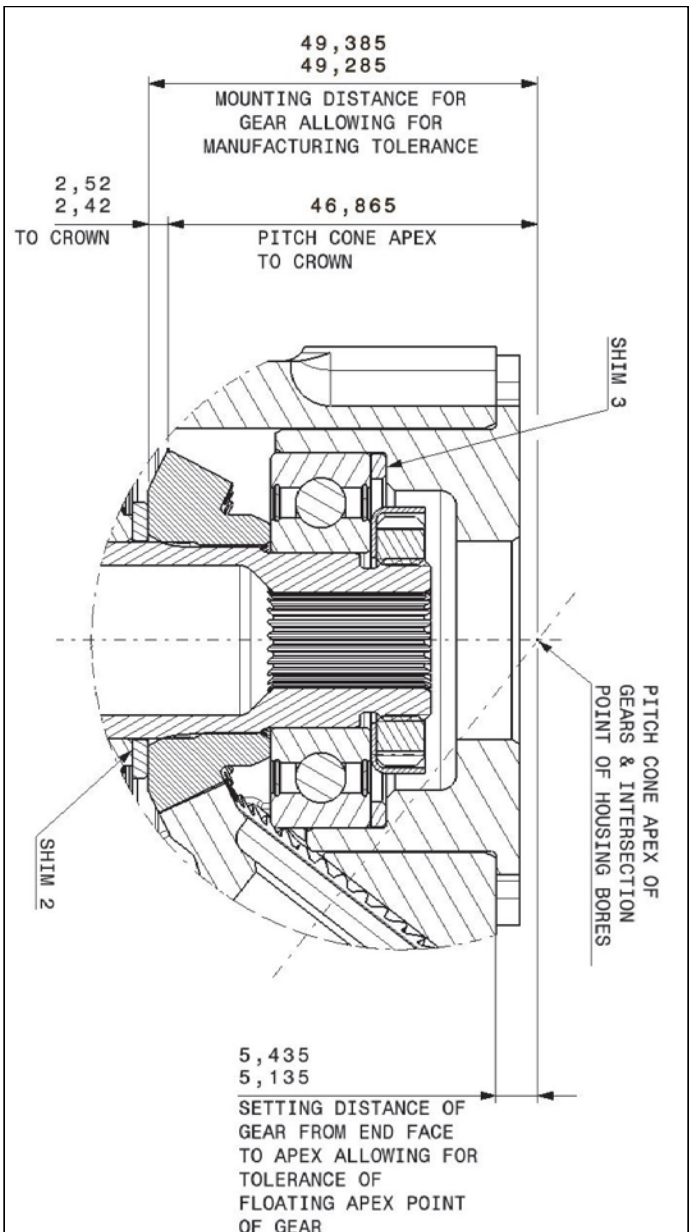

Figure 9 Detail for the second bevel gear on the shaft.

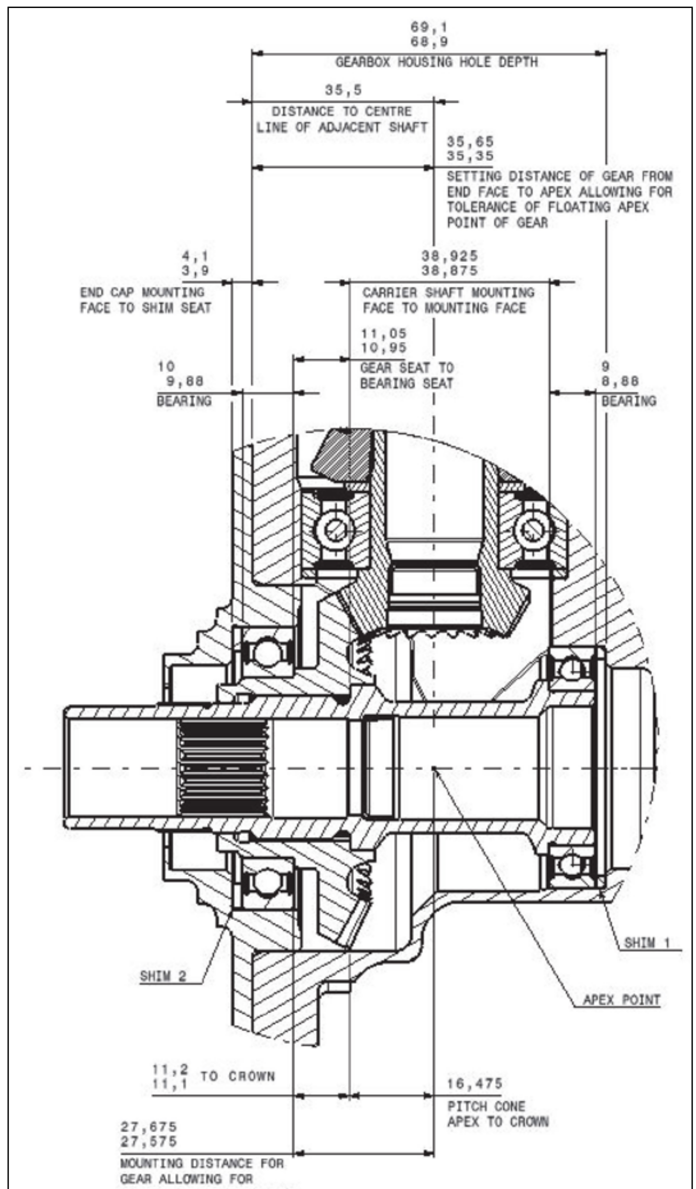

Figure 10 Designers can now calculate the size range of shims for the third and final gear.

Determining the Minimum Second Shim

Minimum Bore Depth + Minimum End-Cap Mounting Face to Shim Seat = A

68.9 + 3.9 = 72.8

Maximum First Shim + Maximum Thickness of all Components = B

2.72 + 9.00 + 0.025 + 38.925 + 11.05 + 10 + 0.025 = 71.745

A-B = Minimum Shim

72.8-71.745 = 1.055

It must be pointed out again that at this stage the designer is not calculating the actual thickness of the shims, but is determining their maximum and minimum thickness limits. The range will most likely be conservative, so as to avoid any future problems at the assembly stage. The designer is allowing for: tolerance build-up of all the individual parts; the true value of the mounting distance; how the fitter positions the pinion/gear inside the housing (which determines where the pitch-cone apex of the gear will be in relation to the intersection point of the bores in the housing); tooth thickness; how the teeth contact and interact with one another; and how the fitter determines “smooth running.” It is better to have a large range of shim sizes on hand than to find out later that the fitter needed a 0.2 mm shim to set the gear at its proper mounting distance at the assembly stage. Proper design will allow the gears to interact smoothly so that the contact pattern on each tooth shares the load evenly and that the contact pattern does not shift towards the edge of the teeth.

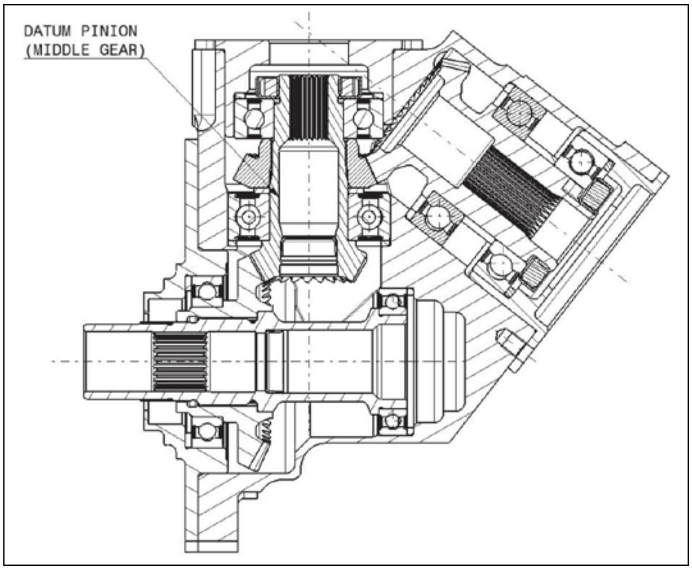

Figure 11 **Section of a gearbox; the fitter should start with the center gear as the datum pinion.**

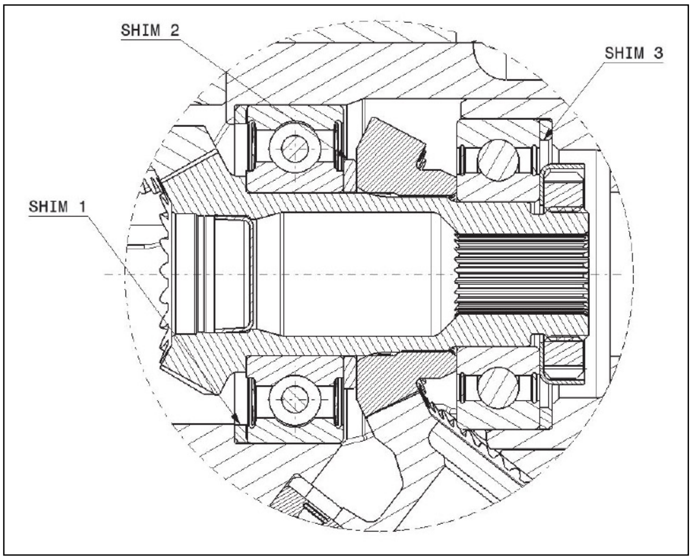

Figure 12 **Location of shims.**

Assembling Bevel Gears and Determining Actual Shim Size

Once the designer is satisfied with the design of the gearbox and has made allowances for the range in shim sizes, the fitter is charged with ensuring proper assembly. The only adjustments the fitter can control are those that axially position pinion and gear at assembly. In certain circumstances the fitter may not be provided with a means of shimming or other methods for positively locating the axial positions of the gears. The assemblies resulting from such inadequate designs will be affected by maximum tolerance accumulations and, in many cases, will not exhibit a good tooth contact pattern.

In gear sets where the mounting distances are known (either marked on the gear or the nominal size on the drawing) and provisions are made for shimming, the fitter should shim to achieve these mounting distances. These adjustments eliminate the effects of axial tolerance accumulations in both the gears and mountings. Shimming however, cannot correct shaft angle or offset errors.

In most bevel gear sets, particularly those with ratios above 2:1, the pinion position (mounting distance) affects tooth contact (the most important parameter for good performance) to a larger extent than does gear position. Conversely, the gear position has a larger effect on backlash.

For this reason the manufacturer may have marked the mounting distance only on the pinion. In such cases, be sure to first accurately position the pinion according to

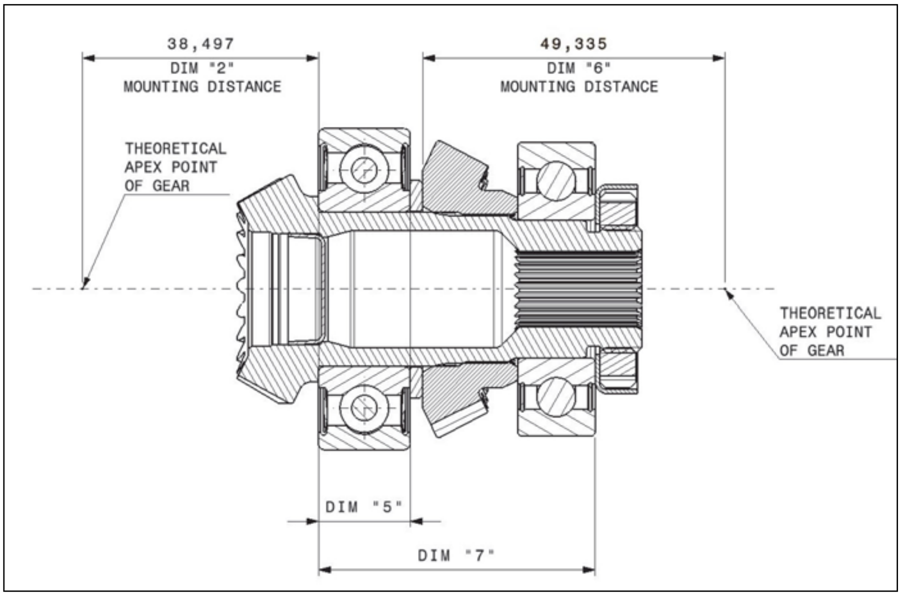

Figure 13 **Bevel gear assembly.**

its marked mounting distance (or nominal value from drawing), and then adjust the gear position for the required backlash.

Where mounting distances have been marked on both pinion and gear, adjust these distances first and then use the normal backlash value to verify proper assembly. Again, the backlash value should be measured *after* assembling the pinion at its marked mounting distance.

It is recommended that the pinion be positioned first to its correct mounting distance when installing a pair of bevel gears. The principal methods for positioning the pinion are:

- ***Gaging.*** A set-up gage assembled in the housing in place of the pinion.
- ***Direct measurement***. Measurements from the locating surface on the pinion to the axis of the mating gear.
- ***Flush surfaces***. In the case of large bevel gears where direct measurement of mounting distance is difficult, a flat is hand-ground on the back cone faces (back angles) of the gear and pinion when in correct position on the testing machine. When the gears are assembled, they are positioned so that the ground flats on the back cone surfaces are flush.

The first method of using gages is used mostly for large gear production runs and is based on the direct measurement method. For this reason, only the measurement method is described in detail in this guide. These measurements may be made by means of inside and outside micrometers, depth micrometer, gauge blocks or a special gauge manufactured for this purpose.

The third method for positioning the pinion exists but is not recommended because the pinion and gear need to be manufactured as a gear set (matched pair). If the surfaces have not been ground as in the third method, then note that when mating gears are adjusted to their optimum position, their back angles will probably not be flush with each other. *Do not attempt to position bevel gears by making the back angles flush.*

To position the pinion by the measurement method, all the components that affect the location of the pinion must be directly measured.

In an ideal world, to aid the fitter at assembly all the housing measurements would be marked on the gearbox housing after it has been machined and inspected. For example, the distance from the locating surface to the adjacent bore center-line could be marked.

Measurements involving the pinion and gear should be taken on the sub-assemblies, which consist of the gear, shaft, spacers, bearings and gear-mounting components. This will minimize any variations due to fits between these components and greatly reduces the number of measurements required.

This example gearbox incorporates bevel gears at right angles and bevel gears at an acute angle (method can also be used for bevel gears at an obtuse angle). It is the same example that was used to determine the range of shims earlier in this guide. Figure 11 shows a section of a gearbox; the fitter should start with the center gear as the datum pinion.

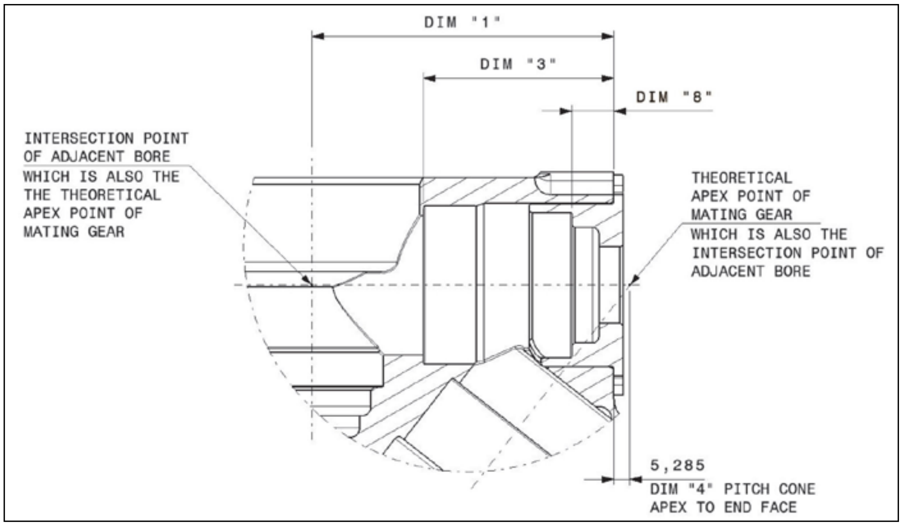

Figure 14 Intersecting bores inside housing.

The fitter would measure to the following method:

i) Dimension 1 = Measure gearbox housing end-face to adjacent bore center-line.

If the mounting distance is not marked on the part, the middle value (nominal figure) of the maximum and minimum size of the mounting distance is to be used.

ii) Dimension 2 = Bevel gear assembly mounting distance = 38.497

iii) Dimension 3 = Measure Depth of Housing Bore

SHIM 1 = DIM 2 - (DIM 1 - DIM 3)

In this example the intersection of the adjacent bore is at an acute angle and the intersection point exists in space; this makes it extremely difficult for the fitter to measure this dimension without highly specialized measuring equipment. The designer therefore has to assign a nominal dimension on the sectional drawing to aid the fitter in assembly.

When the shaft angle is other than 90°, the distance from the pinion mounting face of the housing to the crossing point is not easily measured, as stated above. This dimension can, however, be easily obtained during the machining of the housing, and either the actual dimension or the deviation from the mean can be marked onto the housing. With this dimension available, the necessary shim can be determined in a manner similar to a pinion having a 90° shaft angle.

iv) Dimension 4 = Intersection point of bores from end-face = 5.285

v) Dimension 5 = Measure over inner race of bearing

vi) Dimension 6 = Second bevel gear mounting distance = 49.335

SHIM 2 = (DIM 3 + DIM 4) - (SHIM 1 THICKNESS + DIM 5 + DIM 6)

vii) Dimension 7 = Measure over two bearings, shim 2 and second bevel gear width

viii) Dimension 8 = Measure end-cover depth (mounting surface to bearing seat)

SHIM 3 = DIM 3 - (SHIM 1 THICKNESS + DIM 7 + DIM 8)

The fitter must now adjust the shim to ensure pre-load on the bearings. This can be achieved by ensuring a 0.05 mm (for example) space exists between the end-cover and housing mounting face. The fitter would achieve this by adding 0.05 mm

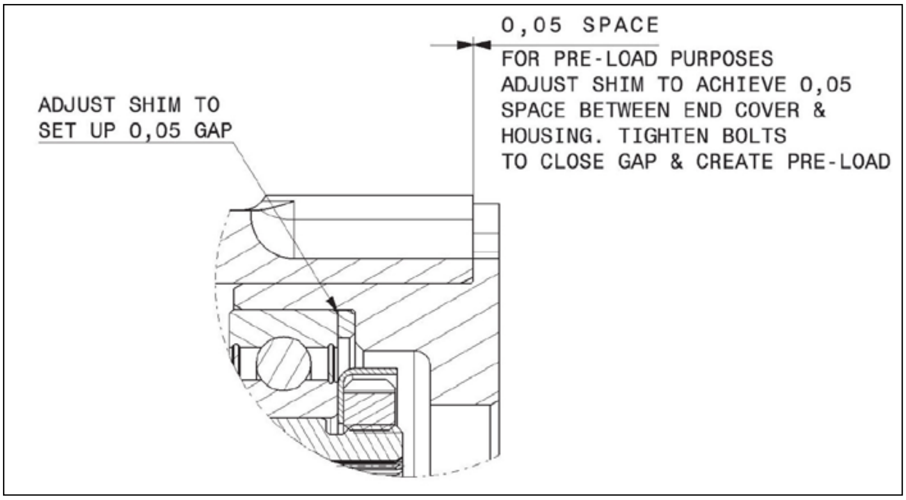

Figure 15 **How pre-load is applied to the bevel gear.**

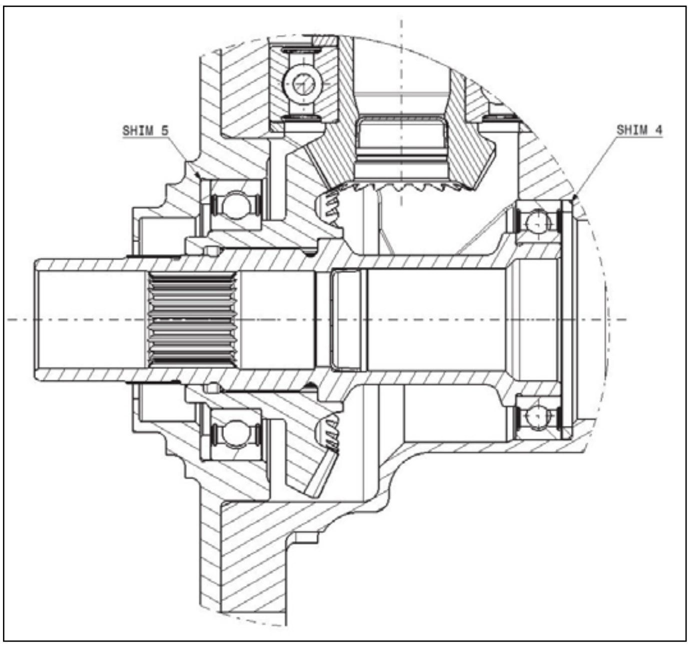

Figure 16 **Location of shims.**

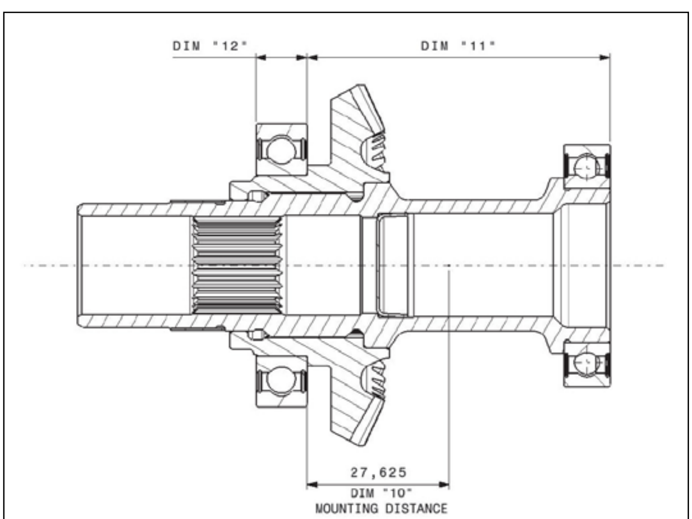

Figure 17 **Bevel gear assembly.**

to the already calculated shim size, so when the bolts are tightened it axially loads all the bearings.

The fitter would measure to the following method for the second gear:

i) Dimension 9 = Measure gearbox housing (shim seat) to adjacent bore center-line
ii) Dimension 10 = Bevel gear assembly mounting distance = 27.625
iii) Dimension 11 = Measure assembly over bevel gear width, carrier shaft and bearing
iv) Dimension 12 = Measure over inner race of bearing and record for later

SHIM 4 = (DIM 9 + DIM 10) - DIM 11

Assemble gear to pinion. If assembling a gear set, make sure that the matched teeth are properly engaged. The fitter now has to rotate the gears and check for smooth running. If necessary, adjust the shim to ensure smooth running. Only after positioning both gears at their mounting distances, and the fitter is satisfied with the "smooth" running, should the fitter measure the normal backlash to verify proper assembly. The fitter should measure at the tightest point of the mesh between the gears.

vi) Dimension 13 - Measure Housing Bore Depth from Flange Face
vii) Dimension 14 - Measure End Cap mounting face to Bearing Seat

SHIM 5 = (DIM 13 + DIM 14) - (SHIM 4 + DIM 11 + DIM 12)

The fitter must now adjust the shim to ensure pre-load on the bearings, as before.

The fitter should measure using the following method for the third and final gear:

i) Dimension 15 = Bevel gear assembly mounting distance = 62.4
ii) Dimension 16 = Measure assembly over bearings and spacer and record for later
iii) Dimension 17 = Measure depth of housing bore
iv) Dimension 18 = Intersection point of bores from End-face = 116

SHIM 6 = DIM 15 - (DIM 18 - DIM 17)

Assemble gear to pinion. If assembling a gear set, make sure that the matched teeth are properly engaged. The fitter now has to rotate the gears and check for smooth running. If necessary, adjust the shim to ensure smooth running. Only after positioning both gears at their mounting distances, and the fitter is satisfied with the "smooth" running, should the fitter measure the normal backlash to verify proper assembly. The fitter should measure at the tightest point of the mesh between the gears.

v) Dimension 19 = Measure end-cap mounting face to bearing seat

SHIM 7 = DIM 17 - (SHIM 6 + DIM 16 + DIM 19)

The fitter must now adjust the shim to ensure pre-load on the bearings as before.

Conclusion

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The gearbox used in this guide is just an example; it incorporates right-angled bevel gears and acute-angled bevel gears. The way the pre-load is applied in this gearbox does not necessarily mean that this is the preferred method. This guide aims to provide the designer sufficient knowledge to design any type of gearbox involving bevel gears.

It must be remembered that due to the complex nature of the forces generated by transmitting power through bevel gears, along with the combination of thermal forces and tolerances of individual components, it is virtually impossible to fully predict the behavior of gears under full-load during its operating service.

However, if the designer and fitter use this guide as a basis to design and install bevel gears, it should in theory be possible to design and build a gearbox that runs under full-load with an ideal tooth contact pattern.

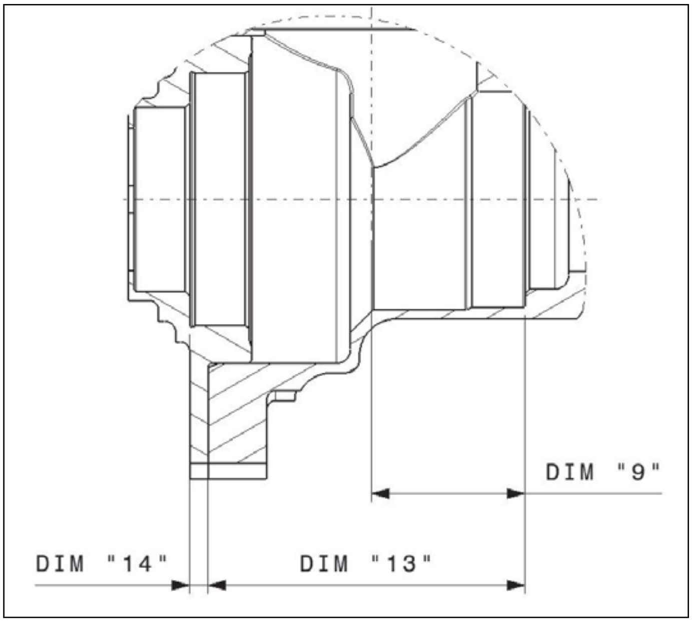

Figure 18 Intersecting bores inside housing.

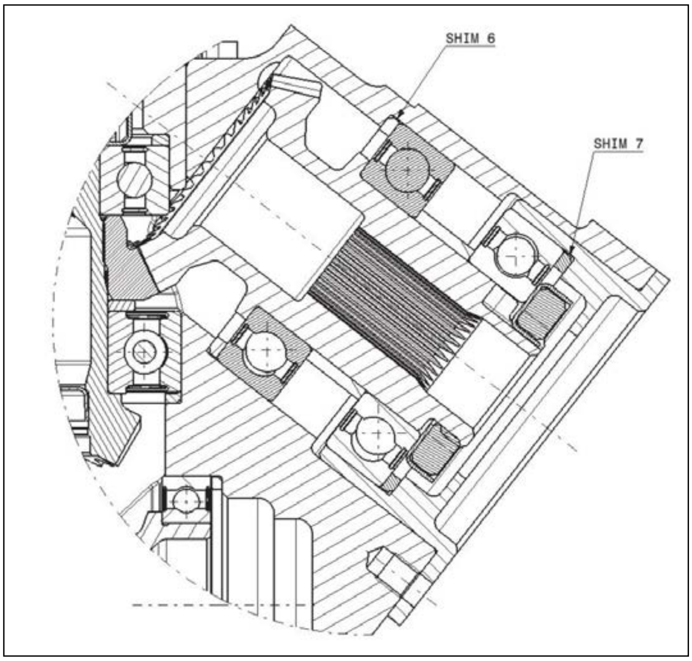

Figure 19 Location of shims.

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Stephen Marsh is an incorporated engineer (IEng) with the Engineering Council (UK) and the Institution of Mechanical Engineers, working primarily in the aerospace and nuclear industries. He began in 2000 a technical modern apprenticeship and an HNC in mechanical engineering with Rolls-Royce plc., and in 2005 graduated from the University of Central England with a BEng (Hons) degree in mechanical engineering, while working as a CAD engineer and draftsman on the HP compressor for the V2500 International Aero engines. Marsh later transferred to the Civil Aerospace Group—Transmissions, Structures and Drives in 2007, working on projects such as Trent 900, Trent 1000, ANTLE and EFE. He subsequently changed industries, moving to the nuclear sector as a design engineer for Nuclear Engineering Services Ltd., and then to UTC Aerospace Systems (formerly Goodrich Actuation Systems), where he currently works as a concept design engineer on projects such as designing the hydraulic actuators for the thrust reverser actuation system (TRAS) on the A320 NEO and designing the gearboxes and telescopic coupling for the variable-area fan nozzle (VAFN) actuation system.

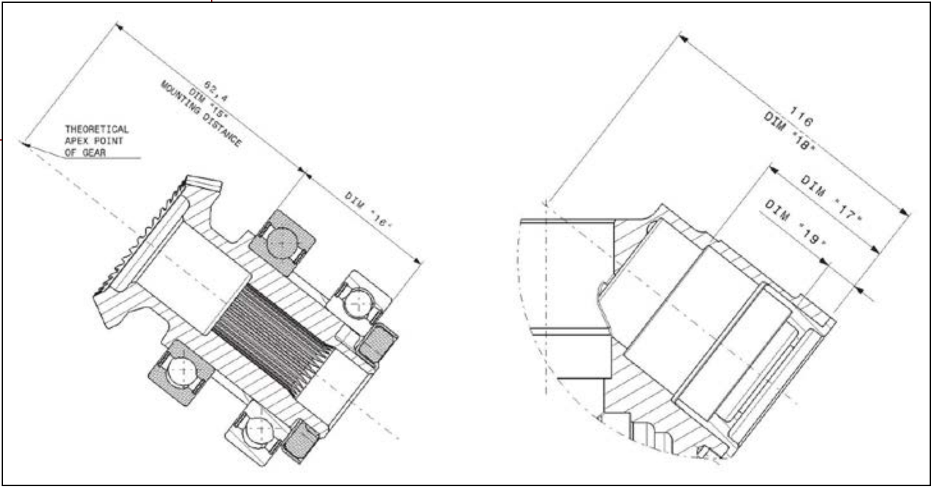

Figure 20 Bevel gear assembly and intersecting bores inside housing.

Building the Perfect Gearbox

ROMAX SUPPORTS WIND INDUSTRY STANDARDS WITH SOFTWARE TECHNOLOGY

Mirelle Ball, ACIM and Dr. P. Gibbs, Romax Technology

Dr. Karl Steingroever (left) presents Dr. Peter Poon, CEO, Romax Technology with 25th Certification.

With the wind energy industry undergoing rapid expansion, pressure is growing on manufacturers to adhere to global framework policies and standards. The certification of wind turbines and wind turbine components is a crucial requirement in order to export products in different markets, to differentiate them, and for standards to be raised. It is therefore increasingly important for manufacturers to apply for gearbox certification, which requires highly technical engineering knowledge and capability.

Certification on its own is mainly concerned with safety, not profitability. To design a safe, reliable and profitable gearbox requires a combination of agreed industry rules and standards (certification guidelines) plus proprietary methods from the component designers and manufacturers. As such, it requires highly experienced experts with a broad knowledge of international and national market information, in addition to a deep understanding of the availability of components worldwide and the associated challenges in each field. Gearbox and wind turbine manufacturers going through this process unaided can find it complex and long-winded. For example, the latest level of certification guidelines for wind turbine drivetrains (GL 2012) requires a complete model of the main shaft, gearbox and generator to take into account static and dynamic system interactions.

Standards benchmarked by three to four key global certification bodies also need to be interpreted through experience to ensure a cost competitive product is developed that works. There is great benefit in perfecting this process to reduce time, eliminate design errors, and decrease the risk of data loss between the different stages of drivetrain design process.

A major difference between wind and other industries is the lack of testing. Large components are too expensive to test and even if budgets allowed, actually applying real-world loads to components is extremely difficult and only a handful of test facilities are available worldwide. This increases the risk of a poor design dramatically as there is no "safety net" from comprehensive physical prototyping.

There has been limited industry take-up so far in simulation software capable of bridging this testing gap by adequately modeling the behavior of the wind turbine under different external conditions and different manufacturing tolerances/processes. Romax believes there is an industry need for the adoption of early stage analysis within the design process in order to assist the designer in working through huge numbers of potential design scenarios. Using simulation software packages programmed with intelligent "rule of thumb" parameters, designers can very quickly go through a large number of potential configurations and home in on the optimum solution. Without such tools this optimization and virtual prototyping is simply not possible within the short development times required by the industry.

More importantly, it allows a lot of the risk-reduction work to be done up front, so that issues within the design can be 'ironed out' prior to deeper design development (and the large additional costs that come with it).

Optimizing the Process

Engineers must first and foremost always look for ways to optimize drivetrain performance and increase reliability. In particular, engineers must constantly look at ways to solve component failures, reduce costs or speed up development time. Top engineering consultancy firms have managed to reduce gearbox development time by up to six months, through the increase of parametric studies at the concept stage with software, leading to a right first time approach to gearbox design.

For the design and certification process, Romax spends an extensive amount of time looking at the way things work within the drivetrain design and development process, analyzing them, removing bottlenecks and automating tasks where possible. Our engineering knowledge has led to the creation of bespoke software modules, packages and consultancy services to support many industries (including the wind industry) in making the much needed leap from basic design to virtual prototyping.

Best Practice

After painstakingly going through the certification process to perfect it, like many organizations Romax have adopted a best practice solution which looks at improving all aspects of the design, development and analysis of the drivetrain in order to support manufacturers in achieving certification targets.

For the last three years, Romax has solidly worked on developing this best practice, which has resulted in many large scale successful global projects, with a total of 29 gearbox designs in production for wind turbines ranging from 1.5 MW to 6 MW upward, of which 25 (to date) have been awarded GL's Statement of Compliance to A-Design Assessment Certification. "Looking back over the last five years working with Romax Technology, I can clearly see how Romax has developed in significantly raising the quality of the gearbox designs," says Dr. K. Steingroever, Germanischer Lloyd (GL). "Spending time and effort putting into place the correct processes to analyze and review the gearboxes allows Romax to continuously develop and become rapid in perfecting design details. This attention to detail has made them very successful in the market, as the only global company to achieve 25 gearbox design A-Level certificates."

"Through my experience of working with Romax, I find them to be very professional across the globe. We know what we can expect from each other's organizations and this allows us to achieve certification quickly and efficiently."

As part of its on-going investment in drivetrain development, Romax utilizes state-of-the-art design and analysis techniques supported by *RomaxDesigner* for wind, simulation software. *RomaxDesigner* is the only software certified by Germanischer Lloyd (GL) for gear calculations to design and analyze drivetrains to certification standards. This is supported by its life cycle engineering services including manufacturing assessment and support, technical training, operations and maintenance services and its advanced predictive maintenance tools.

Romax's recent release of key new products including *Concept*, *CAD Fusion* and *Dynamic Fusion* further highlights the company's commitment to supporting the principles behind the certification process by enabling a rapid, seamless and lossless workflow for driveline design and development from planning to manufacture.

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MTB

REPRESENTS BURRI GEAR GRINDING IN NORTH AMERICA

Machine Tool Builders (MTB) has been appointed as official representative of sales and service for Burri gear grinding equipment in North America. Burri Machine Tools GmbH & Co. KG is involved in remanufacturing or upgrading Reishauer gear grinding machines and the manufacturer of profiling (dressing) machines. Located in Villingendorf near Rottweil in the southwest of Germany, Burri machines are rebuilt to be modern and innovative technology, competitive with today's expectations for reduced set-up time and lower costs of gear grinding. Burri machines are built on the mechanical basis of Reishauer type AZO/ AZA / RZ 301/ RZ 361 / RZ 362 / RZS platforms. When quality becomes an issue on existing equipment or the demand for additional production is required, that's when a trade-in or acquiring a rebuilt machine makes sense. MTB will offer two main products of Burri in North America and provide sales, installation and service:

1. CNC Gear Grinding Machines BZ330 / BZ331 / BZ362; a CNC continuous generating gear grinding machine. The machine is built on the mechanical basis of a Reishauer RZ301/RZ362. After the retrofit, the machine will be technically comparable to the latest machine generation.
2. Grinding Wheel-Profiling Machine PM 550T (dressing machine) is utilized for profiling of grinding wheels for threaded wheel grinding machines; profile grinding machines; bevel gear grinding machines; rack grinding machines; surface and

circular grinding machines; honing rings for gear honing machines; and balancing of grinding wheels.

Controls: B&R Automation

The machines are designed and developed completely at Burri's premises in Germany. Software Engineering develops a dialogue-controlled user interface for all gear grinding and profiling machines, which proves itself with simple menu navigation and logic structure. This enables the user to learn the operation of the machine within a very short time. The open B&R controls system is equipped with all the latest extras and a modern control unit for machine tools, allowing Burri to design all machines to match exactly with their requirements. One unique benefit is the reduced power consumption which has been noted to be reduced by one-third with the high-tech B&R control unit.

Installations and Customers References

Installations of Burri grinders and dressing machines are in many prominent, well-known companies throughout the world. Industries include automotive, machine tools, agriculture and other market sectors.

Service and Support

Machine Tool Builders was chosen as the official sales and service representative for Burri equipment because of its reputation for excellence. MTB has been rebuilding, recontrolling, servicing and repairing gear cutting and grinding equipment for twenty years. For more information, visit *www.machinetoolbuilders.com*.

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For publication guidelines and more information, please contact Jack McGuinn at *jmcguinn@geartechnology.com.*

Liebherr

TOUTS TECHNOLOGY AT LATEST GEAR SEMINAR

For two days in Saline, Michigan, Liebherr's clients, customers and friends came together to discuss the latest gear products and technology. Peter Wiedemann, president of Liebherr Gear Technology Inc., along with Dr.-Ing. Alois Mundt, managing director, Dr.-Ing. Oliver Winkel, head of application technology, and Dr.-Ing. Andreas Mehr, technology development shaping and grinding, hosted a variety of informative presentations.

Topics included techniques in dry hobbing, gear coating, gear cutting, gear shaping, gear inspection, automation solutions, hobbing tools, CVD-reinforced dressing tools and precision-shaped grains for gear grinding. In addition to Liebherr's own presentations, experts from LMT-Fette, Ingersoll, Oerlikon Balzers, Dr. Kaiser, 3M-Winterthur, Wenzel and WZL RWTH-Aachen shared insights and technologies that are changing the gear industry now and in the future. Here are a few highlights from the event:

Gear Inspection

Wenzel, who became a partner with Liebherr earlier this year, discussed its latest gear measuring machine, the WGT 280, during the gear seminar. The WGT 280 is the first model of a new generation of gear measuring machines that made its debut at Control 2013 in Stuttgart.

It will also be exhibited in the Liebherr booth during Gear Expo 2013 in Indianapolis. "The WGT 280 can inspect gears starting with a module of 0.5 as standard and gives us greater flexibility for small gears and rotationally symmetrical parts," said Heinrich Bruderle from Wenzel. "With this machine and the rest of the gear measuring technology provided by Wenzel, we'll be able to provide gear inspection for almost all automotive applications."

Hobbing Solutions

Since its introduction in 2011, SpeedCore by LMT Fette has provided cutting speed increases and significantly more parts in less time. Together, Thomas Falk from LMT Tool Systems and Winkel from Liebherr shared insights into the user benefits of SpeedCore including regrinding and recoating without any process change, higher cutting speeds and significant cost reduction. "SpeedCore closes the gap between carbide and PM-HSS and offers benefits with high performance coating," explained Falk. In a comparison study between a PM-HSS hob and SpeedCore, Liebherr found that SpeedCore provided increased productivity, performance benefits and a time savings of more than 46 percent.

Gear Grinding

In a presentation from Chad Wesner of 3M-Winterthur, the Cubitron II bonded abrasives were examined on applications for an automotive gear as well as a heavy industrial gear. The economic and quality benefits of the Cubitron II provide a more uniform surface finish and no grinding burn if used properly. In addition, it can cut the standard production time, dressing amounts and number of dressing cycles in half.

By reducing the number of passes and increasing the axial feed rates, Liebherr was able to lower the cost pert part for the automotive gear and the heavy industrial gear. All LCS machines from Liebherr have enough spindle power and dynamic stiffness to utilize the Cubitron II.

During the seminar, the Cubitron II was demonstrated on an LCS machine that displayed its extremely fast cutting speeds. It also provided attendees a firsthand account of the many tool life advantages versus a standard ceramic abrasive.

Many grinding innovations and technologies (including a significant reduction in grinding time) have been developed for both the machine and the tools in recent years. Oliver Schramm, division manager at Dr. Kaiser, discussed these technologies regarding CVD-Reinforced Dressing Tools. Schramm—with assistance from Liebherr's Dr.-Ing. Mehr—discussed how CBN (in combination with vitrified bonding) offers the highest metal removal rate and shows economical potential for mass production. In order to achieve full potential, the machine, CBN worms and dressers must fulfill the highest process requirements. In the future, investigations will be made in grinding tests with worms which allow a cutting speed up to 120 m/s and the influence on the residual stress and the life time of a gear ground with vitrified CBN worms.

Feedback:

The two-day event gave engineers an opportunity to get a comprehensive understanding of the new technologies and solutions that Liebherr can provide in gear manufacturing.

"80 percent of all gears are used in the automotive industry," said Mundt. "Today, automotive companies are going outside more and more to buy gears instead of making their own. Our goal is to develop more relationships with job shops and suppliers in the United States and let them know exactly what Liebherr can provide to them. I think with the attendees of this gear seminar, you see that we're starting to get more and more of these people as our customers."

"We're continuing our diversification approach," adds Wiedemann. "Broaden our customer base, help the suppliers benefit from the processes we offer. We don't like to supply just a machine tool. We want to offer solutions. This can be automation; it can be process developments, whatever the customer needs. We'd like to supply a solutions package. This approach is what we're showing at the gear seminar. We work with partners and industry leaders within the gear industry to make that happen."

For more information:
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- ❏ Manufacturing Engineering Department (F)
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- ❏ Product Design, R&D Department (I)
- ❏ Purchasing (L)
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5

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July 7–11—ICME 2013. Salt Lake Marriott Downtown at City Creek, Salt Lake City, Utah. Integrated Computational Materials Engineering (ICME) has received international attention due to its potential to shorten product and process development time, while lowering cost and improving outcome. Building on the success of the 2011 conference, the 2nd World Congress on ICME will bring together researchers, educators and engineers to examine topics relevant to the global advancement of ICME as an engineering discipline. This conference will provide a forum for presentations and discussions on the ICME approach, challenges of integrating models, data management issues and engineering education, and it will set the stage for further growth in the worldwide application of ICME. The event is sponsored by Ford Motor Company, University of Michigan, Carnegie Mellon University, NIST and Fraunhofer Institute for Mechanics of Materials. For more information, visit *www.tms.org*.

July 22–24—Review of Progress in Quantitative Nondestructive Testing Conference. Hilton Baltimore, Baltimore, Maryland. Following the success of the previous nine programs, the 10th International Conference on Barkhausen Noise and Micromagnetic Testing will be held in Baltimore, Maryland in July of 2013. It is the first time the conference will be held in the USA and it will be held in conjunction with the 40th Review of Progress in Quantitative NDE. The ICBM Organization is honored to be included within the 2013 QNDE Conference. The ICBM Special Sessions will be held from July 22–24, 2013 within the QNDE Conference, which will be held from July 21–26, 2013. The Review of Progress in Quantitative Nondestructive Evaluation (QNDE) is a premiere international NDE meeting designed to be an interface between research and early engineering for the presentation of ideas and results to transfer to engineering development. The annual QNDE conference is sponsored by QNDE Programs and organized by the Center for Nondestructive Evaluation at Iowa State University in cooperation with the AFRL, ARL, ASNT, IEEE, DOE, FAA, NASA-LaRC and the NSF. The proceedings from the conference are published by the American Institute of Physics. For more information, visit *www.qndeprograms.org*.

August 8–10—International Gear Transmission and Equipment Expo 2013. China International Exhibition Center, Beijing, China. With an emphasis on high-end manufacturing in China, the Expo is a great starting point for business meetings and technical exchange. Forum topics for the 2013 show include: new materials and techniques, high accuracy machine tools, heat treatment technology, shot-blasting technology, trends in gear cutting, gear measuring developments, automotive transmission technology, fatigue resistance and a technology roadmap for the gear industry between 2013 and 2030. The reoccurring themes for the 2013 show include the reallocation of upstream and downstream resources, tackling core technology and industrial upgrading. 680 exhibitors and more than 50,000 visitors will take part in the 9th GTE Expo. For more information, visit *www.chinagte.com*.

August 19–21—Gear Failure Analysis Seminar. Big Sky Resort, Summit Lodge, Big Sky, Montana. In AGMA's Gear Failure Analysis Seminar, attendees will examine the various types of gear failure, such as macropitting, micropitting, scuffing, tooth wear and breakage. Possible causes of these failures will be presented, along with some suggested ways to avoid them. Robert Errichello will use a variety of tools and methods – lectures, slide presentations, hands-on workshops with failed gears and Q&A sessions – to give you a comprehensive understanding of the reasons for gear failure. Participants are encouraged to bring their own failed gears or photographs and discuss them during the Q&A sessions. The seminar brings together a vast amount of knowledge not available elsewhere. It will help you solve everyday problems whether you are a gear engineer, user, researcher, maintenance technician, lubricant expert, or manager. The course manual can be used as a permanent reference and guide for failure analysis. For more information, visit *www.agma.org*.

September 16–21—EMO Hannover 2013. Hannover, Germany. Under the motto "Intelligence in Production," the EMO will be showing what modern-day production technology looks like and who is offering it. "Everyone wants to be there. That's why once again the EMO Hannover is well set to continue its success story," says Carl Martin Welcker, general commissioner of the EMO Hannover 2013. At the beginning of the year, more than 1,600 companies from 34 different countries had already registered: they will be occupying around 145,000 m^2 of net exhibition space. Thus the current registration status is significantly higher than the comparable figure for the preceding event. The flourishing demand among vendors of production technology evidences the high perceived importance of the EMO Hannover as one of the sector's international highlights and as a superlative platform for innovations. "Meet the world at EMO" is one of the most important arguments for participating. It's not only German manufacturers who have registered for large-size stands. Asian companies are particularly prominent in showing the flag, firms from Japan, China, Taiwan and Korea who are keen to play a bigger role on the global market. They have once again upsized their areas compared to the preceding event's equivalent period, a trend that's been observable for some years now. In all, Asia currently accounts for a good fifth of the EMO's exhibitors. For more information, visit *www.emo-hannover.de*.

September 17–19—Gear Expo 2013. Indiana Convention Center, Indianapolis, Indiana. Gear Expo is a biennial event and the world's only conference and expo designed exclusively for the gear industry. For three days, gear buyers and manufacturers network and build relationships that benefit their respective companies. Attendees see firsthand the latest technology on the market and discuss trends in the industry with experts. Exhibitors have the opportunity to meet face-to-face with attendees and other exhibitors and will display more than 750,000 pounds of machinery on the show floor. Thousands of professionals from around the United States, international manufacturing hubs, and emerging markets conduct profitable business transactions and collaborate on the innovations that make their operations more streamlined. The ASM Heat Treating Society Conference and Exposition is co-located with Gear Expo 2013. For more information, visit *www.gearexpo.com*.

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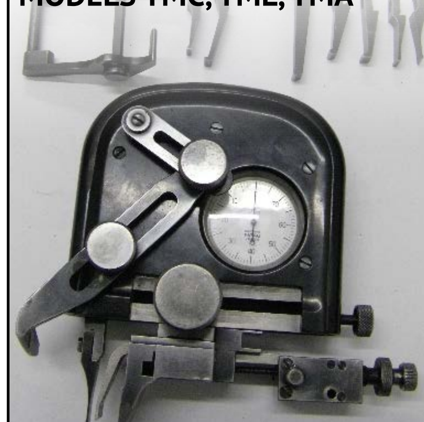

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Need a Snack?

Try 3-D printing technology

Made in Space CTO Jason Dunn (left) and P.I. of the 3DP Experiment Mike Snyder look to optimize the first 3-D printer for space.

We're hungry.

Mounting deadlines, trade show planning and Friday afternoon trivia give the Addendum Staff (recently on a brief hiatus) little or no time to enjoy much food outside of a Milky Way and a carbonated beverage. Sure, there's a Subway right down the street and enough soup cans in the office to survive the zombie apocalypse (3x), but some variety is in order.

3-D printing technology (touted heavily during the last couple of IMTS shows) might make lunch a little easier in the future. How valuable would it be to whip up a dozen gear-shaped cookies in the office in less than five minutes?

The Fab@Home project began at the Cornell University Computational Synthesis Laboratory in 2006 (*www.fabathome.org*). It's an open-source mass collaboration aimed at bringing personal fabrication technology to your home and/or office. The community consists of engineers, inventors, artists, students and hobbyists across six continents.

Using personal fabricators made from off-the-shelf components, Fab@Home's Jeffrey Lipton hosted an instructional video (*www.youtube.com/watch?v=1GG4tWhkxcc*) on creating gear-shaped cookies using 3-D printing technology. He explained the fairly simple process in the video, "The tools you're going to need are a 3-D printer and a deep fryer. The ingredients are half a cup of corn dough, three quarters cup of water and then you simply mix it together in a bowl."

That's it? That's all it's going to take to serve up a fresh baker's dozen whenever your sweet tooth needs a fix? Not exactly, you'll also have to assemble the 3-D printer if you're mechanically inclined to do so.

Lipton remarked that putting the printer together isn't a difficult task at all. "It's completely at-home doable—anybody can build it. This one we like to call IKEA-plus level skill, a little more than putting together IKEA furniture. When we release the next model, we hope for it to be right around IKEA level: just screw it together and you're good to go."

With this ever-changing technology, there's absolutely no reason whatsoever to stop at gear cookies. NASA's Small Business Innovation Research (SBIR) department issued a grant in 2012 to research advanced food systems technology for long space missions using similar technology. Here's the Technical Abstract from the project:

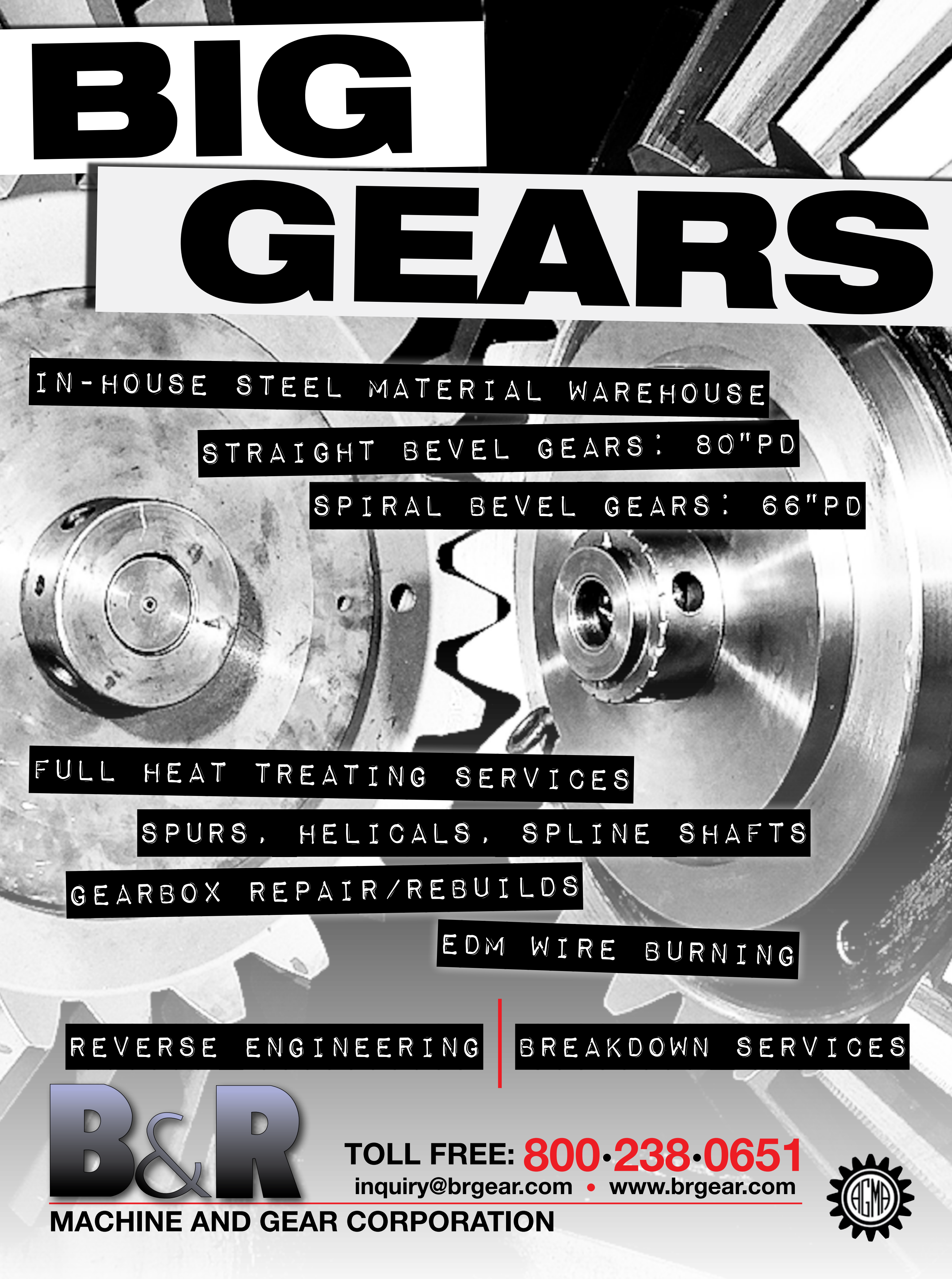
"Systems and Materials Research Corporation (SMRC) proposes combining its Manufacturing Technology and Materials Science expertise to address NASA's Advanced Food System Technology needs. Using progressive 3-D printing and inkjet technologies, SMRC will design, build, and test a complete nutritional system for long duration missions beyond low earth orbit. The 3-D printing component will deliver macronutrients (starch, protein, and fat), structure, and texture while the inkjet will deliver micronutrients, flavor, and smell. SMRC will team with the food science program at North Carolina State University and International Flavors and Fragrances to ensure the production of nutritious and flavorful mission supplies. SMRC proposes producing synthetic food which meets the nutritional needs of each and every mission specialist and astronaut. Using unflavored macronutrients, such as protein, starch and fat, the sustenance portion of the diet can be rapidly produced in a variety of shapes and textures directly from the 3-D printer (already warm). Since basic sustenance will not ensure the long-term physical and mental health of the crew, this is where the microjetting will add value."

The Made in Space team conducts tests during a reduced gravity flight. Made in Space has over 400 parabolas of testing 3-D printing technologies in zero-gravity.

In a separate project, NASA is planning to launch a 3-D printer to the International Space Station to test space manufacturing technology for long-duration missions. The 3-D Printing Zero G Experiment, a partnership between the company Made in Space and NASA's Marshall Space Flight Center in Huntsville, Alabama, will send a Made in Space 3-D printer to the space station in 2014 to demonstrate the feasibility of using the technology to construct spare parts and tools from raw materials on a deep-space mission. If they can find a way to combine both projects, we could live in a world where Captain Jean-Luc Picard could eat a Chicago deep dish pizza the size of an aspirin.

Where do we sign up?

Photos courtesy of Made in Space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