gear TECHNOLOGY®

JUN 2014

30th ANNIVERSARY

1984–2014

30 YEARS OF MECHANICAL ENGINEERING SOFTWARE

TECHNICAL

ADDENDUM

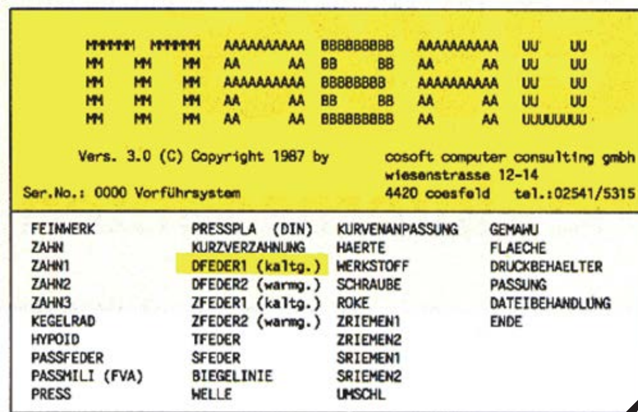
www.geartechnology.com

THE JOURNAL OF GEAR MANUFACTURING

gear TECHNOLOGY

contents

JUN 2014



42

features

technical

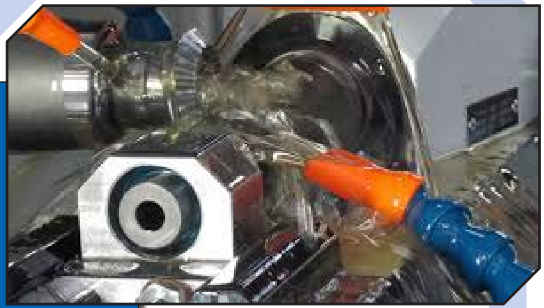
Vol. 31, No. 4 GEAR TECHNOLOGY, The Journal of Gear Manufacturing (ISSN 0743-6858) is published monthly, except in February, April, July and December by Randall Publications LLC, 1840 Jarvis Avenue, Elk Grove Village, IL 60007, (847) 437-6604. Cover price $7.00 U.S. Periodical postage paid at Arlington Heights, IL, and at additional mailing office (USPS No. 749-290). Randall Publications makes every effort to ensure that the processes described in GEAR TECHNOLOGY conform to sound engineering practice. Neither the authors nor the publisher can be held responsible for injuries sustained while following the procedures described. Postmaster: Send address changes to GEAR TECHNOLOGY, The Journal of Gear Manufacturing, 1840 Jarvis Avenue, Elk Grove Village, IL, 60007.

gear TECHNOLOGY Vol. 31, No. 4

departments

THE GEAR INDUSTRY'S INFORMATION SOURCE

www.geartechnology.com



Koepfer America

Our latest video featured on the *Gear Technology* homepage comes courtesy of Koepfer America. This video shows a bevel gear being cut using a special Conikron bevel gear cutter. Since the process is hobbing, it is generating all of the teeth around the circumference of the gear in one pass. See the demonstration at *www.youtube.com/watch?v=hn-RXqXUVIg.*

Success in Manufacturing

Gear Technology blogger Charles D. Schultz recently wrote an excellent post on the perception of skilled trades today as well as a tribute to his uncle, a man that became a tool-and-die maker after coming home from the Korean War. Read the latest posts at *www.geartechnology.com/blog/.*

IMTS Product Preview

Did you know the August issue of *Gear Technology* magazine will feature previews of IMTS 2014? We'll have all the new technologies, products and can't-miss events available on our website as well as Twitter and LinkedIn.

GEAR UP!

Sandvik Coromant recently hosted "Gear Up! New Innovations in Gear Milling Technologies," at the Sandvik Productivity Center in Schaumburg, Illinois. Look for a recap in the July issue of *Gear Technology* and check out the latest article from Sandvik online here:

http://www.geartechnology.com/articles/0514/The_Technology_Shift

Technology Days

EMAG's Technology Days showcase took place in May, marking 20 years of operation here in the United States. For the latest information from EMAG in automation visit: *http://www.geartechnology.com/articles/0514/Moving_Parts.*

Stay Connected

Follow us on Twitter
twitter.com/#!/Gear_Technology

Connect with us on LinkedIn
www.linkedin.com/groups/Gear-Technology-Magazine-3893880

Subscribe Online
www.geartechnology.com/subscribe.htm

gear TECHNOLOGY

RANDALL PUBLICATIONS LLC
1840 JARVIS AVENUE
ELK GROVE VILLAGE, IL 60007

(847) 437-6604
FAX: (847) 437-6618

EDITORIAL

Publisher & Editor-in-Chief
Michael Goldstein
publisher@geartechnology.com

Associate Publisher & Managing Editor
Randy Stott
wrs@geartechnology.com

Senior Editor
Jack McGuinn
jmcguinn@geartechnology.com

Senior Editor
Matthew Jaster
mjaster@geartechnology.com

Editorial Consultant
Paul R. Goldstein

Technical Editors
William (Bill) Bradley
Robert Errichello
Octave Labath, P.E.
Joseph Mihelick
Charles D. Schultz, P.E.
Robert E. Smith

DESIGN

Art Director
David Ropinski
dropinski@geartechnology.com

ADVERTISING

Associate Publisher & Advertising Sales Manager
Dave Friedman
dave@geartechnology.com

Materials Coordinator
Dorothy Fiandaca
dee@randallpublications.com

CIRCULATION

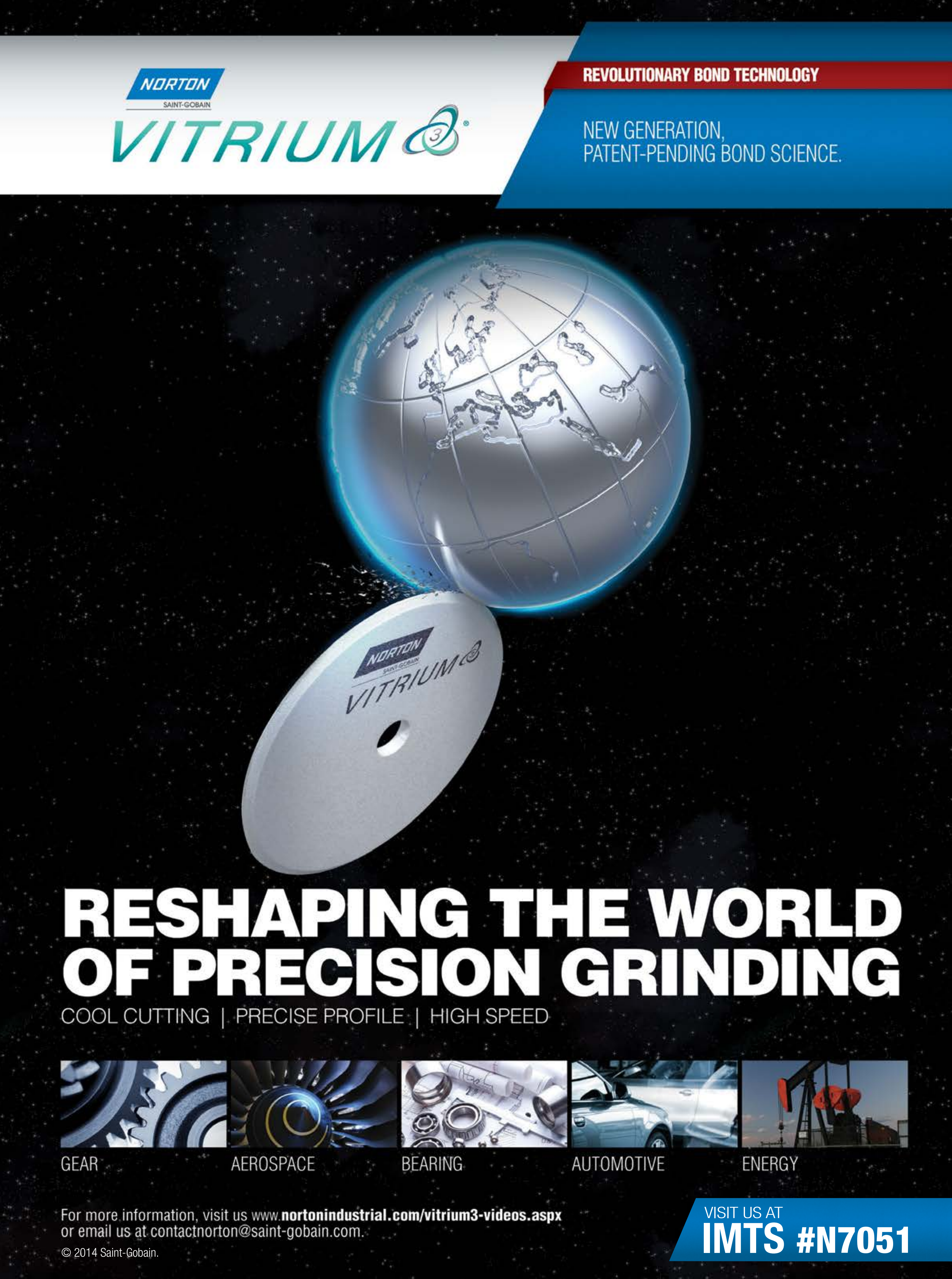
Circulation Manager
Carol Tratar
subscribe@geartechnology.com

RANDALL STAFF

President
Michael Goldstein

Accounting
Luann Harrold

BPA WORLDWIDE



NORTON
SAINT-GOBAIN
VITRIUM3
REVOLUTIONARY BOND TECHNOLOGY
NEW GENERATION,
PATENT-PENDING BOND SCIENCE.
NORTON
SAINT-GOBAIN
VITRIUM
RESHAPING THE WORLD
OF PRECISION GRINDING
COOL CUTTING | PRECISE PROFILE | HIGH SPEED
GEAR
AEROSPACE
BEARING
AUTOMOTIVE
ENERGY
For more information, visit us www.nortonindustrial.com/vitrium3-videos.aspx
or email us at contactnorton@saint-gobain.com.
© 2014 Saint-Gobain.
VISIT US AT
IMTS #N7051

Proverbs

Publisher & Editor-in-Chief
Michael Goldstein

My father, Harold Goldstein, was born in 1917 and came of age during the Great Depression. Like many of his contemporaries, he had to drop out of school to go to work. But my dad didn't allow his lack of formal education to hold him back. He had an insatiable desire to learn, devouring all kinds of books, teaching himself accounting, law, marketing and all the things he needed to become a successful businessman. Working as he did, he learned a thing or two not to be found in schools or textbooks.

My father often spoke in proverbs, and his words of wisdom have guided me my entire life.

"You don't know what you don't know."

"The more you study what seems to be a simple subject, the more complex it becomes."

"If we live in a land where everyone is a taker, then we will live in a very dreary, unhappy society."

That last one was especially significant to me. My dad taught me that I had an obligation to use my knowledge, experience and talents, especially as a small businessman, to give back to our society. He taught me that most people are willing to volunteer at their schools, at their churches or synagogues, or in their communities, but he felt that those with special skills were especially obligated, and he encouraged me to see where I could use my unique skills and experience to give back.

In the beginning of my career, as a used machinery dealer with Cadillac Machinery Company, I volunteered at the Machinery Dealers National Association (MDNA). There, I eventually became president of the board of the association's for-profit publishing company, which published *The Locator*, a monthly magazine listing all the used machinery available on the marketplace. Effectively, each month we put out the equivalent of a printed telephone directory with over 32,000 listings of machinery and mailing over 100,000 copies per month. It was during my eight years at *The Locator* that I became fascinated by the publishing business and where I realized the unique skills that could allow me to give something back to society as more than just a volunteer.

My stewardship of *The Locator* ended 1983, when the United States manufacturing community was in a very deep recession. But at the same time, the CNC revolution was just beginning, first in turning, forming and drilling, soon to be followed by the more complicated machines used in gear manufacturing. At that time, the gear industry was thirsty for knowledge. Gear manufacturers needed to know how to use these new and changing technologies, and they needed to understand the specific engineering concepts of gearing.

While there were technical conferences, produced by AGMA and others, all around the world, the information from those conferences was disseminated to only a few—maybe the owner, president or engineering manager who was able to attend. He'd come home with a stack of technical papers in a blue binder, which would end up on a shelf and often known only to him.

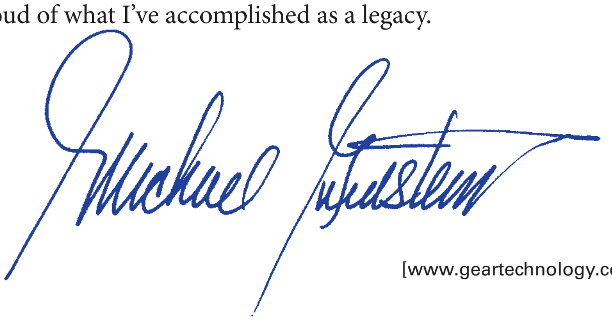
So I started *Gear Technology* as a means to distribute all of this technical information directly to the companies and individuals who needed it—from the executive office to the shop floor.

To make *Gear Technology* a commercial reality, I had to convince some advertisers to support my efforts. The first person I visited was Marty Woodhouse of Star Cutter, who cut off my spiel almost as soon as I began it, saying simply, "I'm in." I said, "Marty, don't you want to hear my ideas?" And he replied, "I've heard enough. I love the idea. I know you can do this, and the industry needs what you're trying to do. You've got the support of Star Cutter Company." I talked to my friend David Goodfellow, who at that time was president of American Pfauter. His immediate reaction was, "I want the inside front cover." My next supporter was Hank Boehm of Liebherr America, who was followed by Klingelnberg and Gleason. Without those early supporters, this magazine would never have gotten off the ground.

I'd like to thank those early supporters, as well as the other companies who have advertised over the years and all of the authors who have so generously given their time to help make *Gear Technology* a reality,

Our first issue was published in May/June 1984, and so, with this issue, we celebrate our 30th anniversary. From the beginning, my vision was to build a body of knowledge that future generations could rely upon—and we continue to take that very seriously. All of our technical articles are vetted for accuracy and relevance, and we include authors from all around the world, many of whom now seek us out for the prestige of publishing in *Gear Technology*. That vision was also to create a continually evolving reference set that was available to anyone who wanted it. Today, because of the Internet, that's truer than ever. All you have to do is visit *www.geartechnology.com* to access that body of knowledge free of charge. You can download any issue from 1984 to today, and more importantly, you can search by keyword for any article from the past 30 years that is relevant to you.

When I look back at all the issues we've published and all the subjects we've covered, I'm somewhat in awe of the sheer volume of information and the value it represents. I can honestly say, now, that I *have* given back, and I'm sure my dad would be proud of what I've accomplished as a legacy.



Inspiring the World Beyond the Theory of Gearing

Celebrating Dr. Faydor Litvin: Remarkable Scientist, Dedicated Mentor, Continuing Inspiration

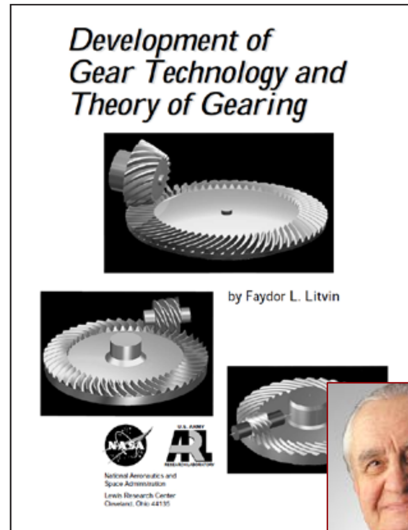
The first time that I had heard of Dr. Faydor Litvin was when I was working on my master's degree thesis about the development of a new type of helical gear drive at a university in China. During that time, my advisor, Professor Zuodu Zhang, shared with me his admiration for Dr. Litvin and how his academic career and research on gear geometries benefited significantly from Litvin's works. Professor Zhang introduced to me a Chinese version of the *Theory of Gearing*, a renowned piece of Dr. Litvin's work, and said to me, "I have long been hoping to meet Dr. Litvin since I read his works in Russian, and although I may not be able to make it in my lifetime, I do hope that you could be his Ph.D. student someday." Since that particular moment, a dream was formed in my mind.

Several years later, my dream came true; Dr. Litvin generously supported my Ph.D. candidacy and became my advisor at the University of Illinois at Chicago (UIC). A year later, as I became fully involved in multiple research projects at the UIC Gear Research Center, I was informed that Professor Zhang had become very sick after fighting cancer for several years. Having heard the news, I wanted to do something for Professor Zhang. After pondering for a while, I came up with an idea and went to Professor Litvin's office, where I told him about Professor Zhang's story and his current situation. I hesitated to ask whether professor Litvin could present professor Zhang with a copy of his book, *Development of Gear Technology and Theory of Gearing*, published by NASA, and autograph it as well. To my surprise, right after I finished my story, Professor Litvin said to me, "Qi, I am touched by your story about your former professor and proud of you for your kindness and thoughtful idea. I am very happy to do this and please also send him my best wishes and a (quick) recovery." In addition, Professor Litvin wrote a personalized note addressed to professor Zhang and proposed to take a picture with me and requested that I send the book together with the picture to Professor Zhang. I felt very thankful and overwhelmed with joy at that moment, knowing just how happy this would make my former professor in China.

In 2010, I traveled back to China and visited Professor Zhang's family but was told upon arrival that he had just passed away. His wife greeted me while tears quickly filled up in her eyes as she recalled just how happy and honored her husband had felt when he received the autographed book and the picture. She recalled that whenever her husband read the book, she could see a feeling of joy and happiness on his face. Reading the book gave him renewed optimism and a feeling of satisfaction, as well as an invisible power to fight the cancer, Mrs. Zhang added. She also told me that her husband had been keeping the book and the picture beside his pillow all the time. Mrs. Zhang requested that I convey her strong feelings of gratitude to Dr. Litvin and thank him deeply for providing her husband with such a wonderful spiritual medication that had helped him courageously fight his illness for so many years.

The power beyond Dr. Litvin's theory of gearing is truly immeasurable and remains an ongoing source of inspiration, touching the personal lives of many gear professionals all over the world. Professor Litvin is not only a remarkable scientist and a dedicated mentor who has advised hundreds of graduate students and visiting scholars from over the world. He is also a great father who is always caring and willing to help his students and their families in their course of seeking professional fulfillment.

> The power beyond Dr. Litvin's theory of gearing is truly immeasurable and remains an ongoing source of inspiration, touching the personal lives of many gear professionals all over the world.
>
> **Qi Fan, Ph.D.**



Dr. Faydor Litvin turned 100 years old in January. Some of his long-time friends, colleagues and collaborators have taken this opportunity to reflect on his career, thank him for his contributions and wish him congratulations.

It is my lifelong honor to have been a Ph.D. student of Dr. Litvin's; without his training and guidance, I would not have had the opportunity to realize my career goal and to be presented with the opportunity to work at The Gleason Works as a gear theoretician.

I would like to take this moment to say Happy 100^{th} Birthday to Professor Litvin and best wishes to him and his entire family.

And lastly, "Professor, thank you very much for everything, and I sincerely wish you continued great health and happiness!"

Qi Fan, Ph.D.

With Dr. Litvin's strong backing of **Qi Fan**'s Ph.D. candidacy, Fan gained admittance in 2000 to the University of Illinois at Chicago (UIC) and Litvin's renowned Gear Research Center. Since 2001 Dr. Fan has worked as Senior Gear Theoretician—Research and Development—for The Gleason Works. Fan is also an associate editor for the ASME Journal of Mechanical Design and a member of the Technical Committee of the ASME Power Transmission and Gearing Committee. He has authored 26 technical papers.

Dr. Faydor Litvin came to the United States in 1978 after a long teaching career in Russia and his retirement as Department Head and Professor of Mechanical Engineering from the Leningrad Institute of Precision of Mechanics and Optics. Upon coming to this country, Dr. Litvin then embarked upon a second career starting in 1979 at the University of Illinois at Chicago (UIC). His second career spanned in excess of 30 years.

During these 30 years, Dr. Litvin had a continual working relationship with NASA, the U.S. Army, and the U.S. rotorcraft industry through the funding of university research grants and contracts. His close working relationship with the U.S. government and the aerospace industry resulted in a multitude of reference publications, contractor reports, conference papers, presentations, and refereed journal articles.

As an educator, Dr. Litvin and the program that he ran from UIC resulted in many specialized gear geometry experts (Ph.D. students) that now work in the gear manufacturing field here and around the world. His personal attention to his students — both foreign and American — was as if each and every one were family members, and he closely mentored them to produce high-quality graduate level research.

> His impact on the gearing world has been substantial.
> **Robert F. Handschuh**

Dr. Litvin also has left his mark in several areas from his dedicated efforts to move the gear geometry field forward. During his government and private industry funding years, all types of gears as used in aerospace drive systems were studied from the gear geometry point of view. His impact on the gearing world has been substantial.

Two of his important contributions came on spiral bevel and face gears. Both of these gear types are utilized in rotorcraft main transmissions to turn the corner from the horizontal gas turbine engines to the vertical rotor shaft. His spiral bevel gear work led to low-noise, high-power gearing through the careful consideration of the gear tooth machine tool settings during manufacture to produce gears with low transmission errors. This practice has been accepted and implemented in various companies in the aerospace gearing business.

The second important aerospace contribution has to do with the development of face gear grinding technology. An Army-funded program produced a design utilizing this type of gearing that resulted in a substantial overall drive system weight advantage. The problem was that the manufacturing technology of this type of gear was stuck in the 1950s. A method had to be developed to grind the gears after carburization. A manufacturing technique was developed by Dr. Litvin and his Ph.D. students to provide the high-accuracy gearing necessary for rotorcraft drive systems. This technology has found its way into the upgraded attack helicopter used by the U.S. Army.

The U.S. gear manufacturing industry is a better place due to the dedicated work of Dr. Faydor Litvin.

A true patriot of his adopted country.

Robert F. Handschuh

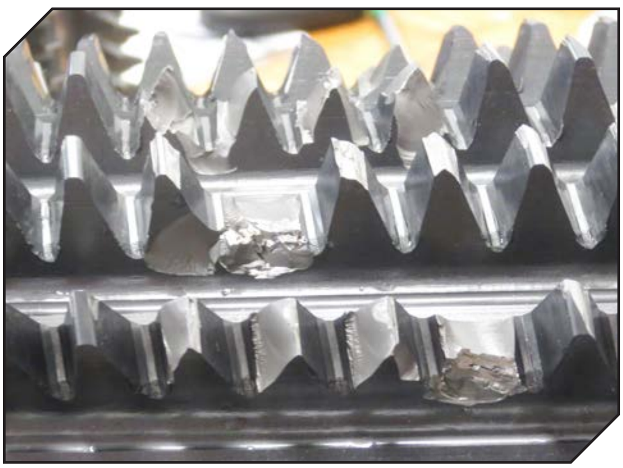
Dr. Robert Handschuh is a 30-year NASA veteran with invaluable experience in DOD rotorcraft drive system analysis and experimental methods. He is credited with successfully developing many experimental research test facilities as Team Leader for the Drive Systems Tribology & Mechanical Components Branch at NASA Glenn Research Center in Cleveland, Ohio, and has conducted testing in areas such as spiral bevel gears and face gears; high-speed, helical gear trains; planetary gear trains; single-tooth-bending fatigue; and high-speed gear windage.

ARTIS Monitoring and Control System

OFFERS THE LATEST IN HOBBING PRODUCTIVITY

Jorge Pena-Mena, General Manager & Application Engineering,
Artis Systems Inc./Marposs Corp.

A modern six- or eight-speed automatic transmission can contain 25 or more precision gears, a fact that makes hobbing a prime target for productivity-improving technologies. For some perspective, a well-known transmission builder produces one million transmissions annually for two lines of luxury automobiles. The eight-speed gearbox contains four planet wheel sets containing a total of 25 gears in two sizes.



The larger gears are hobbed in 40 seconds while the smaller are produced in 4.3 seconds. Advances in tooling and machine technology have quadrupled the productivity of these operations in the last 10 years.

But that performance comes at a price. Today's hobbing tools are considerably more expensive than those of 10 years ago, so users need to optimize the process to achieve the greatest possible number of parts per tool before re-sharpening. Of equal or greater concern, however, is the need to avoid breaking these costly tools.

Traditionally, manufacturers have relied on "rule of thumb" guideliones and hob supplier's expected tool life recommendations to avoid catastrophic failures. But, this approach works by building in extremely conservative safety margins at the expense of productivity, and is paid for in unnecessary machine downtime for tool changes, unnecessary tool maintenance costs and parts not made.

The solution is a process monitoring system with the ability to detect and track tool wear, account for workpiece variations without generating false alarms and instantly stop the process in case of chip welding, damaged or broken teeth, or other conditions such as peeling coatings. Since an individual hob may be re-sharpened up to 15 times, and each re-sharpening changes its diameter and, therefore, its operating parameters, the monitoring system also needs to be self- calibrating to accommodate this factor.

Various attempts have been made to adapt existing process monitoring technologies to hobbing operations, but due to the complexity of the process none have been completely satisfactory. Based on a track record of successful monitoring solutions for other processes, ARTIS—a Marposs company— worked in cooperation with a major transmission builder to develop a hobbing-specific application that would allow the user to achieve maximum output along with maximum tool life.

The project began by collecting real-world data from operating hobbing machines in the customer's plant. The end result was an algorithm representing the life cycle of a hobbing tool that can be used to identify the optimum time to take it out of service for re-sharpening based on its actual condition.

ARTIS then created a system of machine-mounted sensors to monitor process parameters including spindle torque, spindle vibration, power consumption and a number of others depending on the specific application. Using these inputs the system then captures the exact signature of each operation in the process and automatically generates a "good" tolerance band for the process based on that signature.

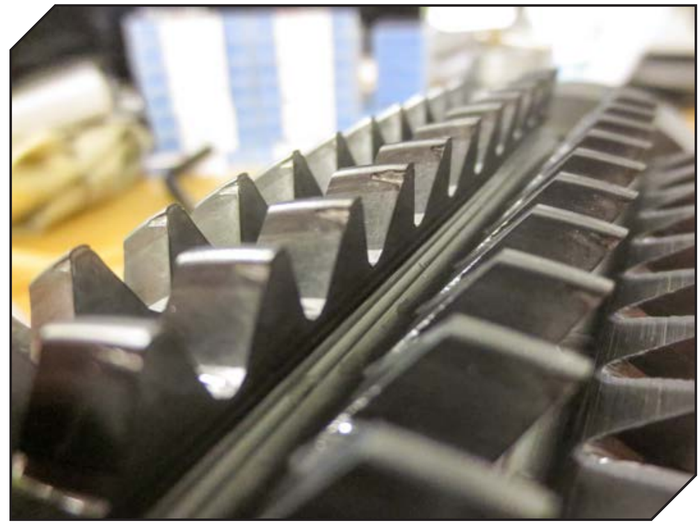
While the concept of monitoring process inputs is not unique, the ARTIS system couples it with powerful software specifically designed to detect the exact kinds of anomalies produced by worn and/or damaged hobs. This data was generated during the in-plant monitoring project which revealed that normal wear, welded chips, peeling coatings and broken teeth all generated distinct signatures before and during the ultimate failure of the tool.

The software can identify and quantify each of these signatures to generate either an approaching end of life warning for normal wear, or an automatic machine stop in case of actual tool damage. In the case of normal wear, the ARTIS system notifies with ample time to schedule the downtime required to minimize the impact on production.

Another unique aspect of the system is the ability to detect and automatically compensate for tool diameter changes after re-sharpening. This is very important in a high-volume environment where individual tools may be used on different machines before and after re-sharpening. A smaller diameter tool changes the operation's power consumption, so the ability to detect diameter changes and re-calculate the optimum process signature without re-mastering the tool eliminates a great deal of machine downtime.

On the test machine, the ARTIS system resulted in a 17-percent increase in tool life without changing any of the process parameters. The increase represents the difference between tool changes based on "rules of thumb" or arbitrary part counts, and tool changes based on actual tool condition as reported by the monitoring system.

During the testing, this customer used the data generated to identify and optimize the coating used on the hobbing tools. While not a direct benefit of the ARTIS system per se, the coating selected on the basis of the process data delivered a 60 percent increase in tool life, an increase that almost certainly would not

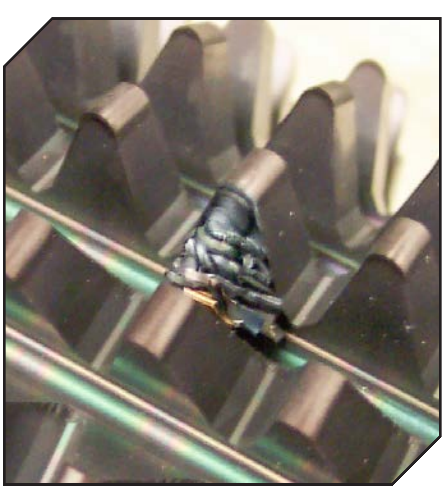


have happened had the data not been available.

In another application, a gear manufacturer reported that their ARTIS system, which had been running for some time without incident, suddenly began triggering alarms, but the hobs showed no signs of wear or damage when inspected. As the manufacturer was preparing to call for service on the ARTIS system, the hob head bearings failed and the machine was immediately taken out of production. The customer credits the ARTIS system for minimizing both repair cost and downtime by allowing them to react promptly to the bearing failure predicted by the repeated alarms.

It is important to note that the initial ARTIS system described here was developed in cooperation with a customer for a specific set of application parameters for gears with a module of 14 or less. Since that customer used only Siemens CNCs and Liebherr hobbing machines, the ARTIS application was developed for that specific combination.

Software with the ARTIS CTM-FP-Gear Hobbing option can be integrated into Siemens 840D controls and is also is available pre-installed in the control. The ability to automatically compensate for tool diameter changes after re-sharpening, and the adaptive control constraint of the feed rate where cycle time can be reduced automatically, are currently available only with Liebherr gear hobbing machine tools.



However, the ARTIS software has also been added to other control systems such as Fanuc via a dedicated PC and control interface. The Artis monitoring components can be applied to virtually any new or existing hobbing machine including those from Liebherr, Gleason Pfauter, Felsomat, MAG-Samutensili, Kashifuji and Mitsubishi, and many others.

ARTIS software is available for soft material applications on gears with module up to 100 and cutting times up to a few hours. All implementations are application specific. Better tools, more capable machines, and improved controls have quadrupled the productivity of the hobbing process in the last decade, just in time to meet the growing demand for precision gears in the transportation and other industries. Now, real-time monitoring and control systems are ready to help gear makers set even higher benchmarks for productivity and quality from the equipment that's already on their production floor.

For more information:

ARTIS GmbH
Phone: +(49) 51 94 9 50-0
www.artis.de

Marposs Corporation
Phone: (248) 370-0404
www.marposs.com

Sunnen

OFFERS SINGLE-STROKE HONING OPTIONS

Sunnen Products Company introduces a quantum improvement in a key manufacturing process for cast-iron, hydraulic valve bodies using a precision single stroke honing process that achieves cylindricity/straightness under two microns in bores up to 10 times longer than diameter. Developed in Europe to meet leak-free requirements for high-pressure hydraulic systems, the high-precision, single-stroke honing process is almost 200 percent more accurate than anything previously achievable for long, small-diameter, tandem bores, according to the manufacturer.

"European mobile equipment hydraulic systems established this cylindricity specification to reduce internal leaks and achieve higher valve performance," said Juerg Huber, managing director, Sunnen AG. "The precise fit between the sliding spool and valve body maintains the internal seal in this area. Precision cylindricity and straightness in this bore ensures uniform clearance between the moving parts from top to bottom in the valve, allowing free movement of the spool, without leakage around it."

Single-stroke honing is a single-pass process that sizes and finishes a bore with a series of progressively larger superabrasive tools adjusted to a preset diameter. Single-stroke honing tools rotate while passing through the bore one time and then withdrawing. The machine's servo-controlled stroking system provides flexibility with adjustable speed and various feed profiles, such as pecking, short stroke, dwell, etc.

The single-stroke honing tool consists of a tapered arbor with an expandable diamond-plated abrasive sleeve mated to it. The external profile of the sleeve is designed for the application. Single-pass honing is ideally suited for solid bore parts with L/D (length/diameter) of 1:1 or less. Parts with much larger length-to-bore ratio can be honed under certain circumstances. Bores that have interruptions allow better chip flushing and reduce the load on the abrasive sleeve. In the case of cast-iron valve bodies, bore length-to-diameter ratios of 10 or greater can be single-pass honed with great success.

"Single-pass honing is a stable, high-production process that can achieve one micron cylindricity in a short bore – one with L/D less than one – assuming the part has a sufficiently rigid wall," explained Huber. "However, as the bore length increases, it becomes more difficult to achieve good straightness and cylindricity. Among other reasons, the honing tool's sizing land tends to follow the bore. On cast-iron valve bodies with L/D of 10, the standard process has typically produced five microns cylindricity under optimum conditions. High-precision single- stroke honing takes this to two microns cylindricity or less. And with a tool life that can be 60,000 to 80,000 parts, the process is economical. A typical part suitable for this process might have a 16 mm bore, over 160 mm long, with 11 lands. This type of part can now be honed to less than two microns cylindricity, less than two microns straightness and one micron roundness."

The high-precision single-stroke process utilizes a combination of proprietary tool processing, tool holding, workpiece fixturing and process parameters, without any penalty on cycle time. It is already in use by manufacturers of hydraulic valves in Turkey, Italy, Germany, Switzerland, Brazil, Sweden and the United States.

Sunnen offers single-stroke honing on three different VSS-2 Series models that incorporate up to six spindles to progressively size and finish part bores. The machines are ideal for precision sizing of bores 3.9 - 50 mm (0.149 - 2.0") diameter in stamped parts, hydraulic valve bodies, gears and sprockets, parking pawls, rocker arms, turbocharger housings and similar parts. Ideal materials include cast iron, powdered metals, ceramic, glass, graphite and other free-cutting materials.

The VSS-2 Series 2 is available in three models – the 84 (eight-station, four-spindle), the 86 (eight-station, 6-spindle) and the 64 (six-station, four-spindle) – to meet various mid- to high- production needs. Spacing between spindles is 190 mm (7.48"). The 7.5 kW (10 hp) spindle drive provides a speed range of 100-2,500 rpm.

For more information:
Sunnen Products Company
Phone: (314) 781-2100
www.sunnen.com

GWJ Technology

INTRODUCES *TBK 2014*

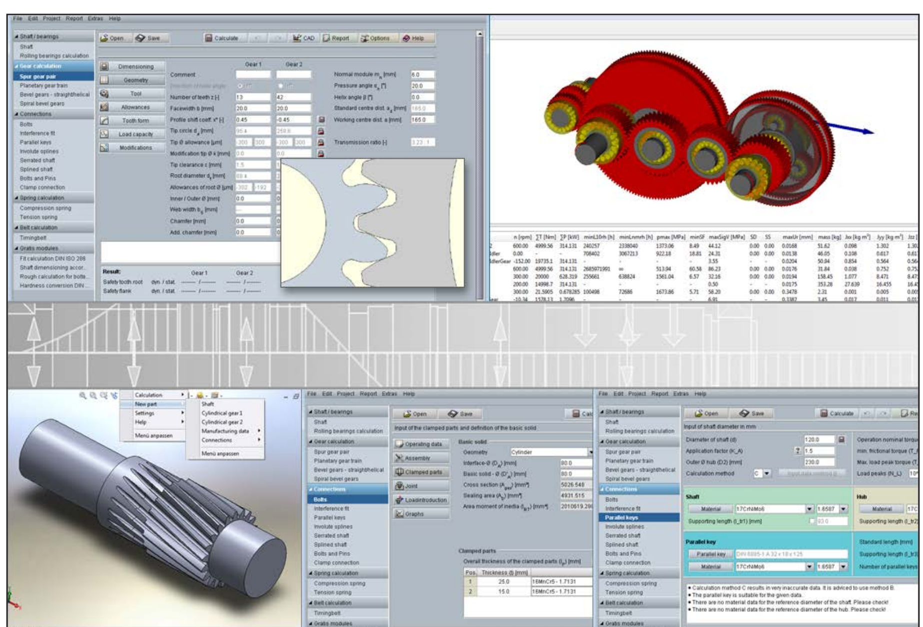
Software company GWJ Technology, headquartered in Braunschweig, Germany, recently introduced a completely new software generation of its *TBK* calculation software for gearboxes. *TBK* is standard calculation software for machine elements focusing on gear manufacturing. For more than 30 years, *TBK* software has been a widely accepted calculation software and is being successfully used by many engineers worldwide. The updated *TBK 2014* version offers a new and more modern look, but keeps the easy-to-use and intuitive interface. The software application comes with a completely new calculation technology, new calculation modules and numerous extension possibilities.

TBK 2014 is full of new features (e.g., calculation modules for bolted joints, bolts and pins, clamp connections, springs and timing belts) and refinements. Already existing modules have been improved significantly. For example, the calculation module for cylindrical gears provides a new dimensioning function. This functionality offers proposals for possible design variations. The module for involute splines supports not only DIN 5480, but also DIN 5482, ISO 4156, ANSI B92.1 and B92.2. *TBK 2014* allows a calculation with two or more bearings. Eigenfrequencies and critical speeds (bending and torsion) can be also determined. Powerful CAD plugins for different CAD systems are available in order to combine calculation and design as well as an output option for the 2-D DXF tooth form.

TBK 2014 software can be extended with the SystemManager add-on application. It allows for a fast and easy design of complete systems. By using the add-on, multi-stage cylindrical gears, planetary gear trains and bevel gear drives can be easily determined. Power-distributed transmissions and manual gearboxes with load spectra are supported as well. The calculation of eigenfrequencies of the complete system is also possible.

TBK 2014 provides a high level of comfort with a redo and undo function, automatical re-calculation, tool tips, user and project information, extensive user default settings, calculation reports in HTML and PDF format with individual report templates. The language of the software can be easily changed. In addition, a new and enhanced user manual with technical information and calculation examples is available.

For more information:
GWJ Technology
Phone: +(49) 531-129 399 0
www.gwj.de



RUF US, Inc.

OFFERS GEAR GRINDERS VALUE IN BRIQUETTING SYSTEMS

Gear and other grinding operations are finding the value they thought was lost in their grinding sludge. The bonuses of increased operational efficiencies and safer working conditions are realized once companies start using a briquetting system.

Corrugated Replacements, Inc., a developer of machine replacement parts for the corrugated board, steel, poultry and satellite industries, and Horsburgh-Scott, a provider of industrial gears and custom gear drives for steel, aluminum, tire, rubber plants and wind turbines, are just two of the many companies who looked to RUF Briquetting Systems to help squeeze value out of product they previously viewed as lost.

Just one year ago, managing monthly expenses of time, money and resources dealing with swarf produced in the grinding process was a reality for Corrugated Replacements. Before purchasing a briquetting system, dirty swarf was stored in barrels on the production floor, and then disposed of by a costly hazardous waste removal company. With the briquetting system, things have changed. "We were able to turn something that was a 100 percent expenditure process into a profitable revenue stream and reuse reclaimed coolant," said Corrugated Replacements plant and engineering manager Scott Wallis. "It has been a win-win for our business."

Similarly, the Horsburgh-Scott production process was positively impacted after the installation of a briquetting system in February of this year. "The briquetter has helped our operation reclaim more than 1,650 gallons of oil from our swarf," said Horsburgh-Scott manufacturing engineer Luciana Talpa. "The RUF machine works amazingly; it simplified the business and will pay for itself in less than a year."

Horsburgh-Scott's grinding sludge contained 63 percent oil by weight. The briquetting process is able to reclaim nearly all of the oil from this saturated grinding sludge. The reclaimed oil is then reused, reducing the amount of new oil purchases needed for their gear grinding process.

Reducing oil and coolant loss is just part of the added efficiency of the briquetting system. Both companies have also reduced or eliminated hazardous waste costs, turning it into a revenue stream by selling the briquettes as scrap to steel mills. Although it was not part of the initial goal, by improving operation efficiency and reducing expenses for wasted material and swarf, briquetting and reclaiming product has made for a safer work environment.

"The barrels where swarf was previously stored, waiting for disposal, created a heat safety issue," said Wallis. "Not only did we regain a significant amount of floor space, but we also eliminated the need to store hazardous waste."

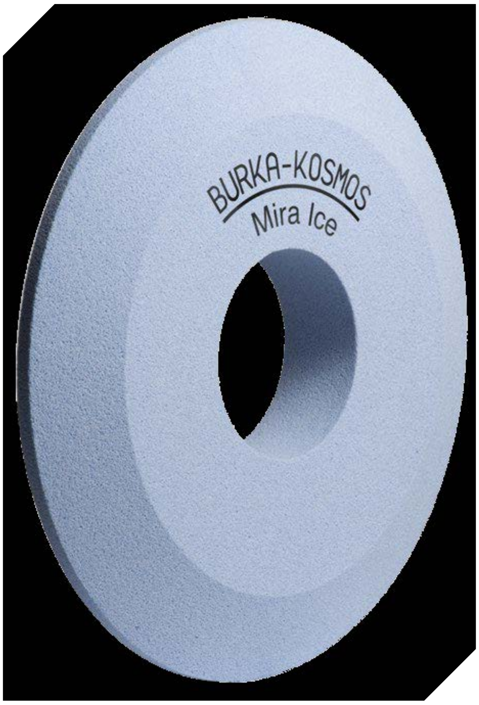
These two companies, along with others in the metal grinding industry, have improved efficiency, safety, productivity and developed a new revenue stream, simply by recycling their own product through the briquetting process.

For more information:
RUF US, Inc.
Phone: (440) 779-2747
www.ruf-briquetter.com

Burka-Kosmos

OFFERS NEW GEAR GRINDING WHEEL

The Mira Ice product line of gear grinding wheels was developed in order to meet the requirements of profile grinding larger gears. A new approach in the development of this grinding wheel enables results to be achieved in terms of economic efficiency and cool grinding that were previously unheard of in the profile grinding of gears. The Mira Ice grinding wheel has been further optimized on the basis of the leading product line from Burka-Kosmos. The use of new grain qualities and the new high-strength bond system guarantees particularly cool grinding. Longer cycle times between dressing and shorter grinding times significantly reduce the processing costs per gear. During testing against the Burka-Kosmos SK23 ceramic grinding wheel stock removal volume was increased by almost 400 percent on a 3.6 diametral pitch gear. This increase, along with the need for fewer dressing cycles, resulted in a 30 percent reduction in total machining time.



For more information:
JRM International, Inc.
Phone: (815) 282-9330
www.jrminternational.com

Mitutoyo America

MICROMETERS IMPROVE MEASUREMENT CAPABILITIES

Mitutoyo America is now offering enhanced versions of its popular Mitutoyo IP65 Coolant Proof and MDC-Lite Micrometer line with exclusive, newly developed Absolute sensor technology. This patented electromagnetic sensor offers improved measurement dependability by increasing resistance to harsh workshop conditions. This re-launch of a product line that has become a global industry standard is an exciting development for Mitutoyo and offers improved performance and reliability to its customers.

Highlighted features of the new Mitutoyo IP65 Coolant Proof and MDC-Lite Micrometers include the new electromagnetic Absolute sensor which provides improved resistance to environmental conditions such as dirt, oil and water that can cause false readings; a function lock system to prevent inadvertent setting changes; new ratchet thimble models that improve one-hand operability and an anti-slip finish for an improved grip while taking measurements. The micrometers will offer long battery life - 2.4 years under normal operation and SPC output models will be fully compatible with existing Mitutoyo data management accessories. These new micrometers will replace previous IP65 Coolant Proof and MDC-Lite models and will be introduced at the same prices as the models they will replace.

For more information:
Mitutoyo America Corporation
Phone: (888) 648-8869
www.mitutoyo.com

GF Machining Solutions

INTRODUCES MODULAR AUTOMATION SYSTEM

GF Machining Solutions has announced the System 3R Transformer, a modular automation system designed to integrate with a wide variety of technologies. With Transformer, manufacturers can begin by simply automating a single machine and then easily expand to include up to 12 machines within the cell.

A Transformer cell accommodates a wide range of machining technologies and allows components from different manufacturers to be included within the same call. This open architecture approach allows end users to create an automation cell where each individual component decision is optimized, as opposed to having choice constrained by compatibility.

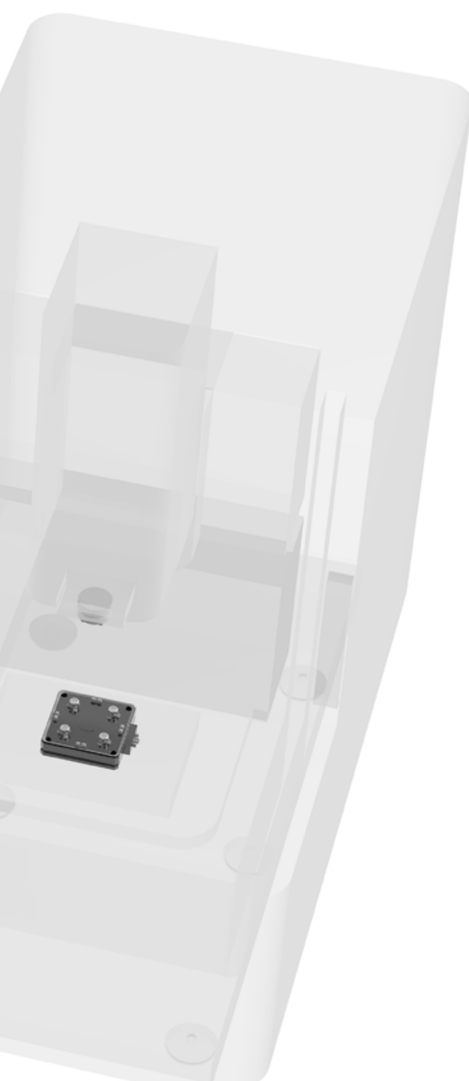
GF Machining Solutions offers wide range of System 3R tooling systems that allow palletization of workpieces and electrodes of a tremendous range of sizes. The Transformer system can handle all of these, as well as different tooling systems, allowing it to be incorporated into production systems with existing tooling systems.

For maximum flexibility, Transformer offers additional in-process accessories that can easily be integrated into a cell, including:

- Multiple loading station that can be used without stopping production
- Draining station for emptying workpiece cavities after machining processes
- Washing machine to clean workpieces
- Coordinate measuring machine (CMM) for pre-setting and/or part inspection

The Transformer system provides user-friendly cell management software whereby all production data is entered in a structured manner or imported through a data exchange interface for major enterprise resource planning (ERP) systems. The software then controls every aspect of the cell, from automatically loading jobs to machines to recording and monitoring cycle times for each job. The core of the system is an efficient database that uses chip identification of the pallets to ensure that correct data is used for every part in the cell.

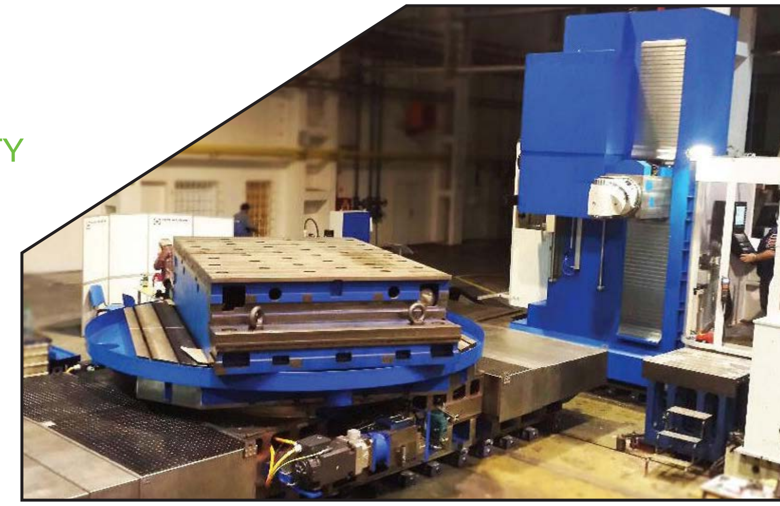
For more information:
GF Machining Solutions
Phone: (800) 282-1336
www.gfms.com/us



Havlik Gear

EXPANDS PLANT CAPACITY AND ABILITY

Havlik Gear, a Renaissance Power Systems company, announces the addition of a TSO FUQ 150-VR6 CNC machine to expand plant capacity and ability. The multipurpose CNC floor type mill has an impressive 60 metric ton table capacity. The custom designed hydrostatic table has a maximum rotational speed of 80 rpm. In addition to general purpose heavy duty turning and milling, large gears can be rough cut and finish cut using one of four custom machining heads that best suit the application.



Additionally, the new machine will increase the company's gear cutting capacity from eight meters to 12 meters, allowing it to undertake some of the largest gearing projects in the world. Products such as multi-segment ball mill, kiln, or SAG mill gears, as well as riding rings or mill heads, can be supplied on a turn-key or subcontract basis to best meet customer needs.

The addition of the new CNC machine also will allow Havlik to provide vertical lathe capability as the machine doubles as a large vertical lathe, able to swing 12 meters in diameter. It also will provide the ability to support subcontract large diameter turning, which will be a major focus for the company moving forward. "Not only will this new machine – one of only 2 or 3 in the world – allow Havlik to take on large gearing projects, but it also will increase capacity as many products can be more efficiently produced, including gearbox housings, press frames and components, compressor components and more," said Mark Readinger, president and CEO of Renaissance Power Systems. Havlik Gear, an ISO9001:2000 certified company, has been a leader in gear manufacturing since 1886.

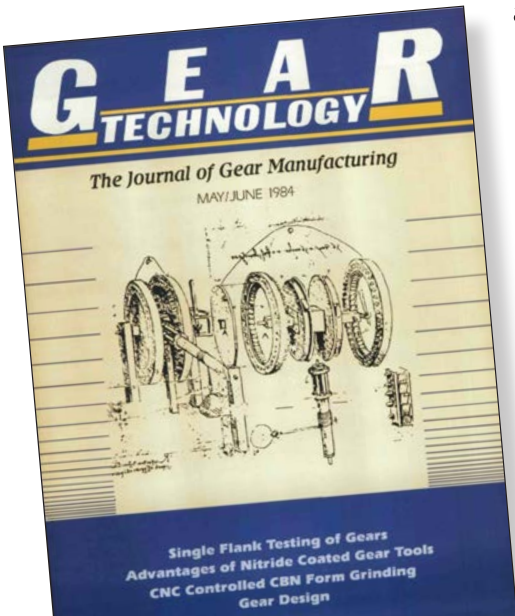
For more information:
Renaissance Power Systems
Phone: (414) 732-2400
www.renaissancepowersystems.com

The Past, Present and Future of Gear Manufacturing

Celebrating 30 Years of Gears

Matthew Jaster, Senior Editor

The gear industry is full of storytellers. It's a niche market that boasts a remarkable cast of characters that have been sharing their stories with us for 30 years. In that time, the editors and staff of *Gear Technology* magazine have had the privilege to report the ins and outs of this highly-specialized industry. From technical articles to case studies and features, the main focus of this magazine has been to "provide a forum of discovery and innovation for you, the gear manufacturing industry." Our Publisher, Michael Goldstein, said as much in our inaugural issue of May/June 1984.



For our 30 year celebration, we turned it over once again to our advertisers, contributors and extended *Gear Technology* family to discuss the past, present and future of this exciting and innovative market. While the information both readers and advertisers have contributed over the years has been instrumental to the success of both *Gear Technology* and its "younger sister" publication, *Power Transmission Engineering*, I'd argue that the "off-the-record" stories are quite possibly more interesting in the grand scheme of things. Getting to know the people and the personnel responsible for making wind turbines spin, helicopters fly, cars shift and rovers rove has been equal parts educational and entertaining.

"Whenever someone wants to connote mechanical precision and complexity, invariably they use among other things, imagery of gearing," says Alan R. Finegan, director of marketing, Gleason Corporation. "Gears are mathematically complicated to design and are relatively difficult to manufacture. The final analysis of their quality and performance is not as a single component but as part of a moving mechanical device. Despite numerous innovations aimed at eliminating gears and all their complexities, gears remain as key drivers of power and motion *simply because they work*."

How has technology changed in gear manufacturing in the last 30 years?

"In the days before high-technology machine tools, workers typically were required to have approximately seven years of experience to be optimally productive, with top pay. The machine tools of today allow workers to be highly productive without the same level of experience. Along with technology innovations, over the past 30 years the quality assurance paperwork has quadrupled. As an example, for some of the aerospace gears we manufacture, the paperwork outweighs the parts."

Joseph L. Arvin, President and COO, Arrow Gear

"Machining technology and cutting tool technology have been nipping at each other's heels over the last thirty years. Driven by base materials, PVD coatings, CBN grains and grinding wheel technology, the speeds in which these tools can be operated have increased exponentially. Machine tool technologies have been forced to keep up with these changes by utilizing CNC controls, high-speed linear guide ways, high-precision ball screws and highly precise, high-speed, direct-drive work spindles."

Thomas Kelly, Senior Vice President-Machine Tool Division, MHI America Inc.

"Everyone is aware of the inroads lean manufacturing has contributed to improved productivity and faster changeovers. Even though this has been going on for years, many of us are still only in the beginning stages of implementation. In order to keep up with our foreign competition's cheaper labor rates, we must incorporate lean in everything we do in manufacturing."

Fred Young, CEO, Forest City Gear

"The fundamentals haven't changed in that quality is still directly related to runout of the tool and the workpiece. What has changed dramatically was humorously summarized by an engineer who offered that 'poor quality can now be produced incredibly fast.'" Of course quality is improved as well as productivity due to the precision of modern CNC machines."

Bill Miller, Vice President, Kapp Technologies

"Computers and gear grinders! In 1984, we were just getting desktop PCs and commercial software. Gear grinding was an exotic aerospace/high-speed technology."

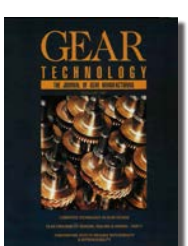
Charles D. Schultz, Beyta Gear Service and *Gear Technology* Technical Editor and Blogger

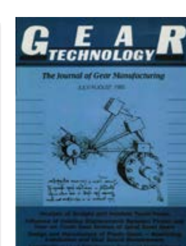
"(There has been a) significant growth of design software. Early on, most design software was generated within a gear company, and so its underpinnings and theory were well understood by the users, as they were also the program developers. Today, some believe that is lost on engineers that use off-the-shelf design software."

Joe Mihelick, consultant and Gear Technology technical editor

"The evolution of CNC gear manufacturing machines, as well as the CNC gear inspection instruments (have been significant). It is now possible to do higher speed cutting with carbide tools, as well as the ability to "hard finish" gears. These much simpler mechanisms have increased productivity as well as quality, and the modern CNC gear inspection instruments have also provided much better computer data analysis in line with the newer gear accuracy standards."

Robert E. Smith, gear consultant





What are the key technologies/innovations driving the gear industry today?

"Although basic manufacturing technology has not changed much, process technology has changed a great deal. Changes in manufacturing are in the widespread use of CNC technology in all phases of manufacture, as well as much wider use of robotics and automation. Demand for better quality gears has increased the use of gear grinding also. Most manufacturers are now employing lean manufacturing principles as well as manufacturing cells."

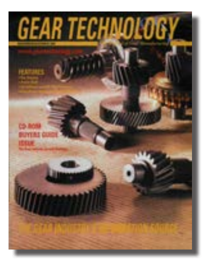
Bipin Doshi, President, Schafer Gear Works, Inc.

"A new generation of fearless manufacturing engineers is accelerating implementation of tool and machine technologies. Technologies (that) evolved over the last 10 years naturally take time to reach full potential, and perhaps to a greater extent in our industry, due to inertia of traditional conservative practices."

Bill Miller, Vice President, Kapp Technologies

"In general we are seeing reduced cycle times and lower-cost-per-component. The competitive landscape of machines and tools can also be seen in gear manufacturing today where there's more competition between conventional gear machining and multi-axis gear machining."

Mats Wennmo, Senior Technical Manager, Sandvik Coromant



30 Years In, We're Still Here

By Jack McGuinn, Senior Editor

Thirty years is not a particularly long history for most industries. Take the gear industry; its beginnings go waaay back—like, B.C. back.

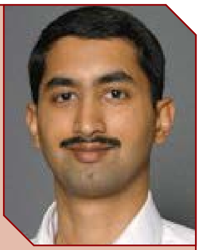
Regardless of perspective, quite a bit has happened in the last three decades in the industry, and this "whippersnapper" of a gear industry journal has been there right along to report on the who-what-when-where of it. And it has been quite a ride.

Following is a distillation of comments offered by individuals who have enjoyed long, successful careers in the gear industry — certainly long enough to back up any opinion they may offer as to what constitutes the "magic moments" occurring in the gear industry from 1984 to present. Their observations will be plotted throughout the Timeline near the date in time cited in their opinions.

As you will see, each person offered up more than a few milestone-moments, although their "homework assignment" only asked for one or two. But let's face it: that's like being asked to name your favorite movie or your favorite child — can't be done.

As you time-travel your way along, you are of course most welcome to disagree with our contributors' choices; or if there is a glaring omission in our Timeline, please don't hesitate to let us know at *geartechnology.com*.

“I believe that power skiving will be extremely beneficial in the high-volume production of ID/OD gears, and multi-tasking machines are going to change the way small- to medium-volume gear manufacturing is done. Multi-tasking machines are doing many of the tasks previously done by special-purpose machines using sequential operations. This trend is seen not only in gear manufacturing but manufacturing in general.”



Nitin Chaphalkar, Manager, Advanced Solutions, DMG Mori ASI

“In 2014 the continued improvement of form grinding machines is important, but not as important as the change to 3-D design software and improved thermal processing.”

Charles D. Schultz, Beyta Gear Service and *Gear Technology* Technical Editor and Blogger

“There are some interesting advancements in the forging industry whereas gears that have traditionally been machined are now forged to a net or near-net shape. Heat treating still remains to be a challenge which many companies are addressing.”

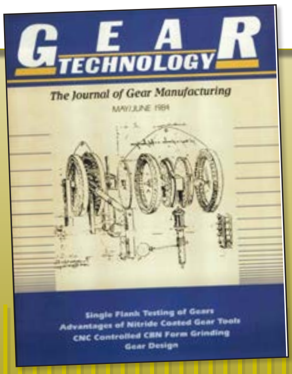
Thomas Kelly, Senior Vice President-Machine Tool Division, MHI America Inc.

1984

The publication of *Dudley's Handbook of Practical Gear Design*

1984

Gear Technology debuts.



Mid-1980s

From **Dr. Robert Handschuh**, *a 30-year NASA veteran with invaluable experience in DOD rotorcraft drive system analysis and experimental methods.*

Through much of the entire **1980s**, “(Development of) coordinate measurement machine use for spiral bevel gear manufacture (Army-funded, Sikorsky Aircraft contractor, early **1980s**); Applying CNC technology to spiral bevel gear manufacture on Gleason 463 gear grinder led to full CNC Phoenix machine line (Army-funded, Bell Helicopter contractor, mid-to-late 1980s); Linking the two above items together improved gear quality and reduced production costs; Face gear development from basic research—by Dr. Faydor Litvin and many Ph.D. students, as well as the Army-funded Adv Rotorcraft Transmission Project led by then-McDonnell–Douglas Helicopter company (now Boeing–Mesa), included technology development of gear load capacity at NASA Glenn to use in the Apache Block III Helicopters now being produced.

1985

The software *Load Distribution Program* (*LDP*) is first distributed to OSU GearLab sponsors, which now has over 1,000 license holders; OSU writes papers on incorporating *LDP* microgeometry analysis into many gear design analysis procedures, resulting in a great reduction in the noise of automotive transmissions.

How will technology in gear manufacturing change in the next 30 years?

"Continuing changes in materials, including plastics and carbon fiber. A continued emphasis on designs and processes for ever-quieter gear sets, including more and improved fine finishing processes, integration of system solutions as part of smart factories. In 30 years there is no way to predict the needs of the applications and end users, much less the manufacturing technology. That's what makes the role of a solutions provider challenging and exciting."

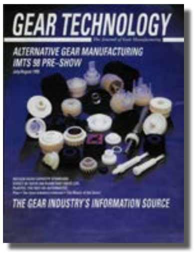
Alan R. Finegan, Director of Marketing, Gleason Corporation

"We will see even more flexibility in gear design and gear machining, as well as increased power density and a considerable change in end user sustainability. Another advancement we'll see is improved end user control and influence in gear design/gear machining ratio."

Mats Wennmo, Senior Technical Manager, Sandvik Coromant

"Of course new turning equipment has been appearing for years that incorporates gear cutting in the blanking phase. This is especially beneficial to those with little gear work and little time to satisfy demands for faster throughput. Just as hard turning is a supplement for cylindrical grinding, I believe these equipment developments will continue to flow down the processing chain. I foresee the addition of gear inspection on this equipment to satisfy the need for quality documentation just as it has been implemented on gear grinders."

Fred Young, CEO, Forest City Gear



1985

From **Dr. Hermann J. Stadtfeld,** *Gleason Corporation vice president, bevel gear technology, R&D.* By **1985** coordinate measurement of bevel gears with electronic master (nominal data file) including machine corrections is developed by Gleason & Zeiss; From **1989-1995**, single flank technology becomes the world standard in bevel gear quality evaluation (T20 by Oerlikon and 500HCT by Gleason); Thee first bevel gear dry cutting process is introduced in **1997** at EMO in Hannover, Germany by Gleason; From **1998–2005** bevel gear grinding in automotive becomes a worldwide standard and replaces lapping widely (Gleason and Kingelnberg); The first bevel gear cutting machine without machine bed for first true dry chip removal (Phoenix II by Gleason) is introduced (**2001**) at EMO in Hannover.

1987

From **Dr. Donald Houser**, *Professor Emeritus at Ohio State University; Houser is the founder of the Gear Dynamics and Gear Noise Research Laboratory (GearLab), an industrial research consortium with 45 participating companies.* The **1987** PhD thesis of Ohio State University graduate student Sandeep Vijayakar provided the basis for his *Calyx* contact analysis algorithm that has been used by numerous companies to solve contact problems of multi-mesh gears.

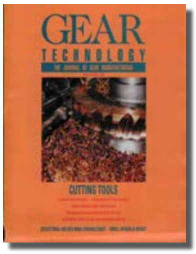
1988

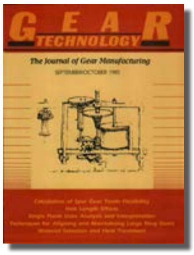
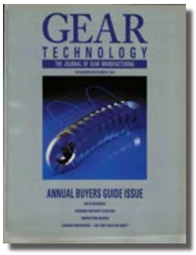
From **Alan R. Finegan,** *director of marketing for the Gleason Corporation.* Gleason Corp. introduces Phoenix (**1988**), the world's first 6-axis CNC bevel gear cutting machine, a genuine game-changer that rocked the foundations of bevel gear manufacturing.

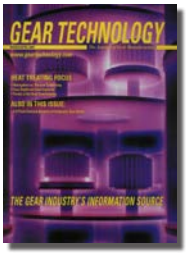
"Hybrid manufacturing is a big trend on everyone's mind. While it is hard to imagine that process being used for production volumes, I think it definitely will change the design of parts and make the designs more optimized. It will reduce the product development times. On the heat treatment side, I think laser heat treatment methods offer the best promise. These methods enjoy a strong advantage due to selective hardening and energy efficiency point of view."
Nitin Chaphalkar, Manager, Advanced Solutions, DMG Mori ASI

"I see non-traditional rough cutting as the emerging technology of the next ten years. Between wire cutting, 3-D printing and CNC milling, companies will have alternatives to hobbing and bevel cutting."
Charles D. Schultz, Beyta Gear Service and Gear Technology Technical Editor and Blogger

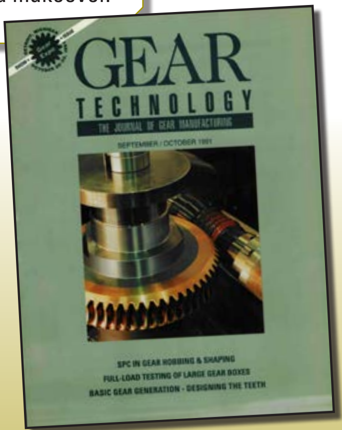
"Just as we've seen computer technology bring about sweeping changes in the past decade, I believe this will continue to evolve. I foresee an increased use of robotics in the gear industry, as well as multi-purpose machine tools."
Joseph L. Arvin, President and COO, Arrow Gear



1991
Gear Technology gets a makeover. Our first re-design.



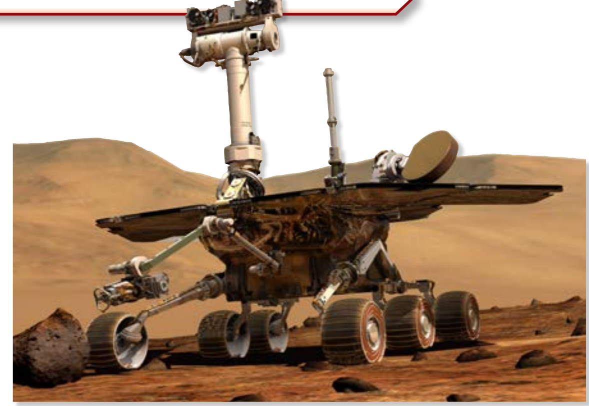
1993
AGMA takes over as Secretariat of ISO TC-60.

1995
Gear Technology re-design #2. "Addendum" column is first introduced.
Gleason acquired Carl Hurth Maschinen und Zahnradfabrik (**1995**).

What sets the gear industry apart from other areas of manufacturing?

"Our current gear industry is anything but stagnant and will continue to offer many opportunities for aspiring engineers to reach levels of productivity and quality unimaginable today. I feel truly rewarded to have rubbed shoulders with brilliant people in our industry who for years have advanced the state of the art. I would remind us that whenever we think there is no room for improvement, our children (who don't know what is currently impossible) will figure out a way to (overcome) the current limitations we have."

Fred Young, CEO, Forest City Gear



"Technically, one can still draw on and apply lessons learned 30 years ago and nearly every day brings a new challenge. Personally, the relationships developed can flourish for an entire career as the people in our industry are generally not transient."

Bill Miller, Vice President, Kapp Technologies

1996
The Gear Industry Home Page is launched at *www.geartechnology.com.*

2003
E-GT (Electronic Gear Technology Magazine)
Launched with the January/February printed issue.

2005
Gear Product News begins publishing in alternating months with *Gear Technology* (*Gear Product News* ceased publication after 2006).

1997
The Power Transmission Home Page is launched at *www.powertransmission.com.*

"Custom manufacturing in the gear industry, vs. large OEMs' involvement in production of gears, is a small and intimate network. It is rewarding to be working in a competitive environment with like-minded people. Niche markets do reduce cutthroat competition, yet offer an exciting place for innovation and improvements."

Bipin Doshi, President, Schafer Gear Works, Inc.

"I feel that gear manufacturing offers the highest level of metal cutting, and this industry produces gears with a high standard of quality that is simply amazing. It is very rewarding to be a part of an industry which is trying to achieve what was once considered impossible and always pushing the boundaries."

Kenneth Sundberg, General Manager Gear Milling Solutions at Sandvik Coromant

"The gear industry is a special and unique industry with its AGMA platform. This platform allows everyone to participate and learn from each other. The camaraderie between rivals seen in this industry is unparalleled. This also comes from the good natured people and talented engineers that work in the gear industry. I have been involved in AGMA activities for the past two-three years and everyone in the industry has always treated me as family."

Nitin Chaphalkar, Manager, Advanced Solutions, DMG Mori ASI

2007

Power Transmission Engineering magazine is born. *Gear Technology* redesign #3, and increased frequency to eight issues per year.

2013

Gear Technology re-design #4—It's tough getting old!

2014

30 years and we're still here!

Development of Wind Turbine Standards

By Robert Errichello, Owner, Geartech and Gear Technology Technical Editor)

"My most memorable story begins in the early 1980s, when the wind turbine industry started in California. Many of the wind turbines were Danish machines that were designed to operate in the perfect wind environment of the steady, non-turbulent winds at the Danish coastline. When they were installed in the California mountain passes of the Altamont, Tehachapi, and Palm Springs, the wind turbines were subjected to severely turbulent winds. Consequently, there were thousands of failed gearboxes literally overnight. Some users made the mistake of designing their own gearboxes or purchasing off-the-shelf gearboxes that lasted no longer than the European gearboxes.

I had finished my studies at the University of California at Berkeley in 1978 and was still living in Berkeley. Suddenly, I received telephone calls from all three wind sites from frantic people who couldn't get their government tax credits because they couldn't keep their wind turbines running due to the gearbox failures. For many years up to the mid-1990s, my partner, Jane Muller, and I traveled to all three wind sites and inspected and investigated the gearbox failures. We also traveled to Europe to visit wind turbine manufacturers and gear manufacturers to resolve the gearbox failures.

On September 23, 1991, during the American Wind Energy Association (AWEA) Windpower Conference in Palm Springs, California, the following people met to discuss gearbox failures:

Robert Errichello (Geartech), Brian McNiff (McNiff Light Industries), Jane Muller (Geartech), Walt Musial (NREL), and Brent Reardon (MSA).

At this ad-hoc meeting it was decided the participants' collective experience should be documented. Therefore, in October 1991, I asked the AGMA if an AGMA committee could be formed in cooperation with AWEA. Approval was subsequently granted and the first meeting of the AGMA/AWEA Wind Turbine Committee convened on October 19, 1992. Brent Reardon was elected chairman and Jane Muller was elected vice chairman.

In September 1994, Geartech completed NREL Report NREL/TP-442-7076: *Application Requirements for*

Wind Turbine Gearboxes (Ref. 1). The report documented Geartech's experience with wind turbine gearboxes and gave guidelines for selecting, designing, manufacturing, procuring, operating, and maintaining gearboxes for use in wind turbines. The report was accepted by the wind turbine committee as a working draft.

The wind turbine committee met ten times between 1993 and 1996. In 1996 they submitted their draft standard to AGMA for approval. In October of 1997 AGMA published

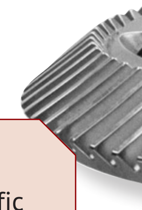
AGMA/AWEA 921-A97: *Recommended Practices for Design and Specification of Gearboxes for Wind Turbine Generator Systems* (Ref. 2). The wind turbine committee set a record for the shortest time from proposal of an AGMA standard to approval and publication.

In 1999, work began on the revision of AGMA/AWEA 921-A97. Because information sheets are less legally binding than standards, the committee decided to upgrade the document from an AGMA "Information Sheet" to an AGMA "Standard." They changed the document title to ANSI/AGMA/AWEA 6006-A03: *Standard for Design and Specification of Gearboxes for Wind Turbines* (Ref. 3). The wind turbine committee met nine times between 1999 and May of 2002, and the document was finally published in 2003. Once again, the committee set records for the first truly multi-discipline AGMA committee and the highest number of participants of any AGMA committee.

The knowledge gained from our failure investigations and our work in designing the wind turbine standard proved to be a very enjoyable experience for Geartech, primarily because we met so many wonderful people who were passionately committed to making wind turbines a viable source of renewable energy. Many of the people we met are now our close friends and we meet often to share colorful stories about our wind turbine experiences."

References

1. Errichello, R., and J. Muller. "Application Requirements for Wind Turbine Gearboxes," NREL/TP-442-7076, Sept., 1994.
2. AGMA/AWEA 921-A97: *Recommended Practices for Design and Specification of Gearboxes for Wind Turbine Generator Systems*, 1997.
3. ANSI/AGMA/AWEA 6006-A03: *Standard for Design and Specification of Gearboxes for Wind Turbines*, 2003.



Share some of your most memorable moments in the gear industry:

"When MHI introduced their Superdry gear hob, it totally blew my mind. The tool was made from HSS, had a proprietary PVD coating and was cutting gears at 200 meters/min without the use of coolant. They broke all the rules that I was taught about cutting tools."

Thomas Kelly, Senior Vice President-Machine Tool Division, MHI America Inc.

"Many may not remember the origins of the AGMA's Gear Expo, but I was the one who first approached AGMA's Board of Directors in 1981, asking them to hold a trade show specific to the gear industry. Initially, they thought I was crazy, stating there would not be enough attendees for such a show. But my perseverance paid off; approaching them again in 1984, this led to the first show in 1985. The show was a table-top exhibit and was held in Chicago in the fall of 1985. Today, Gear Expo is now the leading show for the U.S. gear industry and is a major source of financial assistance for the AGMA."

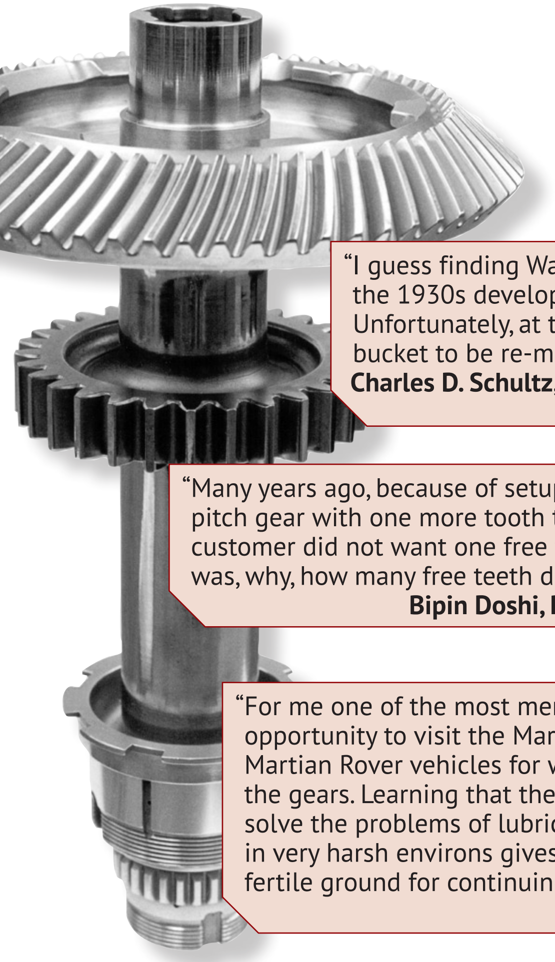
Joseph L. Arvin, President and COO, Arrow Gear

"I guess for a story I can only point to my own company. In 2015 Gleason will celebrate its 150th Anniversary, and I have spent a good number of years (more than I care to admit) here. Stepping back, I continue to be amazed by how Gleason has maintained a position of global leadership through so many years, adapting to change and thriving."

Alan R. Finegan, Director of Marketing, Gleason Corporation

"The journey working with indexable hobbing tools for small modules has been very eventful, and we have many unique and interesting stories from these projects. One specific case I recall very well is the time we implemented an indexable CM 176 hob on an old machine built around 1950 and reached a savings in machining time of 80 percent at the required level of quality. This was an amazing result for the owner of this small sub-contractor, giving him quite a competitive advantage and opening up new business opportunities for Sandvik Coromant."

Kenneth Sundberg, General Manager Gear Milling Solutions at Sandvik Coromant



"I guess finding Walter Schmitter's giant gear tooth with strain gages still intact from the 1930s development of AGMA's bending strength formula remains a high point. Unfortunately, at the direction of my supervisor –Walter's own son- I put it in the scrap bucket to be re-melted into another Falk product."
Charles D. Schultz, Beyta Gear Service and Gear Technology Technical Editor and Blogger

"Many years ago, because of setup error, we shipped a very fine pitch gear with one more tooth than the print called for, and the customer did not want one free tooth! Our foreman's comment was, why, how many free teeth do they want?"
Bipin Doshi, President, Schafer Gear Works, Inc.

"For me one of the most memorable experiences was the opportunity to visit the Mars Yard at JPL where they test the Martian Rover vehicles for which our company helped build the gears. Learning that the engineers are quite keen to solve the problems of lubrication and longevity of the gears in very harsh environs gives me hope that the USA will be a fertile ground for continuing development of our industry."
Fred Young, CEO, Forest City Gear

"I suggested to the chief engineer of our company, Manfred Lorenz, that perhaps our future business might be weighted towards non-gear components. His reply was 'Most certainly not. In fact, gear grinding will be more cost-effective than shaving.' That was in 1992. Quite prophetic, we now know."
Bill Miller, Vice President, Kapp Technologies

Closed-Loop Bevel Gear Development

Spiral bevel gears never had it so good

Joseph L. Arvin, President and COO, Arrow Gear

I would have to say that the most significant change to the gear industry during the 30 years of Gear Technology has to do with the evolution of computer-based design, development, manufacturing, and inspection of gear teeth.

Here is a brief background. The main challenge in producing high-precision gears is designing the gear teeth so the contact pattern of the meshing gear teeth will be correct when the gears experience various gearbox deflections during loaded operation. Historically, the process for getting this correct was that of trial-and-error; a process that would take many months and cost hundreds of thousands of dollars — especially in aerospace applications.

Then, in the mid-1980s, Dr. Dieter Wiener of West Germany developed and manufactured the first CNC spiral bevel gear grinding machine tool, which was later purchased by Klingelnberg. In 1988, I was invited to watch the operation of this original machine in Ettlingen; after viewing this technology in operation I was extremely impressed and I knew this had to be the future for the gear industry.

Meanwhile, Gleason was working with Zeiss in the development of software which could interface between CNC inspection systems and CNC tooth cutting and grinding machine tools. This software could send correction settings directly to the machine tool.

By the early 1990s, using Gleason's software and their new CNC Phoenix tooth cutting and grinding machines, Arrow Gear worked in developing and implementing the world's first hard- wired, integrated, closed-loop system for the design, manufacturing, and inspection of spiral bevel gears.

Using Gleason's software packages allowed Arrow's engineers to predict tooth contact pattern under various displacement conditions. This software also allowed for any settings changes required for the machine tools to be made automatically. By the mid-1990s Arrow's successful implementation was online and this allowed for the trial-and-error process to be eliminated.

Adoption of this new technology by the gear industry came slowly. Today, this capability is the standard among gear machine tool producers and is utilized extensively by gear makers. But I feel that this approach to high-precision gear manufacturing ushered in a new era for our industry and remains as the single most significant evolutionary step in gearing over the past 30 years.

Several years after this closed-loop system was successfully implemented, *Gear Technology* reported on Arrow's capability for advanced design and development in its January/February 2003 issue. I'm sure this article can provide you with all the details you would need.

Thank You

to all the companies who have supported us for 30 years...


Pfauter & Lorenz. Powerful flexibility for any size job.
Now closer than ever... Shave tools made in Oak Park, MI!
MORE BITES FOR YOUR BUCK
Fellows — always on the cutting edge of technology.
FELLOWS
CELL
THE ANATOMY OF A GEAR PRODUCTION SYSTEM
automation
FLEXIBLE
SYSTEM
QUALITY
Gleason
G-TECH holds the lead at high speeds.
GLEASON
Unbeatably fast.
Our new gear grinding machines.
HELIX
Productive
Flexible
Easy
Accurate
Economical
HÖFLER
KAPP
The best is now available.
LOVE AMERICAN STYLE.
KOEPFER GOES DIRECT
NEW UNITED STATES FACILITY
TECHNICAL CALENDAR

30 Years of Calculation

Examining the history of software in mechanical engineering

Dr. Stefan Beermann, CEO, KISSsoft AG

The Early Days

Calculating strength has always been part of engineering. However, the introduction and widespread use of powerful computers has brought about significant changes to the character of these calculations. In the beginning, computer-aided sizing primarily involved Finite Element (FE) modeling. One of the very first uses of these newly available tools was to analyze the NC machines used to manufacture the Apollo modules: NASA required proof that the machines were rigid enough to satisfy their stringent requirement for accuracy. This proof was provided by an FE calculation. In the 1970s, FE methods were implemented for gear problems. At that time, these types of calculations usually took 24 hours, running on a VAX 6400 mainframe.

Development of Standards for Calculation Methods

Although machine element calculations are, in themselves, less time-consuming than FE modeling, there was also a developing trend towards using computers to run them. As customers, inspection organizations, or government bodies often required strength verification for parts, it was essential that the strength calculation programs could perform these calculations according to recognized methods. These methods included ISO and DIN standards, VDI guidelines, and the approved technical literature. However, with regard to technical literature, a distinction must be made between well-defined calculation methods such as in Niemann/Winter (Ref. 1) and methods that are better suited for rough calculations such as Roloff/Matek (Ref. 2).

Although DIN standards have a long history (for example, the first edition of DIN 5481 dates from 1940), they addressed geometry exclusively. The first standardized strength calculation was DIN 3990 (December 1970), which described a method for calculating the strength of cylindrical gears. This was the first, crucial step and was followed (albeit very slowly) by other calculation standards. It took another 30 years (DIN 743 in 2000) before the first comprehensive regulations for rotating shafts were issued.

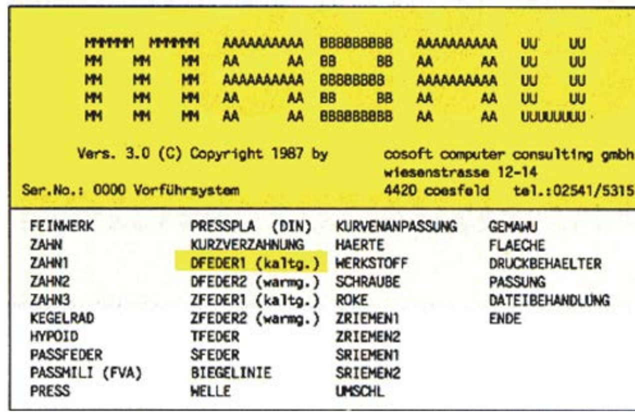


Figure 1 Selection menu from a classic calculation program.

One of the main obstacles preventing the implementation of calculation standards in a program of sufficiently high quality was the problem of how to display the calculation rules. As the guidelines were originally designed for engineers, who worked with slide rules and calculated proofs by hand, most of the data was provided in diagrams (sets of curves) so it could be read easily. The associated formulae were never specified, which meant that programming these curves took a great deal of time and effort — because the data had to be extrapolated from these diagrams and interpolated with higher-order polynomials. The approach taken by simpler programs — in which the user had to read the values from the diagram and then enter them in the program — cannot be considered acceptable.

From 1985 onwards, the standards did become more programming-friendly, because they listed the formulae along with the diagrams. This also made it possible to implement the standards more accurately and reliably in software programs.

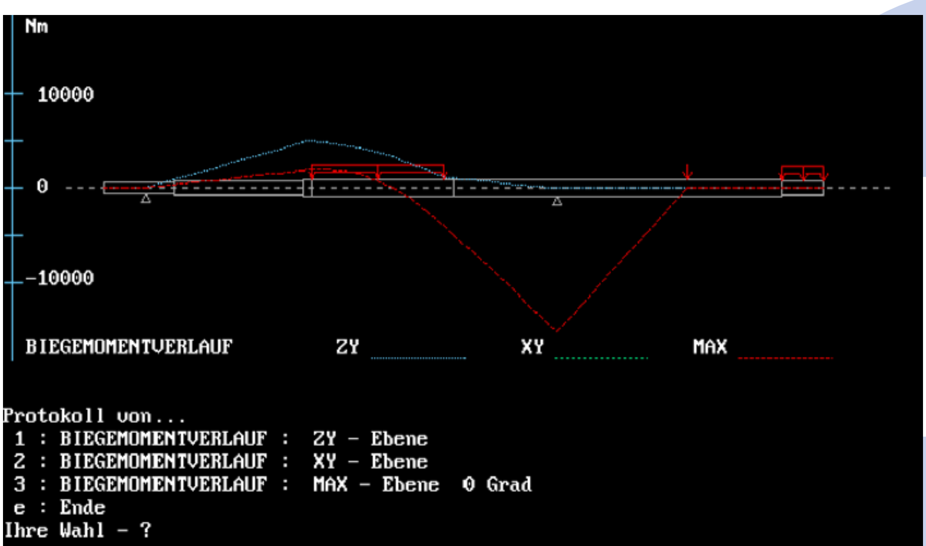


Figure 2 Typical graphics output of the DOS era.

DOS Programs

The first calculation programs that worked with the machine element concept were in-house developments by large companies, designed to meet their own requirements. These companies generally used the computers and programming languages they had available at the time. Commercially developed programs only really became widely available with the introduction and widespread use of IBM-compatible PCs. These generally ran the DOS operating system, with *BASIC* and *PASCAL* as the most commonly used programming languages. For example, *KISSsoft* was originally created on a Commodore PET, in *BASIC*, for in-house use by the company L. Kissling & Co. AG. However, before the program was released for general sale, it was ported to *Quick-BASIC* running under DOS.

One of the first commercial programs, *Mabau*, from the German company CoSoft, was also initially developed for Commodore computers (Commodore 8000). The first version of the gear calculation program had three parts: presizing, strength, and geometry, because it did not have enough memory to run a program that handled everything. Its user interface (Fig. 1) recalls the days of mainframe computers.

The 1980s saw a great expansion in the number of programs, because the demands made on user interfaces were still relatively low. Many engineers had learned programming during their studies, most of them probably running in batch on mainframes. They were then able to apply this knowledge to programming in the PC development environment (which was almost unbelievably convenient, in comparison to mainframes), and create solutions to a multitude of problems in mechanical engineering.

A feature of these early programs was the sequential querying of input data, followed by the question "Input correct? (Y/N)" which either triggered the calculation or sent you back to the start to input all the data again. Even then, one of the handiest functions was that you could simply press <Return> to confirm your entries. Programming a plausibility check for these entries was relatively easy, because the programmer could tell which entries were available at any stage in the program. However, it was not possible to interrupt a predefined process. This was very inconvenient, because it meant all the additional data required at the end of a query sequence had to be input again, starting from the very beginning.

First Borland and then Microsoft implemented support for window technology under DOS. However, the more important mechanical engineering programs did not implement this technology extensively. Development in this area was stopped by the dawn of the Windows programming era.

In the German-speaking world, the most widely used programs in this period (1985-1990) were *Mabau*, *Hirnware*, *TBK*, *Hexagon* and *KISSsoft*.

Porting to Windows

Porting to Windows was a real milestone in the evolution of every calculation program in use today. Up to the mid-1990s, DOS was the most commonly used platform. As these calculation programs were most often programmed by mechanical engineers who knew something about IT rather than by actual software specialists, there was a degree of reticence when it came to converting to a different operating system. Here Windows provided the option of

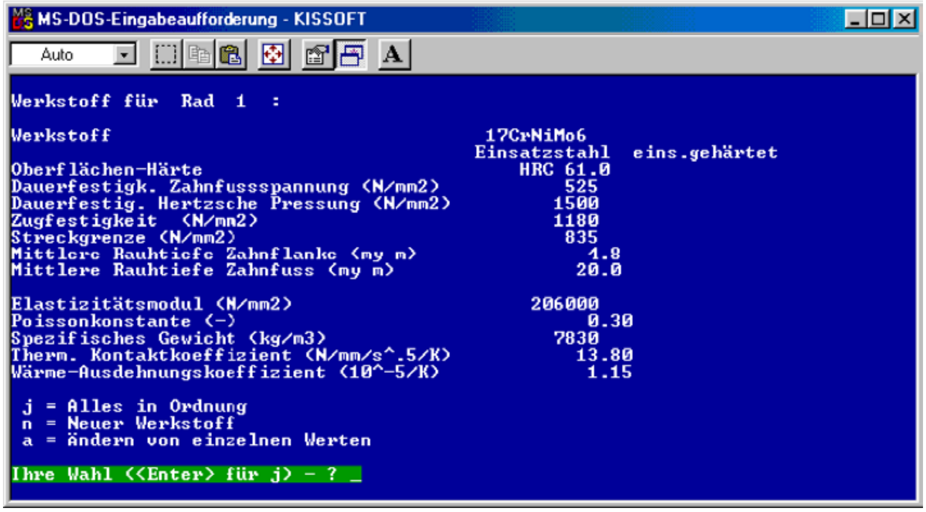


Figure 3 **DOS programs processed keystrokes immediately. Consequently, nearly every screen ended with a prompt asking whether the user's inputs were correct (KISSsoft shaft calculation 1995).**

delegating a whole range of things that would take a great deal of time and effort to program to the operating system: it provided powerful commands for graphics (see the DOS screen display in Fig. 2), enabled the use of a system printer (under DOS you had to program a separate driver for every printer), and greatly simplified file handling. This was all well and good, but only in theory. In practice, the programming effort required to display a line on screen under Windows was much greater than under DOS. As a consequence, every program producer grossly underestimated the effort required to port DOS to Windows: This was the beginning of hard times for providers and customers alike. According to bold announcements, a new version for Windows would soon be available, but was always postponed for some reason or other. Then came the first version — mostly a mish-mash of DOS and Windows which proved to be highly unstable. Some program manufacturers had the idea of keeping the old DOS programs just as they were, and simply putting a user interface on top. Although this provided results very quickly, it soon led to a dead end in development.

Attempts to implement genuine system porting went in two main directions: one direction involved completely restructuring the user interface, whereas the other tried to rescue as much as possible from the old DOS user interface. For existing customers this had the benefit that they did not have to come to grips with an entirely new user interface. Unfortunately, the result was also a program which failed to meet any of the Windows standards.

The most important programs which were able to make the transition to the world of Windows are *Hexagon*, *Mdesign* and *KISSsoft*. These were joined briefly by *Delphi*, which has since disappeared from the market.

Forms or Dialog Technology

Real Windows programs use two main approaches: Form-based programs such as Softwert's *Delphi* or Tedata's *Mdesign* are well suited to running simple calculations. The user interface in these programs looks very like a calculation report (Fig. 3). They also use a top-down workflow, as did the first DOS programs. Every calculation has its own form (key verification, key sizing, etc.). And as long as the entries fit on two screen pages, it is easy to keep the form's content clear in your head. However, the limitations of using forms quickly become clear when you use calculations that require more entries, or where different data variants have to be input.

The vast majority of programs use dialog technology. In other words, you input the parameters required for a calculation in a series of dialog screens. This has the benefit that you can sort the parameters according to group. In principle, you can also input your data in any sequence. In addition, bespoke solutions can be implemented to resolve specific problems. To achieve this, *KISSsoft* has a uniform button, which appears in every screen. This is used to either optimize the values or define them according to specified criteria (Fig. 4).

The Third Dimension

In the last decade CAD programs have increasingly converted to using 3-D. The use of 3-D displays has now become standard in calculation programs, although initially these displays were just an "added extra," because the calculations were performed completely independently of the 3-D display. Although it is debatable whether a three-dimensional image of a cylindrical gear actually provides more information than a 2-D drawing, these images are good to look at, nonetheless.

However, a new interest has come to the forefront in recent years: the direct display of complicated toothing in three-

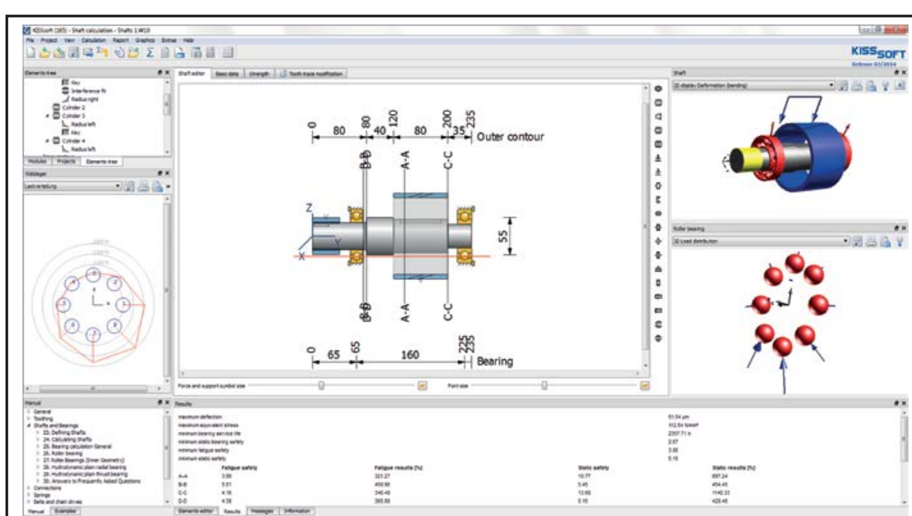


Figure 4 **KISSsoft shaft calculation (2014).**

dimensional models — most critically in bevel gears with spiral teeth. As large bevel gears were in short supply, there was a demand to be able to mill bevel gears directly on five-axis machines. The 3-D models needed to achieve this would then be supplied by the gear calculation programs.

The main problem here was the system performance this would require: i.e., to create realistic gear models you would need not only sufficient computing power, but also the amount of memory required by a mid-range, 3-D CAD system.

Calculating Variants

Since its very earliest days, the mathematical sizing of parts has faced a particular problem. Although the design phase may involve numerous design variants, which are handled in parallel, almost as separate strands, the designer will only select one variant in the end. Nowadays, it is also fairly usual for several variants of a finished design to be processed at the same time, either as standard gear series or as special designs that can be customized to meet specific customer requirements.

The first calculation programs provided absolutely no support for variants. At that time, the limitations on computing power meant that using even one variant in a calculation was quite an achievement. As the hardware has become more powerful, the number of options available in this area has also increased. In particular, the ability to systematically vary parameters to find the best possible solution plays a significant role in calculation programs.

KISSsoft has now implemented variant options at every level of its calculations, starting from individual parameters, such as those used to calculate the profile shift for gears, or find the tolerance pairs for interference fits, through the systematic variation of several parameters at the same time, as required for fine sizing the macro and micro geometry of gears, up to varying the parts in a transmission at system level. It is these additional functionalities that make a calculation program into a modern, efficient system.

Apps

Nowadays, everyone involved in IT is talking about apps. These are relatively small programs which are primarily designed to run on tablets and smart phones. A range of apps, most of which are available free of charge, have already been created for performing mechanical engineering calculations. Each of these apps can handle a small-scale calculation, e.g., determine Hertzian pressure or the L_{10} service life of a roller bearing. However, it is unlikely that the current methods of approach can be used to create apps for more complex problems. This is mainly due to the fact that users are unwilling to input large volumes of data on a 4" screen with a virtual keyboard. And a multilevel calculation would simply demand too much from these devices.

Less of a problem is posed by the relatively weak hardware used by these handheld devices, because most machine element calculations do not require a lot of power. These devices could also be used to find solutions with approaches designed to run the front-end on a handheld device and the back-end on a server over the Internet.

FEM vs. Machine Elements

By the 1980s, computers had become powerful enough to run the FEM calculations described above on a standard designer's workstation. People immediately began forecasting the preeminence of FEM programs, claiming that all the problems encountered in mechanical engineering would now be resolved once and for all. However, in the meantime, this euphoria has been tempered by the sobering realization that, although computers are becoming ever more powerful and sophisticated, the same cannot always be said for their users.

FE models still have to be created by specialists who are capable of defining suitable constraints and interpreting the results. Creating useful and accurate documentation also requires time and effort. But that is not to say that the FEM methods are generally superfluous. They are ideal for handling complicated structures or even only a cylinder head cover. It is just that standardized calculation methods are easier to use for simpler machine elements and provide more meaningful results. In the meantime, the basic calculation methods in the standards have been used to create more sophisticated enhancements. These can now be used to process reliable cases, which previously required a part to be investigated using FEM. These include, for example, the method according to Obsieger (Ref.3) for the detailed investigation of tooth root stresses in a gear. This method has proven to be so reliable that it is even used in aero-engineering.

There is now even a certain trend towards combining FEM with the classic definition of machine elements as the next step in the evolution of calculation programs. This is still based on the idea of using just a few parameters to describe a machine element. The program then creates an FE model in the background to determine the stresses in the part. These stresses are then used as the basis on which strength is verified as specified in the standard. Sheet 2 of VDI 2737 for the proof of bolted joints is one of the first published methods that specifically applies this approach. However, the alluring simplicity of this approach conceals such a multitude of traps in its implementation that all previous advances made in this direction are reduced to side issues.

Transition to a System

Classic machine element calculations only involve a single machine element; it is hard to keep track of much more data at once. Nowadays, computers are powerful enough to process an entire system of machine elements, at least "almost" simultaneously. The principles involved here mean that machine elements which depend on each other can only be processed successively, because the results for one element may affect the next. If a ring is closed (A depends on B, B depends on C, and C depends on A), iteration is the only approach. But this is just what computers do best: they can process huge volumes of calculations at such a speed that you have the feeling they could do everything simultaneously. In our opinion, this is why system programs such as *KISSsys* are the way forward. Once a system has been defined, the engineer can optimize data at a particular point and also constantly keep an overview of the entire system.

CAD Integration

CAD integration has been an ongoing topic of debate ever since the first calculation programs were designed. The first thing many interested parties want to know is whether a calculation program is compatible with their CAD system. Unfortunately, there is not a great deal of data that can sensibly be changed between CAD programs and calculation programs. The difference between machine element calculation and FEM is very clear: the former runs using only a small number of parameters whereas the latter requires the exact geometry. And because the parameters required for a machine element calculation are only rarely directly available in a CAD program, it is usually quicker and easier to input them manually rather than use an interface.

The limited number of options available is primarily due to the way CAD programs are currently structured. Efficient integration will only be possible if all the values involved in the calculation (e.g., speed, torque, and also material data) can also be managed and transferred to the (in-house or external) calculation program.

Full integration also means that designers who use the CAD system must be extremely disciplined and work exclusively with parameters. To explain this, a shaft calculation is used here as an example: The shaft contour is transferred to the calculation program, which identifies that the shaft diameter is not sufficiently dimensioned. When the modified diameter is returned to the CAD program, it immediately causes a number of problems because the changed diameter affects the design. For example, a bearing with a larger internal diameter must now be used. If the design has not been thoroughly parameterized, this type of change will require a lot of data to be input manually.

Summary and Outlook

In summary, our evaluation is that the greatest challenges we face in the coming years involve the implementation of efficient variant calculation processes for optimizing data at every level, up to system level, at the design phase, and the integration of machine elements and FEM.

References

1. Niemann, G. *Maschinenelemente*, Vols I-III, 1981, Springer Verlag.
2. Roloff, Matek. *Maschinenelemente*, Vieweg Verlag.
3. Obsieger, B. "Zahnformfaktoren von Aussen- und Innenverzahnungen," *Konstruktion* 32 (1980), pp. 443-447.

For more information:
KISSsoft AG
Phone: +(41) 55254 2050
www.kisssoft.ag

Bevel & Hypoid Gears: Measuring Backlash

QUESTION

Please tell me the best method(s) for measuring backlash in bevel gears.

Expert answer provided by: Dr. Hermann J. Stadtfeld, Gleason Corp.

There are three commonly used methods to determine the backlash of a bevel or hypoid gearset:

- ***Method 1***. Indicator method on a roll tester or in the gearbox
- ***Method 2***. Metal-to-metal movement of gear cone
- ***Method 3***. Encoder backlash determination in a CNC roll tester

Method 1. The most common method is to determine the normal backlash by clamping pinion and gear in a roll tester. First, it must be certain that pinion and gear are rotated to the so called "tight spot." Because of pinion and gear runout, the difference between tight spot and loose spot may be significant. It is important to determine the minimal backlash because this is the value that must be assured for the operation of the gearset: The tight spot can be found on a manual roll tester by setting the pinion cone to the exact mounting distance and rotating the gear by hand until the first metal-to-metal contact occurs. Further rotation will show if the gear cone has to be increased or reduced in order to maintain slight metal-to-metal contact. The angular ring gear position with the largest gear cone adjustment represents the tight spot. Now, the ring gear cone is adjusted to the correct mounting distance and a dial indicator is positioned at the outside — in the middle of the profile of a convex ring gear tooth.

The indicator probe stem should be normal to the surface it is contacting. If possible, the pinion rotation should be locked before the indicator is positioned. After these preliminaries the ring gear is rotated in clockwise direction until the coast-side flanks are in firm contact and the indicator is set to zero (pinion convex and ring gear concave = coast-side). A slight rotation in counterclockwise direction until a firm contact of the drive-side flanks is achieved (pinion concave and ring gear convex = drive-side). The indicator reading after this procedure is defined as the "minimal backlash in the plane of rotation" Δt. The relevant value relating to the backlash values in the dimension sheet must then be calculated as:

$$\Delta s = \Delta t \bullet \cos ß \bullet \cos \Phi$$

Whereas:

Δs Normal backlash
Δt Backlash in the plane of rotation
$ß$ Spiral angle
Φ Pressure angle

Figure 1 visualizes the setup and indicator position in a 90° roll testing machine. The indicator shaft direction includes a 90° angle to the radius connection between probe contacting point at the flank and the center of the ring gear.

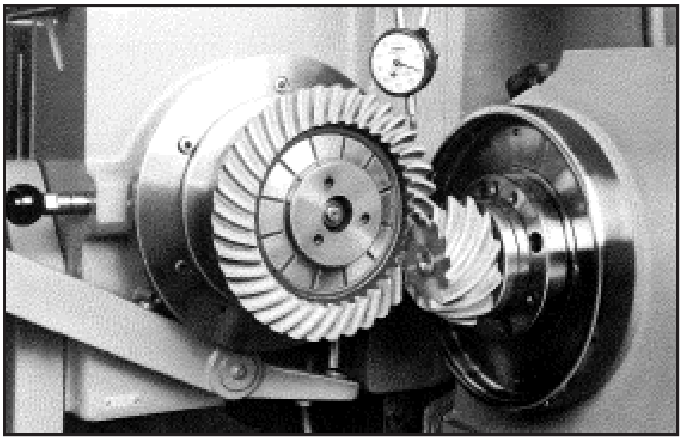


Figure 1 Measurement of normal backlash.

Method 2. Backlash adjustment is done with an axial change of the ring gear position (gear cone). The axial gear cone change between the correct mounting distance setup to the metal-to-metal condition can therefore be used to determine the normal backlash. Pinion and gear are clamped in a roll tester. Then, make certain that pinion and gear are rotated to the so called "tight spot." The tight spot can be found using a manual roll tester by setting the pinion cone to the exact mounting distance and rotating the gear by hand — while simultaneously reducing the gear cone setting with the hand-wheel — until the first metal-to-metal contact occurs. Further rotation will show if the gear cone must be increased or reduced in order to maintain a slight metal-to-metal contact. The angular ring gear position with the largest gear cone adjustment represents the desired tight spot. Assure firm double-flank contact at the tight spot and read the gear cone position on the electronic readout or on the vernier scale.

Now the ring gear cone is adjusted to the correct mounting distance.

The difference in the values of the gear cone at the tight spot, metal-to-metal position to the correct ring gear mounting distance is recorded as "Δz" and used in the following formulae in order to calculate the normal backlash:

$$\Delta s = \Delta z \bullet \sin\left[\arctan\left(n_1/n_2\right)\right] \bullet \left(\tan \Phi_1 + \tan \Phi_2\right)$$

Whereas:

Δz Axial ring gear move from nominal to metal-to-metal
n_1 Number of pinion teeth
n_2 Number of ring gear teeth
F_1 Pressure angle convex gear flank
F_2 Pressure angle concave gear flank


MITSUBISHI
HEAVY INDUSTRIES AMERICA, INC.
Our Technologies, Your Tommorow
The Power of
Dynamic
Motion


SE25A
MITSUBISHI

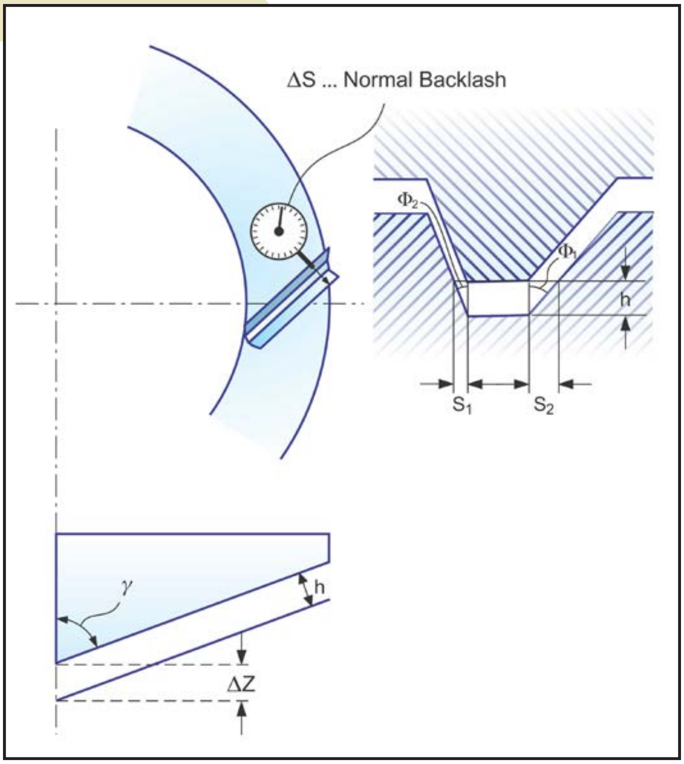


Figure 2 Relationship between withdraw and backlash.

If n_1/n_2 is below 0.3 and $\Phi_1+\Phi_2=40°$, then the following simplification delivers good results:

$$\Delta s=\Delta z \bullet n_1/n_2 \bullet 0.728$$

Figure 2 shows a front view onto a ring gear with a symbolized tooth slot at the top, left. Below the front view, a top view of the ring gear with the pitch cone angle γ. and the relationship between "h" and "Δz." the simplified profile section to the right in Figure 2 relates "h" to s_1 and s_2, where the normal backlash is defined as $s_1+s_2=\Delta s$ in the plane shown in the profile section view.

The technique shown in Figure 2 is also used in CNC roll testing machines in order to automatically set the backlash by an axial ring gear adjustment, after the tight spot has been found.

Method 3. In contrast to Method 2, which uses a simplified, contact geometry, Method 3 is the most accurate way of determining the correct backlash. Method 3 can only be practiced on roll testing machines with single-flank recording capabilities. After pinion and ring gear are positioned at their correct mounting distance, the pinion is rotated in coast direction against a small ring gear torque (e.g., 0.5 Nm). The encoder signal of the ring gear spindle encoder is recorded as shown (Fig. 3, top graph). The diagram shows in the ordinate direction the motion variation in $\Delta\varphi$, and in the abscissa direction the rotational angle φ of the ring gear. Ideally, the number of pinion rotations equals the number of teeth of the ring gear (hunting tooth condition); but in fact one full ring gear revolution will deliver acceptable results. From there the pinion spindle reverses its rotational direction in order to establish drive-side contact and to rotate the same number of prior revolutions. Now, the encoder records the drive-side motion errors — tooth pair after tooth pair. The motion error of a single tooth pair has a parabola shape. The parabolas of the different tooth pairs have a variation in the ordinate direction of the diagram featured in Figure 3. This spacing variation in the ordinate direction of the parabola extrema is caused by an interaction between pinion and ring gear runouts; ring gear runout is dominated by face runout (wobble), where pinion runout originates from the radial runout of the pinion "head" vs. the shaft.

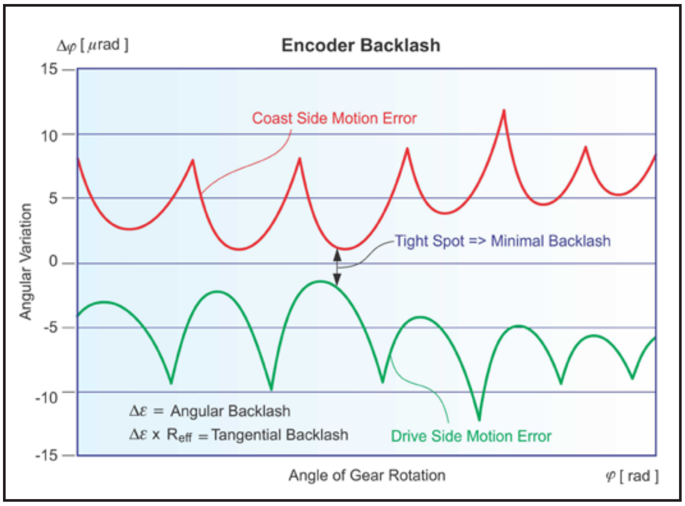


Figure 3 Encoder backlash detection method.

Because the graphs for both coast-side and drive-side meshing have been recorded without an interruption of the encoder signal, the relationship of the upper and lower diagram reflects a double-flank measurement without a double-flank contact. Ordinate (vertical) distances between the two graphs are equal to the angular backlash at the rotational position of the vertical line. The roll testing machine software searches for the shortest distance between the coast- and drive-side motion error graph, and determines the "minimal backlash" (Fig. 3) that is the most relevant number for the gearset. Initially this backlash has the units in microradians, or radians, and is called "angular backlash." The calculation of normal backlash with "length units" can be simply expressed as:

$$\Delta s=\Delta\varphi \bullet R_{eff}/cos\beta$$

Whereas:

$\Delta\varphi$ Angular backlash (rad)
R_{eff} Reference point diameter of ring gear
β Ring gear spiral angle

Dr. Hermann J. Stadtfeld — GT's bevel gear "expert of experts" — indicating setup for measuring backlash on a roll tester. Stadtfeld is Vice President, Bevel Gear Technology, R&D for Gleason Corporation (Photo by Jasmin K. Saewe for Gleason Corp).

Calculation of the Tooth Root Load Carrying Capacity of Beveloid Gears

C. Brecher, M. Brumm and J. Henser

In this paper, two developed methods of tooth root load carrying capacity calculations for beveloid gears with parallel axes are presented, in part utilizing WZL software *GearGenerator* and *ZaKo3D*. One method calculates the tooth root load-carrying capacity in an FE-based approach. For the other, analytic formulas are employed to calculate the tooth root load-carrying capacity of beveloid gears. To conclude, both methods are applied to a test gear. The methods are compared both to each other and to other tests on beveloid gears with parallel axes in test bench trials.

Introduction and Challenge

A particular gear type which becomes more and more important is the beveloid gear, also known as conical involute gear. This is mainly due to their ability to realize small crossing angles between shafts and they can be produced economically on conventional gear grinding machines (Refs. 1–2). Beveloid gears have been used in marine applications, for example, for many years (Refs. 3–5). In recent years the use of beveloid gears in the automotive sector has increased (Refs. 6–7). Here the beveloid gear is used, for example in four wheel drives to transmit torque and rotation from the output of the gearbox to a front axle that may not be parallel.

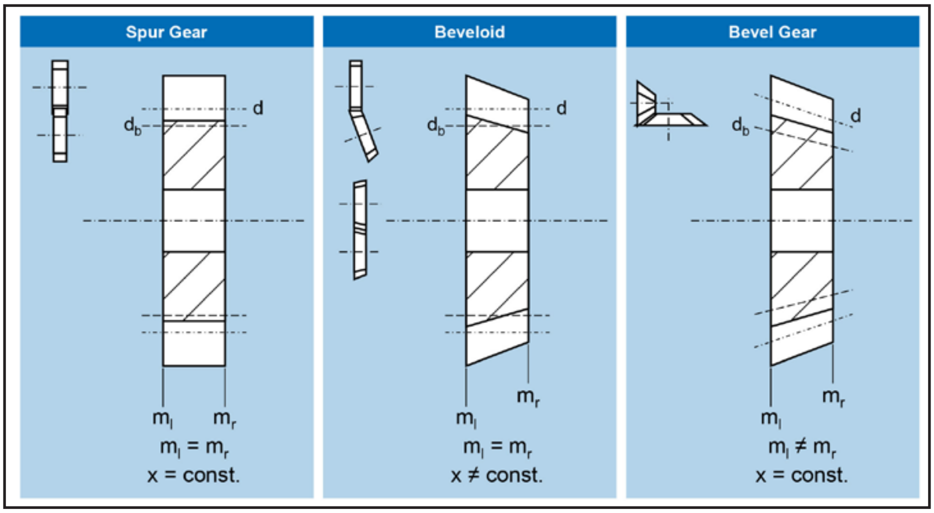


Figure 1 Geometrical characteristics of beveloid gears (Ref. 2).

Geometrical characteristics of beveloid gears. Beveloid gears are used to transmit torque and rotation between elements of crossing, skew or parallel axes (Ref. 6). The geometry of beveloid gears is derived from cylindrical spur or helical gears. The base circle and the pitch circle of beveloids are cylindrical, as presented in the middle section of Figure 1. The pitch and the module are constant along the tooth width. The difference between beveloid gears and cylindrical gears is the varying profile shift along the tooth width to realize crossed or skew axes. For realizing the varying profile shift, the root cone angle θ_f is defined which is generated during gear cutting and gear grinding by a change of the feed during the process. The form of the tip of a beveloid gear is usually conical. The tip cone angle θ_a is determined by the geometry of the workpiece.

A special use of beveloid gears is the arrangement with parallel axes. This is realized by two meshing beveloids that have a cone angle θ of the same absolute value but with opposite orientation (Ref. 2).

Another gear type which is used for realizing crossing or skew axes is the bevel gear. Bevel gears have a conical pitch and base envelope. This results in a varying module m along the tooth width (Ref. 2). Beveloids are usually preferred to bevel gears when small crossing angles must be realized due to manufacturing limits of bevel gears. This is related to the long cone distances of gears with small cone angles which require substantial dimensions of the bevel gear cutting machine (Ref. 8).

Contact behavior of beveloid gears. Beveloid gears can be mounted with parallel, crossed or skew axes. The axis orientation has substantial influence on the gear mesh (Fig. 2). In Figure 2 (left) a typical contact pattern of beveloid with parallel axes is presented. The contact pattern is spread over the whole flank. On the right side of Figure 2 a typical contact pattern of beveloid gears with crossed axes is illustrated. Two involute beveloid gears with crossed axes have point contact. The resulting contact pattern is narrower than the contact pattern of beveloid gears with parallel axes. To achieve a full contact pattern of beveloid gears with crossed or skew axes at least one gear has to be designed with non-involute flanks. In this case, the manufacturing with standard methods like generating grinding is no longer possible. For some applications beveloid gears with conjugated flanks are manufactured by topological grinding to achieve nearly full contact (Ref. 6) but for most applications this manufacturing method is avoided for economic reasons.

Challenge. To achieve a high power/weight ratio, a precise calculation of the gear load and load carrying capacity is necessary to design gears in an economical way. At the state of the art, no approved method for the tooth root load carrying capacity calculation for beveloid gears exists. Therefore the beveloid gear is approximated by a substitute spur gear with the gear data of the middle transverse section of the beveloid gear. The imprecision of this method is shown in Figure 3.

In the diagrams the tooth root stresses of a beveloid gear and a substitute spur gear are compared. The beveloid gear has an axis angle of 7.2°. The substitute spur gear is derived from the gear data of the middle transverse section of the beveloid gear. It can be seen that the stresses of the beveloid are significant higher. Reasons for this are the different root fillet geometry and the different contact behavior. Thus the calculation of the tooth root load carrying capacity of beveloid gears with a substitute spur gear according to existing standards for cylindrical gears is not possible without further ado.

A more precise calculation method can lead to a better design of beveloid gears with a higher power/weight ratio. Furthermore no simulation method for the running behavior of beveloid gears with and without load exists. Such a method could determine the tooth root load carrying capacity for a large number of variants in a short time. Therefore the project "Development and Verification of a Method to Calculate the Tooth Root Load-Carrying Capacity of Beveloid Gears," which is sponsored by the German research funding organization Deutsche Forschungsgemeinschaft (DFG), has been initialized.

Objective and Approach

In this paper the development of two calculation methods for the tooth root load carrying capacity of beveloid gears with parallel axes is described. In Error! Reference source not found. the approach for the development of these methods is illustrated. The initial point is the determination of the tooth root fatigue strength of beveloid gears on a test rig. The results are used to validate a local based calculation method to calculate the tooth root load carrying capacity of beveloid gears.

The initial step of the local based calculation method is the manufacturing simulation with the WZL software

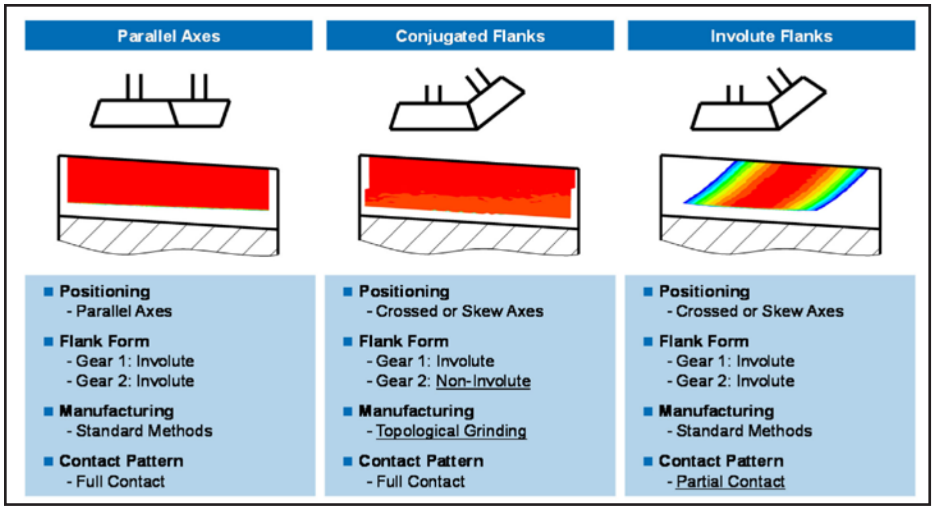


Figure 2 **Contact characteristics of beveloid gears.**

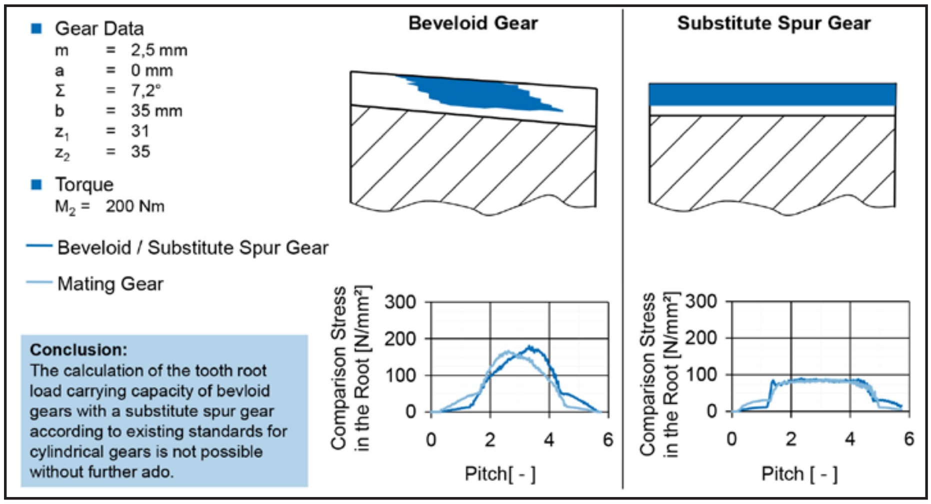


Figure 3 **Comparison of the tooth root stresses of a beveloid gear and its substitute cylindrical gear.**

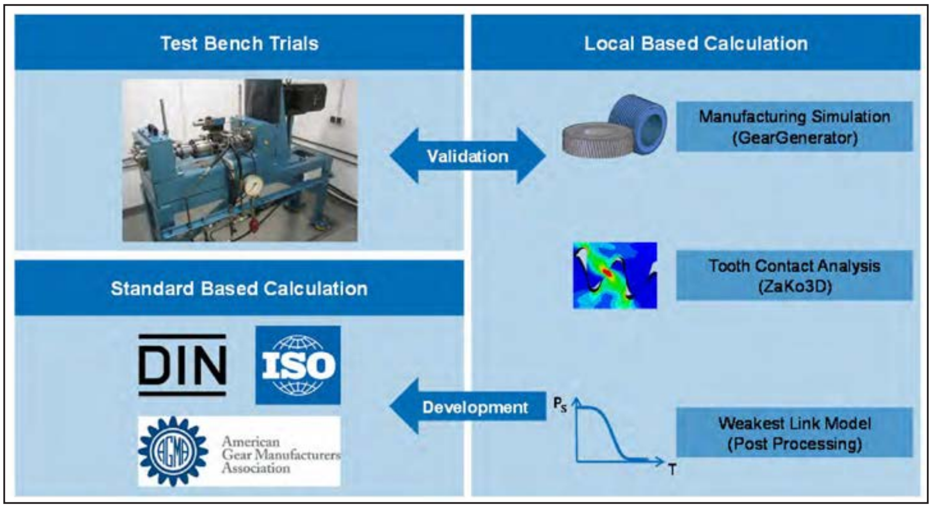


Figure 4 **Approach for developing calculation methods for tooth root load carrying capacity of beveloid gears.**

GearGenerator. In the manufacturing simulation, a 3D geometry of the beveloid gear is created by simulating the generating grinding process (Refs. 1, 9). The resulting beveloid geometries are used in an FE-based tooth contact analysis with the WZL software *ZaKo3D,* which is able to calculate the tooth root stresses of several gear types during meshing (Ref. 10). From these stresses and further parameters (e.g., local material properties) the tooth root load- carrying capacity is calculated in an approach based on the weakest link model of Weibull (Refs. 11–13).

After the local-based calculation method is validated, this method is used to derive a standard based calculation method for beveloid gears. The standard based calculation method uses analytic formulas to calculate the tooth root load carrying capacity of beveloid gears. In this method the tooth root stresses of beveloid gears are compared to the tooth root stresses of cylindrical gears. The effects which, observed during this comparison, are described and formulas are derived to take these effects into account.

Test Bench Trials

To detect the tooth root bending strength of a beveloid gear, a back-to-back test rig is used according to DIN 51354–Part 1, which uses the power circuit principle. The setup is illustrated in Figure 5. The tested beveloid gears are mounted in a gear box. They are connected by shafts to a transmission gear box. This setup is called power circuit. The test gear box is equipped with cylindrical gears. The cylindrical gears have the same gear data as the beveloids, but the cone angle is $\theta = 0°$. The profile shift of the cylindrical gears is taken from the middle section of the beveloid gears. To avoid damage, the test gears are designed significantly wider than the beveloid gears. It is possible to include a torque into the power circuit at the coupling which is mounted at one of the shafts. The other shaft is designed as torque shaft. An electric motor is used to drive the gears. Since the torque is realized by the power circuit, the motor only needs to apply power into the system that corresponds to the power losses due to, for example, friction.

The gear data of the test gears is presented in Figure 6 (left). To use the test principle of the back-to-back test rig according to DIN 51354, Part 1, parallel axes are used with a center distance of $a = 91.5$ mm. The module of the gears is $m_n = 2$ mm; the helix angle is $\beta_{½} = 3.024°$; and the number of teeth are $z_{½} = 45/39$; further, the cone angle of Gear 1 is $\theta_1 = 3.6°$. To realize parallel axes, the cone angle of Gear 2 is $\theta_2 = -3.6°$.

The goal of the tests is to determine the fatigue limit of the test gears for a probability of survival of $P_S = 50\%$; the principle used is the staircase method according to Hück. In this method the test load is dependent on the result of the previous test run. The load is reduced at breakage and increased at run-out. In these tests the load step is fixed at $\Delta T_2 = 25$ Nm. A complete test run is reached if Gear 2 experiences $n_2 = 3{,}000{,}000$ load cycles without root breakage.

The results of the test are presented in Figure 6 (left). In the diagrams the test results are marked at the torque used in each test. The cross represents a breakage during the test; a filled circle represents test run-out. Invalid results are marked with a void circle. To take the test result of the last test (damage or test run-out) into account, a fictitious point is added after the last test. The fictitious point is marked with a void square. For the evaluation, all valid points and the fictitious point are used. This results in a torque for a probability of survival of 50% of $T_2 = 563.64$ Nm for Flank 1, and of $T_2 = 565.91$ Nm for Flank 2; the fatigue strengths of both flank sides are similar.

Local-Based Calculation of Tooth Root Load-Carrying Capacity

The first method to calculate tooth root load-carrying capacity of beveloid gears is a local-based method. In this method the probability of survival is calculated locally for each point in the tooth root.

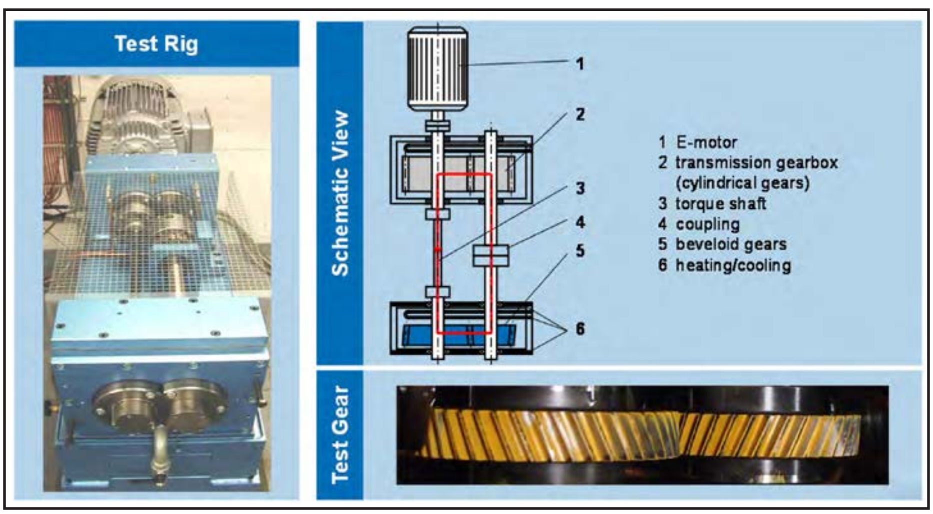


Figure 5 Back-to-back gear test rig according to DIN 51354, Part 1.

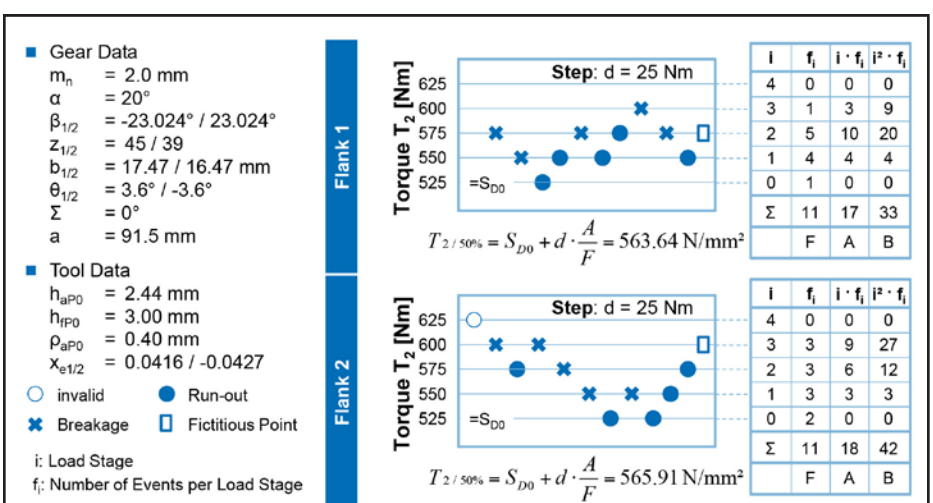


Figure 6 Results of the test bench trials.

In the next section the calculation is presented briefly. This is followed by the application of the calculation method to the test gears that were already used for the test rig trials in the previous chapter.

Simulation method. In this approach three programs are used in sequence to calculate tooth root load-carrying capacity for beveloid gears. Figure 7 presents a brief overview of the three programs.

The first program in the simulation chain is the software *GearGenerator*, which calculates a 3D model of the beveloid gear via generating grinding simulation. The software is based on the calculation method of Röthlingshöfer (Refs. 14, 1). The simulation uses the tool data, the gear data and the information about the axis setup of the machine (e.g., tilting or linked feeds) to calculate the tool geometry, the tool movements and, finally, the resulting gear geometry according to the laws of gearing (Ref. 15). A 3D model of the gear is provided as output. For the microgeometry analysis, the resulting geometry is compared to an ideally shaped involute and then plotted as profile and lead plot. Supplementing the manufacturing simulation, an algorithm was developed according to the Verein Deutscher Ingenieure (VDI) standard, VDI 2607 to evaluate the flank deviations (Refs. 16, 9). The excellent simulation accuracy of this method is shown by Röthlingshöfer in Chapter 5.3.2 of his dissertation (Ref. 1).

The 3D models of the gears generated by *GearGenerator* are used as input for the tooth contact analysis software *ZaKo3D*. The general approach of *ZaKo3D* is the simulation of the 3D tooth contact. Therefore the geometric data of the flank and an FE model of a gear section are needed as input. Furthermore, pitch and assembly deviations can be considered. During the simulation the contact distances, loads, and deflections on the tooth are calculated. The results of the calculation can be displayed in established diagrams to support the gear designer during the development process.

The flank geometry is provided at the outset; in this work the geometry is taken from *GearGenerator*. Alternative input files, such as measurement data files from coordinate measurement machines, are possible as well. Regardless of the source of the input data, the flank must be defined by points in Cartesian coordinates and the direction of the normal vectors at each flank point.

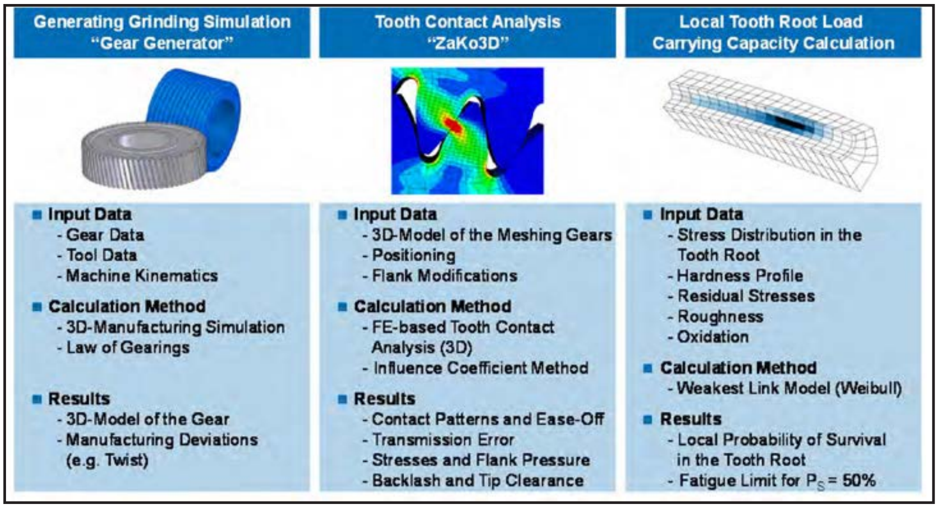


Figure 7 Simulation methodology (Ref. 13).

In order to be able to simulate loaded condition, FE data has to be generated out of the flank data. The FE model contains the information about the stiffness of the gear; it is created by an automatic FE mesh generator for gear teeth (Ref. 10). The FE mesh generator needs the flank geometry to create the mesh. Furthermore, information about the FE node distribution inside the tooth is needed. Finally, the material properties must be defined. Each point of the modeled flanks is loaded with unit forces in each, *x*-, *y*- and *z*-direction. Using this model, a standard FE solver is used to calculate so-called influence coefficients. The influence coefficients hold the information about the deflection of all points during the application of each unit force. This contains the displacement influence coefficients α_{ii} that are on the diagonal of the influence coefficient matrix, as well as the cross influence coefficients α_{ij}. To complete the input data, information about the positioning of the gears is needed. Different gear types can be positioned in *ZaKo3D*; e.g., spur gears, bevel gears, face gears and beveloid gears. The input of pitch deviations, microgeometry deviations and corrections, assembly deviations or different loads can be done by the user and is optional.

The tooth contact analysis starts with the calculation of the contact distances of the flanks during the mesh after reading the input data. This is done for the given number of rolling positions and for each flank point of all the flanks in contact. With these contact distances and the information about the stiffness from the influence coefficients, a mathematic spring model is defined (Ref.18). Since the number of contact points in a rolling position and the force at each contact point influence each other, it is necessary to solve the spring model iteratively.

From these calculations the contact pattern and the transmission error can be derived load-free and under load. The transmission error of a gear is caused by geometric errors of the flanks (load-free content) and deflections (load content) of the gear, and gives a good impression of the dynamic gear excitation (Ref. 17). The course of the transmission error can be displayed over time and by performing a fast Fourier transformation in the frequency domain.

Using the forces on the nodes and the flank areas to which these forces are applied, the resulting pressure can be calculated. The flank area corresponding to a node is defined by the grid size (Ref. 18). The surface stress distribution on the flank has a major influence on the wear resistance of the flanks, and a reduction can lower the risk of pitting (surface fatigue) and improve the flank load-carrying capacity (Refs. 19–20). Furthermore, the ease-off, which represents the contact distances in the mesh area, is calculated load-free. This output data provides information about the gear behavior and can be used to predict the quality of the calculated gear design. This is necessary to reduce the needed number of design validation tests.

Tooth root stresses are calculated by *ZaKo3D* in an FE-based approach. The FE model, which is used for the influence coefficient simulation, is applied with the forces that occur during discrete mesh positions. The stress tensors are calculated at each FE node in the whole root section and are evaluated according to several stress hypotheses (e.g., the von Mises stress hypothesis). The stress tensors can also be used to calculate the probability of survival in a local-based approach (Ref. 10).

The local-based calculation of the tooth root load-carrying capacity is based on the "weakest link" concept, invented by Waloddi Weibull. The weakest link concept says that not just the maximum stress must be taken into account for the fatigue determination; the distribution of weakest links in the material must be considered as well. In the weakest link concept the load stresses σ_a, the fatigue limit σ_D, the volume V and the Weibull module k (for the statistical distribution) are taken into account. The possibility to apply the weakest link model to gears was first investigated by Dr. Brömsen and Dr. Zuber. In their dissertations the calculation of the 50% probability of survival PS/50% in pulsator tests is developed and verified (Refs. 21, 11, 12).

An FE model of the gear is used to calculate the tooth root load-carrying capacity of gears; input data are the stress tensors which were calculated in *ZaKo3D*. The material parameters — hardness, residual stresses and oxidation — are used in this method and the surface roughness in the root fillet is considered. With this input data the probability of survival P_S is calculated for each FE element. This is done by comparing the stress amplitude σ_a to the fatigue strength σ_D in the so-called integration points of the FE elements. The fatigue strength is calculated by empirical expressions from the material parameters. By numerical integration this comparison is extended to the whole FE element. Multiplying the probabilities of survival of each FE element in the root equals the probability of survival of the total tooth root.

Application to the test gears. The method described in the previous section is applied to the gear used in the fatigue tests; the load is applied on Flank 2 (see Fig. 8). In the two diagrams at the top, the hardness profile and the residual stress profile used for the calculations are presented. For adjusting alignment errors and microgeometry corrections, the contact pattern was used. In the lower-left diagram the probability of survival is plotted over the torque at Gear 2. For low loads the probability of survival approaches $P_S = 1$. For high loads the probability of survival approaches $P_S = 0$. The probability of survival P_S drops significantly — between $T_2 = 500$ Nm and $T_2 = 600$ Nm. At approximately $T_2 = 576$ Nm, the probability of survival reaches 50%. In the test bench trials, described earlier, the torque $T_2 = 565.91$ Nm leads to a probability of survival of 50% (see Fig. 8, lower right). The difference between the simulation and the test rig results is lower than 2%. This shows the good correlation between simulation and testing. Hence it is possible to use the simulation to develop a standard-based approach.

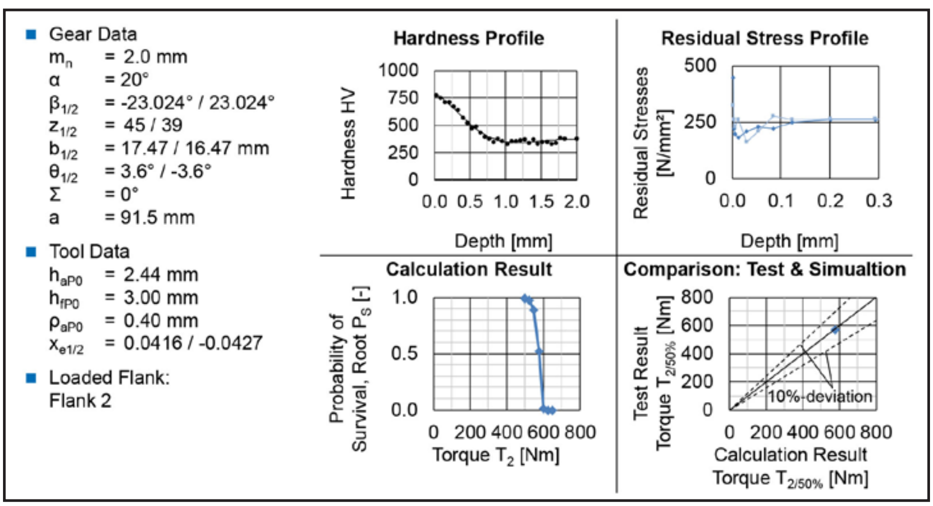


Figure 8 Application of local-based calculation method to a test gear.

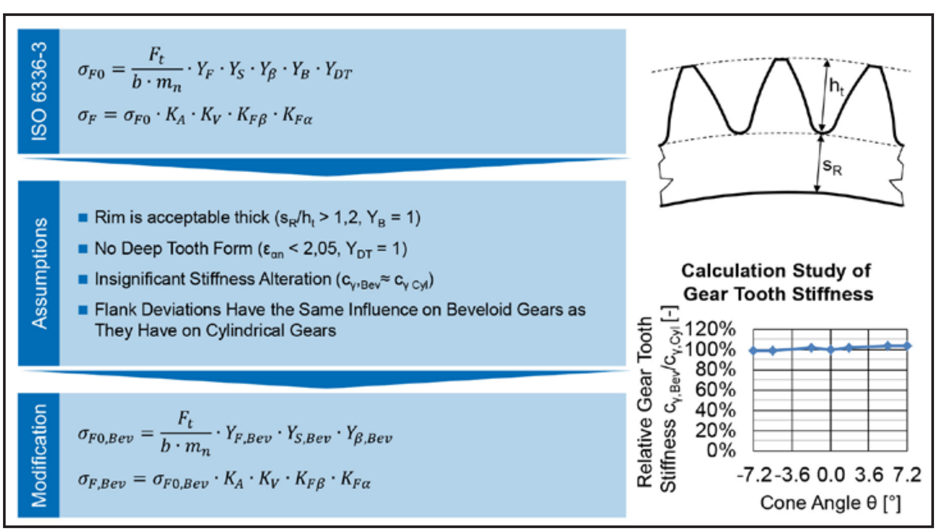


Figure 9 Analytical calculation for tooth root load carrying capacity of beveloid gears with parallel axes.

Standard-Based Calculation Method for Tooth Root Load-Carrying Capacity

Earlier it was shown that the stresses calculated in *ZaKo3D* can be used to calculate the tooth root load-carrying capacity of beveloid gears. Thus the software is used to develop a standard-based method for calculating tooth root stresses of beveloid gears. The approach of the standard-based calculation method is to use the existing calculation method of ISO 6336 by modifying its factors (Ref. 20). The approach for the modification is plotted in Figure 9 (left). In the upper part, the formula from ISO 6336 (Ref. 20) used to calculate the tooth root stress of cylindrical gears is presented. The Y- and K- factors, which are defined for cylindrical gear, have to be adapted to calculate tooth root stresses for beveloid gears.

In the approach presented, it is assumed that the rim is sufficiently thick and that no deep tooth forms are investigated. Accordingly, the rim thickness factor *YB* and the deep tooth factor *YDT* have the value 1 and will not be considered further.

Under the assumption that helix angle deviations, profile corrections, and the pitch errors have the same effects for cylindrical gears and beveloid gears, and that the average gear stiffness of a beveloid and its derived cylindrical gear are similar, the formulas for the transverse load factor $KF\beta$, the face load factor $KF\alpha$ and the dynamic factor KV for the calculation of beveloid gears can be directly used from the standard calculation for cylindrical gears. Beveloid gears have been derived with cone angles between $\theta = -7.2°$ and $\theta = 7.2°$ from a cylindrical. For all versions the average tooth stiffness c_γ has been calculated with *ZaKo3D*. In Figure 9, bottom right, the relationship between the average tooth stiffness of the beveloid variants $c_{\gamma,Bev}$ and the relation of the average tooth stiffness of the cylindrical gears $c_{\gamma,Cyl}$ is shown; only minor differences can be observed. The maximal deviation is 3.3%, so that the previously mentioned assumption of a similar average tooth stiffness of cylindrical gears and beveloid gears is achieved.

The application factor K_A depends only on the engine and the load of the gear box. The factor can be transferred from the calculation method of cylindrical gears into the calculation method of beveloid gears without any adaptation.

What remains are the form factor Y_F, the stress correction factor Y_S and the helix angle factor Y_β; these are related to the geometry and the contact conditions. Due to the significant changes between cylindrical gears and beveloid gears, the factors are redefined. In the next section, these geometry-dependent factors for beveloid gears are developed. Included are the beveloid form factor $Y_{F,Bev}$ and the beveloid stress correction factor $Y_{S,Bev}$. After these the influence of the overlap ratio and of the helix angle on the tooth root stress of beveloid gears are specified by the beveloid helix angle factor $Y_{\beta,Bev}$ later sections of this paper.

Form factor $Y_{F,Bev}$ and stress correction factor $Y_{S,Bev}$. Beveloid gears change their profile shift with the tooth width (Ref. 1). That results in changes of the cross-section of the gearing and the notch in the tooth root. Figure 10 shows a beveloid gear that illustrates that relationship. On the left side, the beveloid gear face side with a low profile shift is illustrated. On that face side thin tooth roots with a small normal chord s_{Fn} occur. Furthermore, the curvature of the tooth root curve is low; therefore the root radius ρ_F on that face side is high. The right side shows the beveloid gear's face side with a high profile shift. On that side the teeth are thick, which results in a long normal chord s_{Fn}. The root curve displays a low root radius ρ_F. The bending moment arm h_{Fe} for load incidence at the outer point of contact depends on the transverse contact ratio ε_α. This depends on several factors, which make a simple statement about the relative change of the profile shift impossible.

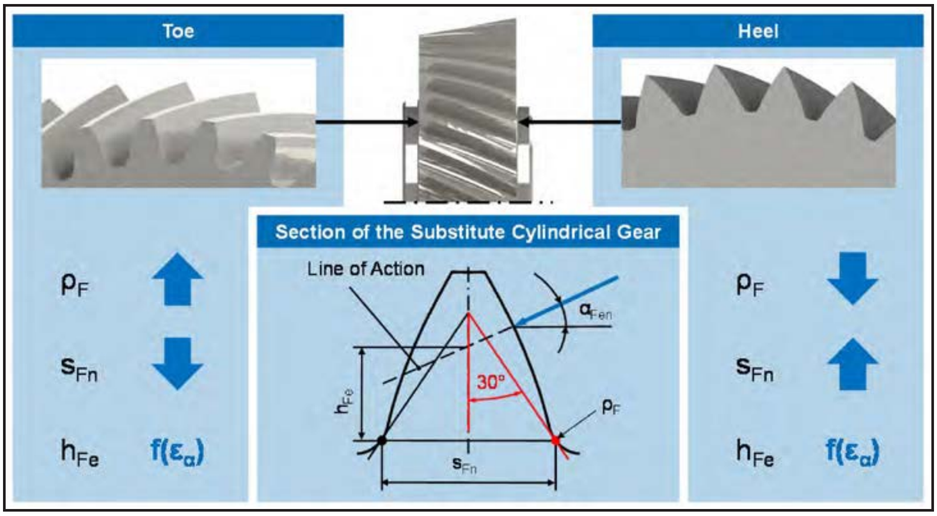


Figure 10 Determining values for the calculation of the beveloid form factor $Y_{F,Bev}$ and the beveloid stress correction factor $Y_{S,Bev}$.

To detect the influence from the varying transverse sections on the tooth root stresses, the profile shift at the axial coordinate with the maximal tooth root stresses is used. First, this is determined in FE calculations. The estimation by analytical equations has to be developed.

Supplied with the data of the face section of the maximal tooth root stress, a cylindrical, substitute helical gear is generated that is used to determine the helix angle factor and the stress correction factor. Since these depend on the profile shift of the critical face section, they are called beveloid form factor $Y_{F,Bev}$ and beveloid stress correction factor $Y_{S,Bev}$.

***Helix factor,* Y_β.** The contact conditions of beveloid gears with parallel axes are significantly different from the contact conditions of cylindrical gears (Refs. 22, 2). The differences are illustrated in Figure 11 by comparing the fields of action of both gear types. The left side shows the field of action of a helical cylindrical gear; the field of action is rectangular. The width is limited by the active tooth width b of both gears. The length is formed by the root and the tip length of engagement. The lines of contact are shown as dashed lines and are inclined at the base helix angle within the field of action. In direction of the length of engagement, the distance of the lines of action is described by the transverse base pitch on the path of contact p_{et}. The overlap ratio ε_β is calculated with Equation 1 from the tooth width b, the base helix angle β_b and the transverse base pitch on the path of contact p_{et} (Ref. 23):

$$\varepsilon_\beta = \frac{b \tan \beta_b}{p_{et}} \tag{1}$$

where

ε_β is overlap ratio
b is (active) tooth width, mm
β_b is base helix angle, degrees
p_{et} is transverse base pitch on the path of contact, mm

The right side of Figure 11 shows a field of action of a helical beveloid gear. Like cylindrical gears, the width of the field of action is limited by the active tooth width b. However, different tip and root lines of action on the face sections lead to a parallelogram-shaped field of action. For a fully accurate illustration of the field of action, the boundaries at the inlet and outlet must be drawn hyperbolically (Ref. 23). The curvatures of hyperbolas of conical spur gears are usually so small that they can be replaced by straight lines. The lines of action are in the field of action diagonally, as it is observed for helical cylindrical gears, but for the fact that the base helix angle on the right flank $_{\beta b,R}$ and the base helix angle of the left flank $\beta_{b,L}$ differ from each other (Ref. 22).

The overlap ratio of beveloid gears is composed of two parts. The first component is the overlap angle of the flank lines $\varphi_{\beta R,L}$. This describes the angle that is enclosed by the axial planes at the endpoints of the flank lines. This part of the overlap ratio is calculated analogous to cylindrical gears (Eq. 2). For beveloid gears, the helix angles on the right and left flank must be considered, dependent on the flank side, so that the overlap angle is calculated separately for both sides. The second part is the overlap angle of the field of action $\varphi_{FMR,L}$. The field of action is inclined at the angles $\beta_{FRf,L}$, and $\beta_{FaR,L}$ on the entry and on the exit side (Fig. 11). This results in the field entry overlap angle $\varphi_{FMR,L}$ and the field exit overlap angle $\varphi_{FMR,L}$, (Eqs. 3, 4). The overlap angle of the field of action $\varphi_{FMR,L}$ is determined by the average of the field entry overlap angle $\varphi_{FfR,L}$ and the field exit overlap angles $\varphi_{FaR,L}$ (Ref. 22).

$$\varphi_{\beta R,L} = \frac{2\,b \tan \beta_{bR,L}}{d_{bR,L}} \tag{2}$$

$$\varphi_{FfR,L} = \frac{2\,b \tan \beta_{FfR,L}}{d_{bR,L}} \tag{3}$$

$$\varphi_{FaR,L} = \frac{2\,b \tan \beta_{FaR,L}}{d_{bR,L}} \tag{4}$$

where

$\varphi_{\beta R,L}$ is overlap angle of the flank line, degrees
b is (active) tooth width, mm
$\beta_{bR,L}$ is base helix angle, degrees
$d_{bR,L}$ is base circle diameter, mm
$\varphi_{fR,L}$ is field entry overlap angles, degrees
$\varphi_{aR,L}$ is field exit overlap angles, degrees
$\beta_{FfR,L}$ is field entry inclination angle, degrees
$\beta_{FaR,L}$ is field exit inclination angle, degrees

The overlap ratio of beveloid gears with parallel axis is calculated with Equation 5 by the difference between the overlap angle of the flank lines $\varphi_{\beta R,L}$ and the mean field overlap angle $\varphi_{FMR,L}$, divided by the angular pitch τ, which is calculated according to Equation 6 (Ref. 22). Depending on orientation of the base helix angle of the line of action, the contact ratio can be increased or decreased. This relation can be observed by comparing the field of action of a beveloid gear and the field of action of a cylindrical gear. On the right section of Figure 11, the field of action of a beveloid gear and the field of action of a cylindrical gear with the data of the middle transverse section are superimposed. It is shown that the contact at the beveloid gear starts earlier due to the helix angle of the field of action at the inlet side $\beta_{FfR,L}$ and the contact at the beveloid gear ends later due to the helix angle of the field of action at the outlet side $\beta_{FaR,L}$. The contact ratio is consequently higher than the contact ratio of a cylindrical gear; however, if the helix angle is more right-oriented, than left so, the relationship then changes and smaller contact ratio occurs for the beveloid gear.

$$\varepsilon_{\beta,BevR,L} = \frac{\varphi_{bR,L} - \varphi_{MR,L}}{\tau} \tag{5}$$

$$\tau = \frac{2\pi}{z} \tag{6}$$

where

$\varepsilon_{\beta,BevR,L}$ is overlap ratio (beveloid)
$\varphi_{bR,L}$ is overlap angle of the flank line, degrees
$\varphi_{MR,L}$ is mean field overlap angle, degrees
τ is pitch angle, degrees
z is number of teeth

The calculation of the beveloid helix factor $Y_{\beta,BevR,L}$ for beveloid gears is analogous to the calculation of the contact ratio factor of a cylindrical gear. For the overlap ratio, the beveloid helix factor of beveloid gears according to Equation 5 is used. The helix angle of the appropriate flank side is used since the contact ratio and the helix angle usually differ for both flank sides. The beveloid helix factor $Y_{\beta,BevR,L}$ must also be calculated separately for each flank.

$$Y_{\beta BevR,L} = 1 - \varepsilon_{\beta BevR,L} \frac{\beta_{R,L}}{120} \tag{7}$$

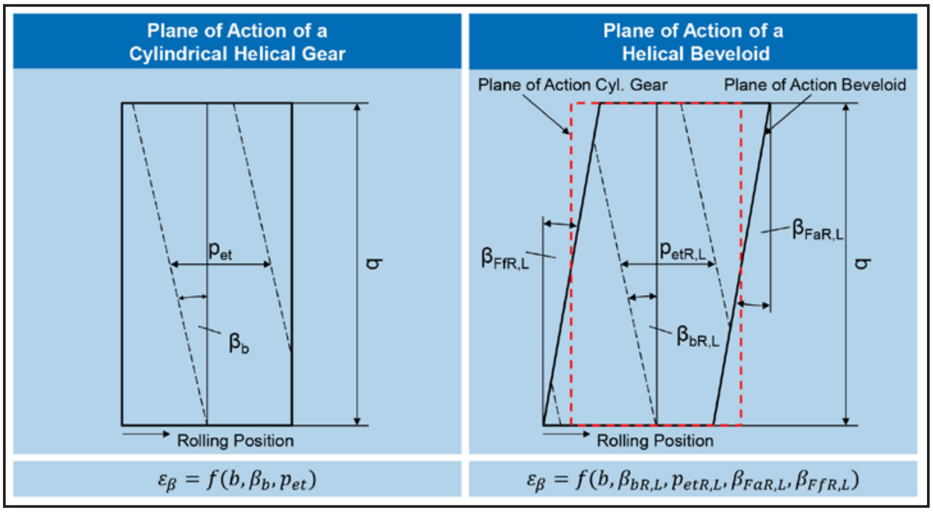


Figure 11 Calculation of overlap ratio of beveloids.

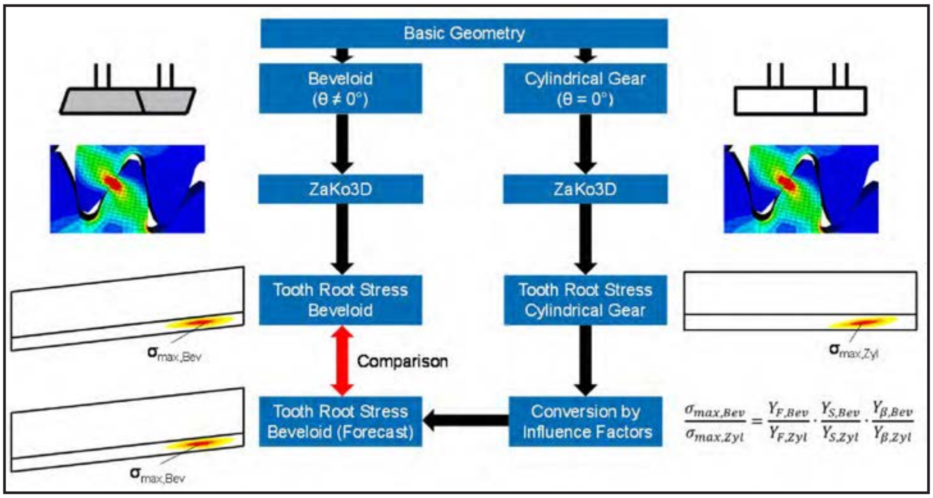


Figure 12 Method for investigation of beveloid factors.

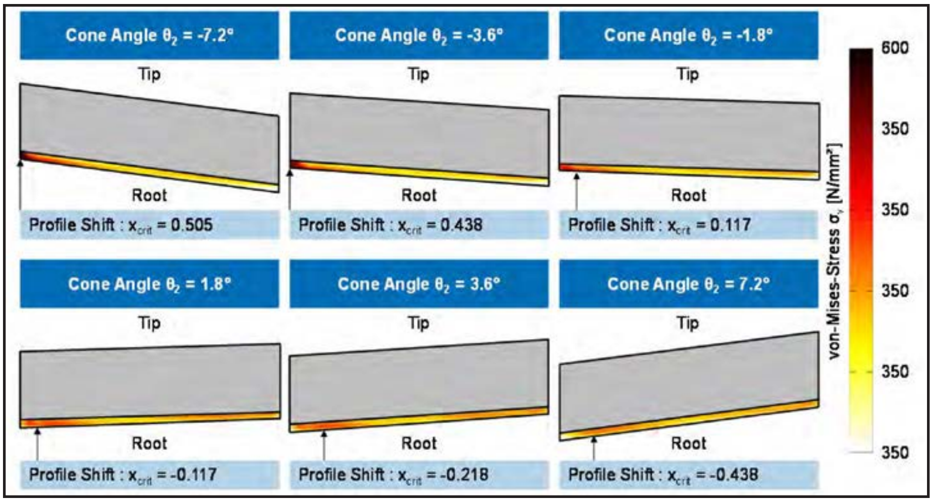


Figure 13 Stress maxima position of the sample gear in dependence of the cone angle (torque $T_2 = 475$ Nm).

where

$Y_{\beta BevR,L}$ is beveloid helix factor

$\beta_{R,L}$ is helix angle, degrees

$\varepsilon_{\beta BevR,L}$ is overlap ratio (beveloid)

Application to a sample gear. The calculation method presented earlier in this paper will be applied to two sample gears in the following paragraphs. A detailed analysis of the individual calculation factors and their combinations will be performed. The strategy in order to do so is shown in Figure 12.

Basic geometry will be defined for a test gear. From that basic geometry a cylindrical gear — also described as a beveloid gear with the cone angle $\theta = 0°$ — as well as several beveloid gears with parallel axes — will be derived. For all variants the maximum tooth root stresses will be calculated with *ZaKo3D*. Subsequently a conversion of the tooth root stress of the cylindrical gear into the tooth root stress of the derived beveloid gear will be performed using the approach presented earlier in this paper. Those calculated tooth root stresses will be compared to the results from *ZaKo3D.*

The sample gear to which the calculation method will be applied has the following gear data: The normal module is $m_n = 2$ mm and the helix angle is $\beta = 23.024°$. The cone angle will be varied between $\theta = -7.2°$ and $\theta = 7.2°$. Gear 2 has a face width of $b_2 = 16.0$ mm, which is narrower than Gear 1, with a face width of $b_1 = 16.8$ mm.

For all variants a tooth contact analysis with *ZaKo3D* was performed. The model used to calculate the tooth root stresses in *ZaKo3D* has a flank resolution of 30 grid points in profile direction and 30 grid points in flank direction for Gear 1. Gear 2 has a flank resolution of 31 grid points in profile direction and 31 grid points in flank direction.

The gearing will be loaded with a torque of $T_2 = 475$ Nm on Gear 2. In the whole tooth root area the tooth root stresses will be evaluated at the FE nodes according to the von Mises criteria. In Figure 13 the maxima of the von Mises-equivalent stress throughout the whole tooth root area for some of the sample gearings with the considered cone angles are shown. The place where the maximum von-Mises-equivalent stress appears is indicated by an arrow. For all variants this place is located on the left of the tooth flank. The maximum tooth root stresses are located directly at the heel for the variants with negative cone angles. Note that with increasing cone angle the maximum stress location moves slightly towards the center of the gearing.

The face section with the maximum tooth root stresses are used to calculate the factors $Y_{F,Bev}$ and $Y_{F,Bev}$ for all variants for the following calculations: the face sections differ in the addendum modifications x_1 and x_2 and in the tip circle diameters d_{a1} and d_{a2} of gear and pinion. As the tip circle diameter is related to the addendum modification through the addendum factor, only the corresponding addendum modification will be listed.

In order to evaluate the influence of the *Y* factors on the calculation of the tooth root stresses, a conversion of the maximum tooth root stress of a cylindrical gear to the maximum tooth root stress of the beveloid variants was performed. The converted stresses were compared to the calculated tooth root stresses of *ZaKo3D*. The conversion is described through Equation 8. In the following the single factors of Equation 8 as well as combinations of those factors are examined. Thereby other factors shall not be considered. The respective fractions were deleted from Equation 8.

$$\frac{\sigma_{v,2Bev}}{\sigma_{v,2Cyl}} = \frac{Y_{F,Bev}}{Y_{F,Cyl}} \cdot \frac{Y_{S,Bev}}{Y_{S,Cyl}} \cdot \frac{Y_{\beta,Bev}}{Y_{\beta,Cyl}} \quad (8)$$

where

$\sigma_{v,2.Bev}$ is maximum Von-Mises-Stress in the beveloid tooth root, N/mm²

$\sigma_{v,2Cyl}$ is maximum Von-Mises-Stress in the cylindrical gear's tooth root, N/mm²

$Y_{F,Bev}$ is beveloid form factor

$Y_{F,Cyl}$ is form factor

$Y_{S,Bev}$ is beveloid stress correction factor

$Y_{S,Cyl}$ is stress correction factor

$Y_{\beta,Bev}$ is beveloid helix factor

$Y_{\beta,Cyl}$ is helix factor

In the upper left diagram of Figure 14, the tooth root stresses calculated with *ZaKo3D* are compared to the tooth root stresses calculated with Equation 8; here all *Y*-factors are considered. With the helix factor Y_β, the dependence of the maximum tooth root stress on the cone angle turns out to have a contrary course between the predicted maximum tooth root stresses and the ones calculated with *ZaKo3D*. Due to that, the combination of all factors is inappropriate in order to predict the maximum tooth root stresses for the considered beveloid gears.

In the three remaining plots in Figure 14, only two of the three possible *Y*-factors are taken into consideration. In the upper right diagram the calculation of the tooth root stresses of the beveloid variants is performed on the basis of the tooth root stresses of the cylindrical gear $\sigma_{v2,Cyl}$, the form factor *YF*, and the stress correction factor Y_S. In the range of negative cone angles, the tooth root stress are predicted considerably lower than in the calculation with *ZaKo3D*. In the range of positive cone angles, higher tooth root stresses are predicted than were calculated with *ZaKo3D*. However, the deviations are small. Due to the underestimation of the tooth root stresses for negative cone angles, the combination of the form factor Y_F and the stress correction factor Y_S is not sufficient to predict the tooth root stresses of the considered beveloid variants.

In the diagram on the lower left hand side the conversion of the tooth root stresses is performed on the basis of the tooth root stresses of the cylindrical gear $\sigma_{v2,Cyl}$, the form factor Y_F and the helix factor Y_β. The relationship between cone angle and maximum tooth root stresses shows strongly contrary tendencies. Hence combination of form factor Y_F and stress correction factor Y_S is not sufficient to predict the tooth root stresses of the considered beveloid variants.

In the lower right hand diagram the conversion of the tooth root stresses is performed on the basis of the tooth root stresses of the cylindrical gear $\sigma_{v2,Cyl}$, the stress correction factor Y_S and the helix factor Y_β. The tooth root stresses being calculated through the conversion match the tooth root stresses calculated with *ZaKo3D* very well; the maximum deviation is 2.33%. Consequently the combination of the stress correction factor Y_S and helix factor Y_β suits very well in order to calculate the maximum tooth root stresses of the considered beveloid variants $\sigma_{v2,Bev}$ from the tooth root stresses of the cylindrical gear version $\sigma_{v2,Cyl}$.

Conclusion

Earlier in this paper, an investigation into which *Y*-factors allow a conversion of the tooth root stresses of a cylindrical gear to the tooth root stresses of a beveloid gear with a high accordance to the calculation results of *ZaKo3D*. The highest accordance to the results of *ZaKo3D* is achieved if the stress correction factors Y_S and the helix factor Y_β are used in the conversion while the form factor Y_F is not taken into consideration.

Due to the high accordance with the sample gears the conversion performed on the basis of the stress correction factor Y_S and the helix factor Y_β in order to calculate the tooth root stresses of beveloids will be added to the calculation method. The form factor Y_F will furthermore be deducted from the data of the average transverse section. Under the boundary conditions described earlier, the calculation method for the tooth root stresses of beveloids results in Equations 9 and 10:

$$\sigma_{F0,Bev} = \frac{F_t}{m_n\, b}\, Y_{F,Zyl}\, Y_{S,Bev}\, Y_{\beta,Bev} \tag{9}$$

$$\sigma_{F,Bev} = \sigma_{F0,Bev}\, K_A\, K_V\, K_{F\beta}\, K_{F\alpha} \tag{10}$$

where

$\sigma_{F0,Bev}$ is nominal tooth stress, N/mm^2
F_t is nominal tangential load, N
b is tooth width, mm
m_n is normal module, mm
$Y_{F,Zyl}$ is form factor (mean transverse section)
$Y_{s,Bev}$ is beveloid stress correction factor (critical transverse section
$Y_{b,Bev}$ is beveloid helix factor
$\sigma_{F,Bev}$ is tooth root stress, N/mm^2
K_A is application factor;
K_V is dynamic factor;
$K_{F\beta}$ is face load factor;
$K_{F\alpha}$ is transverse load factor.

An important part of calculating the form factor is the calculation of the tooth root critical section with the tooth root normal chord s_{Fn} and the tooth width b. High addendum modifications, as they appear at a beveloid's heel, result in long tooth root normal chords and therefore a high section modulus of the cross-section. Low addendum modifications, as they appear at a beveloid's toe, result in short tooth root normal chords and a low section modulus. With beveloids, the profile shift changes alongside the tooth width. Therefore the length of the tooth root normal chords changes along the cross-section. This is outlined by a trapezoidal cross-section of the beveloid at a section through the tooth root (Fig. 15). This is compared to the cross-section of a substitute gear with the gear data of the heel and a substitute gear with the gear data of the toe.

The gear data of the heel lead to an overestimation of the section modulus, while the toe's section modulus is low. The use of the gear data of the average cross-section offers a good approach to approximate the section modulus of a beveloid gearing. This approach will require further investigation.

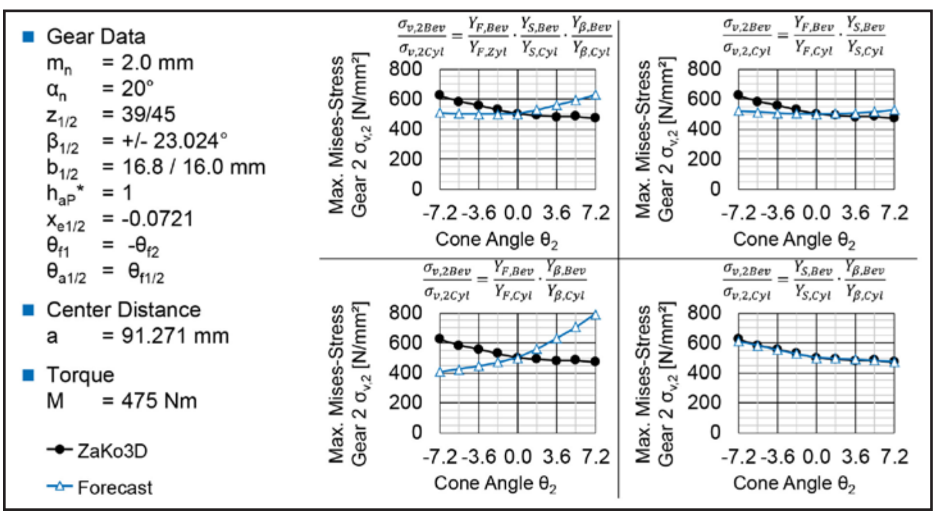


Figure 14 Influence of the Y-factors on the tooth root stress calculation.

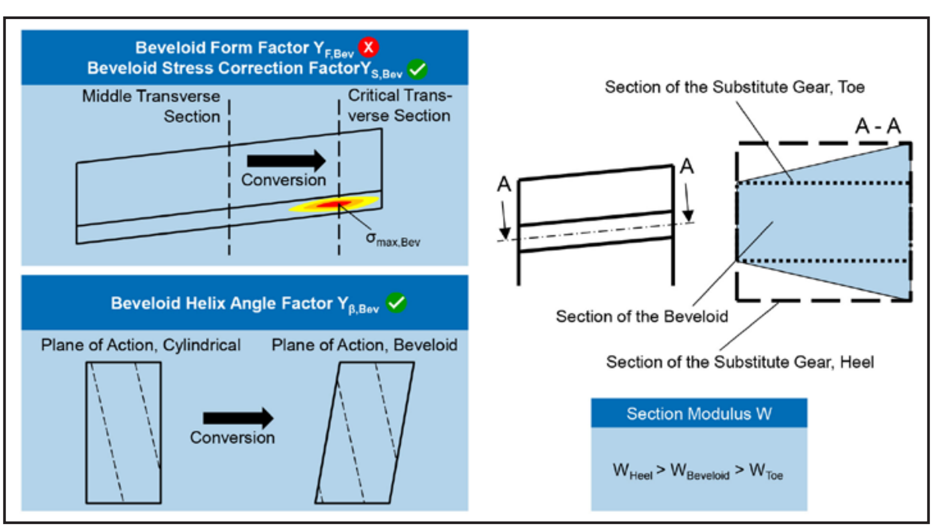


Figure 15 Conclusion.

Summary and outlook

In this paper, two developed methods for the tooth root load-carrying capacity calculation for beveloid gears with parallel axes are presented. The first method calculates the tooth root load-carrying capacity in an FE-based approach. The model is used to calculate the tooth root load-carrying capacity of a test gear. The results show good correlation to investigations on a back-to-back test rig.

The second method uses analytic formulas to calculate the tooth root load-carrying capacity of beveloid gears. In this method the tooth root load-carrying capacity of beveloid gears is compared to the tooth root load-carrying capacity of conventional cylindrical gears. The effects observed during this comparison are described and formulas are derived to take these effects into account.

The new method submits changes on three *Y*-factors. The presented method shows for the sample gear a good correlation. A validation on further cases has not been made yet. To define a method for an extensive scope more comparisons on further gear cases are scheduled.

Furthermore, the method introduced here is presently based on a numerical calculation of the location of the maximum tooth root stresses. To provide a closed analytical solution, the method has to be expanded by an analytical forecast of the location of the maximum tooth stresses. Finally, further verification tests are necessary to confirm whether the analytical calculation — especially the differences in the loads of the flank — can give indication of the influence of the cone angle on the stress distribution.

References:

1. Röthlingshöfer, T. "Auslegungsmethodik zur Optimierung des Einsatzverhaltens von Beveloidverzahnungen," Dissertation, RWTH Aachen, 2012.
2. Roth, K. Zahnradtechnik, "Evolventen-Sonderverzahnungen zur Getriebeverbesserung," Springer, Berlin, 1998.
3. Winkler, T. "Untersuchungen zur Belastbarkeit Hohlkorrigierter Beveloidgetriebe für Schiffsgetriebe mittlerer Leistung" Dissertation, Universität Stuttgart, 2002.
4. Zhu, C., C. Song, T. Lim, and S. Vijayakar. "Geometry Design and Tooth Contact Analysis of Crossed Beveloid Gears for Marine Transmissions," *Chinese Journal of Mechanical Engineering,* Bd. 25, Nr. 2, S. 328–337, 2012.
5. Wagner, M. "Beitrag zur Geometrischen Auslegung von Stirnradpaarungen mit Kleinen Achskreuzungswinkeln," Dissertation, Universität Stuttgart, 1993.
6. Börner, J., K. Humm, K. and F. J. Joachim. "Development of Conical Involute Gears (Beveloids) for Vehicle Transmissions," ***Gear Technology***, Nr. 6, S. 28–35, 2005.
7. Alxneit, H. "Optimierung des Übertragungsverhaltens Konischer Außenstirnräder mit Kreuzenden Achsen," Dissertation, Universität Stuttgart, 2010.
8. Weck, M. and C. Brecher. "Werkzeugmaschinen," Springer, Berlin (U.A.), 2005.
9. Brecher, C., M. Brumm, F. Hübner and J. Henser. "Influence of the Manufacturing Method on the Running Behavior of Beveloid Gears," *Production Engineering*, Bd. 7, Nr. 2-3, S. 265–274, 2013.
10. Brecher, C., M. Brumm, and J. Henser. "Simulation von Beveloidverzahnungen — B erechnung der Zahnfußspannungen," DMK 2011: *Dresdner Maschinenelemente Kolloquium*, 29-30, November 2011, 2011.
11. Brömsen, O. "Steigerung der Zahnfusstragfähigkeit von Einsatzgehärteten Stirnrädern durch Rechnerische Zahnfußoptimierung," Dissertation, RWTH Aachen, Shaker, Aachen, 2005.
12. Zuber, D. Fußtragfähigkeit Einsatzgehärteter Zahnräder unter Berücksichtigung Lokaler Materialeigenschaften, Shaker, Aachen, 2008.
13. Brecher, C., M. Brumm and J. Henser. "Local Approach for the Evaluation of Tooth Root Load Carrying Capacity of Beveloid Gears," *Fourth WZL Gear Conference*, Schaumburg, Illinois, 2012.
14. Brecher, C., T. Röthlingshöfer and C. Gorgels. "Manufacturing Simulation of Beveloid Gears for the Use in a General Tooth Contact Analysis Software," *Production Engineering,* Bd. 3, Nr. 1, S. 103–109, 2009.
15. Litvin, F. L. *Gear Geometry and Applied Theory*, PTR Prentice Hall, Englewood Cliffs, N.J, 1994.
16. VDI 2607. "Computer-Aided Evaluation of Profile and Helix Measurements on Cylindrical Gears with Involute Profile," Beuth Verlag, Berlin, 2000.
17. Smith, J. D. *Gear Noise and Vibration*, Marcel Dekker Inc., New York, 2003.
18. Hemmelmann, J. E. "Simulation des Lastfreien und Belasteten Zahneingriffs zur Analyse der Drehübertragung von Zahnradgetrieben," Dissertation, RWTH Aachen, Shaker, Aachen, 2007.
19. Niemann, G., H. Winter and B. –R. Höhn. Maschinenelemente. Konstruktion und Berechnung von Verbindungen, Lagern, Wellen. Konstruktion und Berechnung von Verbindungen, Lagern, Wellen, 2005.
20. ISO 6336: *Calculation of Load Capacity of Spur and Helical Gears;* 2007.
21. Weibull, W. *A Statistical Theory of the Strength of Materials,* Ingeniörsvetenskapsakademiens Handlingar, Bd. 151, Generalstabens Litografiska Anstalts Förlag, Stockholm, 1939.
22. Zierau, S. "Die Geometrische Auslegung Konischer Zahnräder und Paarungen mit Parallelen Achsen," Dissertation, Technische Universität Braunschweig, 1993.
23. DIN 3960: Begriffe und Bestimmungsgrößen für Stirnräder (Zylinderräder) und Stirnradpaare (Zylinderradpaare) mit Evolventenverzahnung; Beuth Verlag, Berlin, 1987.
24. Haupt, U. "Keilschrägverzahnung für Getriebe mit Einstellbarem Verdrehlankenspiel," Dissertation, TU Braunschweig, 1981.

Prof. Dr.-Ing. Christian Brecher has since 2004 served as Ordinary Professor for Machine Tools at the Laboratory for Machine Tools and Production Engineering (WZL) of the RWTH Aachen, as well as director of the Department for Production Machines at the Fraunhofer Institute for Production Technology IPT. Upond receipt of his engineering degree he began his career as a research assistant and later as team leader in the Department for Machine Investigation and Evaluation at the WZL. From 1999–2001 Brecher worked as a senior engineer with responsibility for machine tools and director (2001–2003) for development and construction at the DS Technologie Werkzeugmaschinenbau GmbH, Mönchengladbach. Brecher has received numerous honors and awards, the Springorum Commemorative Coin and the Borchers Medal of the RWTH Aachen among them.

Markus Brumm, an RWTH graduate with a degree in mechanical engineering, began his career in 2005 as a research assistant in gear investigation at the Laboratory for Machine Tools and Production Engineering (WZL) of the RWTH Aachen. He subsequently became that group's team leader in 2010.

Jannik Henser has studied mechanical engineering at RWTH Aachen University in Germany. During his studies he worked for four years as student assistant at the Laboratory for Machine Tools and Production Engineering (WZL) of RWTH Aachen University and interned at the iron powder manufacturer Höganäs AB in Sweden. He finished his studies successfully in 2010 with the diploma thesis "Micro Geometry Optimization of Powder Metal Gears" and started working as a PhD-student at WZL Gear Department for Prof. C. Brecher. His fields of research are tooth contact analysis, tooth root load-carrying capacity and beveloid gears. More than 10 papers were published on conferences and in journals under his direction.

High Gear Ratio Epicyclic Drives Analysis

Dr. Alexander Kapelevich

It has been documented that epicyclic gear stages provide high load capacity and compactness to gear drives. This paper will focus on analysis and design of epicyclic gear arrangements that provide extremely high gear ratios. Indeed, a special, two-stage planetary arrangement may utilize a gear ratio of over one hundred thousand to one. This paper presents an analysis of such uncommon gear drive arrangements and defines their major parameters, limitations, and gear ratio maximization approaches. It also demonstrates numerical examples, existing designs, and potential applications.

Introduction

Epicyclic gear stages provide high load capacity and compactness to gear drives. There is a wide variety of different combinations of planetary gear arrangements (Refs. 1–2). For simple, epicyclic planetary stages when the ring gear is stationary, the practical gear ratio range varies from 3:1 to 9:1. For similar epicyclic planetary stages with compound planet gears, the practical gear ratio range varies from 8:1 to 30:1.

This paper presents analysis and design of epicyclic gear arrangements that provide extremely high gear ratios. Using differential-planetary gear arrangements it is possible to achieve gear ratios of several-hundred-to-one in one-stage-drive with common planet gears, and several-thousand-to-one in one-stage drive with compound planet gears. A special two-stage planetary arrangement may utilize a gear ratio of over one-hundred-thousand-to-one.

This paper provides an analysis of such uncommon gear drive arrangements and defines their major parameters, limitations, and gear ratio maximization approaches. It also demonstrates numerical examples, existing designs, and potential applications.

One-Stage Arrangements

There are one-stage differential-planetary arrangements that provide much higher gear ratios. In these arrangements the output shaft is connected to the second rotating ring gear rather than the carrier, as with the epicyclic planetary stages. In this case a carrier does not transmit torque and is called a "cage" because it is simply used to support planet gears.

Figures 1a and 1b present differential-planetary arrangements with compound planet gears. In the arrangement in Figure 1a, the sun gear is engaged with a portion of the planet gear that is in mesh with the stationary ring gear. In this case the gear ratio is:

$$u = \frac{1 + \frac{z_{3a}}{z_1}}{1 - \frac{z_{2b}\,z_{3a}}{z_{2a}\,z_{3b}}} \tag{1}$$

where

u = gear ratio

z_1 = sun gear number of teeth

z_{2a} = number of teeth the planet gear engaged with the sun gear and stationary ring gear

z_{2b} = number of teeth the planet gear engaged with the rotating ring gear

z_{3a} = stationary ring gear number of teeth

z_{3b} = rotating ring gear number of teeth

In the arrangement in Figure 1b, the sun gear is engaged with a portion of the planet gear that is in mesh with the rotating ring gear. In this case the gear ratio is:

$$u = \frac{1 + \frac{z_{3a}\,z_{2b}}{z_1\,z_{2a}}}{1 - \frac{z_{3a}\,z_{2b}}{z_{3b}\,z_{2a}}} \tag{2}$$

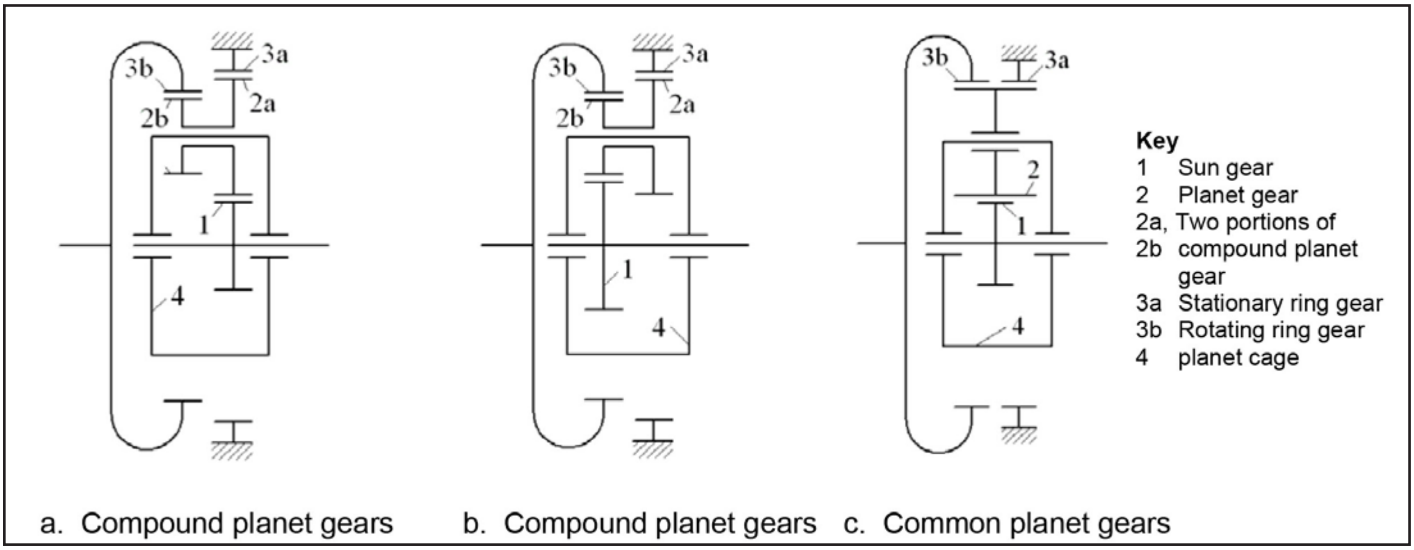


Figure 1 Differential-planetary arrangements.

If a gear ratio is negative, the input and output shaft rotation directions are opposite.

All gear meshes in differential-planetary arrangements have the same center distance. This condition allows for definition of relations between the operating modules, m_w, or diametral pitches, DP_w. For the arrangement in Figure 1a they are:

$$m_{w12a}(z_1+z_{2a})=m_{w2a3a}(z_{3a}-z_{2a})=m_{w2b3b}(z_{3b}-z_{2b}) \quad (3)$$

or

$$\frac{z_1+z_{2a}}{DP_{w12a}}=\frac{z_{3a}-z_{2a}}{DP_{w2a3a}}=\frac{z_{3b}-z_{2b}}{DP_{w2b3b}} \quad (4)$$

The relationship between operating pressure angles in the gear meshes z_1-z_{2a} and $z_{2a}-z_{3a}$ is defined by Equation 5 as:

$$\frac{\cos\alpha_{w2a-3a}}{\cos\alpha_{w1-2a}}=\frac{z_1+z_{2a}}{z_{3a}-z_{2a}} \quad (5)$$

where

α_{w1-2a} operating pressure angle in a mesh of the sun gear and the planet gear engaged with the stationary ring gear

α_{w2a-3a} operating pressure angle in the planet/stationary ring gear mesh

Similar to the arrangement in Figure 1b:

$$m_{w12b}(z_1+z_{2b})=m_{w2b3b}(z_{3b}-z_{2b})=m_{w2a3a}(z_{3a}-z_{2a}) \quad (6)$$

or

$$\frac{z_1+z_{2b}}{DP_{w12b}}=\frac{z_{3b}-z_{2b}}{DP_{w2b3b}}=\frac{z_{3a}-z_{2a}}{DP_{w2a3a}} \quad (7)$$

The relationship between operating pressure angles in the gear meshes z_1-z_{2b} and $z_{2b}-z_{3b}$ is defined by equation:

$$\frac{\cos\alpha_{w2b-3b}}{\cos\alpha_{w1-2b}}=\frac{z_1+z_{2b}}{z_{3a}-z_{2b}} \quad (8)$$

where

α_{w1-2b} operating pressure angle in a mesh of the sun gear and the planet gear engaged with the rotating ring gear

α_{w2b-3b} operating pressure angle in the planet/rotating ring gear mesh

In differential-planetary arrangements with compound planet gears, operating pressure angles in the planet/stationary ring gear mesh and in the planet the planet/rotating ring gear mesh can be selected independently. This allows for balancing specific sliding velocities in these meshes to maximize gear efficiency, which could be 80–90% — depending on the gear ratio (Ref. 2). The maximum gear ratio in such arrangements is limited by possible tip/tip interference of the neighboring planet gears. In order to avoid this interference the following condition should be satisfied:

For the arrangement in Figure 1a:

$$z_{2a}=\frac{z_1\sin\left(\frac{\pi}{n_w}\right)-2h_{2a}}{1-\sin\left(\frac{\pi}{n_w}\right)} \quad (9)$$

For the arrangement in Figure 1b:

$$z_{2b}=\frac{z_1\sin\left(\frac{\pi}{n_w}\right)-2h_{2b}}{1-\sin\left(\frac{\pi}{n_w}\right)} \quad (10)$$

where

n_w number of planets

h_{2a}, h_{2b} operating addendum coefficients of the planet gears z_{2a} and z_{2b}, accordingly.

Maximum gear ratio values for the differential-planetary arrangement with the compound planet gears (assuming $h_{2a}=h_{2b}=1.0$) are shown in Table 1.

The assembly condition for these gear arrangements is:

$$\frac{z_{3a}-z_{3b}}{n_w}=\text{integer} \quad (11)$$

Two parts of a compound planet gear should be angularly aligned for proper assembly. This is typically achieved by aligning the axes of one tooth of each part of the compound planet gear, which makes its fabrication more complicated. Assembly of such gear drives requires certain angular positioning of planet gears. All these factors increase the cost of this type of gear drive. Examples of differential-planetary gear actuators with compound planet gears are shown in Figure 2.

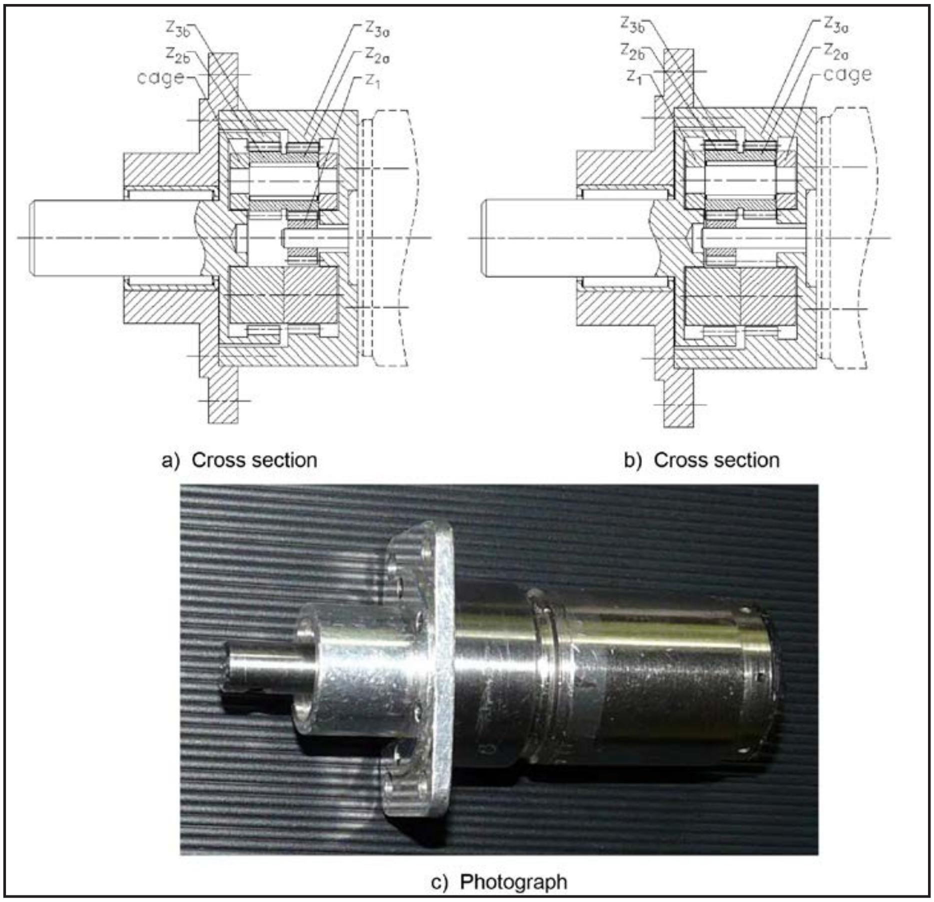


Figure 2 **Differential-planetary gear actuators with compound planet gears.**

Table 1 Maximum gear ratio values for differential-planetary arrangements with compound planet gears

Number of planets	Sun gear tooth number	Maximum gear ratio*
3	10	±1579:1
	15	±2857:1
	25	±5183:1
4	10	±144:1
	15	±273:1
	25	±518:1
5	10	±49:1
	15	±80:1
	25	±162:1

* Sign "+" if the input and output shaft rotation directions are the same, sign"-" if they are opposite.

A simplified version of the one-stage, differential-planetary arrangement is shown in Figure 1c; this arrangement does not use the compound planet gear. The common planet gear is engaged with the sun gear, and both the stationary and the rotating ring gears. This does not allow for specific sliding velocities in each mesh to be equalized, resulting in a reduction of gear efficiency of about 70–84% (Ref. 2). However, the assembly of such gear drives does not require certain angular positioning of planet gears, and their manufacturing cost is significantly lower for drives with the compound planet gears. An example of the differential-planetary gear actuator with common planet gears is shown in Figure 3.

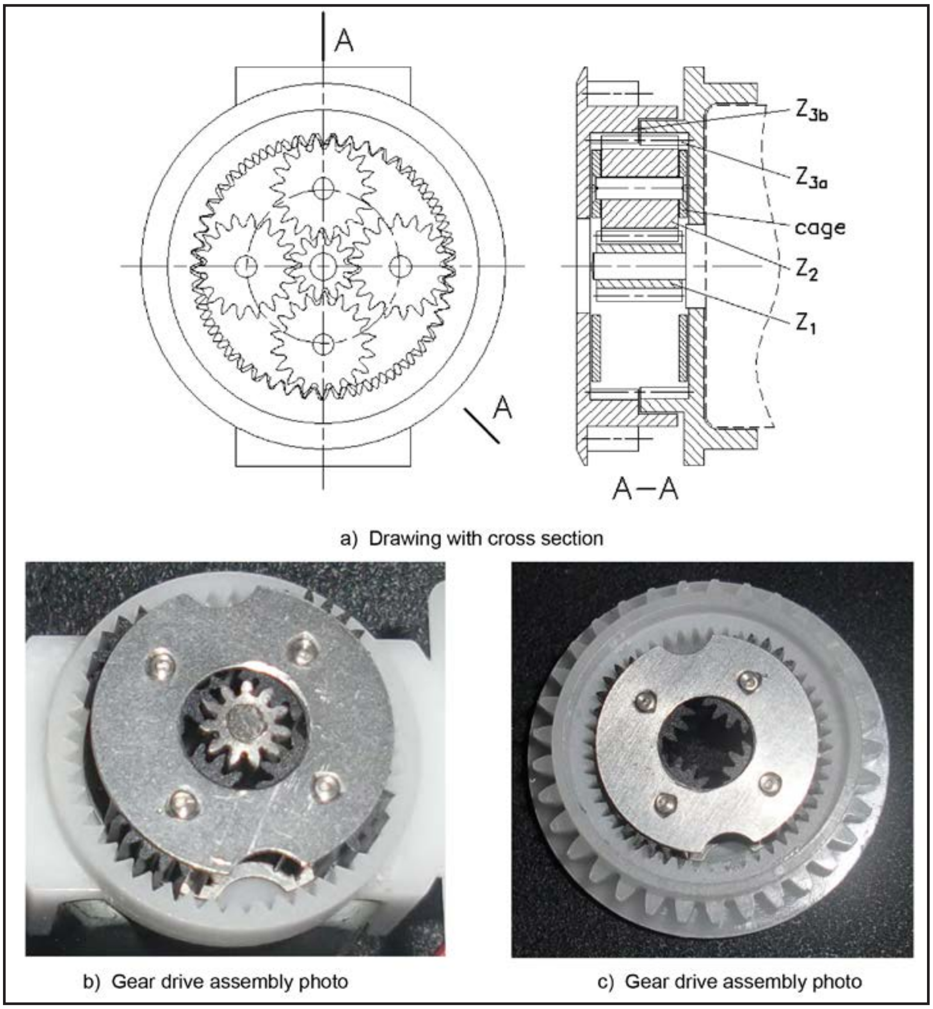


Figure 3 Differential-planetary gear actuator.

Relations between operating pressure angles in the gear meshes are defined by Equations 12–15:

$$\frac{\cos\alpha_{w2\text{-}3a}}{\cos\alpha_{w1\text{-}2}} = \frac{z_1 + z_2}{z_{3a} - z_2} \tag{12}$$

$$\frac{\cos\alpha_{w2\text{-}3b}}{\cos\alpha_{w1\text{-}2}} = \frac{z_1 + z_2}{z_{3b} - z_2} \tag{13}$$

$$\frac{\cos\alpha_{w2\text{-}3b}}{\cos\alpha_{w2\text{-}3a}} = \frac{z_{3a} + z_2}{z_{3b} - z_2} \tag{14}$$

where

$\alpha_{w1\text{-}2}$ operating pressure angle in sun/planet gear mesh

$\alpha_{w2\text{-}3a}$ operating pressure angle in planet/stationary ring gear mesh

$\alpha_{w2\text{-}3b}$ operating pressure angle in planet/rotating ring gear mesh

A gear ratio is

$$u = \frac{1 + \frac{z_{3a}}{z_1}}{1 - \frac{z_{3a}}{z_{3b}}} \tag{15}$$

Maximum gear ratio values for the differential-planetary arrangement with the common planet gears (assuming $h_{2a} = h_{2b} = 1.0$) are shown in the Table 2.

In differential-planetary arrangements (Fig. 1), tangent forces applied to the planet gear teeth from the stationary and rotating ring gears are unbalanced,

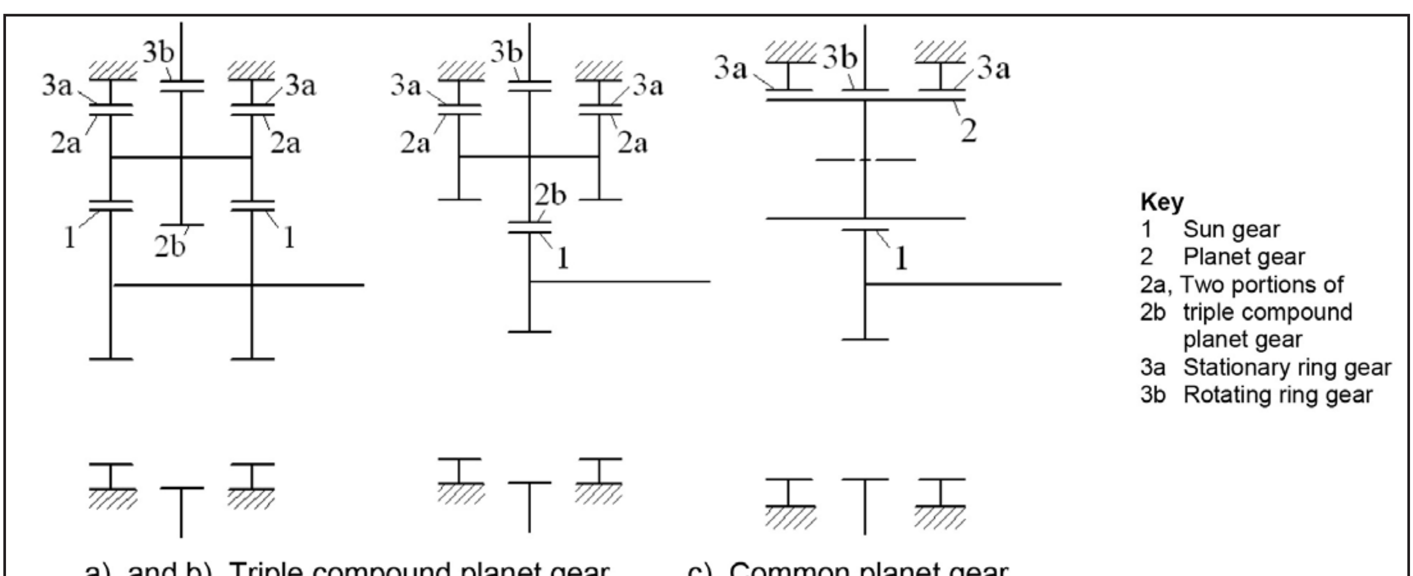


Figure 4 Differential-planetary arrangements without planet gear cage.

as they lie on different parallel planes and have opposite directions. A sturdy planet cage is required to avoid severe planet gear mesh misalignment. There are some gear drives that use the differential-planetary arrangements with balanced planet gear tangent forces (Fig. 4). In this case, the triple-compound planet gears (Figs. 4a and 4b) are used. They have identical gear profiles on their end portions that are engaged with the two identical stationary ring gears. The middle portion of such planet gears has a different profile than those on the ends, and is engaged with the rotating ring gear. The arrangement in Figure 4c has common planet gears engaged with the sun gear, two stationary ring gears, and one rotating ring gear. These types of differential-planetary drives typically do not have the cage and bearings, because the planet gear forces are balanced and planet gears themselves work like the roll bearings for radial support of the rotating ring gear.

Two-Stage Arrangements

In most conventional, two-stage, epicyclic arrangements, the gear ratio usually does not exceed 100:1 — although there are possible arrangements that allow a significant increase in the gear ratio (Ref. 3). Figure 5 shows the planetary gear arrangement "A" with the sun gears of the first and second stages connected together and the compound cage supporting the planet gears of both the first and second stages.

A sketch of the gearbox with arrangement A is presented in Figure 6. Both sun gears are connected to the input shaft and are engaged with the planet gears of the first and second stages, respectively. The first-stage ring gear is stationary and is

Table 2 Maximum gear ratio values for differential-planetary arrangements with compound planet gears		
Number of planets	Sun gear tooth number	Maximum gear ratio
3	10	±405:1
	15	±767:1
	25	±1432:1
4	10	±59:1
	15	±101:1
	25	±198:1
5	10	±20:1
	15	±32:1
	25	±70:1

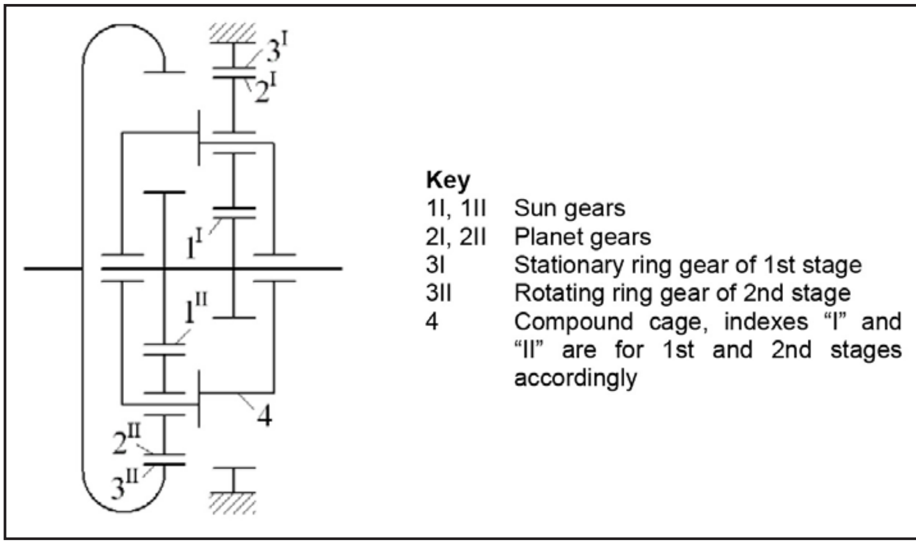


Figure 5 Two-stage planetary (arrangement A) with connected sun gears of first and second stages and compound cage.

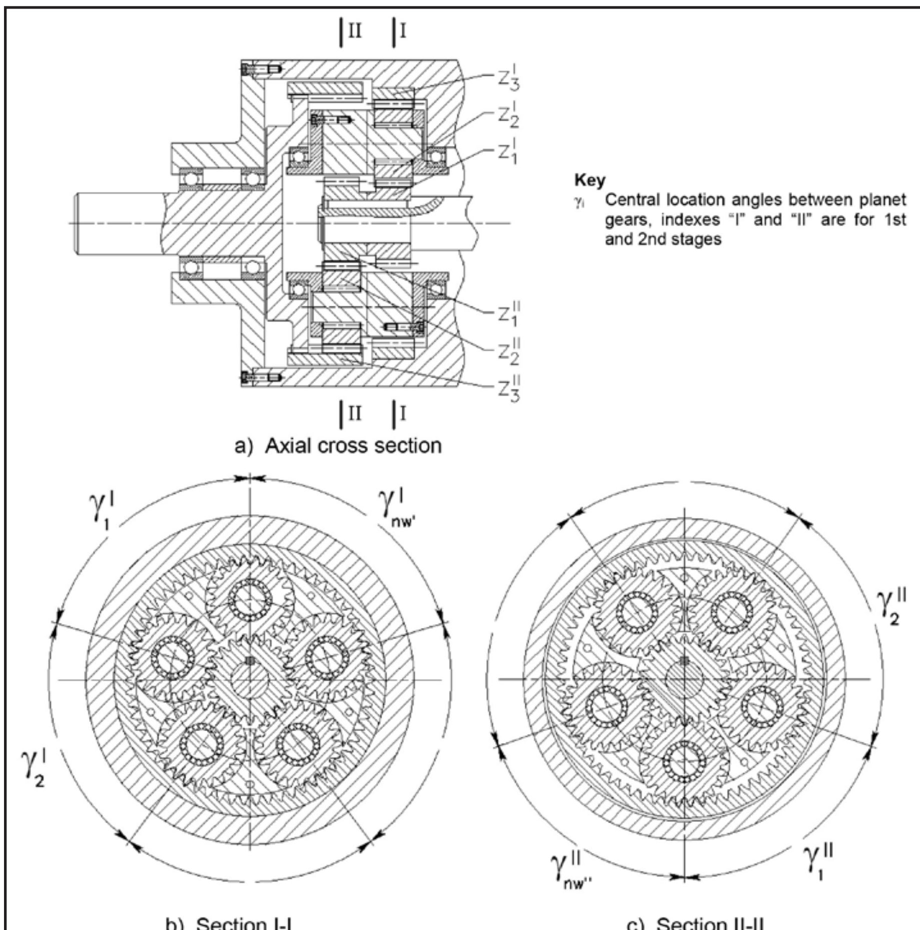


Figure 6 Two-stage planetary gearbox (arrangement A) with connected sun gears of first and second stages and compound cage.

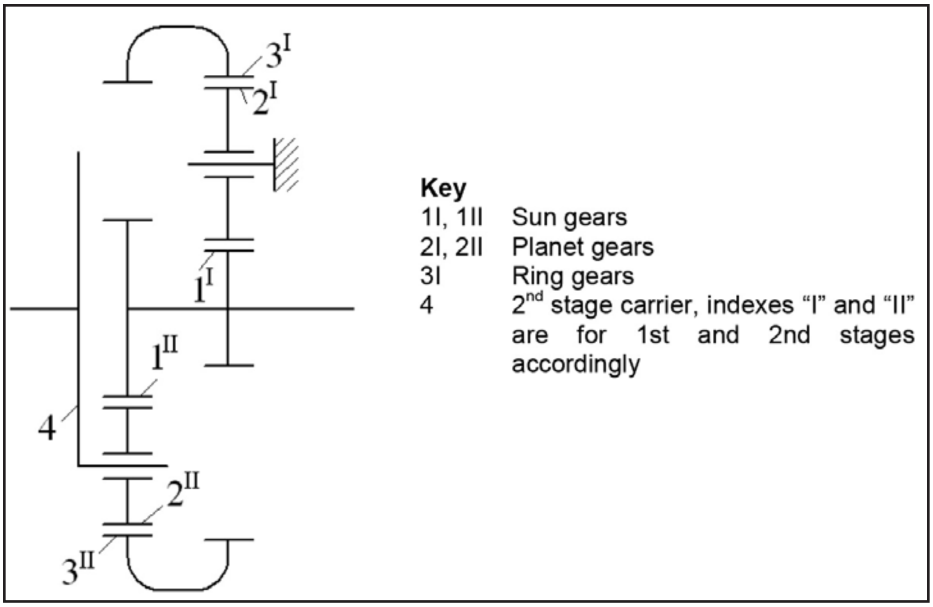


Figure 7 **Two-stage planetary (arrangement B) with sun gears and ring gears of first and second stages joined.**

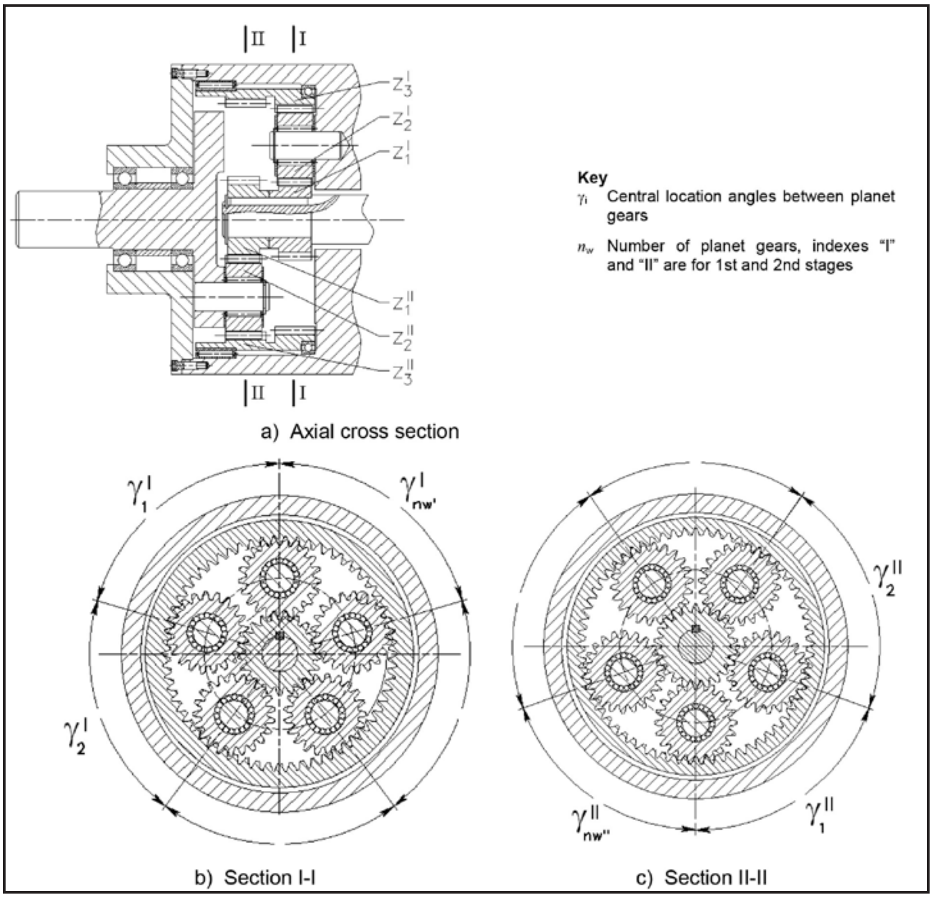


Figure 8 **Two-stage planetary gearbox (arrangement B) with sun gears and ring gears of 1st and 2nd stages joined.**

connected with the gearbox housing. It is engaged with the first-stage planet gears. The compound cage practically contains the first- and second-stage cages joined together. The ring gear of the second stage is engaged with the second-stage planet gears and connected to the output shaft. The gear ratio of arrangement A is:

$$u = \frac{z_3^{II}\left(z_1^{I} + z_3^{I}\right)}{z_1^{I} z_3^{II} - z_1^{II} z_3^{I}} \tag{16}$$

where

z_1^I, z_1^{II} = numbers of teeth of the sun gears of the 1st and 2nd stages

z_2^I, z_2^{II} = numbers of teeth of the planet gears of the 1st and 2nd stages

z_3^I, z_3^{II} = numbers of teeth of the ring gears of the 1st and 2nd stages

Figure 7 shows the alternative gear arrangement B with the sun gears of both stages joined together and the ring gears of both stages also connected.

A sketch of the gearbox with the alternative arrangement B is presented in Figure 8.

Both sun gears are connected to the input shaft and engaged with the planet gears of the first and second stages, respectively. The shafts supporting the first-stage planet gears are connected (pressed in, for example) to the gearbox housing. Both ring gears are joined and engaged with the planet gears of the first and second stages; the second-stage carrier is joined with the output shaft. The gear ratio of arrangement B is:

$$u = \frac{z_3^{I}\left(z_1^{II} + z_3^{II}\right)}{z_1^{II} z_3^{I} - z_1^{I} z_3^{II}} \tag{17}$$

The maximum gear ratios of these two-stage planetary arrangements A and B are achieved when the denominator of Equations 16 and 17 is equal to 1 or –1. This condition can be presented as:

$$\left| z_1^{I} z_3^{II} - z_1^{II} z_3^{I} \right| = 1 \tag{18}$$

When this denominator is 1, the input and output shafts are rotating in the same direction. When it is less –1, the input and output shafts are rotating in opposite directions. If a number of planet gears are more than 1 (n_w^I>1 and n_w^{II}>1), Equation 18 requires irregular, angular positioning of the planet gears in one or both planetary stages. This means that the central location angles γ_i between planet gears in one or both stages are not identical (see Figs. 6b, 6c, 8b and 8c). A definition of

Table 3 Maximum achievable gear ratios depending on numbers of teeth	
Ring gear number of teeth, z_3^{I} and z_3^{II}	Maximum gear ratio
100	±14,000:1
200	±66,000:1
300	±160,000:1
400	±280,000:1

Table 4 Gear ratio example for planetary arrangements A and B			
Arrangement		A (Figure 5)	B (Figure 7)
1st stage	Sun gear number of teeth	21	21
	Planet gear number of teeth	21	21
	Ring gear number of teeth	62	62
	Number of planet gears	5	5
2nd stage	Sun gear number of teeth	22	22
	Planet gear number of teeth	22	22
	Ring gear number of teeth	65	65
	Number of planet gears	5	5
Gear ratio		5395:1	-5394:1

the central angles with irregular angular positioning of the planet gears that provide proper assembly is described in Reference 3.

The neighboring planet gears located at the minimum central angles must be checked for the possibility of tip/tip interference. Irregular angular positioning of the planet gears may result in an imbalance in the planetary stage. This must be avoided by carrier assembly balancing.

Application of the two-stage planetary arrangements A and B allows very high gear ratio values to be achieved. In practice, these values are limited only by the number of teeth of ring gears z_3^{I} and z_3^{II}. Table 3 presents maximum achievable gear ratios depending on the number of teeth of ring gears z_3^{I} and z_3^{II}.

Unlike conventional, two-stage epicyclic arrangements, in the planetary arrangements A and B a total gear ratio does not depend on internal gear ratios in each stage; this allows increasing the number of planet gears. An example of the gear ratio for the planetary arrangements A and B is shown in Table 4.

The efficiency of these two-stage planetary gear arrangements is in opposite proportion to gear ratio, and is much lower than for conventional, two-stage, epicyclic gear arrangements. One potential area of application is in different positioning systems that need very low-output RPM and typically do not require high-output torque.

Potential applications

Potential areas of application of high gear ratio, one-stage differential-planetary arrangements include different aerospace drives, such as flap actuators, robotic mechanisms, etc.

Extremely high gear ratio, two-stage planetary arrangements can be applied in different positioning systems that need very low-output RPM and typically do not require very high-output torque, such as the tracking system gear drives of solar batteries or the mirrors of solar power stations.

Summary

High gear ratio, one-stage differential-planetary arrangements with compound and common planet gears are described. Gear ratio equations and maximum values are defined.

Extremely high gear ratio, two-stage planetary arrangements are described. Gear ratio equations and maximum achievable values are defined.

Potential applications of high gear ratio, one-stage and two-stage planetary drives are suggested.

References

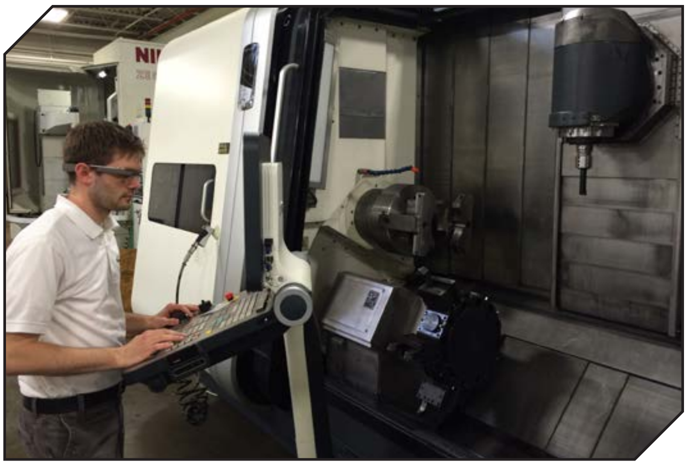
1. Henriot, G. "Gears and Planetary Geartrains," Brevini, Reggio nell'Emilia, 2001.
2. Kudriavtsev, V.N. "Planetary Gears," *Mashinostroenie*, Leningrad, 1966.
3. Kapelevich, A.L. *Direct Gear Design,* CRC Press, 2013.

Alex Kapelevich possesses more than 30 years of custom gear research and design experience, as well as over 100 successfully accomplished projects for a variety of gear applications. His company, AKGears, provides consulting services—from complete gear train design (for customers without sufficient gear expertise) to retouching (typically tooth and fillet profile optimization) of existing customers' designs—in the following specific areas: traditional or direct gear design; current design refinement; R&D; failure and testing analysis. The company provides gear drive design optimization for increased load capacity; size and weight reduction; noise and vibration reduction; higher gear efficiency; backlash minimization; increased lifetime; higher reliability; cost reduction; and gear ratio modification and adjustment. Kapelevich is the author of numerous technical publications and patents, and is a member of the AGMA Aerospace and Plastic Gearing Committees, SME, ASME and SAE International. He holds a Ph.D. in mechanical engineering from Moscow State Technical University and a Master Degree in mechanical engineering from the Moscow Aviation Institute.

ITAMCO

ENGINEER WINS CASH PRIZE FOR GOOGLE GLASS APPLICATION

Joel Neidig, an engineer at Indiana Gear (a division of ITAMCO), was awarded $75,000 for his research project titled "Expanding Manufacturing's Vision: MTConnect and Google Glass."



Neidig's winning project was his contribution to the MTConnect Challenge 2. The National Center for Defense Manufacturing and Machining (NCDMM), the Office of the Secretary of Defense (OSD), Defense-wide Manufacturing Science and Technology (DMS&T), the Association for Manufacturing Technology (AMT) and the U.S. Army Benét Labs sponsored the contest. Participants were asked to develop innovative and unique software applications using the MTConnect standard that could be easily adopted by manufacturing enterprises, especially lower-tier producers. MTConnect is an open, royalty-free set of communication standards intended to foster greater interoperability and information sharing between manufacturing equipment, devices, and software applications.

Neidig coupled MTConnect functionality with Google Glass — a camera, touchpad, microphone, email and Internet connection built into a spectacle frame. Neidig's application, called the MTConnect Glassware app, will reveal a view of the manufacturing process that has never been seen before. The app user will be liberated from laptops and handheld smart devices and be able to travel the entire shop floor, gathering and sharing machine data provided by MTConnect, and accessing the Internet for more information.

"We're using the MTConnect Glassware app on our shop floor at Indiana Gear and have received inquiries from companies that want a customized solution. We plan to use the prize money to develop the next generation of the app, which will be available in safety glasses," said Neidig.

The award-winning product benefits every department in a manufacturing facility, from the shop floor to the management suite. Getting and sharing information will be as intuitive and non-threatening as using a smartphone or playing a video game, but glasses do not distract from a task like handheld devices. For example, a new machine operator watches YouTube training videos while at the machine, supplementing his training program. A machine operator sends an email to the maintenance staff as soon as he sees a problem. The CEO travels the shop floor, getting accurate real-time machine data and comparing it to the company's accounting, quality control, sales and engineering data. Project managers evaluate workflow and machine readiness before scheduling future work.

They will access 100 percent accurate data and share information with their customers to ensure smooth delivery between vendors. The maintenance staff becomes even more astute at monitoring machines and they respond quicker to problems while developing stronger preventative maintenance programs. With Google Glass, they can "see" instruction manuals overlaid on the equipment when installing or repairing machinery. The sales staff provides guided Google Glass tours, impressing prospects with the capabilities of the facility and helping them visualize the final product.

STAR SU & FFG Werke

EXTEND PARTNERSHIP

Illinois-based Star SU LLC and FFG Werke GmbH, a German machine tool manufacturer, signed a contract extending their partnership for machine tool sales and service in the Americas. Having worked together successfully with FFG's Modul hobbing machines brand, a seller of gear manufacturing equipment for more than 20 years, the Star SU portfolio now also covers VDF Boehringer and Hessapp turning machines as well as Hüller Hille machining centers and Witzig & Frank multi-way, multi-spindle machines. The partners anticipate substantial synergies based on Star SU's extensive footprint in the United States, Canada, Mexico and Brazil, and its strong ties into the automotive, tier supplier, energy and other industries, combined with the technological expertise and application know-how of FFG's brands in the target industries.

David Goodfellow, CEO of Star SU, expects that his customers in the Americas will benefit substantially from the new portfolio: "In the past, we proved that, together with our German partners, we can deliver the right solution, at the right time and quality, and secure long-term productivity with our highly skilled service team. We have thus become a reliable and innovative partner for the automotive industry, tier suppliers, and other manufacturers across the continent, especially for transmission component applications. I am excited to present this new range of high-end manufacturing technology brands for turning and milling applications to our customers." As full-range partner, Star SU is able to handle entire projects for FFG products, from the initial request to import, on-site installation and commissioning, including financing and leasing services, all in

local currency. Besides manufacturing and engineering capacities for precision tools and machinery, Star SU has six service centers in the United States, Mexico and Brazil, and a total of approx. 500 employees.

The American market is of strategic importance for FFG, says **Martin Winterstein,** vice president of sales and marketing: "Apart from strengthening our service capacities for our installed base in the Americas, we see considerable potential in various industries. Whether it's automotive and truck components, or applications in the energy, rail, agricultural or engineering sectors, we are sure that Star SU is the right partner to bring our solutions to the growing North and South American markets." FFG Werke GmbH's manufacturing base in Germany comprises four manufacturing plants and more than two hundred years of metal cutting knowhow. At the IMTS in Chicago, starting September 8, 2014, Star SU and FFG will present their extended portfolio to the general public. For more information, visit *www.star-su.com* or *www.ffg-werke.com*.

EMAG

SHOWCASES MANUFACTURING SYSTEMS AT TECHNOLOGY DAYS

To celebrate their 20th anniversary, EMAG USA opened its doors to manufacturers and vendors to join them for their open house Technology Days May 13-14 at their Farmington Hills, Michigan headquarters. Throughout the two-day event manufacturing decision-makers attended technology seminars held by EMAG's industry experts. Here attendees gained valuable knowledge in both industry trends and unsurpassed EMAG manufacturing solutions. Participants expanded their process know-how in eight presentations, including CV-Joint production, brake disc machining with continuous workpiece flow, electro-chemical machining and production laser welding. EMAG's first spindle down machine was also introduced in the VM Platform world-premier seminar. Paul Traub, business economist from the Federal Reserve Bank of Chicago

and Lisa Katz, executive director of the Workforce Intelligence Network of Southeast Michigan provided insights into the economy and manufacturing's role in supporting economic recovery and a highly-skilled workforce. These hot-topics not only provided the opportunity for EMAG to re-introduce its new Mechatronics Apprenticeship Program, but also provided guests the chance to pose questions and sparked many discussions amongst participants.

Another highlight of the event was the live metalworking demonstrations on multiple EMAG machines. Attendees witnessed world-class manufacturing solutions for carrier production on the VLC 250, combination machining with the VSC 400 DDS hard-turn and grinding, and two-sided shaft machining on the VT 2-4 from EMAG's new Modular Standard product family. The unique event allowed participants to secure greater process knowledge in their markets and provided numerous networking opportunities within the industry. "We like to open

our doors and welcome in both known and new customers to see the capabilities of our production facility and meet our team while sharing with them the wide range of solutions EMAG has to offer for any manufacturing challenge," stated Peter Loetzner, CEO of EMAG LLC. "We are proud to celebrate 20 years of operation in the US with our customers and share with them how we are advancing technologies to make the next 20 years even more successful."

Ipsen U

OFFERS EDUCATION IN HEAT TREATMENT

With an ever-increasing need for first-hand knowledge of thermal processing and the intricate functions of specific equipment, companies are recognizing the importance of sending employees to Ipsen U. Not only was Ipsen U's April class full, but the upcoming June course is also completely booked. Ipsen U's three-day course allows attendees to ask questions about specific equipment and processes, as well as learn directly from Ipsen experts in a casual, open-forum environment. This proves useful for those wanting to expand their thermal processing knowledge base, as well as for first-time furnace buyers.

Attendees from the most recent Ipsen U course came from a wide variety of locales, including California, Georgia, Texas, Florida, Wisconsin, Michigan and Mexico. A survey of Ipsen U's April participants found they thought the course provided a

"much better understanding of the overall function of vacuum furnaces, as well as the key principles to how they work and troubleshooting when issues arise." Another participant surveyed praised the instructors' "incredible amount of knowledge, hints, tips and real world advice."

Ipsen U's instructors cover an extensive range of topics, from an introduction to vacuum furnaces and heat treating to furnace sub-systems, leak detection, maintenance and so much more. Most notable about Ipsen U is participants' ability to bring up any topic pertaining to their specific vacuum furnace operations for discussion. This allows for some fluidity and permits the instructors to shape the course to individuals' specific needs and concerns.

In addition to making experts available for one-on-one discussions, Ipsen U also puts information concerning furnace maintenance and upkeep into the participants' own hands, providing them with the tools necessary for the continued, successful operation of their own equipment. With equipment that contains a multitude of parts that are expensive to replace and can be damaged in various ways, sending employees to learn from the experts has its benefits.

The value of attending such a course is immeasurable because employees have the opportunity to learn first-hand about the equipment and processes they will be utilizing in the workplace. Learn more and register for one of the two remaining Ipsen U courses this year, August 5-7 or October 14-16, at *www.ipsenusa.com/ipsenu.*

CMSC

SELECTS PRESENTATIONS FOR 2014 CONFERENCE

The Coordinate Metrology Society (CMS) announced that it has selected a comprehensive slate of 24 technical presentations for the 2014 Coordinate Metrology Systems Conference (CMSC), July 21-25, 2014, at the Embassy Suites, North Charleston, SC. The organization celebrates its 30th Anniversary this year, and continues its long tradition of serving the specialized needs of 3-D measurement/inspection professionals and scientists worldwide. In addition to original technical presentations, the CMSC serves as a platform for discovering and discussing new and future technologies. During the five-day event, attendees and exhibitors share their experiences, research, concepts and theory in an open, educational atmosphere. A full listing of the CMSC 2014 technical presentations and Conference Registration information can be found at *www.CMSC.org*.

The CMS completed its peer review of a record number of abstract submissions, and selected 24 technical presentations from industry experts representing leading international companies and organizations such as NASA Goddard Space Flight Center, Rolls-Royce, NIST, National Solar Observatory, Espace2001, The Boeing Company, Lockheed Martin, Premium AeroTech, Electroimpact, National Propulsion Laboratory (NPL-UK), Hitachi Power Solutions Co., BMW Manufacturing, Sigma Space, and the National Research Council of Canada. The roster also includes speakers from prestigious educational institutions including the University of California Davis, Escola Politecnica da Universidade de Sao Paulo, Wichita State University, and the School of Mechatronic Engineering and Automation from Shanghai University.

The CMSC presentation roster will include wide-ranging topics related to best practices, trends, and the application of 3-D metrology techniques and technology. Subjects range from "Automated Metrology in a Business Jet Final Assembly Line" to "Metrology-Enhanced Robotic Milling" to "100% Automated Production Measurement of a Car Body Welding Line with Absolute Reference." Speakers will cover the use of portable 3D coordinate measurement systems, laser trackers, photogrammetry, laser radar, 3-D scanners and sensors, inspection software and more. The CMSC 2014 Presentation Agenda can be found online at 2014 CMSC Presenters.

For three decades, the event has attracted thousands of attendees from prominent science/research laboratories, educational institutions, and industries such as aerospace, satellite, automotive, shipbuilding, power generation, and general engineering. The CMSC Conference Agenda also includes Level-1 and Level-2 Certification examinations, a new Measurement Study, hands-on workshops, and networking events. More than 40 exhibitors will pack the exhibition hall with portable measurement systems (PCMMs), software, and accessories supported by OEMs and service providers.

Hainbuch

WELCOMES NORTHEAST REGIONAL SALES MANAGER

Hainbuch America is pleased to announce that **Shawn Harty** has joined the team as the northeast regional sales manager. Harty has had a successful 20 year career in manufacturing management and sales, most recently as the owner of PMDG, a turnkey integrator of manufacturing solutions to several large manufacturing companies on the east coast. PMDG successfully used Hainbuch components in a number of applications, and Harty's appreciation for the quality and versatility of Hainbuch products led to his decision to join our team. Harty attended Penn State University and served in the Unites States Marine Corps where he trained in aviation electronics. Harty has also been successful selling cutting tools for Iscar Metals in the same territory he will cover for Hainbuch. He looks forward to leveraging his technical expertise in manufacturing design to distributors and end users in his territory and to bringing Hainbuch's adaptable precision workholding devices to an expanding list of customers.

GMTA

TREASURER RECEIVES MBA FROM EASTERN MICHIGAN

GMTA proudly announces that **Claudia Hambleton,** corporate treasurer and administrative manager, received her MBA from Eastern Michigan University (EMU) on April 27, 2014. Hambleton has been a GMTA associate since the company opened its doors in January, 1991. "I am very grateful to GMTA for supporting my efforts to acquire an MBA. I look forward to using my new skills to further enhance the business at our company. It was also very rewarding to network with the business school students and staff at EMU, where GMTA now has a working relationship with their German business development track," Hambleton says. GMTA routinely hosts a program in German for students in the EMU business school who plan to work for or with German companies as a career path.

AGMA

Recognizes Member Achievements at Annual Meeting

The AGMA/ABMA Annual Meeting took place April 10-12 in St. Petersburg, Florida. Between committee meetings, networking opportunities and social events, many individuals in the gear industry were presented awards. Here are some of the highlights:

Lifetime Achievement

Fred Young, CEO of Forest City Gear, was presented the Lifetime Achievement Award by Louis Ertel, AGMA chairman and chairman of Overton Chicago Gear Corp. Young has been an active member of the AGMA for more than 25 years, serving various committees and councils, reviewing technical papers and attending over 150 meetings and conferences.

“Fred grew up in the gear industry. His parents, Stetler and Evelyn, started Forest City Gear when Fred was 11 years old. As a boy Fred would help out around the shop, doing everything from cleaning the chips out of the oil sumps, to washing and centering motor shafts, to cutting gear teeth,” Ertel said during the award presentation.

With Young at the helm, Forest City Gear has grown to a 35,000-sq.-ft. facility with $14 million in annual sales. “He prides himself on having the latest technology, reinvesting 25-40 percent of their annual revenue back into new equipment each year. Fred and his employees truly embody their motto, ‘Excellence without exception,’” Ertel said. “The company is truly a family affair – his wife, Wendy, is now president of the company, his daughters, Appy, Kika and Mindy, have all had roles with the company, and now his son-in-law Rustin Mikel is heavily involved in the company too.”

AGMA Chairman Lou Ertel presents Forest City Gear’s Fred Young with the Lifetime Achievement Award.

Ertel went on to explain that Young has been active in training the next generation of gear manufacturers. “Forest City Gear prides itself on mentoring of high school students who work part time while attending school, hiring summer interns who are attending college to expose them to the demands of manufacturing, and encouraging current employees to advance their knowledge of mathematics, science and technology.”

AGMA Chairman Lou Ertel presents GMTA’s Scott Knoy with the Next Generation Award.

Young has served as AGMA Chairman, Chairman of the Foundation, a member of the Board of Directors, the Education Council, the Nominating Committee, the Awards Committee, the Strategic Planning Committee, the Marketing Council, the Membership Committee, the 60th Anniversary Committee, and the Small Business Council. He has received the Past Presidents Award, Board of Directors Award, Old Goats Award, and the Administrative Division Executive Committee Award.

“His greatest contribution to AGMA has been as its ambassador throughout the world and at the over 150 meetings and conferences that he has attended. So for all of these reasons and many more, AGMA presents Fred Young with the highest honor we have, the Lifetime Achievement Award. Thank you Fred, for everything you have done for AGMA, and the industry,” Ertel said.

The Next Generation Award

The Next Generation Award, in the words of the AGMA Awards Committee, “honors individuals who, while early in their career, are emerging as contributors, innovators and/or leaders in the gear industry, and who serve as role models for others in the next generation of the gear industry.” It was presented to Scott Knoy, vice president of German Machine Tools of America (GMTA). Knoy has been active in service to the gear industry for 20 years, serving first for 11 years with major machine tool supplier Gleason and now in his ninth year with GMTA, where he oversees the sales, strategic planning, project management and marketing efforts for this major supplier of machinery to the gear, powertrain and other industries.

In addition, along with company president Walter Friedrich, he spearheads the ongoing effort at GMTA to educate and nurture the next generation of leaders in the gear industry, an endeavor to which the company has made a great commitment. GMTA supports local schools in their tech training and also Eastern Michigan University in its unique "business as a second language" program.

Knoy grew up in the industry with his father, Jerry Knoy, working at the Gleason Corporation and followed in his father's footsteps, joining Gleason Cutting Tools. Scott began his career with Gleason as a regional sales manager, where he was the key contact for clients such as TRW, Ford Motor Company, General Motors, Dana Corporation, Parker-Hannifin and General Electric. Following his time at Gleason, Scott joined German Machine Tools of America (formerly American Wera), where he now serves as GMTA vice president of sales and marketing. He is active with the AGMA Strategic Resources Network and has been attending the Annual Meeting, Gear Expo and other AGMA activities for many years.

Education Award

During the 2014 AGMA Annual Meeting, several members were recognized for their achievements over the past year. Seven members of the Education Task Force were recognized for their work on the AGMA Skills Assessment Tool. Released in 2013, the Skills Assessment Tool allows members and others in the industry to set standards for a variety of skills and abilities and for several broadly defined job titles. The tool allows employers to assess each employee's proficiency in each skill. It also allows the employer to set a minimum score and automatically determine where each employee needs additional education or training. The tool is available for free on the AGMA website, *www.agma.org*. The Skills Assessment Tool is an important achievement for AGMA, and would not have been possible without the dedication of the Skills Assessment Task Force. The BMEC Award is presented to individuals who have made noteworthy efforts to enhance the art of management in the gear industry. In that spirit, AGMA recognized the members of the Education Task Force with individual BMEC awards for their dedicated work developing the Skills Assessment Tool.

From left to right: Bob Sakuta, Bill Miller, Rustin Mikel, Tom Marino and Jeff Reynolds receive awards for developing the Skills Assessment Tool.

The award was presented to: Jeff Reynolds, Rolls Royce Corp.; Briggs Forelli, Precision Gear; Bob Maggetti, Triumph Group; Tom Marino, Gear Technology; Rustin Mikel, Forest City Gear; Bill Miller, Kapp Technologies, and Bob Sakuta, Delta Gear.

50 Years of Membership

Organizations like AGMA would not be able to exist and continue to thrive without the support of the members, and this year AGMA recognizes a company that has been a member of AGMA for 50 years. Progressive Engineering Company (PEC) is a precision gear and machine parts manufacturer. Established in 1923, the company has celebrated many years of service, and joined AGMA in 1964. The company provides "job shop made to order parts" for many Fortune 500 companies, manufacturers, parts companies, and metal processors in the Richmond area and throughout the United States. PEC also rebuilds gearboxes; has the largest surface grinder in the area, and a full complement of CNC and manual machines for general machine parts production.

In 1992 an Ion Nitride Heat Treatment department was added to provide low distortion hardening of high quality parts in order to reduce cost and increase the life of machine parts. In 2001 PEC expanded with the addition of Progressive Heat Treating Co., a commercial heat treating company that specializes in plasma ion nitriding, vacuum hardening, oil hardening, induction hardening and carburizing. President Mel Belcher from Progressive Engineering Company accepted the 50 Year Award on the company's behalf during the 2014 AGMA Annual Meeting.

New Members of the Board of Directors

AGMA also announced the election of four members to the AGMA Board of Directors. The new board members are: Kenneth J. Flowers, owner and vice president

Machine Tool Builders, Inc.; Bill Gornicki, vice president sales and marketing, ALD-Holcroft Vacuum Technologies Co., Inc.; John E. Grazia, president, GearTec Inc.; and Brian L. Schultz, president, Great Lakes Industry, Inc. The new board members began their three-year term at the AGMA Annual Meeting.

July 7–11—Additive Manufacturing and 3-D Printing International Conference. The Nottingham Belfry Hotel, England. The conference includes a dedicated exhibition open only to event delegates. The exhibition is the perfect opportunity to talk in detail with metallic and polymeric additive machine manufacturing companies, material suppliers and service organizations. The exhibition will take place in a parallel room to the conference and is the hub of delegate and vendor networking. Presentations made at the conference (subject to speaker permission) will be made available on a USB card supported by a printed booklet containing key conference information, speaker biographies and abstracts. A complimentary copy will be issued to each delegate on registration. Further copies will be available for purchase at the conference and after the event. For more information, visit *www.am-conference.com*.

July 14–17—EMTE-EASTPO. Shanghai, China. This exhibition offers a high quality showcase of innovative technology and practical solutions for Asia's manufacturing industry. Jointly organized by CECIMO, EASTPO and the MP Organization, the inaugural joint exhibition will unveil sustainable and cost-effective solutions to buyers in China and the rest of Asia. It provides a one-stop business platform for leading manufacturers seeking to penetrate the dynamic marketplace in Asia. The joint exhibition will unveil a high quality showcase to buyers from China and other parts of Asia. It will follow the strong tradition of major exhibitions in Europe, where strict exhibitor admission rules, intellectual property protection and live demonstrations of exhibits are featured. With key industrial sectors like electronics, steel, chemicals, pharma and industrial machinery, Shanghai is an important manufacturing hub in China and is geographically located near Hangzhou, Nanjing, Ningbo, Shaoxing, Suzhou and Yangzhou, all of which are manufacturing bases for automotive, industrial machinery and metals. For more information, visit *www.emte-eastpo.com*.

July 21–25—CMSC Conference 2014. Embassy Suites, North Charleston, South Carolina. The Coordinate Metrology Systems Conference (CMSC) provides a professional venue where ideas, concepts and theory flow freely among participants. The educational atmosphere encourages attendees to network and learn about the latest innovations in the field of portable 3-D industrial measurement technologies. CMSC provides technical presentations by industry experts, advanced workshops and seminars along with an exhibition hall filled with the world's leading providers of metrology systems. Those companies providing technical presentations include NASA Goddard Space Flight Center, Rolls-Royce, NIST, National Solar Observatory, Espace2001, The Boeing Company, Lockheed Martin, Premium AeroTech, Electroimpact , National Propulsion Laboratory (NPL-UK), Hitachi Power Solutions Co., BMW Manufacturing, Sigma Space, and the National Research Council of Canada. For more information, visit *www.cmsc.org*.

August 26–28—International Gear Conference 2014. Lyon-Villeurbanne, France. Mechanical transmission components (gears, bearings, CVTs, belts, chains, etc.) are present in every industrial sector and range from nano-gears to large gearboxes. Over recent years, increasing competitive pressure and environmental concerns have provided an impetus for cleaner, more efficient and quieter units. Moreover, the emergence of relatively new applications in wind turbines, hybrid transmissions, jet engines has led to even more severe constraints. The main objective of this conference is to provide a forum for the most recent advances, addressing the challenges in modern mechanical transmissions. Topics include gear noise, gear design, gear materials, gear failure, lubrication, gearbox efficiency and more. For more information, visit *http://int-gear-conf14.sciencesconf.org*.

September 8–13—IMTS 2014. The International Manufacturing Technology Show (IMTS) is the largest manufacturing technology show in the Western Hemisphere. IMTS 2012 drew more than 100,000 industry decision-makers in areas like metal cutting, tooling, metal forming, abrasives, controls, CAD-CAM, EDM, gear generation, industrial automation and more. The IMTS conference brings the industry together to discuss new opportunities and network with the manufacturing community. Other highlights include the Smartforce Student Summit, Exhibitor Workshops, the Emerging Technology Center and IMTSTV. IMTS is co-located with Industrial Automation North America and Motion, Drive & Automation North America. For more information, visit *www.imts.com*.

October 8–10—RMGFS 2014. Boulder, CO. The Rocky Mountain Gear Finishing School (RMGFS) is the premier gear finishing school in the western United States. Kapp-Niles presents this multi-layered program designed to optimize learning and strengthen your understanding of gear finishing processes no matter what level you are at. The curriculum features sessions which are interconnected and lead each step to the next. Participants study the underlying principles and mechanics of different gear finishing processes, apply them through practical sessions on a Kapp-Niles machine, and take part in group workshops for more in-depth discussions. Attendees are encouraged to bring applications to the school for small group, or one-on-one discussions. Presenters include Jim Buschy, Bill Miller, Dwight Smith, Paul Brazda, Michael Ruppert, Sascha Ungewiss, Thomas Schenk, Nidam Meharzi, Eric Dixon and Hans-Helmut Rauth. For more information, visit www.kapp-usa.com.

October 27–30 Gear Dynamics and Gear Noise Course. Ohio State University Campus. The Gear Dynamics and Gear Noise Short Course has been offered for 35 years and is considered extremely valuable for gear designers and noise specialists who encounter gear noise and transmission design problems. Attendees will learn how to design gears to minimize the major excitations of gear noise: transmission error, dynamic friction forces and shuttling forces. Fundamentals of gear noise generation and gear noise measurement will be covered along with topics on gear rattle, transmission dynamics and housing acoustics. This four-day course includes extensive demonstrations of specialized gear analysis software in addition to the demonstrations of many Ohio State gear test rigs. A unique feature of the course is the interactive workshop session (on Day 3) that invites attendees to discuss their specific gear and transmission noise concerns. The round table discussions on Day 4 are intended to foster interactive problem solving discussions on a variety of topics. Cost is $1,950 per person. For more information, visit www.nvhgear.org.

CORRECTION

The May issue of *Gear Technology* included a technical paper—**"Load Distribution Analysis of Spline Joints"**—written by Jiazheng Hong, David Talbot and Ahmet Kahraman. Unfortunately, we published an incorrect version of figures/tables, thus creating contradictory statements made in the body copy relative to the data contained in the figures/tables. In addition, the same mistake occurred with the paper's research references, creating confusion there as well. To see a corrected version, please go to *geartechnology.com*.

Gear Technology regrets the errors.

For Immediate Release:

Orwell Inc. Promises All Things Optimistic with the BBMS

Contrary to popular belief, top executives from Orwell Inc. (founded in 1984) believe the manufacturing community has taken the paranoia and dystopian despair one step too far.

"Sure, we've had drones, color-coded terror alerts, waterboarding, CCTV cameras, global positioning software, Edward Snowden, metadata, social media and the NSA, but at the end of the day, there's no cautionary tale or government takeover to worry the minds of an entire generation of robots (err, I mean skilled workers)," says an ominous vice president of something or other lurking in the shadows of his energy-efficient managing office.

Today, Orwell Inc. boasts the blueprint for absolute power with its Big Brother Monitoring System (BBMS). This state-of-the-art sensor technology is simply implanted in the wrist and head of each employee, giving Human Resources an unparalleled advantage when it comes to productivity and quality control.

"Studies show that the human brain is responsible for 95.5 percent of the average bottleneck on any given shop floor," says O'Brien, product specialist at Orwell. "The BBMS eliminates free thought with the push of a button, allowing your workforce the luxury to concentrate on the task at hand."

Monitoring your production team has never been easier, as the BBMS decodes brain activity to determine when an employee might be making unproductive choices.

"Maybe he or she is about to turn a 15 minute break into 16," O'Brien says. "Perhaps they're thinking about taking the afternoon off to catch a baseball game. Some troublemakers will just stand around the office attempting to solve world peace! The BBMS has the analytics in place to keep humans from—well, frankly—being human," O'Brien adds.

The Winston Smith Case Study

Winston Smith, a young engineer with a major gear manufacturing conglomerate, was recently fitted with his own BBMS. Orwell Inc. is happy to report that his productivity has increased 400 percent. "I was in the gear industry, but I really wasn't IN the gear industry, you know what I mean?" Smith says. "I used to space out a little bit at the desk, think about history, literature, romance even those cooking shows on the Food Network. Now, I think about helical and bevel gears, 24 hours, seven days a week."

Recently Smith was also fitted with Orwell's Executive Chain, a power transmission component that binds the eager worker with an overzealous young executive from the boardroom. This takes micromanagement to the next level, giving leadership an opportunity to find the true value in Smith's work *and* home life. This chain keeps Smith linked 24/7 to a productivity professional that will offer direct criticism on a variety of topics. "It's nice to have someone always available to ask questions," Smith says. "It's a little awkward when I want to take a shower…"

"The gear company is able to break down exactly what Smith brings to the table with the latest generation of these Orwell products," O'Brien says. "If you want a picture of the future, imagine a boot stamping on a human face–forever!" To order Orwell Inc.'s BBMS or Executive Chain, contact your local Orwell representative via telescreen today while supplies last.

(Ed's Note: The Addendum team thought it fitting to celebrate George Orwell's 1984 with the 30th Anniversary of Gear Technology. We do not condone the extreme tactics discussed in this fictional press release unless instructed by the proper authorities.)