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APRIL 2015

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DRIVES

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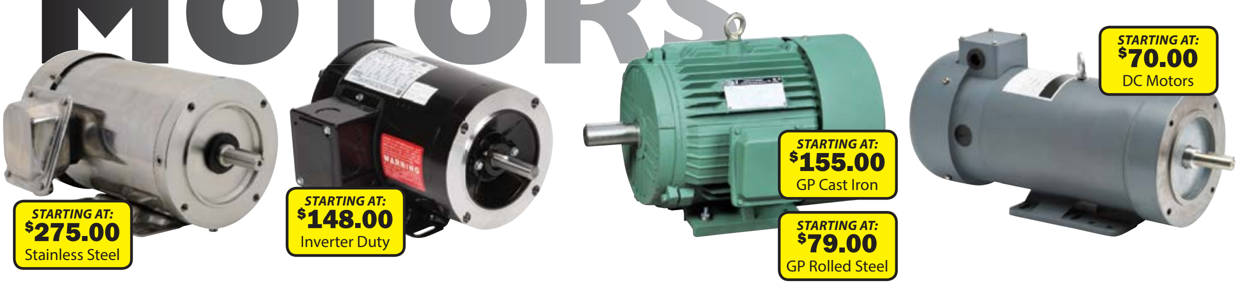
MOTORS



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Marathon Electric inverter-duty motors have been carefully selected to be performance matched with our DURApulse and GS series AC drives. They are available in 1/4 to 100 hp with 1200 and 1800 RPM base speeds. Dual 230/460V and 575 VAC models are available and factory-mounted encoders are offered on select models. NEMA Premium Efficiency XRI series from 1 to 10 hp are compliant with Energy Independence and Security Act of 2007.

Ironhorse AC motors are available in rolled steel (1/3 to 2 hp), stainless steel (1/3 to 2 hp) and cast iron (1 to 300 hp) 1800 RPM models. 1200 and 3600 RPM units are also available in the most popular horsepower ratings. Our premium efficiency models meet current NEMA standards from 1 to 150 hp, and start at $155.

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FEATURE ARTICLES

TECHNICAL ARTICLES

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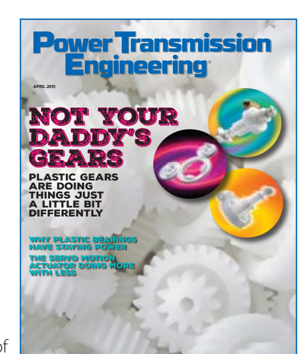
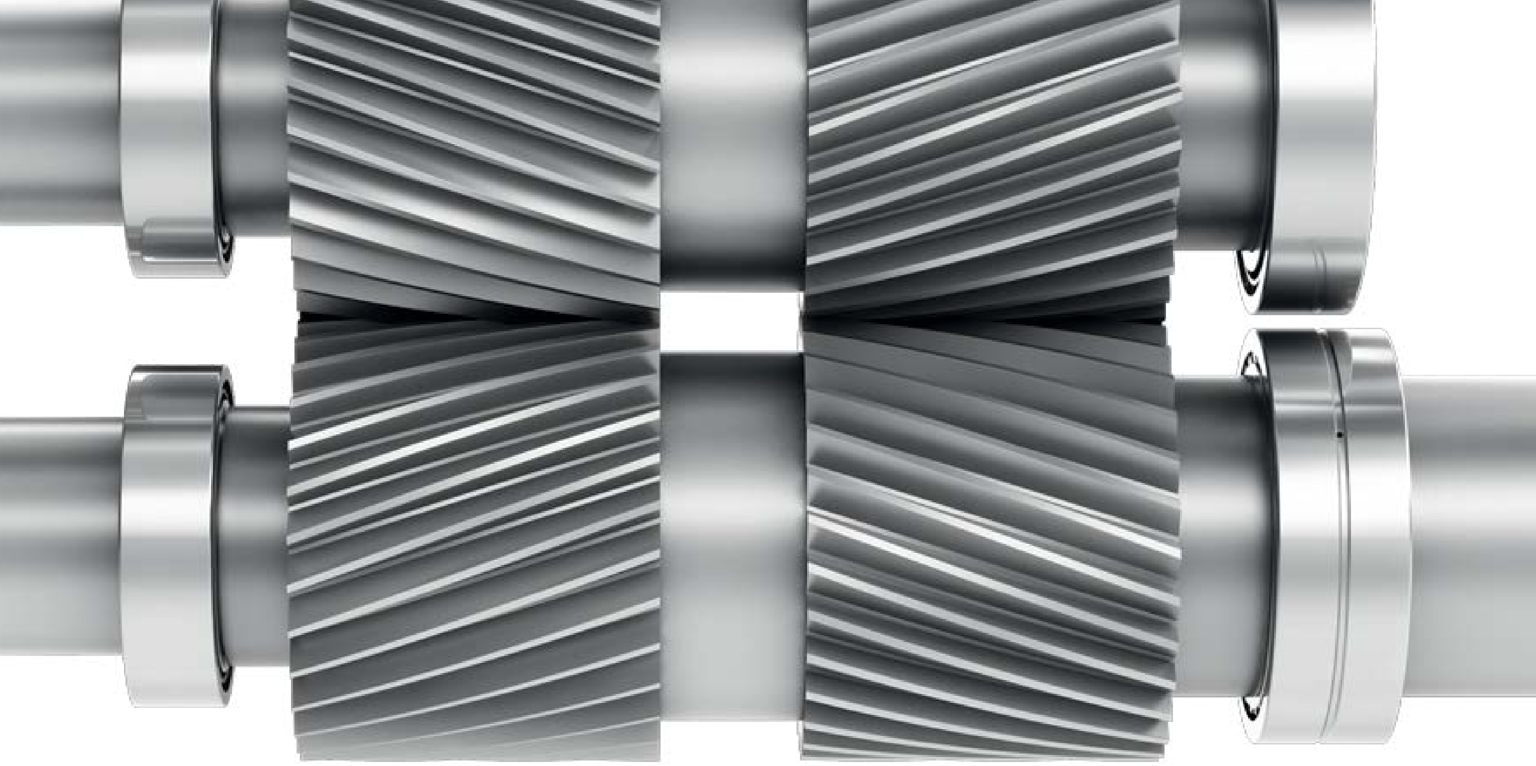


Inset cover photos courtesy of Winzeler Gear, Inc.



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PTE Videos

Missed Hannover Fair? No Worries! We've Got You Covered

Stop by powertransmission.com *to see videos of the exhibits you missed this year. Learn about the latest technologies from Siemens, Contitech and others at the home page.*

PTE Videos

How It's Made – Ball Bearings

In this video from NTN Bearing Corporation of America, we see how ball bearings are manufactured. The footage was originally shot at NTN's Mississauga, Ontario ball bearing factory for the Discovery Channel's "How It's Made" show. Watch the video online at www.powertransmission.com

More Suppliers, More Choices

We're constantly improving the online **Buyers Guide** at *powertransmission.com* by adding new suppliers of gears, bearings, belts, clutches, couplings, motors and other mechanical power transmission components. Stop by the site to see who's been added recently:

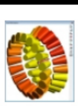
MIKI PULLEY

TECHTOP

Latest News and Products

Powertransmission.com is your resource for the latest information on mechanical power transmission products. Learn about the latest technology as soon as it's announced.

Products (see more products)



Mesys AG Will Present New Shaft and Bearing Analysis Software at Hannover Messe (April/13/2015)
Mesys AG will present its new version of shaft and bearing analysis software in Hall 25, Stand A21 at Hannover Messe.The bearing analysis...

Klüber Lubrication Introduces Synthetic Grease for Extreme Climatic Conditions (April/10/2015)
Klüber Lubrication recently introduced Klübersynth EM 94-102, a fully synthetic lubricating grease incorporating a calcium comp...

HMM Debuts Cable-less Robot Gantry Technology at Automate (April/9/2015)
For the first time, global automated systems provider Harry Major Machine (HMM) displayed a new cable-less robot gantry technology for th...

Bonfiglioli Announces New TQK Series Right-Angle Precision Gearbox (April/6/2015)
Bonfiglioli recently announced the new TQK series -- Right-Angle Precision Gearbox version. This precision gearbox is complementary to th...

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Renee Stott, Guest Editor

Girls Love Science

Hi. My name is Renee. I'm 12.

You might remember me from about a year ago, when my dad wrote about me and the home-made motor project we did together ("Make the Connection," February 2014).

That was a really fun project, and I got to learn a little bit about how electric motors work. After that article ran, we made a way better motor and I took it to school to show my class.

I like science, especially when I get to make things and see how they work. My dad says that sounds like something an engineer would say, and I guess that's true. In March I got to meet a bunch of engineers when I went to the "Introduce a Girl to Engineering" event held at the Siemens factory in West Chicago, Illinois.

The engineers there work with motors that are a little bit fancier than the one my dad and I made. At Siemens, they make motor control cabinets, which are basically big closets full of wires and electronics. They're used to turn motors on an off, and to make sure the motors and equipment don't get damaged, which is really important in a big factory or building where there are a lot of motors running.

I got to see how metal is bent to form the cabinets, how they use lasers to cut shapes in the metal, how the cabinets are painted and how everything is put together and tested. We talked to a manufacturing engineer at the factory's "copper fabrication cell." That's where they punch holes and bend pieces of copper for their cabinets. He said his job was to figure out how to make things faster, better or cheaper. We also met an industrial engineer. She designed the system Siemens uses to automatically load sheet metal into a machine that punches holes in the metal.

One of the coolest things was when we got to stand on the giant scale they use to weigh the cabinets before they go on trucks to be shipped out. In case you're interested, our group weighed 1,794 lbs.

During the event, we also got to work on a few science projects. Working in teams, we designed and built a windmill out of notecards, paper clips and a cork. Then we tested them to see whose generated the most electricity. Unfortunately, my team didn't win, but it was fun anyway. My favorite project was building a structure out of marshmallows and dry spaghetti. My team *would* have won, but our tower developed a leaning problem right at the end.

We also got to hear a number of people speak about engineering and what they do every day. Did you know that only about 12% of engineers are women? I think that's too bad, because most of the smart people I know are girls.

Anyway, I'd like to thank Siemens for hosting the event. It was a lot of fun, and I learned a lot, too. If you have a chance, you should all find a way to share *your* love of science and engineering with kids my age (even if they're boys). It was a great experience that I would definitely do again.

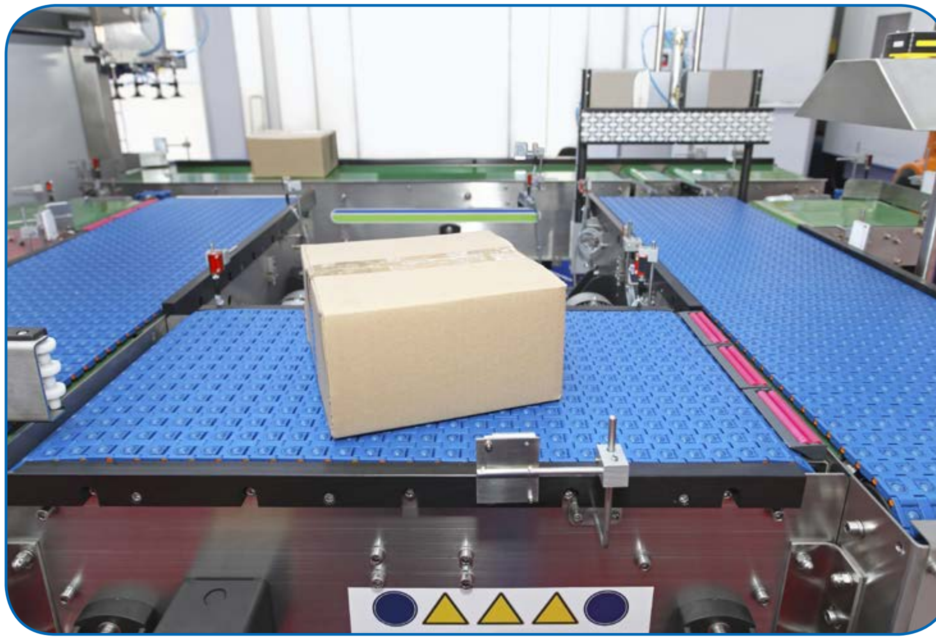
Besides, visiting the factory and talking to all of those engineers has given me some new ideas about that motor we made last year. Do you think my dad will let me add a control cabinet to our design?

Renee

More than 130 5th-12th grade girls took part in the 11th annual "Introduce a Girl to Engineering" day at the Siemens factory in West Chicago, where they manufacture motor control centers, switchboards and enclosed controls for companies in the food and beverage, aerospace, automotive, metals and paper industries.

Regal

INTRODUCES TURNKEY RETROFITS OF LIVE-ROLLER CONVEYOR



Regal Power Transmission Solutions recently introduced a turnkey retrofit program that transforms live-roller conveyor for sustainable, high-uptime performance.

Combining its System Plast modular plastic belt and chain, Hub City or Grove Gear drives and long-lived Sealmaster material handling bearings, Regal provides single-source or à la carte capabilities for on-site evaluation, consulting, layout design, validation and installation. The variety of conveying surfaces available from System Plast creates new options for every material handling need from single bottles to cases, people and even cars.

"Live-roller conveyor is old technology that has many weaknesses compared to recent advances in plastic chain and belt," said Mike Suter, vice president for Regal Power Transmission Solutions. "Live-roller is noisy with pinch points from day one, and gradually turns into a downtime and maintenance problem with dead zones, belt wear, tracking and replacement, and complex snub roller center drives. Converting a system makes economical use of existing conveyor frame, while the new solution reduces product damage, part count, maintenance, downtime, noise, and energy consumption."

System Plast engineers a variety of conveyor belt and chain for low-maintenance, high-efficiency performance, utilizing proprietary polymer formulations to suit application requirements. Showcasing its flexibility at Promat, Regal demonstrated its directionally unlimited roller top belt that can align products to any angle, divert them, sort to multiple lanes, rotate, combine, or gap them—all done softly without contact.

System Plast 2253RT roller top belt simplifies conveyor design and installation with its independently controlled moving surface that handles large or small, flat-bottom products, moving them on half-inch (12.5 mm) balls spaced on one inch (25.4 mm) centers. It supports loads up to 617 lb/ft (9,000 Nm/m) or 1.1 lb/ball (0.5 kg/ball). The belt's non-contact product manipulation eliminates impact and abrasion damage from pushers, as well as the need for mechanical adjustment or changeovers for conventional diverters or guides.

System Plast heavy duty and extra heavy duty plastic belts provide a non-skid, energy efficient conveyor surface for moving walkways, automotive assembly lines, car washes, and general material handling of heavy loads, such as barrels, drums, kegs and pallets. The belts handle loads up to 5,400 lb/ft and 7,800 lb/ft (80 kN/m and 115 kN/m), respectively. Both styles are ideal for moving assembly line floors where the belt's non-skid, solid surface protects against slips and prevents tools and small parts from being trapped. The belts' design allows pusher bars to be attached, removed or replaced by the user at any time without disassembly of the belt, using molded-in drilling locators. Both belts utilize a patent-pending retention clip for the hinge pin. It can be installed or removed from either side of the belt with simple tools, as well as re-used. The molded-in non-skid surface of hexagonal dimples is ideal for people movers, offering a high coefficient of slip resistance under oily or dry conditions.

Hub City High Efficiency Right Angle (HERA) reducers are the 21st century replacement of industrial worm-gear drives, providing 90% efficiency in all ratios for up to 40% reduction in energy requirements. Torque-dense with double the capacity of worm drives, HERA drives reduce motor size requirements, as well as the physical size of the drive package. HERA reducers save up to $550 per year with each increment of motor horsepower for rapid payback, and they are warranted for three years. Wide torque capacity and modular shaft, base and flange components allow just four sizes of HERA reducers to interchange with worm reducers from 1.75" (44.45 mm) to 6" (152.4 mm) center distance, reducing inventory requirements. Maximum torque ratings for the four sizes are 1,100, 2,600, 4,500 and 8,500 in-lbs (124 Nm to 960 Nm). The smallest and largest HERA reducers are available in stainless steel, and packaged drives with Marathon or LEESON motors are available as well.

Grove Gear Bravo gear reducers feature a lightweight, single-piece aluminum housing that is vent-free and protected with high-temperature nitrile output seals. Bravo reducers weigh as much as two thirds less and are one-third smaller than cast-iron designs. Interchangeable with drives commonly found in material handling and packaging systems, Bravo reducers are designed for frequent reversing and start-stop cycles with oversized high-speed bearings and a bronze alloy worm gear centrifugally cast onto an iron hub for maximum strength and lubricity. IEC

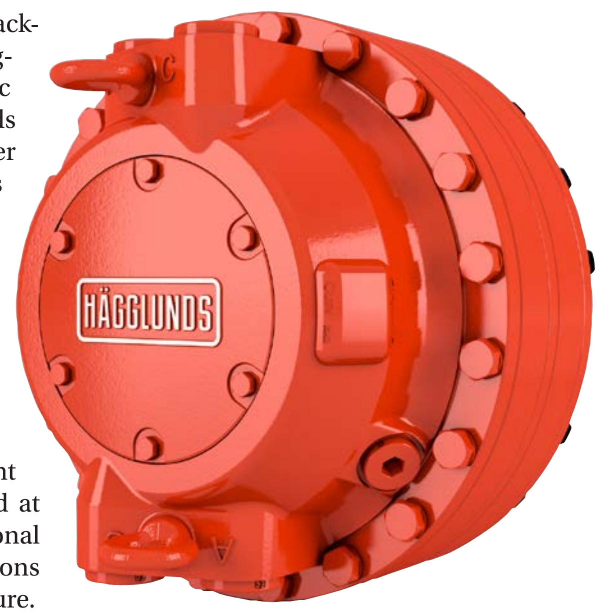
motor inputs and metric outputs make Bravo drives an excellent replacement for popular worldwide material handling reducers. Five sizes are available in ratings to 7.5 hp (5.6 kW) and ratios from 7:1 to 70:1, with center distances from 1.18" (29.9 mm) through 3.35" (85 mm). They are also available as integrated drives with Marathon or LEESON.

Sealmaster material handling bearings are designed with features to provide years of maintenance-free performance. Available in pillow block, tapped base pillow block, flanged and wide-slot take-up housings, all of them start with a solid cast iron base or housing for improved stability and resistance to shock and vibration. A standard contact nitrile rubber seal helps keep contamination out and facilitates grease purge, and an auxiliary flinger helps protect the seal from direct contamination. Both SKWEZLOC and setscrew locking styles are available, and optional snap-on end caps provide an extra level of personal protection from rotating shafts. Sealmaster's zone hardening of the inner race improves lock reliability by reducing distortion at setscrew locations.

For more information:
Regal Beloit Corporation
Phone: (608) 364-8800
www.regalpts.com

Bosch Rexroth

INTRODUCES NEW SERIES OF HÄGGLUNDS HYDRAULIC MOTORS



Bosch Rexroth recently repackaged its compact series of Hägglunds industrial hydraulic motors. The new Hägglunds CA 10 to 40 motors are smaller and lighter, yet nothing has changed in their output.

The new motors will extend the Hägglunds compact series downward to 10 Nm/bar. Initially, the focus for the Hägglunds CA 10 to 40 series will be injection molding machines in the plastics industry. These machines represent a large portion of the demand at 50 Nm/bar and below. Additional industries and further applications are set to follow in the near future.

To achieve the power density of the CA 10 to 40 motors, Bosch Rexroth has reduced the motor diameter while refining the internal construction. These changes have no effect on reliability or output. Full speed and full torque are available simultaneously, and both can be sustained for an indefinite time.

"In designing the series, we've used principles similar to those in our proven Hägglunds CBM motors," said Bengt Liljedahl, technical product manager. "The new CA 10 to 40 motors provide the same power as before but with far less weight — up to three times less than a competitor solution."

Further efficiency comes from the flexibility in the series, which includes four basic models and a wide variety of displacements. Fourteen configurations in small torque steps allow tight dimensioning, as well as full optimization in relation to the hydraulic pumps. The configurations are achieved using just two mechanical interfaces.

For more information:
Bosch Rexroth Corporation
Phone: (610) 694-8352
www.boschrexroth-us.com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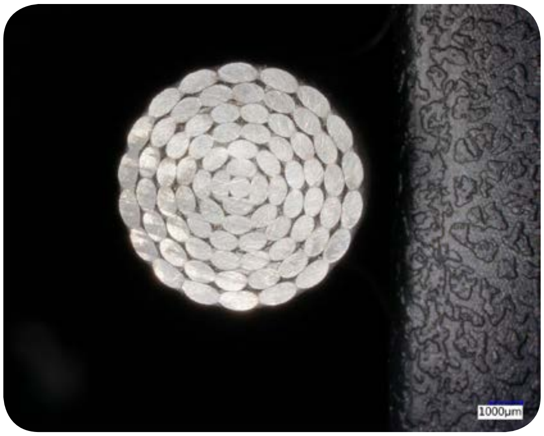
Asahi Intecc

OFFERING DRIVE CABLES IN EXPANDED SIZE RANGE



Asahi Intecc now offers high speed rotation stainless steel miniature drive cables for medical device and industrial applications requiring diameters less than ⅓".

The Asahi drive cable is a multi-layer, multi-filar, stranded flexible shaft with high bi-directional torque transmission properties with minimal backlash or lag. The multi-stranded shaft may be cut to short lengths for a smooth finish 1" to 2 m lengths depending on the application, and in-house assembly and coating services are offered.

The layers and size of wires, along with the way in which the wires are processed, may be customized to the degree of flexibility, torque force, rotation speed required by the application. Secondary operations for the drive cable include laser welding, soldering, crimping hypotubes, cutters, micromachined parts, housings; extrusion, heatshrink, dip, spray PTFE, ETFE, PU, PE, FEP with lubricity and bonding strength; centerless grind the distal end for multi-segmented flexibility; rotary swaging for torsional rigidity, and augering the drive cable with a stainless wire for additional cutting and aspirating capability.

A cannulated 3-layer version is also offered for applications in which material in which aspirated material is extracted through the lumen.

For more information:
Asahi Intecc
Phone: (662) 501-1302
www.asahi-intecc.com

Miki Pulley

INTRODUCES ITS JAW TYPE, HIGH RPM SHAFT COUPLING

Miki Pulley recently introduced its Jaw Type, High RPM coupling for direct sale to OEM's in North America. This ALS-SGN model handles speeds to 22,000 RPM.

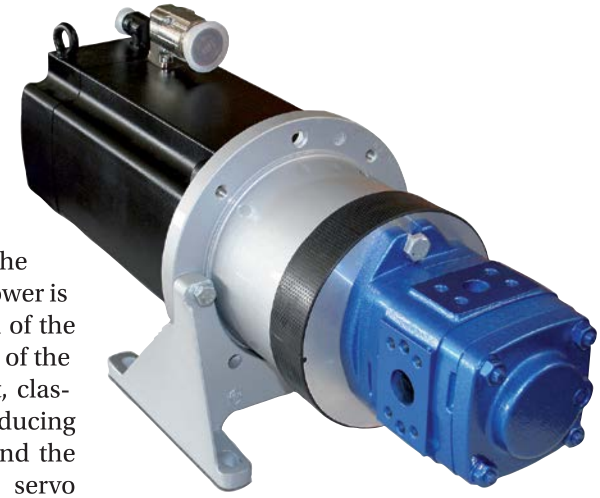
Designed for use in the main spindle shaft on CNC machining centers, the Miki Pulley ALS-SGN coupling features taper-lock hubs. This hub design ensures concentricity and rotational balance during high-speed operation.

The ALS-SGN coupling's high-torque elastomer center element absorbs resonance and mitigates conductive heat while providing over 22,000 Nm/rad torsional stiffness. The coupling's aerodynamic profile decreases noise while optimizing performance at high speeds.

For more information:
Miki Pulley
Phone: (800) 533-1731
www.mikipulley-us.com

Voith

SHOWCASES SERVO PUMP AT HANNOVER MESSE



Hydraulic applications with high demands on control technology are subject to fluctuating stresses during operation. To ensure cost-effective operation under these conditions, the Voith servo pump adjusts to the power needed over varying volume flows and motor speeds. This provides advantages over conventional systems, particularly in the part load range, with a high volume flow that is needed only within an operating cycle. Optimized operation reduces energy consumption by up to 70% and the total cost of ownership (TCO) of the entire hydraulic system by up to 3%. In this way, it is usually amortized within one to two years.

Featuring the motto "Saving Energy while Increasing Productivity," Voith presented the servo pump to visitors at the Hannover Messe trade fair from the April 13-17 in Hall 23, Booth 41.

The pump system consists of three main components: the servo inverter, the synchronous servo motor and the Voith internal gear pump. The servo inverter analyzes and processes the set-point and the actual values of the pressure and speed. It controls the servo motor, which supplies the required torque in the shortest amount of time. This power is fed to the process as a function of the pressure or volume flow by way of the internal gear pump. As a result, classic valves are unnecessary, reducing the complexity of the system and the cooling capacity needed. The servo pump facilitates dynamic control that is further supported by the low mass moment of inertia of the internal gear pump. This shortens the cycle time of the actuators by up to 50% and increases the productivity of the machine.

In addition to optimized energy consumption, the Voith variable-speed solution also reduces the noise emissions of the system by up to 20 dB(A). This reduces the cost and effort for noise abatement measures — in many cases, workplace guidelines are met without any additional measures.

Previously integrated control parameters make the Voith servo pump ready for operation immediately. The pressures and volume flows are individually adjusted to the specific cycle data and harmonized with existing control concepts and systems.

The servo pump can be used in a variety of ways. Typical areas of application are plastics machines, die casting machines, machine tools and presses. As the central component of machines and systems, it continuously measures the current operating parameters and thus provides a node for integration within the framework of Industry 4.0.

For more information:
Voith Turbo GmbH & Co. KG
Phone: +49 7321 37 8303
www.voith.com

Warner Linear Actuators

RECEIVE IP69K RATING

Warner Linear recently announced that select linear actuators and controls received the IP69K rating, the highest rating available for protection from high pressure washdowns, high temperature water and dust.

Specific Warner Linear products with the IP69K rating include B-Track K2 DC actuators and actuator-mounted controls, S-Track actuators, I-Track actuators and M-Track actuators. The new rating makes these models ideal for use in food processing, pharmaceutical, medical equipment, marine, construction equipment, agriculture and outdoor applications.

The IP69K rating insures ingress protection at 176˚F/80˚C (1450psi/100bars), ingress protection of solid foreign objects (including dust) and testing according to DIN 40505 Part 9 Standard.

For more information:
Warner Linear
Phone: (815) 847-7532
www.warnerlinear.com

Bell-Everman

INTRODUCES NEW SERVOSPLINE WITH BACKLASH-FREE MOTION

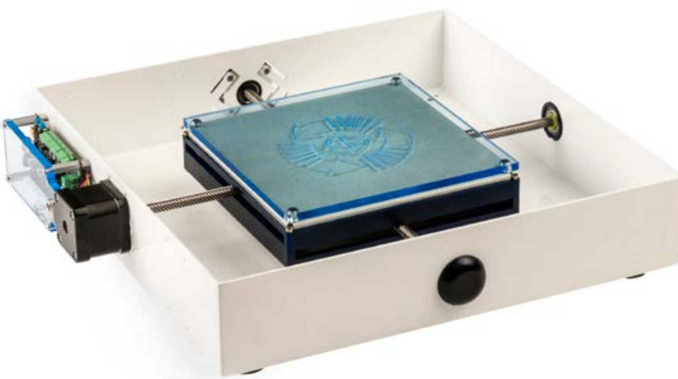
Bell-Everman's new ServoSpline positioning stage offers a drive mechanism that provides backlash-free motion and simplified control.

ServoSpline is an extension of a classic rack-and-pinion design that provides simple motion commands and positions the stage in the X- and Y-axis by independently driving the two splines. It provides engineers with a stage that is easy to deploy. Technical advantages include no backlash, long wearing and easy implementation.

This positioning stage can serve in a variety of machines requiring easily-implemented motion stages. Applications include 3-D printers, automated assembly, dispensing, engraving, additive manufacturing, biomedical titer plate indexing and pipetting for diagnostic equipment.

For more information:

Bell-Everman, Inc
Phone: (805) 685-1029
www.bell-everman.com



Ruland

INTRODUCES CLAMP STYLE SHAFT COLLARS FOR MEDICAL EQUIPMENT

Ruland clamp-style shaft collars are manufactured from select materials with fine finishes, high-holding power and a precise face to bore relationship making them ideal for the demands of medical equipment. They are used for guiding, spacing, stopping and component alignment.

Medical equipment manufacturers benefit from the tightly controlled face to bore perpendicularity of Ruland shaft collars (TIR of ≤ .002" or .05 mm) which is critical when the collar is used as a load-bearing face or for aligning components such as bearings or gears. Clamp style shaft collars distribute compressive forces evenly around the shaft for a tight fit and improved holding power when compared to set screw style collars. Two-piece style collars offer increased holding power, in-place installation and disassembly, and are mated throughout the manufacturing process for proper fit and alignment.

Ruland shaft collars utilize fastening hardware that tests beyond industry standards for maximum torque capabilities and holding power. Shaft collars made from 303 or 316 stainless steel have hardware of like material for consistent corrosion resistance and to meet regulatory standards. All stainless steel hardware undergoes an in-house surface treatment process to prevent galling. Ruland uses a proprietary black oxide process on steel shaft collars to provide corrosion protection and ensure a smooth screw installation to increase the transfer of clamping forces from the screw to the collar/shaft interface while preventing stick-slip.

Clamp-style shaft collars from Ruland are available in 1215 lead-free steel with a black oxide or zinc plated finish, 303 and 316 stainless steel, 2024 aluminum with a bright or anodized finish, grade 5 titanium, and engineered plastic. They are machined to a fine burr free finish that complements or enhances the appearance of medical devices. Ruland offers clamp shaft collars with smooth or threaded bores ranging in size from ⅛" (3 mm) to 6" (150 mm).

Shaft collars are part of Ruland's complete product line which includes rigid couplings with precision honed bores and five styles of motion control couplings: beam, bellows, disc and zero-backlash jaw. All Ruland products are manufactured in its Marlborough, Massachusetts factory and are RoHS2 and REACH compliant.

For more information:
Ruland Manufacturing Co., Inc.
Phone: (508) 485-1000
www.ruland.com

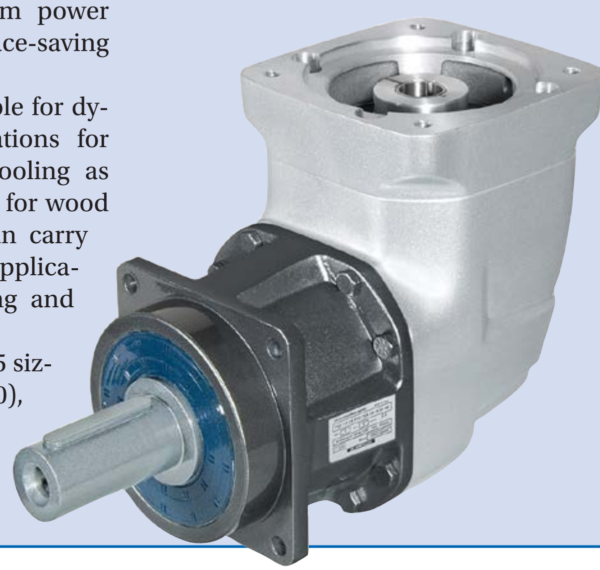
Bonfiglioli

ANNOUNCES NEW TQK SERIES RIGHT-ANGLE PRECISION GEARBOX

Bonfiglioli recently announced the new TQK series, Right-Angle Precision Gearbox version. This precision gearbox is complementary to the TQ series and offers maximum power density and features a space-saving installation solution.

TQK is particularly suitable for dynamic positioning applications for packaging and machine tooling as well for flat-bed machinery for wood working. In addition, it can carry out continuous running applications necessary for printing and paper converting.

TQK features a design in 5 sizes (060, 070, 090, 130 and 160), ratios from 6 up to 200 due to its design based on one and two reduction sizes, all this with a nominal torque from 30-800 Nm. Furthermore, its housing design and monoblock planetary carrier provide a high



Harry Major Machine

DEBUTS CABLE-LESS ROBOT GANTRY TECHNOLOGY AT AUTOMATE

For the first time, global automated systems provider Harry Major Machine (HMM) displayed a new cable-less robot gantry technology for the manufacturing industry during Automate 2015 in Chicago, IL.The technology is the first of its kind available in North America.

HMM is the provider in North America for the cable-less gantry, which is made by MaxRoTec Co Ltd., a manufacturer if gantry robots and a HMM partner since 2014.

"The benefits that the cable-less gantry system provides to our customers are numerous," said H. Curtis Major, president of HMM. "The system is not only compatible with a variety of robot technologies, it also has a compact footprint and helps to lower maintenance and infrastructure expenses. The system is a speedy, reliable workhorse that maximizes productivity while reducing downtime."

With the cable-less gantry, power is supplied to the system via an insulated trolley and collector arm and its communication is powered by an optical Ethernet transmitter using infrared light. The system can work at faster speeds than traditional cabled gantries and has the ability to handle up to five carriages on a single track.

The line configuration and programming are easy to change, since no cable-laying is required, even if the beam length is changed due to line modification. Since no cable track and guarding is needed and multi-carriages can be handled on a single track, less space is required for the system, which allows for a more compact line configuration.

The system is compatible with a variety of robot technologies such as ABB, Fanuc, Kawasaki, Kuka, Nachi and Yaskawa Motoman and can be utilized by manufacturers in various industry sectors in their operations.

HMM is also highlighting several of its other technologies at Automate 2015, including its solutions in the areas of automated parts handling systems, robot integration, industrial parts washers and assembly machines.

For more information:
Harry Major Machine
Phone: (586) 783-7030
www.harrymajormachine.com

torsional stiffness that can meet any application requirement from a faster dynamic to a higher number of stops and reverse cycles. The optimized planetary full-needle roller bearings allow for a higher torque output due to the maximizing of the contact points and increased stiffness, reduced wear and backlash.

The TQK Series is equipped with higher-rating bearings that can handle higher radial and axial loads, and the reinforced bearing options extend higher performance. Its design is also suited for S1 & S5 operation with a very reliable catalog rating and selection procedure.

For more information:
Bonfiglioli USA
Phone: (859) 334-3333
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Klüber Lubrication

INTRODUCES SYNTHETIC GREASE FOR EXTREME CLIMATIC CONDITIONS

Klüber Lubrication recently introduced Klübersynth EM 94-102, a fully synthetic lubricating grease incorporating a calcium complex soap thickener. The thickener enables formation of a resilient lubricating film that provides high resistance to mechanical-dynamic loads while enabling wear protection.

Klübersynth EM 94-102 can be used in a variety of applications under different climatic conditions due to its wide service temperature range. As a result, various friction points can be supplied with one single lubricant, avoiding confusion due to multiple lubricant selection and use.

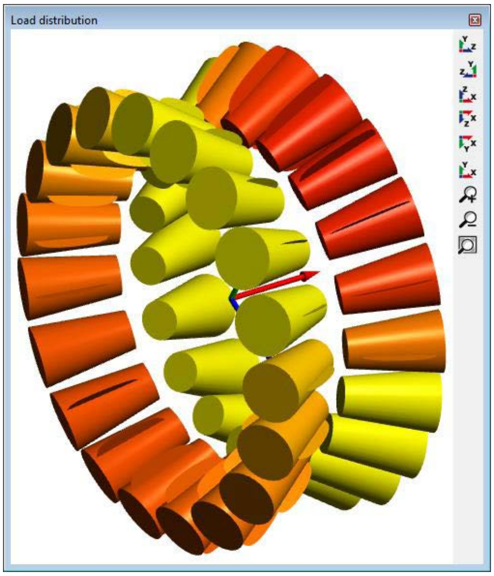
The resistance to water and corrosion protection properties of Klübersynth EM 94-102 make it suitable for use in wet and humid areas. Examples include rolling and plain bearings for winter sport applications, like ski lift equipment and snow grooming vehicles, or industrial offshore and marine applications.

Klübersynth EM 94-102 is suitable for an array of applications because of its wear protection, adhesion characteristics, load capacity and low-temperature behavior.

For more information:
Klüber Lubrication North America L.P.
Phone: (603) 647-4104
www.klubersolutions.com

Mesys AG

WILL PRESENT NEW SHAFT AND BEARING ANALYSIS SOFTWARE AT HANNOVER MESSE



Mesys AG will present its new version of shaft and bearing analysis software in Hall 25, Stand A21 at Hannover Messe.

The bearing analysis software allows the calculation of bearing load distribution and bearing life according ISO/TS 16281 and is integrated in a shaft system calculation with additional possibilities like strength calculation for shafts, modal analysis and interfaces to gear calculation programs.

MesysAG can now do a variation of clearance within bearing tolerances, consider misalignment of one bearing up to 0.1mm and use a tolerance for loading. For all cases a normal distribution is used with three times standard deviation fitting into the tolerance field.

The number of calculations can be specified by the user. A first insight is already possible with a smaller number of calculations, but for nice-looking curves several thousand calculations are needed. The probability for the results of basic reference life were calculated with 20,000 load cases, which took about 12 minutes on a standard desktop computer. For the initial calculation the maxima of the curves are close to each other, but the variance for the roller bearing B2 is much larger. The reason is the misalignment of the shaft, which has a larger influence on the roller bearing. The report will also show minimum, maximum, mean value and standard deviation for each selected result.

The default version of parameter variation is helpful to visualize the influence of one parameter on one or multiple results. The statistical version can be used if the influence of multiple uncertainties should be evaluated. Instead of manually checking multiple parameters an automatic calculation of the whole ranges can be done. In order to understand the reasons behind the results it will still be necessary to carry out additional evaluations, but the software shows the ranges of results that have to be expected.

In addition to the shaft and rolling bearing calculation software a new calculation of load distribution for ball screws is available now. Life is calculated from load distribution analogously to ISO/TS 16281. This allows the consideration of radial and moment loads in addition to axial loads. As in the other programs, an automatic parameter variation is available as well.

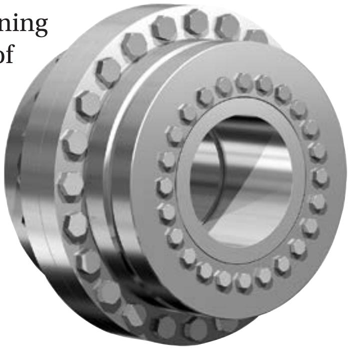
For more information:
Mesys AG
Phone: +41 44 4556800

Ringfeder

OFFERS NEW SERIES OF FLANGE COUPLINGS FOR HEAVY DUTY APPLICATIONS

Ringfeder Power Transmission is now offering its new RfN 5571 series of flange couplings for heavy duty applications. These couplings offer easier installation and have higher torque capacity than standard press fits.

Equipped with Ringfeder shrink discs, the RfN 5571 flange couplings can be integrated into machines without any heating or cooling, making them easily installable for engineers. They are slip fit and come with either hexagon-head or hexagon socket-head cap screws. Other advantages include no wear parts, backlash-free shaft-hub connections, high true running accuracy and elimination of additional components such as keyways or shims.



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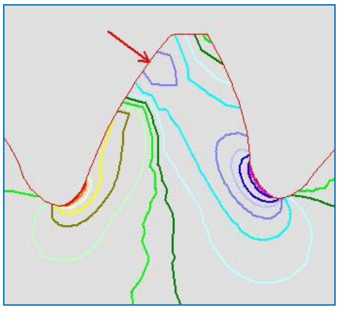
AKGears

UNVEILS LATEST TOOTH ROOT FILLET OPTIMIZATION SOFTWARE

AKGears recently introduced the only commercially available tooth root fillet optimization software that defines the tooth root fillet profile to provide minimum bending stress concentration.

The software allows increasing gear load capacity by 15-25% and is applicable to involute and non-involute gears with external and internal, symmetric and asymmetric teeth. This optimization technique is presently employed in various gear drives for aerospace (Boeing), automotive (Delphi), medical devices (Abbott Labs), consumer products (Panasonic), and many other industries.

For a free, downloadable demonstration version of this program, go to www.akgears.com/software.htm.



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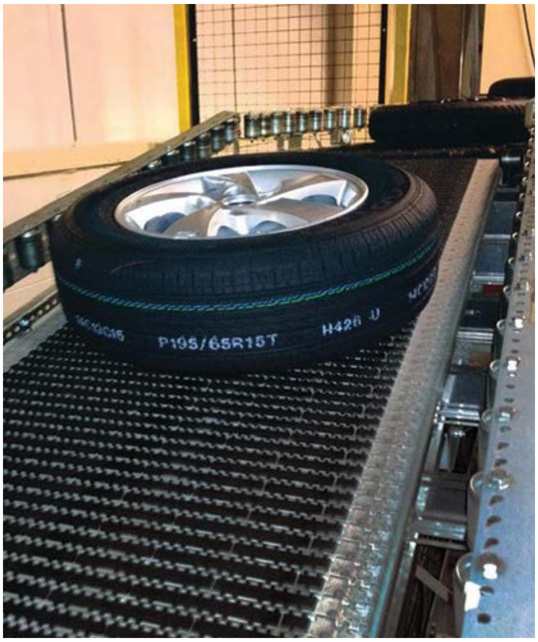

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There's a Great Future in Plastics

It would appear that Mr. McGuire was right

Erik Schmidt, Assistant Editor

The first thing you see when you walk into Winzeler Gear is a pretty face.

No, scratch that—the face is beautiful. Streamlined with chiseled cheek bones, delicate yet bold and strong, a half dozen photos of the feminine face line the building's entranceway, enclosed tastefully in simple black picture frames, beckoning you to come further in with its pleasant warmth, repelling you to turn and leave with its foreign presence in such a location.

This can't be the right place, can it?

Then you look closer.

Dangling loosely on dainty wrists, hanging down from an exquisite pair of ears, sparkling loud and clear even in the black and white photos, is jewelry made of tiny plastic gears. You look to the corner of the foyer and see a fashionable white dress resting on a wardrobe—and that too is made of gears.

OK, this *is* the place.

So you step inside.

The unmistakable bangs and clangs of heavy machinery roar dully in the distance. More pictures hang on a wall to the left, the first of a gear-shaped vodka bottle—"Absolut Winzeler" scrawled across the bottom in white typeface—the second of a square-toothed rock formation that juts sharply into a blue sky in the background.

Before there's time to question the building's interior decorator, an elderly man appears from around the corner. A pair of chic glasses rest over kind eyes; his head is shaven bald like a Shaolin monk.

His presence here seems about as fitting as everything else.

"We don't look like a gear factory," says the man, John Winzeler. "And that's the point."

Doing it Differently

Winzeler, president of Winzeler Gear, has been in the industry a long while, and the whole time he's been trying to cultivate a different (*different*—get used to that word) kind of culture at his factory in Hardwood Heights, IL.

"Think differently," the company's website says. And Winzeler does:

Gear jewelry, gear dresses, gear decorative headpieces, gear paintings, gear sculptures… the entire Winzeler factory is a shrine to indomitable creative thought. Perhaps in a different place, with a different man, the motif may come across as forced, but at Winzeler Gear it is integrated so seamlessly

All photos courtesy of Kleiss Gears, Inc.

and with such genuineness that is feels completely organic.

And, when you really think about it, the connection makes perfect sense. Plastic gears *are* different.

Metal gears, naturally, are much more commonly applied within the industry and are better known. But over the last decade or so, plastic gears — lighter, quieter alternatives to their metallic cousins — have begun to gain some serious traction as a viable, and oftentimes better, solution.

"Thirty years ago when I started doing this, plastic gears were 'cute,'" says Rod Kleiss, president of Kleiss Gears, Inc. (Granstburg, WI). "They were used in a few little things, but anything that was quite serious you would expect failure if the material was plastic. So as I got more into it I realized there were a lot of convenient but poor choices being made on the design and everything about the gear.

"We started working on gear design and started making the normal starting material work much, much better in gearing. The more we got into it, the more we got challenged with just far we could take plastic gears."

Most plastic gear manufacturers use, or have used, a material known as polyoxymethylene (POM), a thermoplastic that offers high stiffness, low friction and excellent dimensional stability. Kleiss used a variation of POM, also commonly referred to as acetal, for most of its applications up until about 15 years ago. Winzeler, meanwhile, has used a popular POM produced by DuPont called Delrin for over 50 years. Celanese (Irving, TX), a specialty materials company that supplies plastic to gear manufacturers, makes its own version known as Hostaform, a product Celanese Senior Design Engineer David Sheridan called the company's "workhorse." POM/acetal-based gears are used in the majority of interior applications within the automotive industry.

"Acetal is a proven gear material," Sheridan said. "People know how to design with it, they're comfortable designing with it and in typical automotive temperatres up to 85°C, there's no reason acetal can't keep working. It works beautifully now."

Essentially any interior application in an automobile, from power windows to car seats, can be driven by an acetal-based plastic gear. Over the years, company's like Celanese have been able to improve and innovate on these POM materials to make them more durable and versatile.

"We have continued to make improvements to our acetal product line," Sheridan said. "We have a new line of glass-reinforced acetal that is a game changer. It's not just a little bit of an improvement, it's a significant improvement. It's not just additives, there's actually a difference in the base polymer that shows a drastic improvement."

Because of these recent innovations, acetal-based plastic gears are now applied in places that were once impossible

"We have begun making plastic gears for much more rugged applications than were previously possible," Kleiss says. "Right now, we're the primary producer of gears that go into corn seeders. These things get bounced over fields and they have 60 to 120 of [plastic gears] on a single implement. If they had metal gears on these things they would weigh a ton and it would be useless.

"Plastic gears are providing an answer that wasn't available with the metal technology."

So while plastics gears are certainly different (i.e. better) than they were a decade ago, there are, of course, still limitations — the main barrier being heat. Winzeler summed it up with proper panache when he said that the "nemesis of plastic gears is high temperatures."

"Delrin is our favorite material, and it does everything we want in a material up to about 85°C," Winzeler says. "The one thing that is very important to understand about many of these engineering materials is that having knowledge of how these materials process has a dramatic impact on the ultimate strength and durability of that part, and also its long term stability. There's a lot of science that goes into it once we have a part designed so that we can achieve consistency over very large volumes.

"In terms of gear applications for the amount of work we can do within a certain amount of space, Delrin does everything we want it to do, but it does not do well once it gets over 85°C, meaning it does not work in the engine compartment and it wouldn't work in the transmission. Once we get into those applications we get into more exotic materials that are more difficult to process.

"The biggest problem with plastics — and it hasn't changed — is that at some point they melt."

Taking the Heat

If the big question in plastic gearing is "how far can they go?" (and it is), then what exactly is the answer?

Most seem to think that, realistically, high-speed, high-torque applications in transmissions are both the present and future of plastic gears due to their ability to eliminate noise at the front end of an automobile.

"Under the hood gears have been looked at for a number of years, and I think manufacturers are satisfied that there are plastic solutions," Sheridan said. "The only gears in automotive that have yet to be tackled by plastic gears are drivetrain gears. Whether or not that's going to be feasible, I don't know. It's all going to come down to material peformance and material improvements."

For Kleiss, that material improvement Sheridan spoke of might be PEEK.

A colorless, organic polymer, PEEK (polyether ether ketone) is a semicrystalline thermoplastic with excellent mechanical and chemical resistance properties that are retained to high temperatures.

More importantly — and most relevant to this particular discussion — PEEK melts around 343°C (662°F). Some grades have a useful operating temperature of up to 250°C (482°F).

In other words, it can take the heat — and then some.

"We got to this point where we're taking the normal engineering materials — nylon, acetal, etcetera, and we're beginning to make power gears with them" Kleiss says. "That was all good to a point, but then we got introduced to PEEK, and that is a very unusual plastic.

"It has much higher temperature capabilities than the other materials. You have to mold it at 700°F. In its virgin state, it has a low modulus. Now, low modulus is a detriment in general, because a gear will fail because it bends and breaks. So in a power situation you want a stiff gear to carry a lot of

load. But what we began to discover is that plastic fits a niche of gearing in quiet applications — when you have vibrating drives you have to use scissors gears and do a lot of damping and things like that. With plastic, that low-modulus material sort of acts like a spring. It takes care of a lot of problems just by the nature of the material, if everything else is right.

"About 10 to 15 years ago is when we got our first customer that said, 'What about this PEEK material?' That's when we started working with it and found a fundamental difference between it and all the other plastics. We walked away from fillers and said that we were only interested in [PEEK] in its virgin state, because that's where we can make the most accurate gears.

"Now we have a material that we think can withstand the kind of temperatures we're going to expose them to in these high-torque applications. It is able to withstand higher temperatures than any other material that we know of — and heat is the fundamental enemy of thermoplastics."

While new, more exotic materials like PEEK bring added benefits to the table, they also have an innate problem — they're more difficult to work with.

"If a customer comes to us and says that they have an application and they think it will work in [an exotic material], well it may work in that application but it may be very difficult to develop a mold to use that material and a molding process that is consistent and repeatable over a large volume," Winzeler says. "The more exotic the materials become the more challenges we have as processors.

"Because we're looking at large volume applications, we need to work with a material that offers consistency and repeatability. We have to ensure that we get structural integrity and dimensional accuracy from part to part — from Part 1 to Part 1 million to Part 10 million. If we can't do that consistently than we don't have a viable business model.

"They're all tradeoffs. What the product engineer with our client wants may not necessarily be a good solution for our manufacturing floor. And some of that is learning curves and knowledge and so forth. That's the challenge with some of these newer, more exotic materials, is that we don't have the lessons learned; we don't have the design data; we don't have the history of performance successes of knowing what we can and cannot do. That's currently an ongoing R&D process."

Another issue when dealing with exotic materials is cost. And when you combine the decreased knowledge base with the increased price, it stands to reason that while these newer materials do provide added benefits, their inherent deficiencies will probably prevent them from ever becoming the "workhorse" material that acetal currently is, according to Sheridan.

"Let's be honest, cost is king," he said. "Folks aren't going to pay more than they have to. The higher temperature materials require higher temperature equipment to manufacture. The hotter things are, the more specialty equipment you need to handle them, to operate them, to manufacture them, and all of that is added cost as well. Plus, they're newer and not known as well."

So, instead of doing what Kleiss did — which was to essentially throw previous materials out the window and focus on developing a new, more advanced material like PEEK — Celanese has continued to work intensively with Hostaform, "it's bread and butter," while dabbling in perimeter exotic materials like Fortron PPS, a high-temperature semicrystalline polymer.

Winzeler, too, has continued to perfect DuPont's Delrin by making minor changes and slight tweaks to that pre-existing, familiar source material.

"I don't know how many materials are out there, but let's say feasibly there might be 200 materials that might be used for a gear," Winzeler said. "Our research is focused on maybe 10 materials. So what we've tried to do is learn a lot about a few materials to really apply them well. This is not only taking into account their performance, but the economics. What do they cost in the raw state, and what do they cost to convert into a plastic material?

"We continue to go deep into [Delrin]. That's our sweet spot. The perimeter is these exotics. We're convinced that in our universe — and that includes under the hood — that it can be done with very few materials. You just need a lot more knowledge."

Knowledge is Power

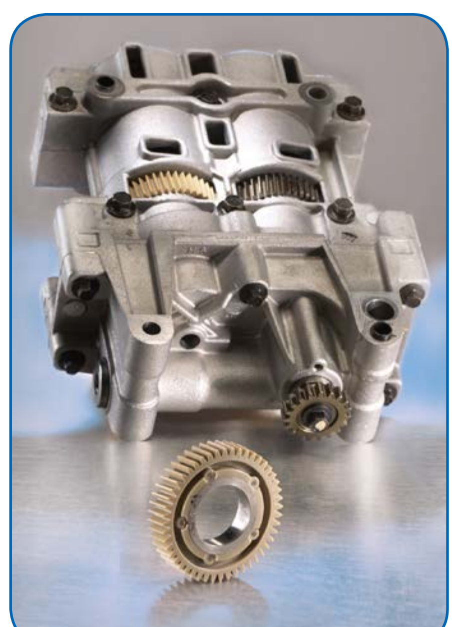
Between Kleiss, Sheridan and Winzeler, you have three fairly different points of view on what the vehicle to the future of plastic gears is.

What the three industry experts do agree on, however, is that whether you're using an exotic material like PEEK or a tried and true substance like Delrin or Hostaform, knowledge is the key to unlocking their potential, because — in theory — the only limitation to how far plastic gears can go is our own lack of knowledge and a dearth of sufficient testing.

Winzeler gave an example of the problem:

"Thermoplastics tend to have fairly high rates of thermal expansion," he says. "So, let's say that we were going to take a transmission gear that's 12-14 inches in diameter, and it can replace the metal gear. If you wanted to understand it's behavior radially and how it's going to grow or shrink depending on whether its –40° or 120° you would have to make parts, measure the delta over the temperature range independent of moisture and other environmental conditions that might affect the size, to understand that.

"The data that they publish on a standard ASTM test bar does not relate to the manufactured product because of how it's gated and how this is done



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or that's done. So that's where this knowledge base is so nebulous.

"We've had applications where automotive has come to us on large gearing and the client said, 'Prove to us on paper that this will work.' The answer from us was that you have to go do it and measure it, and the mentality with a lot of the very large corporations is that if 'you can't prove by computer then we aren't going to consider it.'

"The barrier [for plastic gears] is the knowledge of all the parameters of all these materials in the application so that you can do good product design. And that's huge. It takes a lot of time to accumulate that data."

Kleiss agrees that testing is vital to the continued advancement of plastic gears.

"You have to realize what you're working with and how you need to work with it, he says. "You can't use convenient logic on a plastic gear. You have to be very specific about what you're seeking."

More and more, plastic gears are capable of doing jobs in the automotive forum that could previously only be accomplished by metal. Kleiss says he doesn't see that process stopping — or slowing down — anytime soon.

Because plastic gears aren't worse than metal ones.

They're just different.

"Plastic is not necessarily a weak alternative," Kleiss says. "Sometimes it can be the absolutely preferred choice." **PTE**

For more information:

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Kleiss Gears, Inc.
Phone: (715) 598-4486
www.kleissgears.com

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Phone: (708) 924-1713
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Plastic Bearings Have Staying Power

Nicole Lang, igus product manager

During the past three decades there has been an evolution in the advancement and use of highly engineered plastics in bearing applications. Plastic bearings are no longer designed like their dime-a-dozen, injection-molded nylon ancestors. Today, plastic bearings cost and weigh less than their metal counterparts, but what many design engineers still do not realize, is that they also often last longer in unforgiving environments.

High-performance plastic bearings are working to shed their negative image and continue to forge a path into almost every industry; from packaging machines and medical devices, to automotive, farming equipment, textile machinery, and many more. Plastic plain bearings are an economical replacement for needle, ball, and plain metal bearings. However, they are often not considered a viable choice in the engineering community due to the common misconception that plastic is inferior or weaker compared metal.

The truth is that composite plastic bearings can outperform their metal counterparts in countless rotary, oscillating, and linear-motion applications. In addition, plastic bearings are readily available in many different styles, sizes, materials, and colors to meet the demands of almost any application.

The argument for plastic. igus designs and develops its high-performance, cost-effective plastic bearings for almost any application. The company's plastic bearings are an off-the-shelf solution, and are available in more than 120 different plastic compounds. Each is comprised of three parts:

1. Base polymers, which are responsible for the resistance to friction and wear
2. Reinforcing fibers and filaments, which make the bearings ideal for high forces and edge loads
3. Solid lubricants, which are blended into each material and eliminate the need for any external oils or grease

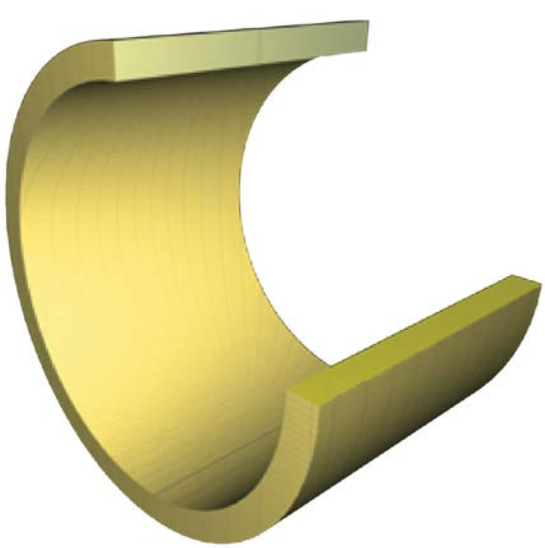

Figure 1 Injection molded iglide plain bearings are homogeneously structured. Base polymer, bonding materials and solid lubricants mutually complement each other.

While most plastic bearings, like the ones from igus, can endure extreme temperatures, heavy loads and high speeds, it is still important to understand both the advantages and disadvantages of the different options available.

iglide plastic bearings vs. bronze bearings. Maintenance-free plastic bearings regularly deliver a longer service life and a cost savings of up to 40 percent when compared to oil-impregnated sintered bronze bearings. Plastic bearings not only outlast their metal counterparts, but prove more economical because no grease or oil lubricants are required, eliminating the need for routine maintenance.

With sintered-bronze bearings, oil is drawn from the bearing as it rotates on the shaft (minimum speed of 200-feet-per-minute). The oil creates a thin film that then separates the bearing and shaft, preventing wear and shaft damage.

At high speeds, a low COF is achieved. Shaft oscillation, slow speeds, irregular use or uneven loads can impede film lubrication from being maintained. As a result, the COF and wear rates increase. In addition, if movement stops completely, the oil on the bearing surface dries up and cause higher friction and squeaking. High temperatures can also break down the oil. In addition, oil film on the shaft can act like a magnet for dust, dirt and airborne debris, which can seize up the bearing or contaminate a product or process, especially in food or medical applications.

Common Misconceptions

Despite the performance advantages, several misconceptions may make engineers reluctant to take full advantage of the benefits of thin-walled plastic bearings:

1. The wall thickness of either bearing does not directly correlate to its strength. Other factors that are more important and should be taken into consideration include the weight, coefficient of friction and wear capabilities of the bearing.

   In addition to the material, a basic difference between thin-walled plastic bearings and thick-walled bronze bearings is thickness. Thick-walled bronze bearings feature a standard wall thickness between 0.0625 and 0.156 inches. In comparison, the wall thickness of a plastic bearing is much less, typically between 0.0468 and 0.0625 inches. Due to their thin walls, plastic bearings not only offer a number of benefits, but also perform equally as well, if not better than a thick-walled bearing.
2. Due to its thin wall, the surface pressure of a press-fit plastic bearing will be negatively affected. Another mistake is to assume the thin wall of press-fit plastic bearings will affect the surface pressure. Actually, the surface pressure of a press-fit bearing, typically rated in pounds per square inch (psi), is determined by the load divided by the surface area it acts on: $Ps = L/(D \times l)$ where Ps = surface pressure, psi; L = load, lb; D = inside diameter, in.; and l = bearing length, in. Whether one is using a thin-

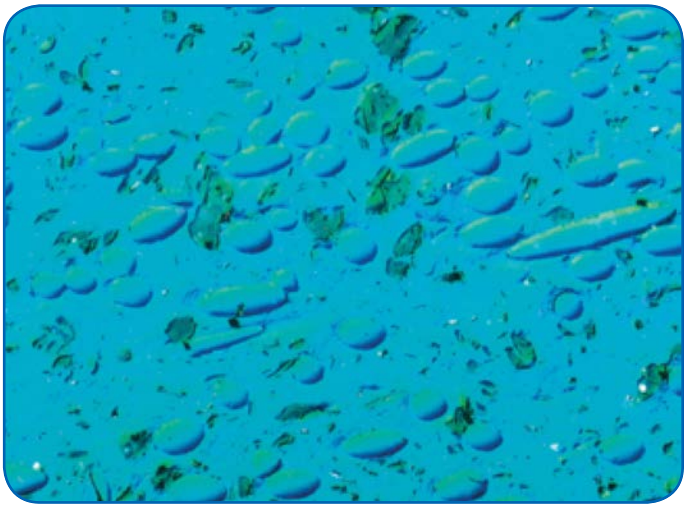

Figure 2 Base polymers with fibers and solid lubricants, magnified 200 times dyed.

walled plastic bearing or a thick-walled bronze bearing, wall thickness has no effect on surface pressure.

3. A thin-walled plastic bearing has a shorter life than its thick-walled bronze counterpart. It is reasonable to assume that since a plastic bearing has less material (a thinner wall), it will not last as long as a thick-walled bronze bearing. This is incorrect because the thin wall of a plastic bearing helps to dissipate any heat buildup, which actually prevents wear. In high-rotation applications, continually re-lubricating the bronze bearing will help prevent wear. However, if a bronze bearing is being used to facilitate other types of motion; excessive wear can lead to added clearance between the shaft and the bearing. If this happens, a number of problems will arise. It is important to remember that wear is dependent on the makeup of the bearing material and not on the wall thickness (refer to misconception one). For this very reason, igus is constantly developing new plastic materials, which minimize wear and provide a long-lasting, maintenance-free solution for a variety of applications.

Reasons to Replace PTFE-Lined Bushings

Plastic bushings are now designed to handle high speeds, loads, temperatures, caustic chemicals and a wide array of other application factors. Here are the top four reasons for replacing PTFE-lined bushings with plastic bushings, which offer more design flexibility.

1. Thinner wear surface. A PTFE-lined bushing is comprised of a metal shell and a very thin polymer coating (PTFE) applied to the inside. These types of bushings typically have a maximum wear surface of 0.06 millimeters (0.002 inches), but as the PTFE coating is stripped off during operation, the metal shell becomes exposed. This creates a metal-on-metal effect between the bushing and the shaft and can cause serious damage. This problem is common when high edge loads or oscillating movements are present.

In comparison, plastic bushings are comprised of advanced compounds, which contain solid lubricants embedded in millions of tiny chambers throughout the material. During operation, lubricant is transferred onto the shaft to help lower the coefficient of friction and wear, and unlike PTFE-lined bushings, plastic bushings eliminate the danger of metal-on-metal contact. This is huge benefit since the acceptable amount of wear can be determined by the type of application (unlike the PTFE-lined bushing, which will fail if the wear rate surpasses 0.06 millimeters).

Figure 3 Washing chain bearings. Reduction of the drive power for bottle washing machines by using iglide under the most difficult conditions in a 2-3% caustic soda and temperature of +176°F (Krones AG).

For example, igus' lifetime calculator uses a preset wear rate of 0.25 millimeters (0.01 inches), but the user can easily increase or decrease this number to meet the wear limit acceptable for the particular application. Unlike PTFE-lined bushings, plastic bushings eliminate the danger of metal-on-metal contact. Almost any of igus' iglide plastic bushings can replace a PTFE-lined bushing. One of the most popular is iglide G300.

iglide G300 is ideal for demanding applications with medium to high loads, average surface speeds, and moderate temperatures. It is available as a sleeve bushing, flange bushing or thrust washer.

2. Increased weight. PTFE-lined bushings weigh more than plastic bushings. When using a heavier bushing, no matter what material it is comprised of, more energy is required for the bushing to operate. This can be troublesome, especially in automotive, aerospace, recreational vehicle, and bicycle applications.

In contrast, plastic bushings are lightweight, which helps decrease fuel consumption and carbon dioxide output. The reduced weight can also help reduce carbon dioxide output, lower masses and, subsequently, lower energy consumption.

In a weight comparison, an iglide plastic bushing weighs approximately 80 percent less than a PTFE-lined bushing.

- iglide G300 plastic bushing = 0.0144 pounds per piece
- PTFE-lined bushing = 0.0750 pounds per piece

3. No corrosion or chemical resistance. The metal shell of a PTFE-lined bushing is not ideal for applications where water or caustic chemicals are present.

In these types of applications, PTFE-lined bushings can rust, corrode, contaminate sensitive areas, and ultimately fail. Since plastic bushings are made solely of high-performance polymers, they offer both corrosion- and chemical resistance and operate unaffected in those types of environments.

4. No resistance to biofuels. The trend towards the increasing use of biofuels and biodiesels creates problems when using PTFE-lined bushings; after limited exposure to moisture, parts of the bushing's metal shell can begin to peel off. However, these types of applications open new doors for plastic bushings. Since they are corrosion-resistant, plastic bushings remain unaffected despite the fact that biodiesel has the tendency to absorb a great deal of water. iglide T500 plastic bushings can be used in applications with biofuels and biodiesels.

Reasons to Use iglide Plastic Bearings

All iglide bearings have excellent antifriction and low-wear characteristics that produce a self-lubricating effect. This is especially important during the start-up phase of an application. With metal bearings, the lubricant film has not yet formed and the bearing begins operating dry, which can accelerate wear. In comparison, iglide plastic bearings are homogeneously impregnated with solid lubricant, and run "lubricated" from the start. During operation, plastic bearings transfer lubricant onto the shaft to help lower the COF. This also minimizes slip-stick conditions and wear, as well as increases operating life, unlike with plain metal, ball, and needle bearings. Dimensional changes to the bearing are essentially non-measurable, and abrasion decreases rapidly following startup and becomes negligible in continuous operation. In addition, the fiber-reinforced materials maintain the bearing's strength and resistance to high forces and edge loads. Plastic bearings can also be used on many different shaft types.

In extremely dirty conditions, particles simply embed into the wall of a plastic bearing with little effect on performance. Plastic bearings offer other advantages as well, including the ability to withstand chemicals and certain types of corrosives such as hydrocarbons, alcohols and alkaline solutions. igus also offers FDA-approved plastic bearings that permit contact with food and pharmaceuticals.

Figure 4 Surgical Light. The motor-powered swiveling LED wings are adjusted with the aid of iglide JVFM bearings. Self-lubricating and maintenance-free (Trumpf iLED Medical Systems Inc.).

Figure 5 Spreaders. Main reasons for iglide bearings: The special design to complement the centrifugal arm results in a significant reduction of manufacturing costs. It is also maintenance-free and has high wear resistance (Fella Werke GmbH & Co. KG).

Many engineers are also surprised to learn that plastic bearings can be used in high temperature applications. For example, certain igus bearings can operate continuously at temperatures approaching 500°F, as well as withstand peaks to 600°F and lows of –148°F.

Plastic bearings also run quietly and absorb mechanical vibrations. The so-called mechanical loss factor, an indicator of vibration-damping capability, is up to 250 times higher than that of plain-metal bearings.

For applications where weight and fuel economy are an issue, for example in racing bikes, snowmobiles, automobiles, and motorcycles, a thin-walled plastic bearing is ideal. The image below compares the weights of different bearing materials.

Successful Applications

Plastic bearings have already replaced plain metallic bearings in thousands of applications from a wide range of different industries, including agricultural machinery, medical equipment, fitness equipment, packaging machinery and more. Engineers are increasingly turning to plastic bearings in a wide range of challenging applications.

In the medical industry, iglide plastic plain bearings from igus are helping to improve the way prostate cancer is detected and treated. A team of researchers from the Worcester Polytechnic Institute (WPI) in Massachusetts have developed a specialized magnetic resonance imaging (MRI) compatible piezoelectric actuated robot. To facilitate different types of motion, the robot uses iglide plastic self-lubricating bearings and DryLin linear guide systems to facilitate translational motion of the positioning module, which provides gross positioning for the robot's needle driver.

The needle driver is a vital part of the system, as it enables the rotation and translational movement of the needle cannula: a flexible tube inserted into the patient's body cavity for MRI-guided diagnosis and therapy. The needle driver has a needle guide sleeve, a collet locking mechanism and passive optical tracking fiducial frame. Two plastic plain bearings are used in the front and rear of the driver to constrain the needle guide. The bearings enable the robot's motor to rotate the needle using the collet mechanism by way of a timing belt. This rotating needle would reduce tissue damage while enhance targeting accuracy. Another 10 plain bearings were used to create a revolute joint, also known as a "pin joint" or "hinge joint", to provide single-axis rotation. The plastic bearings and linear guides operate without messy lubrication, which is important in a sterile medical environment. The plastic bearings also ideal because they are comprised of FDA-compliant polymers specifically designed for applications with contact to food or drugs.

In another case, an OEM turned to plastic bearings for equipment that packages flour, sugar, and various types of pet food. The machines operate around the clock and are expected to last 20 to 30 years. To meet the demanding durability requirements, the company's engineers specified plastic linear bearings on guide rods in the machine's trimming and pressing stations. The linear bearings' aluminum adapter fits over a plastic liner. The beefed-up construction lets them carry up to thirty 50-lb bags/min on each machine, 43,200 times per day. The dry-running bearings are unaffected by flour or sugar dust that gets stirred up during packaging, and will not contaminate food products, unlike bearings that require external lubrication. Plastics bearings have excellent strength, good thermal properties, and need no external lubrication. And the low-cost, lightweight bearings deliver long life despite exposure to harsh chemicals, dust, dirt, and other contaminants. With advances in polymer engineering, plastic bearings now outperform metal in many applications.

A third application is a pasta manufacturer that recently replaced V-grooved, track-guided rollers on its cartooning machines with plastic plain bearings. The machines, which operate 24/7, use a shuttle bucket to carry and unload one-pound portions of pasta. The bucket travels 18 inches, 240 times a minute, to keep up with the machine's load station. Despite the rapid cycling and extreme acceleration, the plastic bearings last more than three times longer than the previous roller bearings and have reduced annual repair costs by $7,800. And the lubrication-free bearings cannot contaminate the pasta or packaging. Replacement, if necessary, takes less than two hours — in contrast to the full day of downtime it takes to rebuild just one set of rollers. And, as an added benefit, the company reports vibration issues have been eliminated and the machines run much quieter. The plastic bearings are also corrosion-resistant and maintenance-free, making them cost-effective replacements for most ball bearings. Their oil-free operation is also a huge advantage, as FDA regulations prohibit most lubricants for sanitary reasons, and even approved lubricants attract dust and dirt, which can eventually cause bearings to seize.

The Ability to Predict Bearing Life

igus offers a service-life calculator called *The Expert System* for its lines of iglide plain bearings and DryLin linear bearings. These convenient tools are based on an extensive tribological test database and have been verified with thousands of hours of actual testing that make it possible to predict plastic-bearing life under almost any operating condition. Users enter various data — proposed bearing dimensions; maximum loads; whether motion is rotating, linear, pivoting, or a combination of the three; speed; whether the motion it is intermittent or constant; operating temperature range; chemical exposure; mating surface; and acceptable limits on bearing wear — and the system uses these factors to calculate bearing life. *The Expert System* then calculates and delivers results, including life in hours and travel distance for various suitable products.

The Expert System delivers trusted results, but igus does encourage testing a selected bearing in the proposed application before releasing a machine to the market. igus supplies free test samples and advice through its expert sales engineers to select the best material and design for a given application. **PTE**

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Figure 6 Tool changer chain. Main reasons for iglide bearings: Enormous cost advantages in comparison to standard metallic rolled bearings as well as low coefficient of friction and with soft shaft materials (Deckel Maho Seebach GmbH).

Figure 7 Axle box arrangement. The edge load is usually a deciding factor for or against the use of bearings. iglide G bearings solve this, also giving high wear resistance, low costs, resistance to corrosion and dirt (Zunhammer GmbH Gülletechnik).

Figure 8 Tubular bag machines. The continuous operating temperature in the bonding arms frequently reach + 320°F and higher. These requirements are met by iglide Z bearings which also offer particularly high resistance to wear (Affeldt Verpackungsmaschinen GmbH).

Nicole Lang — national product manager, dry-tech — joined igus in 1998, moving in 2004 to iglide inside sales. She is a 2001 graduate of Roger Williams University, where she also earned her law degree. Lang was promoted in 2007 to national product manager for iglide plastic bearings. She regularly travels to igus HQ in Germany for intensive technical training, which she in turn passes on to her staff. Lang lives in Rhode Island with her husband and daughter.

Wind Pushing Future of Mechanical Components

Randy Stott, Managing Editor

The global wind power industry is BOOMING again.

After a sluggish 2013, annual installations of new wind turbines grew by 44% in 2014, according to the Global Wind Energy Council. And while much of that growth has been in Asia—particularly China, which now leads the world with 114 GW of installed capacity—the USA, Europe, and the rest of the world expect steady growth for the next couple of years as well (Fig. 1).

The Dry Lake Wind Power Project, online since 2009, was Arizona's first utility-scale wind farm (courtesy AWEA).

In the USA, of course, capital investment in wind energy projects is largely influenced by the infamous Production Tax Credit, which the U.S. government allowed to expire at the end of 2014. Despite that, the projects that began prior to the end of last year will keep the industry busy through 2015 and much of 2016.

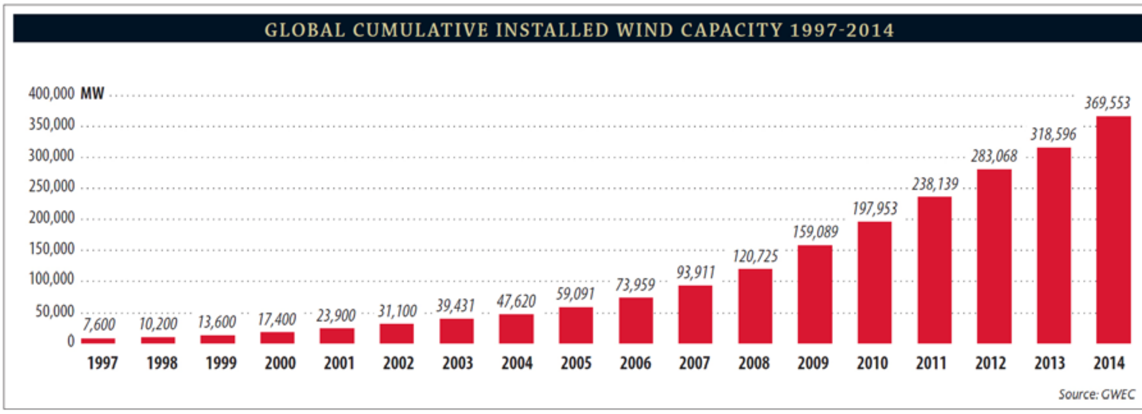

Figure 1 Growth in the global installed capacity of wind turbines (courtesy GWEC).

So there's still reason for optimism, especially if you're a supplier to the industry. Gear drives, bearings, couplings and related components not only help translate the power of wind into electricity, but they're also responsible for the many devices on wind turbines that help position the nacelle and blades to maximize productivity.

According to the American Wind Energy Association (AWEA), the U.S. wind-related manufacturing sector consists of more than 550 manufacturing facilities across 44 states, producing more than 8,000 components that comprise a typical wind turbine. U.S.-based facilities make everything from major components such as blades, towers, rotor hubs, generators and gearboxes, to internal components such as bearings, slip rings, brake systems, fasteners, power converters, sensors and control systems—reason enough for readers of *Power Transmission Engineering* to pay attention.

In fact, the AWEA says that over the past five years, new wind power project installations have grown at an average rate of 36% per year in the USA. This has allowed many U.S. manufacturers to get involved in the industry, bringing down the overall cost of a wind turbine and increasing the amount of U.S. manufactured content from less than 25% in 2005 to more than 67% today.

One of the best places to learn more about how you can get involved in the global wind turbine manufacturing industry is at the AWEA-sponsored Windpower 2015 Conference & Exhibition, which takes place May 18-21 in Orlan-

"This is our annual event, where everyone connected with the industry or interested in the industry comes and is available—so from a networking perspective or from an educational perspective, discovering what's happening in the industry and what opportunities are there for your business—it's *the* event."

—Susan Reilly, AWEA Board Chair

do, Florida. The annual event features major suppliers to the wind power industry, including many gearbox, bearing and other mechanical component providers (see sidebar for highlighted exhibitors from this year's show).

For example, the Schaeffler Group (Booth 2122) will present a number of the company's latest advances for wind turbines, including its FAG X-life cylindrical roller bearings for planetary gears, replace as well as the WiProM portable diagnostic tool, which offers the performance of a permanently mounted system in a rugged, portable unit.

WiProM is Schaeffler's new diagnostic tool for analyzing the performance of wind turbine components such as motors, gearboxes and generators.

Bonfiglioli's 700T Series planetary speed reducers are used by a number of leading wind turbine manufacturers for pitch and yaw control. The 700T are flange-mounted reducers with a torque range from 2,500 up to 300,000 N-m. They can be manufactured with three to five planetary reduction stages, providing ratios from 60 to 3,000 (Fig. 3).

Centa's Centalink couplings offer misalignment capability up to 2 degrees under rated operating conditions (and up to 6 degrees in exceptional cases). They are available for wind power applications from 6 to 50 kNm, and they're rated for temperatures from –45°C to 80°C.

We will be waiting for you at Hannover Messe Germany (13-17th April), and Agrishow Ribeirao Preto Brazil (27th April to 1st May)

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Figure 3 The 700T Series of gearboxes and gearmotors from Bonfiglioli is used for control of pitch and yaw in wind turbine applications.

In addition to the exposition, Windpower 2015 also offers educational opportunities in the accompanying conference. **PTE**

Windpower 2015

AWEA Windpower Conference & Exhibition 2015
Orange County Convention Center
Orlando, FL
May 18-21
www.windpowerexpo.org

Learn more about the show on AWEA’s YouTube Channel at *www.youtube.com/user/americanwindenergy*

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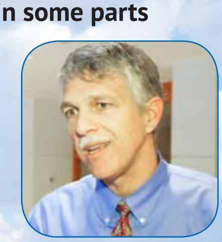
“Wind energy is growing throughout this country, but it’s growing more in some parts of the country than others. In Florida, we think there’s a great opportunity for more.”

—Tom Kiernan, AWEA CEO



Expand Your Design Options with a Visit to Powdermet

By Randy Stott, Managing Editor

Powdermet 2015

MPIF/APMI International Conference on Powder Metallurgy & Particulate Materials

Hilton San Diego Bayfront Hotel
San Diego, CA
May 17-20
www.mpif.org

Many of the best applications for powder metallurgy (PM) are found in mechanical power transmission components. Every year, at the industry's annual conference, when the PM design awards are announced, the winning parts are often gears and transmission parts (Fig 1).

Because PM processes offer significant advantages over other metal forming and metal cutting operations, they are often ideally suited for mechanical parts. Those advantages include minimal material waste and the ability to form complex shapes, making them ideal candidates for many transmission parts.

The industry continues to make significant advancements in terms of part strength and finishing, and Powdermet is the place where industry experts gather to share and exchange knowledge. The conference includes more than 200 technical presentations from industry experts.

The educational sessions are supported by an exhibition featuring more than 100 booths. Exhibitors include leading suppliers of powder metallurgy and particulate materials and processing equipment, powders and related products.

Figure 1 Among the 2014 MPIF Design Excellence Award grand prize winners (pictured here) were an automotive transmission planetary carrier assembly and a dis-engagement mechanism for a snow-blower system. This year's winners will be announced at Powdermet 2015.

Figure 2 The sector gear and fixed ring shown here were winners of the 2014 MPIF Design Excellence Award of Distinction in the automotive transmission category. This year's winners will be announced at Powdermet 2015.

Servo Motor Performance at Stepper Motor Prices?

Robust new sensor system doing more with less

Jack McGuinn, Senior Editor

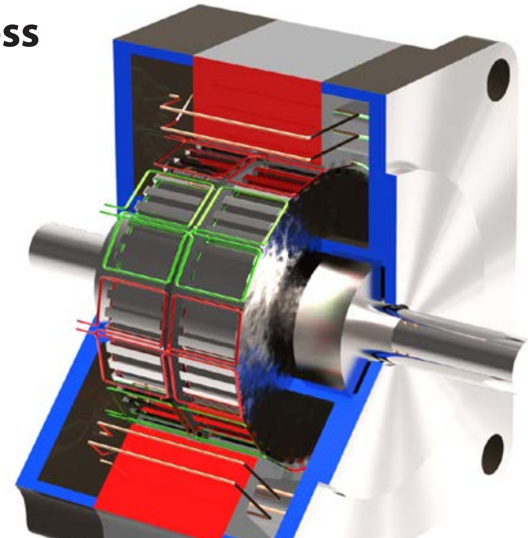


Figure 1 Cutaway view of the Mosolver showing the position sensor coils that have been added to a standard hybrid synchronous motor (All photos courtesy QuickSilver Controls).

Enhancing production with — and for, less — is the standing order in today's manufacturing world. Speeding up production while at the same time looking for ways, to cut, for example, energy costs, is a tricky equation with no single answer; where and how management goes about achieving that can take several paths. But in many manufacturing and countless other end-user environments, easily among the lowest-hanging energy-sucking targets remain NEMA inefficient, non-compliant electric motors, and the systems in which they are incorporated. So any news of a technological advance in that regard is always welcome, and that seems to be exactly what we have with the 2014 rollout of QuickSilver Controls Inc.'s patented Mosolver, i.e. — a servo motion actuator that, according to QuickSilver, "infuses a position feedback sensor into the very structure of a high pole count AC motor," thus eliminating "the need for costly encoders and resolvers."

And Don Labriola, the man behind the curtain whose baby is the Mosolver, offers additional reasons for considering the system a game-changer (*Ed.'s Note: See also sidebar*).

"I believe the Mosolver provides a rugged and cost-effective transition path to transfer up from steppers into Hybrid servo — same form factor. It not only prevents loss of steps, but allows true closed-loop control with a significant improvement in damping. Not only is it smaller than adding an encoder, it is inherently aligned, provides commutation information, and is robust.

"The ruggedness of the polyamide sense circuit within the stator structure of the motor is able to reach or exceed the thermal/radiation/contaminant levels permitted by the motor itself. The flex circuit is stationary, being flexed only for installation. There are no encoder discs to be contaminated or moving parts — other than the motor rotor itself."

And Labriola also explains that "the elimination of a separate feedback device reduces cost and size, eliminates alignment (and loss thereof) and greatly improves ruggedness of the system." What's more, by combining the motor and resolver into one package, the Mosolver enables a compact, inexpensive and robust closed-loop motion control.

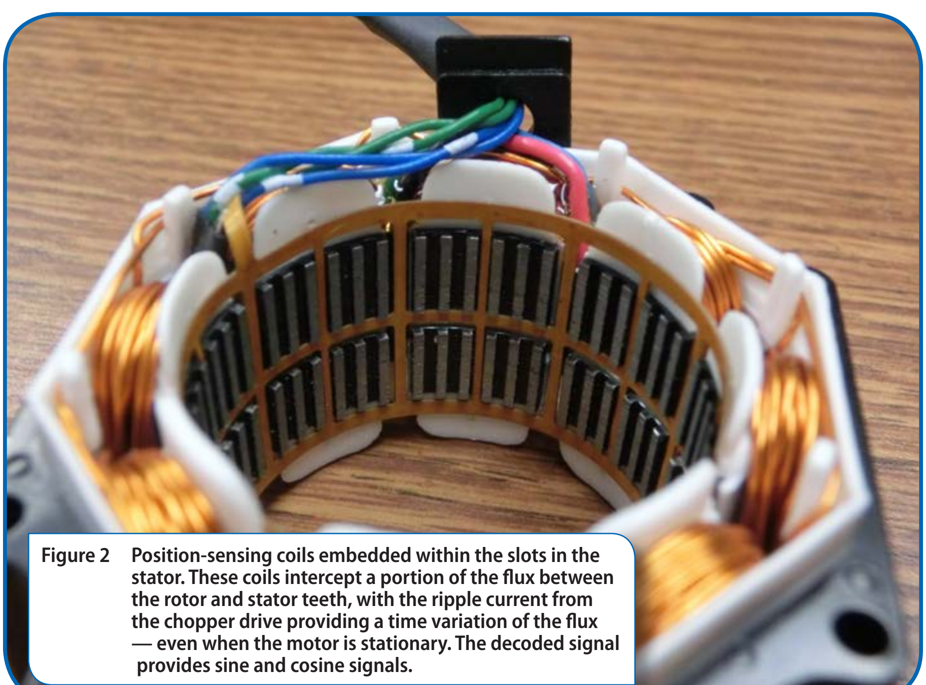


Figure 2 Position-sensing coils embedded within the slots in the stator. These coils intercept a portion of the flux between the rotor and stator teeth, with the ripple current from the chopper drive providing a time variation of the flux — even when the motor is stationary. The decoded signal provides sine and cosine signals.

As you might imagine, the newly developed QuickSilver sensor is critical to the Mosolver's enhanced capabilities and its claims of uniqueness. What impresses is that the sensor was developed in-house — with no outside brainpower.

"The Mosolver was developed in house to allow replacement of encoders for size, cost, resolution, and robustness reasons," says Labriola. "We add sense coils on a polyamide flex circuit to the stator structure. In the case of a 2-phase hybrid stepper/servo, only a slight modification to the stator is needed to

"Operating these motors closed-loop takes advantage of their manufacturing-optimized design, low magnetic materials cost, and excellent motor torque capabilities. Gone are the low-frequency resonance problems, most of the noise and heating problems, and problems with lost steps."

allow passage of the flex circuit along the portion of the stator facing a magnet in the rotor. As the gap there is quite large, the modification does not appreciably change the magnetic paths."

The sense coils appear to be critical to Mosolver's efficacy as well; we asked Labriola to more fully explain that part of the system.

"The sense coil is arranged in the stator to intercept a portion of the flux arising from the motor PWM (pulse width modulation) drive. The ripple current associated with a PWM drive causes a flux variation — even when the motor is not in motion. The sense coil has differential sections arranged in the stator structure to measure the differences in the flux traveling through different sections of the stator. The position of the rotor teeth with respect to the stator teeth determines the path of minimum reluctance for the chopper-induced flux variations; and the paths vary with rotor position. Appropriate placement produces sine and cosine signals with a period corresponding to an electrical cycle of the motor.

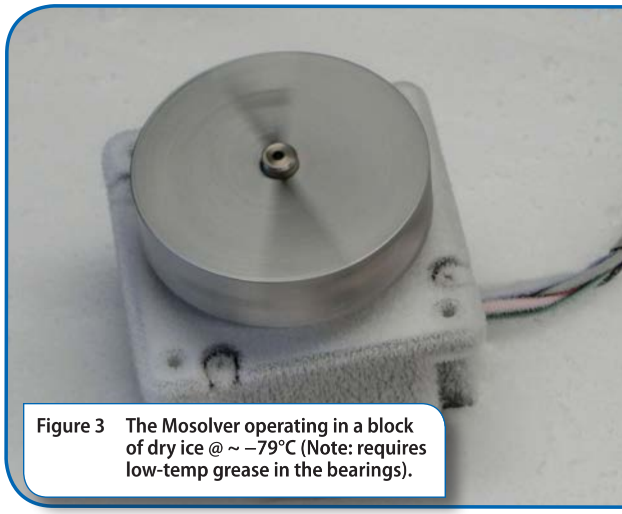


Figure 3 The Mosolver operating in a block of dry ice @ ~ −79°C (Note: requires low-temp grease in the bearings).

"The sensor coil is completely contained within the existing motor frame, requiring no additional volume in the application. The use of a polyamide flex circuit containing only traces makes the sensor very robust, i.e. — polyamide circuits that have a very wide usable temperature range — from cryogenic to as high as 350°C — can be made to be low out gassing for vacuum applications, and can withstand high radiation doses without damage. By using both the existing magnetic structure and the existing motor PWM drive, the sense coils allow both commutation and position feedback with very little electronic overhead."

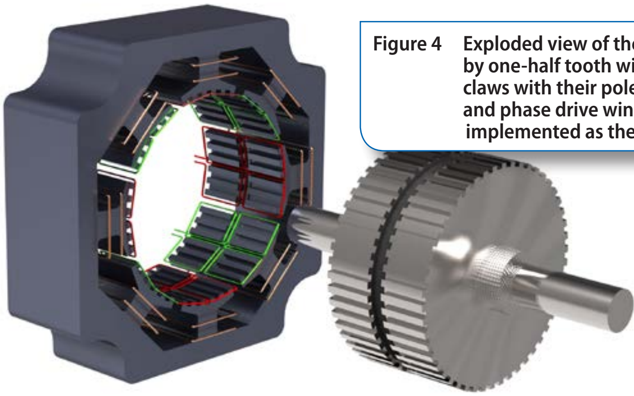


Figure 4 Exploded view of the Mosolver. The rotor has two pole caps, with the north and south poles skewed by one-half tooth width; the magnet is between the pole caps and is axially magnetized. The stator claws with their pole teeth are also exposed, showing the location of both sensor windings (red, green) and phase drive windings (copper). The actual sensor windings have a multiple turns-per-coil, and are implemented as the flex circuit shown in Figure 2.

And of course software is always central to the performance of systems as mentioned above, and the Molsolver is no different. As described by Labriola, "The software manages the trajectory generation, the control system, the digital drive signal timing, and the decoding of the sensor signals. The software effectively eliminates much of the additional hardware required for resolvers by making the existing motor and drive serve multiple purposes. The software also serves the function of the resolver/digital hardware for resolver based-systems."

At this point QuickSilver is focused on incorporating the Mosolver/sensor in hybrid servo motors that are built on a high pole count 2-phase hybrid motor — more commonly known and deployed as "step" motors. The drive and the presence of feedback allow these same motors to fully operate as servo motors, thus taking advantage of their low cost, high reliability, and very high, continuous torque capabilities. Labriola also revealed that the company is in the process of applying these sensors to Sawyer motors, including a planar structure in which the technology is applicable to a range of 2- and 3-phase motors.

And if you're a systems/line integrator or are otherwise involved in specifying motors/systems, you'll want to know that the Mosolver should have no trouble "fitting in."

"The Mosolver was designed and first implemented for the upgrade of hybrid step motors to hybrid servo motors," says Labriola. "There is a

"The elimination of a separate feedback device reduces cost and size, eliminates alignment (and loss thereof) and greatly improves ruggedness of the system."

very large usage of step motors — a much higher quantity of these motors are used than standard servo motors. Operating these motors closed-loop takes advantage of their manufacturing-optimized design, low magnetic materials cost, and excellent motor torque capabilities. Gone are the low-frequency resonance problems, most of the noise and heating problems, and problems with lost steps. The controller needs to be designed to acquire and decode sensing these signals."

When asked, Labriola says that birthing the Mosolver — from prototypes to board layouts to software modifications for enabling optimal operation — took about a year. Since then (2014), a tweak or two has been added.

"A slot is required around the circumference of a 2-phase hybrid motor to accept the sense coil," Labriola explains. "The differential coils optimally cross the center line of the magnet to produce a differential signal. The location of such a slot with other motor types varies. Motors designed for a good sinusoidal back-EMF (electric/magnetic fields) tend to produce more sinusoidal sense signals."

Referring back to motor/energy issue, motor environment ratings are a serious factor, as enforced compliance becomes more and more of a reality — a reality that will not go away, given that most of the motors on factory floors and everywhere else in this country are woefully out-of-date, and therefore NEMA compliance. Quicksilver takes the view that "the Mosolver environmental ratings are essentially limited only by the motor for wide temperature, wet, and vacuum applications, as well as for operation in radiation environments."

"The software effectively eliminates much of the additional hardware required for resolvers by making the existing motor and drive serve multiple purposes."

Expanding further, Labriola adds that, "According to the individual extension of the environmental ratings desired, the motor design must be consistent. For example, some magnet materials do not hold up under radiation. Others may need coating to handle wet or vacuum applications to limit corrosion or outgassing, respectively. Wide temperature ranges may require polyamide-coated wires, and special bearings and or lubricants — as well as wire guide inserts made from appropriate materials. These challenges have already been handled for open-loop steppers operating in these environments. Adding closed-loop capabilities allows these motors to operate with better margins and, typically, significantly cooler, as only the current required for the torque load is applied to the motor."

Can the Mosolver be improved?

"Yes!" Labriola enthuses. "We have patents in place to add single turn absolute capability, and are working to add some significant enhancements which we will reveal when they are ready for market!" **PTE**

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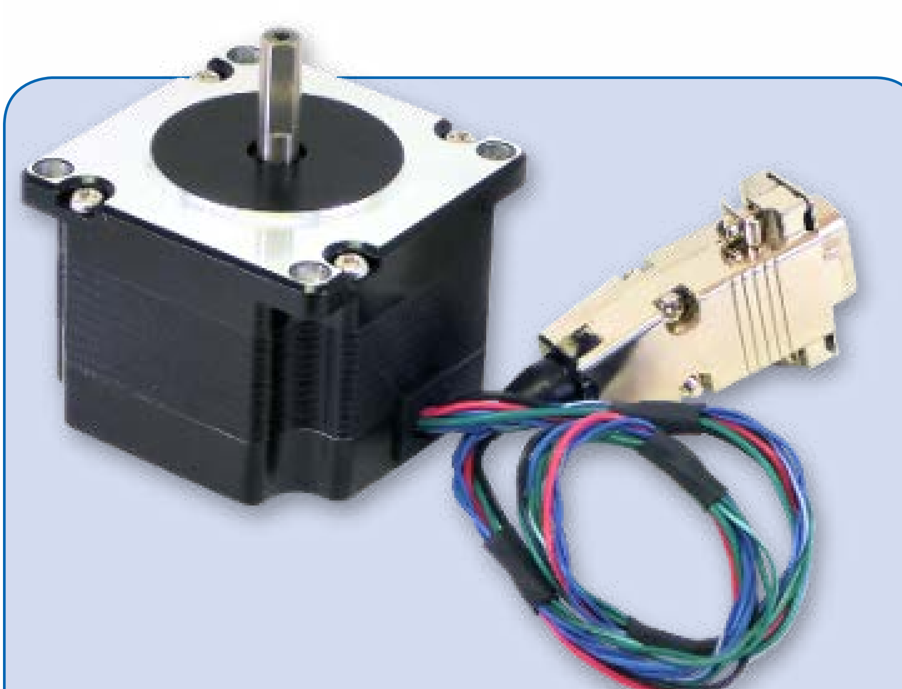

Mosolver Datasheet

NEMA 23 QCI-MV23L-1

Continuous Torque: 40 oz-in
Maximum Current: 2.5A
Maximum Speed: 3,000 RPM
Feedback Resolution: 32,000 counts/rev

General Motor Specifications MV23L - 1

Maximum Speed (RPM)	3,000
48v Optimal Speed (RPM)	2,000
Torque (oz - in/Nm) at Optimal Speed	22 0.15
Continuous Stall Torque oz - in/Nm	40 0.28
Peak Power (Mech. Watts)	44
Rotor Inertia oz – in^2 Kg - m^2	.74 1.35e – 5
Weight pounds/Kg	1.05 0.48
Maximum Driver Input Current (Amps - DC)	2.5
Shaft Diameter in/mm	0.25/6.25
Maximum Axial Force (lbs)	13
Maximum Radial Force (lbs) 0.55" from mounting face	15

Environmental Specifications

Operational Temperature
–10°C to +80°C

Storage Temperature
–40°C to +85°C

Humidity
Continuous specification is 95% RH non-condensing.

Shock
Limitation is approximately 50g/11ms.

IP Rating - Standard
IP50

MORE MOSOLVER: AT-A-GLANCE

There is solid basis to QuickSilver's claims regarding what new—if anything—the Mosolver brings to the dance. The motion control system was awarded an over-all Honorable Mention at the 2014 Create the Future Design Contest, an annual engineering + techno-design + entrepreneurial bazaar started in 2002 by the publishers of *NASA Tech Briefs magazine* "to help stimulate and reward engineering innovation." The contest's principal sponsors are simulation software company COMSOL and semiconductor and electronic component distributor Mouser Electronics, along with Analog Devices, a data conversion and signal conditioning technology company, a supporting sponsor.

—J. Mc Guinn

Applications

- Medical Devices
- Solar Panels
- Industrial
- Manufacturing
- Harsh Environments
- Robotics
- Food Processing
- Textiles
- Automation
- Packaging

Mosolver Benefits:

- Encoder eliminated
- Cost-effective
- Energy-efficient
- Quiet operation
- Physically robust
- Compact; less real estate required
- Precision

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step motors
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Diametral Pitch Calculation

THE QUESTION

Circular pitch gives me the size of the teeth in my mind, but diametral pitch does not. What is the purpose of the diametral pitch concept? Does it merely avoid pi in calculation?

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Expert response provided by Dr. Alexander Kapelevich:

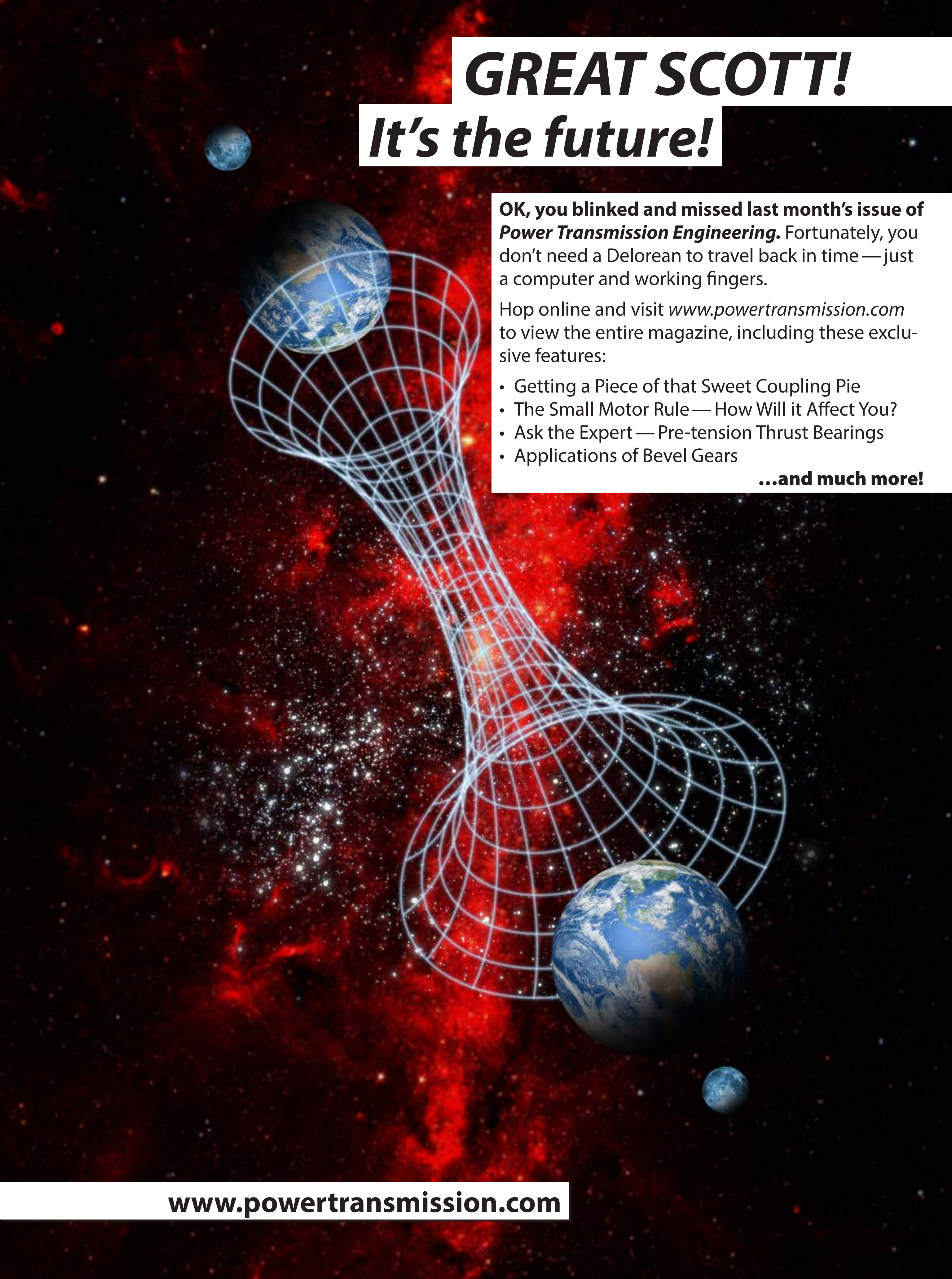
Unlike the circular pitch, diametral pitch (DP) is not a dimension. Historically it is defined as the number of teeth given per inch of a gear's pitch diameter, and its unit is 1/in.

However, DP *is* used for gear dimensions calculation.

For example: the circular pitch is π/DP; the standard tooth addendum is 1/DP; the standard tooth whole depth is 2.2/DP (or 2.25/DP); and the gear pitch diameter is n/DP (where *n* is a number of teeth). An important difference between the circular pitch and diametral pitch is that whole DP values are standardized (for example, DP = 4, 6, 8, 10, 12, 16, 20, 24, etc.) and used for the standard gear design. Standard cutting and measuring tooling (hobs, shaper cutters, gages, etc.) are off-the-shelf available.

While it is true that the circular pitch could be used for nonstandard gear data definition, it hardly makes sense. Such gear data could be confusing for a gear manufacturer and fabrication of such gears will require nonstandard tooling. As well, this approach does not provide any additional benefits for gear drive performance.

Alex Kapelevich has a Master Degree in Mechanical Engineering at Moscow Aviation Institute and a PhD in Mechanical Engineering at Moscow State Technical University, Moscow. He operates the consulting firm AKGears, LLC, a developer of trademarked Direct Gear Design methodology and software. He has 30 years of experience in custom gear drive development. His areas of expertise are gear transmission architecture, planetary systems, gear tooth profile optimization, gears with asymmetric teeth, and gear drive performance maximization. Alex is author of the book titled Direct Gear Design and many technical articles.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Mathematical Modeling for the Design of Spiroid, Helical, Spiral Bevel and Worm Gears

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Introduction

Spiroid and worm gears have superior advantages for high-torque and miniaturization applications. And for this reason they are particularly preferred in aerospace, robotic and medical applications. They are typically manufactured by hobbing technology, a process with a typical overall lead time of 4 to 14 weeks.

Besides the relatively lengthy lead time, and despite the fact that the tooth profile is defined through its pressure angles, 3-D drawings of the gears cannot be produced. This is due to the difficulty of capturing the entire curvature of the gear face from the outside to the inside diameter. Because of this difficulty, 3-D quality control and FEA (finite element analysis) under load are difficult and must be accomplished through classical analysis that incorporate pinion bending stresses, gear tooth shear stresses and compressive stresses between pinion and gear teeth. Due to some of these challenges, these gears have been limited to niche status in the industry.

This paper presents a novel work for Spiroid and worm gears that mathematically calculates the gear tooth profile in terms of the geometry of the cutting tool (hob) and machining set-up. Because of their similarity, the work is also expanded to spiral bevel gears. We have developed software to plot the gear tooth when the parameters of the geometry of the tool and machining set-up are entered. The gear tooth shape can then be altered and optimized by manipulating the input parameters until a desired tooth profile is produced. In effect, the result will be designing the hob and machining set-up for best gear tooth profile on the computer. The result is generated gear tooth data that is entered into CAD software to generate a true 3-D model of the gear. The tool path will also be generated from the same data for CNC machining.

This mathematical modeling allows for direct CNC machining, rather than hobbing, and may reduce the prototype lead time from weeks to hours. The pinion can be designed in a similar process, and its tooth can be graphed inside the gear's groove, showing the contact points and the clearance between the two surfaces. This novel work has already resulted in the invention of a new gear type combining Spiroid and worm gear in a single gear driven by the same pinion. This provides a significant increase in torque capability.

Mathematical modeling is presented in this paper as a tool design to reduce the lead time and cost for designing Spiroid, worm and spiral bevel gears. Mathematical modeling is based on mathematically calculating the 3-D gear tooth profile in terms of the cutting tool (hob) geometry, the machining set-up, and the gear size (inside and outside diameters). Software has been developed to allow designers to enter values of these parameters and observe the resulting 3-D gear tooth profile. The designer can observe the resulting tooth profile on the computer and adjust the input parameter values to obtain the desired profile. The software also generates the profile in numerical *xyz* points, which is necessary to produce 3-D drawings of the gear, conduct direct quality control, and perform FEA under load. It was also demonstrated that mathematical modeling can be a tool for gear innovation. It has already resulted in the invention of a new gear type in that a Spiroid/worm hybrid combines a double Spiroid gear and a worm gear in one. It more than doubled the gear torque capability, in comparison to a single Spiroid gear—with a minimal increase in size or weight.

Spiroid Gears

Oliver Saari invented Spiroid gears in 1954 while working for ITW, and the ITW Spiroid division was created. (*Author's Note: Spiroid is a registered trademark of ITW; the views expressed herein are those of the author alone and do not neces-*

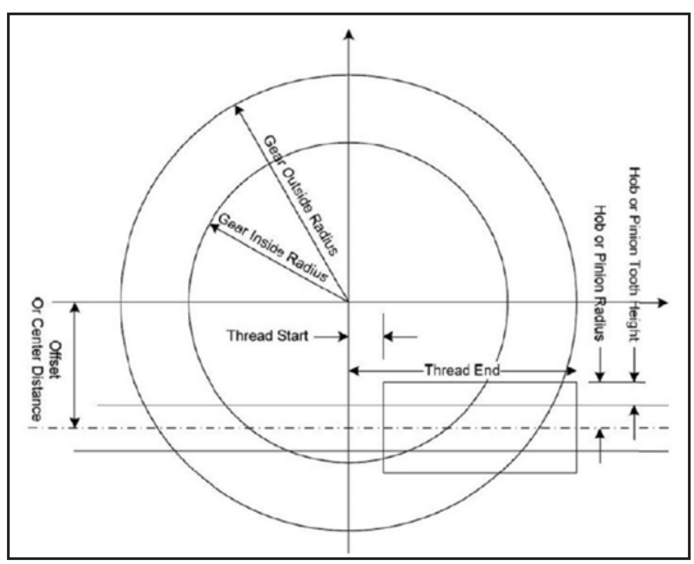

Figure 1 Spiroid gear/pinion assembly or gear/hob set-up.

Figure 2 The three types of Spiroid gear.

sarily reflect the views of ITW.) The gear set consists of a gear and a helical pinion assembled as shown in Figure 1. It is similar to spiral bevel gear set, except the pinion axis is moved a certain distance off the gear axis. This distance is called offset, or center distance. It makes it possible to hold the pinion shaft on both ends to increase leverage and stability. Spiroid gears are special gears with a wide range of RPM ratio, from as little as four to as many as several hundred; this facilitates single-stage designs that reduce size and increase efficiency. They can also have a high contact ratio, which makes them suitable for high-torque and low-noise requirements.

There are three Spiroid gear types (Fig. 2): flat-face; skewed angle with cylindrical pinion; and skewed angle with tapered pinion.

The pinion can be manufactured by any method for making helical gears, including grinding wheel, shaping, or CNC milling — but the grinding wheel is the dominant method. The gear is manufactured in a hobbing process, using a cutting tool (hob). The hob geometry is similar to the pinion geometry, except its tooth sides in the axial direction are straight (Fig. 3). It has a certain number of gashes in the direction perpendicular to the helix that are sharpened for cutting. During machining the gear and the hob will be rotating, with an RPM ratio equal to the ratio of their teeth numbers.

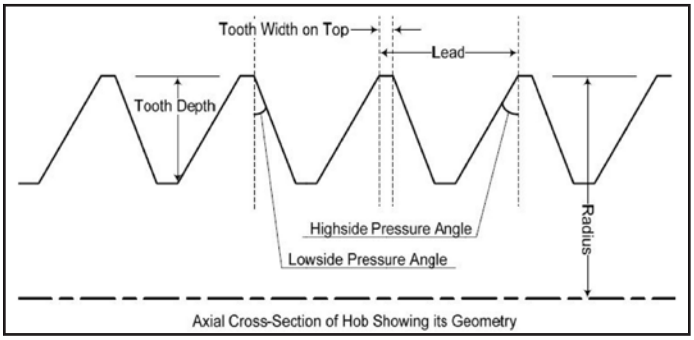

Figure 3 Axial cross-section of a cylindrical hob.

Figure 4 Cylindrical and tapered hobs.

A cylindrical hob and a tapered hob are shown in Figure 4. There are a total of nine basic geometry parameters; they are: outside diameter; tooth height; number of teeth (starts); lead (axial pitch); high-side pressure angle; low pressure angle; tooth width on top in axial direction; and the start and the end of its tooth along the shaft with respect to centerline. The taper angle is an additional parameter for the tapered hob. Other parameters that have to be taken into account when designing a hob are: gear inside radius; gear outside radius; gear number of teeth; and the center distance (off-set) between the gear axis and the hob axis. Also, in the skew angle case the skew angle is an additional parameter. This large number of parameters — 13–15 — makes it hard to predict the gear tooth shape and profile.

Designing a hob for a new gear set has the potential to be a trial-and-error process. The design objective is to have a gear tooth that is free of gouging on its sides or clipping of its top. It also should have the desired width and a balance of pressure angles, and land area at the root and at the top. After selecting the geometry the hob is manufactured by an outside machine shop with a lead time of eight to ten weeks. The gear is then cut in-house. It can be difficult during this process to realize the precise design objectives of the desired gear tooth geometry.

Initially the hob is placed over the gear blank (Fig. 1). During machining the gear and the hob will be rotating. Machining will be complete when the hob penetrates a distance equal to the tooth height. In the case of the skew angle gear, the hob axis will have an incline angle over the gear plane. After cutting a certain number of gears, the hob becomes dull; the hob will be sharpened and its diameter will be reduced by a few thousands of an inch each time. This will have some effect on the gear tooth profile and its contact with the pinion. Many manufactured gears are smaller than five inches in diameter.

Since the full-length tooth profile of the manufactured gear remains unknown, true 3-D models of the gear cannot be produced, and it can be difficult to determine FEA and tooth strength analysis under load outside of classical analytical methods.

Mathematical Modeling Solution

Gearometry has developed math equations and processes to accurately calculate the gear 3-D, *xyz*, tooth profile in terms of the hob geometry, the machining set-up, and the gear parameters. We also developed software in the form of *Excel* spreadsheets where values of the parameters are entered and the 3-D profile of the gear tooth or gear groove is produced in the form of sketches (Figs. 5–7) and in the form of *xyz* point data organized data arrays.

For gear tooth profile representation, the selection of an appropriate coordinate system is very important.

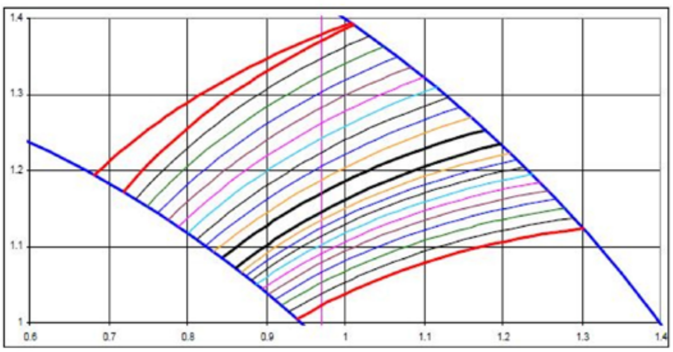

Figure 5 3-D sketch of an optimized, flat-face Spiroid gear groove — high and low sides.

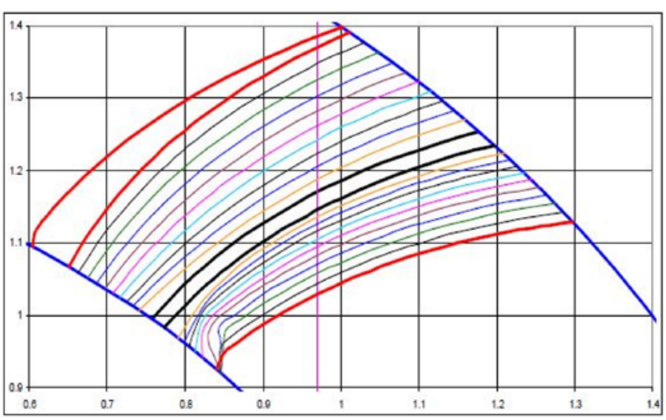

Figure 6 Gear groove profile for a gear inside radius of 1.25".

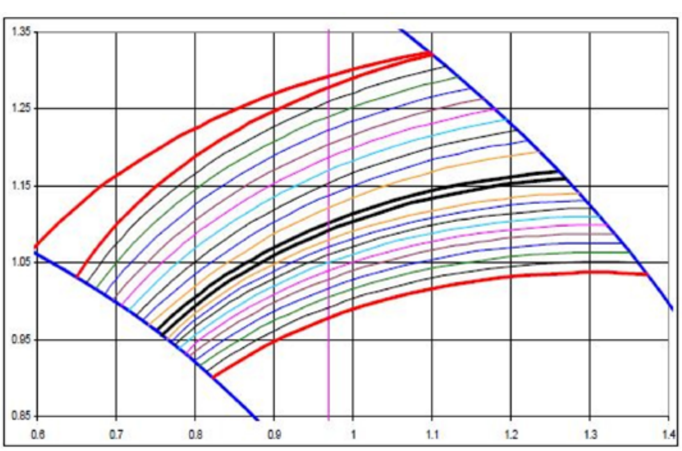

Figure 7 Gear groove profile with enhanced design using *Gearometry* software.

The z axis of our coordinate system is always aligned with the gear axis. For the flat-face gear, $z=0$ is at the root of the tooth. The x axis is parallel to the center distance and the y axis is parallel to the pinion/hob axis. In the case of the flat-face gear, the *xyz* points of the tooth profile are arranged in 11 horizontal lines for the high side (parallel to the x-y plane), and 11 lines for the low side. Each line has 21 points. Therefore, each side has 231 points. The lowest line of each side is at the root where $z=0$. The highest line is at the top where $z=$tooth height. In Figures 5–7 the 11 lines in the southeast corner represent the low side, and the 11 lines in the northwest corner represent the high side. The two heavy black lines in the center are at the root, and the two heavy red lines are located at the tooth top. Therefore the sketches in these figures represent the gear groove where a tooth of a hob or pinion fits. The last heavy red line in the northwest corner is the top of the low-side of the next groove. The area between this line and the previous line is the land area of the tooth top. Each pair of lines of the same color represents a horizontal cross-section, in the x-y plane, of the gear groove. The cross-sections are equally spaced vertically. The separation distance between them is one-tenth of the tooth height.

As in conventional hob design, at the outset one enters values of the design parameters. But in our software the designer will be able to manipulate the entered values in the spread sheet and observe how the tooth profile will change until the desired design is obtained. This optimization process can typically take an hour or two, depending on individual cases and user satisfaction. Ultimately, this process will result in the designing of the hob geometry and determining of the optimum machining set-up parameters to produce the perfect gear.

For illustration we will explain the process of designing a set of Spiroid gear and its pinion. We will consider the flat-face gear set of Figure 2. This set was designed and manufactured in the conventional method. The values of its 13 design parameters are listed in Table 1. They were estimated using reverse engineering of the hob geometry from the available pinion geometry. We used our software to sketch the gear tooth profile with the parameters in Table 1 (also shown in Fig. 5). At this point we ask the question whether the gear tooth profile is truly optimized — or is it merely an example of an "acceptable defect-free" profile. When designing a gear set, the objective is to maximize the torque capability and contact ratio without increasing the set size. This means extending the gear inside radius as much as possible toward the center. Using our software, we

Table 1 Parameters and their optimized values of the flat-face gear design

Hob thread start	0.250"
Hob thread end	1.750"
Hob radius	0.410"
Hob tooth height	0.190"
Hob high pressure angle	40°
Hob low pressure angle	25°
Hob tooth width on top	0.025"
Hob number of teeth	4
Hob lead	1.500"
Gear inside radius	1.375"
Gear outside radius	1.720"
Gear number of teeth	27
Center distance	0.970"

extended the gear inside radius down to 1.25", from 1.375", while keeping all other parameters values the same. Figure 6 shows that the tooth profile begins displaying defects when the inside radius is below 1.30". However, the tooth will be defect free for inside radius of 1.31". Extending the gear inside radius down to 1.31" can significantly increase the contact ratio, increase the tooth strength and increase the gear set torque capability.

The drawing in Figure 7 shows that a perfect gear tooth profile can be obtained for an inside radius of 1.22" when the hob lead is reduced from 1.50" to 1.32"; the high-pressure angle is reduced from 40° to 35°; the low-pressure angle is reduced from 25° to 23°; and the tooth width on the top is reduced from 0.025" to 0.01". The gear outside radius and center distance, as well as the tooth height and the RPM ratio, are kept the same. With this new design the contact ratio is increased from 1 to 2, while the gear tooth is significantly increased in both circumferential and radial width.

However, this improvement is at the expense of the pinion tooth.

Using a narrower hob tooth, the pinion tooth width at tooth mid-height will be reduced by about 17.8%; but now we have two teeth in contact at any time rather than only one. Therefore the load-per-pinion-tooth is reduced by 50%. Depending on the application, the designer has to make decisions on whether to enhance the gear tooth, the pinion tooth, the contact ratio or some other component of the gear set. This software provides a wide range of options to the designer, with accurate information for decision making. In fact, additional good options could be further explored by manipulating the tooth height value. Smaller heights would allow wider groove or larger pressure angles. Both would lead to a wider pinion tooth.

In addition, the *xyz* data points of the curves in (Fig. 7) can be transported into engineering CAD software to generate the true 3-D model of the gear. One model is shown (Fig. 8) for a 35-tooth gear.

Such models can now be used to perform accurate FEA under load that will facilitate design decisions. Figure 9 shows FEA results on the Spiroid/worm hybrid gear (Fig. 21) where the 3-D model is generated from *xyz* data points.

The software also superimposes the hob tooth inside the machined gear groove. Figure 10 shows the gear groove of Figure 7 with the hob tooth superimposed inside it. The figure shows that the high side of the hob tooth is in contact with the gear tooth, along a line from a point at the bottom of the groove exactly below the hob axis, to a point at its rim and to the right of the hob axis. On the low side, the contact line also starts at a point at the bottom of the groove to a point at its rim, to the left of the hob axis.

The hob axis lies at $x=0.97$". The hob and the gear can be incrementally rotated by a software command to show the tooth inside the groove in various positions. This enhances visualization of the pinion tooth and calculation of the contact ratio. In this case the contact ratio is 2; the conventionally designed gear has a contact ratio of 1.

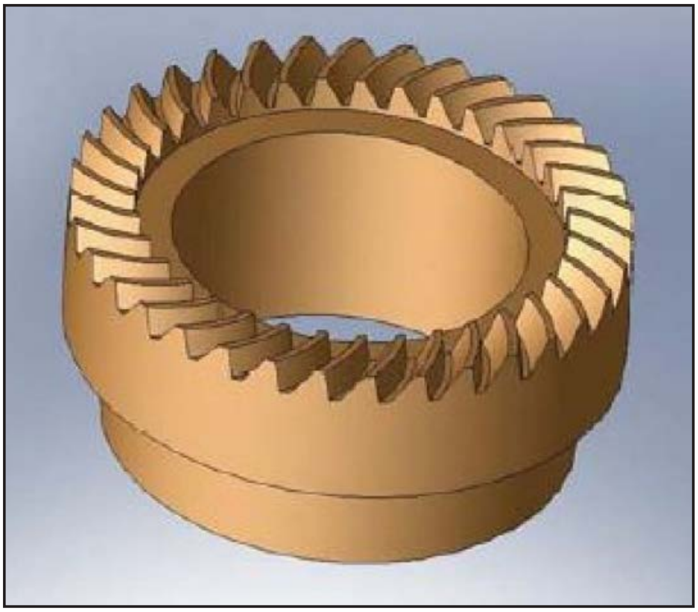

Figure 8 Example of true 3-D model of flat-face Spiroid gear from *Gearometry* software data.

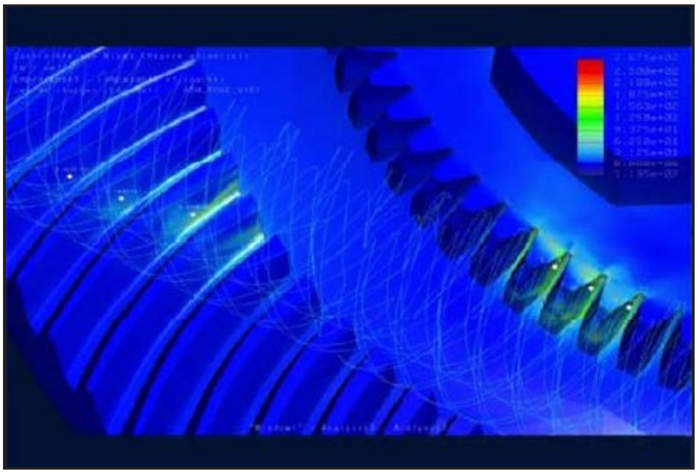

Figure 9 Example of FEA under load from *Gearometry* software data.

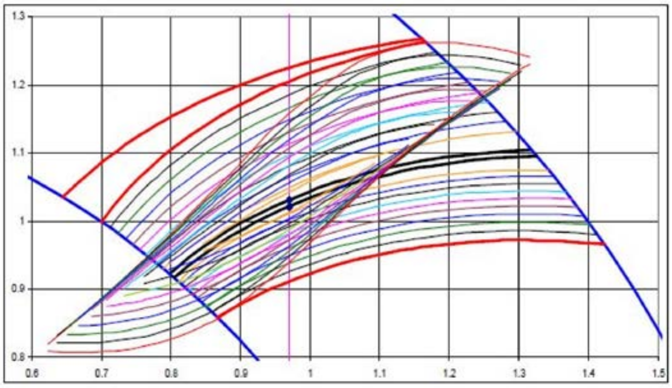

Figure 10 *Gearometry*-enhanced design gear groove with hob tooth superimposed.

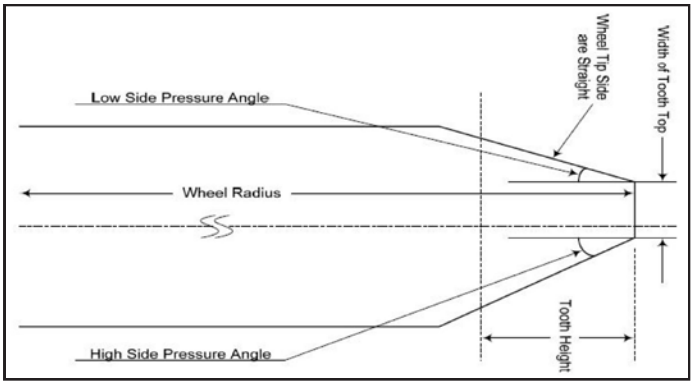

Figure 11 Grinding wheel geometry for Spiroid gear pinion.

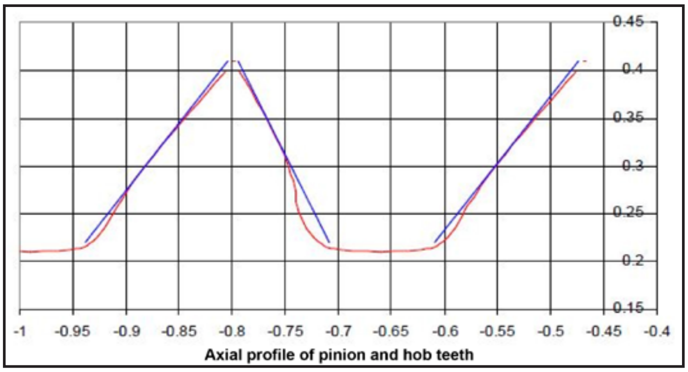

Figure 12 Pinion tooth (in red) inside the hob tooth (in blue).

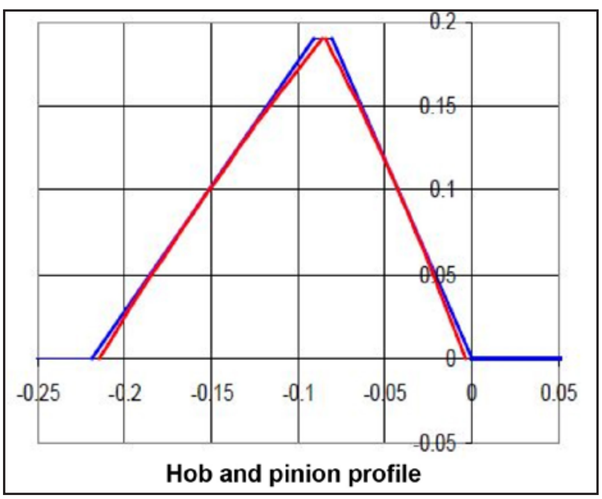

Figure 13 Pinion tooth (in red) with circular sides contained inside the hob tooth (in blue).

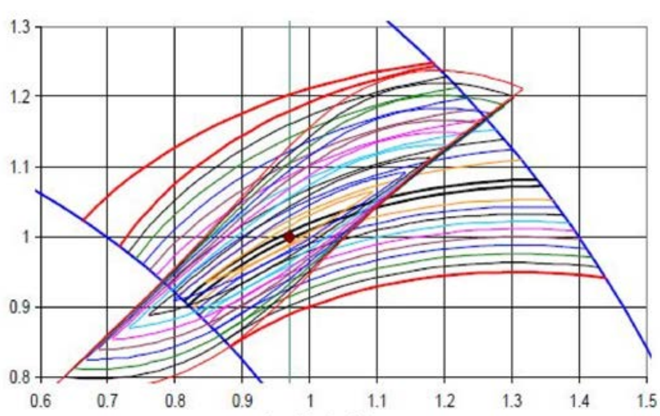

Figure 14 Circular sides pinion tooth superimposed inside the gear groove.

Spiroid Gear Pinion — A Helical Gear

Essentially, the pinion of the Spiroid gear is a helical gear. Thus our mathematical modeling of the pinion will be presented as modeling of the helical gear. In this paper we will design a pinion for the Spiroid flat-face gear. We will discuss two cases; the first is when the pinion is manufactured by a grinding wheel; the other is for using CNC milling, or shaping, by a tool. In any case, the objective is to design a pinion tooth that will be contained inside the hob tooth and touches the hob tooth in one point at about tooth mid-height (Figs. 12–13). In this case, its contact with the gear tooth under no load will be a single point on each side, at about a tooth mid-height.

Grinding Wheel

A typical grinding wheel is shown in Figure 11. Its geometry parameters are: the wheel radius; the high-side pressure angle; the low-side pressure angle; the tooth height; and the top tooth width. The other pinion design parameters are: the pinion outside radius; its number of teeth (starts); its lead; and the wheel tilt angle. We therefore have a total of nine parameters. Again, the profile of the pinion tooth is a function of these parameters. Figure 12 shows a *Gearometry*-optimized pinion tooth (in red) inside the hob tooth (in blue). The pinion and hob will have the same lead if they have the same number of teeth. There are cases where the number of teeth is different, in which their lead would change.

Another way of making a pinion is by CNC milling or shaping. This would only require the generation of the geometry of the pinion tooth. The data could then be entered in a CNC

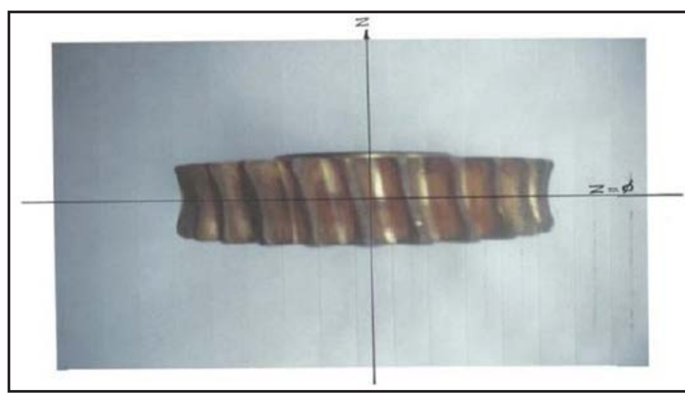

Figure 15 Single-enveloping worm gear.

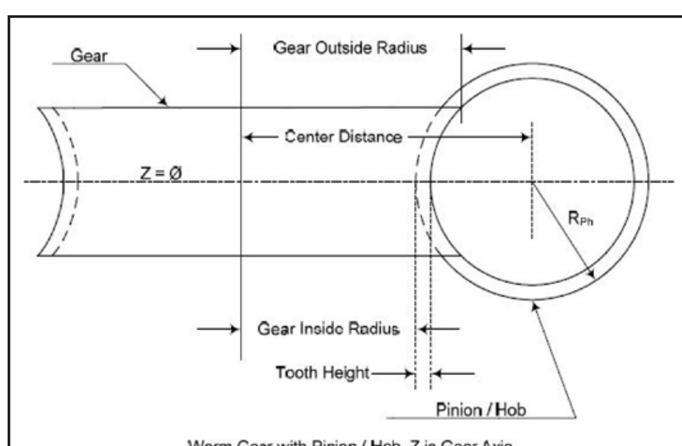

Figure 16 Worm gear/pinion assembly or gear/hob set-up.

machine for milling or used to generate the shaping-tool geometry. Figure 13 shows an axial cross-section of a pinion tooth (in red) contained inside the hob tooth (in blue). The red profile on each side of the tooth is a circular section that touches the blue profile at exactly mid-tooth height. *Gearometry* automatically produces the profile *xyz* point data and the sketch in (Fig. 13) to allow the designer to select the circular profile radius to control the clearance when moving away from the contact point.

Figure 14 shows a sketch of the pinion tooth of Figure 13 superimposed inside the gear groove of Figure 7, under no-load condition. They contact each other in a single point at mid-tooth height along the pink lines as shown. In fact, the software simplifies the calculation of the clearance between the gear groove and the pinion tooth when moving away from the contact point in vertical and horizontal directions. This aids in calculating the spreading of the teeth contact under load for a given material, and in conducting tooth strength and FE analysis on the gear set.

Worm gear. This paper discusses only the single enveloping worm gear. The pinion, or worm, is a cylindrical helical gear — similar to the flat-face Spiroid gear pinion. Like the flat-face Spiroid gear, the worm gear is typically manufactured in by the hobbing process, beginning with a cylindrical gear blank.

During machining the hob — with helical gear shape and sharp gashes — is placed over the cylinder surface. The hob axis is oriented in a perpendicular direction to the cylinder axis. The gear and the hob will be rotating with an RPM ratio equal to the ratio of the number of their teeth. Machining will be complete when the hob is advanced by a tooth height toward the gear center. The resulting gear will look like a helical spur gear with raised edges (Fig. 15); the raised edges will envelope the pinion for increased tooth contact.

The hob geometry and parameters are the same as that of a Spiroid gear hob. Therefore the total number of design parameters is 13. The hob geometry parameters are nine, and the other four are the gear inside radius; gear outside radius; gear number of teeth; and center distance (Fig. 16). (*Note that selecting non-optimized values for the parameters can result in either undercut of the tooth sides or clipping of the gear tooth on top*.)

Table 2 lists all 13 design parameters, along with their optimized values for a worm gear. In this case, the z axis of the coordinate system is aligned with the gear axis. Cross-sections of the gear tooth profile are presented in x-y planes perpendicular to the gear axis at different z values. Figure 17 shows a tooth cross-section in heavy pink at $z = 0$ mm, at exactly the gear center in axial direction where the tooth root is at the gear inside radius (in heavy red). For $z = 3$ mm, the tooth cross-section is shown in Figure 18.

Table 2 Design parameters with optimized values of a worm gear

Hob thread start	-10.0 mm
Hob thread end	10 mm
Hob radius	6 mm
Hob tooth height	2 mm
Hob high pressure angle	20°
Hob low pressure angle	20°
Hob tooth width on top	0.6 mm
Hob number of teeth	4
Hob lead	10 mm
Gear inside radius	28 mm
Gear outside radius	31 mm
Gear number of teeth	76
Center distance	34 mm
Tilt angle	10°

The software includes a new parameter in which the pinion axis is rotated around the centerline by a certain angle, i.e. — the "tilt angle." This adds a degree of versatility to the use of the worm gear, as will be demonstrated. The tooth has good proportions — from one end of the gear at $z = -3.5$ mm — to the other end at $z = 3.5$ mm. This is because the parameters were optimized. Figure 19 shows what the cross-section of the tooth would look like at $z = 0$ mm and when the hob lead is changed from 10 mm to 11 mm. It has steep sides

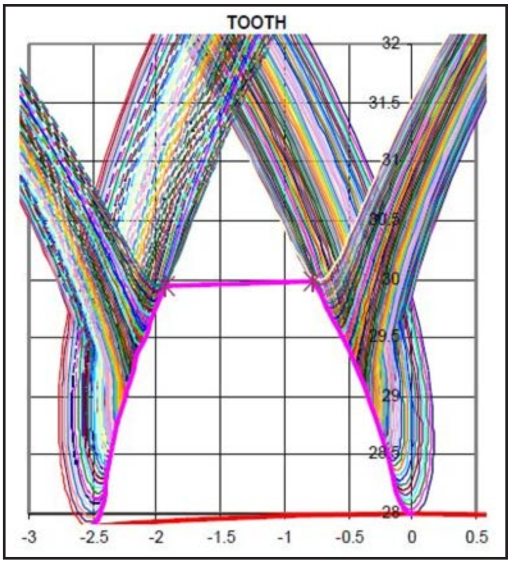

Figure 17 Axial cross-section of the gear tooth at $z = 0$ mm.

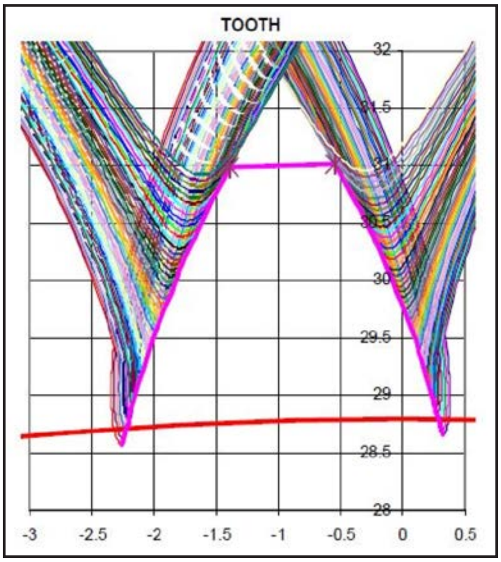

Figure 18 Axial cross-section of the gear tooth at $z = 3$ mm.

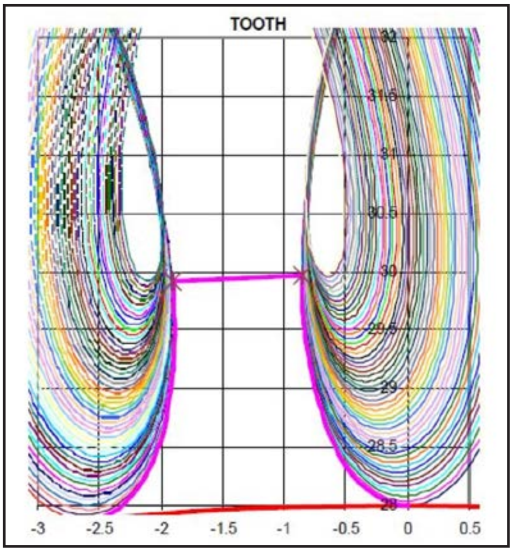

Figure 19 Axial cross-section of the gear tooth: at $z = 0$ mm; lead = 11 mm.

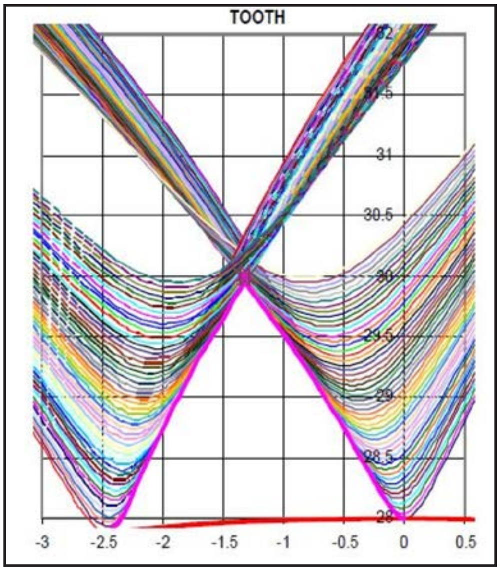

Figure 20 **Axial cross-section of the gear tooth: at $z=0$ mm; lead = 9 mm.**

and some undercut—especially on the tooth left side. For $z=0$ mm and lead = 9 mm, a cross-section of the tooth is shown (Fig. 20) indicating tooth clipping. With this software the user can design the gear tooth to any desired shape by simply manipulating the values of the design parameters and observing how the shape changes. The pinion of the worm gear will be designed in the same manner as the pinion of the Spiroid gear discussed above.

Spiroid Worm Hybrid Gears

For aerospace, robotic, and medical applications there is an increasing demand for smaller and lighter gear sets with higher torque capability. In gear set design this translates to a need for increased tooth contact and contact ratio. As observed previously, Spiroid and worm gears are suitable for these applications. The same pinion that drives a Spiroid gear can have more teeth driving another Spiroid gear on the same shaft, on the opposite side of the centerline. This immediately doubles the torque capability, yet with a minimal increase in weight and size. The Spiroid gear must be of the skew angle-type so that the added teeth will not interfere with the first gear teeth on the other side of the centerline. This combination is called a "double-Spiroid gear." We also determined that there is an unoccupied space between the two Spiroid gears in their axial direction, and between their shaft and the pinion, where a worm gear can fit nicely. This worm gear can be driven with additional teeth on the pinion shaft between the two sets of teeth driving the double Spiroid gears. Adding the worm gear will add more torque capability, but with a negligible weight increase and no size increase. The pinion will have a tilt angle with the worm gear that is the same as the skew angle it makes with Spiroid gears. This is why the tilt angle was added as a new design parameter in the abovementioned worm gear design (Fig. 21).

Both hob and pinion for the Spiroid and worm gear were designed independently. Excepting the outside radius, the number of teeth (starts) and tooth depth were retained. In this special case, with an RPM ratio of 19, to make a good Spiroid gear its hob lead had to be 13.5 mm; and to get a good worm gear, the lead had to be 10 mm. However, in other cases with exactly the same size but an RPM ratio of 38, it was possible to have an exact geometry of hob and pinion for the Spiroid and the worm gear. This makes gear and pinion manufacturing—and assembly—much easier.

This set of gear and pinion was manufactured by a milling CNC machine; it was possible because the *xyz* data points of the tooth profile and the tool path were available. The design optimization was done in a few hours and machining was completed the next day. A hob was not needed, thus saving its cost—and a great deal of time.

For the Spiroid and worm gear, our software is a tool that, in a matter of a few hours, designs the hob and the gear simultaneously. It also provides accurate calculation of the gear tooth profile in terms of the design parameters in sketches and *xyz* data points. The points can be used to produce a true 3-D drawing of the gear. Knowing the gear and pinion tooth profiles in 3-D sketches and data points enables gear CNC milling or shaping, besides hob manufacturing. Contact points between pinion and gear teeth become known. Clearance between the two teeth surfaces when moving away from the contact points will be known. Calculation of teeth contact spread under load for given materials become possible. Accurate FE and load analysis will also be possible (Fig. 22) that was performed on the hybrid gear. Quality control can also be accurately and easily conducted directly on the gear and the pinion.

Mathematical modeling is a new and revolutionary process for improving the design and manufacturing of all gear

Figure 21 **New "hybrid" gear made of a double-Spiroid gear, and worm gear driven by same pinion.**

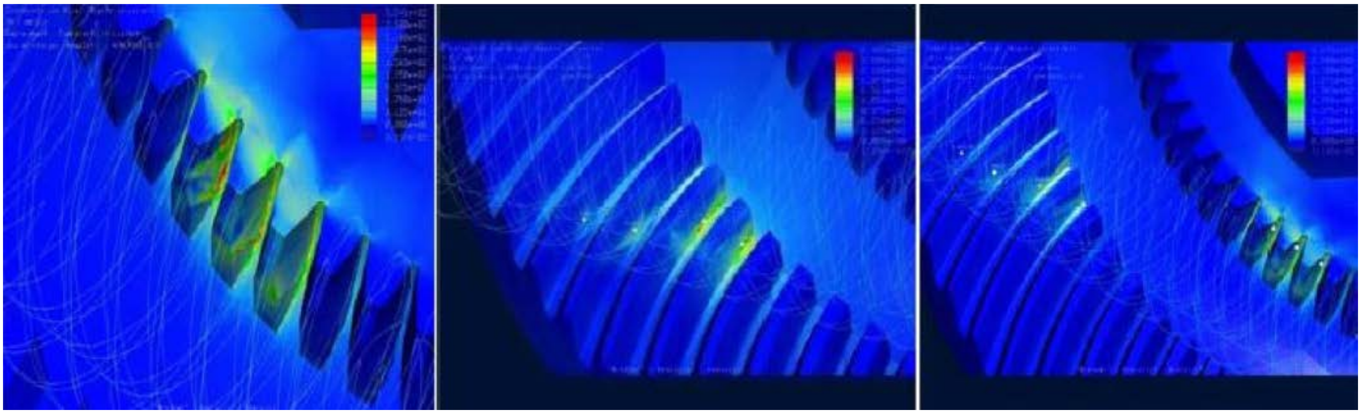

Figure 22 FEA outputs on the hybrid gear.

types—not just the Spiroid and worm gears. It is based on mathematically generating accurate gear tooth profile in terms of the geometry of the tool and the machining process that will be used as input parameters in software. It provides the means for optimization of the parameters in order to obtain the desired gear. The results will be designing the tool and determining the machining set-up to achieve the objectives. The generated gear tooth profile in sketches and *xyz* points can be used to generate the 3-D model of the gear. This opens the door for machining by CNC milling or shaping. With the right approach and diligence it is possible for Spiroid and worm gears to be included in the AGMA standards. There are many improvements and innovations needed in the gear industry, including the spur gear, where mathematical modeling can be the answer.

Spiral Bevel Gears

In this paper we will consider only the spiral bevel tapered hobbing manufacturing method. The gear/pinion assembly or gear/hob set-up is shown in Figure 23; the figure shows the case with no inclination angle.

With an inclination angle, the hob or pinion axis will have a tilt from the horizontal position. The hob geometry is similar to that of a Spiroid tapered hob (Fig. 24) that represents an axial cross-section with straight tooth sides. Therefore the design parameters of the spiral bevel gear are 14 (Table 3).

Table 3 Design parameters of spiral bevel gears	
Hob/pinion number of teeth (starts)	9
Hob/pinion radius at gear OD	5.0"
Hob/pinion axial length	2.0"
Tooth height	0.5"
Hob/pinion lead	8.0"
Hob high side pressure angle	30.0°
Hob low side pressure angle	20.0°
Hob tooth width on top	0.18"
Hob/pinion taper angle	30.0°
Hob pinion inclination angle	0.0°
Gear inside radius	3.0"
Gear outside radius	5.0"
Gear number of teeth	9

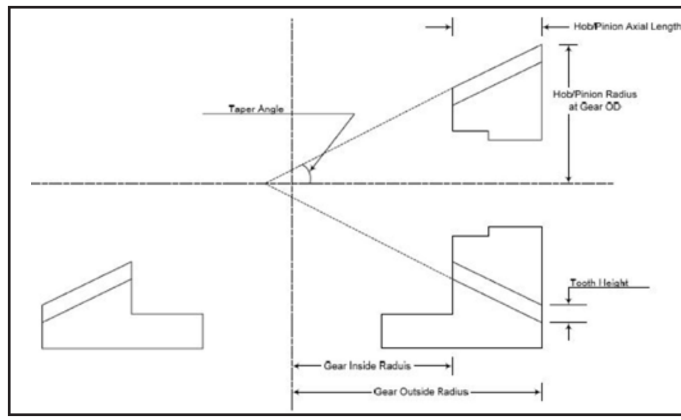

Figure 23 Spiral bevel gear set-up with hob or pinion.

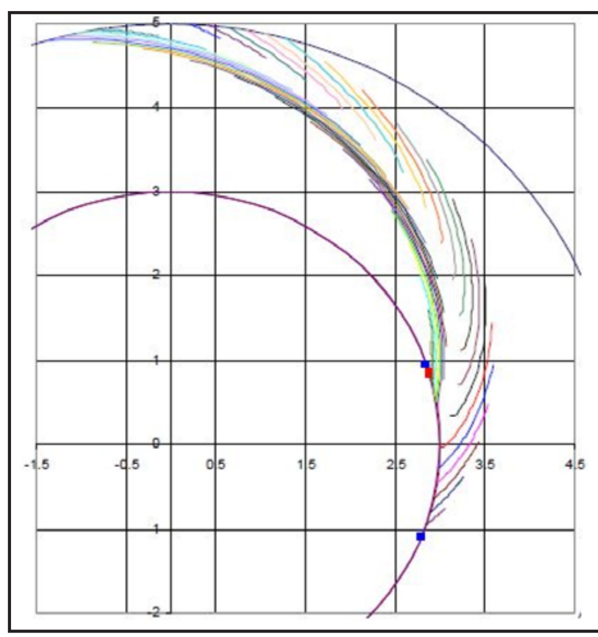

Figure 24 Axial cross-section of spiral bevel gear hob showing its geometry and parameters.

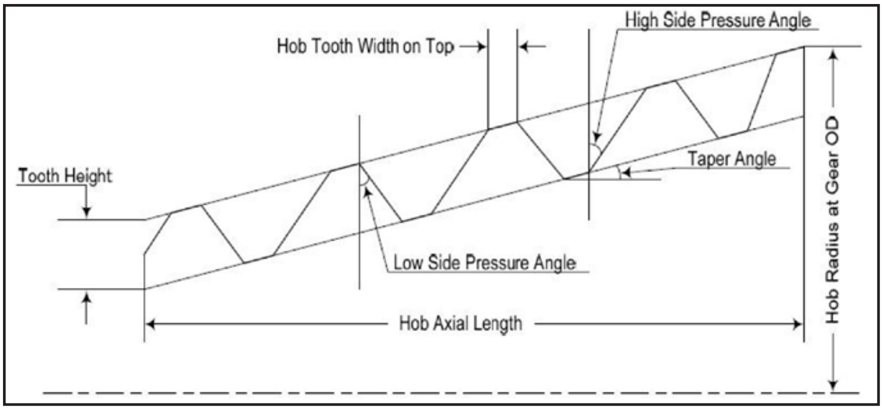

Figure 25 Spiral bevel gear groove with high-side and low-side profiles made of curves in planes perpendicular to the gear axis at different heights.

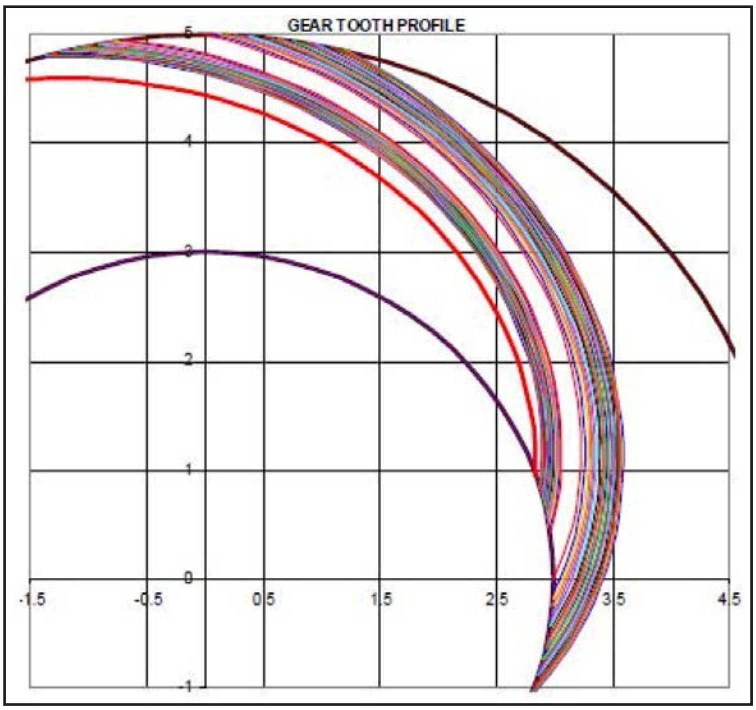

Figure 26 Spiral bevel gear groove with high-side and low-side profiles made of curves in conical surfaces parallel to root or top of the gear.

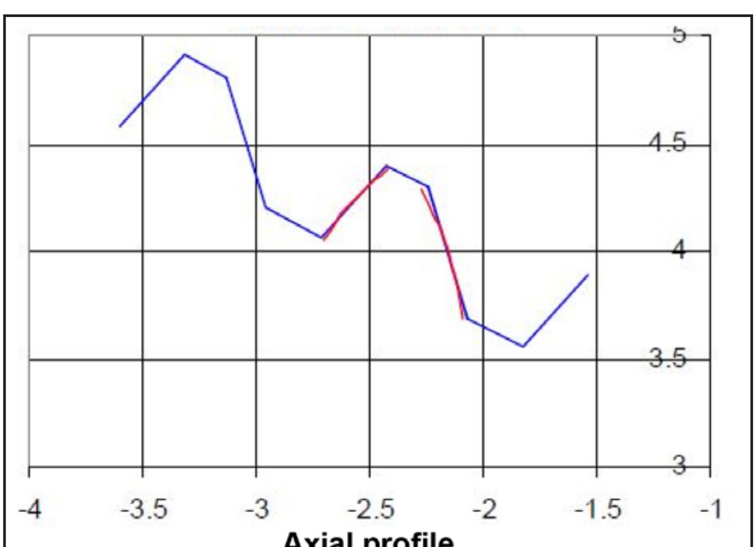

Figure 27 Axial cross-section of the spiral bevel gear hob and pinion.

For illustration we considered designing a set that has an RPM ratio of 1.

The outside and inside gear radii were selected to be 5" and 3", respectively. The outside radius of the hob at gear OD was also selected to be 5". The values of the other parameters were manipulated until a good design was obtained (Table 3).

For gear tooth profile representation we selected a coordinate system where the *z* axis is aligned with the gear axis. As for the flat-face gear, the spiral bevel gear groove will be sketched (Fig. 25).

The curves on the right represent the profile of the high side, and the curves on the left represent the profile of the low side. Each side is made of 21 horizontal curves in planes perpendicular to the gear axis.

Another representation of the same groove is shown (Fig. 26). Here, too, the 21 curves on the right form the high side and the 21 curves on the left form the low side. But they are arranged in pairs, where the pair in the center is at the root. The other pairs are located at equal increments of the z height above the root; the heavy red line at extreme left is the top of the next groove.

As each curve is made of 21 points, each side has 441 points that can be used to generate the 3-D model of the gear groove, using engineering CAD software. The 3-D model of the gear can be generated by circumferentially copying the groove nine times.

Figure 27 shows a sketch (in blue) of the axial cross-section of the hob. The sketch (in red) is an axial cross-section of the pinion tooth to mate with the gear. It is contained inside the hob tooth and, in this case, it is designed with circular sides that touch the hob sides exactly at tooth mid-height. As a result, under no-load, the manufactured pinion tooth will touch the gear tooth in one point at tooth mid-height.

The design method of the spiral bevel gear presented in this paper may be unique to *Gearometry,* based on using tapered hobbing for manufacturing; there are other established design and manufacturing methods in the industry. The details of design and manufacturing set-up are kept as proprietary company secrets by the manufacturers of the machines and the tools that are used to cut the gears.

Conclusion

Mathematical modeling is a new and unique tool for gear design that allows designers to achieve optimal gear sets. It reduces the time and cost of designing the hob and manufacturing the gear.

But more importantly, for these gears, it numerically and accurately generates the gear tooth profile.

This calculation now makes it possible to produce a true 3-D drawing of the gear; accurately determine the contact point and clearance between the gear/pinion teeth surfaces; conduct FE analysis of the gear set under load; and many other useful analyses. It opens the door for CNC machining, in addition to hobbing, teaching, and the research and development of standards for these gear types. Mathematical modeling can improve the already-valuable Spiroid gear form and make it acceptable in large-size gear markets such as wind energy and large machinery at affordable cost. It also opens the door for many innovations in gear technology. The new Spiroid/worm hybrid gear is one example of such innovations. (*Author's Note*: *See U.S. Patent Application Publication No. 2012/0000305, "Hybrid Enveloping Spiroid and Worm Gear," published January 5, 2012, as to which the author is a co-inventor.*) With mathematical modeling, gear technology innovation is limited only by the imagination of the gear designer and the gear manufacturing machine maker. **PTE**

Dr. Ghaffar Kazkaz is currently a consultant in gear technology specializing in gear geometry and design. He started his consulting company, Gearometry, after leaving ITW Technology in 2011. He has a bachelor in pure math and physics from the Syrian University of Damascus, a doctorate in physics from the University of Grenoble, France and a PhD in electrical engineering from the University of Illinois in Chicago. Kazkaz started gear mathematical modeling at ITW to help ITW's Spiroid division in Spiroid and worm gear design. After retirement he made significant improvements and extended mathematical modeling to spiral bevel and spur gears. (Mathematical modeling is based on calculating the gear tooth profile in terms of the cutting or grinding tool geometry and machining set-up parameters. It also calculates the gear tooth profile in terms of the mating gear tooth profile and assembly conditions.) He has developed software formatted in Excel sheet programs for gear design and is looking to work with a team to market the software. Kazkaz is also interested in designing gears and developing custom software in gear design and manufacturing for individual companies.

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How to Get the Most Realistic Efficiency Calculation for Gearboxes

J. Langhart

Introduction

In recent years the estimation of gearbox power loss is attracting more interest — especially in the wind turbine and automotive gearbox industry — but also in industrial gearboxes where heat dissipation is a consideration as well. As new transmissions concepts are being researched to meet both ecological and commercial demands, a quick and reliable estimation of overall efficiency becomes inevitable in designing the optimal gearbox.

Having a closer look at the efficiency calculations for gearboxes, the calculations are relatively well-defined for the gear mesh loss, and plunge or churning losses in the standards, such as ISO 14179. However, the efficiency calculations for the many other machine elements, i.e. — clutches or synchronizers — are relatively new concerns with no as yet established standards, although there exist several pioneering works. The research is still ongoing, and thus far the power losses can be approximated most exactly only by using data maps from measured power losses from a reference gearbox on test rigs. In short, the designer is faced with this task: achieving the most realistic efficiency calculation for gearboxes.

These demands present a major challenge for the software supplier. In response KISSsoft implemented a special template in *KISSsys* to automate the efficiency calculation and thermal rating of a whole gearbox — including gears, shafts, bearings, seal, discs, synchronizers and other machine elements. The template includes two parts; 1) the calculation of the power losses, and 2) the calculation of heat dissipation — both using calculations from standards set by ISO, AGMA, VDI, as well as from research works produced by academia and industry.

Along with the standards, other data sources are needed. So for gear losses the local contact analysis delivers very detailed results, including the microgeometry; as well, the interpolation using measured power loss maps is available. The template has been proven by the comparison with the actual measurement data from our customers in close cooperation with us.

In this paper the overview of the standards for the power loss calculation is shown and applications for an industrial and automotive gearbox are explained with the achieved results.

Application of Thermal Rating for Industrial Gearboxes

The development and production of industrial gear units are subject to considerable general cost pressure that requires gear manufacturers to design their products for maximum efficiency — especially from an economic viewpoint.

The factors involved in the conservation of resources and environmental protection are playing an increasingly significant role in the manufacture and operation of gear units. Interest in monitoring Total Life Cycle Costs is growing among machinery and industrial operators, and is a factor that must also be addressed by gearbox manufacturers. For this reason, gearbox manufacturers now need to conduct thorough investigations into all the relevant influencing factors, such as:

- Production processes (manufacturing costs)
- Gear unit usage (operating costs)
- Service life (operating costs)
- Maintenance (operating costs)

The need to reduce energy losses, and therefore minimize operating costs, is gaining in importance as one of the targets designers must meet when designing gear systems. Energy losses typically manifest themselves as increased operating temperatures that often limit the performance of fast-running, compact gear units. Gear units with low levels of energy loss are not only less expensive to run, but also have less impact on the environment. In addition, customers often ask for exact details about surface temperatures and efficiency as the specific values for a gear unit.

In order to meet the requirements described above, it must be possible, even at the design stage, to specify product characteristics accurately and effectively. The standards developed in the field of mechanical strength calculations provide a broad basis of data which the *KISSsoft* calculation platform, with the added functionalities provided in *KISSsys*, has implemented in a system that is not only comprehensive, but also very user-friendly.

For these reasons KISSsoft AG and ZAE-AntriebsSysteme GmbH & Co KG have been working together to develop a generally applicable calculation tool — within the *KISSsys* system — that can be implemented to determine the power loss and temperatures present in a gear transmission unit.

Calculation of power loss and thermal rating. ISO Technical Report 14179 (Ref. 3) can be used as the basis for a general calculation for determining temperature and power loss in complete gearbox systems. This report contains the basic data required to ascertain the specific losses for the various different machine elements, which can then be added together to determine total power loss. The newly developed calculation tool can also handle calculations defined in other current standards, such as ISO/TR 13593, AGMA 6123-B06, SKF 1994 and 2004, and VDI 2241.

This paper was originally presented at the 2014 International Gear Conference, Lyon Villeurbanne, France and is republished here with the authors' permission.

The heat dissipation calculation covers the heat dissipated via the housing surface, through the base, and the rotating parts, such as shafts and couplings. In it, the user can input more detailed information, for example, about the properties and condition of the surfaces to improve the accuracy of the calculation. The heat dissipation calculation uses the results of the power loss and temperature measurements at the thermal equilibrium, together with data about the ability of the gearbox to absorb heat, to mathematically calculate the temperature progressions and equilibrium temperatures.

A central point in this calculation is the option for users to modify the approach used to calculate individual losses and heat dissipation in order to suit their own specific requirements. In future, the interdependency of the power loss during operation and the operating temperature this generates will also be taken into account. This requires an iterative process to be applied to the calculation of temperature and power loss.

Practical application. The power loss and temperature calculation was applied to a gearbox (Fig. 1, left) recently developed by the company ZAE. The gearbox in question is a three-stage cylindrical bevel gearbox from the latest series, which has a nominal torque of 1,200 Nm. At present this series is produced in four different sizes, with a transmission ratio that ranges from 10:1 to 200:1.

KISSsoft's newly developed calculation tool was then used to calculate the power loss and operating temperature. To achieve this, KISSsoft designed the required gearbox model (Fig. 1, right) and supplied it with the necessary parameters.

The cylindrical bevel gearboxes underwent numerous measuring tests that recorded the temperature of the oil sump and the surrounding environment (Fig. 2). The associated levels of gearbox efficiency were measured at the same time.

To analyze how the gearboxes reacted under load, the operating conditions for the test series were set up in such a way as to ensure that all influences on speed and torque could be clearly identified. The results of these tests were then used as the basis for scaling the calculation model (Fig. 1, left). The sophisticated approach applied here made it possible to modify the power loss and temperature calculations in *KISSsys* to accurately reflect the situation in reality.

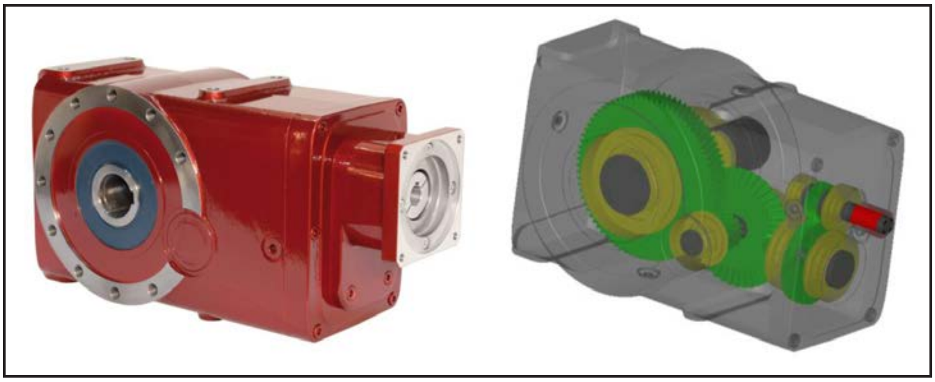

Figure 1 **ZAE cylindrical bevel gearbox and *KISSsoft* model.**

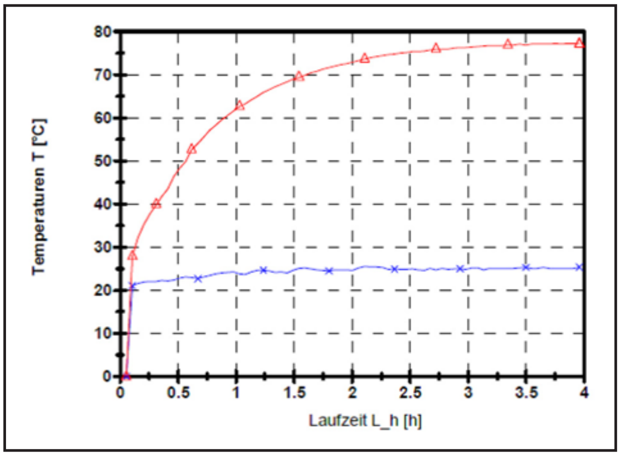

Figure 2 **Test result (temperature progression) of gearbox (red line) and ambient temperature (blue line).**

After being modified in this way, the calculation provided results that were very close to reality. These results then formed an excellent basis for the calculation module applied to other gear units. In future this calculation basis will also make it possible to size and optimize any type of gear unit.

Efficiency Calculation of Automotive Transmissions

Parallel to the industrial gearboxes, the automotive branch needs to focus on fuel economy and emission reduction in automotive transmissions, for example through hybridization or a higher number of gears and larger gear spread. Finally then analyzing and optimizing the efficiency can make a major contribution to reaching future CO_2 limits. In cooperation with IAV GmbH, one of the leading development partners to the automotive industry, KISSsoft AG developed a tool for analyzing, evaluating and optimizing transmission losses.

This paper uses an IAV 7-speed dual-clutch transmission to explain the process for efficient calculation of the transmission's overall efficiency. The primary focus here is on the automated generation of speed- and load-dependent power loss maps which can be used in subsequent cycle simulation. The modular approach to calculating individual losses provides the capability of performing detailed analyses by loss drivers and of validating optimization measures in a simple way.

Based on *KISSsoft* and *KISSsys* software, this calculation method is therefore a helpful tool employed throughout the development process — from concept phase to production layout stage. Combining IAV's experience with calculation tools from KISSsoft resulted creates a toolbox that can be widely used for practical cases.

Dual-clutch transmission IAV 7DCT280. The seven-speed dual-clutch transmission IAV 7DCT280 is a structurally optimized modular transmission in front-transverse design for compact and mid-size vehicles. Compared with current six- and seven-speed transmissions the mechanical components can be reduced to just one main shaft and one countershaft to provide the seven forward speeds. The package for the otherwise conventional

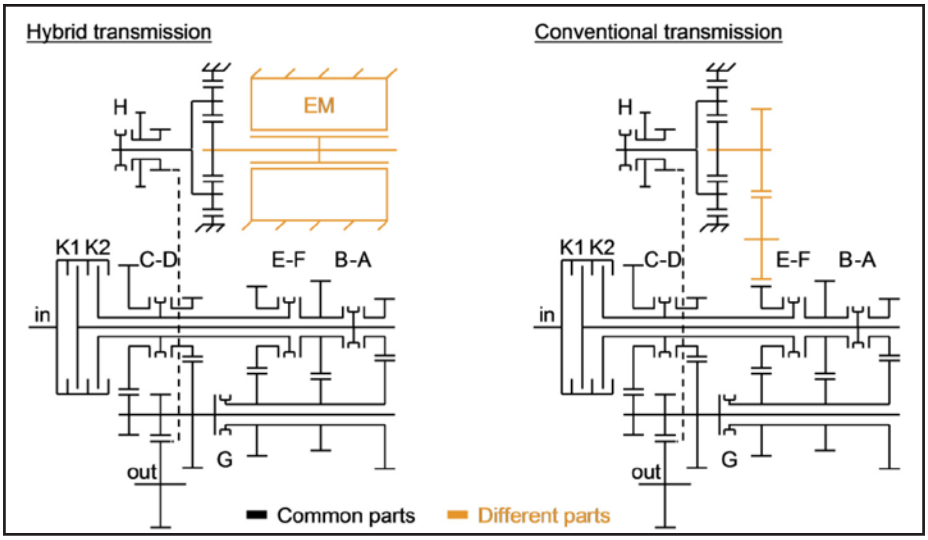

Figure 3 Modular transmission system of the 7-speed hybrid DCT (left) and the 7-speed DCT with mechanical reverse speed (right).

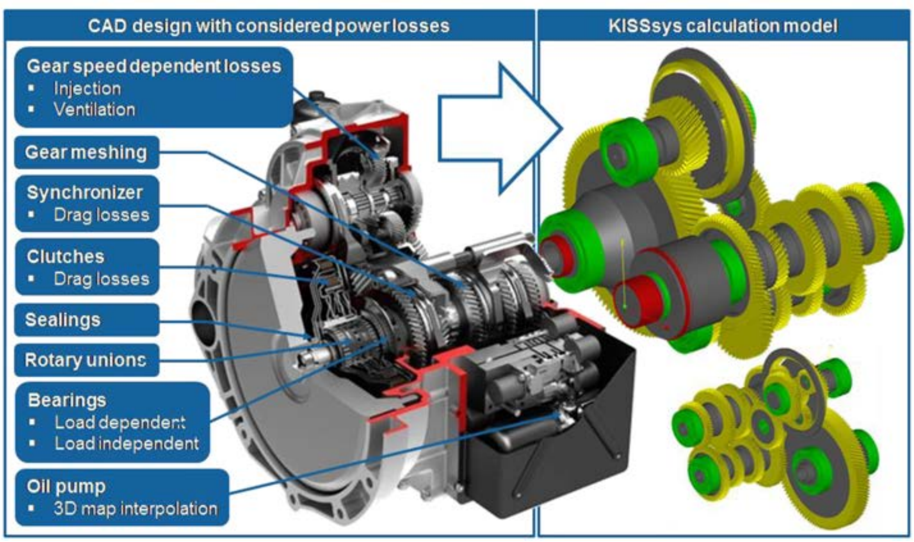

Figure 4 Considered power losses (left) and KISSsys calculation model (right).

second countershaft can be used on a modular basis to integrate an electric traction motor for various hybrid functions, or the mechanical components of a classic reverse speed (Fig. 3).

In the hybrid variant the electric motor is linked to the differential gear via a second output stage (Fig. 3, left). A planetary gear set serves as a booster stage for the electric motor, supplying the hybrid system with high levels of wheel torque — even at low vehicle speeds — and also helping to achieve better efficiency ranges. Synchronizer H provides the optional capability of disengaging the electric motor from the rest of the powertrain to protect the motor at higher vehicle speeds or in the event of a fault in the electric system.

In the conventional transmission variant these elements can be carried over in support of the commonality concept that is aimed for. The mechanical reverse speed is provided merely by replacing the electric motor with a set of spur gears connecting the main shaft's center gearwheel with the sun gear shaft of the planetary transmission (Fig. 3, right). This transmission structure is also shown to provide a high level of flexibility in its ratios, gear steps and spacing in return for slight modifications to the gear set ratio steps (Ref. 1).

Further investigations regarding power losses will be done on the basis of the conventional variant with reverse speed and without hybrid functions.

Model set-up and power loss maps. A calculation model under *KISSsys* (Fig. 4, right) is used to compute transmission power losses. At this system level all kinematic and kinetic interactions between the individual machine elements in the overall transmission are taken into consideration. This means that the most important prerequisite for automated generation of gear-dependent power loss maps is already fulfilled. In addition, the *KISSsys* model is complemented to include the loss modules. As in many cases a *KISSsys* model is already available for dimensioning of the machine elements, power losses can thus be calculated without major extra modeling work.

The analysis claims to be able to calculate the overall transmission efficiency in an integrated way; this is why all relevant loss sources are factored in. These are losses produced by gear meshing; churning and ventilation; drag torque at the synchronizer units and disengaged multi-disk clutches; by radial shaft sealing rings; rotary unions; and bearings as well as by the oil pump's power consumption (Fig. 4, left). The calculation methods to ascertain the different types of losses are based on relevant standards and dissertations or publications. Furthermore, there is great flexibility to choose different approaches for calculating individual loss sources.

The load-dependent losses caused by gear meshing are calculated on the basis of the proposal presented by Niemann/Winter (Ref. 2). In addition, contact analysis can be used to define the optimum microgeometry using profile and tooth trace modifications. A multitude of calculation methods is available for load-independent churning, squeezing and ventilation losses. The rules defined in standard ISO/TR 14179-2 (Ref. 3) are used to determine the losses occurring in the injection lubrication implemented in the dual-clutch transmission. The injection volume flows at the tooth contacts are assumed to be 1 l/min for the gear pairings and 2 l/min for the constant gear ratios respectively.

In the same way as for the gear teeth, a distinction is made for load-dependent and load-independent losses caused by the bearings. These are taken into account on the basis of the information provided in the SKF bearing catalog 1994 (Ref. 4). Alternatively, the new SKF 2004 method (Ref. 5) can be used, which attributes frictional components in relation to their cause.

The losses of the elements as non-actuated synchronizer units, radial shaft sealing rings and open multi-disk clutches are determined by individual literature sources, as i.e. Rao (6).

The power consumption of the oil pump depends to a large degree on the hydraulic concept implemented. This is why it is expedient to describe the pump's power consumption with the help of a map. The map can be defined using interpolation points and saved to *KISSsys*. Linear interpolation serves to ascertain power consumption at individual operating points.

After reading in maps up to the third dimension, proprietary analytical approaches can be defined as well. The first step consists of analyzing a concept with a constant-feed pump connected on the drive side to secure actuating pressure, lubrication and cooling oil demand. Although the IAV 7DCT280 uses a more efficient system, the following section will outline the potential improvement that can be achieved, starting from this basic version often used.

After incorporating all relevant losses in the computation model, automated generation of power loss maps can start for all gears, as well as for the individual components. The maps are generated incrementally based on a grid and a predefined engine speed and torque range for the internal combustion engine. Figure 5 shows the maps for the seventh gear of the dual-clutch transmission investigated here.

The maximum efficiency is 94.1% in seventh gear at low intake engine speeds and high torque. With rising speed and decreasing torque, efficiency drops, as was to be expected. On the one hand, this trend can be explained by the progressive development of speed-dependent losses due, for example, to bearing- and injection-related losses as well power consumption of the oil pump. The drop in efficiency at low intake torque is attributable to load-independent losses that constitute a constant variable at a defined engine speed.

Optimization measures for gears. After the cycle simulation (here the

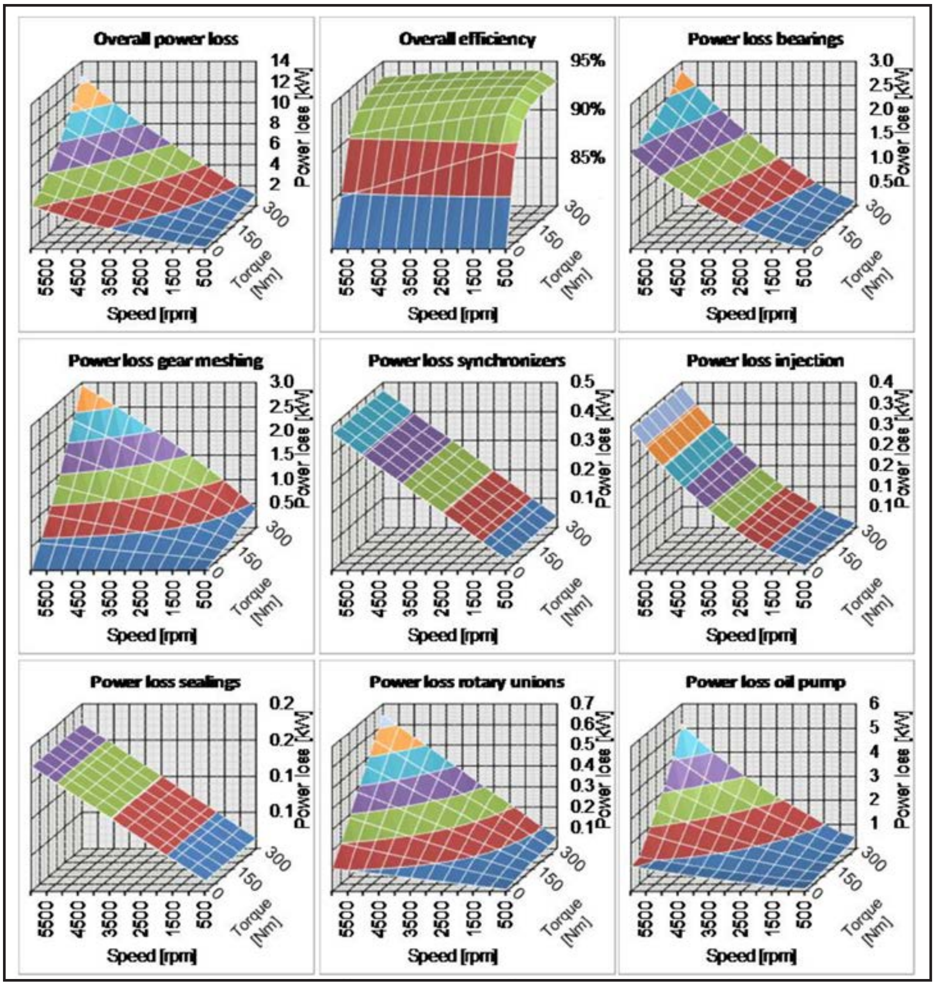

Figure 5 Power losses and efficiency maps of IAV 7DCT280 in 7th gear at 60° Coil temperature.

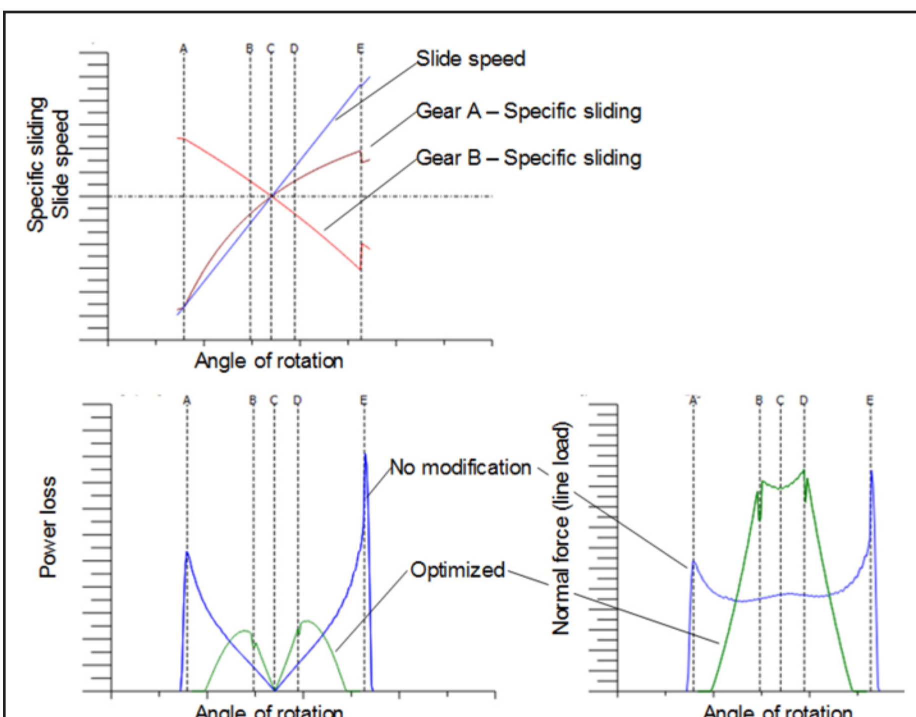

Figure 6 Sliding speed, local power loss, and normal force.

NEDC was applied) and analyzing the distribution of transmission losses, it was found that the actuating concept, the gear teeth, bearings and synchronizers harbor the greatest optimization potential for reducing CO_2 emission.

With a view to reducing gear meshing losses, the goal is to modify the design philosophy. The basic design exhibits a distribution between contact and overlap ratio of $\varepsilon\alpha = 2$ and $\varepsilon\beta = 1$ in order to meet the acoustic requirements. As the power loss mainly depends on the contact ratio, a new design with a distribution of $\varepsilon\alpha = 1.5$ and $\varepsilon\beta = 1.5$ is produced. With the same overlap ratio, a noticeable increase in efficiency is to be expected at the cost of acceptable acoustic disadvantages.

The reviewed gear teeth layout with a contact ratio of $\varepsilon\alpha = 1.5$ allows gear meshing losses to be reduced by 28 W. As axial forces increase as a result of the simultaneous rise in the helix angle, the benefit is partly offset by the bearing-related losses that are 5 W higher.

A possible next step in reducing the gear meshing losses consists in elaborating an alternative distribution between gearwheel transmission ratios and final drive transmission ratio. Using the transmission variant generator in *KISSsys*, a solution with lower, final drive transmission ratio and hence higher gearwheel ratio can be identified for the IAV 7DCT280.

Above and beyond the optimization measures described above, gear meshing losses can be further reduced by appropriate gear teeth modifications, using the load-dependent contact analysis in *KISSsoft*. Among others, the analysis ascertains friction torque at each meshing position based on contact simulation. As local losses depend on sliding speed and friction force, profile modifications and a resulting redistribution of meshing forces can reduce friction losses.

KISSsoft provides a specific sizing function for the gear modification. This function varies modifications within predefined limits, ascertains all possible combinations of up to three sets of modifications and carries out a contact analysis for each variant. At the same time it is possible to vary load as contact behavior strongly depends on load.

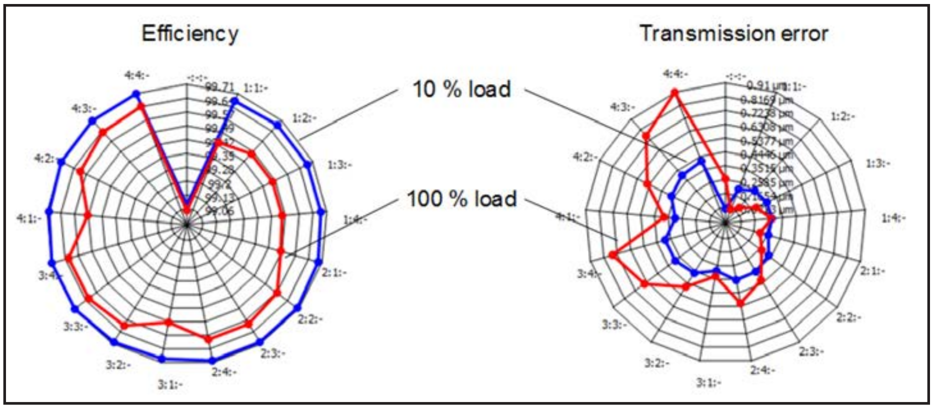

Figure 7 **Radar charts for efficiency (left) and transmission error (right).**

The results are visualized in clear form in radar charts, allowing several parameters to be reviewed simultaneously. This enables the engineer to determine the optimum combination of modifications for the case at hand. Figure 7 shows two radar charts, the left-hand one depicting efficiency, the right-hand one the transmission error. Here, variant 3:2 would constitute a good compromise exhibiting high efficiency (99.6% instead of 99.06%) and moderate transmission error.

Summary

The applications shown above represent the various but also different demands in efficiency and thermal rating calculation for gearboxes.

Whereas the industrial gearboxes include mainly machine elements for which the calculation methods already established and focus on thermal rating finally, the vehicle industry needs to meet the demands on CO_2 reduction targets and hence the efficiency calculation is important. However for many machine elements in automotive gearboxes no calculation methods are provided yet.

The calculation of the transmission power losses is based on a *KISSsys* model. On this system level all kinematic and kinetic interactions between the individual machine elements are taken into consideration. Beside many calculation standards the *KISSsoft* tools allow the possibility to check also about other criteria as gear strength (as, i.e., in case of changing oil viscosity for lower plunging losses) or noise (as, i.e., evaluating the transmission error in parallel to the efficiency of gears).

The process chain presented in this paper is suitable for both efficient evaluation of different transmission concepts and detailed optimization studies on existing transmissions. The calculation method based on KISSsoft and KISSsys is a helpful tool which is employed in the entire development process from the concept phase right through to the production layout stage. **PTE**

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High Contact Ratio Gearing: A Technology Ready for Implementation?

Charles D. Schultz

Today's competitive industrial gear marketplace demands products with excellent reliability, high capacity and low noise. Surface-hardened, ground tooth gearing predominates, but the legacy tooth forms handicap further improvements in capacity and noise generation. Vehicle and aircraft equipment use tooth forms not found in the standard tables to achieve better performance — with little or no increase in cost. This paper will propose adopting these high-contact ratio forms to industrial use.

Discussion

I first became aware of deeper-than-standard tooth forms in 1979. The venerable company had been through tough times but its staff of engineers and designers came up with some creative solutions in the effort to remain competitive. When competitors started to shift to carburized gearing and invest in gear grinding equipment, the owners did not have the cash to follow suit. Some clever engineer decided to use teeth that were 20% deeper than standard and nitride them. The rating methods then in effect gave them competitive power densities with only the purchase of custom cutting tools.

The 1.2 addendum combined with the 25° pressure angle did not result in true high-contact ratio geometry (Fig. 1). Poor tool life, especially when cutting hard pre-nitriding blanks, made for some production challenges. Coming from a through-hardening background I was very skeptical, but over time found the tooth form provided good results in the field. Replacing the special hobs wasn't possible in the reduced order volume of the early 1980s, however, and we did not use the 1.2 addendum system in new design standard products.

My next exposure to high-contact ratio gearing came 11 years later during a tour of the Saturn automobile plant in Spring Hill, Tennessee. The Society of Automotive Engineers (SAE) organized the event and we were keen to see the compact, integrated gear manufacturing cell that had been set up to produce all the components needed for a front-wheel drive transaxle. It was an impressive achievement in 1990 to begin with raw forgings at one end of the line and have complete carburized, hardened, and ground helical gears ready for assembly at the other end. General Motors spent plenty of money on the project and it challenged the best equipment builders in the world to participate.

The gear line included an automated inspection station after the gear grind operation. While watching the charting of parts in the cue, I noticed that the teeth were much deeper than "normal" but did not think to ask our guide a question about it. The equipment supplier gave out sample charts and when we debriefed back at our office we tried to run the geometry shown on it through our gear analysis software. The home-brewed code "blew up" at the dimensions entered and when we dug into the error codes it was found to have exceeded the "allowable" profile

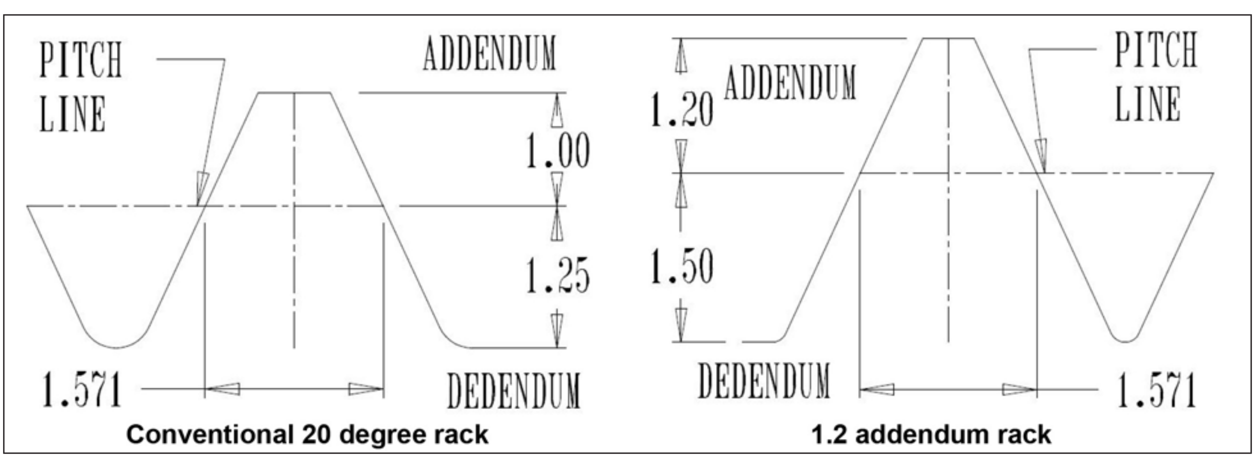

Figure 1 Deeper than standard tooth form.

contact ratio of 1.99. We didn't at first understand the significance of this limit in conventional gear design, but after scouring our engineering library we came across a great paper by J.C. Leming (Ref. 1) that explained things very well. Despite the many advantages of high-contact ratio gearing that Leming pointed out, we put the concept aside and continued to design products with "standard" teeth.

A couple years later, though, one of our salesmen asked us to help a potential customer resolve a noise problem with his equipment. Our firm had a well-deserved reputation as a supplier of high-quality ground tooth gears and we went to work reviewing a consultant's telephone book-thick report on the customer's "problem." Unfortunately the solutions suggested were things we had tried before without much success and we told the salesman we did not think the project was worth pursuing. But this salesman was a very persistent man and he refused to take no for an answer. Under the guise of giving the client a tour of our facility, he arranged for a couple of engineers to meet with my boss and me. We explained our dismal prognosis for quieting his gearbox and figured we were done with the matter. These engineers were just as persistent as our salesman and they knew we wouldn't be able to resist a well-argued challenge — especially after they told us their project motto was "We won't fail because we didn't spend enough money."

During the brainstorming that followed the Saturn tour, the Leming article came up. While I went to retrieve the reference book with the Leming paper in it, my boss committed me to designing a set of high-contact ratio gears in less than a week. There was, after all, a three-day weekend coming up and there would be fewer distractions. Six days later we met again and reviewed the proposed design. We had no way of predicting the possible noise reduction but the geometry worked out and we were ready to make drawings. The customer started expediting delivery of prototypes before the review meeting was over. We thought perhaps two weeks after the hobs arrived, maybe eight to ten weeks total.

But this was not acceptable and the customer promised to use his influence to get the hobs made more quickly. The next day, when the drawings were done, he called back to report that there could be no rush hob delivery. What other options were there? Jokingly reminding him of his project motto, we suggested wire cutting the parts. He didn't find the attempted humor funny and asked for blanks to be ready for his pick-up in two days. Said blanks were back to us three days later with Q9 quality teeth cut in them using tooth plots we provided. The sample gearbox was put on test two weeks later and the results were excellent. Noise reduction goals were easily met with no tooth modifications required.

Knowledgeable observers could not let go of the long, thin teeth appearing to be so delicate; surely those skinny teeth will break, they insisted. Upon completion of the sound tests the prototype gearbox was subjected to the same breakage test used many years earlier to approve the previous gearbox for production. It was still running flawlessly after completing the test three times. The conventional gearbox seldom survived extended testing. A modified version of the high-contact ratio gearbox has now been in production for over 20 years.

Tooling budgets and production schedules prevented me from often using high-contact ratio tooth forms while a gear company engineer. We managed to purchase a few HCR hobs for specific projects where there simply was not enough room for conventional gears to transmit the load but, regrettably, there was not the will to implement this technology in a widespread way. Now that I have my own consulting firm I hope to change that situation and assist clients in developing HCR-geared products.

The History of High-Contact Ratio Gearing

The official "history" of high-contact ratio gears begins with aircraft gearboxes in World War II. Leming's excellent summary of the development work on aircraft systems was published in 1977 but there is also some unofficial history dating back much further that bears study.

We take the "standard" involute tooth forms for granted as they were adopted long before any of today's working engineers were born. The 14½° "full-depth" involute was the first to gain official recognition in April of 1921; but even back then there was an effort to switch to 20°, first at stub-depth and shortly thereafter at full-depth, to meet increasing load requirements for automobiles and trucks. A "composite" 14½° system that combined an involute and cycloidal form into a single reference rack was also adopted in the 1920s, a recognition that not everyone was completely sold on the involute system either.

So where did the "standard" form come from? If you look at old photographs or drawings you will see a variety of tooth proportions, especially prior to the widespread use of hobbing and shaping machines in the late 1880s. Many gears had cast teeth and there is some evidence that the 14½° system became popular in part because the sine of 14½° is 0.25 and that makes it easier to draw the tooth shape into the pattern than other pressure angles. A more plausible reason, based upon my limited foundry experience, is that 14½° teeth have wider top-lands, which would be easier to maintain in the foundry conditions of that time.

In research for this paper I purchased a reprint of the *American Machinist Gear Book* (Ref. 2). Originally published in 1915 (pre-dating AGMA), this volume is a time capsule of our trade. Six different involute tooth systems are described as a prelude to discussing the need for a "standard" tooth form (Table 1). Wilfred Lewis's 1900 speech to the American Society of Mechanical Engineers (ASME) is quoted at length. When he started in gears in 1870 cycloidal teeth were predominant. By 1875 he was sold on the advantages of the involute system but he didn't like the 14.5° and 15° systems proposed. He went with 20° as, "I did not at the time have the courage of my convictions that the obliquity should be 22.5° or one-fourth of a right angle." I mention this as evidence that there is nothing magic about the tooth forms we have settled on as "standard." Using Lewis's dates we have a timeline of involute teeth coming into common use in 1875; a committee being assigned to adopt a standard form in 1891 with ASME; AGMA being formed in 1914; and the 14.5° full-depth tooth not being enshrined as standard until the 1921 AGMA annual meeting. Even in 1921 there was enough debate so that the 20° stub, composite cycloidal/involute rack, and 20° full-depth form were put "on track" for later standardization.

It is reasonably safe to say that the 14.5° form was not selected for its dynamic characteristics, as the 1921 debate recognized the more favorable sliding characteristics of the 20° stub system, along with its purported greater strength. I say "purported" based upon some instances I observed many years later where shaker screen gears were actually found to resist tooth breakage better at 14½° than even 25°. This puzzled us until we discovered the profile contact ratio was 2.47 with the legacy tooth form, and only 1.63 with the supposedly stronger 25° tooth. The same part with 20° full-depth teeth had a 1.93 profile contact ratio and it too suffered tooth breakage in the field. This situation points out the need to avoid single tooth contact entirely when designing HCR sets; the profile contact ratio has to remain over 2.00 at all times, regardless of tip relief or center distance fluctuation.

Many pressure angle and tooth depth systems were in use prior to "standardization" and they continued to be popular long after the 1920s. None had an addendum that exceeded the familiar 1/transverse diametrical pitch until Buckingham (Ref. 3, Section 2, *Spur and Internal Gears*) proposed a 1.35/NDP system for instrument gears (Fig. 2). I confess to using this book for many years and not noticing this gear tooth system until I started researching this paper. Buckingham does not discuss profile ratio in his presentation, despite developing the rack offsets needed to use the tooth form on spur pinions down to 5 teeth.

This is not to say that high-contact ratio gears were not used prior to 1935. One example of very non-standard tooth proportions that I am personally familiar with dates to 1895-vintage Hulet unloading machines. These revolutionary devices created an amazing reduction in the cost of unloading bulk products from the holds of ships on the Great Lakes and are considered national landmarks in Cleveland, Ohio and Wisconsin. The drive mechanism used "finger gears" to allow for a big change in center distance (on the order of 1 inch); finger gears (Fig. 4) were so named because they looked like fingers. The pressure angle was very low, around 8°, but the whole depth was on the order of 5 inches divided by the nominal DP. We were contracted to make spare pinions using our 1916-vintage gear milling machine. As I recall the tooth space was so deep and narrow we had to use three different milling cutters to get the shape and, because of accuracy limitations of the technology, hand-file the transitions to get relatively smooth operation.

Most of the manufacturing techniques in use today were available 100 years ago; the machines were far less accurate and they were a great deal slower. Metallurgy and heat treating were not as sophisticated; bearings were of much lower capacity and quality. Every aspect of machinery was slower and our predecessors, being very practical people, reserved gear grinding for applications where it was the only way to get the gearbox to work. The 14½° full-depth form was still adequate for most applications in 1921, but designers could see that the 20° form, first in stub-depth and later in full-depth, offered advantages for the future.

My purpose in bringing this topic into the discussion of high-contact ratio teeth is simply this: The old answers were based on old conditions. We have different conditions in effect today. Many of the old technology and cost limitations are no longer in effect. We are under great commercial pressure to produce lighter, more compact, longer lasting gearboxes at lower prices. The design rules have to change to help us respond to those commercial pressures.

Table 1 Existing tooth "standards" in 1915, per *American Machinist Gear Book* (pp. 23-24)

	Pressure angle	Addendum	Dedendum	Whole depth
Brown & Sharpe	14.5	1/p	1.157/p	2.157/p
Grant	15	1/p	1.157/p	2.157/p
Sellers	20	1/p	1.157/p	2.157/p
Hunt	14.5	0.7857/p	0.9424/p	1.7278/p
Logue/Nuttall	20	0.7857/p	0.9424/p	1.7278/p
Fellows stub*	20	1/p'	1.157/p'	2.157/p'

- Tooth thickness based on p; tooth height based on p'.
- Examples: 2/2.5, 2.5/3, 3/4, 4/5, 5/7, 7/9, 8/10, 10/12, 12/14, 14/18.

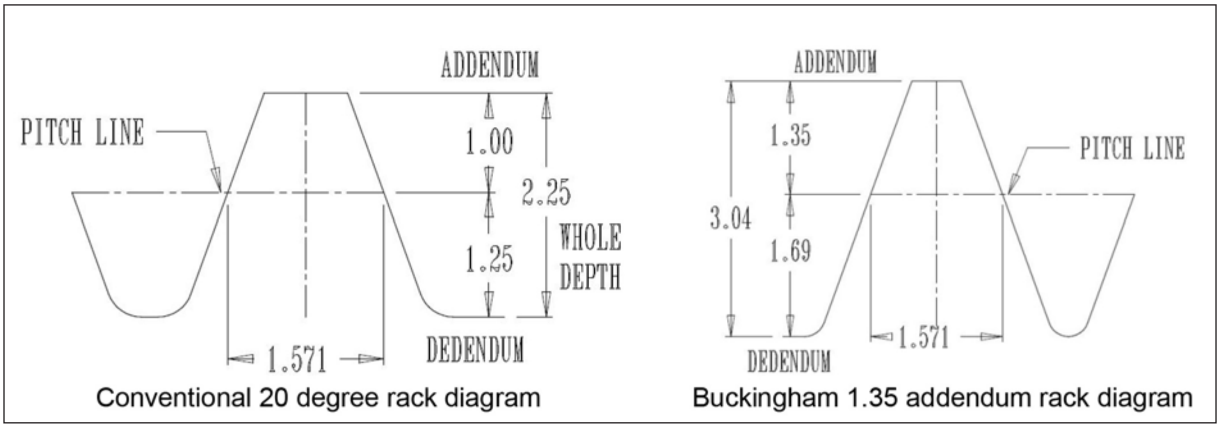

Figure 2 High-contact ratio tooth form.

Design Concerns with HCR Teeth

Since the publication of Leming's paper, high-contact ratio (HCR) gears have been used in many aircraft, defense, and vehicle applications. They have yet to be featured in "catalog" gearboxes, despite the following advantages:

- Increased durability rating
- Increased strength rating
- Reduced noise levels

These advantages, while noteworthy, have been overshadowed by concerns about susceptibility to scoring or other lubrication failures; lower efficiency; narrow top-lands; limited bearing capacity; gearbox thermal limitations; tooling costs; and uncertainty over rating methods. During these past decades many papers have been presented on these concerns as our aircraft and vehicle designing colleagues investigated the best ways to use the immerging technology. I claim no great breakthroughs in this paper but hope to alleviate a few fears and suggest a path forward.

The most difficult advantage to quantify for HCR designs is reduced noise level. In every application of HCR gearing that I know of, the noise level was lower than the conventional gearing it replaced. While mathematical models have been developed to determine optimum tooth modifications for conventional gears, those models require specific information on the load and speed for which noise reduction is needed. Catalog products are sold "off the shelf" with only limited load and speed information. A recent paper on the use of HCR timing gears in diesel truck engines (Ref. 4) revealed that the best noise performance was obtained with gears having little or no profile modification. The worst performers had modifications closer to what conventional math models suggested was "optimum." My own experience with un-modified HCR profiles leads me to believe HCR gearing can be successfully used in catalog gearboxes with no tip or root relief at all.

With regard to the durability rating of HCR gears, the theoretical basis of current AGMA and ISO contact stress formulas has no restrictions on profile contact ratio. The increased capacity of HCR teeth is a matter of tooth curvature and the length of the line of contact. Depending upon the addendum factor chosen for the HCR tooth form, durability ratings can increase from 25 to 50% over similar-sized conventional gears; lab testing has confirmed these results (Ref. 5).

Our current tooth bending strength models are based upon single tooth contact. True HCR designs never see single tooth loading, so a new stress calculation formula will ultimately be needed to accurately predict the success of any HCR tooth form. Photo elastic modeling and finite element analysis results indicate that HCR teeth experience between 57 and 63% of the bending load of conventional gearing. Further testing will be needed before an HCR bending strength formula can be adopted.

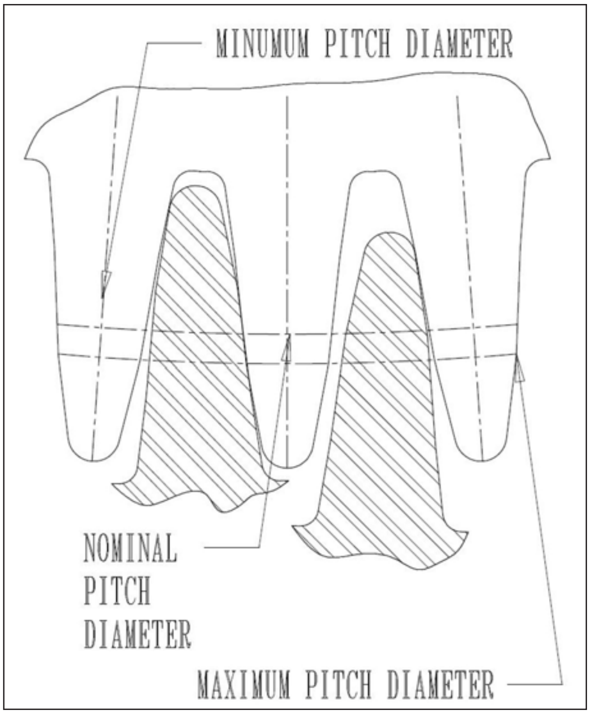

Figure 3 Hulet unloader finger gears.

Math Modeling HCR Gears

For the purposes of this paper I have selected two different size cataloged, parallel shaft, double-reduction speed reducers for study. Since specific design details are proprietary, I began by designing conventional gear, normal contact ratio (NCR) sets that would fit within the housing envelope and then selecting suitable taper roller bearings. These conventional 25° pressure angle helical sets were then rated for durability and strength to confirm that they were capable of the published catalog ratings. The catalog ratings and simulated gear geometry were used to calculate L10 gearing life using the advanced method (a23 factor).

The next step was to design HCR gear sets for the same conditions and repeat the durability and strength calculations before revisiting the bearing life issue. Durability was calculated using the AGMA 2001 method; strength was calculated using the standard method but the result was divided by 0.60 to reflect the load sharing reported in FEA modeling. As there is no "standard" HCR tooth form, I elected to use the 1.35 addendum 20° NPA system that Buckingham proposed for instrument gearing. Occasional minor, warning notes were received from the rating software for top-lands less than 0.250/NDP, but the rating process was otherwise unimpeded. Narrow top-lands are thought to contribute to tooth bending failures; the same warnings were received on some NCR 25° pressure angle sets.

While proposals have been advanced to achieve profile contact ratios of 1.95 or more using standard 20° full-depth tooling (Ref. 6), I chose to study only deeper than standard depth tooth forms. The use of standard depth tools on HCR gears results in reduced operating pressure angles and increased risk of undercutting, without the increased durability rating offered by the deeper tooth form. Catalog ratings are determined by the lowest capacity in a number of categories. Back in the through-hardened days it was expected that products would be durability-limited and that strength ratings would generally be 40 to 50% higher. When we moved to carburized and hardened gearing we found that the durability and strength ratings both came into play in establishing catalog ratings.

The use of standard depth tooling to achieve HCR profile overlaps would return us to durability-limited catalog ratings; overall ratings would probably not increase at all. Contrast this with the move to deeper than standard teeth where durability capacity will increase by 25 to 50% and strength ratings may double. Commercial success comes with high-quality products at lowest prices; high power density contributes to lower prices, as you are more likely to be able to meet a specific application with a "one size smaller" gearbox than a competitor.

Tables 2 through 5 show the results of the two-unit NCR/HCR rating comparison. HCR designs achieved a durability rating increase of 28 to 29%. HCR strength ratings were 44 to 48% more than comparable NCR designs. Greater improvements may be possible with more flexibility in the choice of center distance combinations and stage ratios. These particular examples were chosen to illustrate the potential for HCR redesigns of existing products using existing housing dimensions.

Many existing product lines are also bearing life-limited; the 25° normal pressure angles needed to obtain high bending strengths also increase the forces on the bearings. Space limitations and bearing availability prevent squeezing in more bearing capacity. The lower pressure angles used in the HCR designs have lower bearing forces, but the packaging problem may prevent utilization of increased rating capacity. Allowable "bearing horsepower" for each of the units studied is shown in Tables 6 through 9. With the space available for bearings in the current design units, I was not able to obtain a 10,000-hour L10 on every bearing with the published catalog ratings. Since few gearboxes are sold at a unity service factor, this is not a surprise.

Converting existing gearbox designs to HCR will reduce noise levels and provide additional service factors. To best leverage the technology, however, more flexibility in center distance sequences and ratio combinations will be needed. This is not unprecedented; a review of parallel-shaft gearbox catalogs shows that pre-1964 designs had far different proportions than more recent designs. The first-stage center distance in those through-hardened units is typically 50 to 62% of the second stage. The low-speed gear ratio in those units may be as high as 6.5:1. These are a reflection of the rating methods in effect at the time they were designed. Up until 1964, for example, the durability rating was calculated based upon pinion pitch diameter and pinion rotational speed. This, along with the favorable treatment of allowable stress for second and third reductions, encouraged higher ratios on the output set.

When the "modern" rating method was adopted via AGMA 218 in the 1980s, the durability rating formula changed to the pinion pitch diameter *squared,* and the favorable treatment of second and third reductions went away. This change in rating method is reflected in the design of newer parallel shaft units. The first-stage center distances are now typically 70 to 80% of the second stage. Output stage gear ratios seldom exceed 5:1. Just as the adoption of carburized and ground gearing motivated that shift, HCR designs may also require a different approach to these fundamental design parameters.

With regard to the lubrication concerns with HCR gears, scoring and wear probabilities were calculated for the modeled gears using commercial software. Unfortunately the program wouldn't accept gearing with profile contact ratios over 2.00, so the outside diameters of the HCR gears were reduced to obtain a 1.99. With the surface finish expected for form ground gears (22 AA) and required lubricant conditions (ISO 320EP at 160°F bulk temperature), all sets had scoring and wear probabilities of less than 5%.

Efficiency testing, in conjunction with thermal rating development, would be necessary to determine whether HCR gearing has any disadvantage compared to similar-sized NCR gearing. A review of the factors involved with operating efficiency and thermal limitation shows that the longer line of action and slightly larger outside diameters of the HCR designs could increase power loss. On the other hand, the higher power density of HCR gearing would make the drives smaller in size and potentially render the overall efficiency equal. The author is not privy to the test results of automotive gearbox builders, but doubts they would have moved to HCR designs if efficiency were a problem.

The way forward. The advantages of HCR gear designs are ripe for commercial adoption. Tougher noise restrictions are inevitable and HCR technology has amply demonstrated its ability to reduce noise levels in vehicles. The opportunity to increase power density — whether for overall commercial advantage or just to raise ratings in specific situations, at only a slight increase in material cost — is very attractive in today's competitive market.

Early adopters of any technological change have to temper enthusiasm with common sense. A well thought out test program will be needed to verify the rating advantages and validate the thermal capacity of the products. Theoretical work is needed to support a new high-contact ratio bending strength rating method, along with laboratory testing of HCR sets under standardized conditions.

Table 2 Conventional gearing (2H155 gearbox)

UNIT RATIO>	6.3076	6.8421	8.1092	8.7579	9.9522		10.7094	12.7973	13.8109	16.0105	17.2540		
	single helical	single helical	single helical	single helical	single helical		single helical	single helical	single helical	single helical	single helical		
set #	155H1	155H2	155H3	155H4	155H5	<LOCATION	155H6	155H7	155H8	155H9	155H10	<LOCATION	155L
Catalog HP	239	216	187	189	154	Catalog HP	138	120	108	96	87	Catalog HP	various
CENTERS (mm)	108	109	109	109	109	CENTERS (mm)	108	108	109	108	108	CENTERS (mm)	155
CENTERS (in)	4.291	4.291	4.291	4.291	4.291	CENTERS (in)	4.291	4.291	4.291	4.291	4.291	CENTERS (in)	6.102
GEAR TEETH	59	60	64	64	64	GEAR TEETH	72	101	109	117	116	GEAR TEETH	65
PINION TEETH	32	30	27	25	22	PINION TEETH	23	27	27	25	23	PINION TEETH	19
RATIO	1.8438	2.0000	2.3704	2.5600	2.9091	RATIO	3.1304	3.7407	4.0370	4.6800	5.0435	RATIO	3.4211
FACE WIDTH	1.969	1.969	1.969	1.969	1.969	FACE WIDTH	1.969	1.969	1.969	1.969	1.969	FACE WIDTH	3.543
NDP	11.2889	11.28890	11.2889	11.2889	11.2889	NDP	12.7	16.93330	16.9333	18.1429	16.9333	NDP	7.25714
NPA	25	25	25	25	25	NPA	25	25	25	25	25	NPA	25
HELIX ANGLE	20.0789	21.7360	20.0790	23.2809	27.4250	HELIX ANGLE	29.3596	28.2690	20.6455	24.2270	16.9766	HELIX ANGLE	18.4875
TDP	10.6028	10.4862	10.6028	10.3697	10.0202	TDP	11.0688	14.9138	15.8459	16.5450	16.1954	TDP	6.8826
PINION PD	3.0181	2.8609	2.5465	2.4109	2.1956	PINION PD	2.0779	1.8104	1.7039	1.5110	1.4202	PINION PD	2.7606
GEAR PD	5.5646	5.7218	6.0362	6.1718	6.3871	GEAR PD	6.5048	6.7723	6.8788	7.0716	7.1625	GEAR PD	9.4441
Pinion X1	0.1500	0.1650	0.1900	0.2000	0.2000	Pinion X1	0.2000	0.2000	0.2500	0.2500	0.2650	Pinion X1	0.3000
PINION OD	3.222	3.067	2.757	2.624	2.408	PINION OD	2.267	1.952	1.852	1.649	1.570	PINION OD	3.119
GEAR OD	5.715	5.870	6.180	6.314	6.529	GEAR OD	6.631	6.867	6.967	7.154	7.249	GEAR OD	9.637
Mp	1.38	1.35	1.37	1.32	1.25	Mp	1.23	1.26	1.36	1.31	1.39	Mp	1.34
Mf	2.43	2.62	2.43	2.80	3.26	Mf	3.90	5.03	3.74	4.67	3.10	Mf	2.02
PINION HT	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	GEAR HT	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	GEAR KT	58-62 Rc
GEAR HT	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	PINION HT	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	PINION HT	58-62 Rc
AGMA Q#	11	11	11	11	11	AGMA Q#	11	11	11	11	11	AGMA Q#	11
Cm	1.08	1.08	1.08	1.08	1.08	Cm	1.08	1.08	1.08	1_DB	1.08	Cm	1.15
PINION RPM	1800	1800	1800	1800	1800	PINION RPM	1800	1800	1800	1800	1800	PINION RPM	various
PINION DUR. HP	308	281	241	224	197	PINION DUR.HP	180	147	130	105	98	PINION DUR.HP	
GEAR DUR. HP	317	291	251	234	207	GEAR DUR. HP	190	156	139	113	105	GEAR DUR. HP	
PINION STR. HP	288	277	248	238	213	PINION STR. HP	184	126	117	96	97	PINION STR.HP	
GEAR STR. HP	284	274	247	237	213	GEAR STR. HP	197	129	118	99	100	GEAR STR.HP	
LS Pinion RPM	976	900	759	703	619	LS Pinion RPM	575	481	446	385	357	LS Pinion RPM	
LS Pinion DurHP	249	232	197	184	163	LS Pinion DurHP	152	129	120	105	98	LS Pinion DurHP	
LS Gear DurHP	263	246	208	194	173	LS Gear DurHP	161	137	127	111	103	LS Gear DurHP	
LS Pinion Str HP	313	292	246	229	203	LS Pinion Str HP	189	160	148	129	120	LS Pinion Str HP	
LS Gear Str HP	303	282	238	222	196	LS Gear Str HP	183	154	144	124	116	LS Gear Str HP	
Unit Dur. HP	249	232	197	184	163	Unit Dur. HP	152	129	120	105	98	Unit Dur. HP	
Unit Str. HP	284	274	238	222	196	Unit Str. HP	183	126	117	96	97	Unit Str. HP	
Dur. SF to Cat	1.04	1.07	1.05	1.09	1.06	Dur. SF to Cat	1.10	1.08	1.11	1.09	1.13	Dur. SF to Cat	
Str. SF to Cat	1.19	1.27	1.27	1.31	1.27	Str. SF to Cat	1.33	1.05	1.08	1.00	1.11	Str. SF to Cat	
Thermal HP	84	84	84	84	77	Thermal HP	77	77	77	71	71	Thermal HP	
1 fan	139	139	139	139	127	1 fan	127	127	127	116	116	1 fan	
2 fans	210	210	210	210	193	2 fans	193	193	193	176	176	2 fans	

Table 3 HCR gearing, 1.35 addendum system (2H155 gearbox)

UNIT RATIO>	6.3076	6.8421	8.1092	8.7579	10.0000		10.7519	12.8014	13.8109	16.0526	17.2540			
	single helical	single helical	single helical	single helical	single helical		single helical	single helical	single helical	single helical	single helical		single helical	
set #	155H1	155H2	155H3	155H4	155H5	<LOCATION	155H6	155H7	155H8	155H9	155H10	<LOCATION	155L	
Catalog HP	239	216	187	169	154	Catalog HP	138	120	108	96	87	Catalog HP	various	
CENTERS (mm)	109	109	109	109	109	CENTERS (mm)	109	109	109	109	109	CENTERS (mm)	155	
CENTERS (in)	4.291	4.291	4.291	4.291	4.291	CENTERS (in)	4.291	4.291	4.291	4.291	4.291	CENTERS (in)	6.102	
GEAR TEETH	59	60	64	74	76	GEAR TEETH	88	116	109	122	116	GEAR TEETH	65	
PINION TEETH	32	30	27	29	26	PINION TEETH	28	31	27	26	23	PINION TEETH	19	
RATIO	1.8438	2.0000	2.3704	2.5517	2.9231	RATIO	3.1429	3.7419	4.0370	4.6923	5.0435	RATIO	3.4211	
FACE WIDTH	1.969	1.969	1.969	1.969	1.969	FACE WIDTH	1.969	1.969	1.969	1.969	1.969	FACE WIDTH	3.543	
NDP	11.2889	11.28890	11.2889	12.7	12.7	NDP	14.5143	18.0000	16.9333	18	16.9333	NDP	7.25714	
NPA	20	20	20	20	20	NPA	20	20	20	20	20	NPA	20	
HELIX ANGLE	20.0789	21.7360	20.0790	19.0991	20.6455	HELIX ANGLE	21.3787	17.9122	20.6455	16.6642	16.9766	HELIX ANGLE	18.4875	
TDP	10.6028	10.4862	10.6028	12.0009	11.8844	TDP	13.5156	17.1275	15.8459	17.2440	16.1954	TDP	6.8826	
PINION PD	3.0181	2.8609	2.5465	2.4165	2.1877	PINION PD	2.0717	1.8100	1.7039	1.5078	1.4202	PINION PD	2.7606	
GEAR PD	5.5646	5.7218	6.0362	6.1662	6.3949	GEAR PD	6.5110	6.7727	6.8788	7.0749	7.1625	GEAR PD	9.4441	
Pinion X1	0.1500	0.1650	0.1900	0.1900	0.2200	Pinion X1	0.2200	0.2400	0.2500	0.2600	0.2650	Pinion X1	0.3000	
PINION OD	3.285	3.129	2.819	2.666	2.435	PINION OD	2.288	1.987	1.893	1.687	1.611	PINION OD	3.119	
GEAR OD	5.778	5.932	6.242	6.356	6.573	GEAR OD	6.667	6.896	7.009	7.196	7.291	GEAR OD	9.637	
Mp	2.05	2.01	2.02	2.12	2.01	Mp	2.02	2.11	2.03	2.10	2.06	Mp	1.34	
Mf	2.43	2.62	2.43	2.60	2.81	Mf	3.32	3.47	3.74	3.24	3.10	Mf	2.02	
PINION HT	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	GEAR HT	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	GEAR KT	58-62 Rc	
GEAR HT	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	PINION HT	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	PINION HT	58-62 Rc	
AGMA Q#	11	11	11	11	11	AGMA Q#	11	11	11	11	11	AGMA Q#	11	
Cm	1.08	1.08	1.08	1.08	1.08	Cm	1.08	1.08	1.08	1_DB	1.08	Cm	1.15	
PINION RPM	1800	1800	1800	1800	1800	PINION RPM	1800	1800	1800	1800	1800	PINION RPM	various	
PINION DUR. HP	410	382	322	305	258	PINION DUR.HP	239	192	174	142	129	PINION DUR.HP		
GEAR DUR. HP	422	394	335	319	271	GEAR DUR. HP	251	204	186	153	139	GEAR DUR. HP		
PINION STR. HP	460	435	379	329	293	PINION STR. HP	250	181	177	145	140	PINION STR.HP		
GEAR STR. HP	473	449	401	352	309	GEAR STR. HP	271	198	196	164	162	GEAR STR.HP		
LS Pinion RPM	976	900	759	705	616	LS Pinion RPM	573	481	446	384	357	LS Pinion RPM		
LS Pinion DurHP	320	299	253	236	210	LS Pinion DurHP	196	166	155	135	126	LS Pinion DurHP		
LS Gear DurHP	338	316	268	250	222	LS Gear DurHP	207	176	164	143	133	LS Gear DurHP		
LS Pinion Str HP	408	400	338	315	279	LS Pinion Str HP	259	219	204	176	164	LS Pinion Str HP		
LS Gear Str HP	440	431	364	339	300	LS Gear Str HP	279	236	219	190	177	LS Gear Str HP		
Unit Dur. HP	320	299	253	236	210	Unit Dur. HP	196	166	155	135	126	Unit Dur. HP		
Unit Str. HP	408	400	338	315	279	Unit Str. HP	250	181	177	145	140	Unit Str. HP		
Dur. SF to Cat	1.34	1.38	1.35	1.40	1.36	Dur. SF to Cat	1.42	1.38	1.44	1.41	1.45	Dur. SF to Cat		
Str. SF to Cat	1.71	1.85	1.81	1.86	1.81	Str. SF to Cat	1.81	1.51	1.64	1.51	1.61	Str. SF to Cat		
NCR dur	249	232	197	184	163	NCR dur	152	129	120	105	98	NCR dur		
NCR Str	284	274	238	222	196	NCR Str	183	125	117	96	97	NCR Str		
dur increase	1.29	1.29	1.28	1.28	1.29	dur increase	1.29	1.29	1.29	1.29	1.29	dur increase		average
strength increase	1.44	1.46	1.42	1.42	1.42	strength increase	1.37	1.43	1.51	1.51	1.45	strength increase		average

Table 4 Conventional gearing (2H330 gearbox)

UNIT RATIO>	6.4316	7.0955	8.1579	8.9961	9.8966		10.9211	12.7918	14.0789	15.5373	17.1053		
	single helical	single helical	single helical	single helical	single helical		single helical	single helical	single helical	single helical	single helical		
set #	330H1	330H2	330H3	330H4	330H5	<LOCATION	330H6	330H7	330H8	330H9	330H10	<LOCATION	330L
Catalog HP	1910	1810	1650	1440	1370	Catalog HP	1190	1070	934	889	770	Catalog HP	various
CENTERS (mm)	226	226	226	226	226	CENTERS (mm)	226	226	226	226	226	CENTERS (mm)	330
CENTERS (in)	8.898	8.898	8.898	8.898	8.898	CENTERS (in)	8.898	8.898	8.898	8.898	8.898	CENTERS (in)	12.992
GEAR TEETH	47	56	62	71	81	GEAR TEETH	83	86	107	109	95	GEAR TEETH	65
PINION TEETH	25	27	26	27	28	PINION TEETH	26	23	26	24	19	PINION TEETH	19
RATIO	1.8800	2.0741	2.3846	2.6296	2.8929	RATIO	3.1923	3.7391	4.1154	4.5417	5.0000	RATIO	3.4211
FACE WIDTH	4.000	4.000	4.000	4.000	4.000	FACE WIDTH	4.000	4.000	4.000	4.000	4.000	FACE WIDTH	5.846
NDP	4.2333	5.08000	5.08	6	6.35	NDP	6.35	6.35000	7.8154	7.8154	6.7733	NDP	3.3867
NPA	25	25	25	25	25	NPA	25	25	25	25	25	NPA	25
HELIX ANGLE	17.1078	23.3441	13.2320	23.3867	15.2904	HELIX ANGLE	15.2904	15.2904	169.9998	16.9998	18.9502	HELIX ANGLE	17.3414
TDP	4.0460	4.6642	4.9451	5.5071	6.1252	TDP	6.1252	6.1252	7.4739	7.4739	6.4062	TDP	3.2327
PINION PD	6.1789	5.7888	5.2577	4.9028	4.5713	PINION PD	4.2447	3.7550	3.4788	3.2112	2.9659	PINION PD	5.8774
GEAR PD	11.6164	12.0065	12.5376	12.8925	13.2240	GEAR PD	13.5505	14.0403	14.3165	14.5841	14.8294	GEAR PD	20.1069
Pinion X1	0.1500	0.1750	0.2000	0.2500	0.2700	Pinion X1	0.3000	0.3250	0.3600	0.3750	0.3000	Pinion X1	0.2400
PINION OD	6.722	6.251	5.730	5.318	4.971	PINION OD	4.654	4.172	3.827	3.563	3.350	PINION OD	6.610
GEAR OD	12.018	12.331	12.853	13.142	13.454	GEAR OD	13.771	14.253	14.480	14.744	15.036	GEAR OD	20.556
Mp	1.39	1.32	1.43	1.32	1.42	Mp	1.41	1.39	1.39	1.37	1.34	Mp	1.36
Mf	1.59	2.56	1.48	3.03	2.13	Mf	3.13	2.13	2.91	2.91	2.80	Mf	1.88
PINION HT	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	GEAR HT	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	GEAR KT	58-62 Rc
GEAR HT	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	PINION HT	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	PINION HT	58-62 Rc
AGMA Q#	11	11	11	11	11	AGMA Q#	11	11	11	11	11	AGMA Q#	11
Cm	1.19	1.19	1.19	1.19	1.19	Cm	1.19	1.19	1.19	1.19	1.19	Cm	1.23
PINION RPM	1800	1800	1800	1800	1800	PINION RPM	1800	1800	1800	1800	1800	PINION RPM	various
PINION DUR. HP	2165	2102	1750	1749	1557	PINION DUR.HP	1389	1142	1014	888	750	PINION DUR.HP	
GEAR DUR. HP	2229	2174	1822	1828	1635	GEAR DUR. HP	1465	1214	1083	952	808	GEAR DUR. HP	
PINION STR. HP	2527	2175	1821	1693	1453	PINION STR. HP	1352	1193	934	861	880	PINION STR.HP	
GEAR STR. HP	2488	2139	1793	1644	1410	GEAR STR. HP	1302	1148	897	825	890	GEAR STR.HP	
LS Pinion RPM	957	868	755	685	622	LS Pinion RPM	564	481	437	396	360	LS Pinion RPM	
LS Pinion DurHP	2068	1889	1661	1518	1390	LS Pinion DurHP	1266	1097	1004	916	831	LS Pinion DurHP	
LS Gear DurHP	2189	1999	1758	1607	1471	LS Gear DurHP	1340	1161	1062	969	879	LS Gear DurHP	
LS Pinion Str HP	2578	2348	2057	1875	1712	LS Pinion Str HP	1555	1341	1224	1114	1007	LS Pinion Str HP	
LS Gear Str HP	2599	2367	2074	1890	1726	LS Gear Str HP	1567	1352	1234	1123	1016	LS Gear Str HP	
Unit Dur. HP	2068	1889	1661	1518	1390	Unit Dur. HP	1266	1097	1004	888	750	Unit Dur. HP	
Unit Str. HP	2488	2139	1793	1644	1410	Unit Str. HP	1302	1148	897	825	831	Unit Str. HP	
Dur. SF to Cat	1.08	1.04	1.01	1.05	1.01	Dur. SF to Cat	1.06	1.03	1.07	1.00	0.97	Dur. SF to Cat	
Str. SF to Cat	1.30	1.18	1.09	1.14	1.03	Str. SF to Cat	1.09	1.07	0.96	0.93	1.08	Str. SF to Cat	
Thermal HP	314	314	314	314	309	Thermal HP	309	309	309	293	293	Thermal HP	
1 fan	518	518	518	518	510	1 fan	510	510	510	484	484	1 fan	
2 fans	784	784	784	784	773	2 fans	773	773	773	733	733	2 fans	

Table 5 HCR gearing, 1.35 addendum system (2H330 gearbox)

UNIT RATIO>	6.4091	7.0381	8.1575	8.9646	9.8620		10.8829	12.7470	14.0297	15.4830	17.2004			
	single helical	single helical	single helical	single helical	single helical		single helical	single helical	single helical	single helical	single helical			
set #	330H1	330H2	330H3	330H4	330H5	<LOCATION	330H6	330H7	330H8	330H9	330H10	<LOCATION	330L	
Catalog HP	1910	1810	1050	1440	1370	Catalog HP	1190	1070	934	889	770	Catalog HP	various	
CENTERS (mm)	226	226	226	226	226	CENTERS (mm)	226	226	226	226	226	CENTERS (mm)	330	
CENTERS (in)	8.898	8.898	8.898	8.898	8.898	CENTERS (in)	8.898	8.898	8.898	8.898	8.898	CENTERS (in)	12.992	
GEAR TEETH	47	64	67	71	81	GEAR TEETH	83	86	107	109	111	GEAR TEETH	75	
PINION TEETH	25	31	28	27	28	PINION TEETH	26	23	26	24	22	PINION TEETH	22	
RATIO	1.8800	2.0645	2.3929	2.6296	2.8929	RATIO	3.1923	3.7391	4.1154	4.5417	5.0455	RATIO	3.4091	
FACE WIDTH	4.000	4.000	4.000	4.000	4.000	FACE WIDTH	4.000	4.000	4.000	4.000	4.000	FACE WIDTH	5.846	
NDP	4.2333	5.64440	5.6444	5.6444	6.35	NDP	6.35	6.35000	7.8154	7.8154	7.8154	NDP	3.3867	
NPA	20	20	20	20	20	NPA	20	20	20	20	20	NPA	20	
HELIX ANGLE	17.1078	18.9509	18.9509	12.6661	15.2904	HELIX ANGLE	15.2904	15.2904	16.9998	16.9998	16.9998	HELIX ANGLE	17.1624	
TDP	4.0460	5.3385	5.3385	5.5071	6.1252	TDP	6.1252	6.1252	7.4739	7.4739	7.4739	TDP	3.7330	
PINION PD	6.1789	5.8069	5.2449	4.0028	4.5713	PINION PD	4.2447	3.7550	3.4788	3.2112	2.9436	PINION PD	5.8933	
GEAR PD	11.6164	1 1.9884	12.5504	12.8925	13.2240	GEAR PD	13.5505	14.0403	14.3165	14.5841	14.8517	GEAR PD	20.0909	
Pinion X1	0.1500	0.1550	0.2200	0.2400	0.2600	Pinion X1	0.2900	0.3000	0.2650	0.2900	0.3000	Pinion X1	0.2850	
PINION OD	6.888	6.340	5.801	5.466	5.078	PINION OD	4.761	4.274	3.892	3.631	3.366	PINION OD	6.729	
GEAR OD	12.183	12.412	12.951	13.286	13.567	GEAR OD	13.884	14.371	14.594	14.855	15.120	GEAR OD	20.637	
Mp	2.03	2.06	2.05	2.13	2.11	Mp	2.09	2.06	2.08	2.06	2.04	Mp	2.02	
Mf	1.59	2.33	2.33	1.58	2.13	Mf	2.13	2.13	2.01	2.91	2.91	Mf	1.88	
PINION HT	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	GEAR HT	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	GEAR KT	58-62 Rc	
GEAR HT	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	PINION HT	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	58-62 Rc	PINION HT	58-62 Rc	
AGMA Q#	11	11	11	11	11	AGMA Q#	11	11	11	11	11	AGMA Q#	11	
Cm	1.19	1.19	1.19	1.19	1.19	Cm	1.19	1.19	1.19	1.19	1.19	Cm	1.23	
PINION RPM	1800	1800	1800	1800	1800	PINION RPM	1800	1800	1800	1800	1800	PINION RPM	various	
PINION DUR. HP	3,008	2,841	2,473	2,195	2,040	PINION DUR.HP	1,817	1,497	1,305	1,150	893	PINION DUR.HP		
GEAR DUR. HP	3,097	2,938	2,574	2,295	2,132	GEAR DUR. HP	1,917	1,591	1,392	1,233	962	GEAR DUR. HP		
PINION STR. HP	4,056	3,214	2,851	2,518	2,222	PINION STR. HP	2,048	1,788	1,371	1,254	1010	PINION STR.HP		
GEAR STR. HP	4246	3373	2965	2649	2339	GEAR STR. HP	2158	1909	1525	1402	1149	GEAR STR.HP		
LS Pinion RPM	957	872	752	685	622	LS Pinion RPM	564	481	437	396	357	LS Pinion RPM		
LS Pinion DurHP	2786	2557	2232	2047	1874	LS Pinion DurHP	1711	1479	1353	1235	1120	LS Pinion DurHP		
LS Gear DurHP	2948	2705	2362	2165	1983	LS Gear DurHP	1811	1564	1432	1306	1185	LS Gear DurHP		
LS Pinion Str HP	3400	3112	2706	2474	2260	LS Pinion Str HP	2058	1770	1615	1470	1330	LS Pinion Str HP		
LS Gear Str HP	3674	3363	2924	2673	2442	LS Gear Str HP	2224	1912	1745	1588	1437	LS Gear Str HP		
Unit Dur. HP	2786	2557	2232	2047	1874	Unit Dur. HP	1711	1488	1305	1150	893	Unit Dur. HP		
Unit Str. HP	3400	3112	2706	2474	2222	Unit Str. HP	2048	1770	1371	1254	1010	Unit Str. HP		
Dur. SF to Cat	1.46	1.41	1.35	1.42	1.37	Dur. SF to Cat	1.44	1.39	1.40	1.29	1.16	Dur. SF to Cat		
Str. SF to Cat	1.78	1.72	1.64	1.72	1.62	Str. SF to Cat	1.72	1.65	1.47	1.41	1.31	Str. SF to Cat		
NCR dur	2068	1889	1661	1518	1390	NCR dur	1266	1097	1004	888	750	NCR dur		
NCR Str	2488	2139	1793	1644	1410	NCR Str	1302	1148	897	825	831	NCR Str		
dur increase	1.35	1.35	1.34	1.35	1.35	dur increase	1.35	1.36	1.30	1.30	1.19	dur increase	1.32	average
strength increase	1.37	1.46	1.51	1.50	1.58	strength increase	1.57	1.54	1.53	1.52	1.22	strength increase	1.48	average

Table 6 Bearing life (L10) summary with 25° conventional gearing (2H155 gearbox)

UNIT RATIO>	6.3076	6.8421	8.1092	8.7579	9.9522		10.7094	12.7973	13.8109	16.0105	17.2540	
	single helical	single helical	single helical	single helical	single helical		single helical	single helical	single helical	single helical	single helical	
ID #>	155H1	155H2	155H3	155H4	155H5	ID #>	155H6	155H7	155H8	155H9	155H10	ID #>
Catalog HP	239	216	187	169	154	Catalog HP	138	120	108	96	87	Catalog HP
CENTERS (mm)	109	109	109	109	109	CENTERS (mm)	109	109	109	109	109	CENTERS (mm)
CENTERS (in)	4.291	4.291	4.291	4.291	4.291	CENTERS (in)	4.291	4.291	4.291	4.291	4.291	CENTERS (in)
GEAR TEETH	59	60	64	64	64	GEAR TEETH	72	101	109	117	116	GEAR TEETH
PINION TEETH	32	30	27	25	22	PINION TEETH	23	27	27	25	23	PINION TEETH
RATIO	1.8438	2.0000	2.3704	2.5600	2.9091	RATIO	3.1304	3.7407	4.0370	4.6800	5.0435	RATIO
CENTERS (mm)	155	155	155	155	155	CENTERS (mm)	155	155	155	155	155	CENTERS (mm)
CENTERS (in)	6.102	6.102	6.102	6.102	6.102	CENTERS (in)	6.102	6.102	6.102	6.102	6.102	CENTERS (in)
GEAR TEETH	65	65	65	65	65	GEAR TEETH	65	65	65	65	65	GEAR TEETH
PINION TEETH	19	19	19	19	19	PINION TEETH	19	19	19	19	19	PINION TEETH
RATIO	3.4211	3.4211	3.4211	3.4211	3.4211	RATIO	3.4211	3.4211	3.4211	3.4211	3.4211	RATIO
At Cabin Rating						At Cabin Rating						At Cabin Rating
Shaft 1						Shaft 1						Shaft 1
Left.	3779	3720	4881	4111	2744	Left.	2744	3070	9340	6448	15511	Left.
Right	134463	>200k	>200k	>200k	>200k	Right	>200k	>200k	>200k	>200k	>200k	Right
Shaft 2						Shaft 2						Shaft 2
Left	4191	8223	5300	6877	7892	Left	10358	10321	9674	11205	10699	Left
Right	4333	8014	5630	6501	6468	Right	7792	8343	10499	10820	13262	Right
Shaft 3						Shaft 3						Shaft 3
Left	197885	>200k	>200k	>200k	>200k	Left	>200k	>200k	>2001(	>200k	>200k	Left
Right	18504	42183	22267	24752	25214	Right	26905	25209	28683	27498	30517	Right
Bearing HP SF	1.27	1.29	127.	1.24.	1.39	Bearing HP SF	1.39	1.35.	1.03.	1.15	1	Bearing HP SF
for 10,000 has L-10.	187.7	167.6	159.2	136.8	110.6	for 10,000 has L-10.	99.3	88.2	105.3	83.8	87.5	for 10,000 has L-10.

Table 7 Bearing life (L10) summary with HCR gearing (2H155 gearbox)

UNIT RATIO>	6.3076	6.8421	8.1092	8.7296	10.0000		10.7519	12.8014	13.8109	16.0526	17.2540	
	single helical	single helical	single helical	single helical	single helical		single helical	single helical	single helical	single helical	single helical	
ID #>	155H1	155H2	155H3	155H4	155H5	ID #>	155H6	155H7	155H8	155H9	155H10	ID #>
Catalog HP	239	216	187	189	154	Catalog HP	138	120	108	96	87	Catalog HP
CENTERS (mm)	109	109	109	109	109	CENTERS (mm)	109	109	109	109	109	CENTERS (mm)
CENTERS (in)	4.291	4.291	4.291	4.291	4.291	CENTERS (in)	4.291	4.291	4.291	4.291	4.291	CENTERS (in)
GEAR TEETH	59	60	64	74	76	GEAR TEETH	88	116	109	122	166	GEAR TEETH
PINION TEETH	32	30	27	29	26	PINION TEETH	28	31	27	26	23	PINION TEETH
RATIO	1.8438	2.0000	2.3704	2.5600	2.9091	RATIO	3.1304	3.7407	4.0370	4.6800	5.0435	RATIO
CENTERS (mm)	155	155	155	155	155	CENTERS (mm)	155	155	155	155	155	CENTERS (mm)
CENTERS (in)	6.102	6.102	6.102	6.102	6.102	CENTERS (in)	6.102	6.102	6.102	6.102	6.102	CENTERS (in)
GEAR TEETH	65	65	65	65	65	GEAR TEETH	65	65	65	65	65	GEAR TEETH
PINION TEETH	19	19	19	19	19	PINION TEETH	19	19	19	19	19	PINION TEETH
RATIO	3.4211	3.4211	3.4211	3.4211	3.4211	RATIO	3.4211	3.4211	3.4211	3.4211	3.4211	RATIO
At Cabin Rating						At Cabin Rating						At Cabin Rating
Shaft 1						Shaft 1						Shaft 1
Left	3997	3942	5162	8341	6935	Left.	7736	11426	16818	15078	16530	Left.
Right	>200k	>200k	>200k	>200k	>200k	Right	>200k	>200k	>200k	>200k	>200k	Right
Shaft 2						Shaft 2						Shaft 2
Left	4376	5390	5537	6310	6591	Left	8202	7844	8855	9452	11189	Left
Right	4724	5494	6169	7367	7351	Right	8956	9828	11919	12464	14542	Right
Shaft 3						Shaft 3						Shaft 3
Left	62257	>200k	>200k	>200k	>200k	Left	>200k					Left
Right	6355	22681	8010	26434	24200	Right	28132	26659	30363	28859	32259	Right
Bearing HP SF	1.24	1.27	1.14	1.17	1.15	Bearing HP SF	1.09	1.08	1.04	1.02	0.97	Bearing HP SF
for 10,000 has L-10	193.1	170.6	163.3	144.8	134	for 10,000 has	126.8	L-10. 110.6	104.1	94.2	90	for 10,000 has L-10.

Table 8 Bearing life (L10) summary with 25° conventional gearing (2H330 gearbox)

UNIT RATIO>	6.4316	7.0955	8.1579	8.9961	9.8966		10.9211	12.7918	14.0789	15.5373	17.1053	
	single helical	single helical	single helical	single helical	single helical		single helical	single helical	single helical	single helical	single helical	
ID #>	330H1	330H2	330H3	330H4	330H5	ID #>	330H6	330H7	330H8	330H9	330H10	ID #>
Catalog HP	1,910	1,810	1,650	1,440	1,370	Catalog HP	1,190	1,070	934	889	770	Catalog HP
CENTERS (mm)	226	226	226	226	226	CENTERS (mm)	226	226	226	226	226	CENTERS (mm)
CENTERS (in)	8.898	8.898	8.898	8.898	8.898	CENTERS (in)	8.898	8.898	8.898	8.898	8.898	CENTERS (in)
GEAR TEETH	47	56	62	71	81	GEAR TEETH	83	86	107	109	95	GEAR TEETH
PINION TEETH	25	27	26	27	28	PINION TEETH	26	23	26	24	19	PINION TEETH
RATIO	1.8800	2.0741	2.3846	2.6296	2.8929	RATIO	3.1923	3.7391	4.1154	4.5417	5.0000	RATIO
CENTERS (mm)	330	330	330	330	330	CENTERS (mm)	330	330	330	330	330	CENTERS (mm)
CENTERS (in)	12.992	12.992	12.992	12.992	12.992	CENTERS (in)	12.992	12.992	12.992	12.992	12.992	CENTERS (in)
GEAR TEETH	65	65	65	65	65	GEAR TEETH	65	65	65	65	65	GEAR TEETH
PINION TEETH	19	19	19	19	19	PINION TEETH	19	19	19	19	19	PINION TEETH
RATIO	3.4211	3.4211	3.4211	3.4211	3.4211	RATIO	3.4211	3.4211	3.4211	3.4211	3.4211	RATIO
At Cabin Rating						At Cabin Rating						At Cabin Rating
Shaft 1						Shaft 1						Shaft 1
Left	3098	1546	4675	1903	4265	Left.	5340	5066	6277	4595	5677	Left.
Right	151,608	125,077	166.842	166,489	192,793	Right	>200,000	>200,000	>200,000	>200,000	>200,000	Right
Shaft 2						Shaft 2						Shaft 2
Left	3678	4758	4672	5633	3560	Left	4512	4435	5965	5565	7743	Left
Right	10,175	5,698	5,599	7,179	7,231	Right	9,258	9,233	11,550	10,872	13,889	Right
Shaft 3						Shaft 3						Shaft 3
Left	>200,000	>200,000	>200,000	>200,000	>200,000	Left	>200,000	>200,000	>200,000	>200,000	>200,000	Left
Right	60,283	52,315	56,289.	70,668	66,750	Right	84,823	80,868	97,100	86,785	106,960	Right
Bearing HP SF	1.34	1.65	1.41	1.55	1.21	Bearing HP SF	1.14	1.28	1.19	1.19	1.19	Bearing HP SF
for 10,000 has L-10	1423.8	1099.6	1169.4	932	1129	for 10,000 has L-10.	1046	838.1	786.3	748.3	649.6	for 10,000 has L-10.

Table 9 Conventional gearing (2H155 gearbox)

UNIT RATIO>	6.4091	7.0381	8.1575	8.9646	9.8620		10.8829	12.7470	14.0297	15.4830	17.2004	
	single helical	single helical	single helical	single helical	single helical		single helical	single helical	single helical	single helical	single helical	
ID #>	330H1	330H2	330H3	330H4	330H5	ID #>	330H6	330H7	330H8	330H9	330H10	ID #>
Catalog HP	1,910	1,810	1,650	1,440	1,370	Catalog HP	1,190	1,070	934	889	770	Catalog HP
CENTERS (mm)	226	226	226	226	226	CENTERS (mm)	226	'226	226	226	226	CENTERS (mm)
CENTERS (in)	8.898	8.898	8.898	8.898	8.898	CENTERS (in)	8.898	8.898	8.898	8.898	8.898	CENTERS (in)
GEAR TEETH	47	64	67	71	81	GEAR TEETH	83	86	107	109	111	GEAR TEETH
PINION TEETH	25	31	28	27	28	PINION TEETH	26	23	26	24	22	PINION TEETH
RATIO	1.8800	2.0645	2.3929	2.6296	2.8929	RATIO	3.1923	3.7391	4.1154	4.5417	5.0455	RATIO
CENTERS (mm)	330	330	330	330	330	CENTERS (mm)	330	330	330	330	330	CENTERS (mm)
CENTERS (in)	12.992	12.992	12.992	12.992	12.992	CENTERS (in)	12.992	12.992	12.992	12.992	12.992	CENTERS (in)
GEAR TEETH	75	75	75	75	75	GEAR TEETH	75	75	75	75	75	GEAR TEETH
PINION TEETH	22	22	22	22	22	PINION TEETH	22	22	22	22	22	PINION TEETH
RATIO	3.4091	3.4091	3.4091	3.4091	3.4091	RATIO	3.4091	3.4091	3.4091	3.4091	3.4091	RATIO
At Cabin Rating						At Cabin Rating						At Cabin Rating
Shaft 1						Shaft 1						Shaft 1
Left	3295	3492	2544	6705	4549	Left.	7,018	5,399	6,669	6,030	7,284	Left.
Right	179,014	>200,000	170,958	>200,000	>200,000	Right	>200.000	>200,000	>200,000	>200,000	>200,000	Right
Shaft 2						Shaft 2						Shaft 2
Left	3828	3982	3862	3,609	3,800	Left	4820	4,740	6,387	5,962	7,663	Left
Right	6797	6,631	6,477	8,577	8,011	Right	10,271	10,262	12,862	12,119	15,720	Right
Shaft 3						Shaft 3						Shaft 3
Left	>200,000	>200,000	>200,000	>200,000	>200,000	Left	>200,000	>200,000	>200,000	>200,000	>200,000	Left
Right	65,413	62,817	61,278	77,276	72,283	Right	92,339	88,069	105,728	94,466	116,134	Right
Bearing HP SF	1.31	1.29	1.42	1.34	1.34	Bearing HP SF	1.24	1.25	1.15	1.17	1.10	Bearing HP SF
for 10,000 has L-10	1456	1402.8	1165.3	1077.2	1022.5	for 10,000 has	956.3	L-10. 854.2	815.3	759	700	for 10,000 has L-10.

Conclusion

The industry has devoted much of the last 140 years to exploiting the "standard," full-depth tooth form. We moved from simple cast teeth to highly modified carburized and ground ones as market demands grew and technology evolved. An opportunity now exists to increase the capacity of our products by 25% or more, while simultaneously meeting stringent noise standards, through the adoption of a deeper than "full-depth" tooth geometry that has already been successful in aerospace and vehicle equipment. **PTE**

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Global Industrial Outlook: Oil Slick, Currency Headwinds Worsen

By Brian Langenberg, CFA

End market conditions for the power transmission industry continue to worsen. With the Euro down 13% year to date and U.S. oil production surging we are seeing increasing headwinds, if not storm clouds, for the sector.

***These three factors are challenging to growth*:**

- Oil
- Currency
- War

Oil. The general consensus remains for a (25%) reduction in 2015 capital spending by global oil companies, but those forecasts implicitly assume at least some recovery in oil price from curtailed exploration activity. Unfortunately, cuts in natural gas fracturing — even 4-6% in a week — do not boost oil price.

Currency. The Japanese Yen is already a challenge in the Middle East for U.S. construction equipment manufacturers. Now Europe is about to benefit with the Euro down 13% year to date; expect the Old World to gain an incremental advantage in exporting capital equipment. Not just cranes and excavators — start to think about the advantage to Airbus which sells in dollars and manufactures largely in Euros.

War. The war between ISIS, coupled with lack of U.S. leadership, is a threat to the development and sustainment of oil supplies directly in Iraq and Libya in the near-term. In Europe, Putin continues to rattle his saber with no obvious U.S. confrontation. Conversely, the strong dollar/weak Euro impact of oil/conflict diplomacy will likely end up helping European competitiveness.

OUTLOOK

Here is our outlook for key geographic regions and end markets:

U.S. remains best growth spot. Non-residential construction, consumer durables (auto, housing) and gradually improving employment will offset weaker commodity based demand. Conversely exports will likely start taking a hit.

Europe. Taking a marginal hit (Nordics, resource related parts of German economy) from conflict in Eastern Europe. Conversely the weak Euro should prove a boon for Germany, France and others.

Middle East. Oil & Gas activity should remain strong — even with production cuts — because mature fields require more capital and the region is seeking to capture more of the value stream. Increased Japanese construction equipment competition is a negative for U.S. manufacturers.

Latin America. Mexico is doing well; the rest of the region is seeing the usual political unrest you see when commodity-driven economies are whacked by low-commodity prices. Expect no near-term improvement.

Oil & Gas. One month ago WTI (West Texas Intermediate) was at a "depressed" $52; now we are looking at $45 — so expect more capital spending cuts.

Mining. *Still* awful. Aftermarket is now stabilized despite cash burn at coal companies, but low oil price further impacts coal, as does low steel price (Chinese glut) exacerbated by the strong dollar.

Power generation. U.S. power generation remains weak owing to efficiency gains throughout the economy and lack of regulatory support for new construction. Globally, the industry looks good — including coal and gas.

Transportation infrastructure. More pothole filling; no major infrastructure upgrade anytime soon. While I've heard rumblings of a multi-year highway bill, reality suggests other factors, e.g. — '16 Presidential election, ObamaCare disruptions, etc. — would make for a "status quo" or "status quo with certainty" outcome vs. a big infrastructure rebuild. I would love to be wrong. I doubt I will be wrong.

Machinery: Nobody is feeling the love outside replacement demand for U.S. trucks along with modest incremental demand from non-residential and residential construction markets (cement mixers).

Consumer (auto, appliances). Old cars = continued U.S. strength. Auto related end markets will remain solid. Auto investment in Latin America, particularly Mexico, continues to increase. U.S. residential recovery is on-track and will further support construction equipment demand. Weak Euro could start to impact exports.

Aerospace/Defense. Strong commercial build rates, coupled with two significant wars and depleted U.S. inventories, will continue to support a continued recovery in aftermarket activity. Long-term we expect a U.S. defense recapitalization — but not before 2017 authorization, given the current Administration. Foreign policy matters — and messes — are out there to be cleaned up. As an offset, the stronger dollar hurts Boeing and advantages Airbus as they both sell in dollars but, as mentioned, Airbus manufacturers primarily in Euros.

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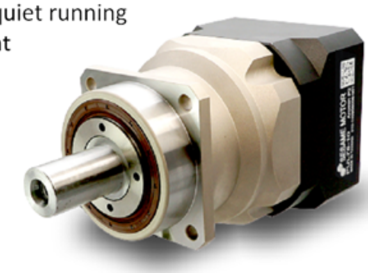

Bill Stevens

RETIRES AS CHAIRMAN OF MOTION INDUSTRIES

After 37 years at Motion Industries, Chairman **Bill Stevens** announced his retirement.

"Bill was the CEO at Motion Industries for 18 years, and Motion has grown to become one of the leading industrial distribution companies in North America, largely attributable to his leadership," said Tom Gallagher, chairman and CEO of Motion's parent, Genuine Parts Company. "We wish him the very best in his retirement."

Tim Breen, named president and chief operating officer in 2013, then president and chief executive officer in 2014, will succeed Stevens to lead the company. The two have worked closely together during the last few years to ensure a smooth transition.

"Tim brings significant experience and skills to lead Motion Industries," Stevens said. "It will be exciting to see the future, as Tim and his management team continue the wonderful tradition of our company."

Stevens began his career with Motion Industries in 1978 as director of human resources. After several promotions to various other roles, he was named president and COO before becoming president and CEO in 1997. With Stevens at the helm, Motion Industries' sales grew from $1.85 billion (1997) to $4.8 billion in 2014.

A portion of this growth, in both sales and footprint, can be attributed to the 38 acquisitions completed during Stevens' tenure as president and CEO. Growth in product offerings, program introductions and system advancements since 1997 has also contributed to Motion's expansion under Stevens' leadership.

Stevens received a number of personal awards, including 1997 and 2006 Genuine Parts Company Manager of the Year, and the 2014 Bearing Specialists Association (BSA) Lifetime Achievement Award. He has been a BSA member since 1989 and has served in many capacities. Stevens has also been an active member of the Power Transmission Distributors Association (PTDA) for many years, as well as serving on numerous boards of both for-profit and nonprofit organizations in the Birmingham area.

CRP Industries

MARKS 60-YEAR MILESTONE

CRP Industries Inc., a company in the automotive parts and industrial products sectors, recently marked its 60th anniversary.

CRP was founded in 1954 as Conti Rubber Products, and was, in essence, Continental Tire's initial outpost in the US. In 1977, CRP became the NAFTA market general agent for Continental AG's ContiTech division, and as it broadened its supplier base, the company changed its name to CRP Industries Inc.

Today, CRP Industries has two divisions, CRP Automotive and CRP Industrial, and provides products from some of the most recognized brand names in the NAFTA market. CRP Automotive features ContiTech Automotive Belts, Rein Automotive Parts, Pentosin Technical Fluids, AJUSA Gaskets and Head Bolts. CRP Industrial provides, Reinflex High-Pressure Thermoplastic Hoses, TUDERTECHNICA Rigid Mandrel Hoses, and Perske High-Speed Motors.

Since its inception, CRP has demonstrated steady growth, expanding its operations from a small organization carrying approximately 200 products to a multi-faceted supplier handling well over 14,000 parts and systematically distributing them from four strategic locations in the NAFTA region. CRP now employs over 120 people and operates facilities in New Jersey, California, Canada, and Mexico.

"We have been very fortunate to have great customers, supplier partners, and employees over the years," said Daniel Schildge, president of CRP. "We have shown a supporting vision that could evolve over time and a commitment to our core values of quality, service, and trust. It is these ideals and partnership opportunities that have allowed CRP to thrive for this long."

Bosch Rexroth

ACCEPTING REGISTRATION FOR HYDRAULICS TRAINING PROGRAM

Bosch Rexroth is now accepting registration for all courses in its training program designed for industrial hydraulics and mobile hydraulics systems engineers.

Training is available on-site for specific hydraulic installations, and customized training programs are available with test stands delivered to the participant's location. Scheduled courses are held at select Bosch Rexroth facilities in the U.S. and Canada, and online self-directed eLearning courses are also available.

Courses are held at Bosch Rexroth facilities in Bethlehem, PA, Houston, TX, and Greenville, SC, as well as at various locations in Canada.

Minnesota Rubber and Plastics

POSITIONS FOR GROWTH

As part of a global corporate strategy to better position for continued growth, Minnesota Rubber and Plastics recently realigned several of its manufacturing operations.

Through a combination of facility upgrades, expansions and consolidations, Minnesota Rubber and Plastics has improved overall efficiencies across the manufacturing group. Recent actions include the expansion of manufacturing in Reynosa, Mexico and Mason City, IW, the relocation of processes from the Watertown, SD and Irvine, CA facilities and the subsequent closing of those locations. In addition to more efficient operations, these expansions strengthen Minnesota Rubber and Plastics capabilities within its key customers industries of water, medical and pharma, power and transportation.

"These moves reflect the company's commitment to growth, and to better meet our customers' needs," said Lih Fang Chew, Minnesota Rubber and Plastics global vice president of marketing. "We expect to see the benefits of these actions throughout 2015 and beyond."

PTDA Foundation

NOW ACCEPTING NOMINATIONS FOR 2015 WENDY B. MCDONALD AWARD

To honor the memory of **Wendy B. McDonald,** one of the power transmission/motion control industry's pioneers, the PTDA Foundation established the Wendy B. McDonald Award in 2014. The award acknowledges a woman who has established herself as a critical contributor to her company's success and has affected positive change within the power transmission/motion control industry.

As a female business owner, McDonald left many legacies through her long career in the industry. The inaugural recipient of the Wendy B. McDonald Award in 2014 was Pat Wheeler of Motion Industries (Canada).

When merited, the Wendy B. McDonald Award will be presented annually during the PTDA Industry Summit. Nominations are now being accepted through May 31, 2015, and will be judged by the following criteria:

Nominees must be female and employed by a PTDA member company within the calendar year for which the nomination is being made. There are no criteria with respect to title, position in company or years of experience.

Nominees must exemplify leadership and integrity in all business relationships.

Although all nominees are considered, those employed by Canadian companies or distributors receive extra consideration.

Lenze Americas

NAMES ROBERT GRADISCHNIG DIRECTOR OF TECHNICAL SERVICES

Lenze Americas recently announced the appointment of **Robert Gradischnig** to the position of director, technical services. As the newest member of the Lenze Americas business leadership team, Gradischnig is chiefly responsible for Lenze technical services and end-to-end support of mechanical and system engineering customers in North America.

"Robert's outstanding sales achievements, coupled with his engineering background, give him a unique perspective and approach to technical support through all phases of the machine building process," said Chuck Edwards, president, Lenze Americas. "His ability to rethink existing systems translates into the implementation of better solutions to balance efficiency and productivity—and deliver maximum benefit to OEM customers and plant operators."

Gradischnig completed technical engineering studies at HTBLA Elektrotechnik and additional business economics studies at the University Mittweida, Germany. After joining the Lenze Austria team in 1993 as an application engineer, he earned a key account manager position there, before transferring in 2011 to a business development role driving marketing strategy and sales of electromechanical products at Lenze in the Americas.

Timken

AWARDS SCHOLARSHIPS VALUED AT $540,000

The Timken Company recently awarded scholarships to 17 sons and daughters of Timken associates in 11 locations around the world. The Timken Company Charitable and Educational Fund, Inc. funds the scholarships, with a total value of up to $540,000 over four years. The program has awarded more than $21 million in scholarships since its founding in 1958.

Chairman John M. Timken, Jr., hosted a recognition event at World Headquarters in North Canton, OH involving students and their parents. Local scholarship finalists attended the event in person, while other finalists and their parents joined by a global webcast.

"As we continue our company's long-standing tradition of promoting education and know-how for our associates and their families, it's an honor to invest in such outstanding students and support them in pursuing their dreams," Timken said. "The 2015 Timken Global Scholars join an elite group, and today we recognize our newest scholars for their impressive academic achievements, extracurricular activities and community commitment."

The $35,000 Henry Timken Scholar Award, which recognizes the top-ranked applicant, is renewable for up to three additional years for a total of $140,000. This year's Henry Timken Scholar was Bogdan Konnerth, the son of Octavian Konnerth, Timken service engineering manager based in Ploiesti, Romania. Bogdan, a graduating senior at Ion Luca Caragiale National College in Ploiesti, plans to study aerospace engineering at the University of Liverpool.

The $25,000 Jack Timken Scholar Award was presented to Mianna Schut, the daughter of Jeffrey Schut, principal inventory planner at World Headquarters. This new award, recognizing the dedicated service of Ward J. Timken, who retired last year from The Timken Company's Board of Directors, is renewable for up to three additional years for a total of $100,000. After graduation from GlenOak High School in Canton, Ohio, Mianna plans to study nursing at Walsh University.

Five students received $10,000 scholarships, renewable for up to three additional years for a total value of $40,000 each.

Aleks Roudnev

NAMED HYDRAULIC INSTITUTE'S 2014 'MEMBER OF THE YEAR'

The Hydraulic Institute (HI) recently named Aleks Roudnev, manager of research and design — applied hydraulics, at Weir Minerals North America as its 2014 "Member of the Year." The award was conferred to Roudnev at the Hydraulic Institute's 2015 annual meeting held recently in St. Petersburg, FL.

The HI Board of Directors selected Roudnev for his commitment to the Hydraulic Institute in both leading and actively participating in numerous HI committees in the advancement of the Institute's technical work as well as in guiding young engineers within the Institute's extensive technical organization. As an active member of HI since 1995, his recent contributions include serving as chair of the HI Standards Committee, the Slurry Pumps Committee, and he serves as vice chair of the HI General Guidelines Committee.

"As an active member of HI for nearly twenty years, Aleks has been actively involved in the different committees that support the future development of standards for the pump system industry as well as mentoring young engineers within the Institute," said Robert Asdal, executive director, Hydraulic Institute, "His ongoing participation and leadership exemplifies the qualities that distinguish an HI Member of the Year."

April 29—SMMA Presents: An Overview of Various Electric Motors and Motion Control Technologies Villas of Grand Cypress, Orlando, FL. This new SMMA Motor & Motion College course will provide fundamental concepts of electric motors and their electronic control methods. Intended for non-technical professionals such as sales and marketing, middle and upper management, and application engineers who want to learn fundamental principles and basic knowledge, course content covers electromagnetic torque production theory, construction and operation of all motor types, their relative differences, as well as electronic hardware topologies, field oriented and sensorless control algorithms, and servo system theory in clear and concise explanation without using lengthy equations. The instructor for An Overview of Various Electric Motors and Motion Control Technologies is Dal Y. Ohm, PhD, president of Drivetech, Inc., a technical consulting firm specializing in the design and development of motor control, drives, and renewable energy converter systems. Dr. Ohm has more than 25 years of industrial and academic experience in research and product development in motor drives, servo systems, and power converters. For more information, visit *www.smma.org*.

April 29-May 1—2015 AGMA/ABMA Annual Meeting The Meritage Resort and Spa, Napa Valley, CA. This year's Annual Meeting will address the key issues facing manufacturing and offer opportunities to network, make memories, forge relationships, and build on future partnerships. Napa Valley provides much to explore and many attendees will bring a spouse or guest. In lieu of the golf tournament, the planning committee opted to keep open the second afternoon for exploration of this unique location. For more information, visit *www.agma.org*.

May 4-6—2015 Gearbox System Design: The Rest of the Story...Everything but the Gears and Bearings Sheraton Sand Key Resort, Clearwater Beach, FL. This course focuses the supporting elements of a gearbox that allow gears and bearings to do their jobs most efficiently. Learn about seals, lubrication, lubricants, housings, breathers, and other details that go into designing gearbox systems. This seminar starts with the basics including some history of design and the varied environments to which gearbox systems are built. It continues by teaching detailed design layout. And it then will focus on individual pros and cons of types of housing construction, drawing practices for housings and related components and selection and role of gearbox accessories, such as breathers, filters, screens, sight gages and other level indication devices. For more information, visit *www.agma.org*.

May 4-7—Offshore Technology Conference NRG Park, Houston, TX. The Offshore Technology Conference (OTC) is where energy professionals meet to exchange ideas and opinions to advance scientific and technical knowledge for offshore resources and environmental matters. Founded in 1969, OTC's flagship conference is held annually at NRG Park (formerly Reliant Park) in Houston. OTC has expanded technically and globally with the Arctic Technology Conference, OTC Brasil, and OTC Asia. OTC is sponsored by 13 industry organizations and societies, who work cooperatively to develop the technical program. OTC also has endorsing and supporting organizations. OTC gives you access to leading-edge technical information, the industry's largest equipment exhibition, and valuable new professional contacts from around the world. Its large international participation provides excellent opportunities for global sharing of technology, expertise, products, and best practices. OTC brings together industry leaders, investors, buyers, and entrepreneurs to develop markets and business partnerships. For more information, visit *2015.otcnet.org*.

May 6-7—Design-2-Part Show Schaumburg Convention Center, Schaumburg, IL. Manufacturers nationwide rely on Design-2-Part Shows as the most efficient place to meet high-quality, reliable American job shops and contract manufacturers. In just a few hours you can find cost effective, quality suppliers, learn about new technologies and materials, see and compare parts and components, and quote jobs and evaluate quality price while getting delivery on the spot. Finding trustworthy contract manufacturers who can provide flexible, cost-effective solutions and scale up growing organizations is critical. There is no better way to identify new outsourcing partners than spending even one half-day at this show. For more information, *visit www.d2p.com*.

May 18-21—AWEA WINDPOWER 2015 Conference & Exhibition Orange County Convention Center, Orlando, FL. Wind energy's premiere industry gathering is a concentration of expertise and innovation that draws thousands of professionals from around the world to trade knowledge, experience and best practices across all industry segments. And in 2015, WINDPOWER will address what you can do now to meet the challenges of today, while preparing for tomorrow. The WINDPOWER 2015 cutting-edge program presents and examines the technical developments and evolving issues that are transforming the industry and increasing the competitiveness of wind power. AWEA has assembled top industry experts for this conference, who will present deep-dive sessions on a variety of relevant topics. For more information, visit *www.windpowerexpo.org*.

June 22-25—2015 Automation Summit Aria Resort, Las Vegas, NV. The Summit will again feature the popular Connect Event, where attendees can network in-person and virtually with Siemens employees, solution providers, integrators, distributors and end users. Innovator and business futurist Nicholas J. Webb will provide the keynote address. Webb, author of "The Innovation Playbook" and "The Digital Innovation Playbook," is a successful inventor with a wide range of patented technologies including one of the world's smallest medical implants and the popular Hanz line of educational toys. For more information or to register, *visit http://sie.ag/1AOihUw*.

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4 → How is THIS LOCATION involved with power transmission products? (Check all that apply)

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- ☐ WE DESIGN products with power transmission components in them. (16)
- ☐ Other (please describe) (15)________________________

5 → My company's principal product or service is:________________________

6 → What is your primary job function responsibility? (Check one)

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- ❑ Design Engineering (3)
- ❑ Marketing & Sales (4)
- ❑ Manufacturing Engineering (5)
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- ❑ Factory Automation (8)
- ❑ Maintenance (9)
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What's In a Meme?

Rosetta Mission mixes with Maxon, Kim K for one unforgettable photo

Erik Schmidt, Assistant Editor

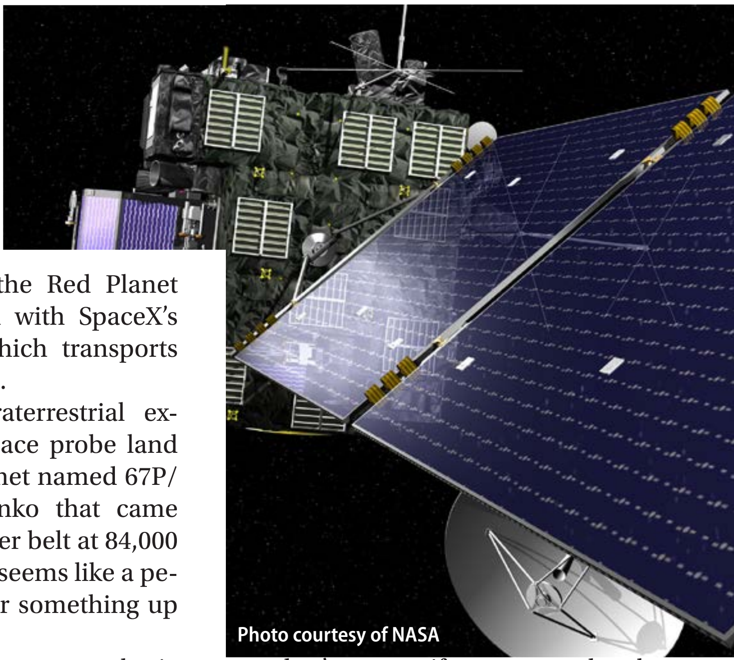
Photo courtesy of NASA

It was a perilous — some would even say *impossible* — undertaking, but on November 12, 2014, Maxon Motor braved the hazardous expanses of space and landed on that mammoth, brownish surface previously untouched by European space vessels.

Maxon had gone where no motor company had gone before:

Kim Kardashian's backside.

And all at once, as the Photoshopped image of a tiny *Rosetta* space probe rested unceremoniously on the — *ahem* — profound posterior of Kanye West's wife, it was clear that Maxon had made it BIG. In a scientific, pioneering sense, of course. But also in the social media-listic, meme-tastic sense (a "meme," by the way, is defined as "an idea, behavior, or style that spreads from person to person within a culture") that transcends the usually constrictive parameters of the power transmission industry and into the boundless world that exists *out there*.

Where?

There.

The place that doesn't forget even when most people do; the place that will live on, powered by "likes" and page views, when we're all dead and in the ground and resting eternally under the all-knowing Cloud.

Yes, Maxon is now a permanent inhabitant of Viral Village, the Internet Age's version of the Hamptons.

But well before that, before Maxon became a buzzword that filled Twitter timelines and Facebook status updates, it was simply a company from Switzerland that made motors, drives and systems of up to 500 W.

Talk about humble beginnings.

Maxon, though, has never been satisfied with staying Swiss-bound (or earthbound, for that matter). For years, Maxon has aided in space missions — first with the Mars rovers that have been perusing the Red Planet since the 1970s, then with SpaceX's Dragon spacecraft, which transports cargo to the ISS station.

Maxon's latest extraterrestrial excursion? Helping a space probe land on a 2.7 mile-long comet named 67P/Churyumov-Gerasimenko that came hurtling from the Kuiper belt at 84,000 mph. If somehow that seems like a pedestrian feat, let's clear something up straightaway: it wasn't.

For ten years, *Rosetta* — a robotic space probe built and launched by the European Space Agency — tailed the comet with steadfast yet futile determination, like an overeager pooch trying to chase down the mail truck. On August 6, 2014 the decade of unbreakable fortitude paid off as the probe finally caught up with "Chury," becoming the first spacecraft to orbit a comet.

That, in itself, was a massive success. But Maxon wasn't done.

Three months later, *Rosetta's* lander, *Philae* — powered by two Maxon DC motors with a diameter of 13 mm each — touched down on the comet's surface. Yes, that was also a first.

And then, ever the overachiever, Maxon went viral.

(Going "viral," by the way, is defined as something "that becomes popular through a viral process of Internet sharing, typically through video sharing websites, social media and email").

The news of the mission's success — landing a manmade vessel on a *comet* isn't exactly something that happens every day — was the talk of the water cooler for 24 hours, slipping insidiously into countless conversations, rooting itself deep into hipster-speak lexicon, and culminating in this:

An unidentified meme wherein the *Rosetta* vessel was inserted brilliantly into that infamous, "break the internet" photo of Kim K from New York-based fashion and pop-culture magazine *Paper* — you know the one (and if you don't, your wife or son or daughter surely does): lavish pearl necklace coiled around Kardashian's tapered neck, elbow-length black gloves offset on top of coffee-colored skin, an impossibly large, oddly lathered derrière front and center in the world's most unsubtle cry for attention.

It was the picture that spawned a million memes.

Rosetta, and by association, Maxon, happened to be one of them.

Perhaps winding up inches from a faux celebrity's rump isn't everyone's idea of great success. It's true that the real story here lies in the science — being an integral part of a pioneering, historical foray into the far reaches of space is more than enough to garner long-lasting notoriety — but we live in a different world now, one in which visibility on social media often serves as the most important measuring stick.

So let's just say this:

Maxon, a world leader in high-precision drives and systems, continues to push the boundaries of how far motors can go. And that is an obvious and unabashed triumph.

But when the dust clears and the mission concludes at the end of 2015, don't be shocked (fairly or unfairly) when it's remembered more for the whirlwind of internet fanfare, the memes and tweets, and of course, Kim Kardashian's ability to stand in admirably for a giant comet.

And that too is a triumph. **PTE**


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