Power Transmission Engineering®

FEBRUARY 2014

Bearing Modeling and Simulation Software Examined

Gear Design "gooFwaRe"?

Technical

[How To Spec a Mill Drive]

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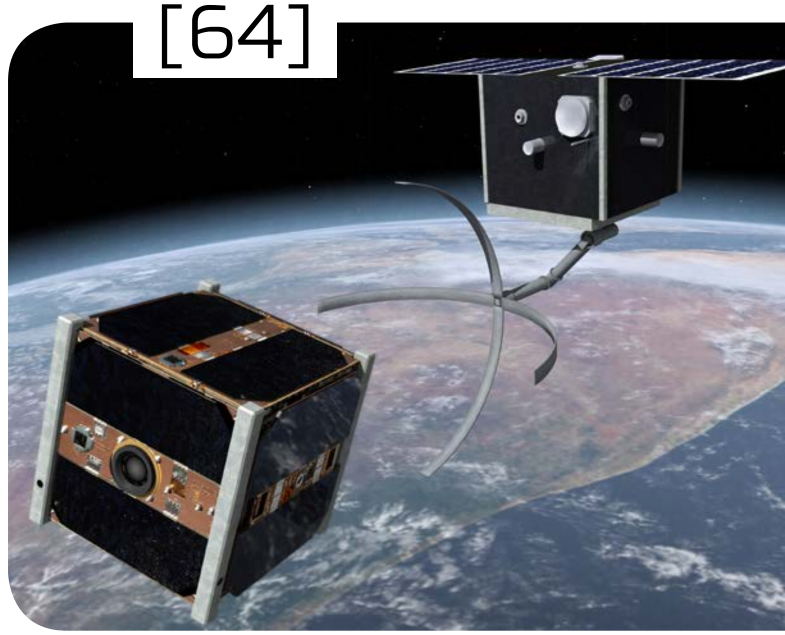

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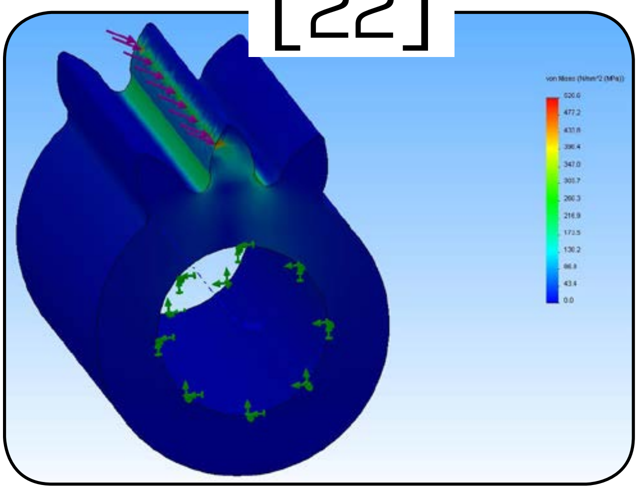

FEATURE ARTICLES

TECHNICAL ARTICLES

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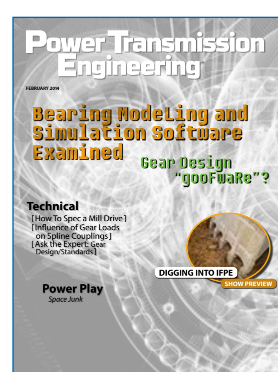
Power Transmission Engineering

VOL. 8, NO. 1



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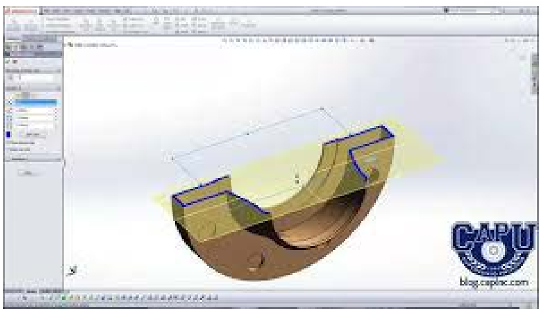
PTE Videos

The SKF Low Friction Hub Bearing Unit, a new wheel end concept developed in Italy, is a solution introduced by SKF to address the technology issues pertaining to both the chassis side and the engine/transmission side. Check out a behind-the-scenes video currently at *www.powertransmission.com*.

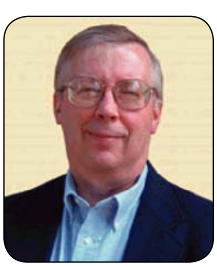


LinkedIn: Dassault Systemes has recently launched *SolidWorks Mechanical Conceptual*, focused on four key elements of design including: Conceptual, Instinctive, Social and Connected. Learn more about this on the PT LinkedIn page (*www.linkedin.com/groups?home=&gid=2950055&trk=anet_ug_hm*).

Twitter: The *PTE Twitter* feed keeps readers up-to-date on the latest products and services available to the PT community. Recent IFPE updates from Parker Hannifin, Bosch Rexroth and the NFPA are currently online. Join us at *https://twitter.com/PowerTransMag*.

Ask the Expert: Do you have a question about gears, bearings, motors, clutches couplings or other mechanical power transmission or motion control devices? Submit your questions to our panel of experts at: *www.powertransmission.com/asktheexpert.php*.

Gear Technology Blog



Charles D. Schultz is offering his insights into the gear manufacturing industry and asking readers to share their knowledge as well. Read his most recent post at *www.geartechnology.com/blog*.

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Make the Connection

Recently I had a disturbing conversation with a colleague here at the office. During the conversation, it became clear to me that my co-worker — a really intelligent guy whom I respect a lot — had no idea how even the most simple electric motor works.

He said something about all the gears that must be in electric motors, and in my head, I heard that noise an old vinyl LP makes when you scratch the needle across the tracks.

"Wait a minute," I said. "You know how an electric motor works, right?"

After all, we work at a publishing company that specializes in writing about engineered components – you know, gears, bearings, motors – that kind of thing.

I had to explain to my friend that, strictly speaking, there are no gears in an electric motor. Of course, there are such things as gearmotors, I said, but no matter how much they're integrated into one package, the gears and the motors are really separate devices.

"Didn't you ever take apart your slot cars when you were a kid?" I asked him.

When there was no reply, I followed up with, "Didn't you ever build an electric motor for a science fair project?"

You could almost hear the crickets chirping.

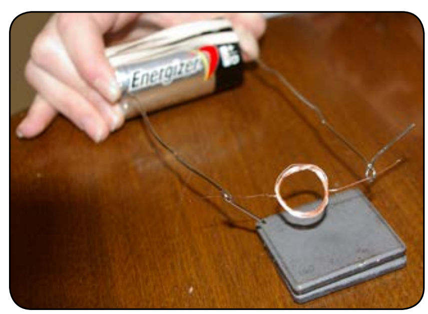
I can appreciate the fact that not everyone is as much of a science nerd as I am. The things that have interested me my entire life are often the same things that make other people's eyes glaze over. I guess many of you can probably relate. But I was shocked that a grown man who I consider to be an intelligent and well-rounded individual didn't know even the basics of what made a motor function.

So I took it upon myself to organize an office science project. One Friday, I went to the hardware store, picked up some wire, a few batteries and a couple of refrigerator clips with decent-sized magnets on the back. After prying the magnets off the refrigerator clips and pilfering a couple of paper clips, a magic marker and a rubber band from the supply closet, we had everything we needed.

The design is relatively simple. Bend the paper clips into shape to form an axle support that also connects to both terminals of the battery. Coil the wire (about 10 loops should do it), and wrap the loose ends around the coil to form the axle. Paint the top half of each axle with the magic marker to form a poor-man's commutator. Put it all together and hold the coil over the magnet, and —*voila*—you've got a working electric motor.

I imagine that most of you reading this *did* take apart your slot cars when you were little, so you probably don't need any more help than what I've given above, but for those who want more detailed instructions, all you have to do is search the Internet. You'll find many examples of simple motors similar to the one I built.

After completing the project, I felt somewhat better for having shared a little bit of knowledge and a passion for science with my co-workers. But I realized all along that I was talking to the wrong audience. Sharing that knowledge and passion is something we all need to do, especially with the young people in our lives.



So I took my science project home and showed it to my 11-year-old daughter, Renee. As soon as she saw it, her response was, "Let me try!" and "Can I take it to school and show my class?" So we're going to build a more permanent version of the simple motor, and yes, Renee, you can take it to class.

If you are interested in doing similar projects with the young people in your life, I recommend you visit the website of DiscoverE (*www.discovere.org*). They're the organization that sponsors Engineering Week (February 16-22). Although Engineering Week will be over by the time you read this, it's never too late to participate. They have hundreds of sample project ideas that are perfect for encouraging young minds, and most of them take no more time or effort than my simple motor.

So please find a young person and share with them your own passion for learning and exploration. I promise, you won't regret it.

Randy Stott

Need High-Quality Gears For Off-Highway Applications? Child's Play...At Forest City Gear

Construction, mining and agricultural vehicles and equipment are pushing into some pretty out-of-the-way places these days. Vital components like gears are expected to operate smoothly, quietly and around the clock in the harshest environments – and reliability is paramount. Forest City Gear is pushing into new frontiers as well – like helping develop and manufacture large gears for some of the leaders in these industries.

Let's have some fun together. Visit:
www.forestcitygear.com.

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11715 Main Street, Roscoe, IL 61073
815-623-2168

Lenze

OFFERS SIMPLE AND VERSATILE SMART MOTOR

Lenze Americas recently introduced its innovative Smart Motor designed for applications ranging from material handling to packaging automation. Blending the simplicity of a conventional AC motor with electronic drive control technology, Lenze's versatile Smart Motor was showcased at Pack Expo last September.

"In a complex world of motors, Smart Motor keeps it simple," states Robert Gradischnig, electro-mechanical business development manager, Lenze Americas. "The Smart Motor gives packaging customers the power and freedom they want in motion control, with freely adjustable speeds, programmable I/O and integrated functions for conveying needs. Fully programmable from a smart phone, the Smart Motor makes commissioning and diagnosis easier than ever."

Offering a highly versatile and powerful package, the Smart Motor delivers maximum standardization—drastically reducing (by up to 70 percent) the number of different motor variations customers might otherwise need. Equipped with control intelligence in the terminal box, the Smart Motor provides design flexibility for speeds from 500 to 2,600 rpm, while maintaining constant torque. The easy-to-install motor enables mains and starter applications to be handled with greater ease than with conventional motors.

The Smart Motor complies with IES2, the top efficiency class for drive systems, while maintaining a small footprint. Among its standard functions, the Smart Motor enables speed switchovers and the ability to set individual start-stop ramps. Integrated ramps ensure smooth acceleration and deceleration to protect the system mechanics and product being handled. The Smart Motor features full motor protection, eco mode for energy savings, and digital outputs for speed selection, direction and status messages.

"Packaging tasks often need higher starting torque, followed by continuous torque. The Smart Motor has been designed and dimensioned to specifically meet this requirement," adds Gradischnig. "By reducing the total number of different motors required for a given operation, our packaging customers can reduce the costs for engineering, documentation, procurement and spare parts—key aspects in the value-added chain."

For more information:
Lenze Americas Corporation
Phone: (508) 278-9100
www.lenzeamericas.com

TOOLS OF THE TRADE

With its Smart Motor Series, Lenze Americas has created an optimal solution for the conveyor industry, one that has generated some attention at trade shows including Pack Expo, SPS/IPC Drives and Hannover Messe. *Power Transmission Engineering* recently spoke with Eric Klein, director of sales at Lenze Americas, to discuss the other tools and resources available to the PT marketplace in 2014.

Lenze is really focused in three key areas, according to Klein: ease of engineering, mechatronics and safety. "These are the three main themes we focus on with our customers," Klein says. "Most companies don't have the design engineers that they used to have. From a machine builder perspective, we look at their buying process. Our goal is to come up with solutions that reduce energy costs, provide more throughput and reliability."

By lowering the cost of ownership, streamlining the design process and integrating a variety of new technologies, Lenze can provide the tools necessary to optimize products found in industries like automotive, robotics, packaging, material handling and pumps/fans. "We look at the business side of each and every customer and see what exactly they're trying to accomplish," Klein says. Some of the more recent tools available to customers include:

FAST: Lenze's standard software modules make it easy to develop modular mechanical control by simply adding the individual modules using the application template. "They might program a drive differently in Spain than they would in the United States," Klein says. "Each engineer would do things a little differently. *FAST* allows a machine builder to pick the applications that matter to them. The engineer can tweak it based on their machine and enhance it from there. What really matters is the engineering time. If they don't have to spend a lot of time re-designing drives, they can focus on what makes a machine better."

Easy Explorer: This updated selection tool can be used in the very early planning phase of a drive solution to provide a fast overview of suitable Lenze products. "No extensive product knowledge is needed," Klein says. "You enter some basic data and it will calculate your application's requirements and offer possible solutions."

Drive Solution Designer(DSD): Want to see your potential energy savings options? How about an overview of the complete drive structure for all of the machine's requirements? *DSD* makes sophisticated and complex dimensioning expertise available and ensures optimal energy efficiency gains. "You can actually model the energy consumption of each potential solution and make sure you're getting the kinematics and mechatronics right," Klein adds.

While technologies continue to converge in machine design, Lenze offers a variety of innovations and solutions to keep up with the growing trends and help machine builders become more efficient, productive and flexible.

Mitsubishi Electric

EXAMINES USER EXPECTATIONS IN MOTION CONTROL AND SERVO TECHNOLOGIES

BARRY WELLER, MITSUBISHI ELECTRIC

User expectations for motion control and servo technologies are changing, with more emphasis being placed on ease of installation, energy efficiency, safety and productivity. **Barry Weller** of Mitsubishi Electric looks at the current trends and the future of these vital technologies.

High-precision servomotors, intelligent servo amplifiers and high speed motion controllers have been available for many years, and have developed steadily over that time. There are many established uses for servos and motion control, and probably just as many potential new users that are yet to adopt the technology.

Today's emerging trends and new developments provide a guide to how servos are likely to advance in the coming years. Mitsubishi Electric's new MR-J4 servos, launched this month, embody many new features that are expected to become the norm over time.

One aspect that the market is always keen on is improving ease of use or "user-friendliness." Commonly one of the most time consuming parts of any servo system is tuning the servomotor to the machine it is installed in. The result of not tuning the servo properly is an underperforming machine or irregular production. Usually the person tuning the system needs a high level of both training and experience. By giving the servo amplifier the intelligence to do this tuning, the commissioning time is reduced. The MR-J4 Servo amplifier achieves this with its "One Touch Tuning Function;" using this function the engineer simply chooses from three different types of machine criteria and lets the servo amplifier do the rest. The servo amplifier then tunes to the machine and any machine resonance points are detected and automatically filtered out. Even machine vibration can be suppressed. Of course some of these resonance points may change over time due to machine wear. To overcome this, the fillers can be set to continually adjust over the life of the machine.

The latest amplifiers also have a "Machine Diagnosis Function" that monitors not just the actual condition of internal components such as capacitors and relays, but also external components like the bearings, belts or ball screws. The system then notifies the user regarding any degradation in operation. This makes both unplanned downtime and machine outage time a thing of the past and can therefore make a significant contribution to long-term productivity. If an alarm does occur, the internal drive recorder

gives the maintenance engineer full diagnostic information before and after the alarm occurrence. This makes the cause of the alarm easy and quick to identify, reducing any downtime.

Safe Operation

Functional safety requirements are continuing to evolve and become more demanding. Mitsubishi's MR-J4 amplifiers have an integrated two-channel STO (Safe Torque Off) input. This can be augmented by an additional option module to provide SS1 (Safe Stop) input in line with EN61800-5-2. The system configuration then also supports SIL2 (EN62061) and PLd (EN13849-1). Embedding the safety on the amplifiers gives greater flexibility in its use and also reduces machine start-up times after the cause of the safety stoppage has been resolved.

Servomotors have always been at the forefront of accuracy. This accuracy helps manufacturers reduce the amount of raw materials used in production and increase the product quality, so there is a constant drive to improve this accuracy. The MR-J4 amplifiers, for example, are being fitted with 22-bit absolute encoders as standard, providing a resolution that corresponds to more than --four million pulses per revolution. This resolution not only increases accuracy but also gives a smoother motor rotation, often known as reduced cogging.

Environment

Environmental issues are on the top of everyone's priory list and often we are looking at how technology can help reduce our carbon footprint. This could be looked at in many ways, from energy saving to the amounts of raw material used. For these reasons Mitsubishi Electric has developed dedicated two- and three-axis amplifiers that provide cost, space and energy savings, compared to using a set of single-axis amplifiers. These amplifiers also offer significant reductions in wiring and commissioning times.

The design of the servomotor itself is also being refined. One of the key ingredients in the motor construction is rare-earth magnets. The MR-J4 range of motors use 30 percent less metal than the previous generation, while still increasing performance.

The MR-J4 motors range is available in 200 V and 400 V and includes power ratings from 100 W to 22 kW. The MR-J4 servomotors are designed for compliance with either IP65 (dustproof), IP67 (wash-down) or to food and beverage hygiene standards.

Therefore, we can summarize that servos and motion control technologies have developed a long way over the last 20 years and are likely to continue developing for many years to come. Similarly, the market is evolving and growing as new application areas are opening up. There are many drivers for this, including steadily falling servo costs and rising performance levels, new fields being automated (e.g. medical procedures), servos replacing other technologies such as hydraulics or steppers, increasing awareness by user groups, and developing geographic regions.

The emerging generation of servo technology represents a significant advance on current standards, and we can be sure that new developments will continue to emerge for many years yet.

For more information:

Mitsubishi Electric UK
Phone: +(44) 1707 288769
www.mitsubishielectric.com

Emerson

OFFERS MORSE RAIDER PLUS SPEED REDUCERS

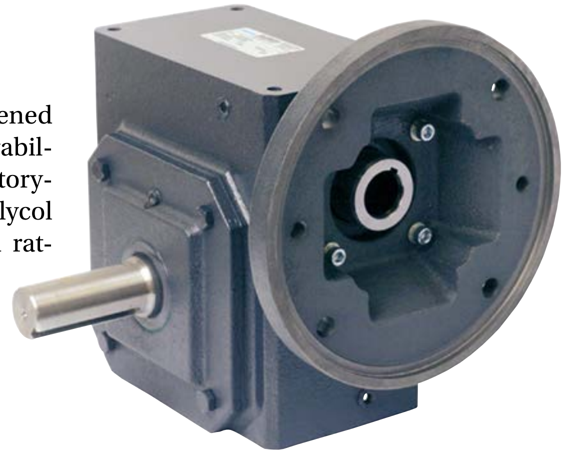
Emerson Industrial Automation's Morse Raider Plus speed reducers utilize advanced finishing techniques, additional bearings and special seals to arm them for leak-free performance, guaranteed for 18 months. Suitable for conveyors, mixers and bucket elevators in the food, beverage and grain industries, the Raider Plus computerized gear centering optimizes the gear mesh for reduced heat and debris formation, eliminating unnecessary backlash and delivering high efficiency and long life.

A patented roll-burnished journal finish extends seal life up to four-fold, and exclusive double-lip seals on both input and output shafts provide leak-proof operation in highly contaminated applications. The Raider Plus also features an exclusive non-metallic quill liner that reduces fretting and corrosion, and eases motor removal. Large, single-row ball bearings absorb radial and thrust loads on higher input speeds, while heavy-duty tapered roller bearings on all output shafts handle inherent gear load and provide maximum overhung load capacity.

"A key design component of this reducer is a second input bearing that reduces wear and fatigue on the input seal," said DeWayne Polley, gearing product specialist for Emerson. "The second input bearing minimizes shaft movement during motor installation and operation. We tracked the performance of the Raider Plus and are confident the materials and assembly of this product support an 18-month, leak-free guarantee."

Four unit models – the "U", a universal style with an output shaft; the "Q", a c-face-style with an output shaft; and both styles with hollow outputs – are available from stock with a center distance range of 1.0" to 6.0", ratios from 4:1 to 3600:1, and output torques from 82 to 22,416 ft-lbs.

All Raider Plus housings, motor adaptors, covers and mounting bases have cast iron single-piece construction for maximum strength and dependability. Forged bronze worm gears, precision manufactured to AGMA specifications, provide greater tensile strength than cast bronze. The integral worm and shaft are hardened to 58 RC for extra strength and durability. The speed reducers are all factory-filled with FDA-approved polyglycol synthetic lubricant for enhanced rating and reduced thermal rise.



For more information:
Emerson Industrial Automation
Phone: (314) 553-2000
industrialautomation@emerson.com
www.emersonindustrial.com

Siemens

RELEASES MOTOR/DRIVE PACKAGE

Siemens Industry, Inc. announces the release of combination motor/drive packages, allowing an OEM or end-user the option to select the optimum solution for a variety of heavy-duty industrial motion control applications from a single source, backed by a full three-year warranty. Choosing from a pre-determined list of motor/drive combinations, the customer simply makes the selection best suited to the application. The motor and drive are packaged on a single pallet, shipped and invoiced together.

The motor and drive combinations are power-matched for 480 V high-overload operation through a 20 hp range, with I^2T protection from thermal damage provided as a standard in both the motor and the drive components. The Siemens Intelligent Operator Panel (IOP) is included with these packages, allowing easy step-by-step drive start-up.

Application macros are provided in the Sinamics G120C drive for easy installation and wiring; the terminals are pre-assigned at the factory and the parameters are automatically set. The Simotics SD100 motors are rugged cast-iron with inverter duty ratings in a 4:1 speed range for constant torque and 20:1 speed range for variable torque. Simotics SD100 units are severe-duty TEFC motors that meet NEMA Premium efficiency.

Communications selections on these matched motor/drive combinations include RS485 with USS and Modbus protocols. A Profibus variant is also offered for a Totally Integrated Automation (TIA) solution. TIA is the proprietary Siemens solution for achieving optimum performance, energy efficiency and sustainability within a machine or manufacturing environment. Standard pricing has been established for a wide variety of motor/drive combinations from 1–20 hp, and is included in the available literature on this new Siemens service.

For more information:
Siemens Industry, Inc.
Phone: (847) 952-4158
www.usa.siemens.com/drives

Revolvo

PROVIDES PUMP SOLUTION AT WASTE WATER PLANT

Revolvo SRB split roller bearings are providing heavy duty support for the crankshaft of a large sludge pump at a United Utilities waste water treatment plant in Oldham, U.K., following operational problems with the pump that resulted in damage to its original bearings and crankshaft. Revolvo was introduced to the project by ERIKS Electro Mechanical Services, who were commissioned to refurbish and repair the sludge pump, including straightening the crankshaft, bearing replacement, drive re-alignment and on-site assembly and commissioning. The total value of the contract was in excess of £200,000.

A total of seven heavy-duty 150 mm SRB split roller bearings and housings were supplied for the application to the ERIKS EMS site who undertook the reassembly and installation of the crankshaft assembly. This was achieved within time and on budget, and the pump is now back in continuous service at the Oldham site.

For more information:
Revolvo Ltd.
Phone: +(44) 01384 245370
www.revolvo.com

Luneta

LAUNCHES CONDITION MONITORING POD

Luneta announces the launch of the Condition Monitoring Pod (CMP), a new multi-parameter inspection device modernizing and expanding daily lubricant inspections. This offering is part of a larger plan to roll out a series of innovative inspection-related products in the next few years. The CMP is a visible inspection window revealing critical health conditions of the lubricant and machine. Its proprietary design serves an unmet industry need, providing an improved method of inspecting the health of machinery without the need for laboratory analysis.

The CMP boasts a robust platform of 16 capabilities (10 of which are uniquely exclusive to this innovative tool). Other features include: built-in oil sampling port, easy-to-remove-and-view magnetic plug, corrosion/varnish inspection probe and a quick lubricant access point for syringe sampling.

Best applications include gearboxes, bearing oil sumps, hydraulic reservoirs, compressor oil reservoirs, turbine-generator main oil tanks and paper machine central reservoirs. "We believe that Luneta has re-invented the sight glass by turning it into the focal point of machine condition monitoring," said Luneta CEO Tom Fitch. "The Condition Monitoring Pod makes oil analysis as simple as performing daily machine inspections. A sight glass should not be limited to simple oil level inspections. Instead, with the CMP, an array of critical machine and lubricant conditions can be examined in just seconds. And, it signals the beginning of Luneta's vision to enhance machine reliability through the development of groundbreaking inspection technologies."

For more information:
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Phone: (888) 742-2021
sales@luneta.com
www.luneta.com

All for One, One for All

Collaboration Moves Bearing Modeling and Simulation Software Forward

Matthew Jaster, Senior Editor

Software is a rather unique beast in the power transmission market. Between all the updates, new technology and user feedback, it's about as fluid a product as you're going to get. Collaboration between software providers and users is imperative. The challenge is creating user-friendly software tools that work for large corporations to small job shops and everything in-between. You have to be technical enough to keep up with the latest trends and conventional enough to keep each and every client happy.

"Customers want bearing modeling and simulation software to help prevent them from making the mistakes that we see coming," says Mark Martens, manager of application analysis systems at Timken. "The way they access this information is one of the greatest challenges. Whether it's online, offline, by computer, smartphone or tablet, there's a lot of neat things you can do with connected technology today."

"There is a constant flow of requests and a continuous exchange of information between the development team and the users," says Andrea Bacchetto, team leader, knowledge and simulation tools at SKF. "Releases are scheduled twice a year to follow-up on the users' feedback and also to prepare the new breakthrough versions."

Addressing the individual needs of each customer is the most essential part of the process.

"You might have a student looking for some bearings for a Baja car or an engineer that needs information for a large industrial gearbox. Each client has different needs and we want to make sure they come to the right answer along the way," Martens adds.

"It is recognized that the software tools are becoming more and more a commodity: customer knowledge levels are growing and also their needs. Even a small size business is nowadays using software tools to predict performance," Bacchetto says.

Timken offers two groups of programs in this area: a comprehensive set of engineering tools available at *www.timken.com* including bearing analysis, bearings searches, 2-D drawings and 3-D models as well as a larger software suite of internal tools.

SKF develops internal software platforms which are distributed on a global scale. Users of such tools include application engineers, researchers and product developers. "SKF simulation software tools are used on a daily basis to support internal development as well as to answer questions from customers via application engineers and handle single bearings up to complete systems of multiple shafts, gears and bearings," Bacchetto says. "Beside the internal offerings, SKF has released an outside SKF Spindle Simulator which is based on one of the internal software platforms as well as more basic tools to support customers with various bearing selection aspects as well as sealing and lubricant selection. Also CAD models are available at *www.skf.com*."

In order to enhance reliability and maintenance of its products, Timken makes sure the modeling and simulation software keeps up with the selection or manufacturing process. "You have to be much more proactive," Martens says. "We can't really stop at just tools. We have to also help our customers realize the optimal performance of our bearings. It's not just the sale of a part in a box. It's also the information and service engineering support to help folks make good choices about their lubrication, sealing and shop practices. All these things go together to achieve that."

One way to ensure reliability is to make sure the customer is properly trained and instructed on how to best utilize the software tools. Customer training is a priority that

SKF Simulator Software

must be met and continually pursued as new updates and technologies are introduced.

"We don't want to hand out sharp knives without safety instructions," Martens says. "We also don't want to push out our tools without proper guidance or training. This is why we encourage our customers to come back to talk to our engineers and help them through each step."

Timken offers a range of onsite and local training courses that are customized to individual industries. "The long-term goal, of course, is to have our customers better educated on our products and the software tools that are available to them," Martens adds.

SKF currently has 21 Solution Factories across the globe and hopes to have 50 total Solution Factories in the next three to four years. An operation like this allows SKF engineers to be near their customers based on the specific industrial needs of the area. Here, an SKF expert can provide the knowledge and services necessary to design and create custom solutions on top of training.

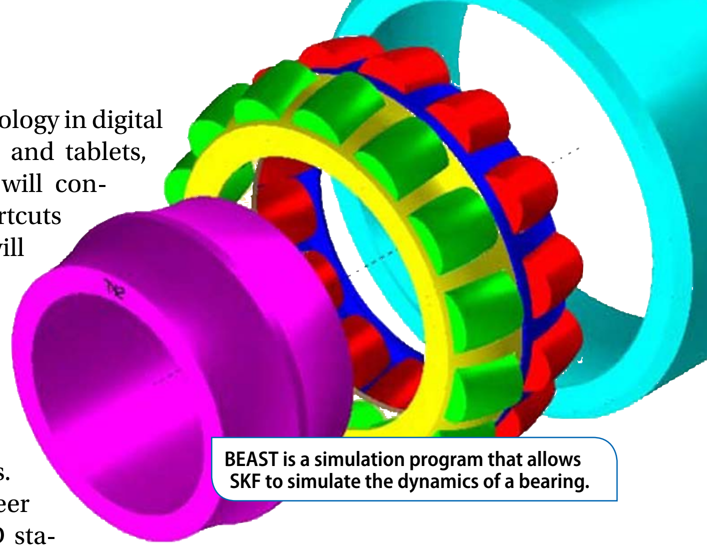
When face-to-face training or travel is out of the question, the user experience online plays a pivotal role.

"Our customers want to work in an environment they're comfortable with. They don't' want to change for us. They want us to come to them," Martens says. "When you see the CAD models on our website, you will find we offer 20 file types for download, and direct import into another 33 specific CAD systems. Anytime a user has to take your data and convert it into something that works for them, that's an extra obstacle. This takes time. We'd want to provide things in as consumable a format as possible."

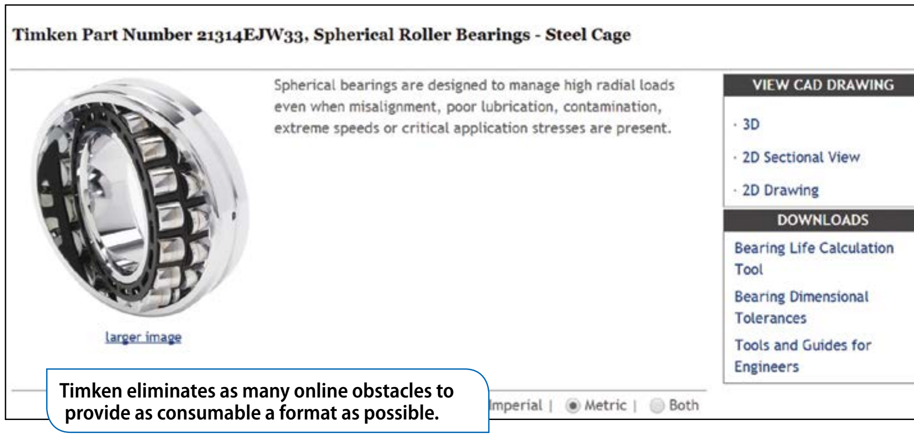
SKF presented their cloud service at Hannover Fair in Germany last year. The service can be used to collect data regarding the status of bearings in a machine. "The SKF cloud system is explained in the first issue of *Evolution* in 2014 (*http://evolution.skf.com/in-the-cloud*)," says Bacchetto.

With advancing technology in digital cameras, smart phones and tablets, bearing software tools will continue to offer new shortcuts and technology that will change how troubleshooting is addressed on the shop floor.

There are two distinct groups that utilize Timken's software tools, according to Martens. "For the design engineer that's working at a CAD sta-



BEAST is a simulation program that allows SKF to simulate the dynamics of a bearing.



Timken eliminates as many online obstacles to provide as consumable a format as possible.

tion, there's very little value in creating a smart phone app to do heavy engineering. But if you think about the guy working on machinery that might need to get more information to grind a bearing seat, shaft or housing diameter, mobile tools make more sense. You don't want to stop and find a computer to do that."

"We are definitely targeting mobile devices for the next few years. SKF already has a heavy presence in the App Store and in Google Play. Advanced calculations are currently based on laptops and/or desktops, but thanks to increasing calculation power from the mobile devices, SKF will soon be in the position to port more advanced tools in such devices," says Bacchetto.

And what other improvements will be made to modeling and simulation software in the future?

"Things are becoming much more sophisticated," Martens says. "If you look at Finite Element Analysis (FEA), 10 or 20 years ago it was special, now it's an integral part of what we do. As our customers design closer to reliability limits, you have to consider not only the bearing but the environment and the application that it is in. I really think we'll see specialty cases becoming more and more routine."

The software solutions, however, can only take you so far. "Technology can't solve everything. There's no substitute for good experience and good engineers and I'm quite proud of those that we have at Timken," Martens adds.

SKF continues to see radical advancements to its software tools and believes there's much more to come. "64-bit capabilities, new report engines, speed improvements, FEM calculation dedicated to seals and large models, e.g. wind turbines," says Bacchetto.

The end result is to not only make products last longer through new technology, but make sure each component reaches its highest level of performance.

While this particular article focuses on bearings, Martens believes the whole PT system should be examined. "Any given component including the gears, bearings, housing, shafts, lubrication, etc, should be considered. None of those components rise or fall on their own or succeed without the others. By

Timken offers 20 CAD Models for download and direct import into another 33 specific CAD systems.

understanding the interaction and the contribution of each component to the system, I think that's where successful design comes into play."

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Software Bits

In addition to SKF and Timken, there are several other companies that offer an array of software tools for bearing modeling and simulation. This sidebar provides a quick analysis of some of the other options on the market from the likes of Romax, MSC, Schaeffler and COBRA.

Romax Technology

From increasing durability in on-road applications to increasing bearing reliability in wind turbine technology, Romax provides support and advice on bearing selection, fitment, analysis and recommendation. Romax software can help users predict bearing life and durability, improve bearing selection, analyze bearing failure and assess and select suppliers. Additionally, *Romax-Designer* contains a comprehensive catalog of production bearings and allows customers to define any custom bearing including conceptual designs.

For more information:
Romax Technology
Phone: +(44) 0 115 951 8800
www.romaxtech.com

MSC Software Adams/Machinery

This module is for engineers who need to predict the impact of the design and behavior of rolling-element bearings on overall system performance. This includes an accurate representation of the bearing stiffness, sensitive to internal dimensions, offsets, misalignments, and clearances.

Customers can choose from 14 different rolling-element bearing types; look up bearing parameter values from a library of over 24,000 off-the-shelf bearings and/or input values directly; calculate bearing reaction forces, optionally leveraging a nonlinear stiffness response from embedded technology delivered by *KISSsoft*, an MSC Software partner; select from over 120 oil- and grease-based bearing lubricants and predict the bearing service life (under the specified simulation conditions) based on industry standards sensitive to the loading, lubrication, speed, and bearing geometry.

For more information:
MSC Software
Phone: (800) 942-2072
americas.contact@mscsoftware.com
www.mscsoftware.com

Schaeffler
INA/FAG Brands

Along with the supply of top quality machine elements matched to customer applications, good service is an important tradition at INA and FAG. Rolling bearing design is one of the focal points of design support. An important task there is to ensure that the product is designed correctly to give customers the competitive edge. To meet this requirement, INA and FAG have been using calculation programs successfully for more than 30 years. *BEARINX* software is a program for performing rolling bearing calculations. It enables rolling bearing supports to be analyzed in detail – from single bearings to complex gear systems and linear guide systems. All calculations are performed in a consistent calculation model. Even for complex gears, the contact pressure on each rolling element is considered in the calculation.

For more information:
Schaeffler Technologies
Phone: +(49) 91 32 82-0
www.schaffler.com

Cobra Software Family
J.V. Poplawski & Associates

Cobra-EHL is an entry-level software package designed for the mechanical designer/engineer. *Cobra-EHL* analyzes the interaction of up to six bearing rows on a rigid shaft under combined radial, thrust, and moment loading in three degrees of freedom. The program has a visual front end that allows the user to interactively change input data and preview results without having to exit the program. Lubricant film thickness and associated life adjustment factors are internally computed.

Cobra-AHS analyzes up to five bearing rows on a flexible or rigid shaft loaded in five degrees of freedom. Bearing models include high-speed ball, cylindrical roller and tapered roller bearings. Housing and shaft distortion effects due to out-of-round, out-of-flat and local slope at each rolling element are included. Bearing heat generation and cage forces are calculated. Roller edge stress concentration is calculated due to local roller misalignment for crowned rollers and displayed as a 3-D color contour plot. Bearing fit-up analysis is performed using classical thick ring press fit theory as well as a seamless interface to *ANSYS FEA* models of the bearing row that are automatically constructed and analyzed with results returned to *Cobra*.

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I Rely on Arrow Gear
SPIRAL BEVEL GEAR CATALOG

FEATURE

Gear Design Software: Engineer Beware?

Gear design software is a wondrous thing. *But watch your back.*

By Jack McGuinn, Senior Editor

When software goes bad, what do we call it? System failure? Human failure? A virus? A number of words will work. How about this? *Glitch.* It has that onomatopoeic quality that fairly screams, "Downtime!" And with good reason — software-generated miscalculations can have very expensive — if not perilous — repercussions. What makes software mistakes particularly sinister is that they may be hard to detect by the QA/Inspection process. You can't put a gage to an algorithm, for example — it's *software*.

Like gears, software is *everywhere.* And like gears, most people never think about the wondrous things that software makes happen around this world on a 24-7 basis. Working seamlessly with gear systems of all types — from the simplest to the most complex — gear software is the friendly "ghost" in the machine that enables robotic surgery, keeps rotocraft rotating, streamlines today's automatic transmissions, keeps gear motor systems running smoothly (and profitably), and ever so much more.

But gears sometimes fail — seemingly for no reason — but we know that's not the case. Software? Same thing. And a design engineer doesn't have to be a closeted Luddite to be a software scoffer. They exist in plain sight. That's one, typically older extreme of the spectrum. And then we have those — typically of a younger vintage — for whom software is the Mothers' Milk of gear design. And finally there is the much larger group — the in-betweeners, we'll call them — who strike that delicate balance of cautious reliance upon software and irreplaceable life/field experience. Is there a right or wrong group? Let's try and find out. To make that happen we talked to a number of people with credentials that qualify them to speak with unquestioned authority to the issues broached in this article. All six of them have authored, co-authored, or overseen the development of their company's proprietary and successful design software programs. As you'll see, they and the companies or consultancies they lead are all *gear knowledge-intensive.* The software expertise comes almost as a value-added premium.

We wondered what, for those designers who cling to software like a favorite stuffed animal, are the most common mistakes that can — and do — occur.

"A permanent source for mistakes is the user himself," said KISSsoft AG CEO Stefan Beermann and president Ulrich Kissling in a joint statement. "He has to enter a large number of parameters; each input can be wrong. Sometimes the meaning of an input is misunderstood. If this happens the program will give correct results, but for another case. So it is essential to make plausibility checks, starting with looking at input and output speed and torque. Checking sense of rotation (important, for instance, if you have a helical gear). And if your final solution is 10 times better than any other in the world, you are a genius, or something is wrong. Usually the latter, no insult meant."

There is no doubt that modern gear software is widely used and very helpful to gear engineers for rating load capacity of existing gear designs or designing new gears. However, calculated results from gear software are accurate only if the input data are correct.

Bob Errichello, president of Geartech, a gear industry consultant, noted AGMA gear failure analysis guru, software developer and *Gear Technology* technical editor, readily concurs with the correct input data concept, adding, "Consequently, it requires an experienced gear engineer to properly formulate the input data. The necessary knowledge and judgment is only gained from years of accumulated experience in designing, manufacturing, and testing gearboxes. The required knowledge includes training in gear materials, heat treatment, gear metrology, gear tribology, gear failure analysis, and bearing technology. Therefore, gear software is a useful tool, but it needs to be used by an experienced gear engineer to obtain meaningful gear designs."

And Mike Fish of Dontyne Systems has perhaps the best answer in terms of why frivolous use of design software can be downright irresponsible. "The most common problem is in believing that the software is a quick-fix or even a replacement for experience

"(There is) no magic gear design software that is capable (of) delivering a perfectly optimized set of gear parameters by itself, (which) is why gear engineer expertise is essential. (In fact) some gear design programs have a warning 'for gear experts only.' Ignoring this warning is a common mistake."

Alex Kapelevich, AKGears

and well-trained personnel within a company."

Indeed, as Alex Kapelevich, president of AKGears and the author of the 2013 release, *Direct Gear Design* (CRC Press), points out, "Software is not magic; it is merely an extension of gear knowledge, fundamentals and experience. "(There is) no magic gear design software that is capable (of) delivering a perfectly optimized set of gear parameters by itself, (which) is why gear engineer expertise is essential. Gear design software is just a tool; it works well in good hands, and vice versa. An engineer that uses gear software should have solid knowledge of gear design fundamentals. (In fact) some gear design programs have a warning '"for gear experts only;"' ignoring this warning is a common mistake."

One more thing on this point: knowing where to start. That may sound obvious. It's not.

Says N.K. "Chinn" Chinnusamy, president of Roscoe, IL-based Excel Gear. "The user should have a clear idea of what he or she is looking for. Incorrect input will result in bad or unreliable output data. Most gear design software available in the market will not tell the user where to start but will simply analyze the data inputted and show results. So the user should know where to start."

And then there's the old truism: software is only as smart as the people that wrote it.

Or is it?

"This is partly true," Fish qualifies, "but it is also only as *beneficial* as the person using it. This includes the psychological barrier of the user departing from a previous procedure for a new one required by the software. The intention is that the new method will benefit long-term in reduced time and more certainty in manufacture (otherwise, why change?), but it is sometimes difficult to demonstrate this to the user if they can't follow the new approach or have a mental block on the principle, perhaps seeing it as a threat. This situation has improved dramatically in the last 20 years with the emergence of much higher computer literacy across the workforce and improved graphics which allow visualization of the gears they have designed."

"Who says that software is only as smart as the person who wrote it?" Beermann and Kissling rejoin. "If the basic concept of a solution is a perfect fit to the problem, often a computer program gets more capabilities than whoever wrote it expects in the beginning. Of course it is also easy and not so rare that a not-so-smart developer spoils even a smart concept."

"Yes, to some extent it is," says Chinn, agreeing with the premise. "But any gear software should be written per standard — either AGMA or DIN or ISO. In this respect it should be the same no matter who writes the software. User-friendly input, clear, concise and easy to interpret output data will depend on the person who wrote it. In some software the user should have knowledge of gear design and metallurgy even to attempt to use it, and it will print out several pages of output data which may be confusing to the user if the user is not a gear engineer."

For Kapelevich, it all comes down to ignoring another old truism at one's own risk. "Insufficient knowledge of gear drive application details that result in improperly prepared input data is another common mistake. Every software user should know the rule: 'garbage in, garbage out.' Disregard of this rule results in poor design that may lead to assembly problems, premature gear drive failure or to over-designed product — i.e., greater in size, weight, and more expensive than necessary."

We mentioned at the top that the extent of reliance on the use of gear design software was probably commensurate with the age of the design engineer; and does anyone not know where this is going? With that in mind, to what extent *do* design engineers rely upon gear design software?

"There are basically three groups of engineers," say Beermann-Kissling. "The first group consists of (typically older) experienced engineers that are, or think they are, sophisticated experts in their field. In this group many don't trust any design software at all. Some of them have problems using it, which is related to growing up without any computers at work. The second group uses the design software as a tool wherever appropriate, understands the theory behind it, but doesn't believe it blindly. The third group wouldn't do anything without software. This group mainly consists of young engineers that grew up with computers and, nowadays, smartphones and tablets. In this group there is a certain danger of relying too much on the computer.

"This becomes a problem when the software is used without understanding the theory. In these cases the software has to replace missing knowledge, and that won't work. Referring to the above, you can say that even smart software is lost if it is not used in a smart way."

For Dontyne's Fish, it, too, is not always a black-or-white answer; on some occasions an engineer's life/field experience also comes into play, among other factors. And it makes sense — even if it reads a bit like a le Carre *Smiley* novel.

"In our experience we see our customers temper the use of the software with a healthy amount of skepticism and scrutinize the results. Software calculates exactly (or at least should!) to ISO, AGMA, or other standards which may differ from each other or an internal rating system.

"There are known problems with ISO and AGMA that result in differences in interpretation, and it is not unusual for different packages to return different answers for a given gear specification. The standards provide standard methods and not recommended solutions.

"If the software calculates better result than experience, then it is best to go with the worst case scenario until proven by testing. If the software indicates failure but the engineer has experience of similar working gears with acceptable performance, then the application knowledge should be used and relied on. This methodology is consistent with the intent of gear rating standards. This decision process requires an application knowledge base to draw on from the engineer or at least within the company."

"The gear engineer must be intimately familiar with each of the industry standards to be able to understand why the (gear) ratings differ, and to properly interpret and apply the ratings. It is imperative that gear engineers test software-designed gears to prove that the software is reliable."

Robert Errichello, Geartech

Despite all the *Sturm und Drang* about catastrophes-in-waiting, gearmaking is, in its way, a forgiving manufacturing process. It's not corneal transplantation. There is no *best* way to make a gear.

But, say Beermann and Kissling, "There are some ways that are not very good. For sure, there are multiple ways to design a gear that are leading to good results in an efficient way. The best basis for a new design is a given design plus some information about the performance of the old gear set. Based on this, it is usually not so hard to find a new variant that fulfills the current requirements. Good design software helps a lot on the way, making proposals and automatically analyzing hundreds or thousands of possible solutions.

"In addition, it is a good idea to go from rough to fine, so start with the macro-parameters like number of teeth, module (or diametral pitch), helix angle, profile shift. Then go down to micro-geometry and further optimize the gear set."

Let's assume Fish agrees: "There is no best way! Have a sure reference point for the application you are making. If this is not available, then there will have to be considerable testing."

And Chinnusamy reminds that "best" should apply to making a gear to its *designed* "size, weight, speed, quality, life, and cost." If that happens, who needs "best"?

What about plastic and PM gears — two hot commodities in play these days — but there's certainly nothing "common" about them. Where does design software fit in their world?

"The properties of these (plastic and PM) are fundamentally different," says Fish. "The standards and models cannot be transposed because it is convenient (though many have tried), since the factors determined by testing are specific to the material and forms originally intended."

As a company, KISSsoft looks at the two materials as two distinct opportunities — but not without challenges. And what's this? — a little "fun" to boot.

"The process of molding gears offers new opportunities and new challenges to the engineer," say Beermann and Kissling. "There is much more freedom in the design, but there are also requirements from the molding process that must be taken into account. Like, for instance, a rounding on the tip that can be produced in a wire erosion process. Then the variety of the materials is huge, and each combination can show significantly different behavior.

The failure modes change — instead of pitting we usually talk about wear. Scuffing will not be a problem, but a complete meltdown of the gear can happen. Temperature becomes crucial for the design. All these challenges make designing plastic gears so much more fun than metal gears (if you don't believe it, try it). The main problem for plastic designs is still the lack of data for material properties. So here it is even more important to have older designs to compare to (also referred to as 'experience').

"Powder metal gears are somewhat in between; more freedom in design, but a material which is close to classic steel with its properties.

"In both cases, it is recommended to use a definition of the tooth form which is closest to the generating process. In this way you avoid a lot of possible errors in the design; like, for instance, collision with the root fillet. If you want to define your tooth shape directly, make sure you check the proper meshing of the gear set afterwards."

"Who says that software is only as smart as the person who wrote it? Of course it is also easy — and not so rare — that a not-so-smart developer spoils even a smart concept."
Stefan Beermann and Ulrich Kissling, KISSsoft AG

Aside from fielding the questions put to them for this article, there is no lack of fodder regarding software-induced — or enabled — errors in gear manufacture. So we pressed for a bit more reaction about software performing unpredictably in circumstances beyond its design parameters.

"If you've ever used a pair of pliers for nailing, this is not coming as a surprise," quip Beermann and Kissling. "A good program should inform the user if he uses the software outside of its well-defined limits. Still, if used carefully, this is not really a problem. It is important to assess the result and check it thoroughly.

"And there will be cases when only a prototype test will give the final answer. But we are talking about the design process here, not controlling a nuclear power plant. So, the worst that can happen is that the design doesn't work in the end. The target of design (and even more analysis) software is not to make prototype testing obsolete; the target is to reduce the number of tests needed."

At Dontyne, Fish points out that, among other things, software is incapable of making a decision — remember, garbage in, etc. "The software should only be an aid for the engineer and used within its bounds. If the engineer recognizes a problem in the calculations it should be reported to the publisher and it will be corrected. If errors occur, then engineers may be trying to apply the calculations for something for which it is not intended."

KISSsoft's team is right with Fish on this one. "You mean except for quan-

> "In some software the user should have knowledge of gear design and metallurgy even to attempt to use it, and it will print out several pages of output data which may be confusing to the user if the user is not a gear engineer."
>
> **N.K. "Chinn" Chinnusamy, Excel Gear**

tum flux? Design software is not thinking, it is following algorithms. Those algorithms might run into dead ends, or simply be not appropriate. But this is a question of software design. We tend to believe that the software should do the stupid part of the design work, proposing and evaluating solutions and do the number crunching. The engineer using it should make the decisions.

"Let me ask you back: What do you think makes the engineer think unpredictably?"

Adds Fish, "There may also be fundamental problems with the approach. FE methods are used increasingly, but can generate a different result depending upon the boundary conditions, method used, and element types. This is confusing and looks unpredictable, but is not strictly a problem with the software. The models should have a thorough testing program behind them to show they are appropriate to the situation. It is important that those who write gear standards provide enough numerical examples to allow test programmers to test their coding. Gear trade bodies can also minimize this risk by arranging informal round robin comparison exercises that provide additional examples to verify software."

Errichello points out that, "A factor often overlooked is that gear software must be validated. This is not a trivial issue, and it takes literally years of testing to determine whether particular software is free of bugs and gives valid results. Typically, a gear engineer tests a candidate software against an archive of example gearsets that were calculated from other trusted software or hand-calculated examples.

"Some software has capability to rate gears according to AGMA, ISO, and DIN industry standards. However, the rated capacity for a given gearset will differ for each of these standards. The gear engineer must be intimately familiar with each of the industry standards to be able to understand why the ratings differ and to properly interpret and apply the ratings. Finally, it is imperative that gear engineers test software-designed gears to prove that the software is reliable."

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So What's to Know About Specifying a Gear?

Only *Everything!*

THE QUESTION

I am trying to specify a few .8 module metric gears and am being asked to include as many gear specifications as possible on our drawings.

For example, one of the gears we need is a .8 module 40t gear, so my plan was to say this:

Spur Gear: *Teeth* = 40, *Module* = .8, *PD* = 32, *Circ Tooth Thickness* = 1.17 (using this to determine backlash, but there is probably a better way)

For all other tolerances and design information, reference AGMA 2000-A88, Q8 (we found some stock gears that listed this number but don't really know what it means). I see so many different AGMA and ISO standards for gears and I'm just not sure which one I need. We haven't purchased any yet and don't want to until we know which one to use. Can you shed some light on this or point me in the right direction? I don't know why but this seems like a real mystery to us!

Thank you!

Expert response provided by Rob Frazer, senior engineer at the Design Unit:

The position that you find yourself in is very common. Gear technology is not a particularly difficult subject to understand, but it covers many fields of expertise and thus there are many elements that need to be appreciated and understood before a full specification can be prepared that will ensure that you get gears that are fit for purpose and meet your needs.

To put it bluntly, if we don't specify gears properly we get the gears we deserve rather than the ones we actually need.

This is a challenging task for people who do not regularly specify gears. Because we have no knowledge of your specific application for fine-pitch gears, the following guidance is generic for most gears and we will assume that specifying the geometry itself is sufficient and that the life and loading is evaluated separately.

Gear standards, whether they are published by AGMA or ISO, are a very valuable source of reference material for gear designers. However these are written for people who have the relevant background gear knowledge and expertise to implement the procedures and, more importantly, interpret the results from applying the standard procedures. The AGMA information sheets and ISO Technical Reports are prepared by the expert working groups to provide guidance on implementing and understanding them. It is also of course important to ensure you are referring to the latest version of the standard. This is not helped by the fact that within AGMA and ISO publications, none fully address the specification requirements of gears.

But before you try and specify gears starting with a blank sheet of paper, there are other options that should be explored first that, although apparently more costly, may save much of your time and potentially avoid costly mistakes:

Employ a consultant to specify the gears for you. The benefits from this are that you will get a full specification that will provide you with the gears you need. The disadvantage is that you won't learn anything yourself and if you need to modify the design or experience quality problems, you will have to re-employ your consultant.

Alternatively, you can seek a reputable gear supplier who is willing to provide the design and manufacturing expertise to supply gears that will meet your needs. You would need to specify life and duty cycle (load and speed) requirements; the drive element (e. g., electric motor specification); the load characteristic; size envelope; gear shaft and gear housing tolerances; manufacturing methods; environmental conditions (temperature and humidity); preferred materials; operating backlash; noise requirements; and annual quantities. Again, if you want to modify anything, you rely on the supplier for these design changes.

The application in this example is not defined, but the 0.8 mm module gears could potentially be supplied by a catalog gear supplier who specialize in small-pitch, standard geometry gears; your question implies that you have already considered this. A number of catalog gear companies offer a range of gears that may be suitable for your requirements. They can supply small quantities of gears but may also be

suitable for larger volume manufacture and also provide guidance on suitable geometry tolerances, tooth thickness tolerances to ensure the gears operate with acceptable backlash. Using off-the-shelf or modified catalog gears often provides the cost-effective solution to prototype or small volume gear applications.

There are many commercial software programs available that can assist you to design, analyze and specify gear geometry. These range considerably in terms of complexity and cost, but the best allow users to invoke ISO and AGMA accuracy standards, use standard proportion cutting tools, define and evaluate tooth thickness (for backlash calculations) and evaluate the gear pair using stress analysis standards such as AGMA 2101 or ISO 6336. Many programs provide graphics that enable the users to properly visualize the gear pair they are specifying. The most basic of these programs is a simple automated gear calculator, while the most sophisticated programs provide help and guidance when things are starting to go wrong (Fig. 1). But users need to understand what the programs are doing and thus it is recommended that proper training is obtained prior to use. Few people are provided with sufficient training in gear technology in college and university courses, but help and guidance is provided by the AGMA in their training program (*www.agma.org*). In the U.K. the British Gear Association (BGA) has an extensive seminar program that allows those new to the gear industry to attend a series of short courses to introduce them to gear technology (*www.bga.org.uk*).

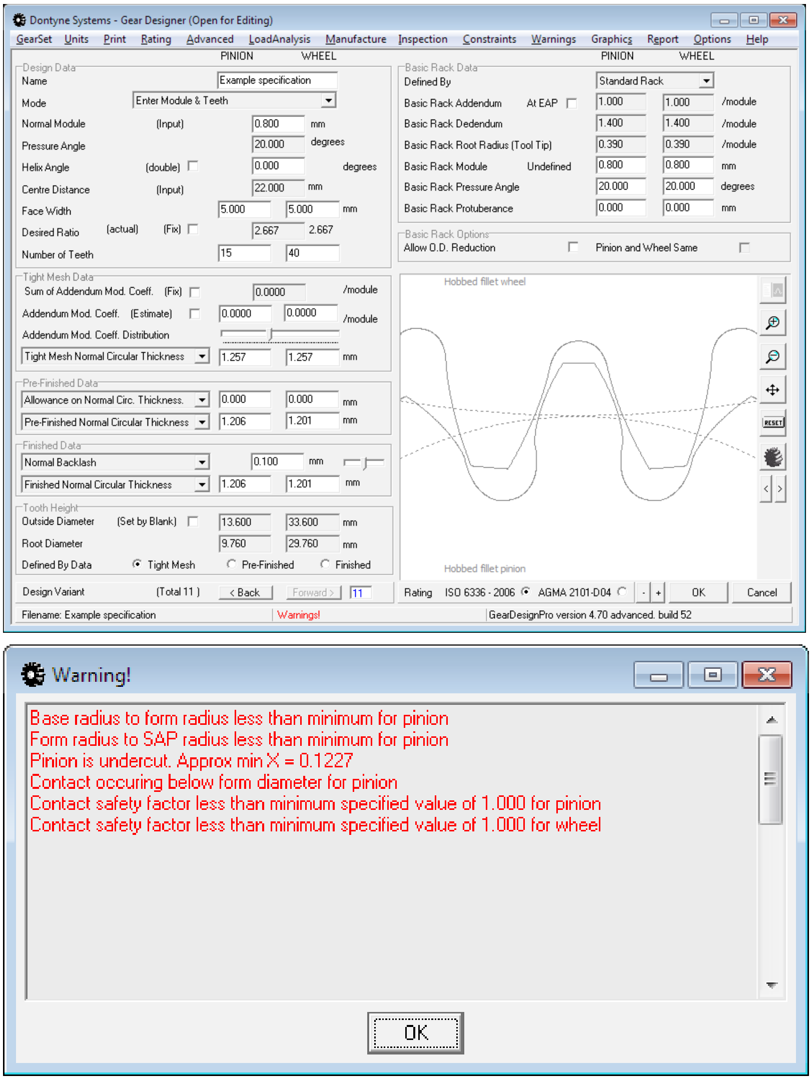

Figure 1 **Example of a software package used to develop a gear specification with built in warnings when you approach normal geometry limits (courtesy Dontyne Systems Ltd).**

The strategy adopted by the Design Unit (at Newcastle University, U.K.) for specifying gears is that you provide unambiguous data relating to the geometry of the finished gear. Our policy is not to specify the details of the manufacturing procedure and thus a full gear specification comprises seven elements:

1. The nominal basic macro gear geometry (module, tooth number, helix angle, tip diameter, root diameter, face width, addendum modification coefficient). ISO 21771 provides formula for these parameters.
2. The specification of microgeometry corrections to the tooth flank (tip relief, helix crowning) on gears that are transmitting significant amounts of power or have stringent noise and vibration requirements.
3. Cutting tool geometry data (depth, pressure angle, cutting tool tip radius used to cut the tooth root region, grinding allowance or backlash allowance). AGMA 1003 and 1006 provide information of the proportions of tooth for fine-pitch gears and plastic gears.
4. Tooth thickness data specifying the tooth thickness tolerances to ensure operating backlash is achieved when the gear is manufactured and assembled. We normally define gear circular tooth thickness indirectly because measuring a circular arc length is difficult. We use parameters such as dimension-over-pins or span size over several teeth. Refer to ISO 21771 for tooth thickness calculations. AGMA 2002 provides guidance on tooth thickness tolerances. There is no ISO standard directly related to tooth thickness allowance and backlash.
5. A gear geometry accuracy specification is defined by ISO or AGMA tolerance classification standards. Two methods are commonly used here:
   a. The measurement of individual errors (profile, helix or tooth alignment, pitch errors, radial runout of the tooth space and tooth thickness). AGMA 2015-1 (replaced AGMA 2000) specifies allowable limits for different tolerance classes and is similar to ISO1328-1. Note that these do not provide guidance on which tolerance grade to pick. For most applications, precision-cut gears of grade 7 or better (lower grade number) are achievable, with molded, fine-pitch gears of grade 9 to grade 10 commonly specified. The tolerances that are specified must consider power transmission, noise and tolerance build-up of the assembled gear assembly. The accuracy of the gear is verified by measurement with CMMs or dedicated gear measuring machines and fine-pitch gears of 0.4 mm module can be easily measured (Fig. 2). The process provides feedback to show that the gears comply with the accuracy specification and also identify what has gone wrong with the manufacturing process.

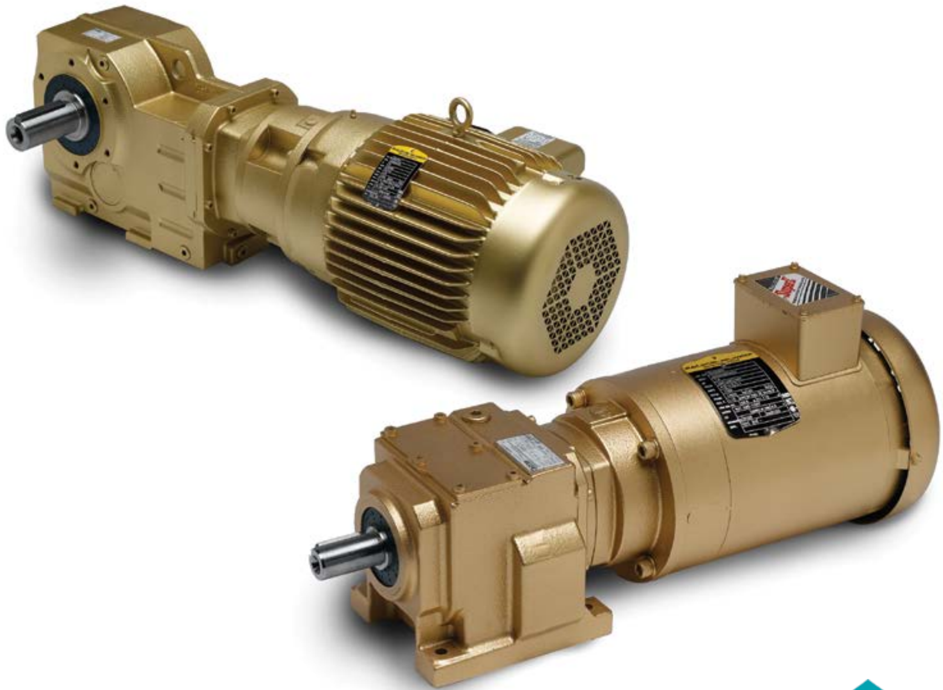

Figure 2 **Klingelnberg P65 with a 0.5 mm diameter probe Inspecting a 0.7 mm module gear in accordance with ISO1328-1 (the minimum standard probe is 0.3 mm diameter).**

b. The second method is the measurement of radial composite errors or dual-flank errors, and is commonly used to control fine-pitch gear tolerances. The method involves meshing a product gear with a precision-ground master gear (a minimum of 3 accuracy grades better than the product gear) under light load with zero backlash, and recording the change in center distance as the gears rotate. AGMA 2015-2 and ISO 1328-2 both provide allowable tolerances for gears of 0.8 mm module, and the quality grades mentioned above equally apply to this measurement method. This also provides a method of measuring tooth thickness by specifying upper and lower indicator limits based on maximum and minimum center distance values.

6. Material specification, including material type and where appropriate the range of acceptable hardness values and case depth requirements.
7. The datum axis that is used to define the gear geometry and provide a functional location for the gears when in service.

A typical realization of a gear specification is illustrated in Figure 4 for ISO 1328-1 grade 9 gears, assuming quality is controlled by helix, profile, pitch tolerances and tooth thickness, defined by dimension over balls or pins. An alternative specification using ISO 3128-2 for dual-flank testing measurement strategy is illustrated in Figure 5.

In conclusion, the specification of gears requires a detailed knowledge of gear geometry, manufacturing methods, inspection methods, material and the functional requirements of the application. Every gear designer has his or her own preferred method, but asking the right questions, using the appropriate standards and support software ensures it is possible to specify gears reliably.

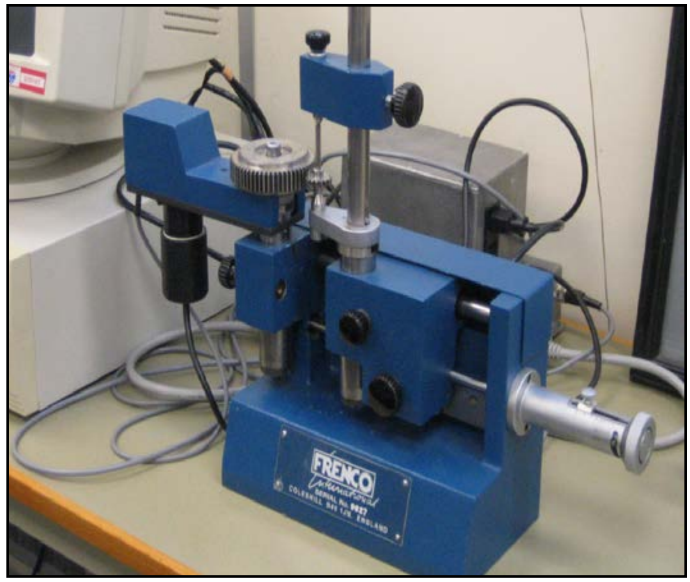

Figure 3 **Frenco-dual flank roll tester for measuring composite Radial deviations in accordance with ISO1328-2.**

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GEARDATA	
Basic Geometry	
Number of teeth	40
Normal module	0.800
Reference pressure angle	20.000
Ref.helix angle (left)	0.000
Addendum Mod. coefficient	0.0000
Nominal tooth depth/Mn	2.400
Reference Data	
Facewidth	5.000
Tip Diameter	33.600
Root Diameter	29.760
Topping	
Base helix angle	0.000
Reference Diameter	32.000
Base Diameter	30.070
Finished Tooth Thickness	
Ball Diameter	1.440
Dimension over balls (nom)	34.022
Dimension over balls (max)	34.022
Dimension over balls (min)	33.943
Flank Tolerances	
Reference axis	datum bore A
Accuracy Standard	ISO 1328-1/95
Grade	9
Adjacent pitch tol	20 µm
Cumulative pitch tol	57 µm
Profile tol	21 µm
Helix tol	25 µm
Tool tip radius	0.312
Meshing Information	
Mating gear	
Centre distance nominal	32.000
Start of active profile dia	30.851
Contact ratio	1.714
Normal backlash max	0.160
Normal backlash min	0.100
Material & Heat Treatment	
Through Hardened (V)	
Surface hardness	200 Hv
Angles are in ° and distances in mm unless otherwise stated	

Figure 4 **Example gear specification for ISO 1328-1 accuracy gears.**

GEARDATA	
Basic Geometry	
Number of teeth	40
Normal module	0.800
Reference pressure angle	20.000
Ref.helix angle (left)	0.000
Addendum Mod. coefficient	0.0000
Nominal tooth depth/Mn	2.400
Reference Data	
Facewidth	5.000
Tip Diameter	33.600
Root Diameter	29.760
Topping	
Base helix angle	0.000
Reference Diameter	32.000
Base Diameter	30.070
Finished Tooth Thickness	
Ball Diameter	1.440
Dimension over balls (nom)	34.022
Dimension over balls (max)	34.022
Dimension over balls (min)	33.943
Flank Tolerances	
Reference axis	datum bore A
Accuracy Standard	ISO 1328-2/97
Grade	9
Single composite tol	11 µm
Total composite tol	56 µm
Tool tip radius	0.312
Meshing Information	
Mating gear	
Centre distance nominal	32.000
Start of active profile dia	30.851
Contact ratio	1.714
Normal backlash max	0.160
Normal backlash min	0.100
Material & Heat Treatment	
Through Hardened (V)	
Surface hardness	200 Hv
Angles are in ° and distances in mm unless otherwise stated	

Figure 5 **Example gear specification for ISO 1328-2 accuracy gears.**

How to Spec a Mill Gear

Frank C. Uherek

This paper outlines the design considerations that go into construction of a drive system in order to explain the importance of specific data, why it is required, and where design freedom is necessary. Apart from loads and speeds, interface dimensions and site specific conditions are also needed. Deciding up front which gear rating practice to select can affect the torque capacity of the drive train by ~15%.

Introduction

An expert in whole-brain learning, Steven Snyder, once said, "There are only two problems in life: (1) You know what you want, and you don't know how to get it; and/or (2) You don't know what you want."

To solve these two problems, a clear understanding and good communication skills are necessary. In terms of getting a great gear set, it requires a coordinated effort between the end user, the gear manufacturer, the gear designer, the consultant, and the original equipment manufacturer. Each of these groups has a key piece of the puzzle necessary for the gear to fulfill its useful operational life. This paper will outline what information needs to be collected and passed on to the gear designer to develop a successful drive train for a specific area of use: gearing for cylindrical grinding mills. It will act as a checklist for information required, outline the impact of certain selections, and resolve ambiguities to address the two problems outlined above.

Background

A grinding mill circuit is an unusual installation for gearing when compared to traditional enclosed gear drive installations, but these applications have been utilized for over eighty six years. The grinding process, more accurately termed a tumbling process, uses horizontal rotating cylinders that contain the material to be broken, potentially augmented by grinding media (Fig. 1). The material moves up the wall of the drum until gravity overcomes centrifugal forces, and it drops to the bottom of the drum to collide with the remaining material. This breaks up the particles and reduces their size. Power required for this process ranges from 75 to 18,000 kW (100 to 24,000 HP), in either single- or dual-motor configurations.

In this type of application, the pinion is mounted on pillow blocks driven by a low-speed motor or a motor and enclosed gear drive. The gear is mounted on the mill using a flange bolted connection (see Figure 2 for one type of flange installation). Both the center distance and alignment are adjustable either by shimming the pillow blocks or moving the mill. Lubricant is typically either high-viscosity oil (1,260 cSt @ 100°C) sprayed on the gear in 15 minute intervals or a lower viscosity oil or grease product sprayed on the pinion every few minutes. Alternately, lubrication can be applied by continuous spray or dip immersion methods.)

Gear sizes can range up to 14 meters (46 feet) in diameter with face widths approaching 1.2 meters (50 inches). Typical tooth sizes range from 20 to 40 module (1.25 DP to 0.64 DP). Single-stage reduction gears range from 8:1 to as much as 20:1. Gear materials are typically through hardened cast steel, fabricated forged and rolled steel, or spheroidal graphitic iron. Pinions are carburized, induction hardened, or through hardened heat treated steels.

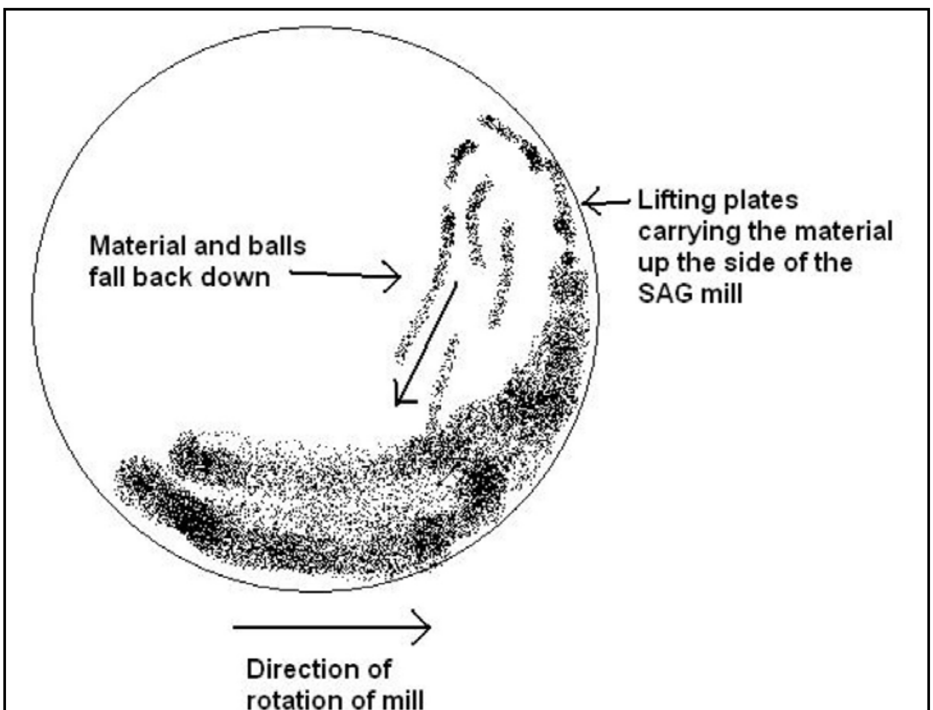

Figure 1 Grinding mill process.

Figure 2 Grinding mill installation.

For small installations, either a one- or two-piece design is used with the split joints located in the root of a tooth. Four- and six-piece designs are also utilized when the weight of the segments exceeds the crane capacity of the facility or pouring capacity for cast segments becomes an issue.

Initial Data

The purpose of a grinding mill is to make large rocks into small rocks. To accomplish this task involves significant calculations on the part of the mill builder. These include reviewing the size of the incoming and outgoing product, the rate of production, the size of the mill in diameter and length, the grinding media, the theoretical critical speed of rotation, and the interior configuration of the mill. Unfortunately, to get what is required, this information needs translation into something that the gear designer can input into the rating calculations. The calculation of actual contact stress s_c does not have an input for tons/hour of mineral produced.

A theoretical relation of mill diameter to power is ~ mill diameter$^{2.5}$. To get torque, we also need the speed of the drum. This is based on the concept of a "theoretical critical speed of rotation (CS)." The critical speed of rotation is the speed (in rpm) at which an infinitely small particle will cling to the inside of the liners of the mill for a complete revolution.

$$CS = \frac{43.305}{\sqrt{\text{Mill Diameter}}} \tag{1}$$

where
CS is the theoretical critical speed of rotation, and is the mill speed, rpm;
Mill diameter is the nominal inside diameter of the mill, m.

Since we actually need the particles to come off the inside diameter of the mill to be processed, the typical mill speed is ~75% of the theoretical critical speed of that mill. Using the above formulas, significant experience of how the grinding process works, and material properties of the ore being ground, the mill builder can provide the gear designer with input power and output speed.

The next step is the interface dimensions. Since the gear needs to turn the mill, it needs to have a bore larger than the mill outside diameter. The mill outside diameter is a function of the grinding process selected. Autogenous mills are the largest in diameter since the feed grinds itself. A semi-autogenous mill uses some metallic or ceramic balls to assist the grinding process and can be slightly smaller. Ball mills are smaller still and use a larger percentage of balls to perform most of the work. Large-diameter mills allow for use of gear ratios not normally thought possible in single-reduction applications — namely, 8:1 to 20:1.

If the gear is to replace an existing gear, then manufacturing drawings or installation drawings complete with gear attributes, center distance and dimensions are required. Although budgetary pricing can be made without dimensional data, once an order is present, full data is required. These gears are made custom for each installation so there are no catalogs available to provide this information.

Site-specific considerations also need to be disclosed. If the gear is expected to operate outdoors or in unheated structures, a minimum and maximum temperature range should be given to assist in lubricant selection and method of application. Transportation limitations can also affect the design. If crane capacity or size limitations exist, the gear designer can increase the number of segments of the gear to allow for reduced handling weights.

Rating Standards

Once mill diameter has been established, the largest cost driver is the actual size of the gear. Gear power capacity is a function of how the ratings are calculated. There are two basic rating practices in use in gearing: ISO 6336 (Ref. 8) and ANSI/AGMA 2001-D04 (Ref. 5). Both exist to provide a common basis for comparing the power capacities of various designs. By their nature, these are general standards in that they apply for fine pitch gears of 4 mm in diameter as well as 13,000 mm gears, made from various materials and accuracy grades. Given that range, we run the risk of missing significant size effects — either large or small — or client expectations that will affect the performance of a gear set. This is why the general standards suggest use of an application-based standard when designing gears for a specific purpose.

The rating committee uses the fundamental standard as a criteria and method source for rating gears and adjusts the component factors to match experience and field performance for existing designs. AGMA and, to a lesser extent, ISO, have developed application standards for a variety of applications such as enclosed drives, high-speed units, drives for wind turbines, marine, automotive, and steel mill rolling applications to narrow the scope of the general rating practice and fine tune it for the nature of service.

For grinding mill service, an early application standard was AGMA 321.05 (Ref. 3); it was first approved for use in October 1943. Various iterations occurred with the last major re-write in 1968, when the standard was updated to use the formulations of AGMA 211.02 (Ref. 9) and AGMA 221.02 (Ref. 10). The last editorial corrections were issued in March 1970.

This rating practice uses concepts that predate our current AGMA 2001 thinking. The rating formulas for gearing in this standard are:

$$P_{ac} = \frac{n_p d^2 C_v}{12600} \frac{F}{C_m} I \left(\frac{s_{ac}}{C_p}\right)^2 C_H^2 \tag{2}$$

$$P_{at} = \frac{n_p d K_v}{12600} \frac{F}{C_m} s_{at} \frac{J}{P_d} \tag{3}$$

where
P_{ac} is allowable transmitted power for pitting, HP;
n_p is pinion speed, rpm;
d is operating pitch diameter, in;
C_v is dynamic factor pitting;
F is face width, in;
C_m is load distribution factor;
I is I factor;
s_{ac} is allowable contact stress number, lbs/in^2;
C_p is elastic coefficient, $(lbs/in^2)^{0.5}$;
C_H is hardness ratio factor;
P_{at} is allowable transmitted power for bending, HP;
K_v is dynamic factor bending;
s_{at} is allowable bending stress number, lbs/in^2;
J is J factor;
P_d is transverse diametral pitch, in^{-1}.

Table 1 Gear geometry and application data

Attribute	Pinion	Set	Gear
Number of teeth	21		314
Ratio		14.95:1	
Normal diametral pitch, in^{-1}		1.0154	
Normal module, mm		25.01	
Face width, in		27.25	
Face width, mm		692	
Axial overlap		1.100	
Outside diameter, in	23.043		313.670
Outside diameter, mm	585.3		8043.4
Tooth accuracy	Q12/A5		Q10/A7
Bore diameter, in			247.000
Bore diameter, mm			6273.8
Material hardness	55 HRC		335 HBW
Application			
Power, HP		5000	
Power, kW		3729	
Speed, rpm	202.42		13.538
Durability service factor		1.75	
Strength service factor		2.50	

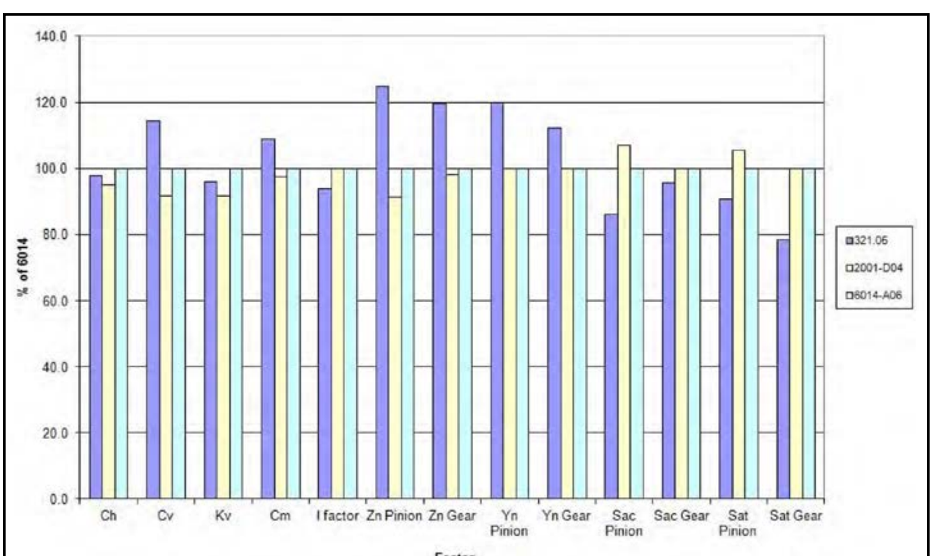

Figure 3 Normalized rating factors for base set.

The major influence factors were assigned specific values based on the size and experience of the industry with this type of gearing. Two dynamic factors were used, but both were a function of the pitch line velocity of the set. Load distribution factor was a function of face width only, covering the range of 50 to 1,016 mm (2 inches to 40 inches) with modification factors to adjust its value when teeth were hardened after completion. The allowable contact stress was reduced by the standard; however no metallurgical properties other than hardness were discussed. The hardness ratio factor was expanded to cover a ratio range of 1:1 to 20:1. The allowable bending stress was also reduced by the standard but it also remained only a function of hardness. Service factors ranged from 1.5 to 1.65 for grinding mill service.

ANSI/AGMA 6004-F88 (Ref. 11) was the first attempt to reflect ratings based on tooth attribute quality for mill and kiln gearing; it was released in 1988. After its limited acceptance in the industry, the AGMA Mill Gearing Committee developed the current standard ANSI/AGMA 6014-A06 (Ref. 4), released in 2006. It is currently in its five-year review cycle with the committee.

The rating formulas used in AGMA 6014 are as follows:

$$P_{acm} = \frac{\pi n_p F}{396000} \frac{I}{K_{vm} K_m} \left(\frac{d\, s_{ac} Z_n C_h}{C_p}\right)^2 \tag{4}$$

$$P_{acm} = \frac{\pi n_p d}{396000} \frac{F}{P_d} \frac{J s_{at} Y_n}{K_m K_{Bm}} \tag{5}$$

where
P_{acm} is allowable transmitted power for pitting, HP;
K_{vm} is dynamic factor bending;
Z_n is stress cycle factor for pitting;
P_{atm} is allowable transmitted power for bending, HP;
Y_n is stress cycle factor for bending;
K_{Bm} is rim thickness factor.

This formula now includes the effect of stress cycle factors as well as making adjustments to the base evaluations of dynamic and rim thickness factor.

The critical changes that the committee made to the standard addressed the fact that these gears mesh through the use of independent bearing support. The gearing is not mounted in a housing where all bearing supports are aligned by machining. Based on the ratios, modules (pitches), and face widths (approaching 1.2 meters, 50 inches), the effect of size and material usage, cast or forged steel or ductile iron needed to be included in the standard. Achievable and measurable accuracy grades limit the values of the dynamic factor. Client expectations of long life indicate values of the stress cycle factor Z_N and Y_N be based on 25 years. Durability service factors were also increased from AGMA 321.05 to C_{SF} = 1.75 for high-power mills over 3,350 kW (4,500 HP) in size. Strength service factors K_{SF} were also specified.

Given two standards designed to rate gears for this service, others occasionally use general standards or their own in-house-developed calculations. When this path is chosen, there can be significant risk that may not be realized by the user of the rating practice. As noted above, an application standard takes into account the narrower range of gear size, operating experience, typical materials, and mounting conditions of the process. A general standard, needing to be "all things to all people" can set requirements or allow mounting practices that are easily achievable when working with 100 kg (220 lb) size gear sets, but are problematic with 118,000 kg (130 ton) designs.

To illustrate this, an existing gear set was selected and rated per various standards (Table 1). Using the data in Table 1, this set was rated to the various AGMA rating practices to illustrate differences in specific rating factors. Each rating factor was normalized to its corresponding AGMA 6014 component (results shown in Figure 3).

The hardness factor C_H is more conservative in AGMA 321.05 and AGMA 2001. The dynamic factor C_V K_V for AGMA 321.05 is not a function of accuracy, so it has a greater de-rating effect than the Q10/A7 values computed with the other standards. Also note that AGMA 2001 is more aggressive than AGMA 6014 for this factor. This was the intent of the mill gearing committee based on their experience with ANSI/AGMA 6004-F88 that adopted dynamic factor from the base standard without modification.

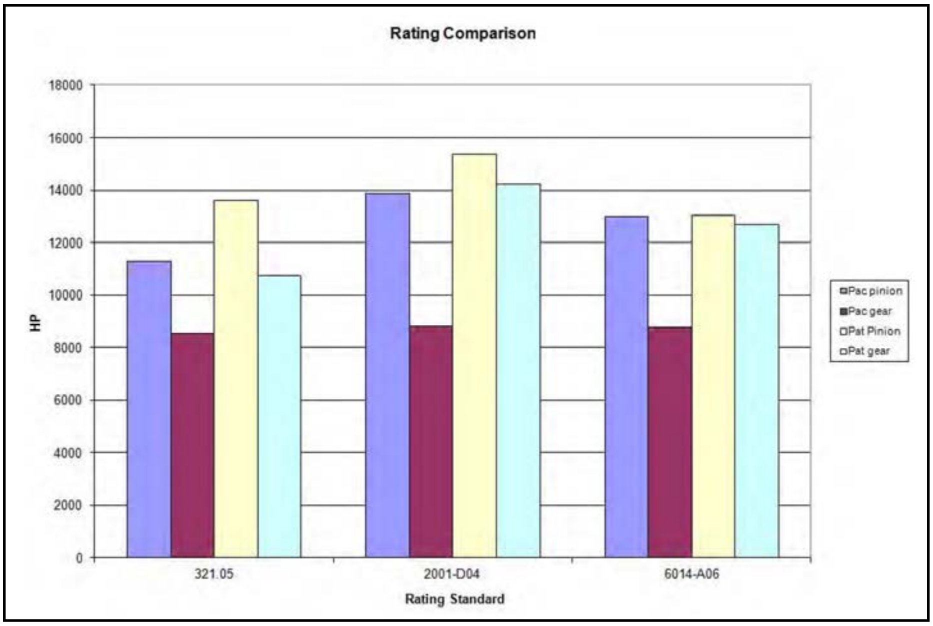

Figure 4 Rating comparison for the base set.

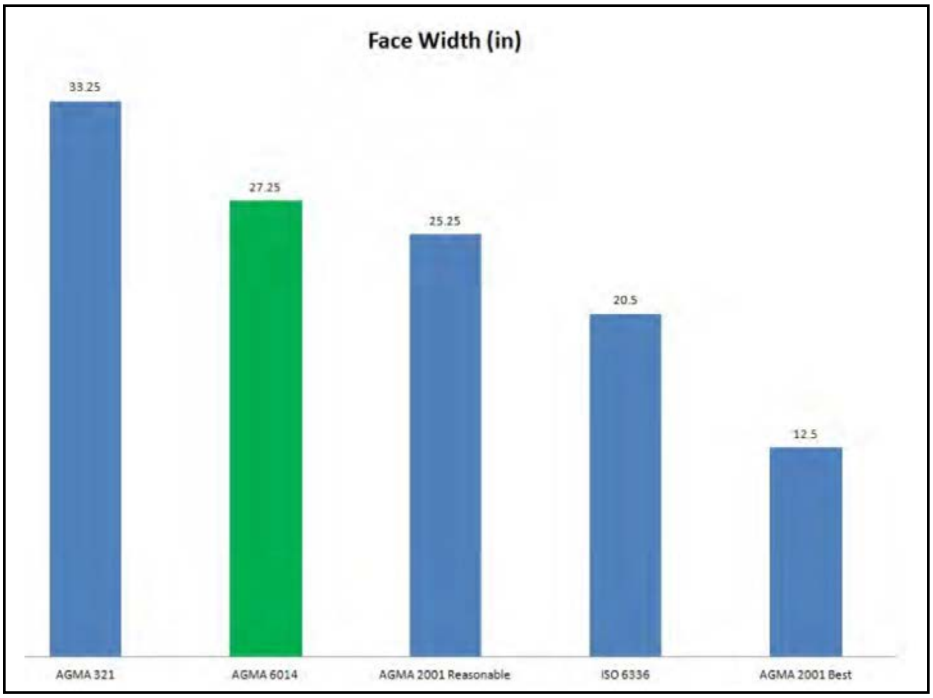

Figure 5 Face width as function of standard selection.

Load distribution C_M K_M follows the same trend as dynamic factor for the same reasons. The change in I factor was caused by the release of the information sheet for its calculation. Stress cycle factors Z_N Y_N were unknown in AGMA 321.05, and AGMA 6014 uses more conservative values than AGMA 2001 to control the power capacity of the set. The expansion of metallurgical specifications in AGMA 2001 and AGMA 6014 over the AGMA 321.05 requirements of hardness and "steel" affected the allowable stress numbers s_{ac} and s_{at}. The use of 55 HRC pinions also lowers s_{ac} and s_{at} in AGMA 6014 over the 58 HRC values in AGMA 2001. Reference 1 further outlines the differences and history of gear rating practice for mill and kiln drives in AGMA.

Given the interaction of the above factors, Figure 4 illustrates the resultant rating. The durability service factors based on transmitted power are 1.71, 1.76, and 1.75 for 321, 2001, and 6014, respectively. The strength service factors are 2.14, 2.84, and 2.53, respectively. We note the lack of strength rating in the AGMA 321.05 rating and the excess strength rating in the 2001 ratings highlighting the problem of rating a set optimized to a different rating practice. The actual power able to be transmitted is 3,204 kW, 3,758 kW, and 3,734 kW (4,297 HP, 5,040 HP, and 5,010 HP). The ~16.6% difference in power capacity between the AGMA 321.05 and AGMA 6014 ratings meets the goal of the Mill Gearing Committee to achieve a rating difference of 15% between the two standards.

To look at the effect of changing the base standard, Figure 5 compares the 6014 base design to sets designed to other standards utilizing face width adjustment. The axial overlap and heat treatment was kept constant as the face width was increased or decreased to meet the service factor requirement of 1.75/2.50.

The more conservative AGMA 321.05 increased the face width by six inches. Use of a general rating practice (AGMA 2001 and ISO 6336) with similar attributes to AGMA 6014 reduced face width by two and 6.75 inches, respectively. Aggressive use of the rating practice, termed "AGMA 2001 Best," enabled a 54% face width reduction. However, with extra precision mounting requirements, high tooth accuracy requirements, and reduced stress cycle performance, it is unlikely that the predicted, optimistic performance of this gear set in this demanding application would meet client expectations.

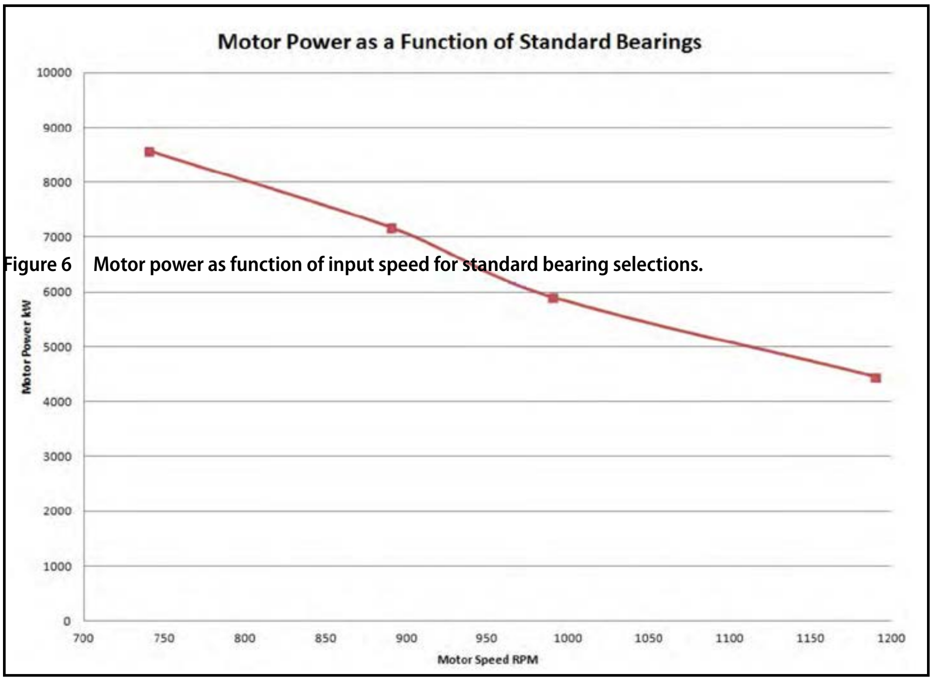

Figure 6 Motor power as function of input speed for standard bearing selections.

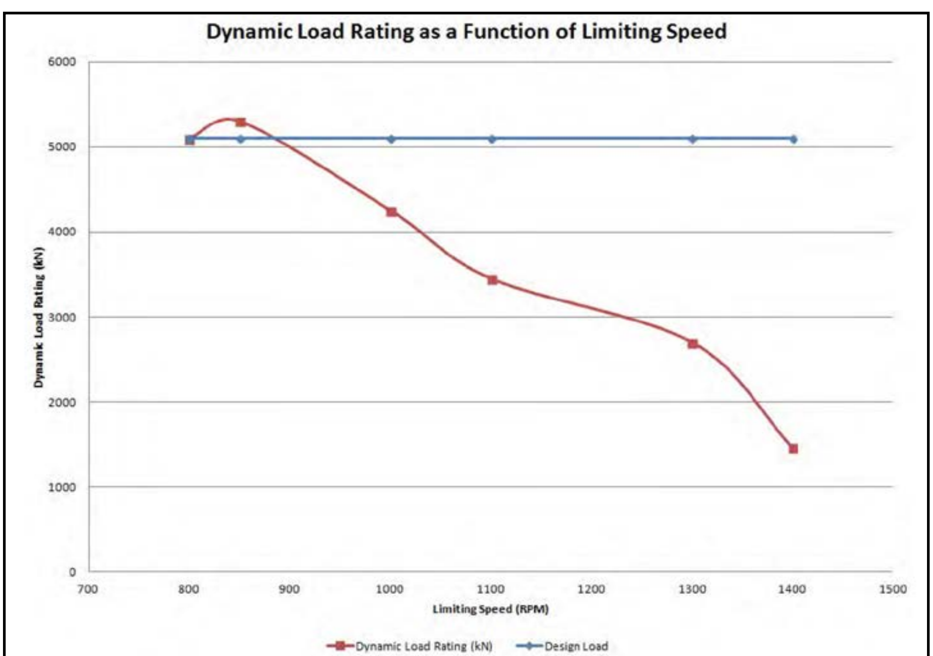

Figure 7 Dynamic load rating as a function of limiting speed.

All gear rating standards stress the need for an experienced gear designer capable of selecting reasonable values for rating factors and who is aware of the performance of similar designs through test results or operating experience. When this is removed from the equation, through the use of an inappropriate rating standard or in combination with OEM or end user in-house practice, a valuable reality check is lost. In most cases, when the gear designer faces such a request, they also check the proposed design under AGMA 6014 or AGMA 321.05 to make sure it satisfies the standard requirements. In cases when the design sufficiently deviates, concerns of suitability or fitness of purpose need to be raised with the client.

Prime Mover Selection

Having resolved the output speed and input power, the next decision point is to determine the input speed to the system. Low speed synchronous motors in the range of 250–150 rpm are one option. This eliminates a source of power loss by removing the gear drive and coupling from the drive train. However there is a cost premium to multi pole (20 – 40) motors over the more conventional kind. Another option is to insert a gear drive between the motor and the mill pinion. If one is trying to minimize motor cost, the tendency is to maximize motor speed (1,500 – 1,200 rpm).

Figure 6 illustrates that as input speed increases, the amount of allowable power in a gear drive decreases. This is mainly due to cage velocity of the gear drive's input shaft bearings. Many bearing manufacturers publish speed limits in their catalogs as a function of thermal loading, above which some method of supplying cool oil to the bearing is required, as well as a limiting speed.

Table 2 Typical load history for mills

	Base load, hp	Speed	Start factor	Actual load, hp	Time per year, seconds	Number of starts per year	Time per year, hrs	Years	Time for 25 years of operation, hrs
Starting load	10806	180	1.5	16209	7	12	0.02	25	0.583
Inching load	123	1.4631	1	123	1800	12	6.00	25	150.000
Running load	10806	180	1	10806	31514316	1	8753.98	25	218849.417
							8760		219000

Table 3 Expected life of mill given a typical duty cycle for 1.0 overload factor

Overload factor K_o = 1.0 all cases				
Duty	**Ratio wear pinion**	**Ratio wear gear**	**Ratio strength pinion**	**Ratio strength gear**
Starting	5.23024E-09	6.6996E-06	1.06083E-14	1.18697E-14
Inching	2.06011E-10	4.25751E-07	4.1522E-18	2.2215E-18
Running	7.29597E-06	0.001922728	1.40726E-14	6.7759E-15
Sums	7.30141E-06	0.001929854	2.4685E-14	1.86478E-14
	Pinion wear	**Gear wear**	**Pinion strength**	**Gear strength**
Life hours	29994206923.06	113480102.36	8871779508117490000.00	11744009610536000000.00
Life years	3423996.22	12954.35	1012760217821630.00	1340640366499550.00

Table 4 Expected life of mill given a typical duty cycle for 1.13 overload factor

Overload factor Ko = 1.13 for running loads				
Duty	**Ratio wear pinion**	**Ratio wear gear**	**Ratio strength pinion**	**Ratio strength gear**
Starting	5.23024E-09	6.6996E-06	1.06083E-14	1.18697E-14
Inching	2.06011E-10	4.25751E-07	4.1522E-18	2.2215E-18
Running	3.42739E-05	0.006479422	6.18972E-13	2.98033E-13
Sums	3.42793E-05	0.006486547	6.29584E-13	3.09905E-13
	Pinion wear	**Gear wear**	**Pinion strength**	**Gear strength**
Life Hours	6388689342.00	33762184.86	347848597742773000.00	706667465221281000.00
Life Years	729302.44	3854.13	39708744034563.20	80669801965899.70

Table 5 Expected life of mill given a typical duty cycle for 1.25 overload factor

Overload factor K_o = 1.25 for running loads				
Duty	**Ratio wear pinion**	**Ratio wear gear**	**Ratio strength pinion**	**Ratio strength gear**
Starting	5.23024E-09	6.6996E-06	1.06083E-14	1.18697E-14
Inching	2.06011E-10	4.25751E-07	4.1522E-18	2.2215E-18
Running	0.000122969	0.017670034	1.40828E-11	6.78081E-12
Sums	0.000122974	0.017677159	1.40934E-11	6.79268E-12
	Pinion wear	**Gear wear**	**Pinion strength**	**Gear strength**
Life Hours	1780862895.23	12388868.30	15539229428435500.00	32240579740876500.00
Life Years	203294.85	1414.25	1773884637949.25	3680431477269.00

Limiting speed is a function of the form, stability, or strength of the bearing cage, lubrication, forces, precision, and other effects. Exceeding the limiting speed of a standard bearing forces the designer into high-precision, limited-production-run bearings that may not be readily available — or feasible.

Figure 7 illustrates the drop-off in load carrying capacity as a function of limiting speed for a 340 mm spherical roller bearing series. This illustrates the problem of increasing shaft speeds, thus limiting bearing selection to cause the rating element in the gear drive to be the high-speed bearing in place of the more typical and more expensive low-speed gear.

Gear Drive Considerations

The next selection point — given use of a high-speed motor — is how to distribute the ratio between the gear drive and the mill set. An initial conjecture is to wrap the gear as closely as possible around the mill or kiln and place the remaining ratio in the gear drive, based on the assumption that a carburized, hardened and ground enclosed drive is more cost-efficient in torque transmittal capabilities than the open set. This needs to be balanced by the loss in efficiency if a multiple-stage reduction drive is necessary for the ratio required. Typical single-reduction drives achieved efficiencies of 98.5 – 99%, whereas double-reduction drives are in the 97 – 98% range. If one is using a line of catalog gear drives, the steps in torque transmittal capacity as a function of unit size will also drive the selection. Forcing a mill pinion speed in a reducer drive train or selecting too fast of a motor speed can lead to low-cost items — such as input shaft bearings in the gear drive — constraining the entire design of the drive train. An example of this is the combination of high power (over 5,000 kW 6,700 HP) high-speed motors with L10 bearing requirements greater than the design amount based on the service factor of the drive. Requesting 100,000 hours of L10 life with a 2.0 service factor that implies 50,000 hours of life in a catalog-designed drive requires the drive designer to increase the size of the input shaft bearings to achieve such a life requirement. This may lead to going to the next unit size to achieve the L10 life requested. Not allowing the ratio in the drive to increase to use more of the excess torque capacity of the gear drive by slowing down the pinion speed causes an uneven distribution of torque generation between the drive and the gear set, thus increasing costs. It is best to advise the gear supplier of either the direct-driven or reducer-driven option and let them work out the most cost-efficient solution to size the gear/gear drive combination.

Duty cycle. A key parameter in gear train selection is the frequency of use. Since these sets are designed for 25 years of life, one needs to review the load cases to ensure that all modes of operation are addressed. ills experience starting loads; bringing the mill from rest to full operation; inching loads; where the mill is slowly turned at ~ 0.1 rpm for inspection or maintenance purposes; and the normal running load during operation. A typical load history is shown in Table 2.

Given this load spectrum, a Miner's rule analysis can be performed to determine expected life. Although the starting loads at 1.5 times and the inching loads are 1.4 times base motor power, they have a minuscule impact on life of the mill. Tables 3–5 list the expected lives of a mill set for selected values of overload factor K_O.

Service factor is made up of overload capacity, life expectation, reliability of stress number data, and economic risk of failure. For this type of service, the major component of service factor is *economic risk of failure*.

Design Considerations

The last items to consider are the arrangement and structure of the gear train. Gear material choices are a large cost driver to the overall design. These gears are typically made from cast steel, fabricated-forged and rolled steel rim with welded steel web, or ductile iron. Each material has its sweet spot in terms of cost-per-inch/pound of torque. Figure 8 illustrates torque capacity as a function of price index, with the most expensive design normalized to a value of 100. Reference 2 further outlines the cost considerations for large gears. As with items outlined above, since one is purchasing torque, it is usually best to allow the gear supplier to determine the optimal material for gear construction.

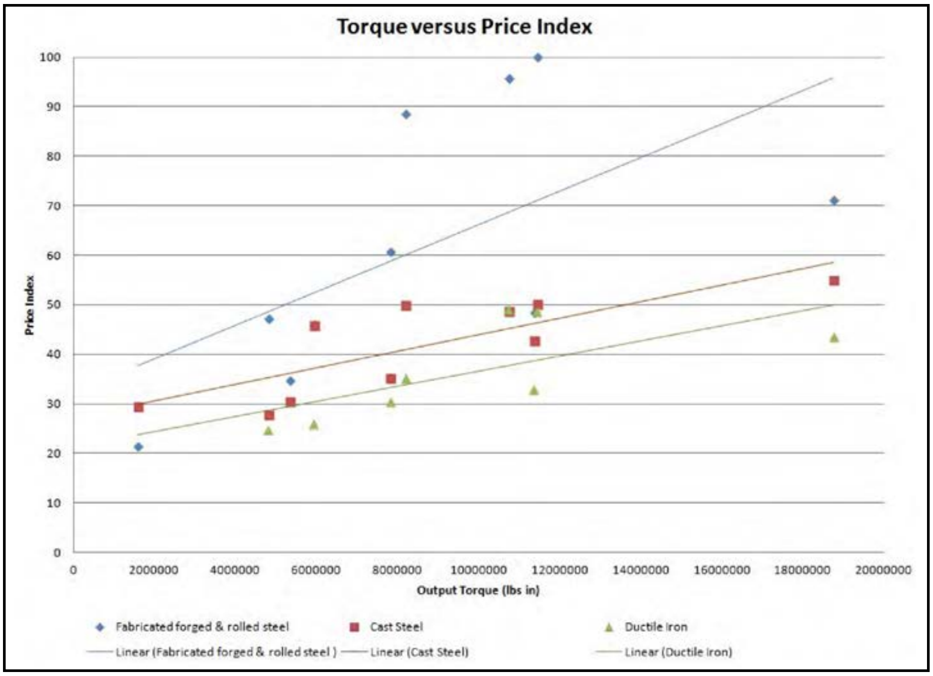

Figure 8 **Cost comparison of blank construction as a function of torque.**

Ambient conditions play a role, usually in the form of thermal considerations. Gear drives of this size are usually cooled by heat exchangers that need either a source of water or air at a reasonable temperature.

Lubrication systems are used to keep a constant flow of oil to the bearings and gear meshes. They need to function across the wide temperature range to ensure that the drive is not starved for lubricant at cold temperatures. In many cases successful oil pumping becomes an issue below 14°C (57°F) for VG320 mineral oils and 9°C (48°F) for synthetic. Immersion heaters, and/or bypass filtration lines may be necessary to ensure an adequate supply of oil. These considerations are avoided in the direct-connect, low-speed motor design.

The mill set typically requires much higher oil viscosities than a gear drive requiring either the use of diluents or heat-traced pipes to ensure flow of lubricant to the mesh. Another consideration is the altitude of the mine site where heat transfer to air may be reduced. Therefore minimum and maximum expected temperatures, altitude, and the availability of cooling methods need to be specified.

Support equipment can also influence drive train size. Pillow blocks are typically used to support the mill pinion. This gives the flexibility to adjust center distance and pinion orientation to optimize load contact.

The economic cost of downtime typically leads to large-diameter pinion extensions to reduce torsional stress. This — combined with a helix angle range of 5 to 11 degrees — usually results in shaft diameter in place of L10 life requirements determining the size of pillow blocks required. Coupling selection will influence the length of the shaft extension on the drive and driven equipment, as well as the torsional resilience of the drive train.

Required Data

Given all the above, the following data is necessary to successfully specify a mill drive set:

- Motor power
- Number of motors
- Mill speed
- Motor speed
- Design standard (for gear set and gear drive if required)
- Service factors based on above standard
- Gear interface dimensions (e.g., bore, minimum center distances, and drive train arrangement, weight limitations if any)
- Inching requirements (% of full load torque, mill speed in inching, desired connection point – mill pinion or gear drive/electric motor shaft)
- Duty cycle (if not continuous)
- Ambient temperature range (low and high)
- Altitude
- Specification requirements (e.g., nondestructive testing such as ultrasonic or magnetic particle, material properties)
- Inspection and witness requirements (on site visits to manufacturing location)
- Documentation requirements
- New or existing installation (if existing, need tooth geometry)

Conclusions

To resolve the two previously cited problems in life, one needs to clearly understand what one wants to do and communicate that to the people who can accomplish the task. Writing a gear train specification requires attention to detail and a realization of the impact that those choices can make. This paper reviewed the drive train outlining the information necessary for the gear designer to successfully develop a gear for this application. It noted various items that can play a dramatic role in the size and cost of a selection and indicated where creative freedom is necessary for an optimized drive that considers capacity, cost and lead time. **PTE**

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Frank Uherek, principal engineer with Rexnord in Milwaukee WI, received a BSME from the Illinois Institute of Technology in 1981 and a MBA from the same institution in 1985. He has been involved in the gear engineering field for over 33 years, holding various positions in design engineering and quality management covering enclosed drives, wind turbine drives, and open gearing for mill and kiln applications. AGMA activities include chairman of the Helical Gear Rating Committee; membership of numerous technical committees; and editor of AGMA 2001, AGMA 2121, AGMA 6014, and AGMA 6015. He received the AGMA TDEC award in 1997 for his outstanding contributions to the art of gear design and utilization. He has previously presented three papers at AGMA Fall Technical meetings and co-wrote two papers for IEEE cement industry conferences. In 2011 he was honored with the AGMA Distinguished Service Award for his work in developing AGMA gear rating standards.

Influence of Gear Loads on Spline Couplings

C.H. Wink and M. Nakandakar

Involute splines are commonly used in gearboxes to connect gears and shafts, especially when high torque is transmitted through the coupling. The load is shared among multiple teeth around the coupling circumference, resulting in higher load capacity than a conventional single key. However, the total load is not equally shared among all spline teeth, mainly because of pitch deviations resulting from the manufacturing process. The load distribution along the spline engagement length is also non-uniform because of tooth misalignments and shaft torsional effects. This paper presents an investigation of the influence of spur gear loads on the load distribution of spline teeth.

Introduction

Spline couplings are often used in power transfer systems to connect mechanical components such as shafts, flanges, brakes, clutches, pulleys, sprockets, and gears. A spline coupling has multiple teeth equally spaced around its circumference, which results in higher load capacity than a conventional single key. Spline teeth can be straight sided, in which both tooth flanks are parallel to each other with the same tooth thickness along the tooth height. Involute profiles are also used in spline teeth. Involute spline teeth are similar to gear teeth but shorter in height to provide great strength and compact size. Involute splines are typically preferred over parallel-side splines because they best center the two connecting components radially, and also provide lower root stresses with a larger tooth base thickness and smooth transition from tooth side to fillet radius. In a typical involute spline coupling of a shaft-gear connection, the shaft has the external teeth machined on it in the same number of internal grooves machined at the gear bore.

Ideally both the external teeth and internal grooves should have the same size to result in no clearance between them. Perfect splines under no clearance condition would evenly share the total load among the spline teeth in the circumferential direction. However, real-life splines are commonly designed to have a certain amount of allowable clearance on tooth sides and diameters to make them easy to assemble, to accommodate manufacturing tolerances, and also to allow lubricant to flow through the splines to help prevent fretting-type wear (Ref. 1).

Depending on the application, the spline fit is defined on tooth sides, between major diameters, or between minor diameters of the splines. Diameter-fits are used in applications where reduced radial clearance is required. In those cases the spline diameters are hard finished after heat treatment to a tight tolerance (Ref. 2). On the other hand, side-fit splines are often soft machined only, with no additional post-heat treatment operation, which provides a cost advantage over diameter-fit splines. The downside is larger variation among parts and larger radial clearance. The side clearance causes non-linearity similar to other components such as gears, bearings, and clutches, which, when combined with manufacturing deviations, such as spacing errors, and heat treatment distortions, result in uneven load sharing among spline teeth, especially in the circumferential direction, with consequent stress increase (Refs. 3-4).

Analytical and experimental studies done by Tjernberg (Ref. 3) showed that about half of the spline teeth carry load because of spacing errors, resulting in between 26% to 36% stress increase and over 50% life reduction. Chaplin (Ref. 1) also recommended assuming that half of the teeth share the full load. When subject to torsional load, splines demonstrate non-uniform load distribution along the engagement length of the tooth, which is in the axial direction, because of shaft torsional effects (Fig. 1) (Refs. 5-7). Volfson (Ref. 6) suggested that about a quarter of the teeth carry the full load. More recently, Chase (Refs. 8-9) presented a statistical approach to determine the load distribution in a spline coupling, and showed for a 10-tooth spline case study that approximately half of the teeth carried the full load.

It becomes clear from previous studies published in the literature that both manufacturing deviations, especially spac-

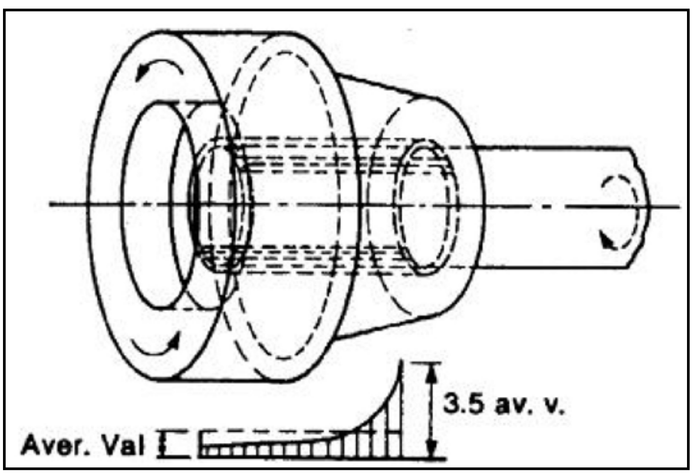

Figure 1 Load distribution in the axial direction for a pure circumferential torsional load case (Ref. 6).

ing errors, and shaft torsion significantly affect spline load distribution in the circumferential direction and in the axial direction. However, the studies were limited to the pure torsion loading condition only. In the particular case of drive train applications where involute splines are often used to connect gears to shafts, the gear mesh loads cause the splined components to be offset from their common center axis, affecting the load distribution of the spline teeth.

The objective of this paper is to investigate the gear load effects on spline load distribution, and propose a generalized and practical technique that can be used in the gear industry to calculate spline load distribution and to determine spline load capacity. A parametric procedure was developed to determine the gaps between internal and external spline teeth, accounting for the gear mesh loads and manufacturing deviations such as spacing errors. The general formulation of the problem of load distribution in gear teeth was applied to splines. The generalized elastic contact problem was solved using a simple procedure given in AGMA 927 (Ref. 10) and ISO 6336-1 (Ref. 11). Elastic deflections of spline teeth were calculated using a constant stiffness value that was obtained through Finite Element Analysis (FEA) of a spline coupling model in Reference 8.

The results showed that gear mesh loads significantly affect the load distribution of spline teeth. The maximum spline tooth load increased as the amount of side clearance between the internal and external splines increased. The procedure only applies to side-fit splines that have sufficient radial clearance between major diameters and minor diameters. Although not as accurate as finite element analysis (FEA), the spreadsheet solution is ideal for the beginning design work because it is fast and does not require FEA capability.

Analytical Model

A simple iterative method for the load distribution evaluation of spline teeth was developed from initial gaps among the spline teeth. Manufacturing deviations such as spacing errors and misalignment in the axial direction, intentional design modifications such as lead crown, and the spline centers offset by the gear loads were considered. When subject to gear loads, the pair of mating spline teeth with the smallest gap comes into contact first and load is transferred through it, resulting in elastic tooth deflections. Those deflections cause the gaps between the other spline teeth to get smaller and eventually other pairs of teeth come into contact and share the load. The final number of teeth in contact and their load sharing depend on the gap distribution, the elastic deflections and the load applied. An iterative procedure based on AGMA 927 (Ref. 10) and ISO 6336-1 (Ref. 11) was used to solve the load distribution problem, which was implemented into a spreadsheet. In the first step of the computational process, the spline teeth were divided into n points — or stations — across the engagement length. The gear loads were calculated from the gear tooth geometry and torque transferred through the gear mesh. The initial gaps were calculated from the spline geometry and radial location of one component to the other. The gaps also included manufacturing deviations and tooth misalignment. Then the load was evenly spread at all stations of all teeth to calculate the initial spline tooth elastic deflections such as bending and torsion. The total displacement of each point was obtained by adding the elastic deflections to the initial gaps under no load. From that point onward an iterative procedure was used to identify the non-contacting points and adjust the load values. The equations, assumptions and other details of the method are described in the following sections.

Gaps Analysis and Tooth Engagement

In perfect splines with no manufacturing deviations, no assembly misalignments, and both splined components sharing a common center axis, the side clearance is equal to the difference between the circular space width of the internal spline grooves and the circular tooth thickness of the external spline teeth taken at the same diameter (Fig. 2). The side clearance is the same for all pairs of mating spline teeth around the coupling, and also across the spline length. When transmitting torque through the coupling, the splined driver component turns in a given direction to eventually close the gap on the drive flanks. In the case of perfect splines, all mating spline teeth come into contact simultaneously and share the full load evenly in the circumferential direction.

One important manufacturing deviation of spline teeth for load distribution is spacing error (Refs. 3, 8). Spacing errors cause the spline teeth to be misplaced in the circumferential direction related to their theoretical location. This means

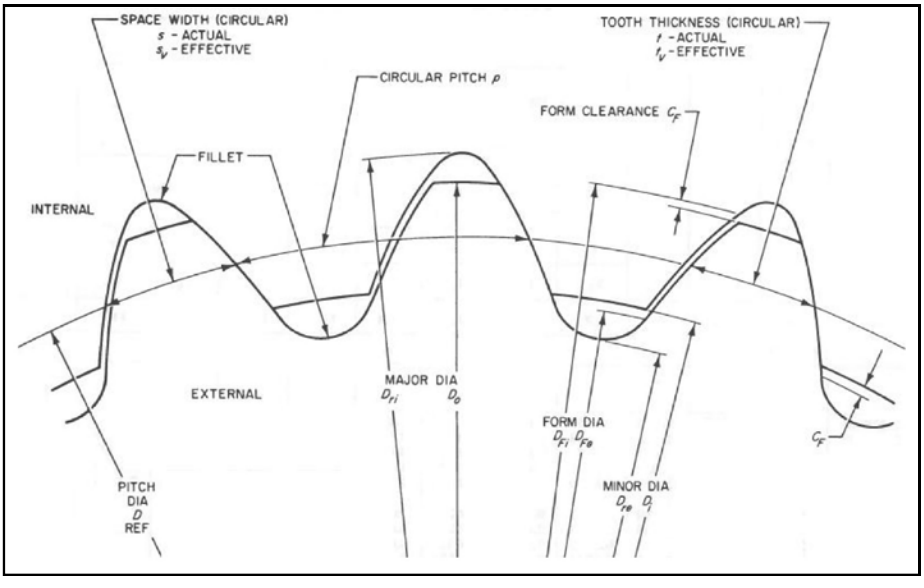

Figure 2 Side clearance of perfect splines (Ref. 12).

the teeth are not equally spaced around the circumference, which results in different gaps (Fig. 3). Spacing errors were entered in this model as a circular value with positive sign to indicate more gap (opposite to what is shown in Figure 3), and negative sign to indicate less gap (Fig. 3). The spacing errors of internal and external spline teeth were added together for a given assembly position, which defines the mating pairs of external teeth and internal groove with their respective spacing errors. The worst case for analysis occurs when the external tooth of maximum spacing error is assembled with the internal groove also with maximum spacing error but in the opposite direction.

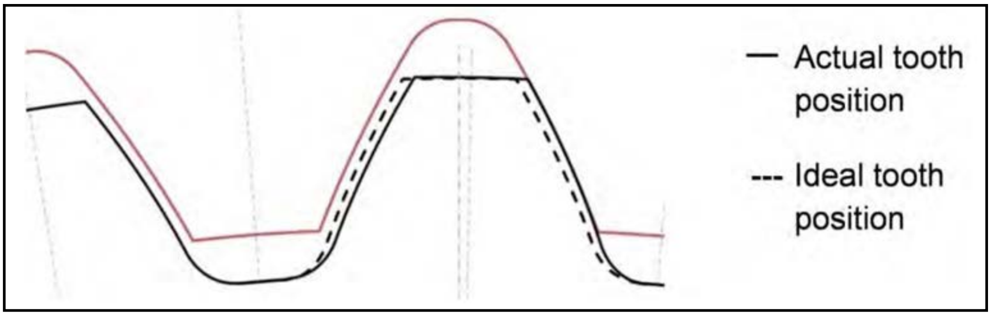

Figure 3 Spacing error of a spline tooth.

Misalignments and linear modifications in the tooth axial direction were also entered as a combination of deviations of the internal and external splines. Positive sign is used to indicate that the gap starts as zero at the left-hand end of the spline coupling and increases linearly towards the right-hand side of the coupling.

Lead crown is a barrel-shape modification across the length of the splines, where its maximum value is found at both ends of the spline teeth. Material is removed from the spline teeth according to a quadratic function from the middle of spline length toward the spline ends. In the model, lead crown was considered for the external splines only. It would be of small practical relevance to consider lead crown for internal splines based on manufacturing difficulty using conventional processes.

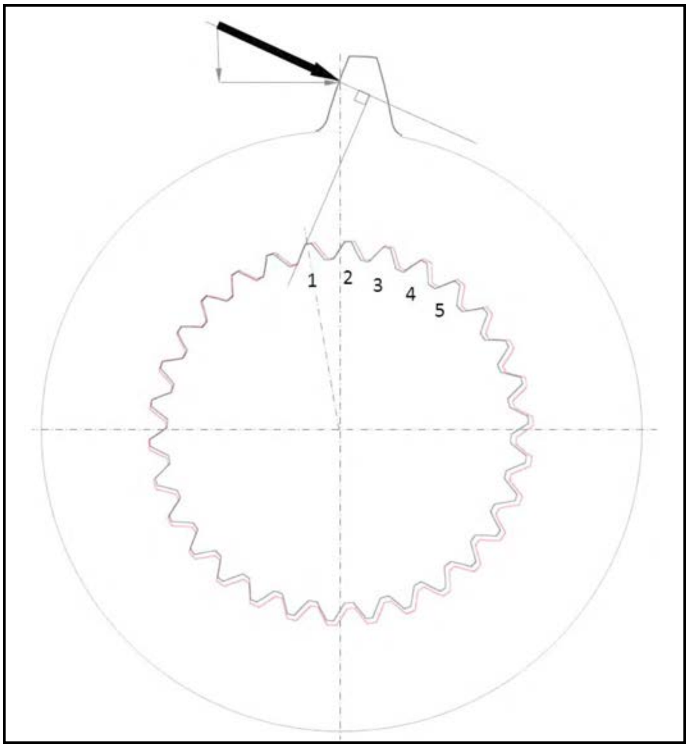

Figure 4 Gear load effect on gap distribution.

All deviations and modifications were calculated at the pitch circle diameter to each station and tooth, and were summed as gaps. Involute profile errors were neglected in the model because they are typically small compared to other factors. The gaps were assumed to be constant along the tooth height.

When the spline coupling is subjected to external loads other than pure torsion, such as gear loads, the gap distribution changes because the center of one component moves away from the center of the other component. This causes the gaps to vary around the spline-coupling circumference. Figure 4 shows an example of the effects of external load from a spur gear on gap distribution. In that case the shaft (external splines) is kept in place, and the gear (internal splines) moves in the line-of-action direction where the gear load is normal to the tooth flank. The gap variation among the pairs of spline teeth increases as the center displacement increases. The maximum displacement of the spline center related to its initial position depends on the geometrical relationship of spline teeth to the line-of-action angle. The worst-case condition was observed when the drive flank of a pair of spline teeth is perpendicular to the line-of-action (Fig. 4). Spline tooth No. 1 enters into contact first when no spline errors are considered. The gaps increase for the spline teeth farther from tooth No. 1.

The center distance displacement to bring the tooth number 1 into contact was calculated using the following Equation 1.

$$C = es \cos \phi_s \tag{1}$$

where

C is the distance from the center of the external splines to the center of the internal splines when an external load is applied;

es is half of the clearance between the circular space width of the internal grooves and the circular tooth thickness;

ϕ_s is the spline tooth pressure angle at the pitch circle diameter.

From the center displacement calculated in Equation 1 the resultant gap at each pair of spline teeth was calculated by Equation 2.

$$hc_i = es - C\frac{\cos\theta_i}{\cos\phi_s} \text{ or } hc_i = es\,(1 - \cos\theta_i) \tag{2}$$

where

hc_i is the additional gap between the internal spline groove and external spline tooth i because of the center displacement;

θ_i is the angular distance from the tooth No. 1 to the tooth I g given by $\theta_i = (i - 1)\,2\pi/N$;

N is the number of spline teeth;

$i = 1, 2, \ldots N$.

The total initial gap is the summation of gaps caused by position and form deviations of spline teeth and grooves, misalignments, and the additional gap when the spline centers are displaced from one to another in the direction of the normal gear load applied to the coupling. The gap of each spline tooth pair was calculated by Equation 3.

$$h_{ij} = hm_{ij} + hr_j + hs_{ij} + hc_i \tag{3}$$

where

h is the total initial gap without deflections, hm is the gap caused by the combined misalignment and modification in the tooth axial direction of external spline tooth and internal spline groove;

hr is the gap caused by lead crown of the external spline teeth;

hs is the gap caused by combined spacing errors of the internal and external splines;

hc is the gap because of center displacement of the internal and the external splines;

i, j are the tooth number and the station number, respectively, $i = 1, 2, \ldots, N$ and $j = 1, 2, \ldots, n$ for n stations.

When load is applied, the tooth engagement follows a sequence based on the gap distribution. The pair of teeth with zero clearance, as the tooth No. 1 of Figure 4, engages first and begins to transmit the torque load. That tooth pair deflects with load until the next pair of teeth engages. At that point two pairs of teeth are carrying different amounts of load, and deflect enough to bring another pair of teeth into contact. The third pair of teeth begins to share load, and also deflects. This process of sequential engagement continues with increasing load until the full load is applied. In the end, the number of pairs of teeth sharing load will depend on the gap distribution and the full load. The amount of load on each pair of teeth is a function of the gap distribution. The most loaded pair of teeth is the first one to enter into contact and is the most likely to fail (Refs. 6, 9).

Spline Tooth Stiffness and Elastic Deflections

Torsional and bending tooth deflections were considered in this study. Hertzian contact deflections were assumed small for spline teeth because of the conformal-type shape of the involute spline teeth in which both external teeth and internal grooves have the same radius of curvature. Thus, no special contact deflection calculation was considered. It was assumed that the elastic deformations are small compared to the size of splines such that the surface curvatures of splines over the contact zone remain unchanged. The supporting shaft was assumed to be solid or hollow cylinders when computing torsional deflection. The gear body was assumed to be rigid. Friction effects between engaged spline tooth pairs were not recorded.

Each spline tooth was divided lengthwise into n stations. Tooth elastic deflections were combined to produce a single stiffness constant of a spline tooth pair, $C_{\gamma m}$, which includes the stiffness of the internal and external teeth. The stiffness at each station was assumed to be an independent spring. The springs are added together when multiple stations and multiple pairs of teeth are in contact. Equation 4 was used to calculate the tooth deflections at each contact point.

$$\delta t_{ij} = L\delta_{ij}\, C_{\gamma m}^{-1} \tag{4}$$

where

δt_{ij} is the deflection of tooth i at Station j, µm;

$L\delta_{ij}$ is load intensity, N/mm;

$C_{\gamma m}$ is the spline tooth constant stiffness, N/mm/µm.

The torsional deflection was calculated over the engagement length of the splines, assuming a cylindrical shaft with circular cross section. The outside effective twist diameter of tooth section was defined as the root diameter plus 0.4 times the normal module (Refs. 10, 11). Shaft torsional deflection at each station across the spline length was calculated using the following equation that is given in (Refs. 10 and 11).

$$\delta t_j = \frac{4d^2 \times 10^3}{G\pi\,(d^4 - d_{in}^4)}\left(\sum_{k=1}^{j} L_k\right)\left(\sum_{k=1}^{j-1} X_k\right) \tag{5}$$

where

δs_j is the torsional deflection at Station j, µm;

d is the effective twist diameter, mm;

G is the shear modulus;

d_{in} is the inside diameter, mm;

L_k is the load at Stations k, N;

X_k is the distance between adjacent stations, mm;

j is the load at station number, $j = 1, 2, \ldots n$ for n stations.

Solution of the load distribution problem. The method described in References 10 and 11 for the solution of the load distribution problem of gear teeth was applied for splines. The method uses the concept of constant tooth stiffness to calculate the load sharing at each station from the overall gap and total load. Overall gap is given by the summation of initial gaps (Equation 3) plus the elastic deflections (Equations 4 and 5). The overall gap is given by Equation 6.

$$\delta_{ij} = h_{ij} + \delta s_{ij} + \delta t_{ij} \tag{6}$$

A pair of spline teeth comes into contact when one of the splined components moves in relation to the other, thus eliminating the gap between the teeth. The pair of teeth with the smallest gap is the first to enter into contact. Using that point as zero, relative gaps can be calculated to all other points related to that point zero. The relative gaps are then defined by Equation 7.

$$\delta r_{ij} = \delta_{ij} - \min(\delta_{ij}) \tag{7}$$

where

$min\,(\delta_{ij})$ is minimum gap among the stations of all teeth, which is the gap of the closest point of a pair of spline teeth.

The load intensity at each tooth and station is a linear function of the gap and the stiffness, and is obtained by rewriting Equations 4 and 7 into Equation 8.

$$L\delta_{ij} = \delta r_{ij} C_{\gamma m} \tag{8}$$

The total load F_g is the summation of the loads applied to all teeth and stations given by Equation 9.

$$F_g = \sum_{k=1}^{N} \sum_{j=1}^{n} L_{ij} \tag{9}$$

where
L_{ij} is the load at a point j of tooth i, N; $L_{ij} = L\delta_{ij} X_j$.

When the torque is applied to the spline coupling through a spur gear pair, the gear mesh load normal to the gear tooth, which is on the plane of action, pushes the internal splines against the shaft external splines in the same direction of the normal load. The normal load can be split into two components based on the gear tooth geometry — i.e., a tangential load and a radial load. The tangential load is applied in the circumferential direction tangent to the spline pitch diameter and is given by Equation 10.

$$F_t = \frac{2T}{d} \tag{10}$$

where
T is the torque applied to the gear that is connected to the splined shaft, Nm;
d is the spline pitch diameter, m.

The normal load pushes the splines down (Fig. 4) and reacts on some of the spline flanks, depending on their position related to the gear load. The gear radial load is given by Equation 11.

$$F_r = \frac{2T}{D_w} \tan \phi_w \tag{11}$$

where
D_w is the gear operating pitch diameter, m;
ϕ_w is the gear operating pressure angle, degrees.

A geometrical factor was developed to calculate the gear radial load component to the spline tooth center line, which is based on the spline tooth geometry and the position of the gear radial load direction to each spline tooth center line. The geometrical factor for the gear radial load is given in Equation 12.

$$a_i = \max\left\{0, \left[\frac{2\pi}{N}(\mathrm{N} - i + 1) + \frac{\pi}{2} - 2\phi_w + \phi_s\right] \cos\phi_s \right. \tag{12}$$

where
a_i is the radial load geometrical factor of spline tooth i;
N is the number of spline teeth;
ϕ_s is the spline tooth pressure angle at pitch diameter, degrees;
ϕ_w is the gear operating pressure angle, degrees.

In Equation 12, the factor a_i must be greater or equal to zero to signify load applied to drive spline flank in the direction of rotation; that is the reason for the maximum value condition.

The total load, F_g, applied to the spline teeth, is equaled to the summation of both gear load components, tangential and radial loads.

The load at the first station of the first tooth in contact is calculated from the gap distribution and total load using Equation 13, whose derivation can be found in References 10 and 11.

$$L_{11} = \frac{1}{nN}\left[F_g + C_{\gamma m} X_j \left(n N \delta r_{11} - \sum_{k=1}^{N} \sum_{j=1}^{n} \delta r_{11}\right)\right] \tag{13}$$

where
δr_{11} is the relative gap at Tooth No. 1 and Station No. 1;
δr_{ij} is the relative gap at Tooth i and Station j.

The difference in load between any two points is proportional to the difference in the relative gap between them. Thus, once the load at Tooth 1/Station 1 is obtained, the load at other points can be calculated by Equation 14.

$$L_{ij} = L_{11} + (\delta r_{11} - \delta r_{ij})\, C_{\gamma m} X_j \tag{14}$$

It is evident from examining Equation 14 that areas with greater gaps have lower load and areas with lower gaps have higher loads. It is also possible from Equation 14 to obtain negative results when the difference in gaps is large, which means that a specific point is not in contact. In that case the load at those points must be set to zero, and the loads at the contacting points need to be adjusted in a way that the summation of all load points equals to the total load as was given in Equation 9. After that, the adjusted loads are used to recalculate the deflections of each station of each tooth, and the gap analysis is updated. The loads can be re-calculated by Equations 13 and 14 with the new gap distribution. The load results are adjusted again if there are points with negative values. This iterative process continues until the difference between the gaps of two consecutive iterations falls within an acceptable error. Less than 3 mm is suggested in (Refs. 10-11) as a convergence criterion, which is usually achieved after a few iterations.

The final load distribution is the last iteration. A load distribution factor K_H is calculated for each spline tooth as the ratio of the total load at each tooth and the average tooth load, as shown in Equation 15 (Refs. 10-11).

$$K_{Hi} \frac{L_i}{L_{avg}} = \frac{N}{F_g} \sum_{j=1}^{n} L_{ij} \tag{15}$$

where
K_{Hi} is the load distribution factor of Tooth i;
L_i is the total load applied on Tooth i, which is the summation of loads applied to n stations across the tooth engagement length;
L_{avg} is the average load, which is the total load transmitted through the spline coupling divided by the number of teeth, N.

A load distribution factor equal to one means the load is evenly shared among all spline teeth. A factor greater than one means Tooth i is carrying more load because of non-uniform load-sharing among the spline teeth.

In a similar way, an axial load distribution factor across the spline tooth length can be calculated as the ratio of the load applied at a station and the average load applied to the stations of that tooth.

$$K_{Aij}\frac{L_{ij}}{L_{avg\,i}} = L_{ij}\,n\left(\sum_{j=1}^{n} L_{ij}\right)^{-1} \tag{16}$$

where

K_{aij} is the axial load distribution factor of tooth i, at station j;

n is the number of stations across the tooth engagement length;

$L_{avg\,i}$ is the average load applied to Tooth i.

The load distribution factor, K_H, can be applied to calculate spline tooth stresses, such as compressive, shear and bending stress, and to determine the spline coupling torque rating. The axial load distribution factor, K_A, can be helpful to assess effects of misalignments, shaft torsional effects — especially for long spline teeth — and to determine the need to use lead modifications such as lead crown.

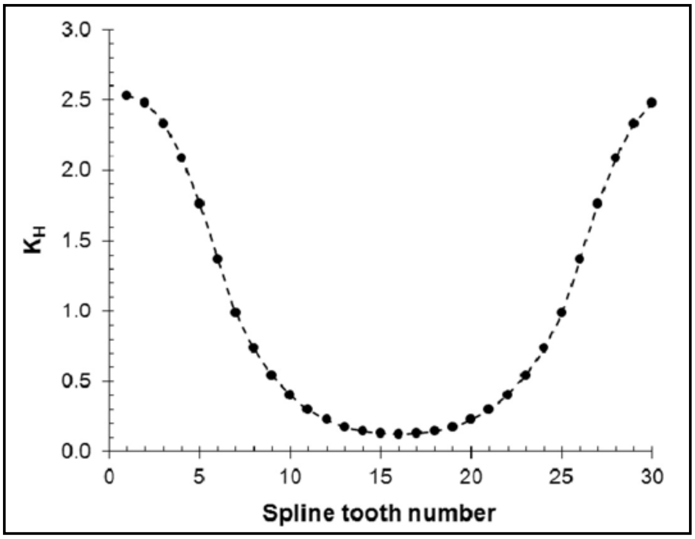

Figure 5 Tooth load distribution factor for 3,000 Nm torque and 0.050 mm side clearance.

Results

The proposed procedure to calculate load distribution of spline teeth was implemented into a spreadsheet and used to evaluate the effects of gear loads of a spur gear pair on the load distribution of spline teeth.

Example problem definition. In the study case, a spline coupling that connects a spur gear with internal splines to a splined shaft was investigated as a numerical example. Torque was transmitted from the gear teeth to the output shaft through the spline coupling. The internal and external spline data is shown in Table 1. The spline engagement length was 30 mm. The spline tooth constant stiffness was assumed as 16 N/mm/μm (Ref. 8). The spline teeth were divided into 18 stations, as recommended (Ref. 11) for gear teeth. Both the gear and the splined shaft were made of steel (modulus of elasticity 2.06 x 10 N/mm^2 and shear modulus 8.3 x 10^4 N/mm^2). The gear teeth and gear body were assumed to be rigid. The spur gear pair used for the case study was a 30-tooth gear pair, 1:1 gear ratio, 4 mm module, 20° pressure angle, and 120 mm center distance. Torque applied to the gear was 3000 Nm.

Tooth load distribution for perfect splines subject to gear loads. The study was done initially assuming perfect splines, which means no misalignments, spacing errors, or other errors, and no intentional modification.

The gap analysis was done using Equations 1 and 2 for different levels of clearances between the circular space width of the internal grooves and the circular tooth thickness according to the tolerances of Table 1. The corresponding load distribution problems were solved to determine the load applied to each tooth and as well as the load distribution factors. Figure 5 shows the results for 3,000 Nm torque that was applied through the gear to the spline coupling, and 0.050 mm side clearance, which resulted in 0.022 mm displacement of the internal spline center to the external spline center.

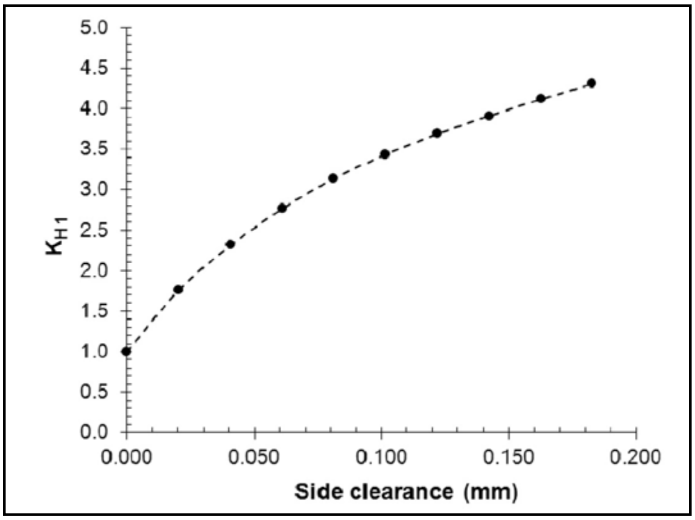

Figure 6 Relationship between maximum tooth load and side clearance.

Table 1 Example: Spline data (ANSI B92.1-1970) (Ref. 12)

Internal involute spline data		External involute spline data	
Fillet root side fit	Tolerance class -5	Fillet root side fit	Tolerance class -5
Number of teeth	30	Number of teeth	30
Spline pitch	16/32	Spline pitch	16/32
Pressure angle, degrees	30°	Pressure angle, degrees	30°
Base diameter, mm	41.2445 ref.	Base diameter, mm	41.2445 ref.
Pitch diameter, mm	47.6250 ref.	Pitch diameter, mm	47.6250 ref.
Major diameter, mm	50.90 max.	Major diameter, mm	49.12/49.07
Form diameter, mm	49.33	Form diameter, mm	45.92
Minor diameter, mm	46.18/46.05	Minor diameter, mm	44.02 min.
Circular space width		Circular tooth thickness	
Actual, mm	2.570/2.537	Maximum effective, mm	2.494
Minimum effective, mm	2.494	Actual, mm	2.421/2.388

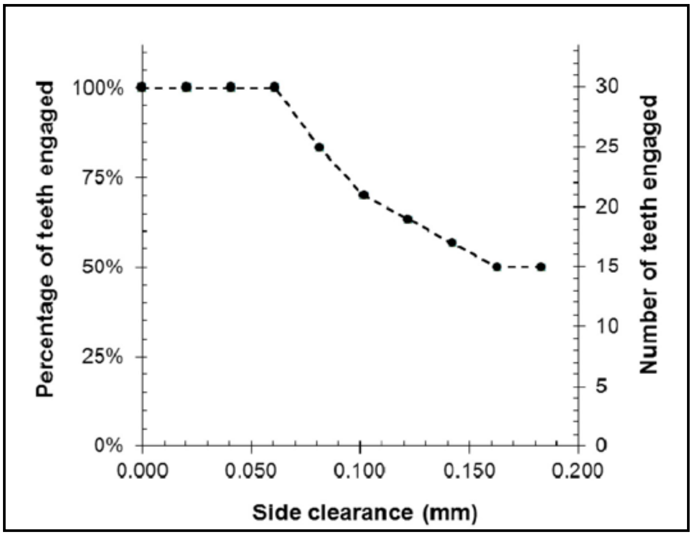

Figure 7 Number of teeth in contact, based on side clearance for perfect splines.

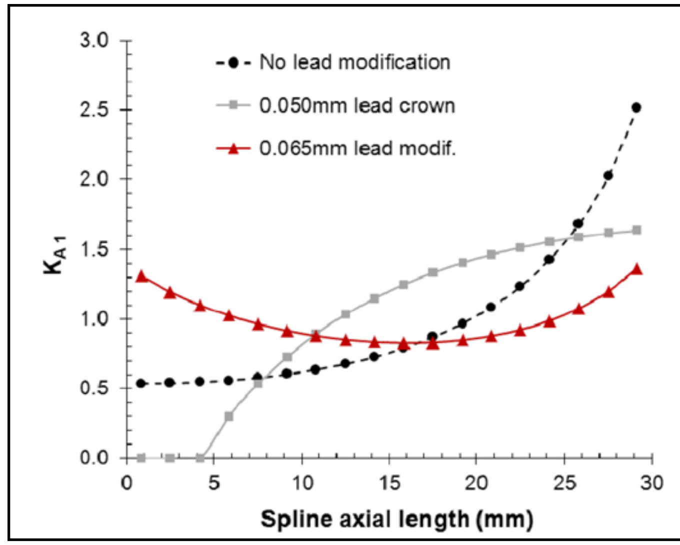

Figure 8 Axial load factor without and with lead modifications.

The results of Figure 5 show that the first pair of spline teeth to engage, which is Tooth No. 1 and with the smallest gap, carried 2.53 more load compared to the average load (uniform load distribution). Tooth No. 16, which is diametrically opposite to Tooth No. 1 and had the largest gap, carries only 12% of the average load ($K_{H16}=0.12$).

Tooth side clearance effect on spline maximum load distribution factor. The maximum load distribution factor K_{H1} changed as a function of the side clearance, which depends on the dimension of both internal and external splines. Figure 6 shows the maximum load distribution factor results, which were found on Tooth No. 1, for different side clearance values. It was observed a significant effect of side clearance on K_{H1} results.

As expected, the worst case was found at the maximum side clearance, which occurs when the actual circular space width is in the upper limit and the actual circular tooth thickness is in the lower limit. At that condition the maximum load distribution factor K_{H1} was 4.31. As the side clearance decreases, K_{H1} goes down towards the value of 1, which means uniform load distribution among the spline teeth.

The overall stiffness of the spline coupling depends on the number of pair of teeth engaged. Looking back to Figure 5, it can be observed that all teeth were engaged and carrying some amount of load. In other cases, with larger side clearance, some of the spline teeth were not in contact because of the large gap difference at the first tooth to engage. Figure 7 shows the percentage of teeth engaged for each side clearance level; 100% means that all 30 teeth were engaged and carrying some load. Under maximum side clearance, 50% of the spline teeth (or 15 teeth) were engaged and carrying some load.

Effect of tooth spacing error on maximum load distribution factor. Both manufacturing deviations and intentional modifications on spline teeth have a significant effect on spline load distribution — especially spacing errors (Refs. 3, 8). Repeating the analysis with a spacing error of 0.015 mm added to the pair of teeth labeled No. 1 for the same case study, with 0.050 mm side clearance, resulted in a 51% load increase on that pair of teeth. Tooth No. 1 carried more load until the next pair of teeth could engage to begin sharing the load. The load distribution factor increased from $K_{H1}=2.53$ (no spacing error) to $K_{H1}=3.83$ (0.015 mm spacing error).

Effects of lead modifications on axial load factor. In the tooth axial direction, the shaft torsional effect was observed as a significant parameter that affects the load distribution across the spline tooth engagement length. The axial load distribution factors were determined in the case study with 0.050 mm side clearance. The maximum factor K_{A1} was 2.51, which is in line with previous studies in the literature (Refs. 4, 6). On Tooth No. 1 the axial load distribution factor varied from 0.54 at one tooth end to 2.51 at the opposite end, which is the tooth end of torque reaction. Lead crown and lead modification were then applied to the spline teeth to investigate their effects on the axial load distribution factor. Figure 8 shows the results without any axial modification, with 0.050 mm lead crown and 0.065 mm lead modification.

The results of Figure 8 show that lead crown reduces the high loads at the end of the tooth, but increases load in the middle of the spline tooth. Moreover, three stations on the free end of the spline teeth carried no load. On the other hand, the lead modification produced a more uniform load distribution across the stations, in the axial direction of the tooth. That means about a 46% reduction on maximum tooth load in the axial direction. **PTE**

Discussions

- A simple model was developed for load distribution analysis of splines that are subjected to spur gear mesh loads and side-fit splines that have sufficient radial clearance between both major and minor diameters.
- A gap analysis was performed, based on the spline tooth dimensions and direction of gear normal load.
- The gear mesh load was calculated and used to determine the sharing among the spline teeth that were divided axially into stations.
- A constant for tooth stiffness was used to calculate tooth deflections.
- The load distribution problem was solved using a simple approach from industry standards for load distribution in gear teeth. The method was implemented into a spreadsheet and used to investigate the load distribution of a spline coupling to connect a spur gear to a splined shaft.
- The results showed a significant effect of side clearance on the load-sharing among the spline teeth for side fit-type splines.
- The case study results — including spacing errors and lead modifications — were in line with previous studies in the literature.
- On spline couplings used to connect gears to shafts, the gear loads — such as radial and tangential loads — add an extra increase to the load factor due to the gap distribution changes caused by the center displacement of one splined component to the other. This is a function of the side clearance.
- The circular space width and circular tooth thickness, which define the side clearance range, are critical parameters for the load distribution factor of spline couplings in gear applications. Those dimensions and tolerances are defined according to the tolerance class and fit class of industry standards. The numerical example used in this study was a tolerance class 5 and fit class h spline. Splines with tolerance classes and fit classes greater than class 5 are expected to present an even larger increase on maximum tooth load and, consequently, higher stresses.
- The simple method incorporated into the spreadsheet could be helpful to designers to quickly assess load distribution of spline teeth in gear applications, determine tooth stresses, and define lead modifications as needed.
- As future work, the model could be extended for helical gears by including two additional momentums of the helix angle.

Acknowledgments. *The authors thank Eaton's Vehicle Group for their support in developing this work.*

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Global Industrial Outlook: The C.H.I.E.F. Issues

By Brian Langenberg, CFA

Global economic activity remains good—noise about China slowing, notwithstanding—and despite the ObamaCare debacle, non-Christmas-related consumer spending looks pretty good. Europe is *still* Europe and China continues to grow 7-8 percent—even as the government seeks to clamp down on its own shadow "banking" system. And India remains a mess.

Backing up our views are meetings we attended with these companies during the month of December:

Marathon Oil; General Electric; United Technologies; AGCO; Danaher Corporation; 3M Company; Lennox International.

In February 2012 we began to write to our clients that the U.S. was becoming a structurally more attractive place to manufacture—despite ourselves—because of cheap natural gas and rising international labor rates. But both drivers are *gradually* playing out.

There is going to be a rising debate in the U.S. and abroad about—depending upon your philosophy of life—economic opportunity or income inequality; they are two sides of the same coin.

The U.S. has five "CHIEF" issues impacting the economic recovery and long-term health of the U.S. economy. Three are more or less going to be sorted out—housing, immigration and financial services reform—while two require action—corporate tax reform (Congress, President Obama), and education/employment (several moving parts).

C.H.I.E.F.

Corporate tax reform
Housing
Immigration
Education and Employers
Finance

Corporate tax reform: This *will* get fixed—either in 2015 with a new Congress or the next presidential election. There is understanding on both sides of the aisle that corporate taxes still keep a lot of work offshore. Instead of being esoteric we will make this simple. Boeing has $35 billion of capital invested in its operations—plant, working capital, tooling, etc. In 2012 they earned $5.9 billion of pre-tax income. They paid $2 billion in tax—34 percent and close to the 39 percent statutory rate. As a result their after tax income was $3.9 billion, or about an 11 percent return on capital. Respectable—but hardly excessive given the incredible complexity of their operations, products and services, and position as a premier U.S. exporter! BAE Systems, a U.K.-based aerospace and defense company, enjoyed a U.K.-based 21 percent tax rate. Had Boeing enjoyed a similar regime, their income would have been *$1.1 billion higher* and generated a more appropriate 13-14 percent return on investment. Instead of being criticized for moving operations to South Carolina, perhaps they should be thanked for *not* moving operations to the U.K.

Housing: For all the political yapping, including any rants I might do on *Facebook*, there is *one* area of broad bipartisan agreement: *no massive foreclosures*. This is why we are going to have QE3, QE4 and QE-to-infinity until home values are comfortably above mortgages. The political classes want to keep their jobs, and all constituencies want to remain in their homes—*period*.

Immigration: Insanity is doing the same thing over and over and expecting a different result. Reality check: 11 million people who are already living here *are not leaving*. As a practical matter, I refuse to waste my time arguing this point or whether it is right, wrong, etc. An accommodation will be made that leaves everybody grumbling sooner or later, but allows the U.S. to move forward. Meanwhile, an improving economy is lowering the temperature on the conversation. Corporate entities are increasingly pushing for "reform" because they want to hire minimum wage labor; think hotels, leisure, retail. I suspect we will see piecemeal reform vs. a grand bargain—but I am not sure.

Education and Employers: Stand by for a soapbox here, and I mean 1) the education mafia at the public school and college level, 2) anybody stupid enough—not to mention their parents—to borrow money and attend college for six years without a goal, and 3) the 85 percent of U.S. management teams that do not adequately invest in employee training, abrogate their labor force engagement to resume-screening software, HR, and legal psychobabble, and then complain when they can't find "good" employees. We are at, or near, a tipping point and change is underway.

Failure of the public education system in the U.S. has multiple factors, but suffice to say that with the majority of kids leaving U.S. public schools either unprepared for college, or trade skills to get hired, the economic costs are huge. Worse—we see no quick fix.

College is another story. Years of hype and marketing by the "college industry"—all of which are "for profit"—fooled many to thinking a college degree is an automatic ticket to a good income. Wrong. As Michael Goldstein, the publisher of this fine magazine has commented, "We make 500,000 sociology majors per year. We need 50,000."

Excluding the "elite" schools, a four-year degree will cost about $120,000 net of grants and discounts and leave the graduate with a large debt load.

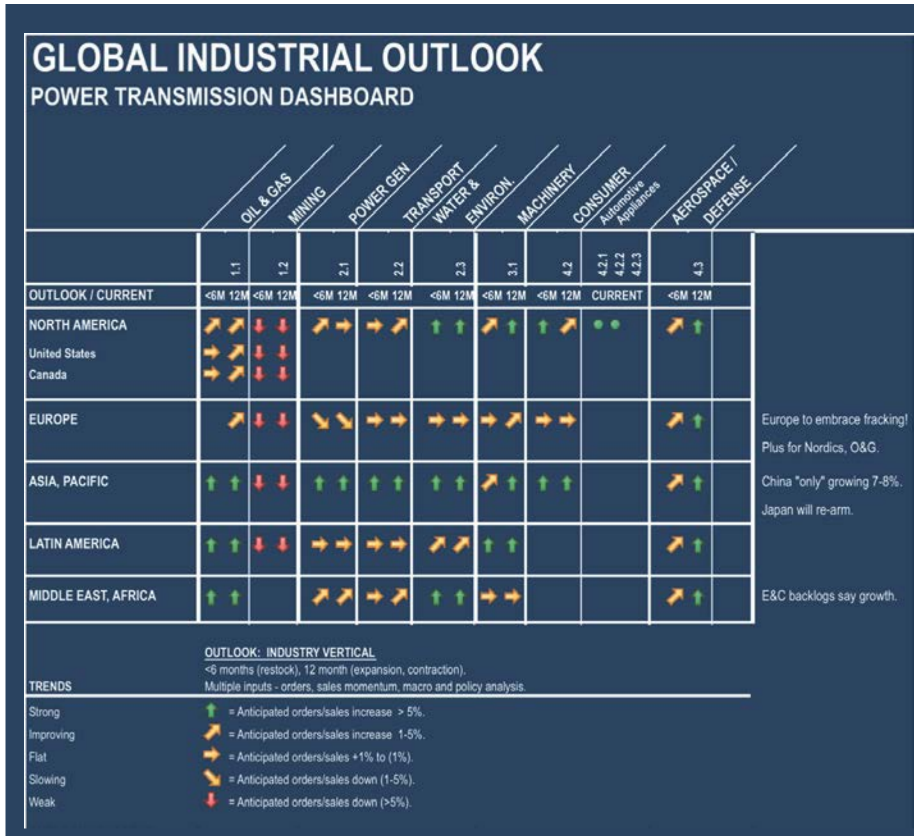
Aside from indisputable facts and figures, the opinions and conclusions are the author's and do not necessarily reflect the position of Randall Publications LLC.

Revolution is afoot. I am an adjunct instructor in finance for Southern New Hampshire University, which is leading the charge to deliver a cost-effective online education that comes to $38,000 for a quality degree. My best research associate graduated from that school. Pricing and enrollment are heading down and schools that do not drive out waste and costs will close. Expect "The Theory of Hair"-type courses and administrators to go first.

Finally, let's talk about the majority of large- and medium-sized U.S. companies. They are going to *have to get their hands dirty instead of outsourcing everything*. For all the complaints about U.S. labor, few companies see leadership, development and retention of a loyal employee base as their responsibility or a competitive advantage. I am not espousing paternalism, but most HR and benefits consultants are there to prevent lawsuits, craft executive compensation schemes for their C suite and boards and to do paperwork.

Here are some fantastic and executable ideas for larger employers that would a) save companies money, b) provide benefits to their employees and c) lead to fewer anti-business voters. To some extent these can be applied even in a small company, or if a few banded together for common purpose.

Healthcare: Average premium for a family of four in an employer healthcare plan is $11,100, according to Hewitt. Primary care physician salary and benefits-per-family of four is $640. Somebody needs to explain to me why a competent manufacturing organization with even 500 people can't do a better job of negotiating discounts and driving out cost than any benefit consultancy, insurance company or hospital. Self-insure the rest except catastrophic and pre-fund long-term health care policies for each employee they can take with them. Save your company money and lower your employees' costs. I think any large manufacturer that took this on could drive 30-40 percent reductions in healthcare costs. Even smaller employers could do this by banding together; a doctor that does not have to pay malpractice and have staff spend 40 percent of their time on paperwork has a much lower cost structure.



Education: Partial tuition reimbursement is the wrong approach to retaining employees. Better yet, negotiate steep price discounts for volume with schools frequented by employees and then reimburse. No excuse not to negotiate with vendors. If an employee attends a more expensive school, reimburse up to a set point.

Training: I am sick of hearing about a qualified labor shortage — particularly from larger companies. The true unemployment rate is about 15 percent — whether young with no skills or older with the wrong skills — maybe 10-15 million people. And yes, there are social issues. Tough; at least half would be good employees with training and leadership. Just do it even if they don't all work out.

Finance: Wall Street, capital markets, Dodd-Frank, and compliance. Expect continued drag for some time, given the higher regulatory costs of Dodd-Frank, anti-business government attitudes and also, lamentably, the fact that banks still get almost "free" money, which they can turn around and hold in government securities as long as the Fed Funds rate is held abnormally low. Not an ideal situation, but at the same time the banking system is rebuilding its capital base.

That is it for the potential enablers; let's talk end markets.

Oil & Gas: Pockets of relative slack potentially occurring as Petrobras and other multinationals slow their capital spending plans and North American onshore drilling activity also slows. Increasingly, the majors are going to focus on earning a return on their huge investments and that means driving production growth. National oil company investment remains strong. Top growth area likely to remain midstream — pipeline, transportation and infrastructure — as investment must catch up with upstream production. Expect stable to higher demand overall, given world energy needs and growth.

Mining: Global equipment orders remain weak though we anticipate rate of decline to slow. We maintain our view that new-equipment demand can decline another 30-35 percent through 2015. U.S. coal, particularly in Appalachia, remains the worst. On the flip side, high utilization rates will continue to support consumables and parts demand; Atlas-Copco is always a great source to track the sector.

Power generation: U.S. wind continues to roar back against easy comparisons, renewed favorable tax treatment and large utilities taking the path of least resistance and installing wind turbines at $2-4 million a crack, instead of fighting to put up a new gas plant (forget coal in the U.S. anytime soon) in an environment devoid of demand growth, rate relief or a reasonable regulatory environment. Wind is by no means the most efficient energy source, but it is a Band-Aid. In the meantime, your company hopefully benefits.

Transportation infrastructure: U.S. infrastructure spending will remain flat until *late* 2016 at the earliest, as it would require bipartisan support and willingness to spend on *infrastructure*. Forget it. Roads with potholes and dangerous bridges don't vote and are thus not a Democratic constituency demanding funding, and the bulk of Republicans (Tea Party) don't want to spend — period. Global passenger rail activity remains strong and we continue to see new order announcements. Beneficiaries include Bombardier, Siemens, Alstom and Wabtec.

Water & environmental: Municipal demand is improving, given pent up maintenance requirements but also higher interest in capital projects. Improving home prices are driving the improved tone, but we still believe significant growth is a couple years out. Industrial activity — particularly dewatering and mobile equipment associated with mining — remains weak.

Machinery: Overall picture continues to improve. Construction equipment production is gradually recovering after three straight quarters of dealer inventory cuts. Mining equipment will remain weak; we still expect a further decline of up to (20-30 percent) in spending over the next two years, and even worse on new equipment. *Truck*: while 2013 production forecasts drifted downward throughout the year, we anticipate solid U.S. demand in '14 driven by high fleet utilization. Internationally we see continued strength in China and Brazil. India remains awful. *Agricultural equipment*: Expect flat markets at best in '14, given farm investment cycle maturity and a pull-back in some grain prices. John Deere growth investment emphasizes Brazil, China, India and Russia. AGCO is emphasizing Russia, Africa and Latin America as providing long-term potential.

Consumer (auto, appliances): Increased consumer confidence supported by recovering home prices and improved stock markets is driving higher demand for durable goods. Auto manufacturers continue to add North American capacity. We expect some continued growth in auto (both OEM and aftermarket service), appliances and housing. Expect the Fed to keep interest rates low, at least through the November 2014 elections, a slight reduction in "QE" notwithstanding. The U.S. economy is better, but not that much better. European comparisons are now easier against a low basis but we do not see any real pickup.

Aerospace/Defense: Commercial aviation spares continue to run strong — General Electric recently reported a 16 percent increase year-over-year on a 5 percent rise in global flight hours; pent up maintenance and restocking by airlines drove the improvement. Build rates at both Boeing and Airbus and increasing order books at Bombardier signal a robust outlook. In Defense, the worst is over given the recent budget deal but do not expect a huge pick up for some time as the U.S. is politically in "withdrawal" mode from exercising global influence. I don't agree with it, but it is happening.

Focus Company: Marathon Oil (MRO)

Energy — exploring, producing, transporting and selling of oil, gas, coal — influences practically every end market we cover. In December we had the opportunity to attend the analyst outlook meeting of Marathon Oil and speak with new CEO Lee Tillman and his management team. Mr. Tillman was recruited to Marathon Oil after a successful 24-year career at Exxon, where he headed engineering for the Exxon Mobil Development Company.

Marathon is a $24 billion market cap exploration and production company. Recruiting an Exxon Mobil executive was no accident; XOM has built a multi-decade reputation as *the* most financially disciplined oil producer. Like others, MRO has spent massive amounts of capital exploring, developing and acquiring properties to drive production growth. Already possessing an above average track record operationally, their priorities are to drive continued operational efficiency (production up time) and *rigorous* portfolio management. In other words the numbers and project risk profiles will drive the decisions.

Their capital spending has increased from about $3.3-3.5 billion per year in 2010-2011 — to about $5 billion annually the past couple of years. What *you* care about is where they will spend capital. As the industry shifts from "find the stuff" to "pump it out" to "pump it out better," i.e. — higher profitability — equipment manufacturers that are aware of customer needs (like MRO) and bring solutions and services that drive oil field uptime and yield will have better growth prospects. Any equipment manufacturer serving Schlumberger, Halliburton and Baker Hughes should still do well.

Our Integrated Company Dashboards (ICD) will give a better sense of these trends. These analyses are available on our website for $199, but readers of *Power Transmission Engineering* magazine can email me directly at *Brian@Langenberg-llc.com* and ask for a copy by putting "*PTE* Offer" in the subject line and the ticker for which company they want. Choose one from: ALFA.IX, AME, ATCOB.IX, CAT, CMI, DOV, EMR, GE, HON, MMM, MTW, ROK, SDVKF, SKFB, UTX, or XYL. We also offer subscriptions at special rates for *PTE* subscribers. **PTE**

Brian K. Langenberg, CFA, has been recognized as a member of the Institutional Investor All-America Research Team, a *Wall Street Journal* All-Star, and *Forbes/Starmine* (#1 earnings estimator for industrials). Langenberg speaks and meets regularly with CEOs and senior executives of companies with over $1 trillion in global revenue. His team publishes the *Quarterly Earnings Monitor/Survey*—gathering intelligence and global insight to support decision-making. You can reach him at *Brian@Langenberg-llc.com* or his website at *www.Langenberg-LLC.com*.

Fluid Power Free-For-All

IFPE Offers the Right Balance of Technology and Education

CONEXPO-CON/AGG 2014 has the construction sector covered. With more than 125,000 attendees and 2,400 exhibitors, the asphalt, aggregates, concrete, earthmoving and mining industries are well represented. IFPE, the International Exposition and Technical Conference co-located with CONEXPO-CON/AGG 2014, integrates fluid power with other technologies for power transmission and motion control applications. It's here in the IFPE hall that you'll find a few names covered in the pages of *Power Transmission Engineering*.

Held every three years, the exposition showcases the newest innovations and expertise. The event includes more than 400 exhibitors, product concentration areas, more than 100 education sessions, keynote presentations, college-level courses in hydraulics and pneumatics. Technologies and products include aerospace, energy, automation, material handling, off-highway, fluid power, power transmission and more.

IFPE 2014 education provides both application-based and research-driven sessions and the IFPE technical conference puts an emphasis on new technologies and methods related to improved analysis, design, manufacture and performance of fluid power components and systems for mobile and industrial markets. One registration allows access to more than 80 presentations and keynotes, and includes a flash drive with conference proceedings. Certificates for PDH or CEUs are available.

Additionally, a new Fluid Power Seminar Series is free to all attendees and will present practical information to help them better understand the operation of hydraulic and pneumatic components, circuits and systems. Eight sessions will be offered, including Best Practices in Mobile Hydraulic Maintenance; Reducing Leakage and its Environment Impact; Contamination Management in Hydraulic Systems; Electronic Control of Mobile Hydraulics; and Troubleshooting Excavators, Loader Backhoes and Aerial Lifts. The series is presented by *Hydraulics & Pneumatics* magazine.

IFPE is also the site of the 2014 International Fluid Power Summit, a meeting of leaders from fluid power associations across the globe. Show co-owner, the National Fluid Power Association (NFPA), is hosting the event. "NFPA is pleased to host this important event; it's an opportunity for fluid power professionals to work together to strengthen the worldwide fluid power industry," stated Eric Lanke, NFPA CEO in a recent IFPE press release.

"As a global gathering place for introduction of the newest products and technologies, IFPE is the ideal venue for this prestigious event, and we welcome their participation," stated Melissa Magestro, IFPE show director. In conjunction with the summit, an International Fluid Power Statistics Committee meeting will be held at the show, with a focus on international statistical/data collection programs.

Exhibitor Highlights

Bosch Rexroth (Booth 80216) will exhibit the newest developments in hydraulics technology, including axial piston variable pumps, mobile controls, hightorque hydraulic motors,

drive train solutions, and more at the CONEXPO-CON/AGG show, March 4-8 in Las Vegas. Visitors to the Rexroth booth can expect to see a broad range of leading hydraulics technology used in loaders, cranes, telehandlers, excavators and other mobile equipment. The company will display several notable products and systems designed to boost energy efficiency, improve productivity, and expand capabilities of mobile machinery (*www.boschrexroth.com/conexpo*).

MICO (Booth 81233) will exhibit modulating hydraulic power brake valves providing directional control of brake system pressure. MICO also has accumulator charging valves for use in open center hydraulic systems and load sensing hydraulic systems (*www.mico.com*).

Schaeffler Group (Booth 81139) will offer innovative products and support services that can dramatically improve the performance and reliability of pumps used in wastewater, dewatering and petrochemical applications. These include ball bearings, linear bearings, plain bearings, roller bearings, spherical bearings and more (*www.schaeffler.us*).

Gates Corporation (Booth 80516) will offer a wide variety of construction solutions, including its mobile equipment, poly chains, industrial hoses and MegaSys assemblies. Condition monitoring equipment, pulleys, belts and fittings will also be included. (*www.gates.com*).

AST Bearings (Booth 82014) will showcase its bearing products and services and has recently added 475 new products to their mounted bearing category. The subsets for the mounted brackets are two-bolt pillow blocks, two-bolt flanges, three-bolt flanges and tapped base housings (*www.astbearings.com*). **PTE**

March 5–6—Lean Transformation Summit 2014. Orlando, Florida. Continuously improving management systems allow companies to provide better quality, more reliable delivery, better service and lower costs. The summit provides attendees with relevant, real-world learning sessions to enhance their lean efforts. Plenary sessions offer feedback from organizations that have developed successful lean programs including Cardinal Health, Food Bank for New York City and GE. Breakout sessions follow that give a firsthand account of the challenges faced during the lean transformation. Additionally, small interactive learning sessions allow attendees to reflect on applications and methodologies on the leading edge of lean thinking. For more information, visit *www.lean.org*.

March 11–13—Gearbox CSI: Forensic Analysis of Gear and Bearing Failures. Sheraton Suites, Philadelphia Airport, Philadelphia. Determining the cause of a failure in a gearbox is like a "who done it" mystery. What caused the failure? The bearings, a gear, the lubrication or a shaft problem? Where do you start, and how can you tell? This seminar helps gear designers gain a better understanding of various types of gears and bearings. Learn about the limitations and capabilities of rolling element bearings and the gears that they support so you can properly apply the best gear-bearing combination to any gearbox, whether simple or complex. A certificate will be awarded upon completion of the seminar. For more information, visit *www.agma.org*.

March 17–20—MODEX 2014. Georgia World Congress Center, Atlanta. MODEX 2014 is the industry's newest expo for the manufacturing and supply chain industries. At MODEX attendees will meet 800 of the leading providers in the supply chain industry. The MODEX Supply Chain Conference includes 150 sessions with keynote presentations from Edward H. Bastian, president of Delta Air Lines; former Walmart CEO Lee Scott; and Scott Sopher, principal with Deloitte Consulting. Some of the areas of interest include: material handling equipment, packaging equipment, dock and warehouse equipment, supply chain management and education. For more information, visit *www.modexshow.com*.

March 25–26—AWEA Wind Energy Regional Summit – Northeast. Portland, Maine. The Wind Energy Summit is for those already involved in wind project development in the Northeast or those that want to learn more about entering into this region's market. Seminar attendees will learn about significant issues related to land-based and offshore wind project development, get expert insight into the region's growth potential, and learn how to position your company for the best return on your investments. From interactive peer-led educational sessions to exclusive networking opportunities, this summit will give you and your company the tools you need to be successful in the Northeast wind market. For more information, visit *www.awea.org*.

April 1–2—Human Error Prevention Seminar. Charlotte, North Carolina. The principles and practices of human error prevention are universally applicable regardless of the type of industrial, commercial or governmental enterprise, and regardless of the type of function performed within the enterprise. This seminar is truly unique and up to date with the latest developments in human error prevention. Ben Marguglio's new taxonomy of human error causal factors and his human error-related models demonstrate his leadership in this subject. Examples and case studies amply reinforce the human error prevention principles and practices. Upon seminar completion, attendees will be able to: improve process productivity, safety and quality using new and unique techniques, tools and behaviors for error prevention, detection and mitigation; Address the four fields of focus – (1) hazards and barriers, (2) error-inducing conditions and counteracting behaviors, (3) non-conservative decisions and counteracting behaviors and (4) prevention of error recurrence; design, implement, manage and assess a Human Error Prevention initiative. For more information, visit *www.hightechnologyseminars.com*.

April 7–11—Basic Training for Gear Manufacturing. Richard J. Daley College, Chicago. The AGMA Training School for Gear Manufacturing will enable you to become more knowledgeable and productive. The Basic Course teaches students to set up machines for maximum efficiency, to inspect gears accurately, and to understand basic gearing. Although the Basic Course is designed primarily for newer employees with at least six months' experience in setup or machine operation, it has proved beneficial to quality control managers, sales representatives, management, and executives. This course offers training in: gearing and nomenclature, principles of inspection, gear manufacturing methods, hobbing, shaping and more. Although all training is basic, on manual machines, everything that students learn is valid and applicable with the CNC equipment commonly in use. By using manual machines, students can see the interaction between the cutting tool and the workpiece. They understand the process and the physics of making a gear. For more information, visit *www.agma.org*.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Kollmorgen

TEAMS UP WITH SEGWAY FOR PACK EXPO

Riding (or some say driving), a Segway Personal Transporter (PT) should be on everyone's bucket list — owning one is an even better idea. Mickey Simmons, research analyst at Proctor & Gamble, Cincinnati, Ohio, is experiencing the fun firsthand, after winning his own Segway x2 PT at the 2013 Pack Expo. This is the second time that Kollmorgen and Segway have teamed up to give away a brand new Segway x2 at Pack Expo as a way to demonstrate the Segway x2 drivetrain servo system designed and developed by Kollmorgen.

The Kollmorgen motor technology developed for Segway has unique and critical technical characteristics integral to its successful design and manufacture. Many of these breakthrough advances offer similarly beneficial application in packaging equipment. First and foremost is the incredibly high level of efficiency and high torque/power density for a servomotor capable of powering a large off-road tire drivetrain and carrying a full grown man — yet small enough to fit in the palm of a hand. Advanced winding technology, including patented dual motor windings to allow the rider to "limp" home in the event the primary winding fails, and multi-design function overmolding are among the breakthrough features of the x2 servo that provide this power and rugged compactness. The performance of the x2 servo drivetrain is best demonstrated by the Segway's excellent torque performance across its full range of speed capability — from a smooth, slow-speed crawl to 10,000 rpm — while ensuring the safety of the passenger with smooth, seamless acceleration and deceleration. An aggressive price point was another critical Kollmorgen development criteria for the servo system to ensure a successful product launch for the x2 PT in the consumer marketplace.

Throughout its design and development process, the x2 drive train provides an excellent demonstration of Kollmorgen's expertise and capability being partnered with the creative solutions objectives of their customer. Specifically for the x2 design technology, Kollmorgen's ability to leverage R&D breakthroughs in controls, drives and motors from their extensive industrial applications expertise was critical. Likewise, the economies of the Segway PT motor manufacturing system originated from Kollmorgen's high volume stepper motor manufacturing technology.

As for Simmons, winning a Segway x2 came as a real surprise, along with learning that a Kollmorgen servo is the power behind the machine and that Kollmorgen provides such technologically advanced solutions far beyond traditional electric servo motors. Back home in Cincinnatti, Simmons and his wife, along with Keary Shaub, Kollmorgen field sales engineer, and Teri Shaub, Kollmorgen technical applications specialist, were treated to a great experience with personalized Segway x2 training and a tour of the beautiful downtown Cincinnati waterfront by the local Segway dealer, Segway of Cincinnati. In short order, Simmons is planning to order another x2 so that he and his wife can both ride the local Loveland Trail in Mason, Ohio and take their x2s camping… Scratch one from the bucket list.

Bosch Rexroth

OFFERS EDUCATIONAL RESOURCE KITS

Bosch Rexroth is offering educational resource kits to help machine builders, systems integrators and manufacturing end-users address engineering challenges in factory automation, linear motion and mobile engineering. Packaged on convenient USB drives, each kit provides useful technical in-

formation and content relevant to its core topic. Specific kits covering factory automation, mechatronics and linear motion, hydraulics, packaging and printing, and mobile technology are available for free by request at the website below. The kit materials are drawn from Bosch Rexroth's extensive library of drive and control application case studies, expert technical papers, current product brochures, links to state-of-the-art engineering and design tools, and the company's latest videos — carefully selected and combined in one convenient digital package. For additional information, visit *www.boschrexroth.com*.

Maxon Motor

NAMES NEW COO

As of 1 December 2013, **Peter M. Grütter**, 47, has been named COO and member of the executive management board of Maxon Motor AG in Sachseln, Switzerland. Grütter previously worked for over 15 years in various management roles within the Schindler Group, most recently as senior vice president of the corporate quality division.

"We are very pleased that Peter M. Grütter, with his extensive international management experience, will be taking on the role of COO. He is a team player who actively maintains direct contact with customers and employees," said Eugen Elmiger, CEO of Maxon Motor.

Grütter worked at Schindler for 15 years. After several years as vice president of manufacturing at the Swiss factory for high-performance elevators, he went on to manage the production sites in the USA from 2005 to 2010. Later, Grütter held the position of senior vice president of the corporate quality division. From 2001 to 2005, he was president of Schindler's vocational training center in Switzerland.

Grütter studied industrial management and manufacturing at the Swiss Federal Institute of Technology in Zurich (ETHZ). After his studies, he worked in various roles, including as factory manager at Rieter AG, before he joined Schindler in 1998. The Swiss-Canadian dual citizen is married with two children.

Manufacturing Institute

HONORS FEMALE EMPLOYEES AT BISON GEAR

Bison Gear and Engineering is pleased to announce that two of its associates, Sylvia Wetzel and Christi Brazener, were recently recognized by the Manufacturing Institute with the Women in Manufacturing STEP (Science, Technology, Engineering and Production) Award. The STEP Awards honor women who have demonstrated excellence and leadership in their careers and represent all levels of the manufacturing industry, from the factory-floor to the C-suite. In 2014, there are a total of 160 women being honored.

Sylvia Wetzel

"We are very excited to have not one, but two honorees this year," said Martin Swarbrick, president and CEO of Bison Gear and Engineering. "Both Sylvia and Christi have shown remarkable dedication and service to Bison, and deserve to be recognized for their excellence."

Nominated by the College of DuPage, Wetzel, chief learning officer at Bison, is a solution-driven leader whose passion for education has led her to a role with a myriad of responsibilities and facets. Through humanitarian efforts, such as BisonCares, internal efforts, and external collaborations, Sylvia is actively engaging the workforce of today while simultaneously cultivating tomorrow's workforce. As an internal resource, she initiated a health program that offers free risk assessments, an on-site health provider, and dollar incentives for employees to embrace a healthy lifestyle.

Christi Brazener

Brazener, customer care manager, personifies the company's approach toward service, PRIDE (People Relentless In Delivering Excellence). She was instrumental in creating a customer service call center so that a customer call never goes to voicemail. Christi has developed a personal relationship with most of Bison's customers that allows the company to get the most accurate information about upcoming demand. She acts as a true customer advocate and a liaison between the customers and the internal workings of Bison.

"These 160 women are the faces of exciting careers in manufacturing," said Jennifer McNelly, president, The Manufacturing Institute. "We chose to honor these women because they each made significant achievements in manufacturing through positive impact on their company and the industry as a whole. The STEP Awards are part of the larger STEP Ahead initiative launched to examine and promote the role of women in the manufacturing industry through recognition, research, and best practices for attracting, advancing, and retaining strong female talent."

A recent survey from Deloitte and The Manufacturing Institute found that nearly 70 percent of American manufacturing companies have a moderate to severe shortage of available, qualified workers. In its second year, the STEP Awards Program was initiated in 2013 while Bison's Chairman and Owner, Ron Bullock, was Chairman of the Manufacturing Institute. On February 6, The Manufacturing Institute recognized 160 recipients of the STEP Awards at a reception in Washington, D.C.

Heidenhain

HIRES NORTH AMERICAN PRODUCT MANAGER FOR CONTROLS

Heidenhain Corporation recently announced the new hire of John Parker as the North American Product Manager for Controls. Bringing nearly 30 years of experience in the machine tool industry to this position, Parker will serve as Heidenhain's point person for the Computer Numerical Control (CNC) product lines (both Heidenhain and Acu-Rite brands), as well as serve as a liaison between the company's sales and marketing and engineering departments. Parker holds an electronics degree from Devry Institute of Technology. He then started his career at Acu-Rite Inc. in the 1980s where he initially served in field and customer service. Moving into a product specialist position, Parker also helped to introduce the well-known Acu-Rite MILLPWR control in 1994. In recent years, Parker has been employed by a machine tool dealership on the East Coast, representing Heidenhain and Acu-Rite brand products. Heidenhain is now pleased to bring him into the corporate family.

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

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

- ❏ Corporate Management (1
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- ❏ Marketing & Sales (4)
- ❏ Manufacturing Engineering (5)
- ❏ Purchasing (6)
- ❏ Quality Control (7)
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- ❏ Maintenance (9)
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GALACTIC HOUSEKEEPING

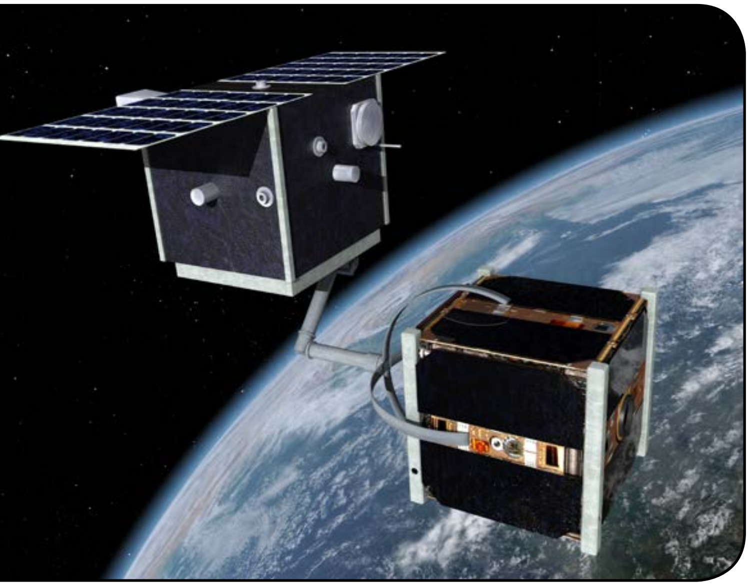
Swiss Space Systems Project Aims to Clean Up Earth's Orbit

With the constant push for sustainable and environmentally friendly procedures on Earth, you'd think we'd apply the same rules high above it. Space junk, also known as "space debris" or "space waste," is the collection of old satellites and multi-stage rocket parts that are floating around Earth's orbit like an intergalactic scrap yard. Our planet's various space programs have launched more than 7,000 crafts into the geosynchronous and low-earth orbits through the years. It's starting to get extremely crowded up there.

On October 21, 2013, a European Space Agency satellite reentered the earth's atmosphere when it ran out of fuel. While most of the 1,110 kg satellite disintegrated, an estimated 25 percent of the wreckage reached the Earth's surface. "The one-tonne Gravity Field and Steady-State Ocean Circulation Explorer (GOCE) satellite is only a small fraction of the 100 – 150 tonnes of man-made space objects that reenter Earth's atmosphere annually," said Heiner Klinkrad, head of ESA's Space Debris Office. "In the 56 years of space flight, some 15,000 tonnes of man-made space objects have reentered the atmosphere without causing a single human injury to date."

With 122+ new space crafts scheduled to launch each year, one has to wonder if those odds will get increasingly worse in the not so distant future.

Swiss Space Systems (S3), headquartered in Payerne, Switzerland, is involved in the CleanSpace One mission to remove the thousands of bits of jettisoned rocket and satellite components orbiting the Earth. The plan is to grab hold of a piece of space junk (for example an out of commission Swiss nanosatellite) and thrust it into the atmosphere, where it will burn up. The engineering know-how necessary to navigate and seize this man-made junk is amazing.



The project will develop a new launch method where a small shuttle rides piggyback atop an A300 jetliner. When the plane reaches cruising altitude, this Suborbital Reusable Shuttle (SOAR) lights its engines and takes off upwards. When it reaches an altitude of 80 km, it ejects a vessel, which after reaching an altitude of 700 km, releases the satellite into Earth's orbit. Both the Airbus and the shuttle are reusable and use standard fuels, making the system very cost effective.

The goal of this three-phase process is to make space more accessible and to make sure that this doesn't end up putting even more space debris in Earth's orbit. S3 will ensure that all the elements in the chain, including the satellites, are equipped with their own reentry systems. CleanSpace One is set to launch in 2018 and will be the first satellite launched into orbit using this method. Although the satellite design is slightly different than originally planned (it's bigger and will weigh about 30 kg), the rest of the design is on track. Scientists have tested many technologies that could potentially be integrated into the satellite in the future.

Finally, as part of a partnership with the European Space Agency, researchers are developing many key technologies targeting space debris propulsion, navigation and reconnaissance systems and, above all, a device that can anchor itself to pieces of debris. ETH Zurich and the Swiss Universities of Applied Science are participating in this as well. They are counting on integrating their developments into the CleanSpace One project.

"You can't democratize space access without having a responsible attitude," says Pascal Jaussi, CEO of Swiss Space Systems. "If we don't deal with the problem of orbiting space debris and its accumulation, future generations' access to space will be compromised."

For more information on the CleanSpace One project, visit *www.s-3.ch.* **PTE**

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