June 2007

GEAR TECHNOLOGY

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The Journal of Gear Manufacturing

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Feature
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Hansen Transmissions' state-of-the-art testing equipment.

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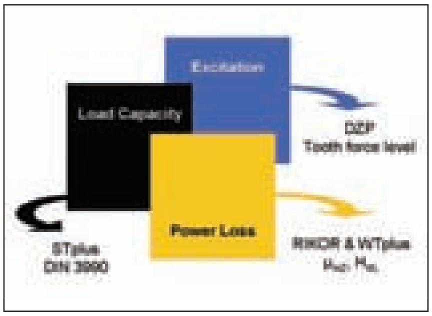

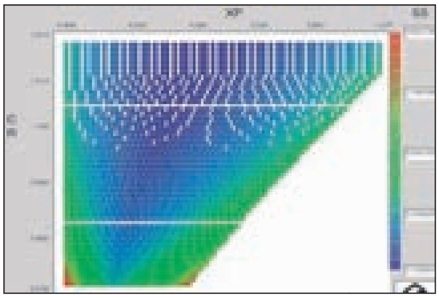

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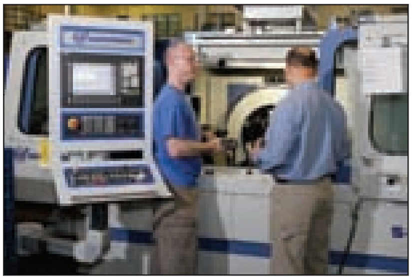

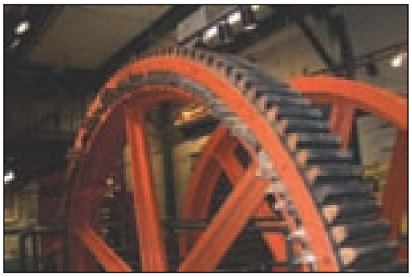

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EXTRA! EXTRA!

Read All About It!

JUNE ISSUE

Like most of the gear industry, we're extremely busy here at *Gear Technology.*

While many of you are working hard to produce more gears, we're doing the same with magazines. While you're adding shifts and hiring employees (or trying to, anyway), we're cranking out extra issues of *Gear Technology*.

What do I mean, extra issues?

Those of you who have been reading *Gear Technology* for the last 23 years know that we've always published on a bi-monthly schedule. Until now.

This year, we decided to publish the magazine eight times instead of six. The first "extra" issue is the one in your hands (June), and there will be another one coming up right before Gear Expo (August). These extra issues allow us the opportunity to include even more important gear-related information than ever before.

Expanding the number of issues has allowed us to bring you features like *Voices,* a column where smart people from the gear industry share their experiences and expertise. It's also where you—the readers—have the opportunity to respond to what you've read in previous issues. Expanding the amount of space we have available also allows us to take on feature subjects like last issue's profile of the work done at the Glenn Research Center and this issue's coverage of lean manufacturing.

Because of the extra issues, we're able to add this content without skimping on anything else. We're still bringing you the best, most relevant and most useful technical and educational articles on gear manufacturing. We're still bringing you the most complete product news, industry news and events coverage in the industry. And we wouldn't think of eliminating the Addendum column (come on—admit it—I know you read it).

Like many of you who have increased production over the past year, we're concerned about maintaining the level of quality our customers have come to expect. Gears made on the third shift have to be just as good as the ones made on the first shift. You can rest assured that we're not skimping on anything here. This issue of *Gear Technology* has been prepared the same way as every other one has been. The technical articles have been subjected to the same level of thorough editorial and technical review to ensure their accuracy and relevance, and the feature articles have been researched to the best of our ability.

I'm confident that you'll be pleased with the results. In fact, I would guess that most of you wouldn't have noticed anything different about this issue if I hadn't called your attention to it.

And that's the way it should be. Hopefully, your customers feel the same way about the gears you make on the third shift.

Michael Goldstein

Michael Goldstein,
Publisher & Editor-in-Chief

P.S. In addition to the extra effort we've put forth in producing two more issues of *Gear Technology*, we've also introduced our new magazine, *Power Transmission Engineering*. If you want to know more about bearings, motors, clutches, couplings, linear motion and other PT products, sign up for a free subscription at *www.powertransmission.com*.

Pushing the Envelope with Plastic

Steve Druley, Marketing Manager, UFE Inc.

We were delighted to see the plastic gear set pictured on the cover of your March/April issue. UFE played the lead role in its design and manufacture. It is indeed a geartrain worth showcasing, and even more so when the largest gears—the 18" internal bevel ring gears—are added to complete the system. This system was integral to the design of a new clothes washing machine bringing many attractive features to the market.

The system was conceived from the start with plastic gears because of the advantages of plastic in the application—especially cost, noise, and weight. UFE was awarded the project, from gear design analysis to mold manufacture and part molding for all the molded parts pictured.

So, of course, we are proud to submit for your readers a photo of the completed gear assembly, including the largest gears in the system. As your articles in the March/April issue discussed, plastic gear design and production requires experience and resourcefulness to be delivered within the allowable time and budget. While plastic gears have long been used in products of all kinds, and their employment is growing, the need for combining the art and science of injection molding increases exponentially as the size of the gear grows. This gear set is an excellent case-in-point of plastic gearing potential and challenges.

The large bevel ring gears presented challenges in gear design, mold design

continued

and moldability. From a gear design standpoint, little has been published regarding the design of an internal bevel ring gear (whether made from metal or plastic). As a result, the desired mesh and tooth profile required gear design resourcefulness, including collaboration with a gear cutting machine manufacturer—The Gleason Works—who ultimately provided the basic tooth form unique to internal bevel gears. This led UFE to the development of an optimized gear tooth form for the application. The next challenge was the design and development of the mold.

The material used in all the gears was a glass-filled resin specified for its strength, resistance to water absorption and compatibility with chemicals. In the final analysis, the most notable production challenges were borne out of the molding properties of the glass-filled resins. Specifically, the uniform alignment of the fibers in glass-filled resins results in non-uniform shrinkage in the molded gears, potentially producing several gear defects. This required UFE to redevelop the original tooth profiles for each gear. But that's not all. Each tooth in each gear then required modification from the original design. In addition, to maintain roundness and optimum part strength, several gating schemes in the mold design were evaluated with Moldflow analysis. Each gear profile in the gear train was revised, and the gating and runner system were optimized in mold design specifically for the resin.

Gears present unique challenges for injection molders. Thanks to many ongoing advances on several fronts in the injection molding industry, the increased use of plastic gears will continue. Every day, reliable plastic gears are shipped and assembled into a widening range of products—from washers and dryers to gas engines and digital printers—and there's no end in sight.

—*Steve Druley,*
Marketing Manager UFE Inc.

Why Do Customers Want to Reinvent OUR Wheel?

Ian Shearing, VP Sales, Mitsubishi Gear Technology Center, Wixom, MI

Over many years of being in the machine tool business, it has been interesting to observe the way we suppliers are forced to quote and sell machine tools to many large companies. Often, we put ourselves through all manner of contortions to win an order, and then—if we do win it—we have to face the eventual ramifications that exist because of that success. Why is this? The answer is very simple: CUSTOMER-SPECIFIC SPECIFICATIONS.

Many large companies have their own machine tool purchasing specifications, which typically include the following:

- Quoting format
- Forms for build reporting
- Frequency of progress report meetings
- Machine configuration to be employed, i.e. horizontal, vertical, etc.
- Individual components to be used in the machine, including electric, hydraulic, lubrication and pneumatic
- Wiring methods to be used, including wire sizes and color
- Piping methods to be used, including type of piping and pipe threads
- Pipe fittings
- Operators screens (HMI)
- Type of PLC logic
- Ancillary equipment to use, such as chillers, oil mist collectors, dust collectors, chip conveyors, etc.
- Special paint specifications
- Vibration sensors and how many are to be used per spindle
- Extended machine acceptance procedures
- Machine noise acceptance levels
- Machine vibration acceptance levels
- Extended warranty demands
- Delivery penalties
- Extended payment demands

The above list is but a small sample of the specifications which must be adhered to when quoting large companies. Normally, these specifications are supplied as a book, but often they are available only via a website whereby we have to print volumes of paper to produce the complete set of specifications, which include specifications for machines

continued

other than the ones you are quoting. These specifications are often written in a language which is foreign to us as a builder and thus, open to varied interpretations.

Once a request for quotation (RFQ) has been received, a supplier is lucky to get as many as 30 days to quote. Normally, it is less. Keep in mind that quoting to these specifications often involves a complete redesign of one's machine, interfacing with unfamiliar suppliers and—because of the timeframe for quoting—a certain degree of guesswork. The end result is a quotation for machines which has some ingredients of conservatism, guesswork, assumptions and hopes that if the order is won, "it will all work out in the end." Typically, it is a recipe for disaster.

Let's review for a minute some of the specification requests and their impact on the machine supplier.

The seemingly simple quoting format. As suppliers we have all developed our own quotation formats, which present the machines and their options in a logical and understandable way. Now we are asked to quote in a totally different format with options being grouped and divided in such a way so that various supplier offerings can be, supposedly, directly compared. The end result involves many revisions until it meets the satisfaction of the customer. The cost of this extra work is rarely factored into the machine price, so it ends up being time and material that is absorbed. And so the diminishing profit margin begins.

Individual components for electric, hydraulic, etc. With the short time given to quote and the unfamiliarity with the component suppliers we have to use, it is impossible to get the same pricing levels per our normal component suppliers. In addition, there is a certain amount of trepidation in using unfamiliar components in our machines. All these unknowns will affect the reliability of our product and its reputation.

Operator screens (HMI). The basic function here is to try and make every machine's startup the same in order for operators to be able to easily switch from one machine to another. However, after the startup screens have been navigated, it is impossible to make the screens the same for a hobber, shaper, machining center and lathe, etc. Therefore why bother? We, as suppliers, have developed screens to make the use of our machines as simple as possible. It seems to us that these special HMI screens only serve to complicate and confuse.

Extended warranties. The reluctance by the machine tool industry to offer warranties longer than a year for large companies is largely based on the fact that we have to use unfamiliar components, and we have a lack of knowledge about their reliability and maintainability.

These specifications, to which we are required to adhere, are updated periodically by our customers to an extent where it is not always possible to assume, because we have supplied machines before, that subsequent machines can be built to the same specifications. In addition, every large company has its own specifications. They are not the same.

We are sure that the selection of components to be used is born of suppliers' lobbying or selling their individual wares to these large companies. Naturally the logic of installing large quantities of machine tools all using the same components is not lost on us as suppliers. The advantages of stocking spare parts, training employees and familiarity with components used are obvious. However, as suppliers we believe that these advantages never seem to be fully exploited, as requests for us to supply these customer-specific parts and to diagnose their faults are numerous.

The consequences, in many cases, are projects which result in much higher purchase prices for the customer; delivery delays through having to deal with unfamiliar component suppliers; redesign of standard products with new, supporting and sometimes inaccurate documentation; probably lower machine reliability; and lower, if any, profit margins. Sometimes we are convinced that large customers do not want to purchase our machines but instead want a supplier to build their machines.

Is this article a complaint? I hope it is perceived more along the lines of presenting facts with the hope that changes for the better will be forthcoming, thereby avoiding the inevitable dissatisfaction on both sides.

Having made the aforementioned observations, it must be stated that there are large companies who do purchase our standard product. The reliability afforded them in this endeavor even encourages them to purchase the machines without any warranty, thereby lowering the purchase price. The companies with this vision employ our service at pre-negotiated rates with commitments to provide service and maintain the equipment at regular intervals. These companies also report back data regarding reliability and maintainability which is meaningful to us because it is what we manufacture and assemble on a daily basis. The advantages are constant product improvements which can be passed on to all customers, not to just a few.

Will the expansion and ever-increasing complication of these specifications ever cease? Probably not in the short term. There is, however, a glimmer of hope which is surfacing because of the unfortunate economic plight of some of these large companies. The desire to purchase capital goods at reduced prices and maintain a healthy supplier base is causing large companies to look again at the way they do business. Does hope spring eternal? As suppliers, we like to think so, because it is our desire to offer products in the full belief that they are the best that we can produce, with documentation which supports that fact.

–Ian Shearing, VP Sales
Mitsubishi Gear Center

THIS IS ABOUT as Lean as it Gets

When thinking of lean and time-saving manufacturing practices, the traditional brick-and-mortar factory is what typically comes to mind. But what if you could design and install a completely automated, high-speed, high-precision micro-assembly facility on a space no larger than a small table-top? Even Henry Ford would sit up and take notice.

That reality is here in the form of CSEM's (Centre Suisse d'Electronique et de Microtechnique SA) robotized microManufacturing system. Intended for an array of microtechnology assembly applications, the 24/7 CSEM "factory" and its "workforce" of PocketDelta robots—the world's smallest according to Swiss-based CSEM—deftly handles multi-step assembly of very small components at speeds of up to 3 cycles per second, with a precision of 2 micrometers. The company is now marketing its system wherever small devices, micromotors, microsystems, watches and other miniature products are manufactured and assembled.

So, how does it work? It's largely about parallel kinematics. Philipp Glocker, CSEM's section head for microassembly and robotics, explains.

"The basic idea behind the Delta parallel robot design is the use of parallelograms, he says. "A parallelogram allows an output link to remain at a fixed orientation with respect to an input link. The use of three such parallelograms restrains completely the orientation of the mobile platform, which remains only with three purely translational degrees of freedom."

In other words, the PocketDelta robots execute a series of pick-and-place parallel cycles in separate modules—each approximately 1 dm^3 in size—to constitute a tabletop factory. The factory is also equipped with a transfer system that advances the product from one robot to another, a feeder system for additional components, and specific tooling for each production step. In addition, the mechanics, motion drives, control electronics and computing are completely integrated into one compact unit. The factory can be configured as a standalone facility, performing a variety of pre-programmed assembly tasks, or in a more sophisticated production line where several robots can be directed by a master station.

Markets and applications served by the tabletop factory include watches, micromotors, and microsensors, as well as MOEMS systems. The factory can also be fitted with a controlled micro-environment, typical Class 100, thus positioning it for use in MEMS and medical device production.

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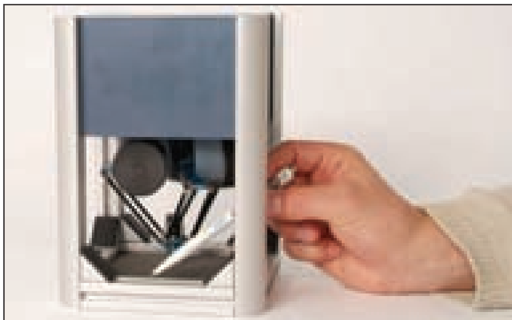

Zeroshift's New Transmission

SAVES 7% FUEL ECONOMY

Zeroshift introduced a second-generation, automatic gearbox technology that uses a new drive ring system.

According to the company's press release, the technology provides a fully automatic transmission that changes ratio with zero torque interruption to facilitate operation as a traditional planetary automatic. Possible applications include passenger cars, commercial vehicles, off-highway vehicles and motorcycles.

Bill Martin, managing director at Zeroshift, says, "Further design innovations, plus our experience with prototype vehicles, CAE analysis and rig testing, has allowed us to create a highly efficient design that is ready for integration with vehicle development programs."

Zeroshift technology provides a fully automatic transmission using the majority of components from any proven manual transmission. According to the company's press release, unlike conventional manual transmissions, the new system changes gear without any torque interruption. In a typical

continued

mixed-drive cycle, the system should also generate a fuel economy saving of approximately 2% compared with a manual transmission and 7% compared with a six-speed planetary automatic.

Zero torque interruption allows either improved acceleration or engine downsizing.

Options include building the shift mechanism into an existing manual gearbox so it can be assembled on the related manual transmission product lines.

"This makes it mechanically so simple that only a few workstations will have to be modified to accommodate a choice of components," explains Martin.

The technology uses a pair of interlocking rings, each incorporating the three drive elements in a single forced component and operated by shift forks to replace the six drive elements used in the first-generation transmission. When shifting from neutral, ring one is engaged to take up drive while ring two is engaged within a few degrees of revolution to take up backlash. The next shift is made with ring two taking up the drive, and ring one, the backlash. As ring two is unloaded during the change, it requires less than 1/20 the axial force required by a conventional synchromesh. The control system coordinates gearshift actuation, engine management and clutch operation to provide full control over the driveline during gear shifting.

One side of the drive element has a retention angle to occupy the drive, while the opposite side has a ramp face to smoothly disengage the drive. Shift forces have also been reduced to 40N. Therefore, all components can be manufactured from proven lightweight materials with no additional surface treatment.

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Moventas

LAUNCHES WIND TURBINE CONDITION MANAGEMENT

Moventas introduced its latest wind turbine condition management system at Hannover Messe 2007.

The system focuses on the gearbox and associated components and monitors operating parameters and changes that could indicate a possible failure mode. Key measurement parameters include oil quality, vibration, load and temperature. Continuous monitoring is handled by an electronic control system with software containing algorithms designed to assess overall performance. Information about the status of individual wind turbine gearboxes can be accessed remotely online. Each condition management system—including the data entered—is tailored to the specific requirements of the wind turbine operator.

The condition management system can be integrated into new turbine design or fitted to machines already in operation. Existing sensors may be integrated into the system that may already be fitted to a wind turbine. In all applications, Moventas offers a turnkey service that provides all the mechanical engineering, installation and connection so customers get the information about the status of each turbine's operation.

"Wind power production is growing extremely rapidly, at an annual rate of over 20%," says Jyrki Virtanen, senior vice president of Moventas. "It is driven by increased need for energy, the need to reduce carbon dioxide emissions and also wind power's improved competitiveness resulting from techno-

continued

logical advances."

According to the company's press release, Moventas invested 100 million euros in wind turbine gears and created approximately 100 new jobs this year.

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Vectron's Viscosity Sensor

DETECTS OIL CHANGE IN GEARBOX

The low-shear ViSmart™ viscosity sensor from Vectron International was recently used to collect real-time viscosity data in an oil-aging detection application. The testing, conducted by a major manufacturer of fluid management systems for industrial applications to determine repeatability and reproducibility, was carried out across the entire temperature spectrum ranging from 0–105°C for a gearbox-specified commercial oil and demonstrated that the ViSmart™ viscosity sensor detected change in oil condition as a function of viscosity to an accuracy and repeatability value of 1%. Vectron is working with the manufacturer to integrate the ViSmart™ sensor into its fluid management systems.

The low-shear ViSmart™ viscosity sensor uses robust and reliable semiconductor technology, has no moving parts, and is sealed for complete immersion. It is not affected by vibration or flow conditions, does not need field calibration and is packaged at 0.5" x 3" and 8 oz. The ViSmart™ low-shear sensor measures viscosity of fluid from 1 to 500 centipoise and is rated for temperatures up to 125°C in the standard package. Custom options and configurations are available for specific industry applications and process requirements.

When integrated with Vectron's eCup handheld viscosity sensor reader, the low-shear ViSmart model series can be connected to any computer platform via the standard USB port to provide a continuous audit trail for process monitoring markets.

Additionally, leveraging Vectron's PC ViscNet™ bench-top viscometer data station, the customer can acquire multiple channels of continuous viscosity, temperature and/or other physical data from up to four ViSmart™ or other physical sensors.

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ZF's Right-Angled Gearheads

OPEN DOOR FOR LUXURY MULTIMEDIA

The SWG gearheads from ZF offer a smoother alternative to screwjack-based door control systems used in the latest luxury media experience—the Oculas 2.0™

Designed by Lee McCormack and engineered by McLaren Applied Technologies, the Oculas 2.0™ is a multimedia product that provides users with a completely immersive environment. Uses include entertainment purposes or business applications.

continued

ZF supplied SWG gearheads to Mitsubishi Electric UK for the door operating system because of the choice of backlash options, which offered additional safety with anti-backdrive operation. The futuristic pod provides advanced immersive branding, conferencing, purchasing, downloading, gaming, music, film, simulation and general entertainment experiences.

A single upward-swinging, gull-wing door is one of the main features of the Oculas 2.0™, and its smooth operation is essential, explains the designer. The McLaren engineering team faced initial problems with the screwjack-based door control system. "Opening and closing operations were functional, but when the 16-kilo door was lowered, the screwjacks caused a juddering motion that wasn't ideal, so Mitsubishi Electric UK was asked to specify a more effective solution," says McCormack.

The ideal door operating system would be based on a lightweight, high-ratio gearbox of the kind used in precision automation and packaging systems. However, these gearheads have traditionally been prohibitively costly for use in even luxury consumer applications.

Mitsubishi Electric then approached ZF to specify the SWG range of gearheads, which offers machine builders the choice of backlash levels of more than 10, 3 and 1 arcmin, explains Dave Morgan, technical sales engineer at ZF Great Britain. "The SWG90 right-angled gearheads were specified with levels of backlash to more than 10 arcmin, offering many of the operational benefits of a higher-precision gearhead at approximately one-third of the cost."

SWG80 units have a 90:1 ratio, thereby providing a smooth door motion. Their precision high gearing eliminates backdrive, ensuring that—in the event of power or system failure—the door does not drop down, trapping or injuring the user. The gearhead design performs under high radial loads, a feature which has made the units suitable for handling systems and machine designers and, ultimately, insures the immediate stopping of the door, despite the high moment of inertia caused by the offset door pivot.

SWG gearheads are designed for maximum mounting flexibility. The standard flanged model attaches directly to the Mitsubishi electric motor. The hollow shaft and shrink disc option also allow easy fitment to the output, which in this case was the Oculas 2.0™ door spindle. In addition, sealed-for-life lubrication enables the gearboxes to be mounted at 45° in the main body of the pod.

According to ZF's press release, the lightweight aluminum/magnesium body of the gearhead helps reduce the final weight of the Oculas design and a computer-optimized worm wheel contact pattern offers performance level of approximately 55 dB. The pattern achieves 90% contact during operation, distributing loads evenly and contributing towards the highly rigid performance. Due to ZF's specifying two sets of oversized taper roller bearings in the SWG design, the gearwheel and spindle can also transmit higher loads with an improved degree of rigidity and accuracy.

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Source Engineering

INTRODUCES PLANETARY GEARMOTORS FOR OEM APPLICATIONS

Source Engineering introduced planetary brush and brushless gearmotors for OEM applications. Gearmotors are best suited to applications requiring high torque and speed reduction and are available in ratios from 3:1 up to 3,729:1.

According to the company's press release, the 22 mm and 56 mm diameter gearmotors can be supplied with 3–36 VDC motors, plastic, powder metal or cut metal gears and 2/3 channel quadrature optical encoders.

The compact, center-shaft planetary gearheads are available in 22, 24, 25, 28, 32, 36, 45 and 56 mm frame sizes with either ball or sleeve output shaft bearings. In addition, the output shafts can be custom machined with cross-holes, splines, keyways and flat or end drilled and tapped. Designed for input speeds up to 6,000 rpm, continuous, rated torque is increased to 392 kg cm and max torque is typically three times rated torque.

Planetary gearmotors can be customized to meet new and existing applications.

For more information:
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Bonfiglioli

UPDATES PARALLEL SHAFT SERIES AND BEVEL-HELICAL SERIES GEARBOXES

Bonfiglioli recently presented five new sizes in the HDP parallel-shaft series and HDO bevel-helical series that was first introduced.

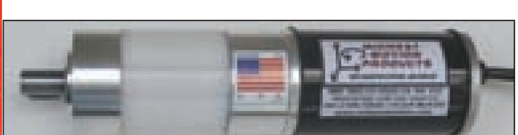
This expansion of the HDP and HDO series will permit the company to fulfill the needs of industrial applications demanding torque ratings of up to 75,000 Nm, which is commonly required in the mining, paper, wood and steel manufacturing industries, and many other processing industries.

According to the company's press release, extensive use was made of FEA (Finite Element Analysis) throughout the design process as well as MBS (multi-body simulations) to ascertain the dynamic effects of stress transmitted between the various components of the gearbox. The result is enhanced structural rigidity and gear geometry for optimized load capacity and gear and bearing life. Noise emissions and vibrations have been kept to a minimum by using structural simulations based on BEM (Boundary Element Method) analysis.

For more information:
Bonfiglioli USA
3541 Hargrave Dr.
Hebron, KY 41048
Phone: (859) 334-3333
Internet: *www.bonfiglioli.com*

Midwest Motion

INTRODUCES NEW GEARMOTOR



Midwest Motion Products announced the availability of the MMP-TM55-12V Gpk52-124 gearmotor.

The motor accepts any 12-volt DC source, including battery power. The gearmotor measures 2.14" diameter by 7.7" long and has a keyed output shaft of 12 mm diameter by 25 mm long. Mounting is accomplished with four "face mount" M5 threaded holes, equally spaced.

According to Midwest Motion's press release, the output is rated for 125 in.-lbs. of continuous torque at 37 rpm and 178 in.-lbs. peak. The gearmotor weighs less than 7 lbs. and requires 7.2 amps and 12 volts to generate its full torque load. Motor windings for 24, 36, 48 and 90 volts are available.

For more information:
Midwest Motion Products
10761 Ahern Ave. SE
Watertown, MN 55388
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E-mail: *randy@midwestmotion.com*
Internet: *www.midwestmotion.com*

Renold

INTRODUCES INTERCHANGEABLE RANGE OF GEARBOXES AND COUPLINGS

Renold will exhibit its SMXtra interchangeable range of gearboxes and fluid couplings, for quarrying applications, at the Hillhead Exhibition from June 26–28 in Buxton, Derbyshire, U.K.

The SMXtra enhanced range of shaft-mounted helical gear reducers is now available in 12 sizes, with four ratios. Hollow bore sizes up to 190 mm enable the units to be fitted in various quarrying applications.

The units are available with a standard parallel sleeve arrangement and with a single bush taper locking system, which can be fitted from either side of the unit, allowing the units to be fitted and removed in the quickest possible time.

Also displayed for the first time is the new range of Hydrastart soft-start fluid couplings, which have been redesigned to be interchangeable with competitors' products.

In addition, Renold is presenting new mid-range metric worm gear units. These are available in the four popular metric sizes (100–200mm) with ratios from 5:1 to 70:1. These units have been designed as 'drop-in' replacements of the Radicon mid-range units from David Brown.

For more information:

Holroyd Gear Works
Milnrow
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Lancashire
OL16 3LS
U.K.
Phone: +44 1706-751-000
E-mail: *sales@gears.renold.com*
Internet: *www.renold.com*

Haas

Simple Innovation #9

HMC Pallet System

A Simple, Dependable Pallet Indexing System

The pallet indexing system for EC Series machines employs a simple, robust and shop-proven design that has been perfected by more than 20 years of experience building rotary tables.

At the heart of the system is a precision worm-gear set that is manufactured to AGMA Class 13 quality using the same high-accuracy techniques used to build Haas rotary tables. This proven design ensures precise positioning and years of trouble-free operation.

Low Loss Gears

Bernd-Robert Höhn, Klaus Michaelis,
Albert Wimmer

Management Summary

In most transmission systems, one of the main power loss sources is the loaded gear mesh. High power losses lead to high energy consumption, high temperatures, early oil aging, increased failure risk and high cooling requirements. In many cases, high efficiency is not the main focus, and design criteria such as load capacity or vibration excitation predominate the gear shape design. Those design criteria mostly counteract the highest possible efficiency. In this article, the influences of gear geometry parameters on gear efficiency, load capacity, and excitation are shown. Therefore, design instructions can be derived, which lead to low-loss gears with equivalent load capacity

Introduction

Power losses occur in different components of a gearbox. Each gearbox element produces some power losses. The total power loss is the sum of the power losses of the single elements. Basic gearbox elements are bearings, gears and seals. Their power losses are usually individually mentioned. Other potentially integrated elements, such as clutches or oil pumps, also produce losses, but these are not usually treated separately. Their power losses are merged in auxiliaries. According to their types, losses can be further divided into no-load losses and load-dependent losses. Equation 1 (on page 29) shows the summation of all power losses P_V in a typical gearbox. Losses in bearings and gears usually predominate in a gearbox.

No-load losses comprise all losses that exist when a gearbox is rotating, but not transmitting power. No-load losses derive from seals or from windage and churning.

Load-dependent losses occur only in elements that carry the transmitted power or portions of it, such as bearings and gears. They encompass all power losses that vary with the power transmission in the concerned element. They evolve when two surfaces under pressure move relative to each other. Power losses in this case depend on the acting force between the solids, the sliding speed, and on the coefficient of friction established in the contact of the surfaces.

For the composition of total power losses P_V in a gearbox, the following four main components are investigated:

- No-load power losses in bearings
- Load-dependent power losses in bearings
- No-load power losses in gears
- Load-dependent power losses in gears

The investigations are based on calculations for which the FVA-software *WTplus, STplus* and *RIKOR* are used (Refs. 1, 10 and 13).

Gearing Model

For the calculations, a gearing model is necessary. The data of the gear set that was used for the calculation were taken from an existing gear set of one of our test gearboxes. Figure 1 (overleaf) shows the main data of the reference gearbox model and a transverse section of the reference gear shape on which the calculations are based. Starting from that, single gearing modifications are applied in order to investigate the influence of each single parameter.

Power Loss Portions

In Figure 2, the amount of power losses for each of the four considered components is depicted versus the rotational speed at the operating conditions given in Figure 1. The investigation of power loss composition in Figure 2 is accomplished with the example of modified reference gearing with spur gears (β = 0°). It shows that the gear no-load losses increase progressively with speed, while the other components seem to depend fairly linearly on the speed. For the vast range of rotational speeds, the main portion of losses are load-dependent gear losses. Only for very high speeds do no-load losses prevail, though the load-dependent gear losses may still occupy an important portion. Bearing losses have only subordinate portions of the total losses

Prof. Dr.-Ing. Bernd-Robert Höhn *is head of the Institute of Machine Elements and the Gear Research Centre FZG at the Technical University Munich. He worked for 10 years at Audi as manager of the departments for gear research, design and experimental investigations and became a professor in 1989.*

Dr.-Ing. Klaus Michaelis *is research group manager at the FZG. His main working areas are load carrying capacity and experimental tribology of cylindrical, bevel and worm gears.*

Dr.-Ing. Albert Wimmer *was assistant at the FZG. The subject of his PhD thesis was theoretical and experimental investigation of load dependent losses in gears. He is now working with Weigl Engineering GmbH in the field of innovation management of geared systems.*

throughout the whole speed range.

In addition, the sum of these losses is rated against the power transmission, which results in the loss degree ζ, the complement of the degree of efficiency η:

$$\zeta = \frac{P_V}{P_{in}} = 1 - \eta \qquad (2)$$

with P_V total power loss (measured in watts), P_{in} input power (measured in watts), η degree of efficiency.

The loss degree ζ shows a significant minimum between 10 and 20 m/s rotational speed. This reflects the basic changes in the coefficient of friction in the mating gears from the mixed lubrication regime for low speeds towards elasto-hydrodynamic (EHD) friction at higher speeds. Depending on the geometry of the transmission and the operating conditions, this minimum occurs at different speed ranges.

The prevailing power loss portion is very dependent on the operating conditions. However, in order to minimize the power losses, a focus must always be set onto load-dependent gear losses since their portion is always significant. With increasing speed, no-load losses of gears need to be considered increasingly.

Basics of Load-Dependent Gear Losses

The load-dependent losses depend on both gear and lubricant properties. The calculation of load-dependent power losses in gears is based on the law of friction according to Coulomb (Ref. 3).

$$F_R = \mu \cdot F_N \qquad (3)$$

$$P_{VP} = F_R \cdot v_g = \mu \cdot F_N \cdot v_g \qquad (4)$$

with F_R friction force (measured in Newtons), μ coefficient of friction, F_N normal force (measured in Newtons), P_{VP} load-dependent power loss (measured in Watts), v_g sliding speed (m/s).

Equation 4 is valid for a single point of contact. In order to receive the mean power loss of two mating gears, all points of contact along the path of contact need to be considered. The power loss is calculated by the integral of the product of sliding speed, coefficient of friction and load over the path of contact.

$$P_{VZP} = \frac{1}{p_{et}} \int_A^E P_{VZP}(x)\, dx \qquad (5)$$

with p_{et} transverse base pitch (measured in mm), $\overline{AE}$ path of contact (measured in mm).

All three parameters (coefficient of friction, normal load, sliding speed) vary along the path of contact (Fig. 3).

Sliding speed is a geometry parameter that is derived from the gear shape and can be calculated exactly.

The load distribution along the path of contact can be approximately set to

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$$P_V = \overbrace{P_{VZ0} + P_{VZP}}^{\text{Z: gears}} + \overbrace{P_{VL0} + P_{VLP}}^{\text{L: bearings}} + \overbrace{P_{VD}}^{\text{D: seals}} + \overbrace{P_{VX}}^{\text{X: auxiliaries}}$$

0: no-load
P: load dependent

Equation 1.

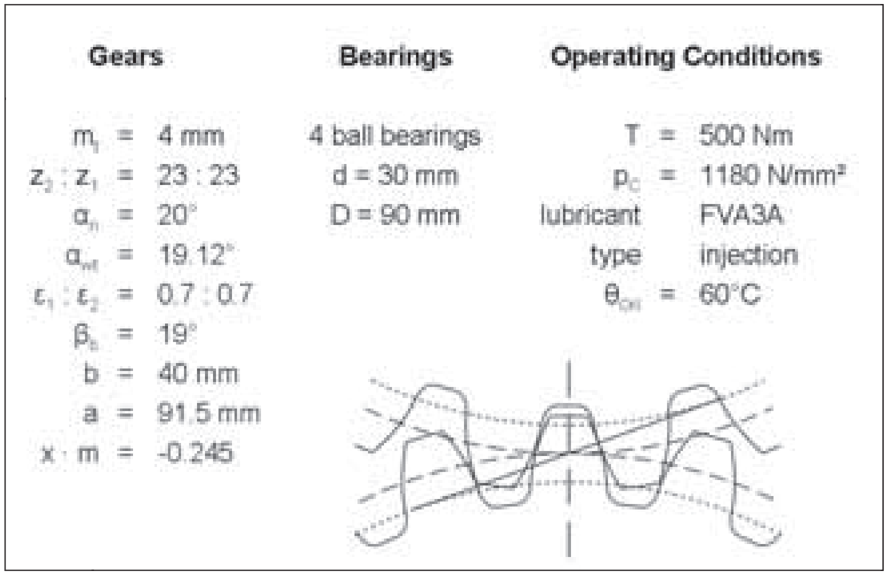

Figure 1—Main data of reference gearing model and gear cross section.

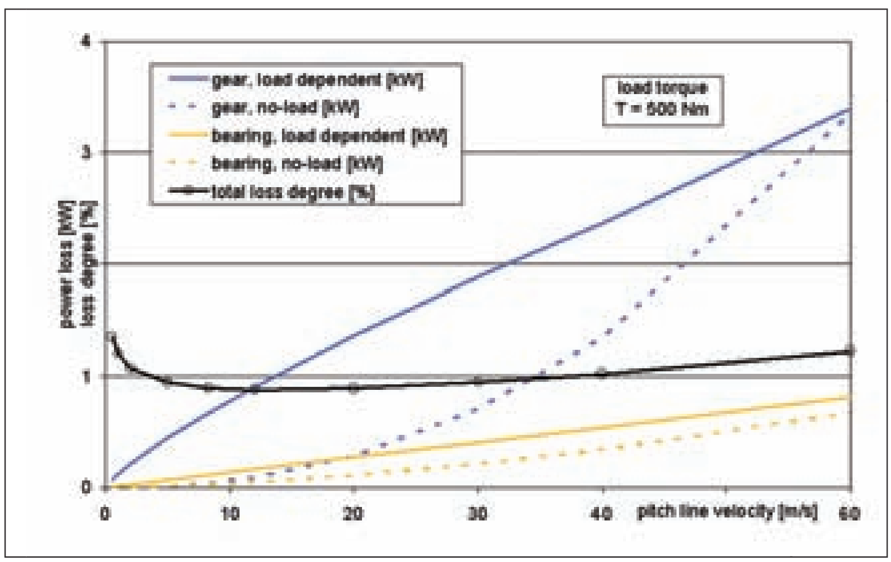

Figure 2—Power loss composition in the model gearbox vs. pitch line velocity.

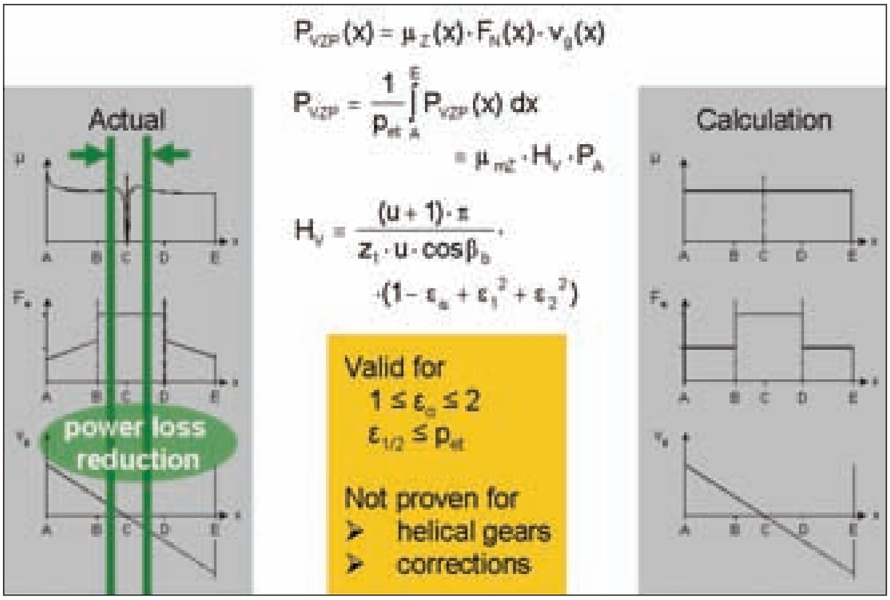

Figure 3—Tribological conditions along the path of contact.

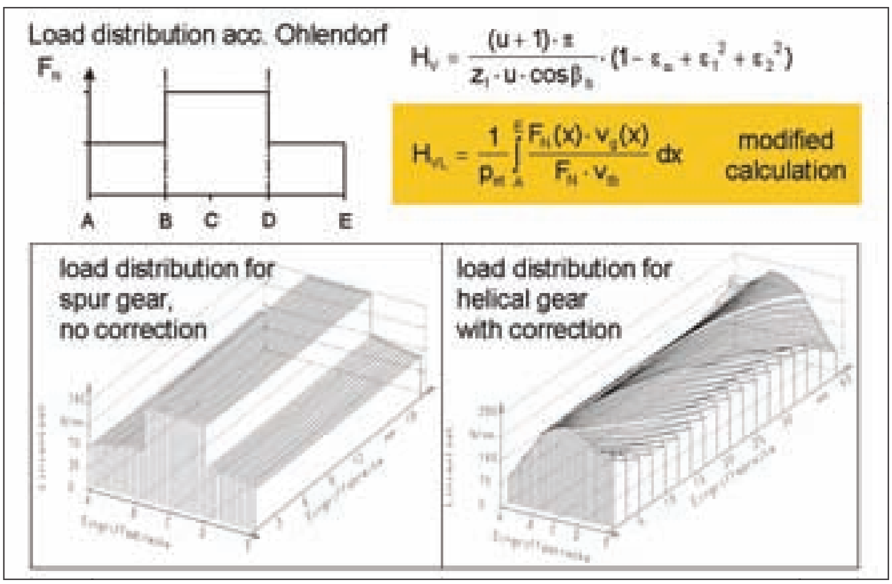

Figure 4—Gear loss factors H_V and H_{VL}.

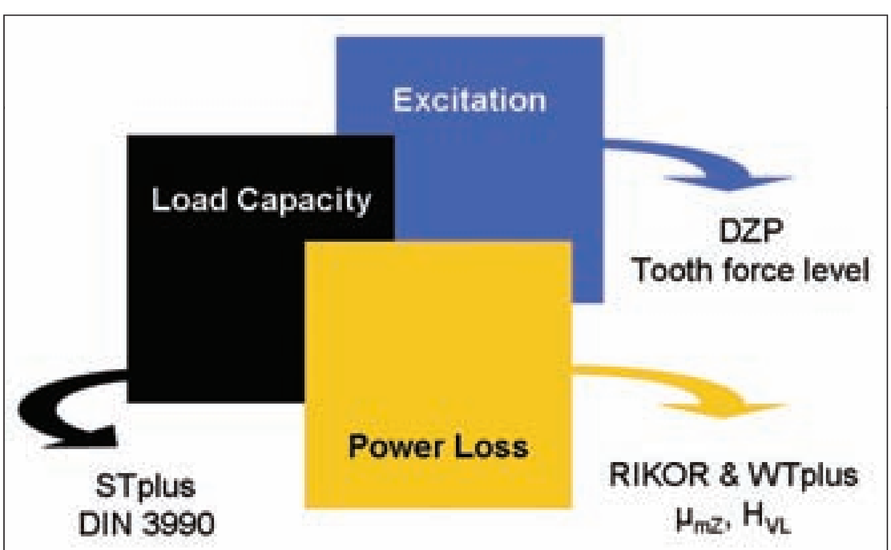

Figure 5—Calculation methods for different parameters.

the total load resulting from the torque and split up into the number of pairs of teeth in contact. This assumption is a simplistic approximation (Ref. 9). The coefficient of friction over the path of contact is assumed to be approximately constant. At the pitch point C, where sliding is zero and pure rolling occurs, the instantaneous drop of the coefficient of friction to zero has to be considered. This deviation of the coefficient of friction takes place where the sliding speed is zero. Hence, in the integral, this deviation is negligible. The coefficient of friction is approximated according to the FVA project No. 166, done by Schlenk (Ref. 11), with the following equation:

$$\mu_{mZ} = 0.048 \cdot \left(\frac{F_{tb} / b}{v_{\Sigma C} \cdot \rho_{redC}} \right)^{0.2} \cdot \eta_{oil}^{-0.05} \cdot Ra^{0.25} \cdot X_L \quad (6)$$

with μ_{mZ} mean coefficient of friction, F_{tb} circumferential force at base circle (measured in newtons), $v_{\Sigma C}$ sum speed at operating pitch circle (m/s), ρ_{redC} reduced radius of curvature at pitch point (mm), η_{oil} dynamic oil viscosity at oil temperature (mPas), Ra arithmetic mean roughness (µm), X_L factor for oil type.

With the introduced simplifications (constant coefficient of friction along the path of contact, equal load distribution onto mating pairs of teeth), Equation 4 can be applied to gears and transformed into Equation 7:

$$P_{VZP} = \mu_{mZ} \cdot H_V \cdot P_A \quad (7)$$

with H_V gear loss factor:

See page 31 for Equation (8)

with $u = z_2/z_1$ gear ratio, z number of teeth (1 pinion, 2 wheel gear), β_b base helix angle (°),

$\varepsilon_\alpha = \dfrac{\overline{AE}}{p_{et}}$ transverse contact ratio,

$\varepsilon_1 = \dfrac{\overline{CE}}{p_{et}}$ addendum contact ratio of pinion,

$\varepsilon_2 = \dfrac{\overline{AC}}{p_{et}}$ addendum contact ratio of wheel gear.

The gear loss factor H_V was introduced by Ohlendorf (Ref. 9) and is only dependent on gear geometry.

These equations were set up for usual spur gear geometries ($1 \le \varepsilon_\alpha \le 2$ and $\varepsilon_{1/2} \le \rho_{et}$) and produce acceptable results in these cases. Extreme gear shapes, however, may result in calculated power losses, which deviate significantly from actual power losses. By more detailed considerations, a better approximation of the real distribution of the load along the path of contact can be obtained by using sophisticated calculation methods such as FEM or the FVA-program *RIKOR*. This is proven by experimental investigations (Ref. 14). Gear loss factors based on such methods are called local gear loss factors H_{VL}. Differences between H_V and H_{VL} are significant for high-contact-ratio gears, helical gears or gears with profile corrections (Fig. 4).

The gear loss factors H_V or H_{VL}, respectively, comprise the integral of the product of the sliding speed and the load distribution (Refs. 9, 14). Here—for the calculation of load-dependent power losses of gears—the local gear loss factor H_{VL} with the more realistic load distribution according to the FVA-program *RIKOR* is used (Ref. 10).

For gear design, power losses are often of subordinate interest, compared to load capacity and excitation level. So, if gears are to be optimized in terms of efficiency, load capacity and excitation must not be neglected. To evaluate single gear geometry parameters and their influence on power loss, load capacity and excitation are investigated by the means of FVA-programs according to Figure 5. Excitation is evaluated by the tooth force level, which represents the dynamic load in the tooth contact without respect to the further environment. This load dynamics is the cause of, but not equal to, real load dynamics, vibration and noise.

Lubricant properties affect the power losses via the coefficient of friction μ and are not subject to this investigation. Their influence is supposed to be constant here.

$$H_V = \frac{(u+1)\cdot\pi}{z_1 \cdot u \cdot \cos(\beta_b)} \cdot (1 - \varepsilon_\alpha + \varepsilon_1^{\,2} + \varepsilon_2^{\,2})$$

Equation 8.

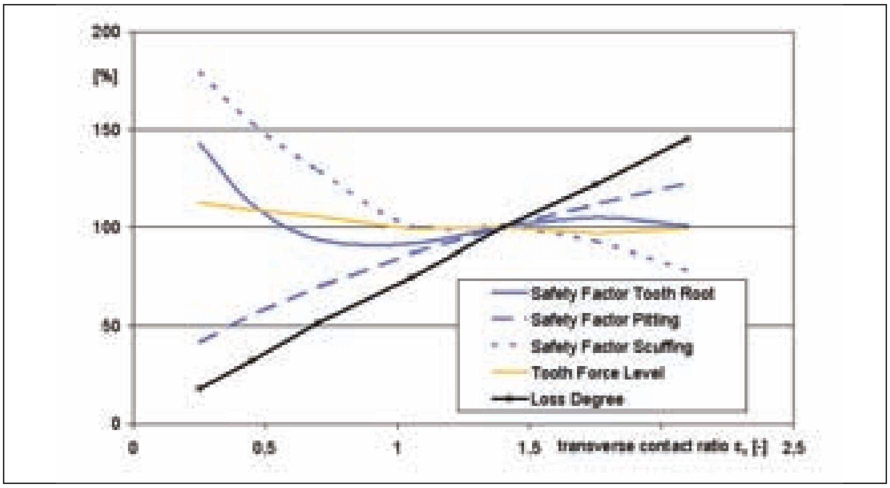

Figure 6—Influence of transverse contact ratio on power loss and load capacities.

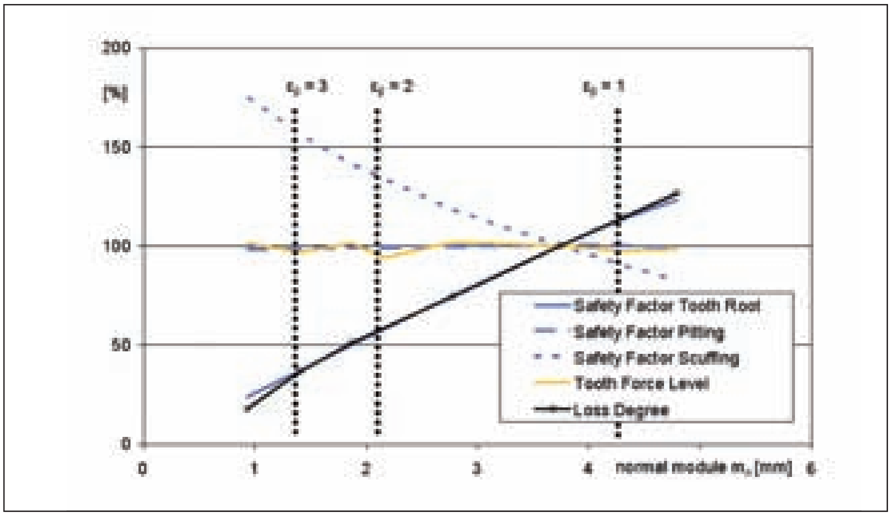

Figure 7—Influence of module on power loss and load capacities.

Influence of Gearing Geometry on Load-Dependent Power Losses

Figures 6–13 show the influences of gear geometry parameters on the gear load-dependent power losses compared to the reference gears given in Figure 1. The influence of these parameters on the coefficient of friction is included. For these parameter variations, the pitting and tooth fracture capacities are provided referring to the reference gearing with capacities of 100%. Ideally, power loss and tooth force level are low while the safety factors of load capacities are high.

The most important geometric parameters are transverse contact ratio (Fig. 6) and module size (Fig. 7). Less strong is the influence of the pressure angle, but its importance comes from the advantage that, in the given range, a higher pressure angle has only advantageous effects both on power loss reduction and higher load capacities, and no unfavorable effects on excitation (Fig. 8).

With helix angle, power losses increase generally but to a limited extent (Fig. 9). For minimum power losses with small transverse contact ratio, there has to be a significant over-

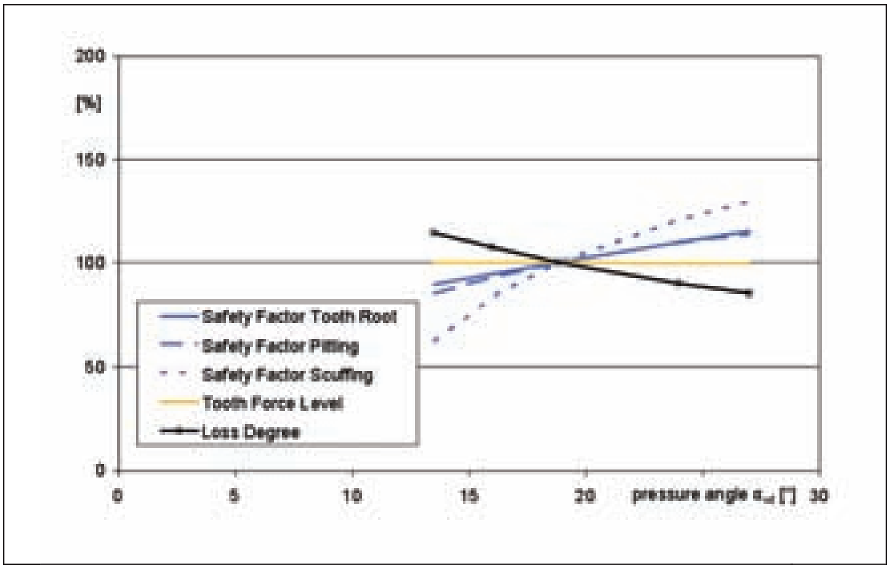

Figure 8—Influence of pressure angle on power loss and load capacities.

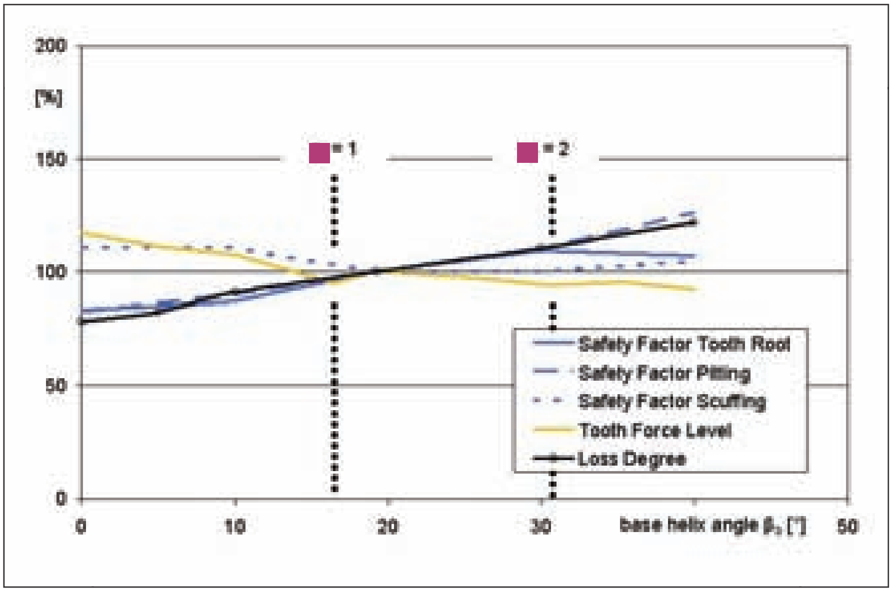

Figure 9—Influence of helix angle on power loss and load capacities.

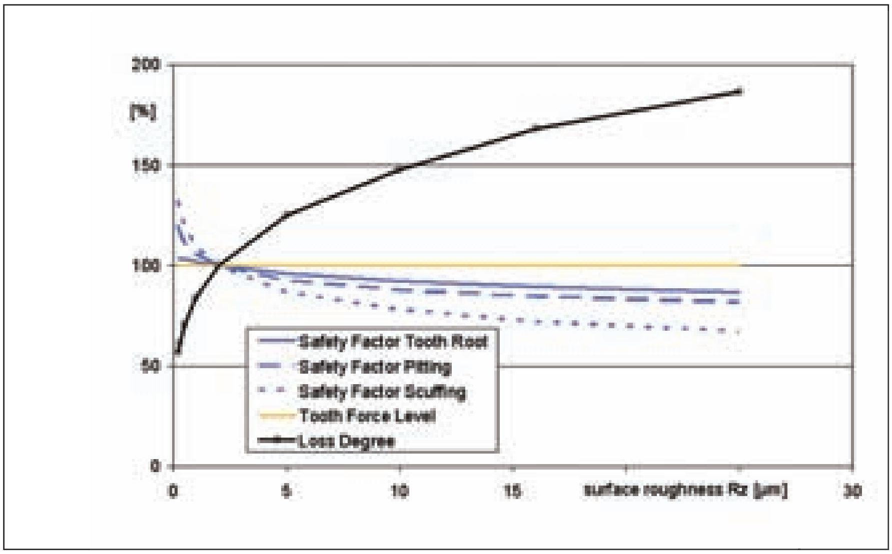

Figure 10—Influence of surface roughness on power loss and load capacities.

lap contact ratio ($\varepsilon_\beta > 1$) for proper load capacity and noise excitation.

The influence of surface roughness shows positive effects if it is reduced (Fig. 10). Unfortunately, an improvement is usually subject to cost increase. Recent investigations show that this effect is limited. Below a certain roughness, there is no further improvement. Moreover, there are other effects of surface structure such as roughness orientation, which are not expressed by surface roughness but can affect the power loss to a substantial extent.

The gear ratio and face width parameters (at constant load per face width) are usually constraints that cannot be changed. Their effect on power loss, capacities, and excitation is shown in Figure 11 and Figure 13, respectively.

Addendum transverse contact ratio is best if equally split between both gears, but small deviations have marginal impact (Fig. 12).

Gearing Optimization Process

The following steps conclude the optimization process towards low-loss gears. Most of these steps are considered unconventional, but prove more efficient to the gear design with respect to power savings. However, it is not a unidirectional process, but rather a loop that has to be run through cyclically.

- Corrections for load reduction in areas of contacts with high sliding speed
- Reduction of module down to tooth root fracture limit
- Reduction of transverse contact ratio down to pitting capacity limit
- Tooth root fillet radius as large as possible
- Increase of pressure angle
- Increase of face width
- Helix angle for adequate overlap contact ratios

Power Loss Reduction with Optimized Gearings

From the reference gearing given in Figure 1, an optimized gearing is derived. It has the same or better load capacities, but lower power losses. Figure 14 compares the absolute figures of power loss components at one operating point (v_t = 10 m/s ≈ n = 2,100 U/min; load torque 500 Nm). In this example, the bearing type has been optimized, but is not relevant here. The main changes applied encompass the following:

Bearings

- Different type (ball/taper roller)
- Size (d_m = 60 mm/43.5 mm)

Gears

- Module reduction (4 mm/2 mm)
- Transverse pressure angle increase (19.1°/41.5°)
- Transverse contact ratio reduction (1.4/0.6)
- Face width increase (40 mm/80 mm)
- Overlap contact ratio increase (1.18/4.73)

For the optimized gears, the module and the transverse contact ratio are radically cut back, compensated by a doubled face width. So, the optimized gears have a low transverse contact ratio, but a high overlap ratio for the same load-carrying capacity. Furthermore, the tooth root fillet has a larger radius in order to support the tooth root fracture capacity, and the transverse pressure angle is significantly increased for a larger radius of curvature, which backs up the pitting capacity and results in a lower coefficient of friction. Total power losses can be reduced by two-thirds in total (68.8%), where the largest portion of loss reduction is achieved for the load-dependent gear losses (see Fig. 14). The reduction of load-dependent gear losses is caused by two advantageous effects: a lower coefficient of friction because of better curvature (μ_m = 0.030; –32%) and lower gear loss factor (H_{VL} = 0.045; –78%). This combination results in a reduction of load-dependent gear losses of [1– (1–0.32)*(1–0.78)] = 85%. In Figure 14, the transverse section of both the reference and the optimized gears is included.

A similar optimization procedure is applied to Type C gears, which is a frequently used test gear geometry for many kinds of gear and lubrication tests. In Figure 15, the main geometry parameters of Type C gears are given,

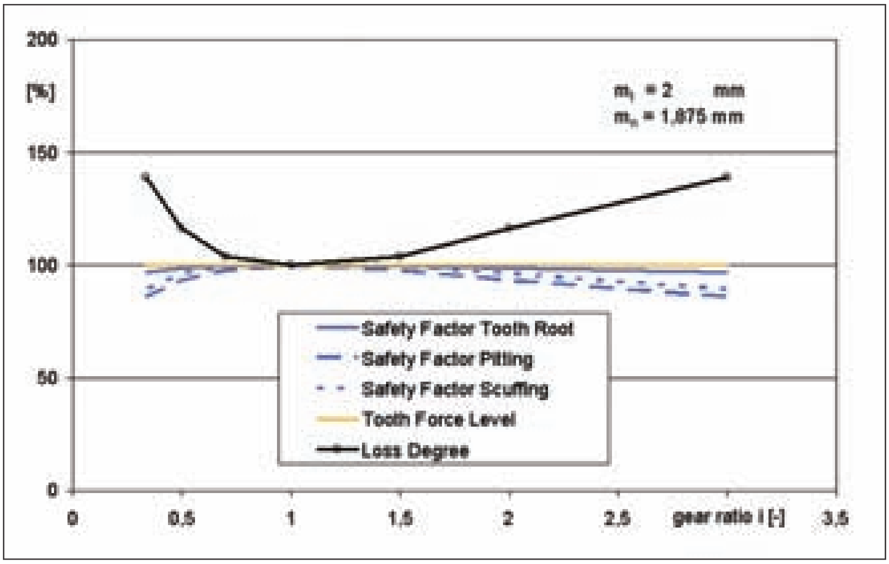

Figure 11—Influence of gear ratio on power loss and load capacities.

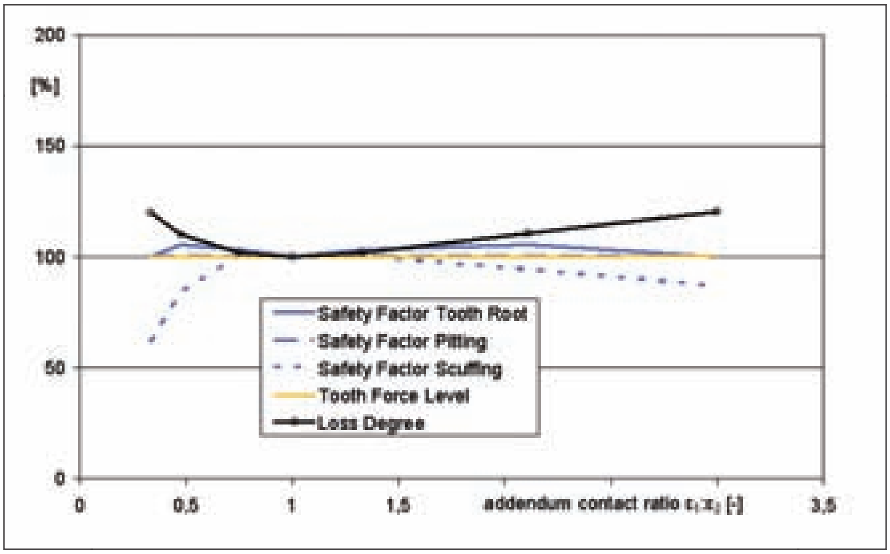

Figure 12—Influence of addendum contact ratio on power loss and load capacities.

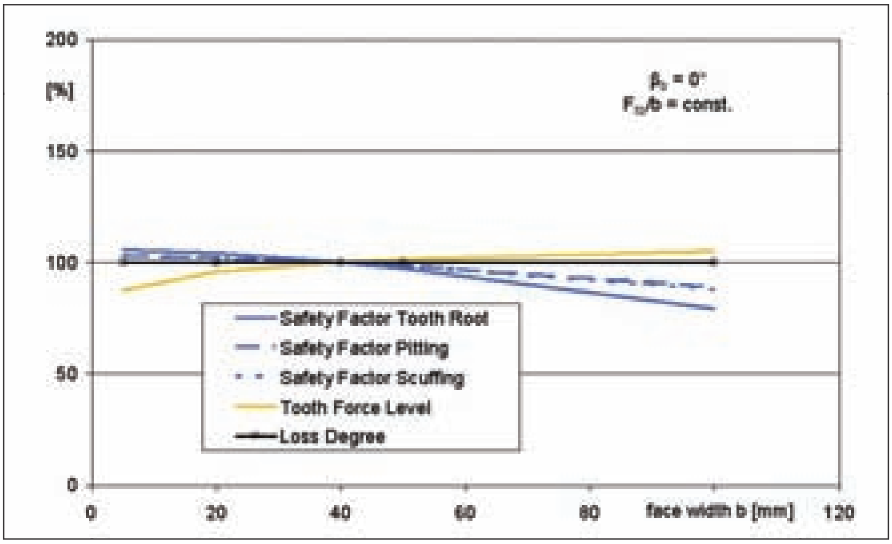

Figure 13—Influence of face width on power loss and load capacities.

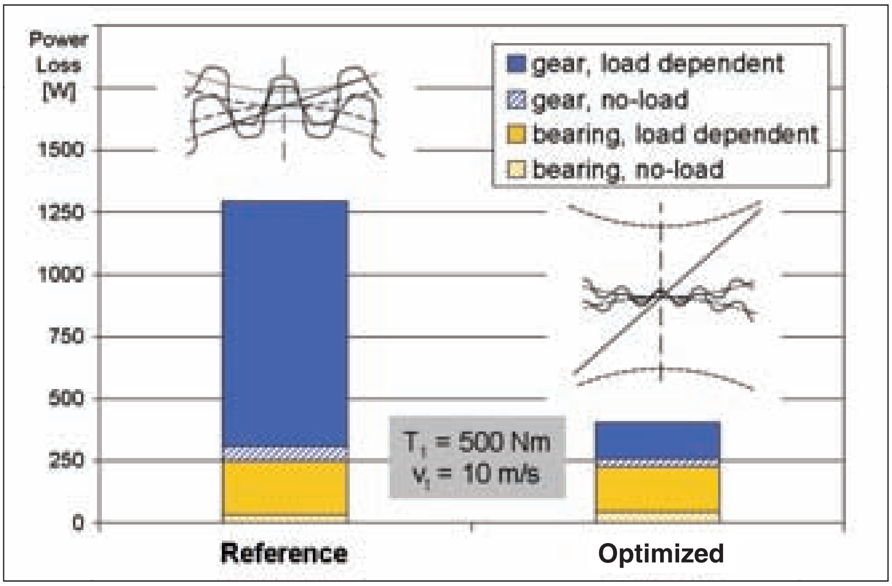

Figure 14—Comparison of power loss composition of reference and optimized gearing.

	Type C	Low Loss
m_n	4.5 mm	1.75 mm
$z_2 : z_1$	16 : 24	40 : 60
α_n	20°	40°
α_{wt}	22.4°	41.6°
$\varepsilon_1 : \varepsilon_2$	0.72 : 0.71	0.29 : 0.26
β_b	0°	15°
b	14 mm	20 mm
ρ_{a0}^*	0.25	0.5
a	91.5 mm	91.5 mm
Ra	0.17 µm	0.38 µm
S_F	0.81 / 0.87	0.89 / 0.89
S_H	1.79 / 1.86	1.86 / 1.82
S_B	1.14	11.72

DIN 3990
KS9
v_t = 8.3 m/s
FVA3A

Figure 15—Gear geometry of Type C gears and corresponding low-loss gears.

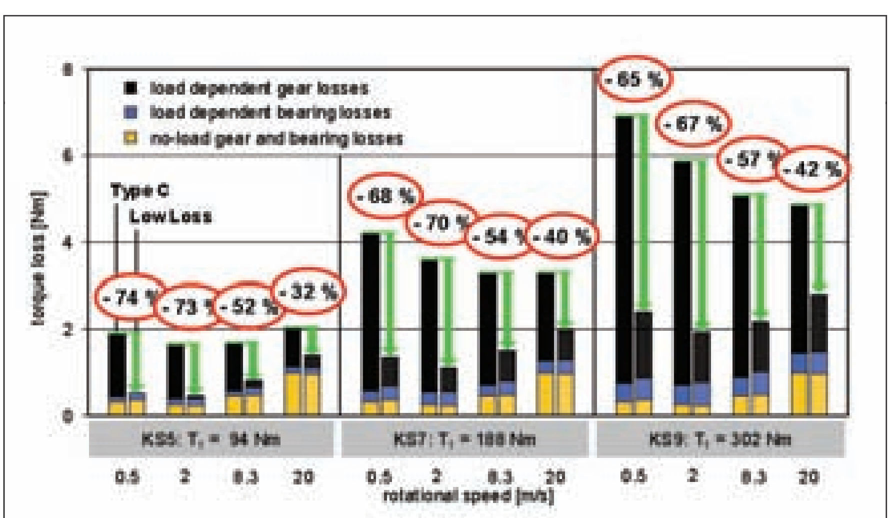

Figure 16—Experimental results of low-loss gears.

as well as those of the corresponding low-loss gears. Additionally, the calculated safety factors are shown. Those of low-loss gears are at least equal to or even higher than those of Type C gears.

Figure 16 shows experimental results of the power losses of both Type C gears and corresponding low-loss gears. Enormous power loss savings of up to two-thirds can be achieved with low-loss gears. The given percentages of power loss reduction refer to the total power loss where higher bearing losses of low-loss gears are included because of the higher pressure angle. So, the pure load-dependent gear losses are even further reduced than the numbers suggest.

Besides power loss reduction, lower bulk and oil temperatures can also be achieved. Measurements during tests without lubrication showed a bulk temperature of 105°C for conventional gears where low-loss gears achieved 67°C. Also, its lifetime may be extended: Sample tests with coated gears—but without lubrication—showed that the coated gears had 15–20 times the load cycles of conventional gears before the coating was damaged.

Certainly, low-loss gears have some less favorable properties, including possible effects on design space and excitation level. Advantages like power loss reduction, temperature decrease and extreme lifetime extension make low-loss gears a viable option for many applications.

Conclusion

From the investigations shown in this report, the following main conclusions can be drawn:

- In a gearbox, bearing losses are subordinate to the losses in gears.
- In a gearbox, no-load losses are subordinate to the load-dependent losses at usual operating conditions. For operation at part-load or very high speeds, no-load losses may exceed the load-dependent losses.
- Load-dependent gear losses can be influenced by a range of

parameters, which either affect the load distribution along the path of contact or the coefficient of friction between the mating gear flanks.

- Conventional gears can be enhanced, but their power loss reduction is limited.
- With basically changed gear geometry, which optimizes the composition and interdependencies of all gear parameters, considerable loss reduction can be achieved.

Further investigations are necessary in the fields of load-carrying capacity and noise excitation properties of unconventional low-loss gear design.

Acknowledgment

This paper contains results which were elaborated on the behalf of the sponsored project No. IPS-2001-80006 'Oil-free Powertrain' of the European Union EU. Therefore, we want to thank all participating industrial associations/groupings:

VDMA—Verband Deutscher Maschinen- und Anlagenbau, Frankfurt, Germany;

FVA—Forschungsvereinigung Antriebstechnik, Frankfurt, Germany;

FVV—Forschungsvereinigung Verbrennungskraftmaschinen, Frankfurt, Germany;

FMS—Fachverband der Maschinen- und Stahlbauindustrie Österreichs, Wien, Austria;

Magosz—National Association of Hungarian Engineering Industry, Budapest, Hungary.

SST—Association of Manufacturers and Suppliers of Engineering Technique, Prague, Czech Republic;

AIA—Automotive Industry Association, Prague, Czech Republic.

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The Use of Nonstandard Tooth Proportions and Center Distance to Improve the Performance of Gear Trains

Isaias Regalado

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Management Summary

During the design of a gear train, there may exist geometric constraints leading to the use of a nonstandard center distance and a profile shift in the gears. That is the case with countershafts, or gears with small numbers of teeth where undercut is to be avoided. Besides these special cases, the use of nonstandard proportions in the teeth of the gears may be used to improve the performance of the transmission in aspects like contact ratio, specific sliding, bending strength, balance in life, scoring capability, efficiency, etc.

With the right selection of nonstandard center distance and tool shifting, it may be possible to use standard tools to improve the gear set capacity with a considerable reduction in cost when compared to the use of special tools.

This paper presents an analysis of the effects in the performance of gears due to a deviation from the standard proportions and proposes an optimization procedure for the selection of the best geometry of the gears assuming generation with a standard rack or hob.

Introduction

The general geometry of the gears for a gear set is defined during the first stage of the design process. This includes mainly the following information:

a) Number of teeth in the pinion
b) Number of teeth in the gear
c) Normal or transverse module
d) Normal or transverse pressure angle
e) Helix angle
f) Operating center distance
g) Addendum coefficient of the pinion
h) Addendum coefficient of the gear
i) Normal or transverse backlash
j) Outside diameter of the pinion
k) Outside diameter of the gear
l) Face width

From the previous list, items "a" to "e" are the minimum necessary for a gear set with standard proportions. For a nonstandard center distance, the operating center distance and two of the items from "g" to "i" need to be defined, while items "j" and "k" must be specified if a special topping hob is going to be used or when special dimensions are required in the gear tooth height.

Information regarding the first step of gear design can be found in the literature, and the AGMA (Refs. 1, 3) has standardized the procedure for evaluation of the performance in bending and pitting of gears.

According to the AGMA (Ref. 1), the basic equations for standard tooth geometry adjusted for helical gears and in metric units are as follows:

$$a = m_n \tag{1}$$

$$b = 1.25 m_n \tag{2}$$

$$c = 0.25 m_n \tag{3}$$

$$t = \frac{\pi}{2} m_n \tag{4}$$

$$D = \frac{N \cdot m_n}{\cos(\psi)} \tag{5}$$

$$D_B = D \cdot \cos(\phi_t) \tag{6}$$

$$D_O = D + 2a \tag{7}$$

$$D_R = D - 2b \tag{8}$$

$$C = \frac{(N_P + N_G) m_n}{2 \cos(\psi)} \tag{9}$$

where:

$$\phi_t = A\tan\left(\frac{\tan(\phi_n)}{\cos(\psi)}\right) \tag{10}$$

We can use N_P or N_G instead of N in Equation 5 to calculate the diameters for pinion and gear respectively.

In nonstandard gears, the profile shifting modifies the tooth thickness of the gear. Also, in order to keep the standard whole depth in the tooth, we must modify its outside diameter and root diameter. This will change Equations 1, 2 and 4 to Equations 11, 12 and 13, respectively.

$$a = (1 + X) m_n \tag{11}$$

$$b = (1.25 - X) m_n \tag{12}$$

Nomenclature

Ah	Addendum coefficient in the hob
A	Addendum
B	Coefficient of normal backlash
Bh	Dedendum coefficient in the hob
B_t	Coefficient of transverse backlash
b	Dedendum
C	Theoretical center distance
C_r	Operating center distance
c	Root clearance
D	Theoretical pitch diameter
D_{BP}, D_{BG}	Base diameter of pinion and gear
D_O	Outside diameter
D_R	Root diameter
D_P, D_G	Operating pitch diameter of pinion and gear
m	Transverse module
m_G	Gear ratio
m_n	Normal module
m_p	Transverse contact ratio
N_P, N_G	Number of teeth in pinion and gear
P_{Bt}	Transverse base pitch
Rh	Tip radius coefficient of the hob
t	Normal theoretical tooth thickness
t_t	Transverse theoretical tooth thickness
v_{rP}, v_{rG}	Rolling velocity in pinion and gear
X	Coefficient of shifting of the profile
X_B	Coefficient of profile shifting for backlash
X_P, X_G	Coefficient of profile shifting in pinion and gear
X_T	Coefficient of total profile shifting
Z_A	Length of approach
Z_P, Z_G	Radius of curvature at the tip of the tooth
Z_R	Length of recess
Z_{PP}, Z_{PG}	Radius of curvature at operating pitch point in the pinion and gear
ϕ_n	Normal theoretical pressure angle
ϕ_t	Transverse pressure angle
ϕ_{to}	Transverse operating pressure angle
γ_P, γ_G	Specific sliding in pinion and gear
ρ_P, ρ_G	Radius of curvature in pinion and gear
ω_P, ω_G	Angular velocity of pinion and gear
ψ	Theoretical helix angle

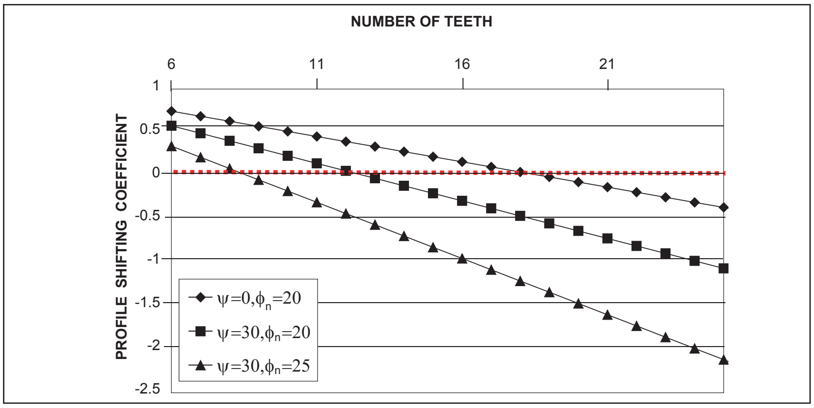

Figure 1—Profile shifting to avoid undercut. In this chart, any point above the red line indicates the need for a positive tool shifting to avoid undercut.

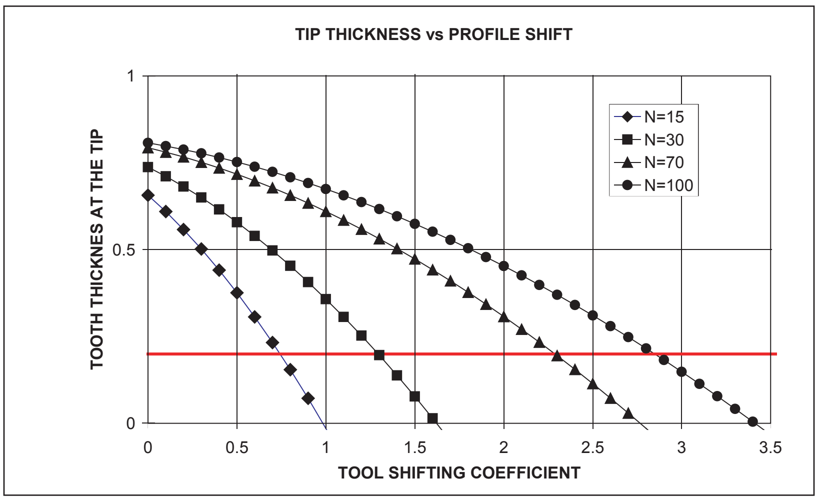

Figure 2—Tip tooth thickness for different profile shifting in spur gears. The red line in this figure represents the minimum allowable tooth tip thickness.

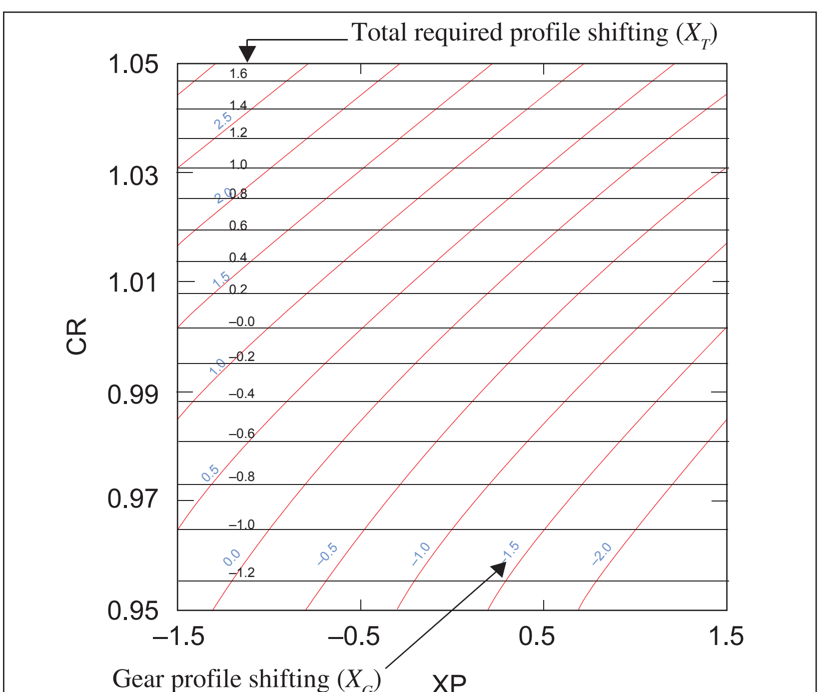

Figure 3—Gear and total required profile shifting in the (XP, CR) space.

$$t = \frac{\left(\frac{\pi}{2} + 2 \cdot X \cdot \tan(\phi_n)\right) m_n}{\cos(\psi)} \qquad (13)$$

We can use X_P or X_G instead of X in Equations 11–13 in order to calculate the addendum, dedendum and tooth thickness for pinion and gear, respectively.

It is well known that the module or diametral pitch of a gear works as a scaling factor; therefore, for the purpose of this paper, the normal module will be considered unitary, and may be removed from the analysis.

The first consideration for the deviation from the standard geometry that must be considered is the profile shifting to avoid undercut (generating interference); this will give us the minimum profile shifting in each individual gear. Mabie and Ocvirk (Ref. 3) present a detailed analysis of the undercut process. Equation 14 gives the minimum coefficient of profile shifting required to avoid undercut.

$$X \geq y - \frac{N}{2\cos(\psi)}(\sin(\phi_t))^2 \qquad (14)$$

where:

$$y = Ah - Rh(1 - \sin(\phi_n)) \qquad (15)$$

Evaluating Equation 14 for different combinations of pressure angle and helix angle, we may get a chart like the one shown in Figure 1. In this chart, any point above the red line indicates the need for a positive tool shifting to avoid undercut.

On the other hand, the profile shifting assigned to a gear cannot go beyond the condition of a pointed tooth; this is particularly important in hardened gears, because a hard-pointed tooth tends to be brittle. Therefore, it is a common practice to limit the minimum tooth tip thickness; in our case, we will use a minimum of $0.2m_n$. Figure 2 shows the tooth tip thickness for spur gears with different numbers of teeth and a 20° normal pressure angle. The red line in this figure represents the minimum allowable tooth tip thickness. It may be observed that the higher the number of teeth, the higher the allowed profile shifting before pointed teeth occur.

For the calculation of the deviations from the standard geometry needed in a pair of gears, we must take into account their operating center distance. Defining CR as the ratio between the operating and theoretical center distances—$CR = C_r/C$—we may consider three cases, namely:

$CR < 1$ Reduced center distance
$CR = 1$ Standard center distance
$CR > 1$ Extended center distance

In any of these cases, the profile shifting in the pinion and gear may be negative, positive or zero; and they must be calculated to get the total required profile shifting. This profile shifting is defined by Equation 16.

$$X_T = X_B + \frac{(N_P + N_G)\left[\mathrm{inv}\,(\phi_{to}) - \mathrm{inv}\,(\phi_t)\right]}{2\tan\ (\phi_n)} \quad (16)$$

where:

$$X_T = X_P + X_G \quad (17)$$

$$\cos(\phi_{to}) = \left(\frac{\cos\ (\phi_t)}{CR}\right) \quad (18)$$

$$X_B = \frac{B_t}{2\tan\ (\phi_{to})} \quad (19)$$

The space defining the geometry possibilities for a specific pair of gears may be handled as a two-dimensional space where one of the axes represents the profile shifting in one of the gears (let's say the pinion X_p) and the other axis represents the center distance ratio CR. In this design space, the standard case corresponds to the point defined by ($X_p = 0$, $CR = 1$).

For the first part of this study, a design space limited by $-1 \leq X_p \leq 1$ and $0.95 \leq CR \leq 1.05$ is evaluated, using a gear set and generating tool defined by:

$N_p = 25$
$N_G = 34$
$\phi_n = 20$
$\psi = 0°$
$B_t = 0.035$
$A_h = 1.25$
$B_h = 1.0$
$R_h = 0.3$
$n_P = 2{,}000$ rpm

Figure 3 shows the required X_T and X_G in the design space; it may be seen that X_T depends only on CR. In this design space, the limits of shifting for undercut and pointed tooth X_p and X_G are defined by Equations 20

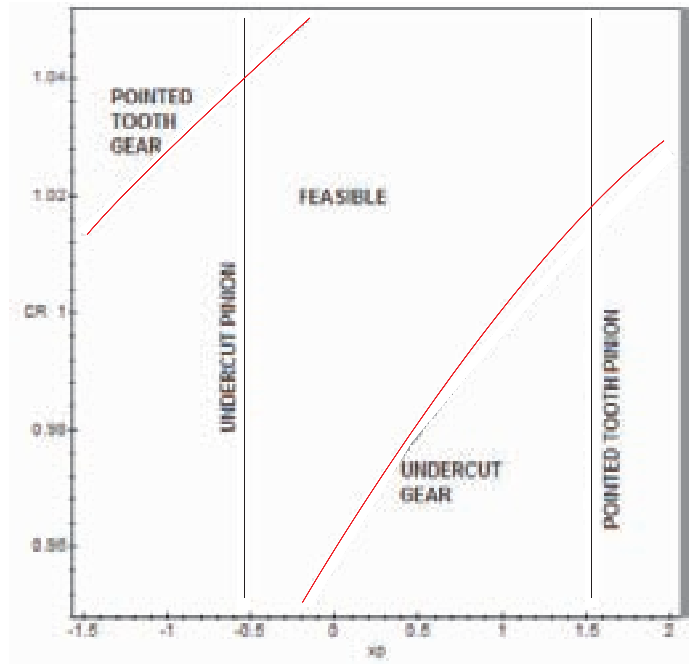

Figure 4—Limits for undercut and pointed tooth in the (XP, CR) space.

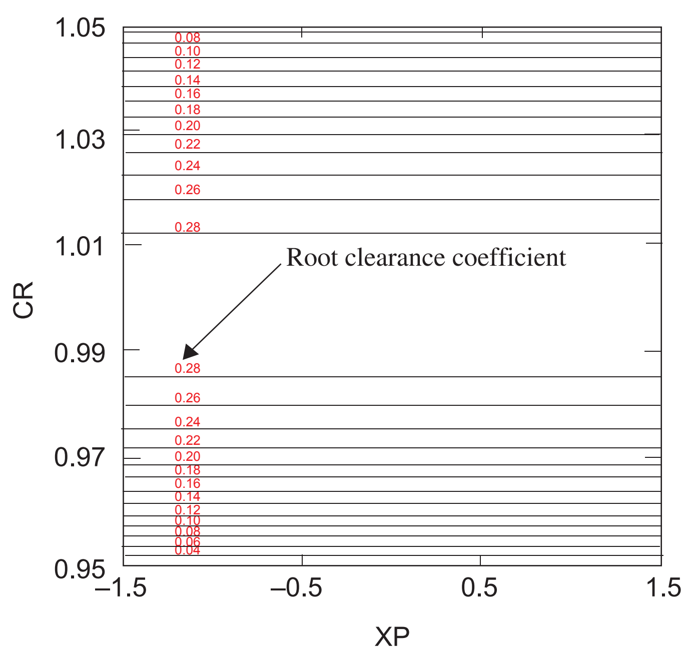

Figure 5—Root clearance in the (XP, CR) space.

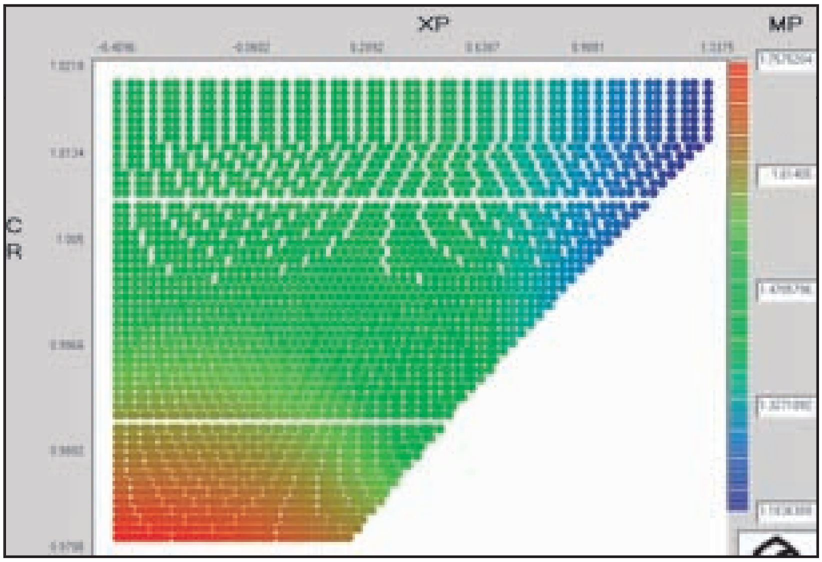

Figure 6—Transverse contact ratio in the (XP, CR) space.

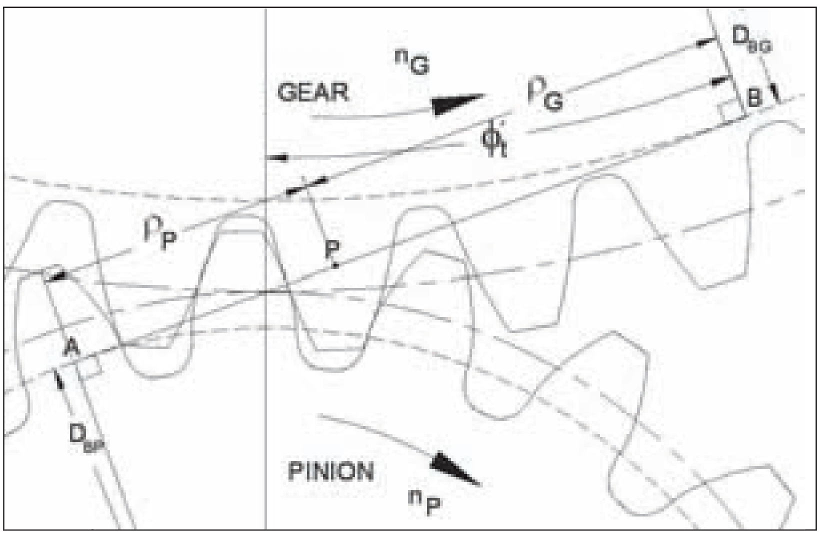

Figure 7—Radius of curvature for pinion and gear for a point "P" along the line of action.

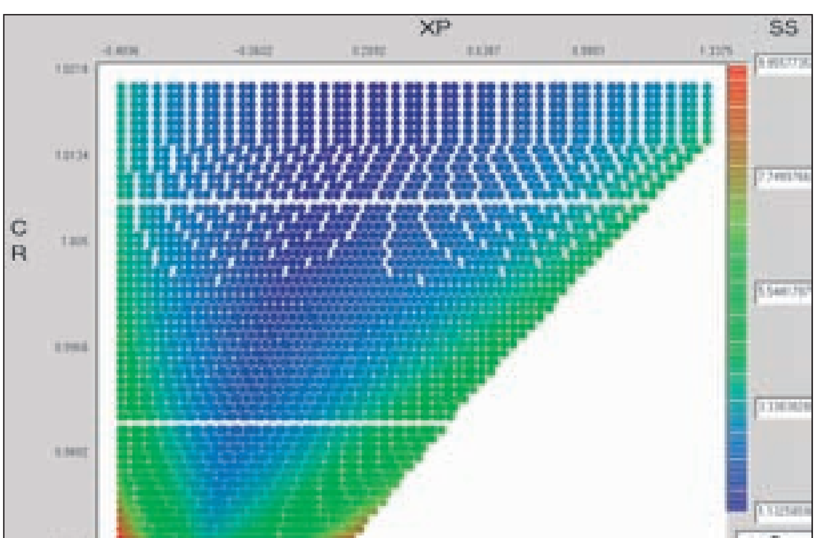

Figure 8—Specific sliding in the (XP, CR) space.

and 21 and are shown in Figure 4.

$$X_{P\,min} = Max\left[X_{Pundercut}\,;(X_T - X_{Gpointed})\right] \quad (20)$$

$$X_{P\,max} = Min\left[X_{Ppointed}\,;(X_T - X_{Gundercut})\right] \quad (21)$$

Note that the limits for undercut and pointed teeth are individual for each gear and do not consider the mating gear. These limits in the (X_p, CR) space are shown in Figure 4, where XG has been calculated using Equation 17.

By definition, the pinion is the member with fewer number of teeth; therefore, the feasible region will always be limited by the undercut line for the pinion and either the undercut line for the gear or the pointed tooth line for the pinion (See Fig. 4).

Now considering the mating gears, some of the parameters that indicate the performance of the gear set are: root clearance, operating interference, contact ratio, specific sliding, recess to approach ratio, balance in life, etc. Each of these parameters will be analyzed separately below.

Root Clearance

The root clearance is given by Equation 22:

$$c = (Ah - Bh) + (C_r - C) - X_T \quad (22)$$

where the first parenthesis corresponds to the theoretical root clearance. Figure 5 shows the root clearance in the (X_p, CR) design space. From the figure, it is evident that the greater the limit for root clearance, the smaller the allowed deviation in center distance. It is also evident that the root clearance depends only on the value of CR and is independent of the profile shifting in the pinion or the gear.

Operating Interference

Operating interference occurs when the tip of the tooth in one of the gears goes beyond the point of tangency between the base circle of the other gear and the line of action. The limiting outside diameters for interference are defined by Equations 23 and 24.

$$\frac{D_{OP}}{2} \le \sqrt{(C_r \sin(\phi_{to}))^2 + \left(\frac{D_{BP}}{2}\right)^2} \quad (23)$$

$$\frac{D_{OG}}{2} \le \sqrt{(C_r \sin(\phi_{to}))^2 + \left(\frac{D_{BG}}{2}\right)^2} \quad (24)$$

It is important to observe that if both the pinion and gear have been generated using a hob or a rack, and none of them have undercut, this guarantees that no operating interference will exist. Therefore, if an adequate profile shifting to avoid undercut is used, interference will never limit the feasible region.

In the next section, the performance of the transmission will be analyzed for transverse contact ratio, specific sliding, recess vs. approach ratio, pitting and bending. The plots will show only the feasible region limited by the undercut line in the pinion and the undercut line of the gear, for a minimum root clearance coefficient of 0.2.

Transverse Contact Ratio

The transverse contact ratio is a numerical indication of the continuity of action in a gear set and is given by Equation 25.

$$m_p = \frac{Z_P + Z_G - C_r \, Sin\,(\phi_{to})}{P_{Bt}} \qquad (25)$$

Where:

$$Z_P = \frac{\sqrt{(D_{OP}{}^2 - D_{BP}{}^2)}}{2} \qquad (26)$$

$$Z_G = \frac{\sqrt{(D_{OG}{}^2 - D_{BG}{}^2)}}{2} \qquad (27)$$

$$P_{Bt} = \frac{\pi D_{BP}}{N_P} \qquad (28)$$

Figure 6 shows the transverse contact ratio in the (X_p, CR) design space. It may be observed that the greater the desired transverse contact ratio, the smaller the size of the feasible space. It is also observed that a reduced center distance and negative profile shift in the pinion produces a better contact ratio in the gear set.

Specific Sliding

The specific sliding is a way to measure the amount of sliding during the mating of the gears, and is defined by Equations 29 and 30 (Ref. 5).

$$\gamma_G = \frac{v_{rG} - v_{rP}}{v_{rG}} \qquad (29)$$

$$\gamma_P = \frac{v_{rP} - v_{rG}}{v_{rP}} \qquad (30)$$

where:

$$v_{rP} = \omega_P \, \rho_P \qquad (31)$$

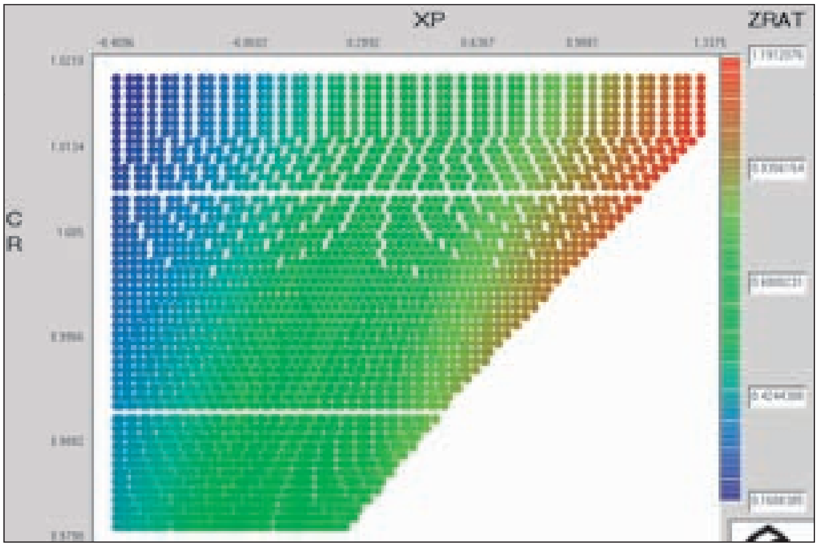

Figure 9—Ratio between recess and approach in the (XP, CR) space.

$$v_{rG} = \varpi_G \, \rho_G = \frac{\omega_P \, \rho_G}{m_G} \qquad (32)$$

$$\varpi_P = \frac{\pi \cdot n_P}{30} \; ; \; \varpi_G = \frac{\pi \cdot n_G}{30} \qquad (33)$$

and the radius of curvature will be measured from the point of tangency of the line of action with the base circle of the pinion and gear respectively to the point of analysis (Fig. 7).

Specific sliding may be evaluated at different points along the line of action, but the points where the maximum specific sliding occurs are the start and end of contact; therefore, only these points are considered in the analysis. Figure 8 shows the maximum absolute value of specific sliding in the (X_p, CR) space. Note that an extended center distance design with zero to positive profile shift for the pinion produces smaller specific sliding values. Some authors recommend a specific sliding no bigger than 3, and that constraint is accomplished inside the feasible region at the upper centered part of Figure 8.

Recess to Approach Ratio

Another way some authors suggest to measure mesh smoothness is the ratio between the length of approach versus the length of recess; some designers even use what is called a full recess action gear set where all the contact takes place above the operating pitch circle of the pinion. The length of approach and recess are given by Equations 34 and 35. Figure 9 shows the recess to approach ratio in the design space. According to the figure, the more posi-

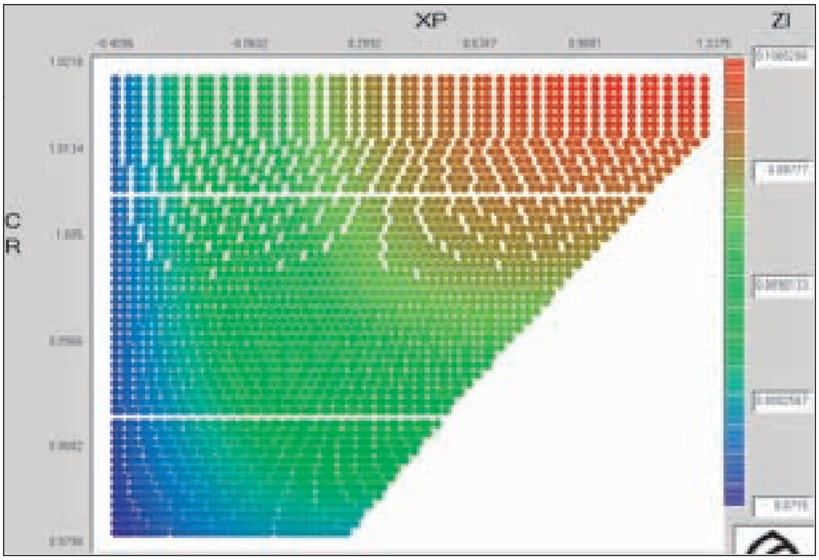

Figure 10—Geometry factor for the calculation of the pitting resistance in the (XP, CR) space.

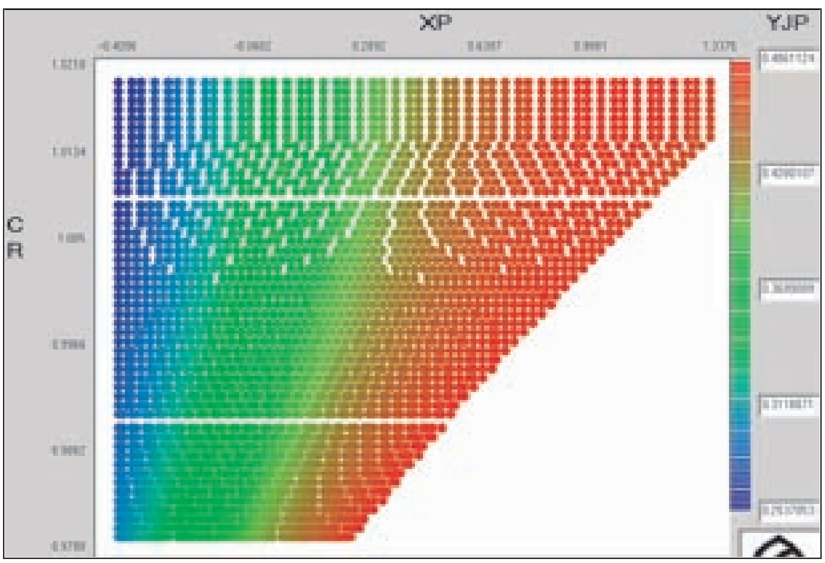

Figure 11—Geometry factor for the calculation of the pinion bending strength in the (XP, CR) space.

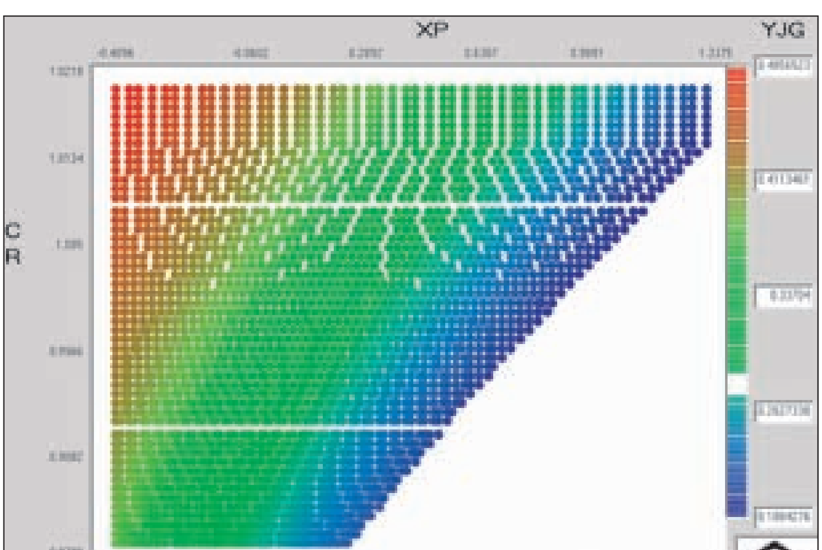

Figure 12—Geometry factor for the calculation of the gear bending strength in the (XP, CR) space.

tive the profile shift of the pinion, the bigger the recess/approach ratio. In fact, the exact full recess action occurs when $X_p = 1$.

$$Z_A = Z_{PP} + Z_G - C_r \sin(\phi_{to}) \tag{34}$$

$$Z_B = Z_{GG} + Z_P - C_r \sin(\phi_{to}) \tag{35}$$

where:

$$Z_{PP} = 0.5\sqrt{D_P^2 - D_{BP}^2} \tag{36}$$

$$Z_{GG} = 0.5\sqrt{D_G^2 - D_{BG}^2} \tag{37}$$

$$D_P = \frac{2C_r\, N_P}{N_G + N_P} \tag{38}$$

$$D_G = 2C_r - D_P \tag{39}$$

Pitting and Bending Strength

Per AGMA (Ref. 2), the pitting stress in the gears is given by Equation 40, and the relationship to evaluate pitting durability is given by Equation 41.

$$\sigma_H = Z_E \sqrt{F_t\, K_o\, K_v\, K_s \frac{K_H}{D_P\, b} \frac{Z_R}{Z_I}} \tag{40}$$

$$\sigma_H \leq \frac{\sigma_{HP}}{S_H} \frac{Z_N}{Y_\vartheta} \frac{Z_W}{Y_Z} \tag{41}$$

Also, the corresponding expressions for bending stress and bending durability in the gears are given by Equations 42 and 43, respectively.

$$\sigma_F = F_t\, K_o\, K_v\, K_s \frac{1}{bm} \frac{K_H\, K_B}{Y_J} \tag{42}$$

$$\sigma_F \leq \frac{\sigma_{FP}}{S_F} \frac{Y_N}{Y_\vartheta Y_Z} \tag{43}$$

where

σ_H is contact stress number N/mm2
Z_E is elastic coefficient of the materials [N/mm2]0.5
F_t is transmitted tangential load N
K_o is overload factor
K_v is dynamic factor
K_s is size factor
K_H is load distribution factor
Z_R is surface condition factor for pitting

resistance

b is net face width of narrowest member

Z_I is geometry factor for pitting resistance

σ_{HP} is allowable contact stress number N/mm2

Z_N is stress cycle factor for pitting resistance

Z_W is hardness ratio factor for pitting resistance

S_H is safety factor for pitting resistance

Y_ϑ is temperature factor

Y_Z is reliability factor

σ_F is bending stress number N/mm2

K_B is rim thickness factor

Y_J is geometry factor for bending strength

σ_{FP} is allowable bending stress number N/mm2

Y_F is stress cycle factor for bending strength

S_F is safety factor for bending strength

For each particular design, most of the factors remain unchanged, and they may be included in de-rating factors defined by Equations 44–46.

$$K_{DH} = K_o\ K_v\ K_s \frac{K_H\ Z_R}{D_P\ b} \qquad (44)$$

$$K_{DPP} = K_o\ K_v\ K_s \frac{K_H\ Z_{BP}}{bm} \qquad (45)$$

$$K_{DFG} = K_o\ K_v\ K_s \frac{K_H\ Z_{BG}}{bm} \qquad (46)$$

Also, AGMA recommends that, for a general-purpose application, $Z_W = S_H = Y_Z = Y_\vartheta = S_F = 1$; therefore, the major effects in the stress calculation are given by the geometry factors Z_I, Y_{JP} and Y_{JG}. Also, these factors appear in the denominator of the stress calculation equation, and thus, the bigger the factor, the smaller the stress and the bigger the strength.

Figures 10–12 show the calculated geometry factors for pitting and bending in the design space.

From Figure 10, it is observed that, for better pitting strength, an extended center distance and a positive tool shifting in the pinion are recommended. Figure 11 shows that in order to improve the bending strength of the pinion, a positive tool shifting in the pinion is required and is almost independent of the value of *CR*. Figure 12, on the other hand, shows that we must use an extended center distance and a negative tool shifting in the pinion (positive in the gear) if we want to improve the bending strength of the gear.

Due to the fact that the number of cycles of operation in the pinion is usually bigger than in the gear (speed reducer)—and in order to get a balanced durability in pinion and gear—it is a common practice to use a stronger material in the pinion than in the gear. A more refined approach to this practice is the selection of a nonstandard proportion gear set, wherein the ratio between the bending geometry factors is equal to the relationship given in Equation 47:

$$\frac{Y_{JP}}{Y_{JG}} = \frac{\sigma_{FG}}{\sigma_{FP}} \frac{Y_{NG}}{Y_{NP}} \qquad (47)$$

For our example, assume that a 400-Brinell, through-hardened steel pinion will be used with a 250-Brinell gear, and an equal durability in both gears corresponding to 106 cycles in the pinion is required. First of all, being a speed reducer, the required durability in the pinion corresponds to 7.35x105 cycles in the gear; therefore, from Figure 13, the required stress cycle factors are:

$Y_{NP} = 1.2232$
$Y_{NG} = 1.2043$

Per AGMA (Ref. 2), the allowable bending stress number for a through-hardened steel is obtained from Figure 14. For our example,

$\sigma_{FP} = 300$ MPa
$\sigma_{FG} = 220$ MPa

Therefore, the following ratio is required:

$$\frac{Y_{JP}}{Y_{JG}} = \frac{220}{300} \frac{1.2043}{1.2232} = 0.7220 \qquad (48)$$

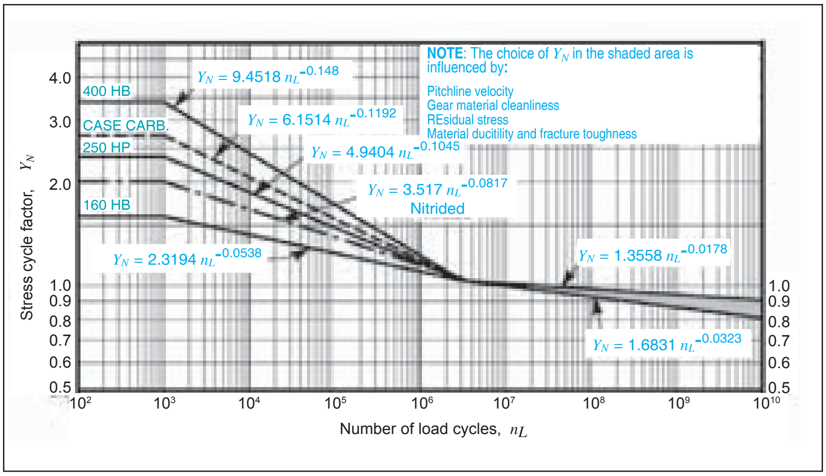

Figure 13—Bending strength stress cycle factor, from AGMA (Ref. 2).

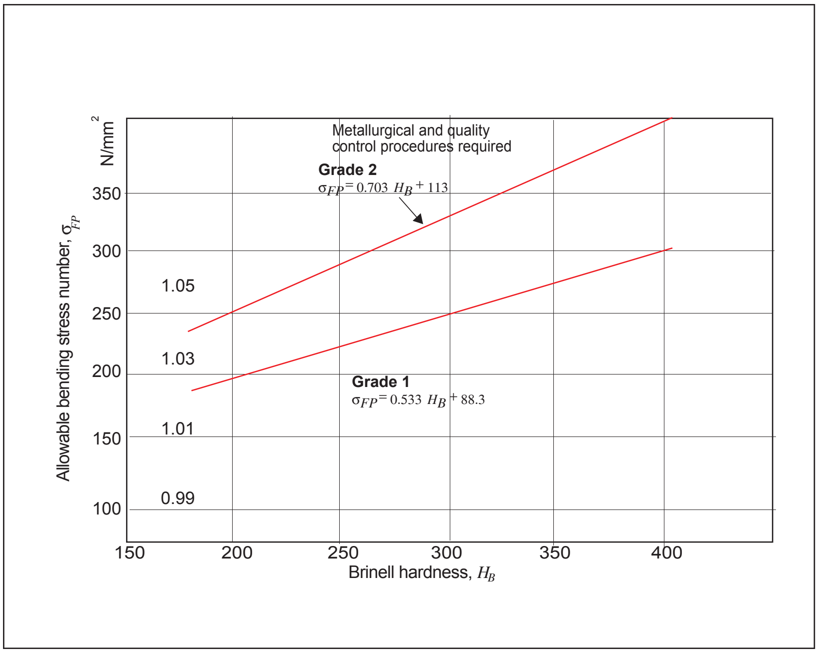

Figure 14—Allowable bending stress number for through-hardened steel gears, from AGMA (Ref. 2).

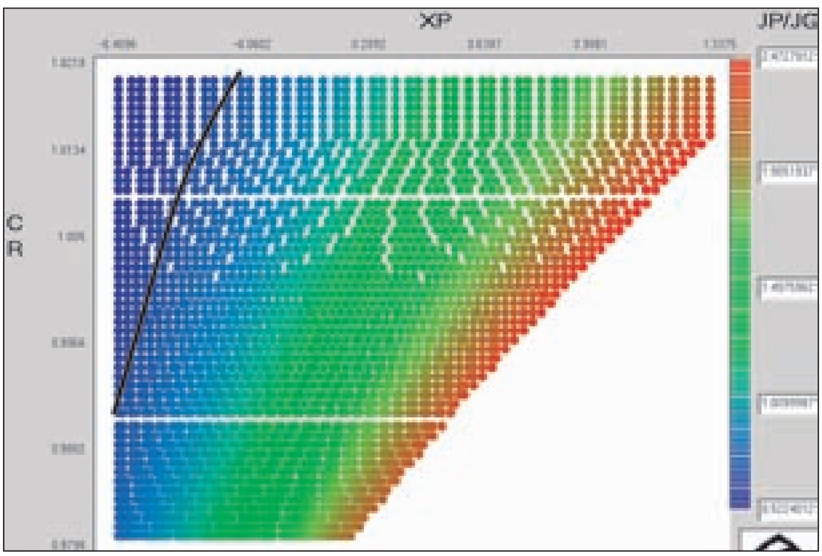

Figure 15—Ratio Y_{JP}/Y_{JG} in the (XP, CR) space.

Figure 15 shows the ratio Y_{JP}/Y_{JG} for the (XP, CR) space, where the contour for the desired ratio is traced with a black line.

A similar analysis may be performed for a desired life relationship between pitting and bending of pinion and/or gear. In general, the selection of the best combination of (XP, CR) for a particular gear set must be based on the specific needs of the application, and some tradeoffs must be made between the performance criteria. These tradeoffs are beyond the scope of this paper, and more information may be found in Reference 6.

As general guidelines, we suggest the following steps:

1) Select the number of teeth in pinion and gear using the procedure outlined by AGMA (Ref. 4) for the preliminary selection of the best number of teeth. When possible, it is recommended to use a hunting tooth.

2) Determine the limits in profile shifting for undercut and pointed tooth in each gear.

3) Define the limits of variation in operating center distance based on the desired root clearance.

4) Define the desired values for transverse contact ratio, specific sliding, recess/approach ratio and expected bending and pitting life.

5) Establish a preference level for each of the required performance criteria (Ref. 6).

6) Find a point in the (Xp, CR) space that best satisfies the required performance criteria with the desired preference levels.

Conclusion

We have shown that the performance of a gear set may be changed considerably by changing the tooth proportions and center distance from the theoretical (standard) values. It has also been observed that the improvement of some performance criteria may lead to the deterioration of others. A tradeoff between them must be done in order to get a better gear design.

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Wind Turbine Market

Leads Hansen Transmissions to India

When Belgium-based Hansen Transmissions was under the ownership of Invensys plc in the late 1990s, the parent company was dropping not-so-subtle hints that the industrial gearbox manufacturer was not part of its long-term plans. Yet Hansen's CEO Ivan Brems never dreamed that, less than a decade later, he would be working for an Indian company.

Hansen was bought by Suzlon Energy in March, 2006. Suzlon ranks number 1 in Asian wind energy turbine production and number 5 worldwide, and employs 9,500 people.

The Hansen organization consists of two business units. The wind energy unit focuses on development and manufacturing of gearboxes for wind turbines. An industrial unit serves customers with its Hansen P4 and Hansen M4 range in applications including cooling tower drives, aerators, screw pumps, conveyors, bucket elevators, mixers and pulp and paper manufacturing. Nearly 99% of Hansen's production is exported worldwide. Last year, Hansen was awarded the Belgium Export Lion Trophy for "Best Performer," recognizing its achievements in exports as well as ethical and sustainable entrepreneurship.

Rapid Expansion All Over the Globe

In recent years, the company has focused much of its efforts on production capacity for wind turbines. In 2004, it opened an ultra-modern, 101 million euro, 120,000-m^2 plant for wind turbine gearboxes in Lommel, northeast Belgium. The plant produces an annual output of 2,200 megawatts in gear drives, which, as Brems points out, is significant when one considers that a 1-megawatt wind turbine supplies energy for 750 households.

Last year, Hansen bought 63,500 m^2 of land and announced plans for a further investment of 140 million euros to double production capacity and expand R&D facilities. They also unveiled blueprints for a 65,000-m^2 wind turbine gearbox plant in Tamil Nadu, India.

Where to Begin? A Template for Starting Operations in India

Despite the new ownership, day-to-day operation has changed very little for Hansen Transmissions.

"There was always a clear understanding that Hansen will continue to operate independently of the Suzlon Group," says Brems. "When Suzlon bought us, they were not yet a customer of Hansen. We worked very hard to develop prototypes and gain Suzlon as a new customer. So there are two relationships at play here between the two organizations, one as an owner—but that is interfaced through a board of directors—and the second relationship of Suzlon as a customer, which is based on terms similar to other customers."

Components are still entirely manufactured in Belgium, although this will change as production in India is ramped-up in the coming years. Where the partnership has paid off is in Suzlon's access to Asian markets and high profile in the wind energy world. While numerous European companies have assembly centers in India, none can benefit from Suzlon's wind energy network as well as Hansen.

This aside, India is geographically well situated to any business seeking access to the Asian market.

Although Hansen was partnered with India's most prominent wind turbine name, hiring in this particular niche of the industry was somewhat of a challenge. Brems elaborates: "The good thing is that we've undergone heavy expansion in the past five years. I feel we have a good template for hiring. When we built the second factory in Belgium, we faced the task of hiring and training 300–400 people. At a very early stage, we realized that recruitment was going to be our greatest challenge. One idea that proved very successful was to undertake a lot of advertising and TV time promoting Hansen and reporting the progress of our factory and the wind energy business. What also worked well was going to local technical schools and organizing a contest, allowing local students to participate. We were looking for metalworkers and went to schools asking for a metal sculpture relating to energy and renewable energy. We ended up with 15 schools participating, held the drawing and an independent panel of judges decided upon the best three entries. We found employees that way and also exhibited all the winning objects in our factory."

It was hard enough to find workers in Belgium, but the factory in India provided an entirely different set of challenges. Hansen was a step ahead of the game in that they had the relationship with Suzlon, who organized travel, found temporary offices, dealt with administration, helped find contractors and introduced Hansen to recruiting teams. Ultimately, Brems said, India has a different culture and the company anticipated difficulty with language barriers, time zones, etc.

Overcoming these obstacles was not as hard as one might think. Brems pointed out that language-wise, India is much easier to deal with than, say, China. English is recognized as the business language of India, so Hansen has never had to employ the services of a translator.

"The legal system in India is very much based upon what the English left behind in the 1940s. Since everything is rooted in the Anglo system, it's very easy for an outsider to come in and understand the mentality and

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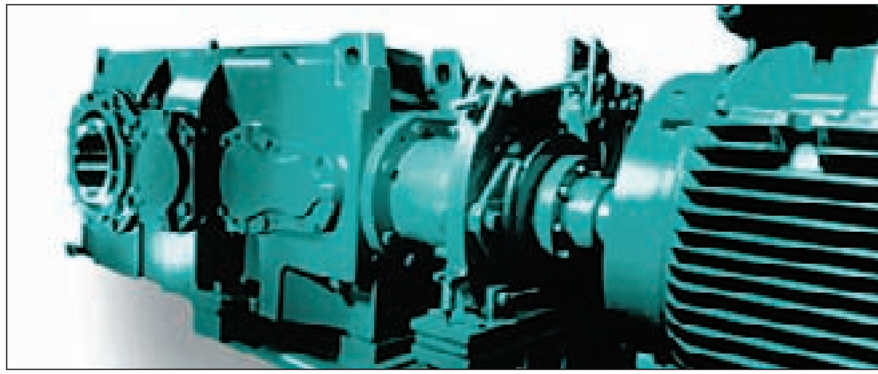
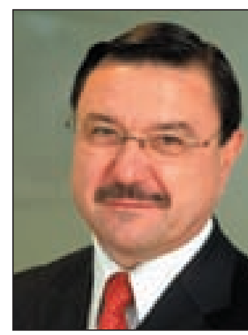


the logic."

When establishing a global operation, hiring new people is only part of the job. Equally important is familiarizing the current employees in Belgium with the day-to-day realities involved in working with their Indian counterparts. To that end, Brems says Hansen has organized cultural training sessions for its European employees to create awareness of certain aspects of Indian culture such as the prevalence of vegetarianism and differences in religion. In the end, he hopes the entire operation comes together to expand the growth of the wind energy market.

"We believe that the future of the wind energy industry is just great. We've been a supporter of the industry in the very, very early days in Denmark in the '70s and through some difficult periods in the '80s. A great mental driver right now in Europe and the U.S. is the concern about energy supply. There is plenty of wind and political willingness to capture that source of energy and that's a main impetus for future growth. India and China have an enormous need for energy, and one of the great advantages of wind energy is that it can be implemented very quickly, usually within a year or two without creating hidden costs caused by pollution. So we are highly optimistic."



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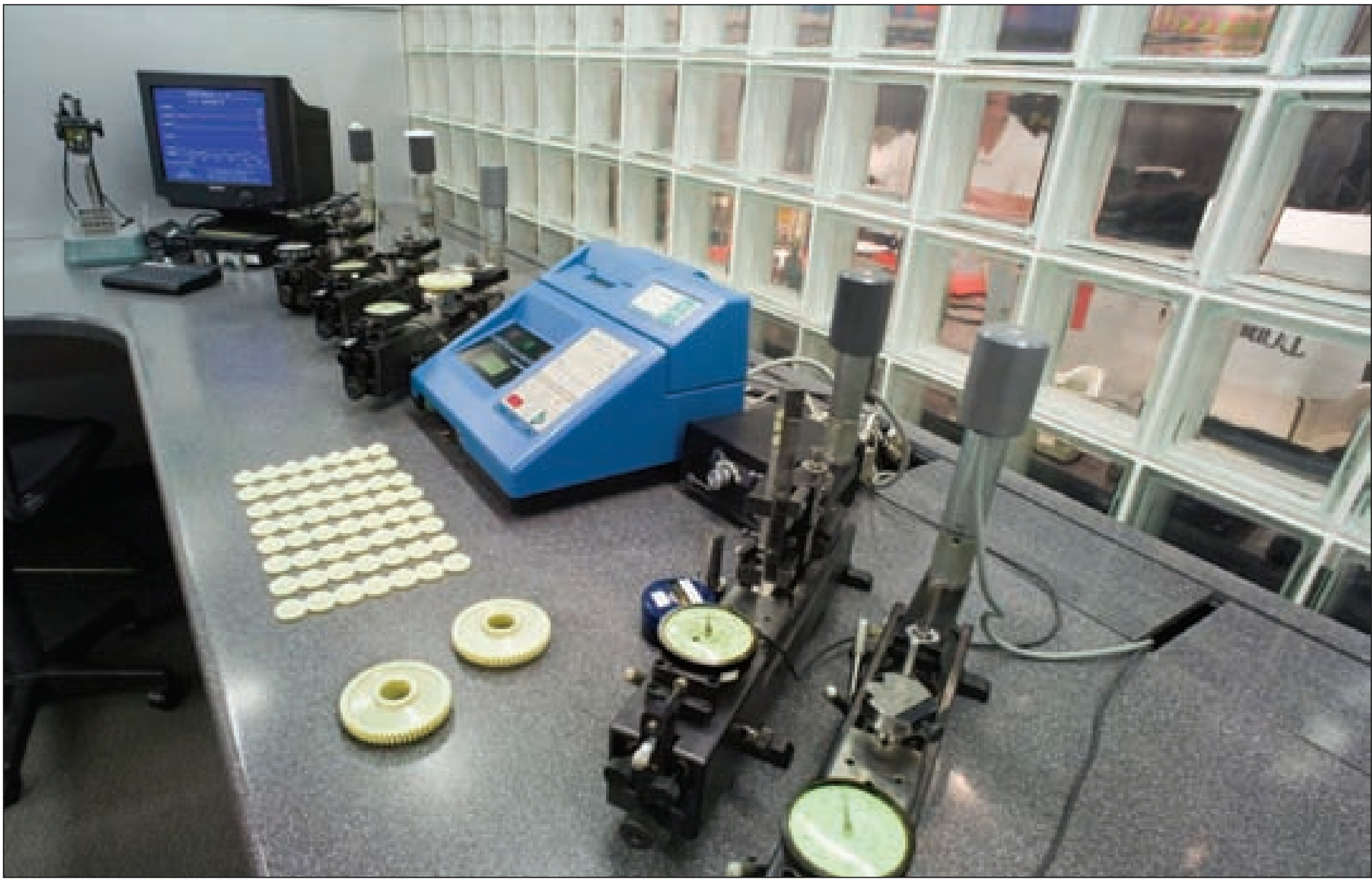
Going Lean is One Thing.

But Getting There? Quite Another.

Jack McGuinn, Senior Editor



Winzeler Gear's quality assurance gear lab, where all of the company's dual-flank gear testers are linked to a centralized data collection system, eliminating the need for manual entry. The collection system significantly reduced the amount of direct labor required to monitor gear production.

Google "lean manufacturing" and you will find a virtually endless font of information regarding formal lean implementation. You'll see definitions for Japanese words such as kaizen, gemba, muda, mura, kanban, and so on. You will also find other variations or iterations of lean, e.g.: Six Sigma, Lean Sigma, TPS (Toyota Production System), TOC (Theory of Constraints), JIT (Just in Time), and others.

But at Forest City Gear and Winzeler Gear, while top management made a strategic decision to commit to lean implementation, they also decided that they were going to Americanize it a bit and lose most of the formal lean jargon that, frankly, might prove somewhat intimidating and confusing to company personnel.

Winzeler began its lean journey about five years ago, and Forest City, just last November. What their approach to manufacturing improvement has in common is their working with the same consultant, Brian Barch's Telosis, Inc. Barch consults with Forest City on their ongoing quality systems certification, and with Winzeler Gear on their trek to the Promised Land of Lean. (By sheer coincidence and like-minded thinking, both companies call their lean effort the Operation Excellence Program.) And given that each company is similar (gears), yet different (steel vs. plastic), it is a testament to lean's far-reaching influence on any number of industries that the same consultant is helping to bring about quality and efficiency upgrades to both companies. Industries now benefiting from lean practices include banking, hospitals, and service industries—whether it's processing a part or processing a mortgage loan.

Developing "rules and tools." Barch's approach to going lean differs perhaps somewhat from more traditional implementation. His is an incremental, or "modified"—his word—approach which, if effectively utilized, guides a company step-by-step through various areas of its operation, ultimately resulting in a lean conclusion. Typically, says Barch, he is first invited in by companies—as is the case with Forest

City—looking for quality systems (ISO, QS, etc.) certification. And it is those efforts, he adds, which will eventually lead a company down the path to lean implementation.

"In developing an ISO program, what you are really doing is identifying, documenting and developing rules and tools," Barch says. "That tends to feed into management practices—not just from a quality system standpoint, but from a management system as well—i.e., how an organization is governed."

By design, Telosis' niche is smaller businesses. He has found that working with larger companies looking to go lean is a bit like navigating a battle ship's 180° turn—very slow. He says the reason for that is an inbred intransigence to change on the part of management.

"With big companies, it is an absolute unwillingness to bring down the ivory towers," he says. "Large organizations live and thrive on towers. You have engineering, purchasing, quality, etc., towers. And no matter what they say, they are keeping those walls around their tower."

It follows then, Barch believes, that lean implementation must include top management's buy-in to the fact that lean is all about people, the people in their employ. (*Ed.Note - Please see sidebar on p.55.*)

"Many companies go through a lean program and most of them fail at it or give up on it, because they do it wrong. They start at the wrong place. There's a function in every organization of getting today's work done today, and then there's a function in an organization of understanding and controlling the resources that that organization utilizes. And typically we (Telosis) are drawn into this whole program because top management wants to see what they call their management team develop and progress as far as their ability to participate in the management of the company allows.

"From there we don't start with things that might be strictly lean organizational things; we start out with job knowledge, what we call 'walking around knowledge'."

Barch explains that every company needs to identify personnel who have "daily" knowledge of key areas of the business; the knowledge-is-power approach. He feels that companies that manage an operation based simply upon monthly, quarterly, or annual reports are working backwards, much like "putting the lookout in the back of the boat."

"What (companies) need to do is manage on a real-time basis, and in order to do that you need to know what is happening in real time," he says. "So we'll go through a list of things that somebody in this organization should be aware of on a day-to-day basis. And then we'll say, given the organization and resources we have, who should have this knowledge? And then we'll go around and ask, who does have this knowledge? And universally we find that nobody has this knowledge."

With that information in hand, Barch then initiates a program to facilitate access to that knowledge and to develop it in such a way that key individuals may have access to it

continued

Forest City Gear's gear lab. The carts in the foreground are stocked with the job- and machine-specific tools and materials before being sent out to the shop floor.

on a real-time basis and can then start managing based upon what's happening on any given day. Barch provides a concrete example of this that any company accountant or controller can appreciate.

"If I were to go to an organization and ask someone to quantify for me what they have in inventory—dollar value—typically no one can answer that question (immediately). If you go to Winzeler Gear and you ask how many dollars do we have in inventory as of the end of business yesterday, there are three people who can answer it to the dollar.

"So as they start getting and developing that information, it is then easy to implement lean concepts. Once people know what they have, they can then start to develop objectives."

Inventory reduction via production. Barch again uses inventory as an example of this. Instead, for instance, of reducing inventory by, say, 30% or 40%, let's ask the question: Why should we have any inventory? "If a company believes its inventory to be excessive, what's the reason behind that?" he says. "And that usually leads us down a path to saying, cycle time. If the cycle time is extensive, it leads to saying, OK, we've got a 30-day cycle time from receipt of order to parts ready for delivery. What's really happening during that 30 days? How much time is going to actual value-added production as opposed to how much queue time is involved?"

As one might expect, getting answers to those questions leads management, production planners, etc., to realize that they are wasting a lot of valuable time. They further realize, says Barch, that if they can reduce cycle time to 15 days, there is little need for extensive—and expensive—warehousing.

Winzeler CEO John Winzeler is in complete agreement.

"Right now we're focusing on inventory reduction, for both raw material and finished goods, because we are running out of warehouse space (optimal space utilization being a key lean principle)," he says. "So before we reorganize the warehouse area, we're going through this exercise of understanding the consumption patterns of our customers and how much raw material we need, how much finished goods. And how can we best make sure that we never shut our customers down without taking up any more space—i.e., money—for that activity."

What Winzeler is alluding to is the difficult but necessary development of reliable metrics, which in turn allows for making sound decisions. It's actually step one in the process.

"It all started—and it took a good year or more—to get good metrics. What's our on-time delivery? What's our internal and external waste? We have several metrics that we track on a daily, weekly and monthly basis. You need that to know whether you're getting better or not.

"Take scrap rate, quoted material usage vs. actual, premium freight. It took a while to even get that data together, and it took a very qualified IT person to be able to capture the data and trust it. If we can't trust the data, we don't know what we're doing."

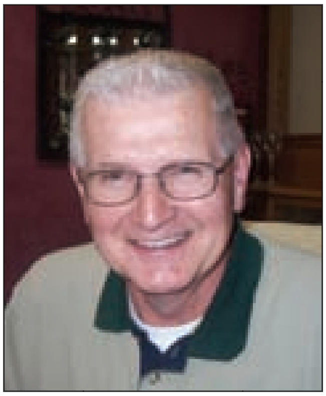
There is a central element of lean that is not lost on Winzeler. He is fully aware and appreciative of his people's efforts. And he also realizes that without the due diligence of his core management team—comprised of the company's director of quality, director of manufacturing and controller—and led by Barch, going lean would not be possible.

"In small manufacturing," says Winzeler, "the people in the trenches doing the work every day don't have a lot of time to do continuous improvement. We've taken more supervisory people to do the planning and training, and they engage the worker team. But it isn't just going out there and saying, 'OK, we're going to form a team. Now go make us better.' That doesn't work.

"And just because you have a team doesn't mean that you have any talent, either. You have to have somebody driving it. It's going to take us years to get the whole culture changed to the point that everybody is doing everything possible. But every day, every week, every month, we're getting better.

"You never reach the finish line because there's always more you can do. It's becoming lifelong learners, being curious enough to challenge everything you do every day."

Highfalutin', simplified. At Forest City Gear, the decision of going lean and committing to getting there made good business sense. Peer pressure (local competition) was one reason; more profits, a better one. Its Operation Excellence Program mirrors that of Winzeler Gear in that the order of the day is to keep things simple, and not, as vice president of manufacturing Everett Hawkins explains it, "highfalutin."


Everett Hawkins

"(Going lean) was being proactive on our behalf, because we know of other companies' successes with lean, especially in the Rockford (IL) area," he says. "We weren't forced into it. We just looked at it as an area that we certainly could use some improvement in. We think we can get a product through the plant a lot faster, and by doing that we can take on more business."

The "faster" Hawkins refers to is a direct result of an ongoing reorganization of the shop floor. Two crucial elements of lean are having the tools and materials necessary to do a job in close proximity to where the work is performed, and the other is an absolute necessity of having a designated place for every production tool. A recent tour

continued

Pat Keeley

of Forest City's shop floor revealed a strict adherence to both directives. All necessary tools and materials were situated at various machine stations; smaller tools were arranged and stored in a ship-shape manner that even the most extreme fussbudget would envy. We were also able to observe new shop floor plans which will eventually be executed and will significantly expedite queue and work flow. This provided an observer with a clear, real-time snapshot of lean in action.

Since the implementation began just six months ago, Hawkins says it will boost sales from $8 million to $9 million annually, with expectations that lean will get them up to around $12 million. Hawkins added that the company has also begun attracting new customers at a faster rate—now averaging at least one to two per week.

Hawkins also believes lean implementation will result soon in new business for which Forest City would otherwise not have been considered.

"I believe lean is going to position us to (attract) more aerospace work," Hawkins says. "It's certainly going to (bring) more medical (device) work. And it's going to increase our customer base, because we're taking on projects now that normally we wouldn't be able to do since now we can get things through the plant much faster."

Hawkins also points out that the company's long-held commitment to staying technologically viable has served to complement its lean efforts in a significant way.

"Due to new equipment, we feel we can get the hard type of work, from the standpoint of special gears. We just landed the 2009 gears for the Mars Rovers. Those people, when they were in here, were very impressed with our going lean."

There is no "I" in team—or lean. Pat Keeley, Forest City's quality manager, is in full support of the company's lean effort. And why not? He says it is already reaping rewards. For him, lean is all about personnel constantly asking themselves how they can best do their job and dedicated commitment to fulfilling their responsibilities.

"Although we are in the beginning stages of lean manufacturing, its benefits are already being realized," he says. "Lean is a thought process or a belief in how you go about performing your job, not necessarily added responsibilities. Being that it is based on a 'team' set of ideals, actually everyone in the 'group' shares in those responsibilities.

"My responsibilities to Forest City Gear are to ensure continued customer satisfaction. Lean will help us do this by improving quality, price and delivery, all of which lean will help us to accomplish."

Lean implementation also means new requirements and

A look at three Phoenix hobbers with the needed parts and tools in close proximity for faster production standards for Forest City suppliers, and it's Keeley's role to make sure they meet them. Or else.

"Our suppliers will be required to implement at least a minor portion of the lean concepts to remain competitive for Forest City Gear. For instance, lead and set-up times will need to be reduced to allow us to in turn reduce our timelines to our customers. Large orders will need to be shipped in smaller batches or lots to satisfy our customers."

As for Hawkins, another lean true believer, he sees his role as more than being the eyes and ears on the factory floor in lean's implementation.

"I refer to myself as a champion; I have to be the champion that keeps people in charge of lean committees involved. And I need to compliment them on a job well done, especially if they get the low-lying fruit (obvious lean targets) taken care of right away. And we do what we can here to share with them a special lunch, a special meal after hours. And that does help because it shows them that we appreciate everything that they do.

"And last but not least, I share with them that we are not turning back."

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The Skinny on Lean Manufacturing

While universally known as a Japanese "invention" that was popularized by Toyota, lean in fact traces its roots to the work of post-World War II American occupation forces in Japan. Utilizing the War Department's Training Within Industry (TWI) learning and training programs to assist Japan in regaining their industrial footing, the TWI programs included standard work job instruction and process improvement job methods. Instituted simultaneously with these programs were a number of statistics-based methodologies championed by the quality gurus W. Edwards Deming and Joseph M. Juran. As early as the 1950s, these programs and methodologies were at the heart of what is called Japan's kaizen (continuous improvement) revolution.

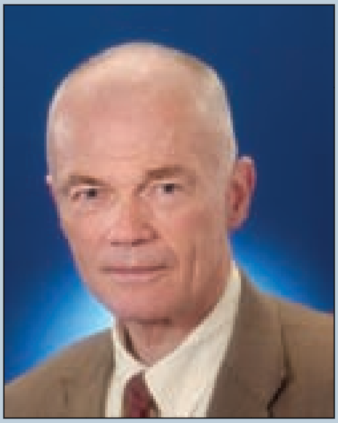
Robert "Doc" Hall is professor emeritus of operation management at Indiana University's Kelley School of Business, a founding member of the Association for Manufacturing Excellence (AME), and has authored six books on lean principles. According to Hall, the term "lean" was unknown until the late 1980s. Even then, there was some overlapping of terms.

"First, JIT was the nickname originally given by Toyota for TPS, and it was used in the 1980s until Womack's book (*The Machine That Changed the World: The Story of Lean Manufacturing*, by James P. Womack, Daniel T. Jones and Daniel Roos) coined the term 'lean,' which still is not a good descriptor," he says. "Lean equals American understanding of Toyota Production System. Six Sigma originated at Motorola as a very techniques-oriented quality program. The other terms represent some companies' attempt to blend everything together into 'their' program, amid much misplaced debate on the differences.

"The truth is that no one understands the thinking from just reading. They have to live it and think about it to begin to 'get it'—a totally different philosophy of work, and even of what a company is. That's why Toyota is reluctant to discuss it with the uninvolved."

Indeed, in any conversation of lean, the need to change a company's culture—from the most senior management to the shop floor—is a bedrock constant. Esprit de corps, unity of purpose, everyone pulling from the same end of the rope—call it what you will, it is an absolute necessity for a company to begin even thinking about lean.

"Culture is the aggregate of all policies and practices that create the milieu of work," says Hall. "It's more than just attitude; it's the consistency—or lack of it—between what is expected of people in improving processes, and the systems-and-reward actually used by the company. Culture today is sometimes referred to as a company's DNA."



Robert "Doc" Hall, a charter member of the AME, author of six books on lean, and a longtime Louis Armstrong fan.

For Hall, rethinking or reinventing what a particular company is goes way beyond a mission statement. It's really all about people. And he believes that is one area where companies contemplating or implementing lean fall short.

"When you say you're going lean, that really is ultimately a change in how you actually think about what the company is, because what you have to do is develop the people, and the company really is the people," he says. "There's a favorite saying, 'Our people are our most important asset.' But people are not an asset; they are the company."

Lastly, Hall warns of impending lean implementation failure if a company's CEO and/or owner is not completely behind the effort. Not just talking the talk, but doing the walking and heavy lifting as well.

"They underestimate the degree to which the company must eventually change and think of it as a miraculous cost-cutting program in which they need be only minimally involved. Sort of, 'While you're up, fetch me a lean program.'"

Bottom line, effective lean implementation is predicated not only upon top management's desire to get lean, but upon its day-to-day understanding of its workings, as well. Hall—by way of jazz great Louis Armstrong—probably puts it best.

"Satchmo Armstrong had a great saying—'Man, if someone has to explain it to you, you don't dig nuthin'."

—***Jack McGuinn, Senior Editor***

June 11–13—AGMA's Gear Manufacturing Technology Course. Lieberr Gear Technology Center, Saline, MI. Geoff Ashcroft and Ron Green present theory as well as practical aspects of gear manufacturing. Materials will be covered on gear theory, gear inspection, gear manufacturing, gear hobbing, gear shaping, hobbing and shaping tools, production estimating and gear shaving. $750 per person includes materials, an AGMA reference manual and certificate of completion. For more information, contact the Gear Consulting Group by telephone at (269) 623-4993.

June 13—ECM Seminar for Heat Treating. Plymouth, MI. One-day technical seminar on new technologies for gas quenching and low-pressure vacuum carburizing; research and development in heat treating; updated mechanical properties and more. The seminar is free and includes meals and a tour of an ECM furnace at a customer plant. For more information, contact ECM by telephone at (262) 605-4810 x 1008 or on the Internet at *www.ecm-usa.com*.

June 14–17—GIMT + AMB 2007. Guangzhou International Convention and Exhibition Center, Guangzhou, China. The GIMT, South China's leading machine tool exhibition, will be combined for the second year in a row with the AMB. The show is jointly organized by Business Media China AG, Messe Stuttgart International and BMC Zhenwei International Exhibitions Co. Ltd. For registration and related additional information, contact GIMB + AMB by e-mail at *rhung520@aol.com* or on the Internet at *www.china-machinetool.com*.

June 19–22—31st Annual Vibration Institute Meeting & Symposium. Menger Hotel, San Antonio, TX. The four-day course includes a presentation on gearbox design and a case study analysis of two-stage planetary gearbox vibration and more. $775 includes all sessions, institute membership for 2007 and all meeting luncheons. For more information, contact the institute by telephone at (210) 223-4361 or online at *www.vibinst.org*.

June 25–28—KISSsoft Calculation Program Seminar. KISSsoft facility, Rapperswill, Switzerland. Dr. Kissling, a member of the Swiss Standard Association, will teach a range of subjects including properties of gears and how to account for them in the sizing and optimization process. The third day will introduce participants to shaft and bearings calculation, other machine elements as well as some basic functions of KISSsoft such as material database and reports. The fourth day is dedicated to programming KISSsys and will teach participants to model entire systems of machine elements such as a gearbox or powertrain. Seminars are conducted in English. For more information, visit the company's website at *www.kisssoft.ch* or contact *info@kisssoft.ch*.

September 4–7—10th International Power Transmission and Gearing Conference. Rio All-Suite Hotel and Casino, Las Vegas, NV. The conference program is separated into ten areas including gear design and analysis; gear strength and durability; gear dynamics and noise; gear diagnostics; gear manufacturing; gear lubrication and efficiency; engineered surfaces and tribology; transmissions; chains, belts and traction drives; couplings, clutches and bearings. Registration fees range from $75–$800. Early registration closes July 20. For more information, contact the ASME on the Internet at *www.asme.org*.

September 12–14—Basic Gear Noise Short Course. Department of Mechanical Engineering, Ohio State University, Columbus, OH. Attendees will be shown how to design gears to minimize the major excitations of gear noise transmission error, dynamic friction forces and shuttling forces. Fundamentals of noise generation and measurement will be covered as well as gear rattle, transmission dynamics and housing acoustics. Course includes extensive demonstrations of computer software for gear analysis in addition to demonstrations of the measurement of transmission error and noise for test rigs at the GearLab. $1,500. For more information, contact the GearLab by telephone at (614) 688-3952 or on the Internet at *www.gearlab.org*.

September 17–18—Advanced Gear Noise Short Course. Department of Mechanical Engineering, Ohio State University, Columbus, OH. The advanced course is designed for individuals who have taken the basic gear noise short course. Topics include computer modeling; transmission error prediction; general system dynamics; bearing/casing dynamics, gear rattle models, experimental approaches, modal analysis of casings, acoustic radiation, advanced signal processing, sound quality analysis and transmission error measurement. A case history workshop takes place on the last day. $1,050 for the advanced course only, $2,350 for both courses. For more information, contact the GearLab by telephone at (614) 688-3952 or on the Internet at *www.gearlab.org*.

September 17–22—EMO Hannover. Hannover Fairgrounds, Hannover, Germany. Featuring products and services for metalworking technology, with special emphasis on machine tools, manufacturing systems, precision tools, automated systems, computer technology, industrial electronics and accessories. Sponsored by the VDW (German Machine Tool Builders Association). 27 euros for a day ticket, 48 euros for a season ticket. For more information, visit the show's website at *www.emo-hannover.de*.

Nord Gear

EXPANDS IN WISCONSIN

Nord Gear added a new gearbox assembly center and a new paint booth to its newly expanded facility in Waunakee, WI.

The large gearbox assembly area allows assembly of the complete line of units, including the large SK12382 unit. According to a company press release, this will allow the company to reduce lead time on large gearboxes.

The assembly area consists of three traditional work cells with a fourth cell consisting of a large swivel press to assemble the largest bevel and clincher units. Each of the cells was designed around the team concept to build the units, which can sometimes take a full day to assemble. Inventory is strategically staged close to each cell, and high-capacity bridge cranes and conveyors complete the large gearbox assembly area.

Once assembled, units are taken to a new paint booth and drying tunnel. Painters prime and then apply a durable, stainless steel solvent-based coating that, once cured, achieves 2H hardness. A new overhead conveyor feeds the paint booth and is designed to handle 6,000-lb. units. The paint booth itself consists of a painting stage, heated drying tunnel and cooling tunnel. Nord expects the new booth, which will be fully operational by mid-2007, will increase painting capacity by 200 units per day.

Kaufer

NAMED SALES DIRECTOR AT UNITED GEAR

Paul Kaufer joined United Gear and Assembly of Hudson, WI, as director of sales and marketing.

According to UGA's press release, he has held previous positions with Toro Corp., 3M and North American Communications Resource.

Northstar Aerospace

ANNOUNCES EXECUTIVE APPOINTMENTS TO ACCELERATE MANUFACTURING RAMP-UP

Harry Schmink

Northstar Aerospace announced several executive management changes in accordance with the company's plans to drastically accelerate its manufacturing.

Harry Schmink was appointed to the newly created position of president—gears and transmissions—and is responsible for the company's Chicago and Anderson, IN, facilities in the U.S., and Canadian facilities at Milton, Cambridge and Windsor, Ontario, Canada. Previously, Schmink was general manager at the Chicago manufacturing plant.

According to Northstar's press release, Schmink's responsibilities will include maximizing company-wide planning and production capabilities associated with the growing orders for the CH-47 Chinook helicopter customers as well as Northstar's lean manufacturing initiative.

Ian Taylor, formerly general manager of Northstar Aerospace in Canada, was named vice president—programs. He will lead the company's face gear initiative as it moves from development into the manufacturing phase as part of a transmission that will be used in the U.S. Army's Block III AH-64D Apache Longbow helicopter modernization program. Taylor will also oversee programs for Rolls-Royce, Bell Helicopter, Textron Canada and others.

The Block III AH-64D Apache modernization program includes the development, testing and qualification of drive system components. The company says it will lead production of 510 transmissions with deliveries beginning in 2011.

The program will be the first aerospace application of the transmission using face gear technology.

Other appointments include Tom Purvin as vice president of sales and marketing for North America and David Fisher as director of business development for North America.

Schafer Gear

ADDS NEW GRINDING EQUIPMENT

Schafer Gear Works invested in new production equipment, including new gear grinding machines in the company's South Bend, IN, and Rockford, IL, facilities.

In Rockford, Schafer has installed the Reishauer RZ400 CNC continuous gear grinding equipment for short or long production runs.

Schafer installed a Star SU 250 G profile and continuous generating grinding machine in its South Bend, IN, facility.

Bipin Doshi, president of Schafer Gear, says "Like many industries, keeping up with the latest advancement in production equipment is very important for us. This equipment will add needed capacity and capability to the company's existing grinding machines."

Eaton

MOVES GEAR MACHINING TO HASTINGS, NB

Eaton Corp. announced plans consolidate some of its business segments, including those involved in machining gears for differentials.

As part of the initative, Eaton's headquarters will move $9 million in equipment from Marshall, MI, to Hastings, NB. The company is consolidating its Hastings operation due in part to its closer proximity with its existing casting facility in Kearney, NB, a manufacturing plant in Reynosa, Mexico, and the gulf ports.

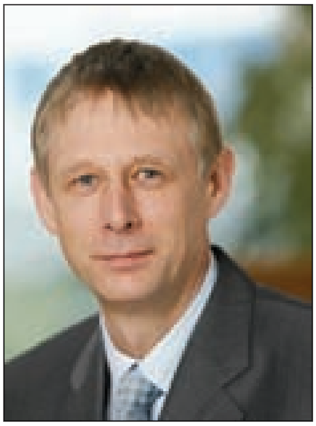
The Hastings plant is part of the Torque Control Products Division, which is part of the Power Train and Specialty Controls Operations Division of Eaton's automotive group. The major market is North American rear drive and four-wheel drive light trucks and SUVs. The plant utilizes four work teams arranged for gears, gear support, assembly and housings.

According to the company's press release, this move will add 30 new full-time jobs to the plant's staff of 95.

The state of Nebraska granted $504,000 in funding to Hastings to help with the move. Of that amount, $500,000 will fund working capital related to the project and $4,000 will be available to the city for administration costs.

Volker Bartelt

NAMED CEO OF PFAFF-SILBERBLAU



Volker Bartelt

Effective April 1, Volker Bartelt started as CEO of Pfaff-Silberblau Group, located in Derching/Friedburg, Germany.

Bartelt will be taking over the position of Martin Kutschka.

According to a company press release, Bartelt worked as a technical director for Oerlikon Geartec AG of Zurich, Switzerland before assuming board-level responsibility for production and logistics at Klingelnberg GmbH of Huckeswagen, Germany. Later, he moved to Körber AG and joined the board of executives for the HAUNI Maschinenbau AG's manufacturing network. As managing director of both Baltic Metaltechnik GmbH and HAUNI Hungaria Kft, Bartelt led four manufacturing plants and 1,100 employees in Germany and Hungary.

In addition, Bartelt also holds German and American patents in electrical and mechanical engineering.

Maag

SELLS MARINE AND TURBO GEAR ACTIVITIES

Maag Gear AG divested its marine and turbo gear business units, effective May 1. Renk AG bought the business unit and plans to continue production of already contracted turbo gears in Winterthur, Switzerland. New marine gear accounts will be executed in Augusburg, Germany.

The new Maag Gear AG plans to reposition itself as a large-scale supplier of gears and services to the cement and minerals industry.

"Maag Gear will therefore become 100% focused on the group's main business—the supply of complete plants, machinery, equipment and services to the cement and mineral industries," says Bjarne Moltke Hansen, group executive vice president of FLSmidth, Maag Gear's Swiss subsidiary.

Bonfiglioli

RELOCATES TO LARGER FACILITY

Bonfiglioli relocated its headquarters to a facility with quadruple the square footage of its previous location.

The new address is:
3541 Hargrave Dr.
Hebron, KY 41048
Phone: (859) 334-3333
Internet: *www.bonfiglioliusa.com*

According to a company press release, Bonfiglioli USA was recently recognized by the Northern Kentucky Chamber of Commerce as one of its "Emerging Thirty" companies. The recognition was for small- or medium-sized companies with strong year-to-year growth as reported in *The Cincinnati Post*.

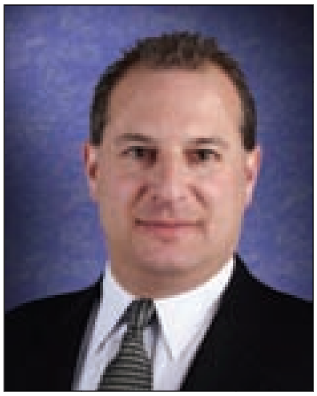
"This is another step towards our goal of becoming a market leader in the American speed reducer market," says Garry O'Neill, industrial solutions sales manager.

Norton Kaplan

APPOINTED HEAD OF METHODS AUTOMATION

Methods Machine Tools has formed a new group—Methods Automation—to assist customers in maximizing performance of the individual machine tools as well as the overall manufacturing process.

The company named Norton Kaplan to lead the new endeavor. Among his new responsibilities will be heading a group that will evaluate the customer's manufacturing process and suggest automation opportunities, including robotic loading/unloading of parts or tools; improved machining methods; integrating/automating processes such as deburring, cleaning, polishing, inspection, marking (for traceability and documentation), assembly, packaging, sealing, labeling and transferring parts between these processes.


Norton Kaplan

Kaplan previously developed robotic automation systems for Cimflex Techknowledge Inc. and designed and marketed systems for Automated Assemblies Inc. According to a company press release, he has more than 28 years' experience in manufacturing automation and advised "Fortune 100" corporations on automated precision assembly and packaging in the production of medical devices, automotive parts, electronics and communication devices.

Westerman Companies

ANNOUNCE PLAN TO SPONSOR KEY MOTORSPORTS

Westerman Companies and Key Motorsports agreed to partner in four NASCAR Craftsman Truck Series races this year, beginning with the July 14 event at Kentucky Speedway.

Westerman will also serve as the primary sponsor of the #40 Key Motorsports Chevrolet Silverado in the scheduled races at the O'Reilly Raceway Park in Indianapolis on July 27, the Nashville Superspeedway in Lebanon, TN, on August 11 and the Martinsville Speedway in Virginia on October 20.

According to a Key Motorsports press release, veteran driver and NCTS champion Mike Bliss will be behind the wheel of the #40 Westerman Companies Chevrolet for the Kentucky and Martinsville races. Team owner Curtis W. Key Sr. will announce his driver selection for the Indianapolis and Nashville events later this season.

Westerman Companies, headquartered in Bremen, OH, manufactures equipment for gears; oil and gas production and storage; industrial tanks; pressure vessels for agriculture,

Terry McGhee, Westerman president

tanks for enriched uranium hexaflouride storage and transportation; and SCADA/Telemetry systems and controls for communication functions.

"We are very pleased to have a champion driver like Mike Bliss driving us in two of the four races in which we will sponsor Key Motorsports," says Terry McGhee of the Westerman Companies. "We believe this is an excellent beginning for us with NASCAR and the Craftsman Truck Series and it will allow us to entertain our customers and excite our employees at the same time."

ZF

EXPANDS IN RUSSIA

ZF and Kamaz announced plans to complete an expansion of a joint assembly plant in Naberezhnye Chelny, Tatarstan (Russian federation) by 2010. Kamaz, a manufacturer of heavy-duty trucks for the Russian Federation, announced plans to completely change over all vehicle models to ZF transmissions. The workforce at the ZF Kama plant in Russia will be increased from its current 60 employees to 700 workers.

ZF Kama launched production operations in January 2006. The current plant produces 9,000 Ecosplit manual truck transmissions annually, both for heavy- and mediumduty commercial vehicles. Kamaz plans to produce 54,000 vehicles this year. The ZF Kama plant currently focuses on assembly operations. Following further expansion, it will also handle manufacturing for the company beginning in mid-2008. According to ZF's press release, approximately 160 million euros will be invested by 2010. Targeted sales volume is expected to reach 250 million euros.

This expansion will require parts supplied from ZF plants in Friedrichshafen, Boutheon (France) and Eger (Hungary). These plants will provide about 25% of the parts required by the joint venture in Tatarstan.

"With this internalization strategy, ZF is taking a decisive step towards market penetration in Russia and, in cooperation with the market leader, has gained an optimal starting position in another growth market," says Wolfgang Vogel, executive vice president of ZF Friedrichshafen AG Commerical Vehicle and Special Driveline Technology Division.

Bison Gear Institutes

SKILLED WORKFORCE INITIATIVE

Bison Gear & Engineering is collaborating with other manufacturing, governmental and educational institutions to remedy the shortage of qualified entry-level workers in manufacturing.

A training program is currently being compiled, consisting of applied math and manufacturing principles. The series of 12-week classes includes topics like shop math; blueprint reading; measuring requirements; use of hand tools; safety skills; employment/life skills; quality improvement; and business success.

Upon completion of the program, students will seek an entry-level position at one of the 10–12 manufacturing organizations supporting the initiative. Students will then be placed in a paid 90-day internship where the focus is on-the-job training. A more specialized Level 2 training will then follow with an additional 72 hours of training in a specific manufacturing area.

According to the company's press release, cost of training for this program is still being researched. Bison is investigating the possibility of establishing grants and funding assistance for tuition costs

Richard Kuhr

JOINS ABA-PGT'S SALES FORCE

Richard Kuhr was hired as a sales and applications engineer at ABA-PGT's Chicago location.

Kuhr has more than 30 years' gear design and manufacturing experience. He has held positions in production control, purchasing, manufacturing engineering and heat treating as a manager, chief engineer, director and vice president.

According to ABA-PGT's press release, Kuhr later moved to the software development and gear design consulting fields. His responsibilities included training engineers in gear design for aerospace throughout the U.S., Canada, Europe and Asia.

In addition, Kuhr has presented numerous technical papers on topics like converting metal gear to plastic designs for cost saving purposes.

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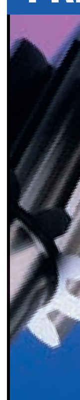

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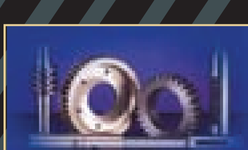

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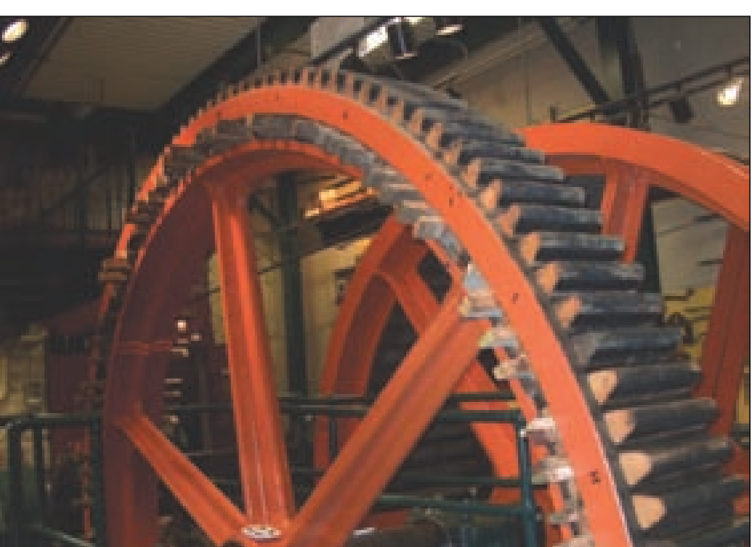

On May 20, the city of Pittsburgh celebrated the 130th anniversary of the Duquesne Incline, a funicular railway that allows passengers to travel via cable car to an observation area and catch a panoromic view of the city and—most importantly—get a bird's eye glimpse of the gear teeth in action.

The Duquesne Incline, at 800' long, 400' high and at a 30° inclination, scales Mt. Washington and zips groups of 25 visitors up the mountain and into downtown Pittsburgh within six minutes. Some visitors love the fact that the Upper Station deposits customers in the heart of Restaurant Row. For three years running, *USA Weekend* has distinguished The Duquesne Incline as the second most scenic spot in the U.S., with the Allegheny and Monongahela Rivers fusing together to create the Ohio River, and the nighttime view of more than 15 lighted bridges illuminating downtown Pittsburgh.

Staff members swear that, for a significant segment of the visitors, it's really the mechanics behind the incline that make every penny of the $1.75 trip worthwhile. Tourists with "gearhead" tendencies may be interested to learn that the incline was steam-driven and remained so until 1932 when it was converted to electricity. In the 1970s, the incline was modernized with Westinghouse receptacles and a maintenance-free drive system.

One of the features that stuns tourists, says Jim Presken, vice president of operations, is the location of the hoisting machinery at right angles with the plane. This method was initially adopted to save the expense of buying an additional piece of real estate. Over the years, it has evolved to two working cables wound together on a single grooved drum.

The single drum is 12' in diameter and 3'10" wide. It has a grooved periphery into which both pulling cables wind. The original cable drum and wooden tooth drive gear are still in use today and operate without any problem. Replacement teeth of aged rock maple are on hand at all times.

Aside from tourists, a large part of the incline's traffic is daily commuters. For those lucky workers, this mechanical masterpiece is part of the daily grind. As for the rest of us, we'll have to make a vacation of it.

For more information:

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