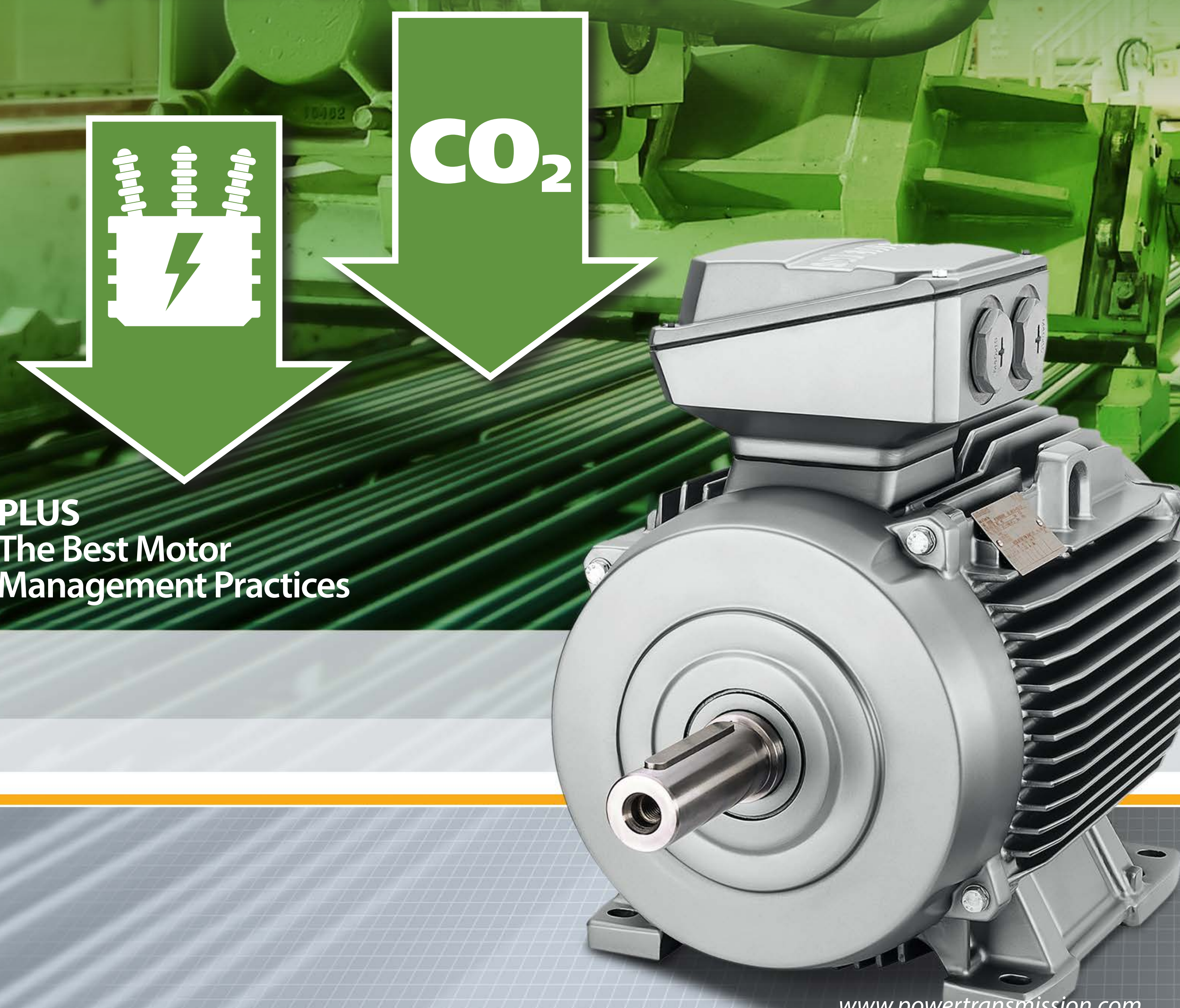
Power Transmission Engineering®

OCTOBER 2015

PREMIUM RUSH

Gearmotor manufacturers upgrade to premium efficiency as impending laws loom



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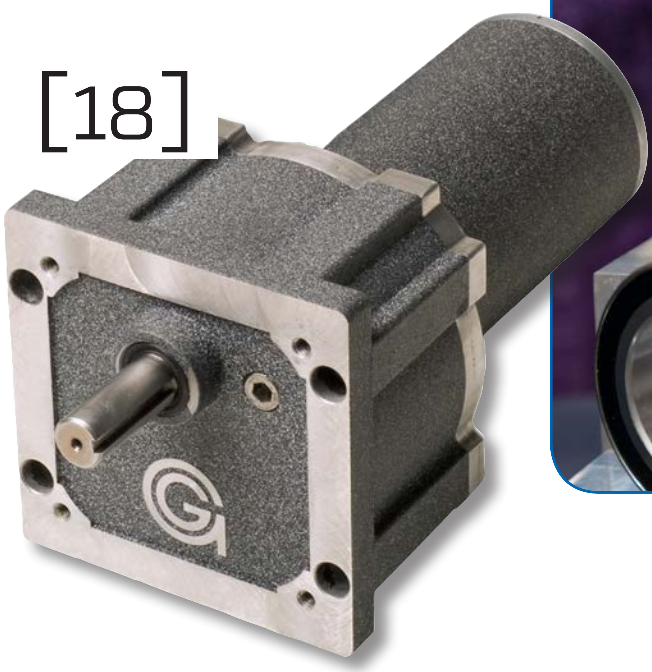

FEATURE ARTICLES

TECHNICAL ARTICLES

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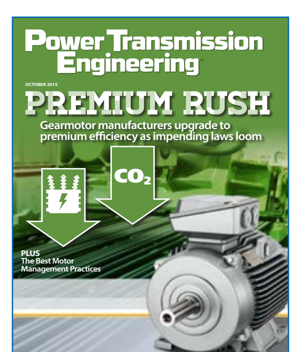
Power Transmission Engineering

VOL. 9, NO. 7



Cover photo courtesy of Siemens

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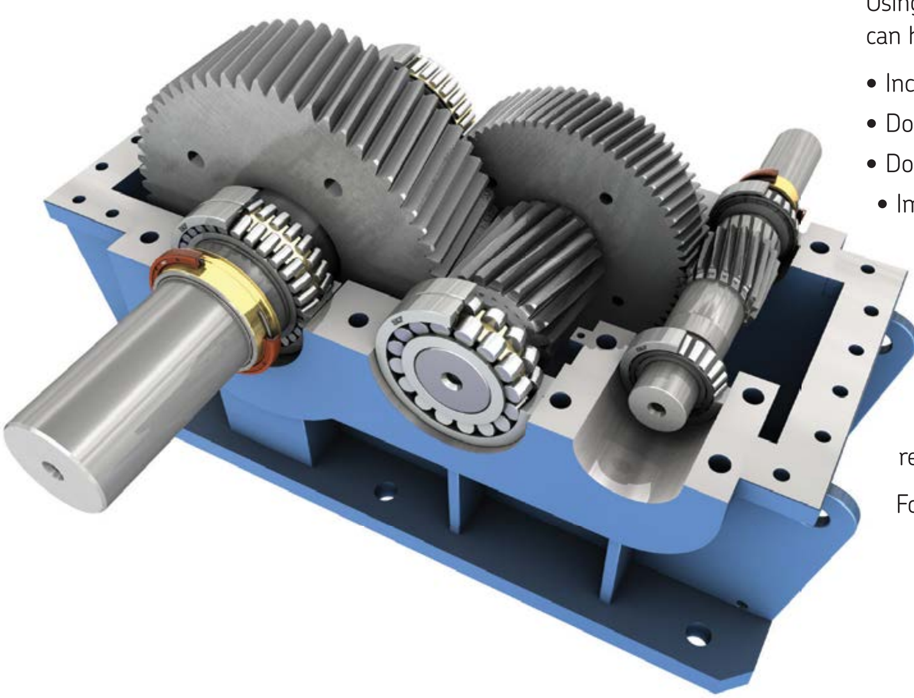

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  - Reduce energy use

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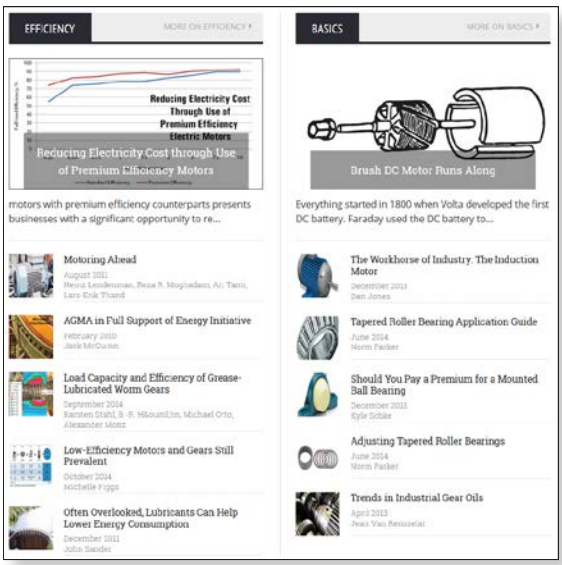
The Bearings Blog – Norm's Back

After a brief hiatus, the bearings blog is back in full swing. Resident blogger Norm Parker offers his insights and instruction for anyone who wants to better understand the process of choosing the right bearing for an application. In a recent post, Norm describes some of the nomenclature and basic principles involved in fitting bearings to shafts and housings.

This Month's Highlighted Topics



Every month we feature two topics from our archives. On the home page you can find a sampling of these key topics along with links to the archive. Stop by *powertransmission.com* to see this month's featured topics:

Efficiency
Basics

Buyers Guide: Don't Be Left Out

The printed version of the *Power Transmission Engineering* Buyers Guide is coming your way in the December issue. If you are a supplier of mechanical power transmission components, now is the time to make sure your company is listed. Visit *www.powertransmission.com/allcos.php* to see if you're already listed (Make sure you scroll all the way to the bottom. The top section shows all of our premium listing advertisers, with descriptions and logos. The bottom section includes all of our free listings. Everyone gets included in the printed buyers guide).

If you don't see your company's name, go to *www.powertransmission.com/getlisted.php* to fill out the form. It's fast, easy and free.

Also, there are many affordable advertising options available in the Buyers Guide. Contact Dave Friedman (*dave@powertransmission.com*) if you're interested.

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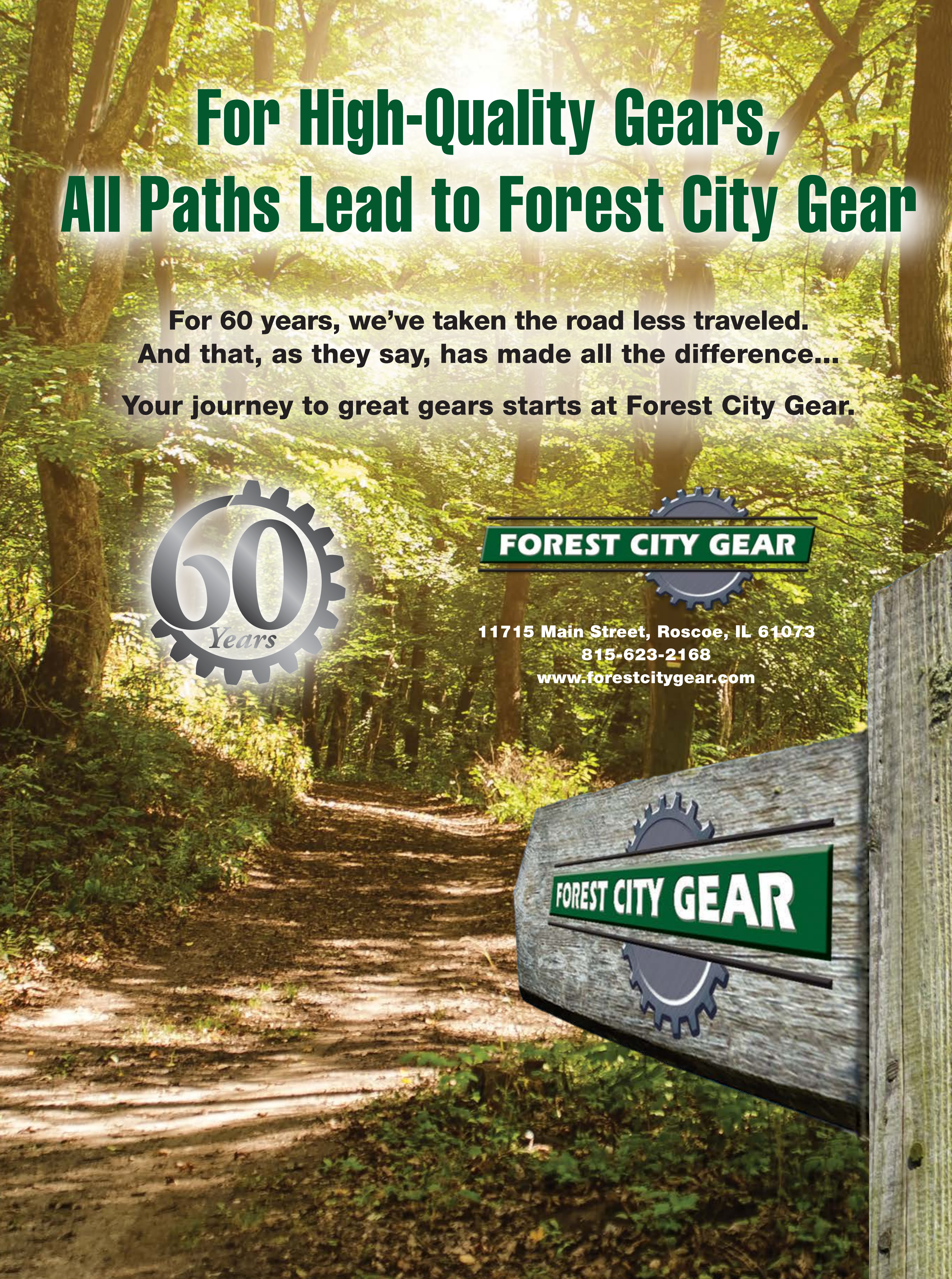
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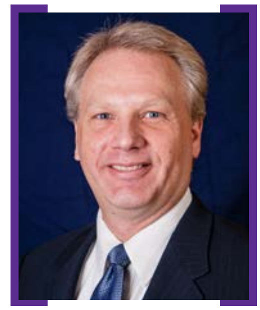
Randy Stott, Managing Editor



Motor Madness

This issue we take a long, hard look at motor efficiency, and you should, too. After all, electric motors used in industrial settings are the single largest consumer of electricity in the United States. Upgrading your electric motors is not only good for the environment, but it's also good for your bottom line. Sure, saving electricity lessens the burden on our country's energy infrastructure. But it also saves you money in the long run.

You're probably aware that there have been many legislative efforts designed to force us to become more efficient. Most of that legislation is aimed at the manufacturers and suppliers of electric motors, who are required to sell only motors meeting certain energy efficiency levels, depending on the type of motor.

It's all very confusing, even to those involved in the industry. Most new electric motors are required to be NEMA Premium efficiency. This is roughly equivalent to the IE3 European designation. But fractional horsepower motors only recently came under these requirements, and even still, there are exceptions, including gearmotors, which are the subject of Assistant Editor Erik Schmidt's article on page 18. One of the last required to upgrade, soon even gearmotor manufacturers will have to conform to NEMA Premium efficiency levels. As you'll see from the article, some of these manufacturers are ahead of schedule and introducing these gearmotors now.

But what does all of this mean for you, the consumer? You're just going to buy the motor that makes most sense for your application, right? Most buyers will choose the least expensive motor that meets or exceeds the specifications. In general, that's true. But if you manage a facility with dozens or hundreds of motors, the decisions and the process become a lot more complicated.

For those facilities, it's important to have a plan. And that's exactly the focus of the article "Motor Management—Best Practices," which begins on page 24. This article is the first in a three-part series provided to us by the Copper Development Association. The series will walk you through the decision-making process for determining when to repair and when to replace. It helps you decide the best options both from an energy efficiency and a cost savings standpoint.

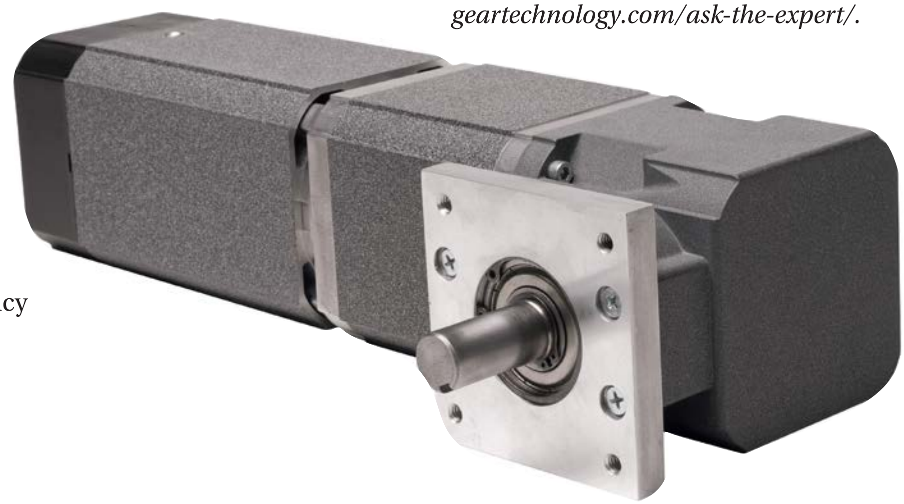
Finally, if you're looking for the latest and greatest in electric motor technology, there's probably no better place to look than the SPS-IPC Drives trade show and conference that takes place November 24-26 in Nuremberg, Germany. News Editor Alex Cannella's article previews the show beginning on page 58.

Of course, if motors aren't your thing, we also have some good articles this issue on custom bearings (page 34), aerospace actuators (page 36) and lubrication (page 48).

As always, we appreciate your feedback. Let us know what you like and don't like, which articles are useful and what topics you'd like to see us cover more frequently. Let us know how we're doing by sending an email to wrs@powertransmission.com.

Randy Stott

P.S. We've just come back from a very successful Gear Expo, and one of the highlights was the Ask the Expert Live presentation in our booth. In four separate sessions, our panels of experts answered gear-related questions with in-depth technical responses. Each session was video recorded and will be available online at *www.geartechnology.com/ask-the-expert/*.



Pittman Motors BGE Series Servo Motor Controllers

OFFER FOUR-QUADRANT DIGITAL SPEED CONTROL FOR DC MOTORS

Pittman Motors recently introduced the BGE series of servo motor controllers, a family of compact four-quadrant positioning motion controllers with integral output stages to control Pittman brushless and brushed DC motors.

The motion controllers can be operated in stand-alone operation with digital or analog IO or as a slave in a CANopen network with device profile DSP 402, protocol DS 301. The family of motion controllers is rated from 12VDC to 60VDC input voltage and 4 amp to 20 amp continuous output current.

The BGE series controllers are suitable for use with Pittman brushless or brushed DC motors. Information about the motor's rotor position can be supplied to the positioning controller by an encoder or integrated Hall sensors contained within a brushless motor. The controls incorporate protection against over-voltage, low voltage and excessive temperature.

If four-quadrant digital speed control is desired, the control can be commanded to run in either direction, stop and hold with torque or stop without torque (coast) through digital inputs. Other inputs can switch between programmed speeds or allow for a variable analog speed reference.

Accel/decel ramps for the motor also can be programmed. The control offers the capability for a motor to function as a stand-alone or programmed servo, which interfaces to the rest of the machine via digital and analog IO.

The BGE series electronic controllers offer different modes of operation to choose from, such as analog or digital torque control, analog or digital speed control and digitally selectable position control (relative, absolute, and modulo). The controls incorporate protection against over-voltage, low voltage and excessive temperature.

Designed for volume OEM applications, BGE series electronic controllers offer a design alternative for restricted space applications where the additional length of an integral control motor may not be feasible. They also can be encapsulated to provide additional protection in extreme environmental conditions.

If only speed control is required for an application, the BGE 3004A, a cost-optimized one-quadrant controller, is available. BGE controllers are available from stock at the Pittman Express E-Commerce store.

For more information:
Phone: (267) 933-2105
www.pittman-motors.com

Stafford Staff-Lok Shaft Collar

COMES WITH COUNTERSINK DRILLED AND TAPPED HOLE, MOUNTING HOLES FOR SENSORS

Stafford Manufacturing Corp. recently introduced a new version of the Staff-Lok hinged shaft collar that incorporates a mounting flat with a countersink drilled and tapped hole and two mounting holes on the face for attaching sensors, cameras, and other devices.

The Staff-Lok shaft collar features an integral hinge with a conformal cam lever and a knurled screw that provides fully adjustable clamping by hand, making it ideal for use as a clamp, stop or spacer. Now offered featuring a mounting flat with a countersink drilled and tapped hole and two mounting holes on the face for attaching sensors, cameras and other devices, this new version is designed for breadboarding mechanical and optical systems.

Providing easy setup and repositioning without tools, Staff-Lok shaft collars are designed for non-rotary applications requiring fast and secure clamping. Machined from steel with a black oxide finish, they are offered in ½" to 2½" I.D. sizes. Custom bore designs and a one-piece version are also available.

The Staff-Lok shaft collar is priced according to version, size, special requirements, and quantity. Price quotations are available upon request.

For more information:
Phone: (800) 695-5551
www.staffordmfg.com

Siemens Sinamics S120 Cabinet Module Drive Packages

NOW COMPLIANT WITH NORTH AMERICAN STANDARDS

Siemens recently announced a new version of its Sinamics S120 Cabinet Module (CM) drive packages compliant with North American standards, with optional UL listing. This module is designed to enable easy configuration of even complex common DC bus line-ups for multi-motor coordinated drive systems as well high horsepower (hp) stand-alone drives for a wide variety of industrial applications.

Pre-designed, fully type-tested modules, including line side components, line infeeds (bus supplies) and motor inverters, all with a broad range of standard options, are selected and configured from a catalog. Compared to the traditional approach of custom-engineered systems, this approach offers a reduction in engineering effort and manufacturing lead times which translates to reduced project costs and a compressed delivery schedule, while minimizing technical and commercial risk for even the most complex drive systems.

Individual cabinet modules have a standardized power and control interface, which allows them to be freely positioned in a line-up that best suits the particular application and makes them easy to install and connect. A range of standard options, such as the DC bus current rating and enclosure type, for example, is available to tailor the line-up to best meet site and environmental conditions. Despite standardization, the design offers a high degree of flexibility for both power and control circuits. For line side converters, there is a choice of non-regenerative Basic Line Module (diode rectifier) or fully regenerative Smart and Active Line Modules. Both of these are IGBT inverters, the Smart Line Module being a more basic six-pulse unit, whereas the Active Line Module offers low harmonics exceeding the demands of IEEE 519, unity or controllable power factor and DC bus voltage control that allows stable operation of motors even on poor power supply systems. Basic and Smart Line Modules can also be configured in 12-, 18- or 24-pulse systems for low harmonic operation. All of these configurations are now compliant with the National Electrical Code (NFPA 70) and Short Circuit Current Ratings per UL508A supplement SB of up to 100 kA.

The Sinamics S120 firmware, combined with Drive-CLiQ (the flexible backplane bus), allows users to assign control units multiple Line and Motor Modules and to mount the control units and associated I/O and sensor modules anywhere within the

line-up or even remotely in a centralized control cabinet or control room. Drive-CLiQ provides automatic electronic nameplate, real-time control data transfer, diagnostic and parameter value gathering and fault tolerant transfer protocols. For external control systems, high-speed industrial Ethernet communications can be parameterized for Ethernet/IP, Profinet or Sinamics Link (peer-to-peer), and programming can be done via a dedicated port or Ethernet TCP on the Ethernet network simultaneously with Profinet or Ethernet/IP.

Sinamics S120 Cabinet Modules were designed to address the need for a complete, ready-to-connect-and-run drive system that enables customers to configure an enclosed drive lineup with a central line infeed (rectifier) and common DC bus supplying power to multiple motor modules (inverters). Typical uses for such systems requiring multi-motor coordinated drive systems include paper machines, steel rolling mills, test stands, cranes, mixers, and oil and gas field equipment. Very high horsepower single drive applications also benefit from this system.

The use of a common DC bus configuration with these new drive packages allows for energy exchange between motors that are motoring and others that are regenerating power back to the AC system, which can save up to 80% of the energy consumed when using standard installations.

For more information:
Phone: (800) 743-6367
www.usa.siemens.com

Mitutoyo KA-200 Counter

UPDATED WITH ABSOLUTE AND INCREMENTAL MODES

Mitutoyo America Corporation recently announced the KA-200 Counter, a multiple feature, intuitive display unit for linear scales. The latest version of the KA-200 Counter has been updated with new features including absolute and incremental modes, non-linear error compensation, function lock, adjustable LED intensity, multiplier function and an option to output directly to spreadsheets using a USB card.

A large, crisp display allows the operator to read the numbers at a glance. In addition, an improved sub-display makes it easy to navigate through parameters. Switch between absolute and incremental modes with the push of a button. Each mode offers 10 presets. A function lock holds the settings and prevents accidental changes of settings. A calculator function allows operators to calculate angles, distances and other measurements.

For more information:
Phone: (630) 978-6483
www.mitutoyo.com

OES Sonata Series

SUPPORTS UP TO THREE AXES OF MOTION IN ANY COMBINATION OF STEPPERS

The Sonata series of motion control systems from Optimal Engineering Systems (OES) is a stand-alone, easy-to-use, plug-and-play, cost effective solution for motion control applications. The systems in this series support up to three axes of motion in any combination of steppers, DC servo and brushless servo motors, and voice coil motors.

Each system includes the power supplies, the motion controller, the micro-stepper and/or servo motor drivers. The operator interface terminal makes the system stand-alone and allows the operator to interact with the motion controller without needing an additional PC. The system can also be operated using an analog joystick or a trackball. The speed of the motor is proportional to the tilt angle of the joystick or the rotational speed of the trackball. Using the 4-line LCD and 32 button keypad, the user can enter motion parameters and select different modes of operation.

Power supply options include: 115 or 230 VAC (standard) and 12 VDC to 80 VDC (Optional). The power supply will be selected to meet the customer's requirements. The available wattages are 80 W, 160 W, 240W, 400 W and 500 W. Options available for these plug-and-play systems include motors, encoders, drivers and cabling. Measuring just 10.0 in. (265 mm) × 10.8 in. (265 mm) × 4.875 in. (124 mm), or in a 19 inch (482.6 mm) rack mount enclosure, the Sonata motion control systems are easy to install.

For more information:
Phone: (888) 777-1826
www.oesincorp.com

Haydon Kerk WGS Integrated Screw/Slide System

DESIGNED FOR STABILITY AND SPEED

Haydon Kerk Motion Solutions recently added the WGS (Wide Guide Screw) to its linear slide product line. Made from the same components used in the RGS Linear Rail Series, the WGS Linear Slide utilizes a screw-driven carriage designed to offer continuous linear speed while maintaining accurate positioning. The length and speed of the WGS are not limited by critical screw speed, allowing high rpm, linear speed and long stroke lengths.

The WGS slide has a compact profile meant to provide improved torsional stiffness and stability versus Haydon Kerk's existing RGS and RGW slide products. An integral mounting base can provide support over the entire length, which can extend up to 8 feet (2.4 meters). Longer lengths are available on a special order basis.

Standard leads include .100", .200", .500" and 1.00" (2.54, 5.08, 12.7 and 25.4 mm) travel per revolution. There are short leads for non-backdriving vertical applications as well as longer leads capable of speeds of more than 60 inches per second (1.5 meters per second).

The WGS utilizes sliding plane bearings on a low-profile aluminum guide rail that keeps the motion smooth throughout the travel distance. The lead-screw is precision made of high-quality stainless steel.

All moving surfaces include Kerkite high-performance polymers running on a Kerkote TFE coating. The slides come with wear-compensating, anti-backlash driven carriages. Additional driven or passive carriages can be added, along with application specific customization. Linear guides without the drive screw also are available.

For more information:
Phone: (800) 243-2715
www.haydonkerk.com

Voith Vorecon Variable Speed Planetary Gear with Dual Torque Converters

IMPROVES NATURAL GAS TRANSPORTATION

Voith's Vorecon recently expanded its footprint in the U.S. market, with five Vorecons equipped in compressor stations transporting natural gas from the Marcellus shale formation into markets in the mid-Atlantic and Southeast states. These Vorecons are increasing transport capacity by over 25 percent, and improving reliability and efficiency in the process. The Vorecon with dual torque converter is a variable speed planetary gear with two matched torque converters. It's built to deliver high efficiency, even into the lower speed range. It is up to two percent more efficient than variable frequency drives (VFD).

"Vorecons with dual torque converters are quickly expanding their presence in markets across the United States," said Jim Kosalek, vice president of Voith Turbo's Power, Oil & Gas Division in North America. "From Marcellus to Eagle Ford, customers are turning to the latest in Voith's cutting edge technology to improve efficiency, reliability, and power in their operating systems. Given the advantages the Vorecon with dual torque converters provides, we expect demand will continue to mirror the tremendous growth of the oil and natural gas sectors in the U.S."

The Vorecon with dual torque converter is marketed for operating points that frequently fall within 60 to 90 percent of maximum speed, and still provides high torque at low speeds – up to 42 percent higher than VFDs. The mean time between failures for the Vorecon is 48 years.

For more information:

Phone: +49 (7321) 37-8497

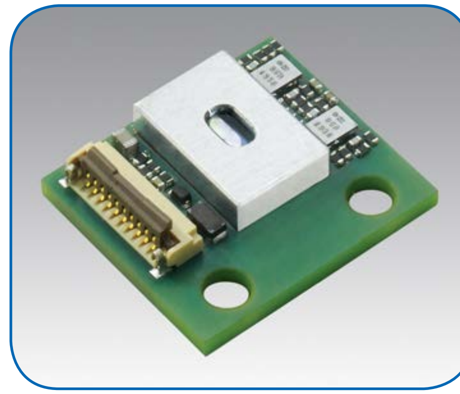
www.voith.com

Celera Motion MicroE Optira Encoders

FEATURE HIGH RESOLUTION AND EASY SETUP

Celera Motion recently introduced its MicroE Optira series encoders. They are the only encoders in their class to provide a resolution of up to 5nm with all automatic gain control (AGC), interpolation, and signal processing carried out in the sensor head. Furthermore, wide alignment tolerances and PurePrecision optical technology make Optira's miniature sensor head easy to setup.

The Optira sensor head comes with two mounting options and a standard FFC connector. In addition, the Optira consumes low power, and a 3.3 VDC version is available for use in precision instruments powered by batteries. The Optira sensor head measures 11.4 × 13 × 3.7 mm.



No additional PCBs, adapters, or dongles are necessary for the full functionality and resolution of the sensors. Optira sensors can also be universally applied with MicroE linear glass scales (to ±1 μm/m accuracy), linear metal tape scales (to ±5 μm/m for Optira), and rotary scales (to ±2 arc-seconds). The interface's options are A-quad-B or 1Vpp sin/cos.

PurePrecision technology has enabled designers of precision motion control systems in medical, semiconductor, advanced industrial and scientific applications to design smaller, lighter and more energy efficient machines and instruments. PurePrecision technology is designed primarily for use in controlled environments that require compact sensor size, a small overall footprint and high accuracy.

For more information:
Phone: (781) 266-5700
www.celeramotion.com

Timken Revolvo Split Cylindrical Roller Bearing

ACCOMMODATES MISALIGNMENT UP TO ±1.5 DEGREES

The Timken Company recently introduced its full Revolvo split cylindrical roller bearing housed unit line to the North American marketplace. Timken added the Revolvo line to its bearing and power transmission product lineup when it acquired the assets of Revolvo Ltd. late last year.

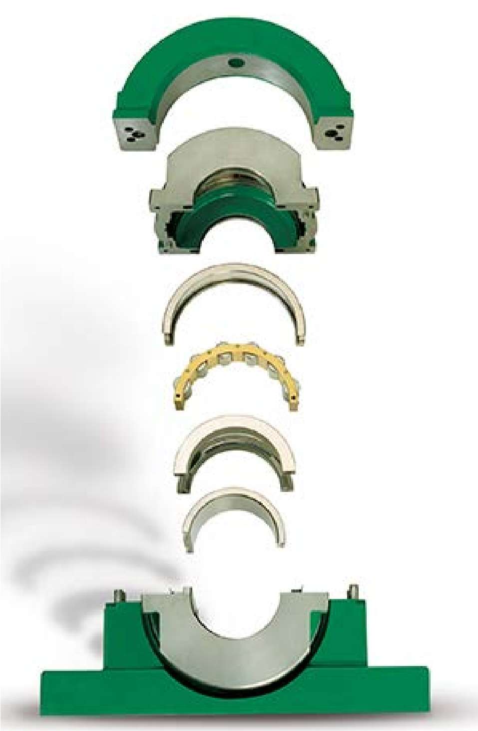
Revolvo split cylindrical roller bearing housed units are used by mining, power generation, food and beverage, pulp and paper, metals, cement, marine and waste-water end users. The units and components are fully interchangeable with most split cylindrical bearing configurations in the market today.

Revolvo housed units can reduce installation time particularly in tight spaces and trapped locations commonly encountered in such applications as fans, conveyors, long shafts, crushers, kiln drives and marine propulsion shafts. The product's design accommodates misalignment up to ±1.5 degrees and is available in a number of housing configurations including special pillow blocks, flanged, take-up and hanger assemblies.

The Revolvo line of split-to-the-shaft cylindrical roller bearing housed units can be installed without requiring access to the shaft ends. The split design allows the bearing to be assembled around the shaft, which reduces downtime because drive components can remain in place during installation or for maintenance. Revolvo units help extend up-time, reduce maintenance costs and typically increase plant efficiency and profitability.

"With the addition of the Revolvo split cylindrical roller bearing housed units, Timken continues to expand its housed unit offering," said Hans Landin, vice president of power transmission and engineering systems for Timken, "and today, we're proud to offer one of the broadest lines of housed units in the industry that can address a wide range of specific customer needs."

For more information:
Phone: (234) 262-3000
www. timken.com



The Doomsday Clock is Ticking for Gearmotor Manufacturers

Within a couple years, all gearmotors will have to be premium efficient — will you be ready?

Erik Schmidt, Assistant Editor

For Groschopp Sales Manager Ed Tullar, the clock is ticking.

Tick.

Tick.

Tick.

The secondhand on the Doomsday dial ominously spins around the face, slowly but ever so surely inching the motor industry towards its inevitable terminus:

Jan. 1, 2017.

The day the IE2 motor will die for good.

"IE2 motors are pretty much dead now," says Kitt Butler, senior program manager for Advanced Energy - Motors & Drives. "IE3 motors were mandated in an EU directive January 1, 2015 and they are equivalent to NEMA Premium which was regulated in 2010 in the US. IE2 are still being used in special applications in Europe and that may be for things like gearmotors. They are to be phased out completely by 2017. But I suspect if you tried to purchase an IE2 motor today on the open market you could not find it."

With another new law — the DOE Electric Motor Rule, which covers electric motors with non-standard end shields or flanges — going into effect on June 1, 2016, all gearmotors over 1 horsepower will soon fall under the sprawling umbrella of legislation pushing for what has been dubbed "premium effiency". *Scampering* is perhaps an imprudent word, but gearmotor manufacturers who have not yet met the IE3 minimum energy performance standards or NEMA Premium are certainly beginning to move with slightly more urgency as Doomsday approaches.

Fractional gearmotors, as fate would have it, are the last unturned stone. And, according to Butler, a law forcing them to upgrade to IE3 or NEMA Premium could be coming "within a couple of years".

"We have to meet the law," Tullar says. "There's a new fractional motor law coming out sometime in the future and we will have to get more efficient. We don't fall under the law yet — but we know we will soon."

Tick.

Tick.

Tick.

Father Time is closing in, and the clock is racing.

Will you be ready when it strikes midnight?

Lightening the Load

There is little doubt that efficiency is one of the buzziest words in the motor industry. The world in its totality is attempting a complete purge of all things wasteful, and with that greener way of thinking has come new legislation encouraging manufacturers to reduce energy consumption.

Picture this: Roughly 30 million new electric motors are sold each year for industrial purposes. Some 300 million motors are in use in industry, infrastructure and large buildings. These electric motors are responsible for 40% of global electricity used to drive pumps, fans, compressors and other mechanical traction equipment, according to Leonard Energy.

Those numbers are certainly big, and the burden that falls on the gearmotor industry to be more efficient remains heavy. So what are companies doing to lighten the load?

"Gearing is one of the most important factors," says Brother Gearmotor Applications Engineer Juan Avalos on how to improve efficiency in a gearmotor. "If you're looking at a helical gear versus a spur gear — those two just have innate differences in terms of efficiency. If you're going to use spur, that's usually more efficient than a helical gear. At the same time, helical has advantages over the spur, such as a smaller size to carry the same amount of load and less noise. So gearing is number one for efficiency.

"That's the gearbox side of it. Then the other side of it is the electrical motor. We're going to be doing the IE3 gearmotor now, so basically it's just a difference in the motor construction. It can handle flux better and the power factor is a lot better. It's going to allow it to draw less actual wattage."

Over at Groschopp, a manufacturer of electric fractional horsepower motors and gearmotors for OEM and distribution products, Tullar and design engineer Scott Lundquist have been working on other ways to increase efficiency.

"We do our best to design for efficiency when looking at our standard products or a customer's specific design," says Lundquist. "One of the first factors that go into that is having the correct sizing of the gearbox. I don't want to get too big or have too big of gears for the application. If the teeth are too thick that may bring the efficiency down. It would have a very high safety factor, but the efficiency may decrease a little bit. Getting that right size of gears in there that is ideal for the application is one thing we look at."

According to Tullar, Groschopp has an extensive testing process it goes through in order to maximize the efficiency of its gearmotors in any given application.

"We have a program where we test all of our motors to run continuously and intermittently, and we check for power as well as thermal capabilities of insulation and everything else. Most motors don't run continuously. We're the only ones that I know of that if you give me a duty cycle I can tell you what the thermal of that motor is.

"Here at Groschopp, we have put in thousands and thousands of hours to calculate what will be the most efficient thing for the application."

Added Lundquist:

"In that case, you're getting the right size motor for the right duty cycle. If we have a duty cycle that is only half a minute and then is off for 10 minutes, but it allows us to size our package down, then that helps efficiency and doesn't

Table 1 Full-Load Efficiencies for NEMA design A, NEMA design B and IEC design N motors (excluding fire pump electric motors) at 60 HZ

Motor Horsepower	2 poles		4 poles		6 poles		8 poles	
	Enclosed	Open	Enclosed	Open	Enclosed	Open	Enclosed	Open
1	77.0	77.0	85.5	85.5	82.5	82.0	75.5	75.5
1.5	84.0	84.0	86.5	86.5	87.5	86.5	78.5	77.0
2	85.5	85.5	86.5	86.5	88.5	87.5	84.0	86.5
3	86.5	85.5	89.5	89.5	89.5	88.5	85.5	87.5
5	88.5	86.5	89.5	89.5	89.5	89.5	86.5	88.5
7.5	89.5	88.5	91.7	91.0	91.0	90.2	86.5	89.5
10	90.2	89.5	91.7	91.7	91.0	91.7	89.5	90.2
15	91.0	90.2	92.4	93.0	91.7	91.7	89.5	90.2
20	91.0	91.0	93.0	93.0	91.7	92.4	90.2	91.0
25	91.7	91.7	93.6	93.6	93.0	93.0	90.2	91.0
30	91.7	91.7	93.6	94.1	93.0	93.6	91.7	91.7
40	92.4	92.4	94.1	94.1	94.1	94.1	91.7	91.7
50	93.0	93.0	94.5	94.5	94.1	94.1	92.4	92.4
60	93.6	93.6	95.0	95.0	94.5	94.5	92.4	93.0
75	93.6	93.6	95.4	95.0	94.5	94.5	93.6	94.1
100	94.1	93.6	95.4	95.4	95.0	95.0	93.6	94.1
125	95.0	94.1	95.4	95.4	95.0	95.0	94.1	94.1
150	95.0	94.1	95.8	95.8	95.8	95.4	94.1	94.1
200	95.4	95.0	96.2	95.8	95.8	95.4	94.5	94.1
250	95.8	95.0	96.2	95.8	95.8	95.8	95.0	95.0
300	95.8	95.4	96.2	95.8	95.8	95.8	--	--
350	95.8	95.4	96.2	95.8	95.8	95.8	--	--
400	95.8	95.8	96.2	95.8	--	--	--	--
450	95.8	96.2	96.2	96.2	--	--	--	--
500	95.8	96.2	96.2	96.2	--	--	--	--

The expanded DOE Electric Motor Rule will go into effect on June 1, 2016. It covers electric motors with non-standard end shields or flanges, which equates to gearmotors.

pull as much current. It's better for the customer in the end because it doesn't cost as much and they're not putting as much power through it either."

A peripheral concern is what lubrication is being used, according to Lundquist.

"It's something we at least consider when we're designing gearboxes," he says. "We can look at the type of lubrication that we use, whether it be oil or grease. We may need to use a type of lubrication for another reason—customer requirements or temperature requirements. But we do pay attention to the viscosity of the lubrication that we use and how it affects efficiency.

"We are aware of how much lubrication is put in the gearbox. We did a design where we basically filled it up with lubrication and it was a very high speed gearmotor. The input speed was close to 10,000 RPM and when we ran our test we had too much heat produced by the gearbox. We ended up having too much oil in the gearbox. We were churning the oil in there and that was increasing heat in there and reducing efficiency.

"These are all things that need to be looked at to increase efficiency."

The Path to Premium Efficiency

For those possessing the complete spectrum of human emotion, tragedy—more so than any other sensation—often sparks change. True enough, we've arrived at this crossroads because of multiple incidents of great distress.

Due to the 1979 oil crisis, the Chernobyl disaster in 1986 and a worldwide need for more power, there has been increased attention paid to energy conservation awareness over the last three decades. Here is a brief timeline of the events:

In 1992 the U.S. Congress, as part of the Energy Policy Act (EPAct) set minimum efficiency levels for electric motors. In 1998 the European Committee of Manufacturers of Electrical Machines and Power systems (CEMEP) issued a voluntary agreement of motor manufacturers on efficiency classification, with three efficiency classes, according to the American Council of Energy-Efficient Economy.

According to a paper produced by Siemens entitled "ABC of Motors", in 1998 CEMEP issued an agreement of motor manufacturers on efficiency classification, with three efficiency classes: Eff 1 for High Efficiency; Eff 2 for Standard Efficiency; and Eff 3 for Low Efficiency.

In June of 2005 the European Union enacted a Directive on establishing a framework for setting Eco-design requirements for all energy using products in the residential, tertiary and industrial sectors, according to the International Electromechanical Commission, according to an article run by ManagEnergy earlier this year. Coherent EU-wide rules for eco-design will ensure that disparities among national regulations do not become obstacles to intra-EU trade, said the article. The directive does not introduce directly binding requirements for specific products, but does define conditions and criteria for setting requirements regarding environmentally relevant product characteristics and allows them to be improved quickly and efficiently. It will be followed by implementing measures which will establish the eco-design requirements. In principle, the Directive applies to all energy using products and covers all energy sources, the article said.

On Dec. 19, 2007, President George W. Bush signed the Energy Independence and Security Act of 2007 (EISA) into law. NEMA participated in crafting major provisions on EISA. One of the most crucial provisions that NEMA focused on was increased motor efficiency levels. The Motor Generator section of NEMA joined forces with the American Council for an Energy Efficient Economy to draft and recommend new motor efficiency regulations covering both general purpose and some categories of definite and special purpose electrical motors, according to NEMA Premium Motors.

On July 22, 2009, Commission Regulation (EC) No 640/2009 implementing Directive 2005/32/EC stated that in the EU, with the exception of some special applications, motors shall not be less efficient than the IE3 efficiency level as of Jan. 1, 2015. The Directive also states all motors (from 0.75 to 375 kW) must be IE3 by Jan. 1, 2017.

And lastly, the Small Motor Rule was passed in 2014. According to an article by Chris Medinger that was published in the December 2014 issue of *Power Transmission Engineering*, the rule covers two-digit NEMA frame single- and three-phase 1/4 through 3 horsepower motors in open enclosures. The mandate is expected to save appoximately seven quads of energy, reduce $41.4 billion in energy costs and cut 395 million metric tons of carbon dioxide.

At face value, this all appears to be an undeniably favorable sequence of events. Tullar, however, isn't yet convinced that premium efficiency is all it's promised to be.

"There's energy efficient washing machines now," he says. "Well, it used to take 20 minutes to wash a load of clothes and now it takes me an hour. I don't know if that's any more efficient. Dishwashers use to run 40 minutes and now they run three hours. Are we really saving the water or the electricity that we thought we were?"

Still, initial projections are promising. Based on U.S. Department of Energy data, it is estimated that the NEMA premium-efficiency motor program would save 5.8 terawatts of electricity and prevent the release of nearly 80 million metric tons of carbon into the atmosphere over the next ten years.

That is equivalent to keeping 16 million cars off the road.

For Tullar and Groschopp, it's not a bridge they have to cross — yet. Fractional gearmotor companies, at this time, still aren't confined to any laws stating that they absolutely must go to IE3 or NEMA Premium. It will be, as the saying goes, the last hat to drop, and Groschopp will be ready when it does.

"We're waiting to see when the new laws come out, but have we checked into the new laminations, building materials and tested all of that stuff? — absolutely," Tullar says. "Do we want to spend the money to buy new laminations right now? — probably not."

Brother, because it manufactures gearmotors over 1 horsepower and falls under the law beginning in mid-

2016, introduced its first IE3 gearmotors at Pack Expo Las Vegas on Sept. 23.

According to Avalos, the new gearmotors offer increased efficiency over standard IE1 models throughout the speed range and initiate less mechanical stress, which directly translates into long-term energy savings. The IE3 gearmotors run cooler and with less noise and, as they are sealed for life and require practically no maintenance, reduce costly downtime. Two of the factors in the enhanced durability of Brother's IE3 gearmotors are the company's long-lasting e-coat paint, and increased lubrication through the use of premium H1 food-grade grease.

Though it's not an entirely apt comparison because they're operating under slightly different circumstances, Brother and Groschopp have still presented us with two different, legitimate options when it comes to upgrading to premium efficiency: be proactive and get ahead of the curve or be cautious and wait it out. What's the line in the sand between the two philosophies? Well, it's cost, of course.

"To increase our cost when we don't have to yet doesn't make sense for us," Tullar says. "We're in a very competitive market. The thing about the 1 horsepower motors and above is that there are a few people that use them. For the fractional, the prices are less and the buyings are higher."

Now, most gearmotor manufacturers will fall somewhere between Brother and Groschopp, and yet there is another, far less traveled path to be taken: IE4.

Yes, while most of the world is still making the transition to IE3 and NEMA Premium low-voltage motors, some companies have decided to join Marty McFly in the future ahead of schedule. IE4 motors, dubbed "Super Premium Efficiency" motors, have actually been on the market for some time, and since 2009 have posted consecutive, double-digit growth rates and almost quadrupled in value in 2013 to reach $114.7 million with nearly 259,000 units shipped, according to a Jan. 20, 2015 article published by *www.automation.com*.

So why have some companies surged ahead to IE4 while others are still spinning their tires at IE2 or below? Once again, cost is the obvious culprit.

"It is possible to go to IE4, but it's more expensive for the company to manufacture," Avalos says. "Therefore, if it's more expensive for the company to make it, it's more expensive for the customer to buy it. There's a big economic factor there. Everyone has to have time to do the proper research and development and weigh benefits against their competitors."

At this point, IE4 is the highest efficiency that has an actual designation. IE5, according to Avalos, is currently just speculative. And after that? Who knows? After all, the clock never stops ticking.

"It's really dependent on the technology," Avalos says of continuously increasing gearmotor efficiency. "To get higher efficiency, you're going to have to do something different. You're going to have to use a new material, you're going to have to change the way the rotors are designed, and you're going to have to make something so there's less losses.

"Especially at our size of motor, the changes could be very minimal — like maybe 1%. If we start getting up into the bigger horsepower motors, it's even less of a change because those are already so efficient. If you have a 200 horsepower motor the change could be less than 1%. Of course, if you want to do that it would be a really big investment and the whole company would have to alter all its motor designs. It will happen eventually, just not today."

Tick.

Tick.

Tick.

You hear that? That's the sound of change.

And it's getting closer and closer. **PTE**

For more information:

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Motor Management — Best Practices

Part I: Creating motor inventory and repair-or-replace guidelines

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Reducing losses and increasing profits by instituting a motor management plan is what this series of articles is all about. Here in Part I, we discuss how to create a motor inventory and establish repair-or-replace motor guidelines. Subsequent topics in this three-part series will address (Part II) motor failure policies and purchasing specifications, and (Part III) repair specifications and preventive and predictive maintenance, respectively.

Why Does My Company Need a Motor Management Plan?

Electric motors tend to be taken for granted, even in technically advanced industries, but they are actually among our least well-managed industrial tools. Complacency breeds ignorance in plant management facilities that depend on motors. Bruce Benkhart, director of Applied Proactive Technologies, Springfield, Massachusetts, has supervised hundreds of motor surveys on behalf of the New York State Energy Research Administration. Even he was surprised to learn that "Most facilities, commercial or industrial, are unaware of their motor population."

Motors that are not properly administered can (and do) cost businesses billions of dollars in annual wasted energy and operating costs. Neglecting motors can also lead to increased downtime and higher maintenance costs — not to mention those moments of near-panic when a critical motor fails and a production line grinds to a halt. Costs spike, profits fall.

You can begin to reduce losses and increase profits by instituting a motor management plan, as outlined in this presentation. Instituting sound motor failure policies and purchasing specifications, as well as writing good repair specifications and preventative and predictive maintenance plans, can further reduce your losses and increase profits. Those will be the subjects as we continue this three-part series on motor inventory management.

What is a Motor Management Plan?

A motor management plan is just what its name implies, i.e. — a set of actions based on a thorough knowledge of a plant's motor inventory that encompasses all aspects of motor ownership. It spells out which types and sizes of motors are optimum for specific operations based on load profile, duty cycles and operating environment; it identifies which motors (or motor classes) should be repaired or replaced when they fail; which ones are critical to manufacturing continuity, and which ones in fact can, to a degree, be taken for granted. The plan can also spell out purchase and repair shop best practices, and cite when and with what model one should replace fully functional motors. It can prioritize maintenance procedures and intervals, and even enable maintenance personnel to predict motor life in order to avoid failure and schedule non-disruptive replacement times.

The Key: Corporate Commitment

A motor management plan is a flexible concept, easily tailored to the size and complexity of an organization and its motor population. There is no one-size-fits-all, but it is important to emphasize that developing and implementing a plan takes serious commitment by top management and training at the shop floor level. Fortunately, there are many excellent (and free!) tools available, several of which are described here. Numerous organizations also provide assistance; utilities and government agencies offer training and hands-on assistance by motor experts.

Selling the concept of motor management to the corporate suite should not be difficult, as the plan offers real dollars-and-cents benefits to all areas of concern, e.g.:

- ***Corporate management will appreciate improved reliability***

and uptime. Good motor management reduces the incidence of motor failures, reducing the fear of that awful day when a failed motor stops a production line and everyone waits while maintenance scrambles for a replacement.

- ***Financial officers will put reduced energy and operating costs at the top of their wish list***. Industrial motor-driven systems are energy gluttons, consuming more than 23% of all electricity produced in the United States. Consumption by motors averages around 66% in manufacturing plants, and in industries like water treatment and mining it can reach 90%. Good motor management can also optimize spares inventory levels, freeing up capital for use elsewhere.
- ***Engineering personnel look at improving performance and efficiency***. The majority of industrial motors now in service in American factories are old. Many of them were installed before 1997, when the Department of Energy (DOE) began mandating increased motor efficiency with the Energy Policy Act of 1992 (EPAct). EPAct became law on October 19, 1992 and became effective on October 19, 1997. NEMA Premium efficiency motors, which cost less to operate and offer higher reliability and longer warranties, are a major element of proper motor management. Their use is now mandated in many applications by the Energy Independence and Security Act of 2007 (EISA) — effective on December 19, 2010.
- ***Purchasing managers see the value in a motor management plan***. A management plan includes a survey of operating motors and of the spares in inventory to provide assurance that NEMA Premium efficiency motors are readily accessible. Faster turnaround is attained. The plan should identify critical motors and ensure that such motors are always available, whether spares in inventory or nearby distributors' warehouse.
- ***Maintenance supervisors and facilities managers see motors from a close perspective on a daily basis***. They appreciate the benefits of programmed maintenance, better predictive capability, simplified replace vs. repair decisions and fewer motor failures, and, when failures do occur, they transform what is often a panic operating mode into a smoothly executed maintenance project.

The Motor Inventory: Where It All Begins

Gilbert A. (Gil) Mc Coy, P.E. is an energy systems engineer with the Washington State University (WSU) Energy Extension Office and a recognized motor expert. For more than 20 years, he and his teams have helped dozens of companies, the military and utilities develop motor management plans — saving untold millions of dollars in the process. He emphasizes that beyond management commitment, the most important element of a good motor management plan is the initial survey — or motor-energy assessment — of all motors in the plant. "Motor energy assessment is a systematic look at motors and motor-driven systems and their uses in an industrial plant," says McCoy. "The object is to identify the energy savings opportunities. The result of the assessment is a motor management plan. That plan not only indicates savings in dollars but also gives the user a path that describes how those savings can be obtained. The first step in a motor assessment is to obtain an in-plant inventory of both in-service and spare motors. Of particular importance is the efficiency class of each motor."

Efficiency is the fundamental driving force behind energy (and cost) savings in any motor management situation. It refers to the amount of shaft or brake power a motor produces, compared to the amount of electrical input power it consumes at any given load level. There are currently three recognized efficiency classes:

1. ***Standard efficiency*** motors are those machines built before 1992. Such motors were neither rated by, nor required to meet, minimum efficiency standards, and their efficiencies are quite low by current standards. Such motors can often be identified in the field by the fact that their nameplates do not display a nominal full-load efficiency value.
2. ***Energy efficient motors*** — also known as ***EPAct motors*** after the Energy Policy Act of 1992 (effective 1997) — mandated modest increases in efficiency. These motors must equal or exceed the minimum full-load efficiency standards given in Table 12-11 of NEMA MG-1-2009/ Revision 1-2010. Nameplates

on these motors list a nominal efficiency value.

3. ***NEMA Premium efficiency motors*** represent next-generation improvement in motor efficiency. The NEMA Premium efficiency standard, which initially covered three-phase, general purpose motors up to 200 hp, was adopted in 2001 and based on earlier recommendations by the Consortium for Energy Efficiency (CEE). Premium efficient motors are several percentage points more efficient than EPAct motors, and conform to Tables 12-12 and 12-13 of NEMA MG-1-2009, Revision 1-2010. Table 12-13 expanded the NEMA Premium definition to include medium-voltage motors in the range of 25-500 hp. The Energy Policy Act of 1992 at first made their use mandatory only in public buildings. The Energy Independence and Security Act of 2007 (EISA) mandated their use in other-than-government applications and extended coverage to several new categories of motors. EISA also extended coverage to higher-horsepower motors — those up to 500hp — but limited mandatory use of NEMA Premium efficiency motors to those smaller than 201 hp.

Manufacturers also offer motors in sizes up to 200 hp that exceed minimum NEMA Premium efficiency levels, but such motors have not received formal recognition as an efficiency class as of the date of this publication. These induction motors achieve their high efficiencies through the use of better technology, higher-quality materials and tighter production controls. One variety utilizes a die-cast copper (rather than aluminum) rotor, whose lower resistivity reduces electrical (I^2R) losses — thus increasing efficiency by one-to-five percentage points. It should be noted that even a one-point rise in efficiency in any class of motors translates into very large savings over a motor's service life. Moreover — since lower electrical losses equate to lower heat gain in the motor, the risk of damage to winding insulation is reduced. Therefore these motors — NEMA Premium in particular — are expected to operate longer before failure. Data supporting that expectation is not available yet, but manufacturers are generally offering longer warranties for these products.

Having sorted out the plant's motors by efficiency category, a performance history must be collected from all relevant nameplate data of all relevant motors, i.e. — those from which energy/cost savings can be realized. This data includes make and model, horsepower rating, synchronous and full-load speed, voltage rating, frame size, enclosure type and special- or definite-purpose requirements for such application as wash-down, severe or inverter duty. A comprehensive motor management plan would also require historical data such as cost, plant application, location and date of installation, plus failure and repair/rewind history. Some companies' motor management plans call for the immediate replacement of "problematic" motors at the outset, based on recurring maintenance requirements.

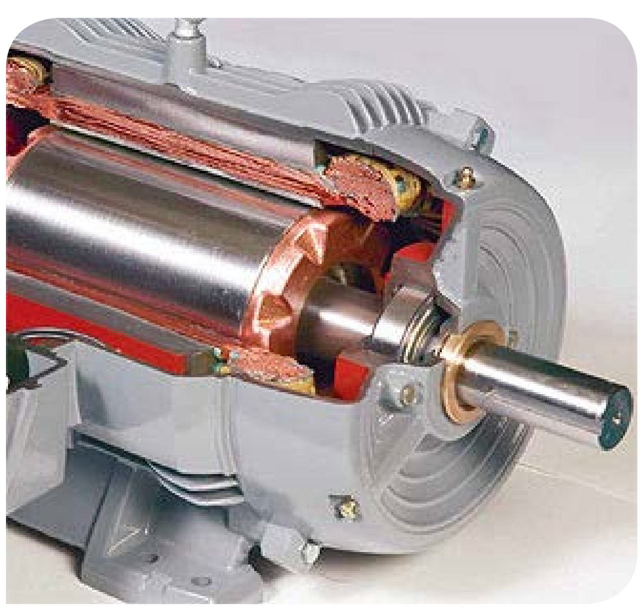


Figure 1 **NEMA Premium Plus motor with die cast copper rotor.**

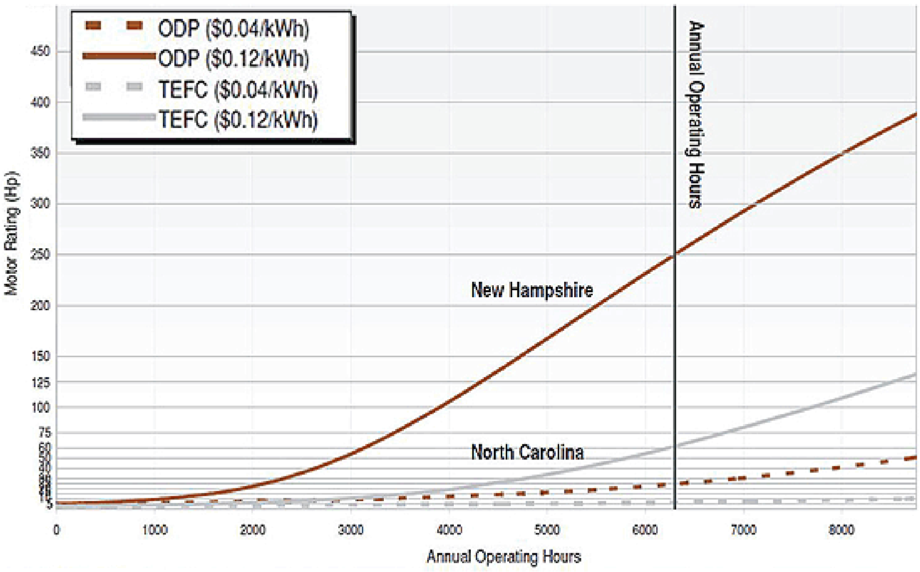


Figure 2 Typical horsepower breakpoint graph showing values for two utility rates (courtesy Advanced Energy Corporation).

Horsepower Breakpoint, *MotorMaster+* and Other Tools

Creating motor inventory requires inspecting the nameplate of every motor in the plant. Or, in the interest of reducing personnel costs, at least assessing the critical motors — to be described below — and then recording the data. Large plants may house several thousand operating motors, and several hundred more as spares — making data-gathering a daunting task. McCoy and other motor experts recommend establishing a screening process to reduce the number of motors that must be assessed.

One such expert is Kitt Butler, director, motors and drives at Advanced Energy Corporation (AEC), Raleigh, North Carolina. AEC was, in fact, the first independent laboratory in the U.S. to gain certification for testing motor efficiency — a service it has offered globally since the early 1990s. Butler recommends a screening process, also developed at AEC, that involves calculating a limit, called a "horsepower breakpoint." The breakpoint defines the motor size for a given duty cycle, at which point it becomes cost-effective to replace an operating motor with a new NEMA Premium efficiency model. Graphically, horsepower breakpoint is the point at which plots of motor rating and annual-operating-hour-data cross for a given utility rate (Fig. 1). It is calculated based on motor size (hp), nameplate efficiency, annual-operating-hours and cost-of-electricity. A breakpoint calculator is available at: *www.advancedenergy.org/md/knowledge_library/resources/ Horsepower%20Bulletin.pdf*.

Making the breakpoint calculation before engaging in the full motor assessment quickly establishes a limit for groups of motors operating under similar circumstances. This eliminates the need to assess all such motors, since the repair/replace decision is already known.

Perhaps the most popular motor assessment tool in current use is *MotorMaster+* software, distributed free of charge to U.S. addresses by the DOE at: *www1.eere.energy.gov/manufacturing/tech_deployment/ software_motormaster.html*. This tool is a data management application with which users can compare the cost of repair with the cost of new replacement for industrial motors under any operating conditions and for any utility rate (Fig. 2). The databases upon which the software relies are updated periodically by the WSU "Extension Energy Program." Databases include nameplate information from thousands of commercially available motors, prices and dealer discounts, plus repair and installation costs. Modules embedded within the software provide the ability to conduct batch analyses and calculate energy and cost savings, project simple payback periods, conduct lifecycle analyses, log maintenance action for both motors and driven equipment, manage spare inventories, verify savings and display aggregate energy and dollar savings from implemented energy efficiency measures at the facility or corporate level. Results can be stored, electronically distributed and/or printed.

Other widely used motor management tools include a spreadsheet-based application offered by AEC — *www.advancedenergy.org/md* — and the 1-2-3 motor analyzer available from CEE's "Motor Decisions Matter" program at *www.motorsmatter.org/index.asp*. Motor manufacturers offer proprietary programs, as well.

Repair or Replace?

After having assembled an inventory of assessed motors, what happens next depends on the need at hand. The data can provide a reference base for future actions. For convenience, each motor can be identified by location, application, and nameplate data to provide reordering information; e.g.:

- Is this a "critical" motor?
- If it is cited "for replacement upon failure," is a spare available in-house?
- Is the spare a new NEMA Premium efficiency model or a repaired stand-by?
- And where is it?
- How soon can the local distributor supply a replacement?
- Should the failed motor be scrapped, or repaired and stored in inventory?
- What periodic maintenance is required and when?
- Has data logging been conducted to reveal a motor's load profile, or have any specialized tests — such as vibration analysis, insulation resistance testing or polarization index tests — been performed on the motor, and what do they suggest?

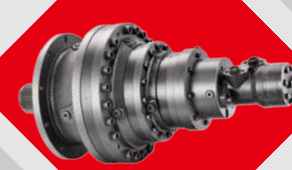
Test results, maintenance actions and field measurements can be appended to the motor's file to provide instant access by facility maintenance staff, millwrights, plant electricians and purchasing department.

But the key question, of course, remains: Which motors should be repaired upon failure, and which ones should be replaced with new NEMA Premium efficiency models?

The HP Breakpoint calculation described above provides a "go/no go" answer to that question. And then *MotorMaster+* does so with considerably more detail — suggesting alternative replacements; spelling out projected energy and cost savings; predicting simple paybacks and performing lifecycle cost analyses; and calculating reductions in greenhouse gas emissions.

As Advanced Energy's motors and drives consultant, Dr. Emmanuel Agamloh, P.E., puts it, "The decision to repair or replace is handled differently by different plants. Some plants settle on a fixed horsepower below which they don't want to consider repair. Other plants use detailed economics of the specific situation, including payback and lifecycle costs. Repair costs are typically lower than the cost of a new motor (for larger motors), but if you factor in the operating costs, there are some differences that you have to take into account. NEMA Premium efficiency motors cost more than EPAct motors, or repairing older

THOTH



motors, but operating costs of old, inefficient motors add up quickly. For motors in the range below 50 hp or so, it probably does not make sense to repair. But as you go higher in motor rating, repair becomes more competitive. Some repair companies, in particular those that are certified under Advanced Energy's Proven Efficiency Verification (PEV) Program, can repair motors and recapture their initial efficiency. Lower-quality repair usually leads to a decrease in efficiency and an increase in operating costs over the life of the repaired motor." For a list of PEV-approved repair shops, see *www.advancedenergy.org/md/consulting/repair_shop_selection.php.*

There are alternatives to setting up a formal *MotorMaster+*-based repair/replace protocol. Some industries have blanket policies of replacing motors when the cost of repair exceeds 60% of the new motor cost, while others immediately scrap failed motors below 50 hp and replace them with NEMA Premium efficiency models. Many large plants systematically discard a failed motor smaller than 60 hp and repair motors above this size. An alternative approach is to conduct a survey of local motor repair costs and compare those costs with discounted new-motor prices. For 20 hp and smaller — 1,800-rpm, TEFC motors — the cost of a new NEMA Premium efficiency motor will generally be less than the cost of repair, according to WSU's McCoy. The numbers change slightly for 900-rpm and 3,600-rpm motors and for ODP (open, drip-proof) motors. The assumption is almost invariably valid for motors smaller than 10 hp; "repair" in this instance implies rewinding the motor and replacing the motor bearings.

A final issue when considering repair/replacement is the size of the replacement motor. For motors that operate below 100% of full-load, there are those who suggest that energy savings may be realized by downsizing to a smaller horsepower of the same efficiency class. Issues of frame size, shaft alignment and over-current protection aside, this action may be technically possible under certain circumstances, but is generally not recommended. Motor efficiency tends to peak at 75-80 percent of full loading, and larger motors may be quite efficient down to 25 percent of rated load. Efficiency increases with horsepower rating; therefore a smaller replacement motor might exhibit lower peak efficiency than a larger motor operating at reduced load. Bottom line — old, inefficient standard efficiency motors that are over-sized and under-loaded could be cost effectively replaced with smaller, more efficient NEMA Premium models, but the analysis must be conducted carefully. It is rarely a good idea to replace a larger motor with a smaller motor of the same efficiency class.

Views from the Operating Floor

CDA spoke with plant operating personnel, purchasing agents, and maintenance staff at a variety of installations. All had undertaken or been provided with a motor inventory-and-assessment within the previous several years, and all had taken actions based upon the results of those assessments. These experienced individuals saw benefits to their motor management program based on the particular circumstances of their respective operations.

Rick Streeter is the purchasing agent for Qubica/AMF — the leading bowling pin manufacturer in the U.S. His plant, in Lowville, New York, turns out between 8,000 – 10,000 bowling pins per day. Electrical energy costs are very high and the company pursues energy savings through everything from improved windows, insulation and lighting to NEMA Premium efficiency motors.

"Our plant is old, and before the inventory we didn't even know what motors we had," Streeter said. "After the inventory, we learned that we had more than 100 really inefficient motors. We started changing them immediately. We looked at 232 motors in total, and we still have a way to go with replacements. We haven't replaced anything with motors that weren't NEMA Premium.

"*MotorMaster*+ works very well for me, because in purchasing I'll have the maintenance guys or supervisors tell me they need a motor on a certain machine, so all I have to do is look it up on *MotorMaster*+ and I know exactly what type it is and where it goes and what to order. That's mainly what I use it for.

"The three things I look for when replacing a motor are cost, the time that it runs, and its efficiency. We look for replacements that give us a payback under two years. I'll only repair special

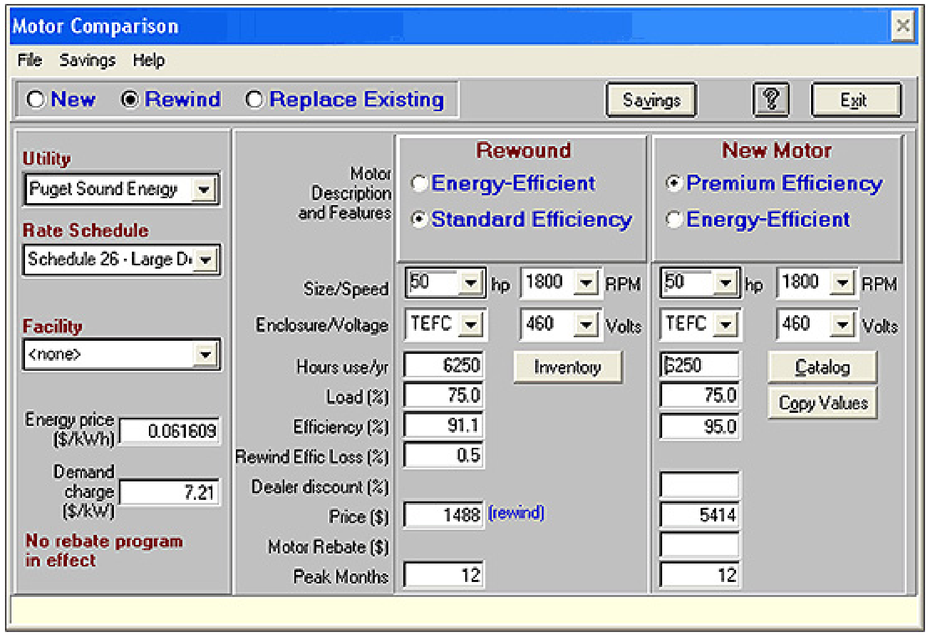


Figure 3 Savings calculation screen from MotorMaster + — the DOE's motor analysis software.

motors. Otherwise, we always go with NEMA Premium because they're more efficient, you have less to worry about, and we see a decrease in downtime."

Joe Anderson is maintenance manager at Interface Solutions, Beaver Falls, New York. The company is the world's largest supplier of automotive gaskets.

"Our motor distributor told us about the motor management program," Anderson said. "He even came in and surveyed our entire motor inventory — including our spares. We have about 150 motors in service, and 125–130 in spares. One of the things we have to do to remain competitive is watch our costs for electricity and steam. We have shift mechanics who work 24/7 and they inspect every motor once a week and inform me when a motor is getting bad. We can change almost any motor within an hour. Keeping our downtime to a minimum, *MotorMaster+* is very helpful. We went after the low-hanging fruit it identified based on 24/7 operation at high loads. There were a few motors that we swapped out with new NEMA Premium replacements that were running fine. When we have a failure, we replace the motor with a NEMA Premium. We also threw out seven or eight stock old motors and replaced them with NEMA Premium. Within the past three years we replaced probably 17–18 motors and keep seven or eight new NEMA Premium motors in stores. We don't rewind." **PTE**

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Acknowledgements. *CDA acknowledges the help of government- and industry-based motor experts in the creation of this Motor Management Best Practices series. Individuals cited in the recent publication include*: **Gilbert A. (Gil) McCoy**, *P.E. Energy Systems Engineer with the Washington State University (WSU) Energy Extension Office, Olympia, Washington. He can be reached at (360) 956-2086, mccoyg@energy.wsu.edu;* **Kitt Butler**, *Director, Motors and Drives at Advanced Energy Corporation (AEC), Raleigh, North Carolina (919) 857-9017, kbutler@advancedenergy.org;* **Emmanuel Agamloh**, *Ph.D., P.E. Motor Systems Engineer Advanced Energy Corporation, Raleigh, North Carolina (919) 857-9023, eagamloh@advancedenergy.org;* **Bruce Benkhart**, *Director, Advanced Proactive Technologies, Springfield, Massachusetts. Benkhart can be reached at (413) 731-6546 bruce@appliedproactive.com.*

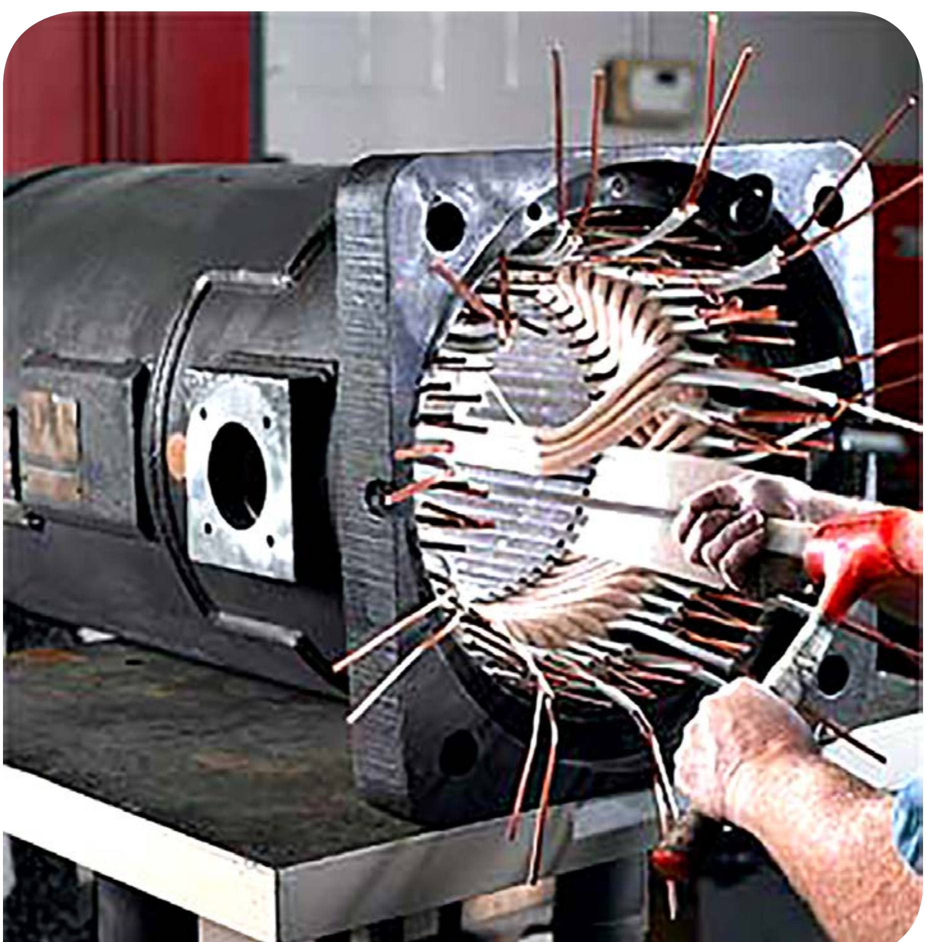


Figure 4 The key question: which motors should be repaired upon failure, and which ones should be replaced with new NEMA Premium Efficiency models?

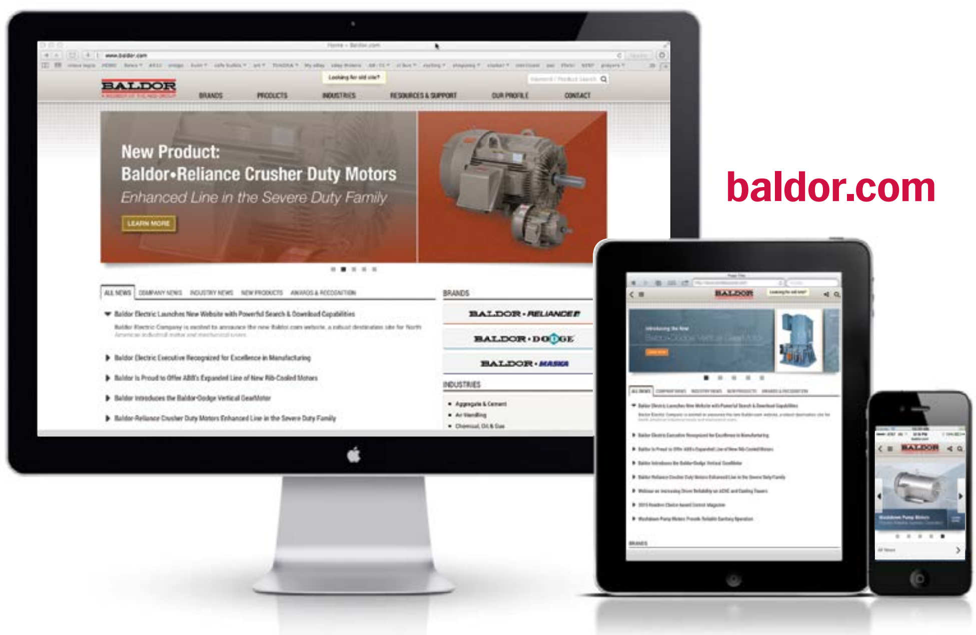
Bruce Benkhart

Kitt Butler

Emmanuel Agamloh

Gilbert A. (Gil) McCoy

The Right Click



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Beyond Bearings Basics

When to consider custom bearings instead of standard bearings

Mark Bos

The bearing selection process is seemingly simple:

1. Select the bearing type based on direction of load in the application;
2. Select the appropriate rolling element based on load and speed requirements;
3. Select the bearing material based on operating environment. Temperature, corrosion, and contamination are the primary considerations.

There are thousands of sizes and types of standard bearings. In many applications, specifying a standard bearing yields an effective result.

But in selecting a standard bearing, can a product designer actually make a costly mistake? In some cases: Yes.

Standard bearings are made to standard sizes, use standard materials, carry a maximum load and operate at a maximum speed for their size. They almost always exceed the functional requirements of an application.

While such a bearing is a workable and safe choice, there are times when a custom bearing might offer decreased cost, increased productivity or greater opportunity for design innovation.

Compare a Standard Bearing to a Custom Bearing

There is a widely held misconception that a standard bearing is cheaper than a custom bearing. A custom bearing may actually reduce costs and allow the designer to create value throughout the production process, from design to assembly to supply chain management.

When can a custom versus standard bearing analysis reduce costs and add value? Suggestions follow.

Reduce Bearing Complexity to Reduce Cost

Does the bearing contain components that don't support functionality?

Standard bearings are made from a full complement of components and are often unnecessarily complex for an application. Many applications do not require all bearing components. For example, a full thrust bearing includes a retainer and two races. Perhaps only the retainer is required. The use of a standard bearing requires the purchase of the full bearing components (unnecessary cost), a larger than necessary housing to ensure that the bearing fits the application (size and cost) and additional elements to protect the bearing (cost).

On the other hand, a custom bearing might allow the designer to simplify the design and contain cost by eliminating unnecessary components. The simplified design can be more compact and cheaper to produce and protect.

Right-size the Bearing to Reduce Cost

Does the bearing's size require a modification to the size or shape of the mounting or mating components?

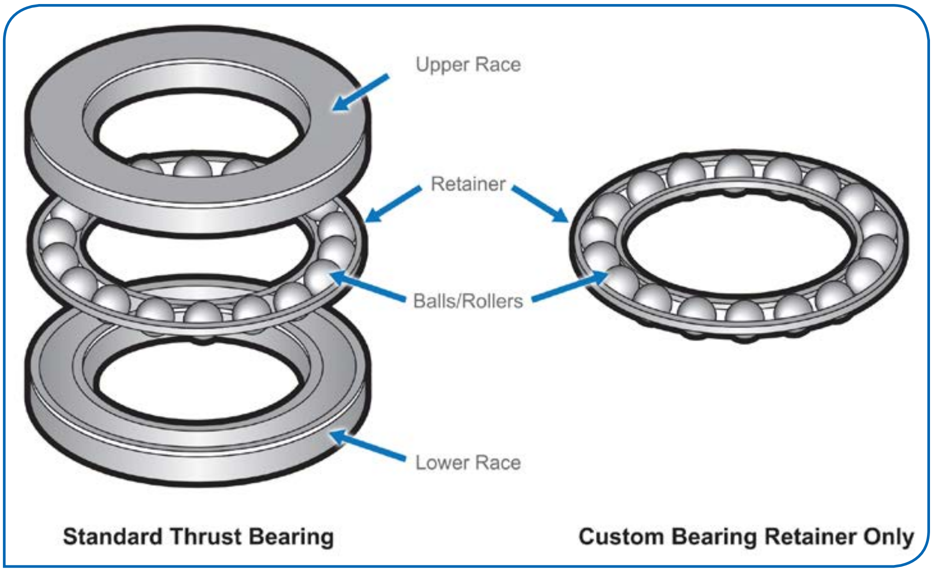
A standard bearing is almost always one size larger than necessary. Consequences of this mismatch in size can be OD, ID or a bore that is a different size than the shaft on which it will turn. If the bore is too big, the size of the shaft must be increased or an adapter sleeve added. If the bore is too small, the shaft must be ground down to fit or re-made to a different size.

It is often easier and cheaper to customize the bearing's dimensions than to bear the increased cost in tooling, processing and materials of more expensive components like shafts, castings or housing.

Choose Proper Materials to Reduce Costs, Streamline Production

Is the standard bearing's material the "right fit" for the operating environment? Will this material contribute to the bearing failing prematurely? Or require expensive efforts to protect the bearing?

Standard bearings are made from a few materials. 52100 bearing steel and 440C stainless steel are the most common. In some environments, the use of a standard bearing material will pre-



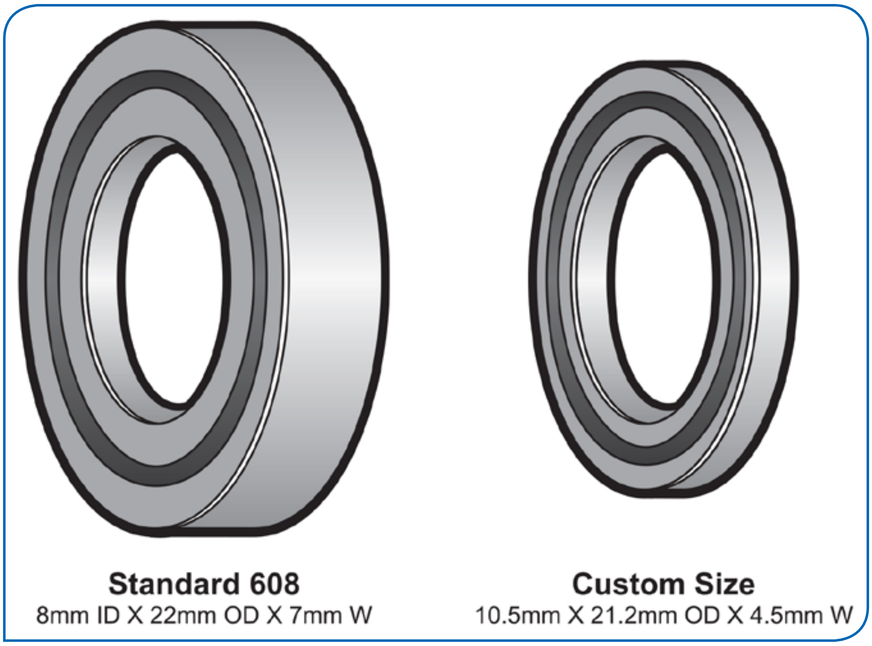
Standard bearings are made of a full complement of components. Custom bearings provide only the components needed for the application.

cipitate premature bearing failure, create unnecessary production expenses or require extraordinary efforts to protect the bearing from its operating environment.

Standard bearings are typically machined, heat treated and ground. Choosing an alternative material allows the bearing to be produced more economically. Example: For jobs that run in large production volumes, the use of stamped washers and races costs a fraction of machined and ground races and washers. 52100 steel and 440C stainless are not readily available in strip steel. Other steels, like high carbon steel, are readily available in strips.

There are instances in which a bearing can be made from a less expensive material and still achieve peak performance. If corrosion resistance is important, for example, and the load on the bearing is light, 300 Series stainless or even plastic may be a better option than the more expensive 440C. Cost is lowered further because heat treating and grinding are not necessary.

A standard bearing may require extraordinary efforts to protect it from operating conditions. A custom bearing, manufactured with a material specific to an operating environment, can eliminate expensive protective measures such as sealed bearing housings, extra seals or layers of grease that will ultimately disappear out of the application or become fouled. Custom bearings can be made from corrosion-resistant materials such as engineered plastics and ceramics. They can also be made to work without lubricants or from materials that can function in extreme temperatures or meet special requirements such as bio-compatibility.



Design the bearing to fit your application, instead of designing your application to fit the bearing.

Enhanced Design Innovation

Does the bearing allow for the incorporation of desirable product features?

With a custom bearing, the product designer can eliminate components while integrating features that improve functionality. For example, adding a threaded stud for mounting a roller or bearing eliminates the need for a shoulder bolt. Installation can be further simplified with a screwdriver slot, hex head or socket head broach. Savings are achieved because the eliminated components do not need to be purchased or inventoried.

And with a custom bearing, aesthetic qualities such as the limitation of movement and audible sounds can easily be integrated into the product design. To add features with a standard bearing, components and cost will be added.

Assembly and Supply Chain Efficiencies

Does the standard bearing provide an opportunity to streamline assembly or supply chain processes?

Subassemblies can have labor- and cost-saving benefits by reducing the number of components or steps in an assembly process. Subassemblies improve productivity and reduce the number of purchase orders, inventory to be stocked, quality inspections, and documentation. A bearing subassembly can include many of the other components that surround the bearing in an application.

Custom bearings can use a custom packaging solution that presents bearings to personnel ready to install and properly sized for the assembly area. In one instance, a simple redesign of the tray resulted in 30% fewer trays, 30% fewer boxes, 30% fewer pallets and a $25,000 per year savings for one OEM.

Closing

By asking a few additional questions about the bearing, the application's design, and of the manufacturing process, the product designer can identify opportunities for value engineering. For the best result, consult a bearing engineer to compare the option of standard vs. custom. **PTE**

Mark Bos is a manufacturing professional with extensive experience in custom bearing and assembly design and manufacturing. In his current position with National Bearings, Bos serves as VP of Business Development, and he is actively involved in product development, engineering, marketing and sales management. Bos has specialized in bearings and bearing component design and manufacturing for the past 18 years.

Space Shuttle Rudder/Speed Brake Actuator

A case study: probabilistic fatigue life and reliability analysis

Fred B. Oswald, Michael Savage and Erwin V. Zaretsky

The U.S. Space Shuttle fleet was originally intended to have a life of 100 flights for each vehicle, lasting over a 10-year period, with minimal scheduled maintenance or inspection. The first space shuttle flight was that of the Space Shuttle Columbia (OV-102), launched April 12, 1981. The disaster that destroyed Columbia occurred on its 28th flight, February 1, 2003, nearly 22 years after its first launch. In order to minimize risk of losing another Space Shuttle, a probabilistic life and reliability analysis was conducted for the Space Shuttle rudder/speed brake actuators to determine the number of flights the actuators could sustain. A life and reliability assessment of the actuator gears was performed in two stages: a contact stress fatigue model and a gear tooth bending fatigue model. For the contact stress analysis, the Lundberg-Palmgren bearing life theory was expanded to include gear-surface pitting for the actuator as a system. The mission spectrum of the Space Shuttle rudder/speed brake actuator was combined into equivalent effective hinge moment loads including an actuator input preload for the contact stress fatigue and tooth bending fatigue models. Gear system reliabilities are reported for both models and their combination. Reliability of the actuator bearings was analyzed separately, based on data provided by the actuator manufacturer. As a result of the analysis, the reliability of one half of a single actuator was calculated to be 98.6 % for 12 flights. Accordingly, each actuator was subsequently limited to 12 flights before removal from service in the Space Shuttle.

Introduction

The U.S. Space Shuttle fleet was originally intended to have a life of 100 flights for each vehicle, lasting over a 10-year period, with minimal scheduled maintenance or inspection. The first space shuttle flight was that of the Space Shuttle Columbia (OV-102), launched April 12, 1981. The disaster that destroyed Columbia occurred on its 28th flight, February 1, 2003, nearly 22 years after its first launch.

The Space Shuttle actuators are lubricated with space-qualified National Lubricating Grease Institute grade 2 perfluoropolyalkyl ether (PFPAE) grease and were intended to operate for life without periodic re-lubrication and maintenance (Refs. 1–2). Many of these actuators were operated without maintenance in excess of 20 years.

During Space Shuttle actuator inspection and refurbishment after the Columbia Space Shuttle disaster, there were external signs of corrosion on one of the Space Shuttle Rudder/Speed Brake (RSB) Actuators. Visible inspection of two partially disassembled RSB actuators from the Space Shuttle Discovery was made on September 16, 2003. Pitted gears and discolored grease were observed inside. The grease-lubricated ball bearings and gears making up the actuators exhibited various degrees of wear. In some cases the wear was severe. Both the bearings and gears operated under a dithering (rotation reversal) motion when these systems were powered during ground operations (Ref. 2).

The observations summarized above led to several research efforts to determine the causes of the grease degradation, damage and wear to estimate the future life and reliability of the shuttle actuators and to investigate design improvements for future heavily-loaded space mechanisms. These works included: Morales et al (Ref. 1), which analyzed the condition of the grease in two actuators after 39 flights; Oswald et al (Ref. 2), which describes a probabilistic analysis of the life and reliability of Body Flap Actuator (BFA) bearings; Krantz et al (Ref. 3); and Krantz and Handschuh (Ref. 4), which examined wear of spur gears lubricated by PFPAE grease; Proctor et al (Ref. 5), which describes experiments to characterize gear scuffing damage observed on one gear of the power drive unit that operates the Rudder/Speed Brake (RSB) Actuators; and Handschuh and Krantz (Ref. 6), which investigated a concern that a tooth might break off from a planet gear — potentially jamming the mechanism.

In order to minimize the risk of losing another Space Shuttle, Oswald et al. (Ref. 2) performed experiments on a test rig under simulated conditions to determine the life and failure mechanism of the grease-lubricated Space Shuttle body flap actuator (BFA) bearings that support the input shaft of the space shuttle body flap actuators. The failure mechanism was wear that can cause loss of bearing preload. The experimental life data was analyzed using the two-parameter Weibull-Johnson method (Refs. 7–8) on experimental life test data from bearings. These tests established life and reliability data for both shuttle flight and ground operation. Test data was used to estimate the failure rate and reliability as a function of the number of shuttle missions flown. The Weibull analysis of the test data for the four actuators on one shuttle — each with a two-bearing shaft assembly — established a reliability level of 96.9 percent for a life of 12 missions. A probabilistic system analysis for four shuttles — each of which has four actuators — predicts a single bearing failure in one actuator of one shuttle after 22 missions (a total of 88 missions for a 4-shuttle fleet). This prediction is comparable with actual shuttle flight history in which a single actuator bearing was found to have failed by wear at 20 missions.

The work of Oswald et al (Ref. 2) was extended to perform a probabilistic life and reliability analysis of the Space Shuttle RSB actuators to determine the number of flights and/or the probability of failure that the actuators could sustain, based on rolling-element (contact) fatigue of the rolling-element bearings and gears and tooth bending fatigue for a minimum of 12 Space Shuttle missions. (The 12-mission life requirement was based on the test data for the Space Shuttle BFA bearings discussed above.) Accordingly, the objectives of the work reported were to 1) determine the life and reliability of each of the gears and bearings in an RSB actuator and 2) extend the analysis to the four RSB actuators on each shuttle vehicle to estimate the system reliability.

Nomenclature

- a Major semi-axis of contact ellipse, m (in.)
- a_1 Life adjustment factor for reliability
- a_2 Life adjustment factor for materials and processing
- a_3 Life adjustment factor for operating conditions including lubrication
- B Gear material constant, N/m 1.979 (lbf/in. 1.979)
- C_D Basic dynamic capacity of a ball or roller bearing, N (lbf)
- C_t Basic dynamic capacity of gear tooth, N (lbf)
- d Diameter of rolling-element, m (in.)
- F_t Normal tooth load, N (lbf)
- f Tooth face width, m (in.)
- f_{cm} Bearing geometry and material coefficient
- i Number of rows of rolling-elements
- J AGMA gear tooth form factor
- K AGMA bending stress adjustment factor
- k Gear tooth stress cycles per input shaft revolution
- L Life, hr, stress cycles, or revolutions
- L_β Characteristic life or life at which 63.2 percent of population fails, hr, stress cycles, or revolutions
- L_{10} 10 percent life or life at which 90 percent of a population survives, hr, stress cycles, or revolutions
- l Length of stressed track, m (in.) or roller length, m (in.)
- m Weibull slope; exponent or gear tooth module, mm
- N Number of gear teeth
- n Life, stress cycles
- P Diametral pitch, 1.0/in.
- P_{eq} Equivalent bearing load, m (in.)
- P_t Normal tooth load, N (lbf)
- p Load-life exponent
- R Reliability for bending fracture, fractional percent
- r Pitch circle radius of gear, m (in.)
- S Probability of survival, fractional percent or material strength MPa (ksi)
- V Sressed volume, m^3 where $V = alz$
- X_n Fraction of time spent at load-speed condition n
- Z Number of rolling-elements per row
- z Depth beneath the surface of maximum orthogonal or maximum shear stress, m (in.)
- α Contact angle, deg
- η_{10t} L_{10} life of single gear tooth, stress cycles
- Λ Lubricant film parameter
- ρ Curvature sum, m^{-1} ($in.^{-1}$)
- σ Gear tooth bending stress
- σ_1, σ_2 Surface roughness of bodies one and two, rms, m (in.)
- τ Maximum orthogonal or maximum shear stress, MPa (ksi)
- ϕ Gear pressure angle, deg

Subscripts

- 1, 2 Bodies one or two; load-life condition one, two, etc.
- B Bearing
- G Gear
- g Gear reliability
- i Individual gear
- n Body n or load-life condition n
- ref Reference
- s, sys System
- t Tooth

Rudder/Speed Brake (RSB) Actuator System

The Space Shuttle RSB Actuator system is contained in the space shuttle orbiter tail section (Fig. 1). The actuator system comprises a power drive unit and four actuators. Each actuator contains two complete gear trains, designated left and right. The system controls the RSB panels through the four actuators that are driven by a single power drive unit. A schematic of an RSB actuator is shown (Fig. 2). The actuators are designed to provide both rudder and speed brake functions by a split design in the vertical tail section (Fig. 1), in which two input shafts enter the actuator. By rotating the shafts in the same direction, the panels are moved concurrently for the rudder function. When the input shafts rotate in opposite directions the panels move apart for the speed brake function; both functions can be used simultaneously.

The 1L (left) rudder/speed brake gearbox shown (Fig. 2) (designated Actuator A in Fig. 1) is composed of a 19.75:1 offset compound spur reduction, followed by a differential planetary with a 24:1 speed ratio; the overall speed ratio is 474:1. The input shafts are shown in the lower right and left corners of the figure. A series of two spur reductions brings the input power to the shaft of a sun gear that drives a nine-planet differential planetary with two ring gears — one of which is fixed. In this arrangement, the fixed ring gear is split into two parts that straddle the output ring gear. The sun gear has a small face width and drives the right-most planet set that meshes with the fixed ring gear; it is placed near the center of the planet spools. The geometric properties of the gears in both the compound spur reduction and the differential planetary reduction are given in Table 1.

In Table 1 the gear face widths were reduced to estimate the gear tooth contact stresses more closely, as the analyses only

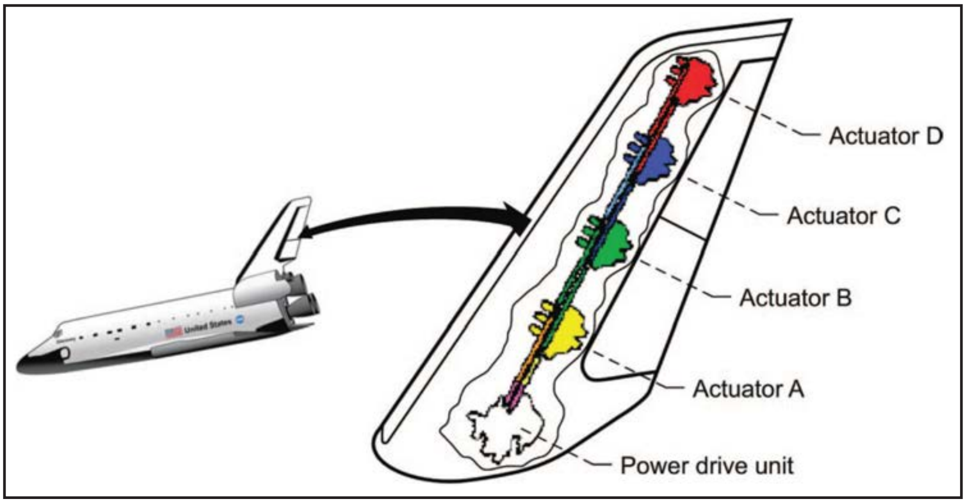


Figure 1 Rudder/speed brake actuating components.

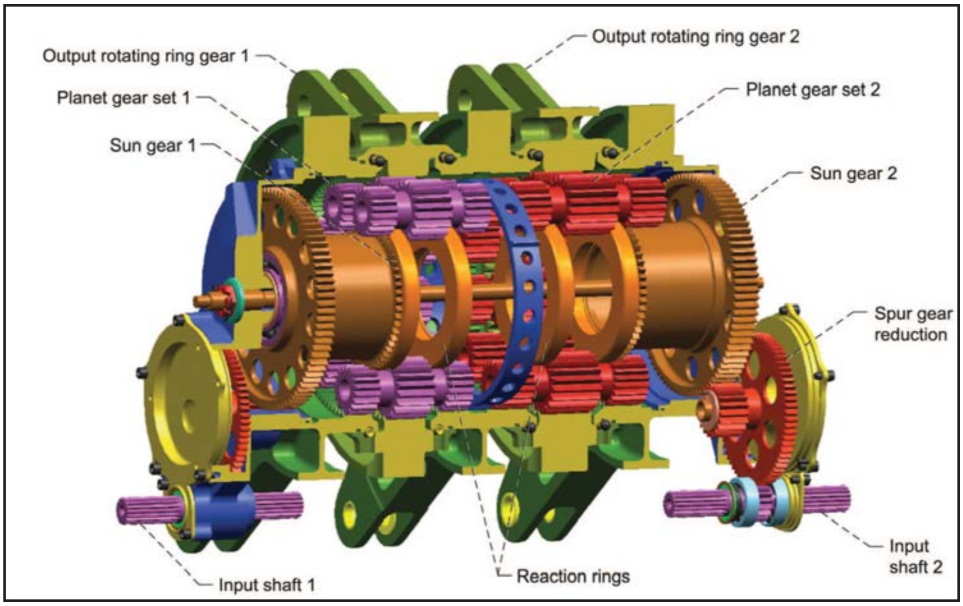


Figure 2 Schematic of space shuttle rudder/speed brake (RSB) actuator.

dealt with a constant stress across the gear face. In the planetary, the actual diametral pitch and pressure angles of the gears differ from the nominal values because the gears operate at non-standard center distances.

The compound spur gear input stage has an offset configuration with a 155° angle between the two gear center-lines connecting its three shafts. The input shaft is supported by two symmetrically placed cylindrical roller bearings. The intermediate shaft is supported by a quill of four one-half-inch-wide rows of 0.125-in.-diameter rollers. This quill is modeled as two half-inch-long roller bearings at its ends, with increased capacity. The sun gear shaft is supported by a single ball bearing in the plane of the spur output gear. An axial thrust bearing is placed between the sun gear and the nearest of the two planet support rings.

The two support rings carry the radial loads applied to the nine planets by the fixed and output ring gears. Finally, there are two ball bearings placed between the output ring and the housing. These bearings carry the radial load associated with the hinge moment of the output arm. They also carry the axial preload applied to the rudder/speed brake actuator by a rod through its center.

The gears comprising the actuators were manufactured from case carburized AISI 9310 steel. Most of the rolling-element bearing rings were manufactured from AISI 9310 steel. Two bearing rings were made from through-hardened AISI 52100 steel. The rolling-elements were made from AISI 52100 steel — except for the bearings that connect the fixed and movable ring gears, which were made from AMS 6491 steel.

Analytic Procedure

For the contact stress analysis, the Lundberg-Palmgren bearing life theory (Ref. 9) was expanded to include gear surface pitting for the actuator as a system. The mission spectrum of the Space Shuttle RSB actuator was combined into equivalent-effective hinge moment loads, including an actuator

Table 1 Gear geometric properties in compound spur reduction and differential planetary reduction

Gear	No. of teeth	Face width,mm (in.)	Nominal tooth size[a], module-mm dia. pitch (1.0/in.)	Nominal pressure angle, deg.	Working Tooth size[a], Module-mm dia. pitch (1.0/in.)	Working Pressure angle, deg.
Input (1)	16	10.85 (0.427)	1.814 (14)	25	1.814 (14)	25.0
Intermediate (2)	69	10.85 (0.427)	1.814 (14)	25	1.814 (14)	25.0
Intermediate (3)	19	23.75 (0935)	2.117 (12)	25	2.117 (12)	25.0
Spur Output (4)	87	23.75 (0935)	2.117 (12)	25	2.117 (12)	25.0
Sun (5)	54	3.18 (0.125)	2.117 (12)	25	2.169 (11.708)	27.839
Planet on Sun (6a)	18	3.18 (0.125)	2.117 (12)	25	2.169 (11.708)	27.839
Planet on fixed ring (6b)	18	52.20 (2.055)	2.117 (12)	25	2.169 (11.708)	27.839
Fixed ring (7)	90	52.20 (2.055)	2.117 (12)	25	2.169 (11.708)	27.839
Output planet (8)	18	37.34 (1.470)	2.54 (10)	25	2.479 (10.245)	21.797
Output ring (9)	81	37.34 (1.470)	2.54 (10)	25	2.479 (10.245)	21.797

a The tooth size standards in metric and in English units are different. The metric module is the pitch diameter divided by the number of teeth. The English Diametral Pitch is the number of teeth divided by the pitch diameter.

input preload for the contact stress fatigue and tooth bending fatigue models.

The reliabilities from the contact stress and tooth bending analyses were calculated for a series of estimated loads and load cycles for 100 missions of the shuttle. The loads considered were the output hinge moments on the gearbox. The analyses were performed for the applied loads and for a combination of the applied loads with a pre-load, due to the 50 lb-in assembly. Once the reliabilities of the separate analyses were obtained for each loading condition they were combined to determine the overall reliability of the gearbox for the analyzed loading conditions and cycle counts. This analysis was performed for the left half of the first actuator, with the assumption that the life and reliability of the right half was identical; life and reliability for the four actuators on each Space Shuttle were assumed equal, as well.

Table 2 Summary of rolling-element bearing lives for rudder speed brake actuator

Bearing no.	1a	1b	2	3a	3b	4	6a	6b
Bearing type	Roller	Roller	Roller	Roller	Roller	Ball	Ball	Ball
L_{10} life (hr)	647,000	647,000	15,823	4,875	4,875	54,973	1,509	1,089
$L_{0.1}$ life (hr)	9,747	9,747	238	73	73	828	23	16
Min life (hr)	34,291	34,291	839	258	258	2,914	80	58

The results were extended to the system of four full actuators on a single shuttle.

The life and reliability assessment of the gears was performed in two stages: a contact stress fatigue model and a gear tooth bending fatigue model. The actuator manufacturer provided L_{10} life data for the actuator bearings, assuming a Weibull slope of 1.11 (Table 2). The analyses did not consider the probable reduction in life and reliability caused by boundary lubrication, wear and grease degradation.

The original intent of the space shuttle design was for a life of 100 flights with minimal maintenance. During the Return to Flight program — post-2003 Columbia disaster — this goal was initially reduced to 20 flights. The reliability analysis for the gear tooth bending, gear tooth surface fatigue and bearing fatigue were calculated separately — along with the total system reliability for 1, 12, 20, and 100 space shuttle flights. The reliability values did not consider other modes of failure, such as wear or lubricant degradation.

Enabling Equations and Analysis

TLIFE*: *transmission life and reliability modeling. The method of probabilistic design was applied to the Space Shuttle RSB left half actuator gearbox number 1L (designated Actuator A in Fig. 1). A fatigue life and reliability assessment of the gearbox was performed in two stages: 1) a contact stress fatigue life and reliability model and 2) a gear tooth bending fatigue life and reliability model. The initial effort to determine the fatigue life and reliability of the 1L rudder/speed brake actuator gearbox was focused on adapting the program *TLIFE* (Ref. 10) to analyze this gearbox. This program performs life and reliability analysis in three stages:

1. Determines the loads and load cycles on the components of the gearbox based on the overall load and speed of the gearbox
2. Determines the lives and reliabilities of the individual components based on these loads and load cycles
3. Combines these lives and reliabilities to determine the life and reliability of the gearbox system

This program did not initially include an analysis for the differential planetary described above. It did have an analysis for the compound spur gear reduction that precedes the differential planetary. The analysis for the differential planetary based on the analysis of the preceding section was added as part of this effort.

The loading for this analysis was obtained from two Shuttle Program office tables. The first three columns of Table 3 are a list of shuttle events, the corresponding rudder/speed brake applied moment, and its duration in minutes. For a flight of 7.604 hr, 7.25 hr are for the ferry, from the landing site to the launch site. In the original table, specific moments and times are given, but no load cycle counts.

The second table, Table 4, lists load ranges for given cycle counts. The total cycle count in this table is 35,297,798. However, 91,500 of these cycles are for reversed-loading (negative moments) at lower moment ranges. These loads are applied to the opposite sides of the teeth and do not add to the positive moment fatigue damage. Removing these cycles from the total gives 35,188,298 positive load cycles for this analysis. Multiplying this total by the percent times of duration gives the estimates of the cycles for each applied moment that are presented in the last column of Table 3. These loads and cycles are the basis of this analysis. The loading condition analyzed represents a median load for the conditions considered. It should be noted that a 57,827 N

Table 3 Event duration

Event	Moment, (N-m) lb-in.	Duration, min	Cycles calculated
Ferry	(1,693) 15,000	420	32,392,514
Ferry	(23,953) 212,000	0.25	19,281
Ascent	(9,039) 80,000	0.2	15,425
Ascent	(5,197) 46,000	1.7	131,113
Ascent	(11,073) 98,000	0.1	7,713
Descent	(15,818) 140,000	33	2,545,126
Descent	(27,116) 240,000	1	77,125

Table 4 Initial applied loading information

Load range, N-m (in.-lb)		Cycles
−50,814 to −39,545	(−450,000 to −350,000)	0
−39,545 to −28,246	(−350,000 to −250,000)	0
−28,246 to −16,948	(−250,000 to −150,000)	2,939
−16,948 to −5,649	(−150,000 to −50,000)	88,561
−5,649 to 5,649	(−50,000 to 50,000)	34,369,892
5,649 to 11,298	(50,000 to 100,000)	20,768
11,298 to 16,948	(100,000 to 150,000)	67,044
16,948 to 22,597	(150,000 to 200,000)	439,135
22,597 to 28,246	(200,000 to 250,000)	291,459
28,246 to 33,895	(250,000 to 300,000)	0
33,895 to 39,545	(300,000 to 350,000)	0
39,545 to 45,194	(350,000 to 400,000)	0
>45,194	(> 400,000)	0
Total cycles		35,297,798

(13,000 lb) axial pre-load is included in the loading on the output ring support ball bearings.

The two-parameter Weibull distribution function was used to model the statistical variations in life and strength for both models. For the contact stress analysis, the Lundberg-Palmgren bearing life theory (Ref. 9) has been expanded to include gear surface pitting and the gearbox as a system (Ref. 11). The mission spectrum of the space shuttle rudder/speed brake unit 1L gearbox has been combined into a single equivalent hinge moment effective load, based on the contact stress fatigue model using the linear damage rule also referred to as the Palmgren-Langer-Miner Rule (Refs. 12–13); this load was also used for the bending fatigue analysis.

Weibull Analysis

In 1939, Weibull (Refs. 16–17) developed a method and equation for statistically evaluating the fracture strength of materials. He also applied the method and equation to fatigue data based upon small sample (population) sizes, where:

$$ln\,ln\left(\frac{1}{S}\right)=m\,ln\left(\frac{L-L_\mu}{L_\beta-L_\mu}\right)\text{ where } 0<L<\infty;\ 0<S<1 \tag{1a}$$

The location parameter L_μ is the time or life at or below which no failures are expected to occur, or there will be 100 percent probability of survival. Equation (1a) is referred to as the three-parameter Weibull equation, relating life and probability of survival. If $L_\mu=0$, the expression is referred to as the two-parameter Weibull equation and is written as follows:

$$ln\,ln\left(\frac{1}{S}\right)=m\,ln\left(\frac{L}{L_\beta}\right)\text{ where } 0<L<\infty;\ 0<S<1 \tag{1b}$$

When plotting the $ln\ ln\ (1/S)$ as the ordinate against the $ln\ L$ as the abscissa, fatigue data are assumed to plot as a straight line (Fig. 3). The ordinate $ln\ ln\ (1/S)$ is graduated in statistical percent of components failed or removed for cause as a function of $ln\ L$, the log of the time or cycles to fail. The tangent of the line is designated the Weibull slope m, which is indicative of the shape of the cumulative distribution or the amount of scatter in the data (Refs. 7 and 15).

The method of using the Weibull distribution function for data analysis to determine component life and reliability was later developed and refined by Johnson (Ref. 17).

Rolling-Element Bearing Life Analysis

Lundberg and Palmgren (Refs. 9 and 18) extended the theoretical work of Weibull (Refs. 16–17) and showed that the probability of survival S could be expressed as a power function of maximum orthogonal shear stress τ, life n, depth of maximum orthogonal shear stress z, and stressed volume V:

$$ln\frac{1}{S}\sim\frac{\tau^c n^m}{z^h}V \tag{2}$$

$$ln\frac{1}{S}\sim\frac{\tau^c n^m a l}{z^{h-1}} \tag{3}$$

By substituting the bearing geometry and the Hertzian contact stresses for a given load into Equation 3, the bearing basic dynamic load capacity C_D can be calculated (Ref. 9). The basic dynamic load capacity C_D is defined as the load that a

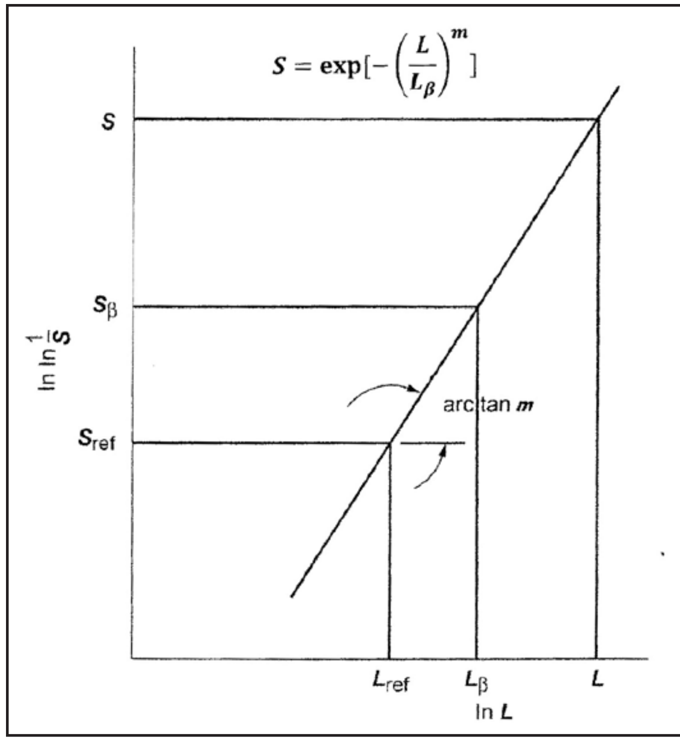


Figure 3 **Weibull plot where (Weibull) Slope of tangent or line is *m*; probability of survival S_β of 36.8 percent at which $L=L_\beta$ or $L/L_\beta=1$.**

bearing can carry for a life of one-million inner-race revolutions with a 90 percent probability of survival (L_{10} life). Lundberg and Palmgren (Ref. 9) obtained the following additional relation:

$$L_{10}\left(\frac{C_D}{P_{eq}}\right)^p \tag{4}$$

where P_{eq} is the equivalent bearing load and p is the load-life exponent (Ref. 9).

Formulas for the basic dynamic load ratings derived by Lundberg and Palmgren (Refs. 9 and 18) and incorporated in the ANSI/ABMA and ISO standards (Refs. 19–21) are as follows:

Radial ball bearings with $d \leq 25.4$ mm:

$$C_D=f_{cm}(i\cos\alpha)^{0.7}Z^{2/3}d^{1.8} \tag{5}$$

Radial ball bearings with $d>25.4$ mm:

$$C_D=3.647f_{cm}(i\cos\alpha)^{0.7}Z^{2/3}d^{1.4} \tag{6}$$

Radial roller bearings:

$$C_D=f_{cm}(i\ell\cos\alpha)^{7/9}Z^{3/4}d^{29/27} \tag{7}$$

Equation 4 can be modified using life factors based on reliability a_1, materials and processing a_2, and operating conditions such as lubrication a_3 (Refs. 22–23) where:

$$L=a_1a_2a_3L_{10} \tag{8}$$

For the boundary lubrication under which the rolling-element bearings in the actuator operate, the lubricant film parameter Λ can be used as an indicator of rolling-element bearing performance and life. For $\Lambda<1$, surface smearing or deformation accompanied by wear will occur on the rolling surfaces, and the factor a_3 in Equation 8 can vary from

Table 5 Rudder speed brake actuator probability of survival

Number of flights	Number of actuators	Gear reliability, percent: Tooth bending fatigue	Tooth surface fatigue	Combined bending and surface	Actuator bearing reliability, percent	Total system reliability, percent
100	1/2	95.943	99.978	95.922	85.950	82.4
100	4	71.793	99.822	71.666	29.782	21.3
20	1/2	99.926	99.999+	99.926	97.495	97.4
20	4	99.409	99.997	99.406	81.632	81.1
12	1/2	99.979	99.999+	99.979	98.571	98.6
12	4	99.835	99.999	99.834	89.126	89.0

0.2–0.5 (Ref. 22). If the effect of boundary lubrication had not been considered by the actuator manufacturer, the bearing lives summarized in Table 2 would be as much as 80 percent less than those shown.

Gear Life Analysis

Between 1975 and 1981, Coy, Townsend and Zaretsky (Refs. 24–26) published a series of papers developing a methodology for calculating the life of spur and helical gears based upon the Lundberg-Palmgren theory and methodology for rolling-element bearings. Coy, Townsend and Zaretsky (Ref. 27) reported that for AISI 9310 spur gears, the Weibull slope m_G is 2.5. Based on Equation 2, for all gears except a planet gear, the gear life can be written as:

$$L_{10_G}=\frac{N^{-1/m_G}\eta_{10_t}}{k} \tag{9}$$

For a planet gear, the life is:

$$L_{10_G}=\frac{N^{-1/e_G}\left(\eta_{10_{t1}}^{-m_G}+\eta_{10_{t2}}^{-m_G}\right)^{-1/m_G}}{k} \tag{10}$$

The L_{10} life of a single gear tooth can be written as:

$$\eta_{10_t}=a_2 a_3\left(\frac{C_t}{F_t}\right)^{p_G} \tag{11}$$

where:

$$C_t=Bf^{0.907}\rho^{-1.165}l^{-0.093.} \tag{12}$$

and:

$$\rho=\left(\frac{1}{r_1}+\frac{L}{r_2}\right)\frac{1}{\sin\varphi} \tag{13}$$

and η_{10t} is the L_{10} life in millions of stress cycles for one particular gear tooth. This number can be determined by using Equation 11, where C_t is the basic load capacity of the gear tooth; F_t is the normal tooth load; p_G is the load-life exponent (usually taken as 4.3 for gears based on experimental data for AISI 9310 steel); and a_2 and a_3 are life adjustment factors similar to that for rolling-element bearings. Life factors a_2 for materials and processing are determined experimentally. The value for C_t can be determined by using Equation 12, where B is a material constant that is based on experimental data and is approximately equal to 1.39×10^8 when calculating C_t in SI units (Newtons and meters) and 21,800 in English units (pounds and inches) for AISI 9310 steel spur gears; f is the tooth width; and ρ is the curvature sum at the start of single-tooth contact.

The L_{10G} life of the gear (all teeth) in millions of output shaft load cycles at which 90 percent will survive can be determined from Equations 9 or 10, where N is the total number of teeth on the gear; m_G is the Weibull slope for the gear, and was taken to be 2.5 (Ref. 28); and k is the number of load (stress) cycles on a gear tooth per output shaft load cycle.

For all gears except the planet gears, each tooth will see load on only one side of its face for a given direction of input shaft rotation. However, each tooth on a planet gear will see contact on both sides of its face for a given direction of input shaft rotation. One side of its face will contact a tooth on the sun gear, and the other side of its face will contact a tooth on the ring gear. Equation 10 takes this into account, where η_{10t1} is the L_{10} life in millions of stress cycles of a planet tooth meshing with the sun gear, and η_{10t2} is the L_{10} life in millions of stress cycles of a planet tooth meshing with the ring gear.

Equations 9–13 are for gears in a single mesh only. For the case of collector gears such as parallel reduction, planetary gear trains and idler gears, the damage accumulates differently. As the load count will be different for these gears, the equations must be modified to account for this variable loading. In planets and idler gears, each tooth is loaded on both faces in one rotation. Since the surface fatigue damage accumulates separately on the faces, the gear faces are treated as separate gears in their own mesh in this simulation. The load count factor l_c is used in other cases; this has units of load-cycles-per-output-load-cycle.

For gears, this analysis used values for the Weibull slope — $m=2.5$; gearing load-life exponent — $p=4$; and dynamic capacity surface strength S_{ac}, 500 ksi. Based on a mission duration of 7.604 hr, the reliability (probability of survival) of the gears based on contact (surface) fatigue is given in Table 5 for 12, 20, and 100 shuttle flights.

Gear Bending Life Analysis

Gear tooth bending stresses are calculated using the AGMA bending stress adaptation of the Lewis bending stress calculation (Ref. 29). This adaptation included the Dolan and Broghammer stress concentration factor (Ref. 30). The maximum bending stress on each tooth is used for the life and reliability calculations. For both pinion and gear, this stress, σ, is the bending stress at the root of the tooth caused by the full mesh load F_t, applied at the highest point of single tooth contact on that tooth.

$$\sigma=\frac{F_t}{fmJ}K(MPa)=\frac{F_tP}{fJ}K(\text{ksi}) \tag{14}$$

The first part of the equation is the metric form, with the tooth size in Equation 14 defined by the gear tooth module m. The second part of the equation is the English version with the tooth size defined by the diametral pitch P. The symbol J is the AGMA tooth form factor and the symbol K is for the

AGMA stress adjustment factors.

This stress is compared to the maximum allowable corresponding fatigue strength, S, in MPa (ksi) for AISI 9310 carburized gear steel (Ref. 31). Using a Goodman diagram, one can determine the corresponding alternate strength S_e, where the ultimate strength S_u, is 1,889 MPa (274 ksi) where:

$$\frac{1}{S} = \frac{0.5}{S_u} + \frac{0.5}{S_e} \tag{15}$$

It is very unusual to have the full alternating strength available in an in-service device, and since there is the presence of a corrosive environment after a period of time, this alternating strength will be de-rated to 80 percent of its full value. The reciprocal of Equation 15, including this de-rating factor, gives the zero to maximum fatigue strength:

$$S = \frac{2}{\frac{1}{S_u} + \frac{1}{0.8\,S_e}} = \frac{2}{\frac{1}{1{,}889} + \frac{1}{752}} = 1{,}076\,\text{MPa} \tag{16a}$$

$$S = \frac{2}{\frac{1}{S_u} + \frac{1}{0.8\,S_e}} = \frac{2}{\frac{1}{274} + \frac{1}{109}} = 150\,\text{ksi} \tag{16b}$$

Using Equations 14–16, a bending stress analysis of the gears can be performed to determine their life and reliability in terms of the maximum fatigue strength S, load-life factor p and Weibull slope m.

The bending stress is directly proportional to the load. Accordingly,

$$L \sim (1/P_t)^p \sim (1/S)^p \tag{17a}$$

The load-life factor can be determined from the slope of the S-N curve in the region between one thousand cycles and one million cycles to failure. For AISI 9310 carburized gear steel, $S = 1{,}076$ MPa (156 ksi). Due to insufficient statistical tooth bending stress life data, a value of $m = 2.5$ is estimated for the bending stress life Weibull slope. This is the same Weibull slope used for the gear tooth contact stress life slope. It is higher than that used for the bearing contact stress life slope. With the 90 percent reliability fatigue strength equal to 0.9 S_u at 103 cycles and the endurance strength equal to S at 10^6 cycles, the load-life factor p becomes:

$$p = \frac{\ln L_2 - \ln L_1}{\ln S_2 - \ln S_1} = \frac{\ln 10^6 - \ln 10^3}{\ln(0.9\,S_u) - \ln S} = \frac{\ln 10^6 - \ln 10^3}{\ln(0.9(1889)) - \ln(1076)} = 11.6 \tag{17b}$$

The maximum bending stress on each tooth is used for the life and reliability calculations. For both the pinion and this stress is the bending stress at the root of the tooth caused by the full mesh load at the highest point of single tooth contact on that tooth. The load-life relationship for bending fatigue is:

$$L_{10,g}\left(\frac{S}{\sigma}\right)^p \tag{18}$$

where, $L1_{0,g}$ is the 90 percent reliability life of the gear for the applied stress, σ. In terms of this life, the gear reliability R_g for a given life, L, is given by:

$$\ln\left(\frac{1}{R_g}\right) = \ln\left(\frac{1}{0.9}\right)\left(\frac{L}{L_{10,g}}\right)^m \tag{19}$$

With the life and reliability of the gears in a single mesh established, the next step is to combine the analyses for all the meshes. For this analysis, the system bending reliability is the product of all the individual bending reliabilities:

$$R_S = \prod R_i \tag{20}$$

The pinion torque for each gear mesh is determined as a ratio to the output hinge moment on the rudder/speed brake. The number of teeth in engagement in each mesh for a single output ring gear tooth engagement is counted.

The load cycles throughout the gearbox for every output ring motion of one tooth engagement, which equals 1⁄81 of a revolution or 4.444°, counted. Since there are nine planets, the fixed and movable ring gears each see nine tooth load cycles. Although each of the nine planets that mesh with the ring gears see only one load cycle, collectively they see nine. Each planet-sun mesh sees four load cycles for each planet-

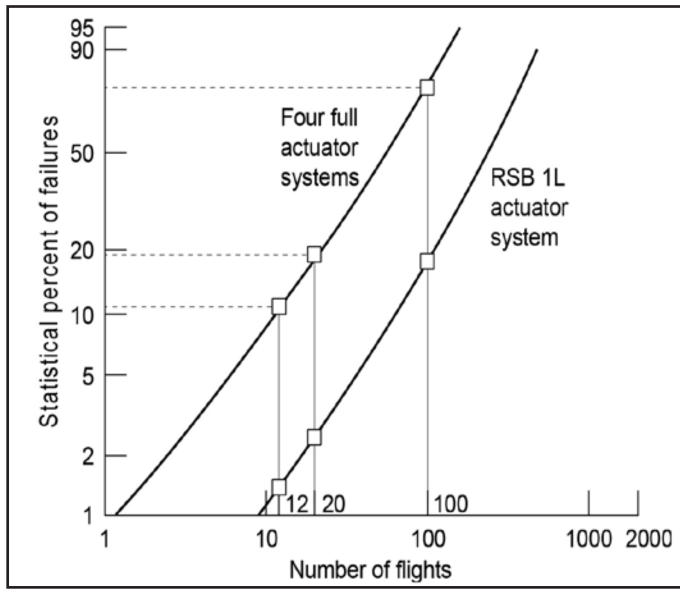


Figure 5 System Weibull plot for 1L actuator and four full actuators.

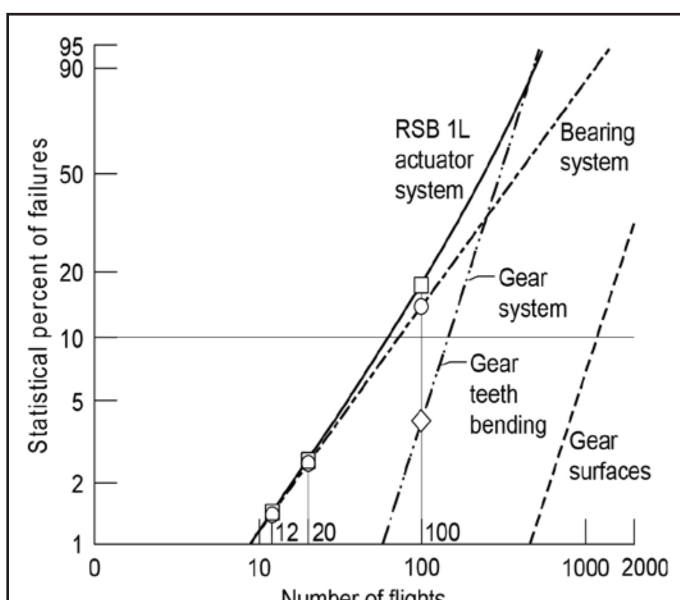


Figure 4 Rudder speed brake 1L actuator component and system Weibull plots.

ring cycle (four teeth mesh at the sun for each tooth at the ring), thus the count for the sun with its planets is 36.

The two meshes in the compound spur reduction are a little different. The spur-sun mesh sees 26 load cycles. However, the intermediate gear that meshes with this gear has only 19 teeth. Thus seven of its teeth see two load cycles and 12 teeth see only one load cycle. The input mesh sees the most load cycles, with 14 of the teeth on the input spur seeing six load cycles for the one tooth rotation of the output.

The overall reliability for all the meshes taken together is the product of all the calculated reliabilities in the analysis. By calculating this analysis with different loads for one-million cycles, the dynamic capacity of the gearbox in bending is determined. It is the load that produces a reliability of 90 percent for one million stress cycles.

System Life Prediction

The L_{10} lives of the individual bearings and gears that make up a rotating machine are calculated for each condition of their operating profiles. For each component, the resulting lives from each of the operating conditions are combined using the linear damage (Palmgren-Langer-Miner) rule (Refs. 12–15) where:

$$\frac{1}{L}=\frac{X_1}{L_1}+\frac{X_2}{L_2}+\cdots\frac{X_n}{L_n} \tag{21}$$

The Lundberg-Palmgren bearing life theory was expanded to include gear surface pitting for the gearbox as a system. A second fatigue life and reliability analysis was also conducted for the bending fatigue lives of the gear teeth in the actuator. Gear and bearing system reliabilities were calculated and combined using strict series reliability where:

$$S=S_1\cdot S_2\cdot S_3\cdot\ldots S_n \tag{22}$$

At a specified reliability, the cumulative lives of each of the machine components are combined to determine the calculated machine system L_{10} life using the Lundberg-Palmgren formula (Ref. 9):

$$\frac{1}{L_{sys}^{m}}=\left(\frac{1}{L_{B_1}^{m_B}}+\frac{1}{L_{B_2}^{m_B}}+\cdots\frac{1}{L_{B_n}^{m_B}}\right)+\left(\frac{1}{L_{G_1}^{m_G}}+\frac{1}{L_{G_2}^{m_G}}+\cdots\frac{1}{L_{G_n}^{m_G}}\right) \tag{23}$$

Unfortunately, Equation 23 is only an approximation since the system Weibull slope varies with load. In a balanced life transmission, the system Weibull slope will be somewhere between the highest and the lowest of the components' Weibull slopes. A form of this equation can be solved numerically for system reliability as a function of life and plotted on Weibull coordinates (Ref. 32). The resulting graph can be fitted with a straight line to determine the system Weibull slope and the system L_{10} life. In the event of an unbalanced life transmission, the lowest lived component will dominate the transmission failures and thus can serve as a good approximation for the system Weibull properties. However, at a given reliability, the system life will always be lower than the lowest lived component because other components can also fail.

Based upon the Lundberg-Palmgren equation, the L_{10} life for the actuator as a system can be calculated where:

$$L_{10}=\left(\frac{C}{T}\right)^{p} \tag{24}$$

C is the theoretical load that produces a life of one million cycles (designated the dynamic capacity), *T* is the equivalent output torque, and *p* is the load-life exponent. Using the linear damage rule, where X_n is the time and T_n is the torque at condition *n*:

$$T=\frac{X_1T_1^p+X_2T_2^p+X_3T_3^p+\ldots+X_nT_n^p}{X_1+X_2+X_3+\ldots+X_n} \tag{25}$$

A mission spectrum for the actuator supplied by the NASA Shuttle Program office was used to compute effective hinge moment loads for both gear tooth contact stress and tooth bending stress, using the Palmgren-Langer-Miner linear damage rule.

A gear contact stress fatigue model established the output hinge moment dynamic capacity for the actuator as 44 860 N-m (397 050 in-lb) with a Weibull slope of 2.5 and load-life exponent of 3.64. A similar analysis for tooth bending fatigue produced a dynamic capacity of 25 280 N-m (223 740 in-lb) with Weibull slope of 2.5 and load-life exponent of 11.6.

A loading spectrum for 100 missions consisting of 35, 188 and 292 load cycles was assumed for the gearbox. Output hinge moment loads varied from 1,700–27, 100 N-m (15,000–240, 000 in-lb). An input pre-load of 5.6 Nm (50 in-lb) added 2,010 Nm (17, 800 in-lb) to the external hinge moment for the analysis.

Results and Discussion

Rudder/speed brake actuator loading. The rudder/speed brake actuator was designed and manufactured in the 1970s for an expected life of 100 missions, over a period of 10 years, with no definitive plans for re-lubrication, maintenance and/or refurbishment. At that time and during the course of operation of the space shuttle fleet, and until the time of the Columbia disaster, it was assumed by the shuttle program office that no maintenance would be required of the actuators and/or that no failure should be anticipated. In other words, there would be a 100 percent probability of survival for the designated 100 missions. As a result, design loads were not available for the analysis we reported.

The shuttle program office provided two simplified load vs. time and load cycle tables (Tables 3–4). These tables give load ranges for an operation in output load cycles. It was assumed that the frequency of loading was constant for all missions and that the full cycle-count of 35, 188 and 298 positive load cycles was the total for the 100 missions. On this basis the cycles were totaled; each time was converted to a percent time and this percent was multiplied by the 35, 188 and 298 cycles to obtain the number of cycles at each listed load. The average mission duration operating time was given as 7.604 hr.

Rolling-element bearing life. The L_{10} lives of the bearings, (the rolling-element fatigue life at a 90 percent probability of survival) were calculated by the actuator manufacturer for the left half of actuator number 2 and were not re-calculated by us. Based on Equation 1a, for each of the bearings the location parameter L_μ is the time below which no failures would be expected (100 percent probability of survival), where $L_{\mu}=0.053L_{10}$ (Ref. 23); these results are shown in Table 2. However, as previously discussed, the analyses reported do not consider the negative effects of grease degradation or the boundary lubrication and resultant wear on the life and reliability of bearings and gears.

The lowest-lived bearing in the system dictates the time below which no bearing failure should occur; from Table 2 this time is 58 hours. Based on an average mission duration operating time of 7.604 hours, and using the two-parameter Weibull Equation 1b, the probability of survival for each bearing was calculated for 1, 12, 20 and 100 missions. The product of the reliabilities of the individual bearings represents the reliability of the system of bearings in one-half actuator.

These results are shown in Table 5 for 12, 20 and 100 shuttle flights. Based on the results reported for bearing 6*b* (Table 2) having a life of 58 hours below which no failure should occur, it can be reasonably concluded that the bearings as a system will operate with no failure for about eight missions (58 hr/7.604 hr-per-mission).

Gear contact fatigue life. Gear tooth failures can be similar to failures of bearing elements. However, there are a few differences due to the complex shape of the gear tooth. Most differences result in sudden tooth breakage in a poorly lubricated, overloaded mesh. For our analysis, we did not consider this as a failure mode.

For the gear analysis, the values for the Weibull slope $m=2.5$, gearing load-life exponent, $p=4$ and dynamic capacity surface strength, $S_{ac}=3{,}450$ MPa (500 ksi). Based on a mission-duration of 7.604 hr, the reliability (probability of survival) of the gears based on contact (surface) fatigue is given in Table 5 for 12, 20 and 100 shuttle flights.

Gear bending fatigue life. A bending stress analysis of the gears was performed to determine their life and reliability in terms of the maximum fatigue strength S, load-life factor p and Weibull slope m that have been determined. These values are: $S=1{,}076$ MPa (156 ksi), $p=11.6$ and $m=2.5$, respectively.

The maximum bending stress on each tooth was used for the life and reliability calculations. For both the pinion and the gear, this stress is the bending stress at the root of the tooth caused by the full mesh load at the highest point of single tooth contact on that tooth.

The result of this analysis is a dynamic capacity of 25,279 N-m (223,741 in-lb). Coupled with the load-life factor of 11.6 and the Weibull slope of 2.5, this dynamic capacity enabled the bending stress life and reliability to be calculated along with the life and reliability for contact stress. The resulting reliabilities based on tooth bending fatigue are summarized in Table 5 for 12, 20 and 100 shuttle flights.

Rudder/speed brake system reliability. The reliabilities for the gears and bearings are summarized for a single half-actuator (the 1L (left) rudder/speed brake gearbox shown in Fig. 2) and for four full actuators as a system in Table 5 for 12, 20 and 100 flights. The bearing reliabilities are summarized under the column "Actual Bearing Reliability" in Table 5. The reliability of the gears as a system is summarized under "Gear Reliability for Each Failure Mode" in Table 5. The column showing "Combined Bending and Surface" reliability is the product of the reliabilities for "Tooth Bending Fatigue" and "Tooth Surface Fatigue." The Total System Reliability shown under the column "Total System Reliability" in Table 5 is the product of the reliabilities shown in each line for the "Combined Bending and Surface Reliability" for the gears and the "Actuator Bearing Reliability."

There are four full actuators on each shuttle; the failure of a single actuator can result in the loss of a shuttle. When considering the probability of survival of four actuators together, these numbers become 89.0, 81.1, and 21.3 percent for 12, 20, and 100 flights, respectively. For 20 flights the probability of a gear failure in one of four actuators is less than 0.6 percent. However, for 100 flights the probability of a gear failure in any one of four actuators increases to 28.3 percent, where bending fatigue failure becomes the predominant mode of gear failure. This is an unacceptable risk.

The reliability of the bearings was determined separately from that of the gears. Combining the bearing and gear statistical analysis for 8 missions results in a 100 percent probability of survival for all four actuators as a system. The reliability of the gears in the actuators for both surface and bending fatigue is higher than the reliability of the bearings. This means the reliability of this system is dominated by the reliability of its bearings.

The Weibull plots of Figure 4 illustrate this; they are for the sub-systems and full system of rudder/speed brake 1L. The gear surfaces' sub-system, shown to the far right as a dashed line, is by far the strongest with the highest reliabilities. The gear teeth-bending sub-system is somewhat less reliable and weaker, and dominates gear system reliability. Finally, the bearings are the weakest system — dominating full system reliability at high reliabilities or low percent of failures.

These life and reliability data for a single half-actuator were assumed to apply to all four full-rudder/speed brake actuators used in the space shuttle. Based on four-RSB-actuators-per-shuttle, the probability of survival of the actuators as a system is the probability of survival of a single half-actuator at a designated number of flights to the 8th power — or S^8. The Weibull plots of Figure 5 show the comparison of the reliabilities of the 1L gearbox and the full system of four gearboxes with their higher statistical percent of failures and lower reliabilities.

The above analysis does not include wear as a failure mode. Wear, as fatigue, is probabilistic and not deterministic. We know that wear occurred in the RSB actuators. However, without an experimental database and predefined, acceptable wear criterion for critical components, wear as a failure mode could not be analytically predicted.

General Comments

From Table 5, the 81.1percent system reliability for 20 flights was deemed an unacceptable risk for the space shuttle. As a result, the number of future flights-per-actuator was limited to 12. The reliability for 12 flights on a single half-actuator was calculated to be 98.6 percent. The reliability of the four full actuators as a system in a single space shuttle was calculated to be 89.0 percent for 12 flights. This means the probability of fatigue failure of the bearings or gears in the system of four full RSB actuators on a single shuttle is 11 percent over a life of 12 flights.

Because of time constraints, a life-and-reliability analysis was not performed by us on the Space Shuttle body flap actuator (BFA) gears and bearings. However, Oswald, et al (Ref. 2) performed experiments on a test rig under simulated conditions to determine the life and failure mechanism of the grease-lubricated space shuttle body flap actuator (BFA) bearings that support the input shaft of the space shuttle body flap actuators. The Weibull analysis of the test data for the four space shuttle body flap actuators (BFA) on one shuttle, each with a two-bearing shaft assembly, established a reliability level of 96.9 percent for a life of 12 missions. For the purpose of our analysis it was assumed by us that this reliability is for the entire BFA assembly. If so, then the combined reliability with the RSB assembly for 12 missions on a single space shuttle was 86.2 percent $[(0.969 \times 0.890) \times 100]$. This yields a 13.8 percent probability that a bearing and/or gear failure will occur on either a BFA or RSB actuator.

In the 1970s, when the Space Shuttle RSB actuator was designed and manufactured, a sophisticated gearbox life probabilistic life and reliability analysis of the type presented in this article was not available. As mentioned, the rudder/speed brake actuator was designed for a life of 100 missions over a period of 10 years with no definitive plans for re-lubrication, maintenance and refurbishment. However, a bearing life and reliability analysis was performed by the manufacturer of the actuators. The bearing analysis performed by us and reported herein was an extension of the one performed by the manufacturer. In previous related studies (Refs. 1–6) are reported the effects of boundary lubrication, grease degradation and wear on BFA and RSB actuators' life and reliability. For this fatigue study we did not consider the negative effects of boundary lubrication, wear and grease degradation on bearing and gear life and reliability.

The risk of a rudder/speed brake actuator bearing failure (not including the risk of gear failures) that could result in the loss of a space shuttle at 100 missions was 70.2 percent. For 20 missions that risk was reduced to 18.4 percent. However, it appears that the NASA Program Office did not examine or extend the bearing analysis to reflect the reliability and/or consider that an actuator bearing failure could be a probable cause for loss of a space shuttle. This omission was further compounded by a failure to provide for a scheduled maintenance program to remove, examine, repair, replace, and/or refurbish these actuators based on time and missions flown.

Summary of Results

A probabilistic life analysis was applied to the space shuttle rudder/speed brake (RSB) actuator 1L. A contact stress fatigue model and a gear tooth bending fatigue model were used for a life and reliability assessment of the gears. Life and reliability of the bearings in the actuator were analyzed separately, based on data provided by the actuator manufacturer using the Lundberg-Palmgren life model. The life and reliability results for the gears and bearings in a single half-actuator were combined, using strict series reliability. The life and reliability of each of the four full actuators was assumed equal and the results extended to the four actuators on each shuttle as a single system. Although the analyses do not consider the probable reduction in life and reliability caused by boundary lubrication, wear and grease degradation, the recommendation to limit actuators to 12 flights before refurbishment addresses this concern. The following results were obtained.

1. Based on the analysis, the space shuttle rudder/speed brake actuators were limited to 12 flights each in order to maintain a reliability of 98.6 percent for any half-actuator, and 89.0 percent for the system of four full actuators on one shuttle.
2. The life and reliability of the actuator gears as a system for both surface and bending fatigue are higher than the life and reliability of the bearings as a system. Thus, the life and reliability of the actuator system is dominated by that of the bearings.
3. Based on the original design requirement of the space shuttle, the rudder/speed brake actuator system, comprising four actuators on each shuttle, has a calculated reliability of 81.1 percent for a life of 20 flights and a reliability of 21.3 percent, for a life of 100 flights.

Epilogue

The body flap actuators (BFA) on all remaining space shuttles were limited to 12 flights each before refurbishment. The remaining two RSB actuators were removed from the Space Shuttle *Discovery*. All four of the removed RSB actuators were replaced with four actuators that had been subjected to three flights and had been removed from a sister space shuttle. On July 26, 2005 *Discovery* returned to flight on a mission to the International Space Station (ISS) — the first Space Shuttle to fly since the Space Shuttle *Columbia* disaster on February 1, 2003. This was the 31st mission and flight of the *Discovery*. On July 4, 2006, the 32nd flight of the *Discovery* occurred on a mission to the ISS. This was the second consecutive return to flight since the Columbia disaster. The *Discovery* was flown seven more times to the ISS without incident, for a total of 39 flights. After the 39th mission, on March 9, 2011 the Space Shuttle *Discovery* was retired from service and was placed on permanent display in the Smithsonian Air and Space Museum, Washington, D.C. The BFA actuators and the RSB actuators functioned as intended — without incident. The RSB actuators in the *Discovery* at the time of its retirement had a total of 12 flights — nine flights from those in the Discovery and 3 previous flights before being removed from the Space Shuttle Endeavor (Ref. 1). **PTE**

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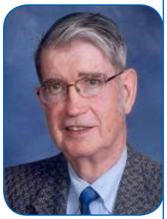

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While **Erwin V. Zaretsky, PE,** is retired from his post as chief engineer/materials and structures, at the NASA Glenn Research Center, he continues as a noted speaker, educator (Case Western Reserve University, University of Wisconsin/Milwaukee and Cleveland State University), writer (at least 180 technical papers and two books) and consultant to both government and industry. A 1957 graduate of the Illinois Institute of Technology in Chicago—and with a 1963 doctorate from Cleveland State University—Zaretsky is also a former head of the NASA Bearing, Gearing and Transmission Section, where he was responsible for most of the NASA mechanical component research for air-breathing engines and helicopter transmissions. With approximately a half-century of experience in mechanical engineering related to rotating machinery and tribology, Zaretsky has performed pioneering research in rolling-element fatigue, lubrication and probabilistic life prediction; his work resulted in the first successful 3 million DN bearing. Zaretsky is an adjunct professor at Case-Western Reserve University and is a member of the executive advisory board of the Northern Illinois University College of Engineering. In 1992 he edited and co-authored the STLE (Society of Tribologists and Lubrication Engineers) book, Life Factors for Rolling Bearings, as he had done previously, in 1997—Tribology for Aerospace Applications. Zaretsky is the recipient of numerous NASA awards for his contributions to the Space Program, among which are the NASA Medal for Exceptional Engineering Achievement, the NESC Director's Award and the astronauts' Silver Snoopy Award. In 1999 the STLE honored him with the Wilber E. Deutsch Memorial Award; he has also received four IR-100 awards. Zaretsky is a Life Fellow of the ASME and a Fellow of STLE.

Is Splash Lubrication Compatible with Efficient Gear Units for High-Speed Applications?

A. Neurouth, C. Changenet, F. Ville, P. Velex and M. Octrue

A thermo-mechanical model of a splash lubricated one-stage gear unit is presented. This system corresponds to a first step towards the design of a hybrid vehicle gearbox that can operate up to 40,000 rpm on its primary shaft. The numerical model is based on the thermal network method and takes into account power losses due to teeth friction, rolling-elements bearings and oil churning. Some calculations underline that oil churning causes a high amount of power loss. A simple method to reduce this source of power losses is presented, and its influence on the gear unit efficiency and its thermal capacity is computed.

Introduction

In automotive applications, there is an increased demand for hybrid technology that combines an internal combustion engine and one or more electric motors. In order to increase power densities of these systems, it is possible to use electric motors that run at high rotational speeds: up to 30,000-50,000 rpm. One limitation to this evolution relies on the design of efficient gearboxes at acceptable cost level. Considering medium-speed applications, splash lubrication appears as the most appropriate and economical solution to move the lubricant onto the gear teeth and bearings. Prior to performing tests on actual gear drives, the aim of this study is to investigate if this lubrication technique is worth considering for the above-mentioned, high-speed application.

To this end, a numerical model of a one-stage helical gear unit has been fashioned. This model can predict the temperature distribution and the efficiency of a mechanical transmission via analytical laws. The following sources of dissipation are taken into account:

- Tooth friction
- Rolling-elements bearings
- Oil churning

As far as the thermal behavior is concerned, the gear unit is divided into lumped elements with a uniform temperature connected by thermal resistances that account for conduction, free or forced convection, and radiation. Particular attention is paid to the oil sump behavior, including the use of specific heat exchange relationships between the lubricant and some rotating elements.

Compared to the finite element method, a thermal network model requires less computing time and provides significant information on temperature distribution. It enables testing of many different assumptions, or operating conditions, and the quantification of their influences on both temperature and power losses, by considering the strong coupling between these physical parameters. This method has been often used to study heat transfer in mechanical transmissions (Refs. 1–4).

In this paper, a thermal network model is presented that can predict the temperature distribution and the power losses of a one-stage gearbox lubricated by an oil sump. Calculations were performed to determine the major sources of power losses; a method to reduce oil churning power loss is exposed and some calculations are drawn to emphasize potential savings on the complete geartrain.

System under Consideration

This study deals with a one-stage helical gear unit that will be designed as the first reduction stage of an electric powertrain. The system is composed of two shafts and the gears are designed to withstand a transmitted power of 100 kW on a speed range from 6,500 – 40,000 rpm. The gear ratio is limited to one-third because the final gearbox is planned to have three stages. All shafts are mounted on ball bearings. The main geometrical data of the gear unit are given in Table 1. The whole set is enclosed in a housing. It is parallel-piped with the following external dimensions: $160 \times 90 \times 200\ mm^3$. Assumptions are made concerning the environment of the gearset: the housing is set on the ground and placed in a ventilation stream (these assumptions are in accordance with the future test rig to be designed). The housing is filled with a certain amount of lubricant to ensure splash lubrication. But bearings are not intended to be immerged in the sump; their lubrication is performed by using channels fed by oil projections via the rotating elements. Oil properties considered in this study are given in Table 2.

Modeling of the Thermal Behavior

Thermal network. In order to simulate the steady-state temperature distribution in the above-mentioned system, the thermal network method has been used. The thermal network relies on the decomposition of the test rig into isothermal elements (gears, shafts, bearings, etc.) and the con-

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Table 1 Main features of the gearset		
	Pinion	**Wheel**
Center distance (mm)	88	88
Tooth face width (mm)	28	27
Module (mm)	2	2
Tooth number	20	61
Pressure angle (°)	20	20
Helix angle (°)	14	14
Arithmetic roughness (µm)	0.8	0.8
Mean diameter of bearings (mm)	51	46
Bearings preload (N)	300	300

Table 2 Lubricant properties		
Kinematic viscosity at 40°C (cSt)	Kinematic viscosity at 100°C (cSt)	Fluid density at 15°C (kg/m^3)
47.6	8.3	887.3

nection of these elements by thermal resistances (Ref. 1). The differences in temperature are created by the heat flow between each element. Thermal resistances depend on the kind of heat transfer encountered, i.e. — conduction, free or forced convection, and radiation. Finally, power losses are computed considering nodes temperature to ensure thermo-mechanical coupling. In this study, thermal resistance and power loss are estimated by using analytical formulas.

The proposed thermal network is given in Figure 1; the gearbox has been divided into 15 elements (Table 3). As far as boundary conditions are concerned, some temperatures were considered as input parameters for the thermal model, i.e. — the ambient air (node #1) and the ground (node #2). The casing is decomposed into (a) a lower part that is in contact with the ground (node #5) — (b) a lateral part where bearings are located (node #4) and (c) an upper part (node #3). The bearings are assumed to be isothermal and are represented by a sole element (nodes #7 – #10). Node #6 corresponds to the lubricant. Node #15 represents the meshing zone of gear teeth — a small area shared between the two gears (nodes #13 and #14) where friction occurs.

In order to evaluate heat exchanges, the model uses four types of thermal resistance:

1. The thermal resistance of striction is based on Blok's works (Ref. 5). Heat generated by teeth friction is localized on a very small area compared to the gears' dimensions. The temperature increase is confined to a thin thermal skin whereas the bulk temperature is not affected. In the thermal network, this resistance links a node associated with the meshing of gear teeth and a node which relies on the gear bulk temperature.
2. To evaluate convection and radiation between the surrounding air and gearbox, Newton's and Stephan-Boltzmann's laws are respectively applied (Ref. 4). The casing is assimilated to an assembly of plates and the classic correlations for flat plates are used to quantify the air-casing convection.
3. Conduction between elements is calculated with classical formulations of heat transfer by conduction (Ref. 1). As an example, several elements (bearings, shafts) are represented as cylindrical bodies and the use of Fourier's law enables to determine the corresponding resistances.
4. Convection with oil is quantified by using different relationships (Ref. 4). Heat transfer with gears running partly immersed in the oil sump is characterized by using standard correlations for a rotating disk, but an additional thermal resistance is also introduced that accounts for the heat removal by centrifugal fling off. As previously mentioned, the casing is modeled as an assembly of flat plates. Then classical heat transfer relationships for forced convection and flows over flat plates are used to quantify the coefficient of convection between the lubricant and the housing.

Table 3 Elements of the thermal network	
Number	**Element reference**
1	Air
2	Ground
3	Upper part of the casing
4	Lateral part of the casing
5	Lower part of the casing
6	Oil sump
7-8	Bearings on pinion's shaft
9-10	Bearings on wheel's shaft
11	Pinion shaft
12	Wheel shaft
13	Pinion
14	Wheel
15	Meshing zone of gear teeth

Power losses. Four sources of power losses are identified in a one-stage gear unit lubricated with an oil sump. Power losses in geared transmissions are traditionally decomposed into no-load and load-dependent contributions. Load-dependent power losses are due to the meshing of gear teeth and internal friction of rolling-elements bearings. No-load-dependent power losses are generated by viscous forces in

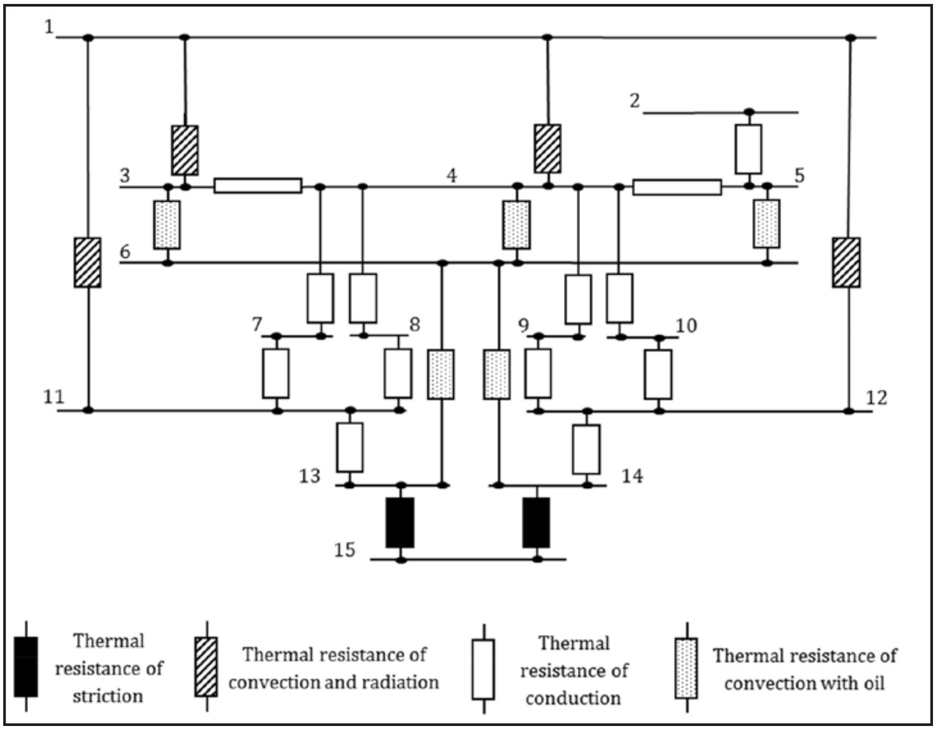

Figure 1 Thermal network of the one-stage gear unit.

rolling-elements bearings, oil churning, and seals friction. In the studied system it is assumed that labyrinth seals are used. They provide non-contact sealing action and then no-friction torque.

For each individual dissipation source implemented in the thermal network, a description is given hereinafter (P refers to power loss and C to friction torque):

(a) *The power losses due to teeth friction*: The heat-generated-per-unit-of-time by the friction between the mating teeth is calculated as a function of the following parameters: (i) the input power P_{input}, (ii) a geometrical factor H_v determined with equation from Velex and Ville (Ref. 6) and (iii) an average friction coefficient f determined with ISO14179-1 (Ref. 7). These power losses are injected to node #15:

$$P_{teeth\,friction} = P_{input} * Hv * f \tag{1}$$

(b) *The power losses due to friction in rolling-elements bearings*: To evaluate this source of power loss, the classical formulas developed by Harris are used (Ref. 8). It depends on the mean diameter of the bearing d_m, a parameter that takes into account bearing characteristics f_1, and the load P_1. These power losses are injected to nodes 7–10:

$$C_{friction} = f_1\, P_1\, d_m \tag{2}$$

(c) *The power losses due to viscous forces in rolling-elements bearings*: Harris' formulas are also used to quantify no-load-dependent power losses in the bearings. As shown (Ref. 1), this source of power loss is a function of oil kinematic viscosity v, the rotational speed n, the bearing mean diameter d_m and a factor f_0, which depends on the type of bearing and lubrication. These power losses are injected to nodes 7–10.

$$C_{hydrodynamic} = 10^{-7} f_0 \,(vn)^{2/3}\, d_m^3 \tag{3}$$

(d) *The power losses due to churning phenomenon*: Formulas from Changenet et al are used to determine oil churning power loss (Ref. 9). The drag torque is expressed as a function of a dimensionless torque $C_{dimentionless}$ that depends on the fluid flow around a rotating gear, its rotational speed w, its pitch diameter D_p and the submerged surface area of the gear S_m. These power losses are injected to node #6.

$$C_{churning} = \frac{\rho\, \omega^2\, S_m\, (D_p/2)^3\, C_{dimentionless}}{2} \tag{4}$$

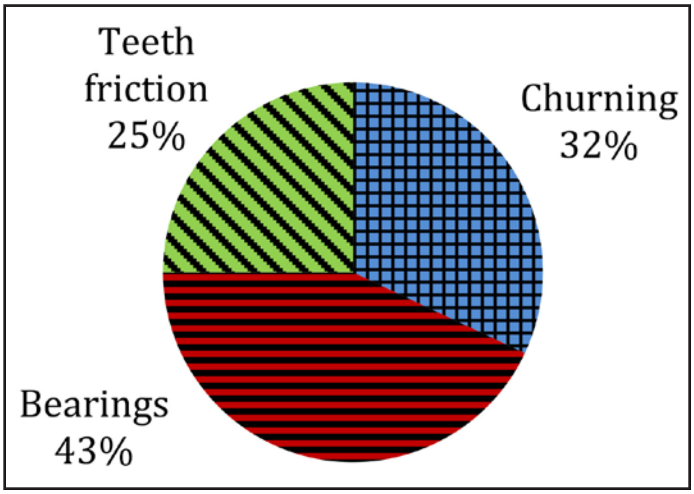

Figure 2 Power losses repartition at 100 kW and 32,000 rpm.

First results. To investigate the power losses repartition, some calculations were performed using different operating conditions. Figure 2 presents the results obtained for an input power of 100 kW at 32,000 rpm, and when oil level is assumed to be at 40 mm under the gears' axis. In this case the teeth friction represents only 25% of the global power losses. The drag torque — due to rolling-element bearings and oil churning — dominates and consequently highlights the strong influence of no-load-dependent power losses.

To improve the overall performance of the gear unit, the power losses must be controlled. A first step to reduce heat generation consists in limiting the dissipation associated with oil churning. Of course it is possible to reduce churning losses by lowering the oil level in the sump, but this action will also lead to higher gear bulk temperature. For example, at 100 kW and 6,400 rpm, if the oil level is decreased by 10 mm, churning loss is lowered by a factor of two. Simultaneously, the temperature difference between oil and the gear wheel increases by 10°C. Another solution to minimize churning losses without modifying temperatures consists of using axial flanges (Ref. 10).

Tests to Reduce Churning Losses

Test rig. A specific test rig was developed to investigate the housing influence on churning (Ref.10). A pinion shaft is operated by an electric motor that allows speeds up to 7,150 rpm. Churning losses are measured with a strain gauged contactless sensor with a full-scale range of 2 Nm. Because the pinion shaft is supported by two pairs of ball bearings, their contribution to the total drag torque has been experimentally determined and subtracted from global torque measurements. The housing is a parallelepiped with a face made of Plexiglas (Fig. 3). Thermocouples are used for measuring the ambient and lubricant temperatures. Several heating covers have been installed on the external bottom face of the housing in order to perform measurements up to temperatures near 100°C. Some movable walls can be inserted in the gearbox to modify the clearances between a wall and a gear face or top land.

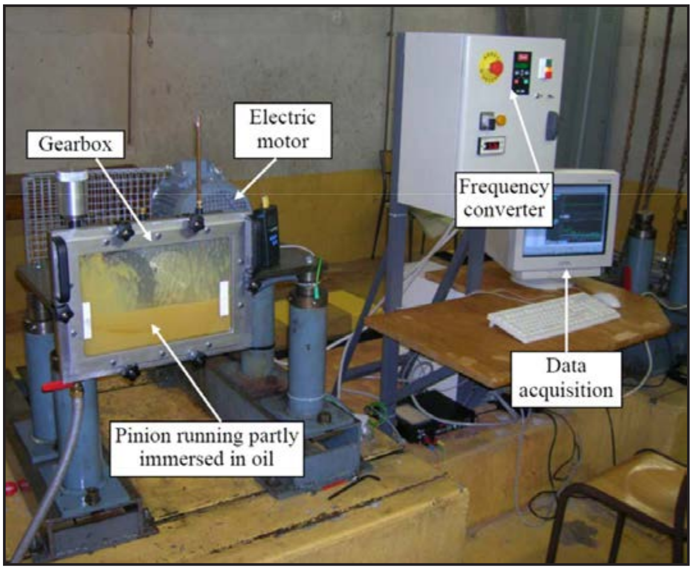

Figure 3 Test rig.

The main parameter generating a decrease in churning power losses is the axial distance between a flange and a gear face (*Ja* in Fig. 4). By mounting flanges close enough to the gear lateral faces, it is possible to divide by two the oil churning losses. In (Ref. 10) several measurements have been conducted and the influence of flanges has been quantified by establishing relationships based on dimensional analysis.

As tests performed in (Ref. 10) deal only with spur gears, some additional experiments have been conducted in order to extend previous results to helical gears. Figure 6 presents some typical results obtained for two different gears: a spur gear and a helical gear. These gears have the same module (3 mm), the same face width (24 mm) and a similar outside diameter (about 162 mm). For a peripheral speed of 58 m/s, churning losses can be substantially reduced and it is noted that the helical gear has the same behavior as the spur gear. Other measurements have been performed on gears with smaller module (1.5 mm). These tests confirm the above-mentioned evolutions.

Numerical results on the complete gear unit. Formulas developed in (Ref. 10) represent a good approximation to estimate the decrease in churning losses. They have been used in the thermal network model to estimate potential savings associated with flanges on the complete geartrain. The following parameters have been used to obtain numerical results: (i) presence of flanges with an axial clearance equal to 4 mm; (ii) flanges with a clearance of 1 mm.

By adding flanges, the power losses repartition is modified. Table 4 presents the total power loss and the relative importance of the dissipation sources for each test. It appears that churning losses can be decreased up to 50 percent with the smallest axial clearance, whereas total power loss can be reduced by 13 percent.

The evolution of power losses leads to a modification in temperature distribution, which is presented in Table 5. As the oil bulk temperature depends on the total power loss in the gear unit, it can be noticed that the presence of flanges generates a decrease of 10°C in the lubricant temperature. Because of high heat transfer coefficient of convection, the bulk temperature of the gear wheel is found to be very close to that of the lubricant. As far as the pinion is concerned, this element is not immersed in the oil sump and stabilizes at higher temperature.

Compared to oil churning and teeth friction, bearings-related losses become predominant; they vary from 657 W in the initial configuration to 750 W with flanges at 1 mm. This behavior can be explained by viscous forces in rolling-element bearings: the evolution of temperature distribution in gear units modifies the local oil viscosity at certain points in the transmission; it highlights the role of temperature-related power loss sources.

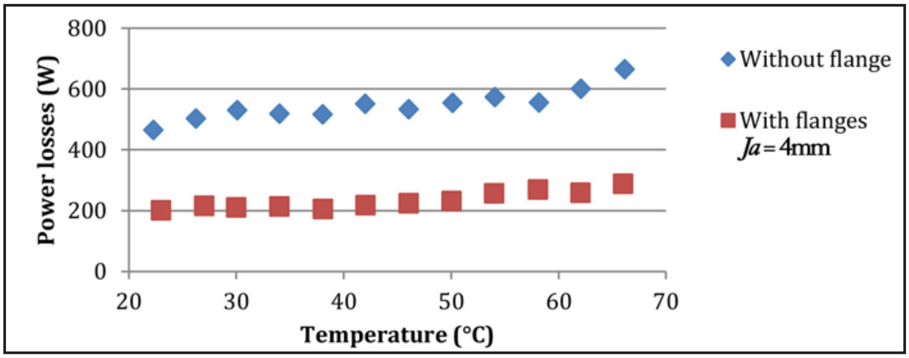

Figure 5 Power losses with and without flanges at 7,000 rpm for the spur gear.

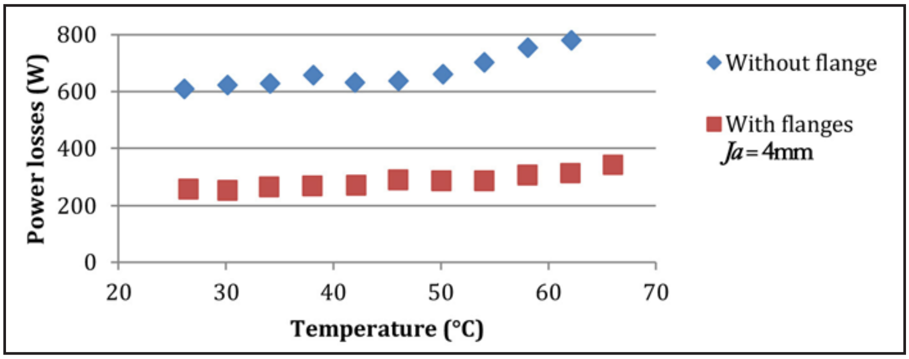

Figure 6 Power losses with and without flanges at 7,000 rpm for the helical gear.

Table 4 Power losses repartition (32,000 rpm–100 kW)

	Initial configuration	Use of flanges *Ja* = 4 mm	Use of flanges *Ja* = 1 mm
Total power loss (W)	1,512	1,389	1,315
Churning (%)	32	22	14
Bearings losses (%)	43	51	57
Teeth friction (%)	25	27	29

Table 5 Temperatures distribution (32,000 rpm–100 kW)

Calculated temperatures	Initial configuration	Use of flanges *Ja* = 4 mm	Use of flanges *Ja* = 4 mm
Oil (°C)	87	81	77
Pinion (°C)	94	89	86
Wheel (°C)	87	81	77

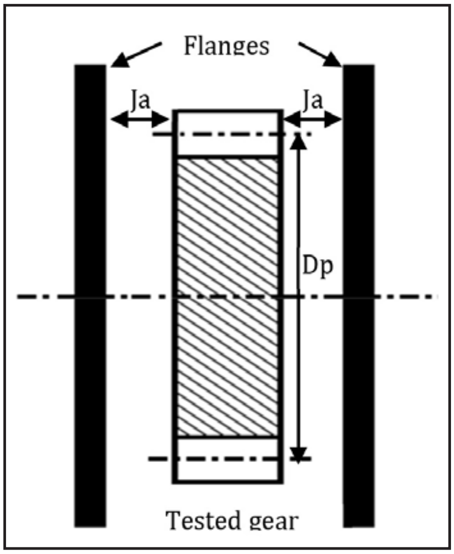

Figure 4 Flanges position relative to the gear.

Conclusion

To investigate power losses generated by splash-lubricated gears at high speed, a thermo-mechanical model of a one-stage gear unit was created. Particular attention was given to churning power losses and to heat transfer between gears and the oil sump. The numerical model shows that for high rotational speeds, churning losses represent a major source of dissipation.

A simple solution to reduce churning losses — without decreasing convection heat transfer with the oil sump — is to insert some flanges in the gearbox in order to modify axial clearances between a wall and a gear face. This solution is presented and implemented in the thermo-mechanical model. The numerical results show that it seems possible to reduce by 13 percent the global power losses at nominal operating conditions. The presence of these flanges also generates a decrease of 10°C in the oil bulk temperature.

It can be noticed that rolling-element bearings also represent an important source of dissipation. For the moment these elements have been considered as isothermal in the numerical model. As an example — no temperature difference is calculated between the inner and outer rings. A more realistic modeling will be implemented in the near future by using the thermal network method (Ref. 11). **PTE**

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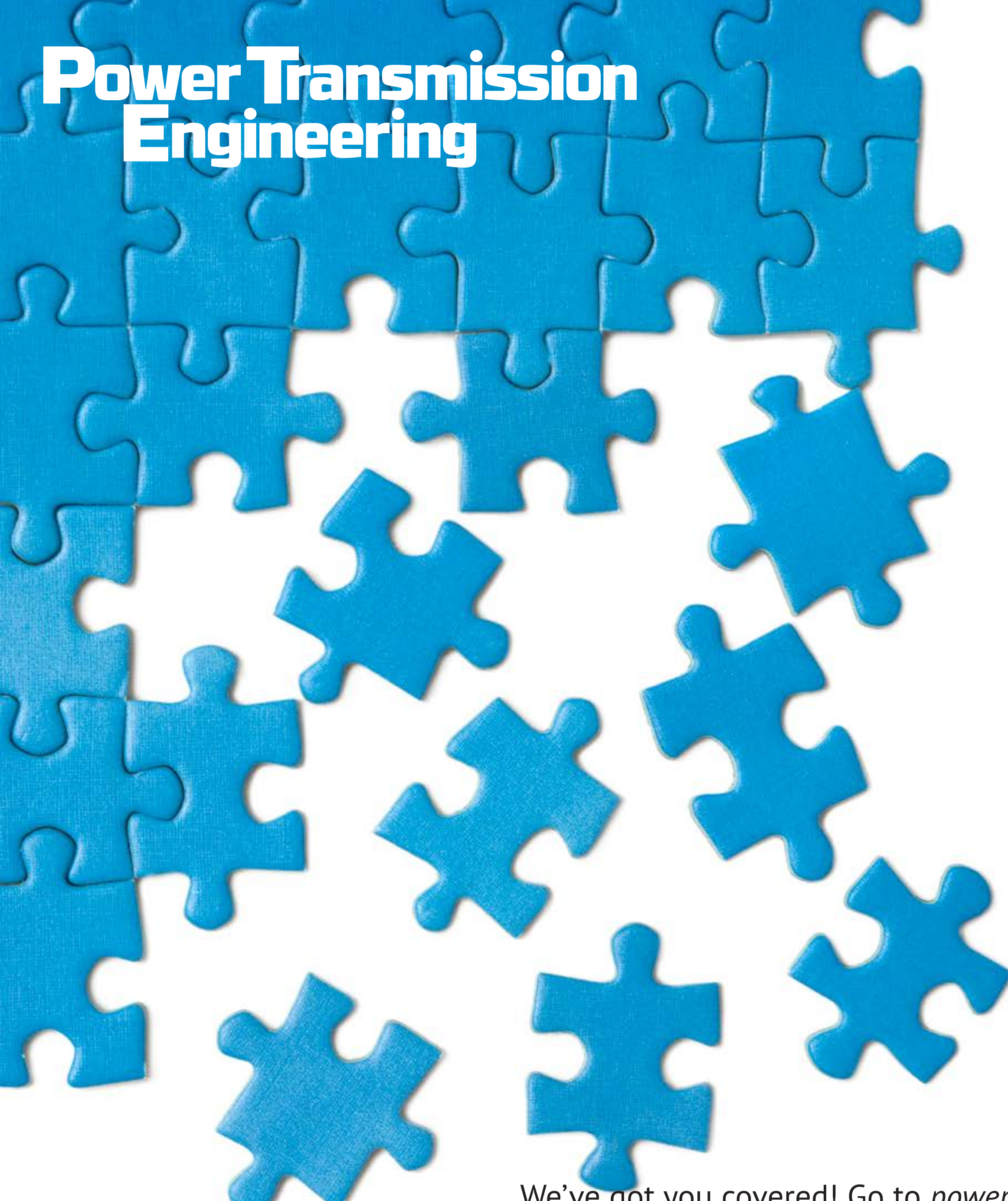
Adrien Neurouth is a student pursuing his Ph.D. at CETIM, where he concentrates on the study of the energy performance of high-speed transmissions.

Prof. Christophe Changenet has since 1992 been a researcher and lecturer at ECAM Lyon (Ecole Catholique d'Arts et Métiers de Lyon) — the institution's graduate school of engineering. From 1998 until 2008, he was head of the Department of Mechanical Engineering and Energetics at ECAM Lyon and, since 2008, Changenet has served as the school's head of research.

Dr. Michel Octrue is a CETIM (Centre Technique des Industries Mécaniques) expert in the field of mechanical power transmission. He specializes in the behavior of mechanical components (gears, roller bearings, etc.), and brings particular expertise to projects involving mechanical power transmission components and their integration in machinery and gearboxes for automotive and transportation devices. His experience covers the different stages — design and calculation; choice of tolerances; selection of materials and heat treatment; and development and validation by numerical simulation or testing devices. He is also involved in testing strategy and test bench design for mechanical components. Octrue is a graduate engineer from ECAM–Lyon, where he subsequently earned his doctorate in engineering. Among his technical achievements are the design and realization of software and training sessions to transfer the knowledge in the fields of gear technology (gear design, gear rating, gear tolerances and gear behavior in noise and vibration; the development of test benches for endurance to characterize cylindrical and worm gears; the design and realization of several test benches for gearboxes and different machine elements; the participation and animation of working groups in standardization activities in the framework of ISO TC60 – gears; and the development of methodologies to reduce gear noise at the source. Octrue also serves as a technical expert for ANVAR, has been published in 20 international publications, and has authored two books on worm gears — *New Method for Designing Worm Gears* and *An Industrial Approach for Load Capacity Calculation of Worm Gears (Verifying and Design).*

Adrien Neurouth, Fabrice Ville and **Philippe Velex** — INSA Lyon


Power Transmission Engineering

Global Industrial Outlook: Weathering the Storm

Brian Langenberg

Last issue (Sept.) we explained how bad energy is getting. So on to alternatives.

We just returned from WEFTEC—the premier U.S. water and wastewater management show. Our recent discussion (Game Changer) highlighted the negative impact of low oil prices.

We confirmed that municipal finances continue to improve, leading to continued activity on projects originally planned for 5, 6, and 7 years ago finally being released.

Could this possibly be enough to offset weak energy? Not in 2015 or 2016.

Speaking with a publicly traded pump manufacturer that tends to focus on small-to-mid-size municipal water systems, we gleaned the following:

Market	Sub-Segment	Commentary
Oil & Gas	Upstream, midstream	Off (50%); no silver lining.
Fire Pumps	Non-residential buildings	"Very good." Same commentary from other participants. Gulf Coast is "a touch" slower.
Irrigation	U.S. farming	California doing well owing to drought. Rest of market is weak.
Municipal	Water, wastewater treatment	Growing. Good conditions.

We think these trends will continue at least through 2016, albeit with oil and gas comparisons eventually (hopefully) stabilizing.

So what/where are the silver linings when the entire machinery sector, ex trucking, is getting whacked?

1. **Low commodity prices enable demand**. Nothing big is happening now. But infrastructure soaks a lot of energy and materials; funding improvements is easier after an election (hopefully) and with lower input costs. The set-up is for early '17 bills getting passed with funding (President-elect's brief "honeymoon"), and rising activity in 2018.
2. **Improving home prices** support continued residential optimism. You (not me) will be happy to know that, for example, my village has just proposed a 20 percent increase in my tax assessment. THAT isn't going to happen, but the direction remains up.
3. **U.S. economic growth continues**, despite headwinds from lower energy and the strong U.S. dollar.
4. **Aerospace and auto market conditions should remain strong**.

In a nutshell, weathering the continuing low commodity price storm (energy, mining, agriculture) will remain your number one priority—but with pockets of potential—as above.

Focus Company: Bombardier (BBD.B)

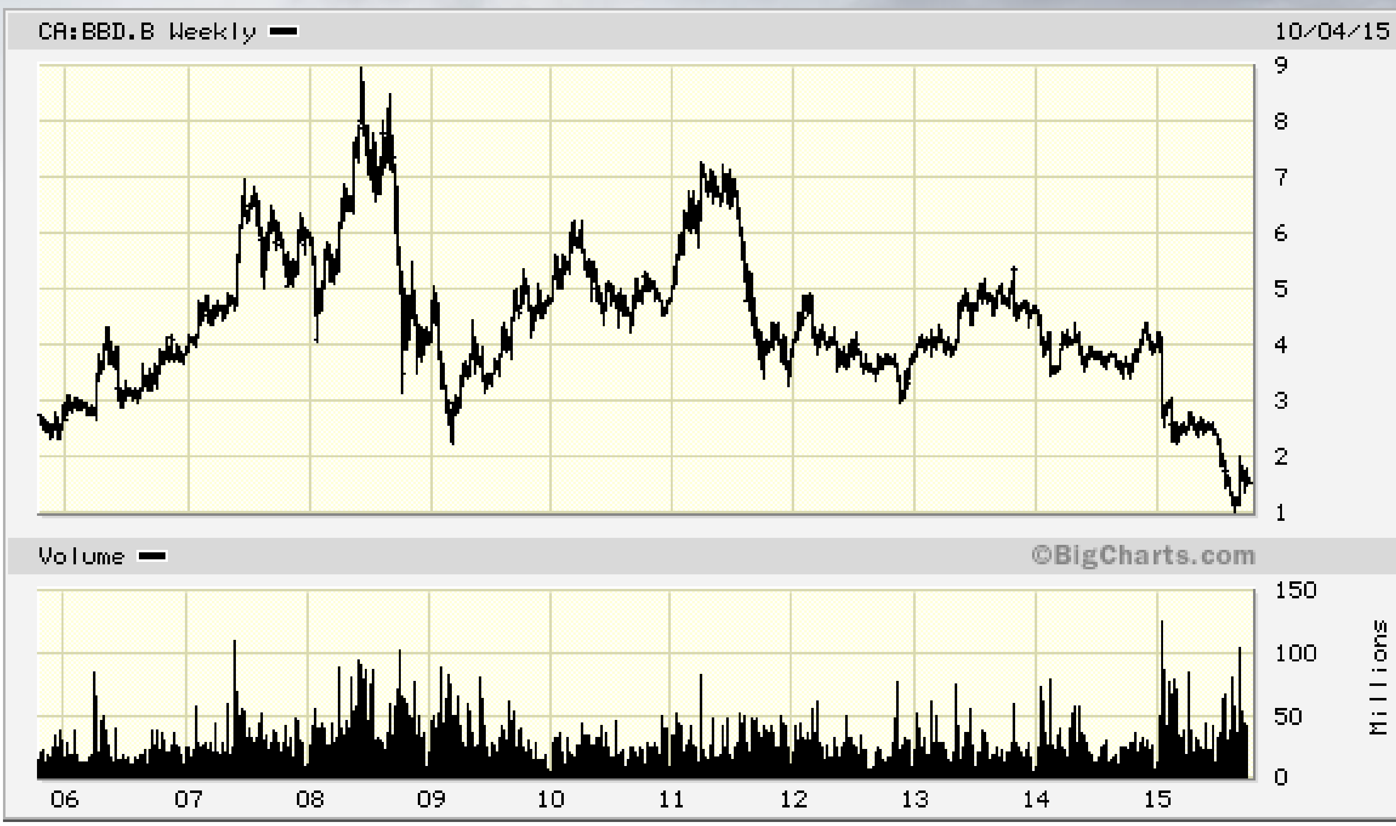
Bombardier operates in two business segments: Commercial Aviation and Rail. This company matters to you if you supply engine components or related systems to Pratt & Whitney on its geared turbo fan (GTF) engine slated to launch on the significantly delayed C-Series jet.

That chart looks...*awful*. The company is over two years late on C-Series entry into service (hoping for certification in November), which has cost orders, replaced senior management, and is currently running around seeking cash so they can ensure getting aircraft into commercial production.

If you are a supplier on the GTF engine, this appears worrisome, but take heart—the GTF is extremely important to United Technologies (owner of Pratt & Whitney) and the AOFUTXEWSF (Association of Former UTX Executives Who Speak French) has taken over at Bombardier. They will get it into the air.

Further your career: On July 1, 2015 I took over as Chair (and Lecturer) of Graduate Business Programs at Aurora University. The Dunham Graduate School of Business MBA with a Leadership concentration is ideally suited for technical or engineering professionals seeking to gain the business skills, tools, and mentorship to further their careers. AU also offers adult degree completion programs.

- Highly accomplished, professional business people teach our courses, whether on ground or online (same professors).
- Front-end career goal discussion and mentorship. We want you to succeed.



- Rolling admission with six 8-week terms per year allows you to flex up/down, depending upon your needs.
- Focused curriculum: 12 classes — seven core, five electives. Each core class is a basic, needed building block; the electives are geared for your objectives.
- Locations: Aurora, Illinois, Woodstock, IL and Online.

Contact *blangenberg@aurora.edu* to arrange for a phone or in-person discussion. **PTE**

Brian K. Langenberg, CFA has earned recognition as a member of the Institutional Investor All-America Research Team, a Wall Street Journal All-Star, and Starmine Best on the Street. As Principal of Langenberg & Company, he advises CEOs and senior executives on strategy and capital markets, and makes numerous public speaking appearances. In July 2015 he was named Chair (and Lecturer)of Graduate Business Programs at Aurora University.

Brother Gearmotors

HIRES TWO NEW REGIONAL SALES MANAGERS

Brother Gearmotors recently added two more regional sales managers: **William (Bill) Miicke** and **Larry Thomas**.

William (Bill) Miicke

Larry Thomas

The sales managers are responsible for the growth and sales performance of their respective regions, while building a portfolio of direct OEM clients and distribution sales channels. Their focus is on prospecting and capturing market share, developing and implementing sales strategies that are aligned with corporate objectives and working closely with product management and marketing to successfully launch strategic campaigns.

Miicke's Northeastern territory covers the states of New York, New Jersey, Pennsylvania, Connecticut, Vermont, New Hampshire, Massachusetts, Maine, Rhode Island and West Virginia. Prior to joining Brother Gearmotors, he held a variety of high level executive positions, including vice president and CEO at Precision Fasteners. Before that, he was a sales engineer at Eaton-Tinerman Corporation, and held multiple sales positions with TRW's fasteners division. Bill holds a B.S. in marketing management from Susquehanna University and resides in Warren, NJ.

Thomas' Midwestern territory covers the states of Kentucky, Michigan and Ohio. Prior to joining Brother Gearmotors, he was account manager at Applied Industrial Technologies. Larry resides in Fairview Park, OH with his wife and three children.

"In the North American market, Brother Gearmotors is in the midst of a rapid surge – one that, in addition to a new facility and expanded product line, includes the addition of talented sales representatives such as Bill and Larry," said Matthew Roberson, senior director of Brother Gearmotors. "We are confident they will provide the active management, support and leadership needed to drive sales and continue to increase our customer base."

Kaman Distribution

ANNOUNCES NEW AUTOMATION, CONTROL & ENERGY OPERATING UNIT

Kaman Distribution recently announced the formation of Kaman Automation, Control & Energy, a new operating unit that is said to be the only fully integrated national provider of industrial solutions serving original equipment manufacturers, industrial production plants and infrastructure facilities throughout North America, according to Kaman.

The new organization, dubbed Kaman AC&E, has been created through the combination of some of Kaman Industrial Technologies' existing resources along with the Minarik Corporation, acquired by Kaman in 2010, and the Zeller Corporation, acquired in 2012. The new organization has greater than 500 employees – including more than 75 degreed engineers – who operate out of 27 strategically located facilities throughout North America and offer three separate service areas: engineering, manufacturing and distribution.

Kaman AC&E's engineering offerings include engineered processes and machine control solutions as well as advanced energy management systems for a wide variety of processing plant, original equipment and infrastructure applications. AC&E solutions include engineering and integration services that include: application engineering, motion control, vision inspection, automation and industrial control, water/wastewater process control, machine safety, productivity monitoring and enhancement, building control and utilities management, facility upgrades, value-engineering services and a number of other value-adding capabilities such as assembly and sub-assembly, custom cabling, system design, motion systems design and other custom engineering and development services.

KAMAN
Industrial Technologies

The organization's manufacturing capabilities include the design, manufacture and testing of OEM control panels as well as documentation, value engineering, component inventory management, code and standard compliance and custom enclosure production.

Kaman AC&E's distribution capabilities include an inventory of more than 100,000 SKUs and 135 different brands, including products from suppliers like Kollmorgen, Phoenix Contact, Schneider Electric, Rittal and many others. Products offered include precision mechanical, electrical and mechanical power transmission, motion control, automation, industrial controls, and power and energy applications for the OEM or MRO (maintenance, repair and operation) applications for most manufacturing and processing industries.

"The formal launch of the Kaman Automation, Control and Energy platform of Kaman Distribution is a significant milestone in the growth of our organization, and represents a tremendous opportunity for our customers, as we can provide them with the entire gamut of products, services and integration from one place," said Gary Haseley, senior vice president and general manager of Kaman AC&E. "We believe our all-encompassing engineering capabilities, expertise in manufacturing and assembly, and extensive inventory of products from world-class suppliers makes us the only organization of its kind on the continent. We feel our capabilities combined with our commitment to putting our customers first will help our customers meet their return-on-investment goals, which ultimately will lead to our continued success."

TimkenSteel Ohio Technology Center

OPENED FOR SBQ STEEL RESEARCH

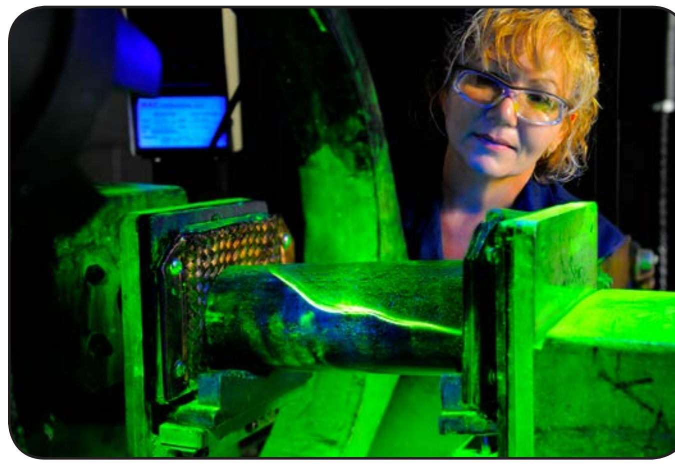
Looking to get an edge in achieving clean special bar quality (SBQ) steel and developing new steel grades, TimkenSteel recently opened a $5 million technology at its Northeast Ohio corporate campus.

"TimkenSteel is the global leader in clean steels because we understand the science and measurement of inclusions better than anyone, and we're the only steelmaker that can take that understanding and use it to predict component life," said Ray Fryan, vice president of technology and quality at TimkenSteel. "The new TimkenSteel Technology Center, along with the talent across our entire team, enhances our ability to develop the SBQ steel that our customers need to solve their toughest challenges."

The TimkenSteel Technology Center features a dozen laboratories, including a brand new scanning electron microscope, an ultrasonic lab, a physical process modeling lab and additional labs where metallurgists and materials scientists can test steel and find inclusions less than the width of a hair.

Steel cleanness is a factor to consider in products such as gears, bearings, axles, crankshafts, down-the-hole drilling equipment and military applications.

The 20,000-square-foot facility, in combination with the company's steelmaking capabilities, allows TimkenSteel to integrate learning and knowledge and achieve new levels of steel performance from the company's new $200 million jumbo bloom vertical caster at the Faircrest Steel Plant in Canton, OH.



Hand in Hand with the Times

SPS IPC Drives 2015 is coming while Industry 4.0 continues to grow in relevancy, and they're a perfect fit for each other

Alex Cannella, News Editor

Out of the endless tide of shows happening this time of year, SPS IPC Drives may not be the brightest blip on your radar. It's not quite the monolithic event its near neighbor, the Hannover Fair, is, and it's a lot farther from home than something like Gear Expo. Only a few thousand Americans and a handful of exhibitors make the trip across the pond to go there each year. Most companies operating in the US understandably leave this show to their more local European branches to cover. But when it comes to electric automation, something a lot of our readers *do* care about, it's the place to be in November, which means that even if you aren't going, Europe's bound to show off something new and shiny that you're going to want to know about.

And in the hype leading up to SPS IPC, two words keep constantly being tossed about: Industry 4.0.

The (relatively) new buzzword for the growing push towards digitalization and automation of the workplace has been on a lot of lips lately. And while SPS IPC is neither the first industry event to focus on it nor the end all authority on it, you can be sure that the show will be as important to Industry 4.0 as Industry 4.0 is to the show.

As a German show entirely about electric automation, it shouldn't come as much of a surprise that SPS IPC would make Industry 4.0 a centerpiece at the show. It isn't just a newly German-coined catchphrase, it's a trend in the same spirit of what SPS IPC has been working towards for over two and a half decades. The two fit together so naturally that it was only a matter of time before they came together.

Taking (literally) center stage in Hall 3A, a joint stand on Industry 4.0 entitled "Automation meets IT" will be devoted to exhibits and lectures on the latest in Germany's industrial revolution. The joint stand, which will be taking up two full booths, is the joint effort of ten separate companies, the most notable amongst them being Bosch Rexroth and Siemens.

On their website, SPS IPC is describing the exhibit as a place where speakers will "give visionary impulses and present IT-based solutions in the field of automation technology on the way to the digital production of the future as well as databased business models."

Rounding out the fair's official Industry 4.0 offerings will be the "MES goes Automation" stand, which will provide visitors with information about manufacturing execution systems, and the"DFKI-SmartFactory" booth, both situated a stone's throw from the main joint stand.

SPS IPC isn't the only one doubling down on Industry 4.0. Individual companies will also be showing off the latest in automation products, not least amongst them being Siemens, whose motto going into this show will be "On the Way to Industrie 4.0 - Driving the Digital Enterprise." Many of their most highlighted products for this year's show, such as the Siemens Cloud and TIA Portal, all have to do with automation.

Other particularly noteworthy products Siemens will be showing off are the Simotics S-1FG1 servo geared drive motor with finely-graduated transmission ratios, and the Sinamics S120, a highly modular and flexible cabinet module drive package (which, coincidentally, you can read up on in our product news section in this very issue or online).

"We want to show the really aligned combination of how a drive system and a motor can work together for maximum energy efficiency, for maximum power, and also for maximum engineering efficiency using the complete, totally integrated automation," Heinz Eisenbeiss, head of the Siemens booth at SPS IPC Drives, said in a pre-show press conference.

Siemens will also be showing off one of their drive systems at work in a paper producer, a complex system with numerous individual drives. Siemens will be using the display to highlight the advantages of using an IDS.

Amongst the small village of Siemens' booths and their dozens of other products on display, they'll also be showing off their Sirius ACT push buttons and signaling devices, version 9.0 of their Simatic Process Device Manager and the Bipex-X and Sipex backlash free Flender couplings.

Voith will also be making an appearance at the show, bringing along their CLDP servo drive, which combines the benefits of electromechanical drives with those of hydraulics. Voith's servo drive features wear-free operation and a service interval of three years. It comes in three different designs: linear, orthogonal and parallel, and covers a power range up to 1,000 kN and speeds of up to 1,000 mm/s.

Amongst the few pioneering souls striking out from the US to SPS IPC is Celera Motion, the newly formed combination of optical encoder producer MicroE and drive manufacturer Applimotion. SPS IPC is nothing new for the MicroE side of Celera, but this is the first time they'll be bringing Applimotion products along to the show, as well.

"This is the first time Applimotion motors will be displayed," David Kosewski, product marketing manager of Celera Motion, said, "and one of the first two shows globally where we are present as Celera Motion. Past experience at the show has been good for MicroE, though we feel that Celera

Motion has a much more compelling value proposition with its expanded capabilities to solve OEM motion control challenges."

At the forefront of Celera's product display will be two new encoders, Optira and Veritas. Optira (also in our Product News section this issue) features a high resolution with multiple processes occuring in the sensor head, as well as wide tolerances and Celera's PurePrecision optical technology for ease of setup.

"Optira is the only encoder in its class to provide a resolution of up to 5nm with all automatic gain control, interpolation, and signal processing carried out in the sensor head," Kosewski said.

The Veritas encoder features many of the same capabilities as the Optira, but its main draw is its size: only 35 × 13.5 × 10 mm. With automatic calibration, Veratus is designed for plug-and-go setup with resolution from 5 µm to 20 nm.

Though Celera's encoders are already regularly shown off at SPS IPC, this year's show is an important debut in the European market for their Applimotion motors.

"We are just beginning to share, beyond the traditionally North American customer base, the capabilities of Applimotion OEM-specific motors assemblies and mechatronic solutions built around those motors," Kosewski said. "So we are excited to share this, as well as our new encoder developments, with the visitors of SPS IPC Drives. Reaction by our customers has been overwhelmingly positive, and we look forward to using the show as a broad opportunity to share information about the innovation, expertise and vision Celera Motion has to offer."

This year's SPS IPC Drives will feature over 1,600 exhibitors in 14 different halls, making this one of its biggest years thus far. As all the main European players in the field gather, SPS IPC is set to be this year's closing statement on the state of the electric automation industry. You may not be going in person, but SPS IPC will be hard to ignore.

For more information:

Siemens Industrial Automation
Phone: (847) 640-1595
www.siemens.com

Voith Turbo
Phone: +49 7321 37 3456, CET (UTC+1h)
www.voith.com

Celera Motion
Phone: (781) 266-5700
www.celeramotion.com

October 27-29 – Discover 2015 Florence, KY. Mazak Corporation encourages those involved in the metalworking industry to attend its Discover 2015 technology and education event. Here, the machine tool builder plans to spotlight new technologies and trends that will change how part manufacturers operate, including unconventional ways to drive operational efficiency via additive manufacturing, CNC technology and the Industrial Internet of Things (IIoT) concept. Additive manufacturing is creating a shift in the way engineers and designers think about product development, and Mazak is leading the way with its additive-capable INTEGREX i-400AM. The HYBRID Multi-Tasking machine will make its North American debut at Discover 2015, and attendees will experience how it integrates laser cladding with advanced full 5-axis milling and turning capabilities. Overall, more than 30 of the latest Mazak machine tools will perform real-world cutting demonstrations throughout the event. Applications experts will be on standby during the demonstrations to discuss total manufacturing solutions as well as part-processing improvements with attendees. The company will also offer a series of seminars that will teach attendees the latest metalworking tools, trends and techniques for improved productivity and profitability. For more information, visit *www.MazakUSA.com/DISCOVER2015*.

October 27-30 – PTC Asia 2015 Shanghai New International Expo Center, China. PTC Asia is the largest annual industrial event in power transmission and control in Asia and the second largest in the world. In the exhibition, the debut of new products will illustrate the integration and development of Industry 4.0, highlighting the utilization of eco-friendly materials as an optimal solution to achieving industrial high efficiency and energy conservation. The expo will boast over 90,000 sq. m and 80,000 visitors. Over 15 ndustries will be included, but PTC will focus on the rail transportation, petrochemical, mining machinery, wind power and food and medicine packing industries in particular. PTC will also be providing over 12 seminars as well as matchmaking services between businesses and potential buyers. For more information, visit *www.ptc-asia.com/EN*.

October 28-30 – ASME DSCC 2015 Hilton Columbus, Columbus, OH. The Dynamic Systems and Control Conference is the showcase technical forum of the ASME Dynamic Systems and Control Division. It provides a focused and intimate setting for dissemination and discussion of the state of the art in dynamic systems and control research, with a mechanical engineering focus. The 2015 DSCC Technical Program will consist of sessions in all of the usual areas of interest to the Division. In addition, the conference will feature specific technical tracks that uniquely identify this particular DSCC. The location of the conference, in the heart of the Manufacturing and Automotive industries, makes these two areas especially appropriate for special tracks. Other special tracks will include Interplay between Biology/Ecology/Life Sciences and Engineering and Information Technology in Mechanical and Aerospace Engineering. The program will include contributed sessions, invited sessions, tutorial sessions, special sessions, workshops and exhibits. For more information, visit *www.asmeconferences.org*.

November 3-5 – 2015 Detailed Gear Design - Beyond Simple Service Factors

Hyatt Place, Las Vegas, NV. This course explores all factors that go into good gear design from life cycle, load, torque, tooth optimization, and evaluating consequences. Students should have a good understanding of basic gear theory and nomenclature. The course is designed for gear engineers, gear designers, application engineers, and others who are responsible for interpreting gear design or who want to better understand all aspects of gear design. AGMA members: $1,395 for first registrant from a company, $1,195 for additional registrants. Non-members: $1,895 for first registrant, $1,695 for additional registrants. For more information, visit *www.agma.org*.

November 4-5 – Advanced Engineering UK 2015 NEC, Birmingham UK. Integrating multiple show exhibit zones with the UK's largest free-to-attend engineering conference program, Advanced Engineering is where the supply chain meets with visiting engineering and procurement decision makers from OEMs and top tier organizations spanning: aerospace; automotive; motorsport; marine, civil engineering, and more. Whether you are a visitor or an exhibitor, Advanced Engineering will not only provide you with a business forum and supply chain showcase for your own sector, but will also introduce you to new opportunities in industries using related technologies and services. For more information, visit *www.advancedengineeringuk.com*.

November 13-19 – 2015 International Mechanical Engineering Congress & Exposition Houston, TX. ASME's International Mechanical Engineering Congress and Exposition (IMECE) is the largest interdisciplinary mechanical engineering conference in the world. IMECE plays a significant role in stimulating innovation from basic discovery to translational application. It fosters new collaborations that engage stakeholders and partners not only from academia, but also from national laboratories, industry, research settings, and funding bodies. Among the 4,000 attendees from 75+ countries are mechanical engineers in advanced manufacturing, aerospace, advanced energy, fluids engineering, heat transfer, design engineering, materials and energy recovery, applied mechanics, power, rail transportation, nanotechnology, bioengineering, internal combustion engines, environmental engineering, and more. For more information, visit *www.asmeconferences.org*.

December 7-10 – 14th International CTI Symposium

Estrel Hotel, Berlin. The automotive industry is about to change drastically due to energy and CO2 discussion, interconnection and automation. This results in new requirements on drive trains and transmissions with regard to efficiency, comfort, dynamics and safety, which in turn lead to a large variety of competing drive and transmission concepts and to a previously unknown wave of innovations for components and assemblies. It does not only apply to the new electrified drive trains, but also to conventional drive trains and transmissions, which will remain the pillars of drive technology for the time being. For more information, visit *www.transmission-symposium.com/en*.

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Automotive Transmissions, HEV and EV Drives

7–10 December 2015, Berlin, Germany

1,350 Delegates in 2014

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Panel discussion

What will the drivetrain be like in 2030?
Will the electric vehicle remain a niche model?

Key Note Speakers

Michael Schöffmann | Dr Eckhard Scholz | Gerald Killmann | Dr Robert Fischer | Larry Nitz | Wolfgang Müller-Pietralla | Dr Rolf Leonhard | Dr Wolfgang Reik | Mihir Kotecha | Tetsuya Takahashi | Guillaume Devauchelle | Dr Peter Rieth

Michael Schöffmann
Head of Transmission Development | Audi AG

Dr Eckhard Scholz
Chief Executive Officer (CEO) | Volkswagen Commercial Vehicles

Gerald Killmann
Vice President R&D2 | Toyota Motor Europe

Dr Robert Fischer
Executive Vice President Engineering and Technology Powertrain Systems | AVL List GmbH

Larry Nitz
Executive Director | General Motors

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Tetsuya Takahashi
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Meet Mayhem's Master

An interview with Dr. Donald E. Simanek, curator of 'The Museum of Unworkable Devices'

Jack Mc Guinn, Senior Editor

Power Play patrons may recall last month's drop-in at the **Museum of Unworkable Devices** — a virtual and pretty damn funny (***www.lhup.edu***) "celebration of fascinating devices that don't work."

In researching the story, we found the story of its founder — Dr. Donald E. Simanek, Emeritus Professor of Physics, Lock Haven University of Pennsylvania — as fascinating as his museum. Here's why.

To begin — this bears repeating (from Sept. *Power Play*, "The Museum of Unworkable Devices"). It is what you might call the museum's "mission statement," while also providing a fine example of the professor's sharply honed sense of humor, as he's the person who wrote it:

"The museum (www.lhup.edu) houses diverse examples of the perverse genius of inventors who refused to let their thinking become intimidated by the laws of nature, remaining optimistic in the face of repeated failures."

With that, some questions for the curious curator.

Power Play (PP). To be clear, the museum is your *brain child* yours and yours alone?

Donald Simanek (DS). Yes. It began as a modest collection of puzzles for my physics students in the 1990s. I wanted students to exercise their understanding of physics by finding the flaws in classic PPM (process & packaging) machines, using only elementary physics. The ground rules were:

(1) Friction is never the reason they don't work; remove all friction and dissipative processes and they still won't work.

(2) Don't assert that "The laws of thermodynamics show they can't work." Of course, but that obscures the interesting details of fundamental physics laws. See (1).

I became aware that available books about perpetual motion history frequently glossed over the physics, and sometimes their "explanations" missed the mark. I try to correct that. Yet I try to keep the explanations at a level that can be understood by high school physics students — or even interested laymen.

Some have called me "the world's expert on perpetual motion" — a dubious honor. Actually there are three of us who take this subject seriously enough to discuss it in detail. The others are Hans-Peter Gramatke and Eric Krieg; both have web sites.

PP. Except for book excerpts, who provided the descriptive copy for the various devices, etc. for the site?

DS. I did. Sometimes I constructed the descriptions to enhance the deception. Most of the devices are classics, described in the references I supply. Some were sent to me by email, and were credited when the inventor wanted credit. Some don't. I have received many more than appear on my web pages.

PP. Is this strictly a "virtual" museum? Do you in fact possess any of the devices described or pictured?

DS. For demonstration purposes, I have made a few models. I have described those in the section "Building Perpetual Motion Machines." I encourage building small-scale models, for only then do inventors realize how badly they perform; most won't complete one revolution. One person took my advice and built a mechanical model to test his assumed motive principle. It showed up in a UPS package on my doorstep as "a contribution to your museum." The inventor — a talented machinist — had found it didn't work. It isn't large, made of machined aluminum and magnets. But it did stimulate the invention of an interesting and seemingly unrelated physics puzzle that I may publish in a physics journal.

There does seem to be considerable activity by people who build "fake" perpetual motions just for fun, and put the results on U-Tube to mystify others. Some of these are quite ingenious. They don't try to sell them or defraud anyone. I liken this activity to the deceptions of a stage magician.

PP. What does *The Museum of Unworkable Devices,* Myths, Mysteries and Legends conference of the Committee for the Scientific Investigation of Claims of the Paranormal (CSICOP), October, 2003 refer to?

DS. It refers to one of the annual conferences of CSICOP. This one was held in Albuquerque, N.M. where I was invited to give a one hour talk on perpetual motion machines. It is now called Committee for Skeptical Inquiry (CSI) (*www.csicop.org/)*. I have given similar invited presentations at colleges and universities.

PP. How active would you say the site is these days?

DS. Its expansion is slowing. People still send me their ideas by email, but mostly they are re-inventions of the square wheel — already described in books and even patents — presenting nothing really new or interesting. I'd be surprised to receive anything really original anymore. Once, I put my mind to devising a really original one myself, and thus I created the "Silly Slinky" machine. Not only does it not work, it illustrates several common misconceptions perpetual motions have. The closest thing to a perpetual motion is a simple wheel with frictionless bearing. Whenever you try to be clever by adding anything to that — e.g., swinging weights, rolling balls, articulated arms, gears, pulleys and gimmicks, the performance decreases.

Adding designer hubcaps to square wheels doesn't improve their performance. **PTE**

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