Power Transmission Engineering®

APRIL 2017

BEARINGS

- MAINTENANCE
- COMPOSITE MATERIALS
- RACEWAY DAMAGE

BRAKES

BACKLASH IN MAMMOGRAPHY SYSTEMS

AUTOMOTIVE

MEET OUR BEARINGS BLOGGER

Power Transmission Engineering®

APRIL 2017

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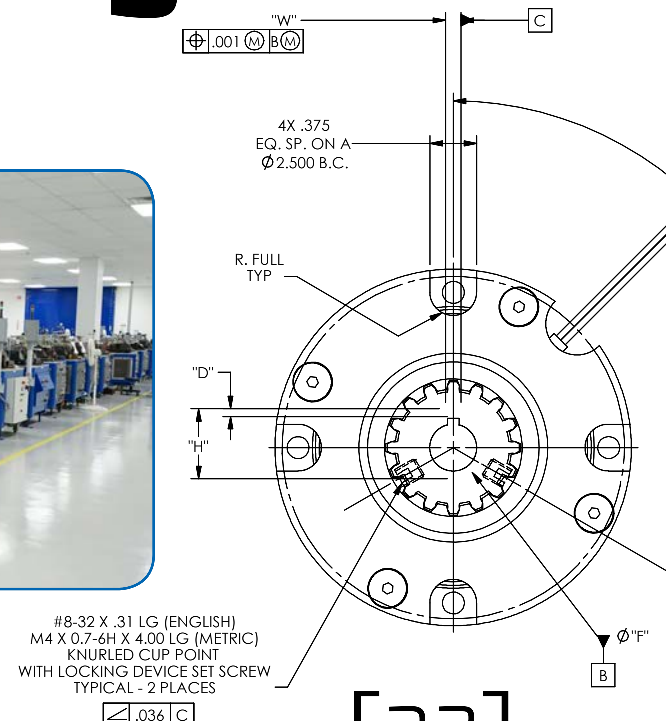

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FEATURE ARTICLES

TECHNICAL ARTICLES

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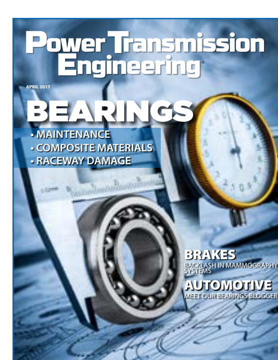
Power Transmission Engineering

VOL. 11, NO. 3



www.powertransmission.com

PTE Videos

Regal Power Transmission Solutions offers the Sealmaster Large Bore Performance Gold Line Mounted Ball Bearings with a patented* TIME SAVING axial groove in the inner ring bore to allow for easier bearing removal and ability to reuse the shafting with minimal clean up. See the video here:

www.powertransmission.com/videos/Regal-Power-Transmission-SealMaster-Bearing-Removal/

KEB's Tooth Clutches provide three times the torque in a given size compared to friction clutches. The clutches are electrically engaged and can be used in both wet and dry operation. Typical applications include medical machinery and test stands. Check out the video here:

http://www.powertransmission.com/videos/Tooth-Clutches-from-KEB-America-/

Event Spotlight: PowderMet 2017

The International Conference on Powder Metallurgy and Particulate Materials will present leading companies featuring the latest PM equipment, powders, products, and services. Over 200 worldwide industry experts will present the latest in powder metallurgy, particulate materials and metal additive manufacturing. For more information, visit: ***www.powertransmission.com/news/8010/PowderMet-2017/***

Motor Matters with George Holling

Georg Holling (PI) is an in-demand consultant to many major U.S. and International corporations for motors and drives as well as a contributing blogger for PTE Magazine. Recent topics include motors in high-temp environments, motor design tools and more. Read his latest entries at:

http://powertransmission.com/blog/category/motor-matters-with-george/

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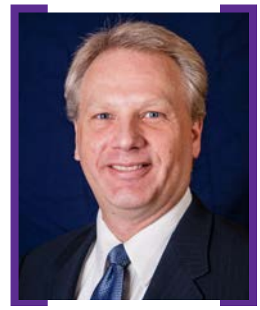


Bearing Good News

This issue's focus on bearings includes a number of important and informative articles about one of the most key components in any machine design.

Some of you are already familiar with our bearings blogger, Norm Parker, who has been writing for us for a number of years. His blog on powertransmission.com includes insights and tips from someone who's in the trenches, specifying and buying bearings every day for a big three automobile manufacturer. But we know that many of you prefer to read your articles in print, and some of Norm's posts are just too good to miss, so we've reproduced some of the recent ones here as a sort of introduction (or reintroduction) to the blog.

We also have a bearings maintenance article from Will Cannon at Baldor Electric. In "The Hidden Cost of Incomplete Hydrodynamic Bearing Maintenance," Cannon explains why less expensive repairs might be much more expensive in the long run. By looking at the maintenance of a bearing over its entire lifetime, it becomes clear that the long-term costs are what's most important.

Leo Dupuis of Bosch Rexroth describes the steps taken to better understand the performance and specification of composite radial bearings used in large hydraulic cylinders. "Performance Testing of Composite Bearing Materials for Large Hydraulic Cyclinders" explores the significance of inclusions in the material structure as well as friction response of different materials.

Finally, we sent Senior Editor Matt Jaster to visit the SKF seals investigation facility in Elgin, IL, where the company investigates the root cause of rotating equipment failure and analyzes how seals react under various environmental conditions.

But as you know, we're not just about bearings here. There's plenty more to read, including articles on clutches and brakes, motor basics and gear design. In this issue's "Ask the Expert" column, *Gear Technology* technical editor and blogger Chuck Schultz answers a reader's question about high-performance gear materials.

Speaking of "Ask the Expert," our team is putting together another version of "Ask the Expert LIVE" to be held at Gear Expo 2017 in Columbus, OH (October 24-26), and one of the sessions will definitely focus on gear design. We're currently putting together our panel of experts and selecting the topics for this live version of our popular column. The first "Ask the Expert LIVE" was held at Gear Expo 2015, and you can see examples of the sessions by visiting *www.geartechnology.com/videos.*

As always, we welcome your submissions of questions for "Ask the Expert" – both the live version at Gear Expo and the print version that appears each issue. If you have technical questions on any power transmission and motion control topic, we'll help you get answers from the most knowledgeable people in the field. Just send your questions directly to Senior Editor Jack McGuinn (*jmcguinn@powertransmission.com*).

But we also encourage you to come to Gear Expo in October. It is definitely not too soon to start making plans. The AGMA has been building the show with each edition, and it has grown to become a significant power transmission show, in addition to being the world's premier gear manufacturing show. We'll be there, and we hope you will be, too.

Randy Stott

ART Technologies

STAMPS OUT DOWNTIME WITH FORCE CONTROL'S OIL SHEAR TECHNOLOGY

In the metal stamping business, precision, repeatability and uptime are key. But stamping accuracy suffers when improper tension on the coil feeders incorrectly supplies metal to the presses, resulting in off-spec parts and increased rejections. ART Technologies relies on an oil shear clutch brake to supply constant, reliable tension on the coil feeding one of their 400-ton presses to give them the precision and repeatability they need, with no downtime for maintenance or adjustment. When the plant is working 20 hours a day, that uptime is as critical as the tolerances they maintain.

ART is a full-service global supplier of precision metal stamping components, thrust bearings and coining services for raceways and washers. With 11 presses ranging in size from 45 to 800 tons, they stamp out a wide range of products for the automotive, truck, bus, solar, HVAC, agriculture, defense and other industries.

They currently operate a single shift, but when business demands it they operate two ten-hour shifts, or nearly around the clock. Whether they are cranking out miniature thrust bearings for the automotive industry or thick stampings for the truck market, repeatability is key in all that they do.

A Pressing Problem

One of their Minster 400-ton presses has a ½ hp motor which pulls the stock strip though the press, keeping tension on the steel at all times. This tension is a necessity because of the Die Design, and ensures consistency of the stamped product and optimal productivity.

Engineering Manager Fred Meinhardt, explains it this way: "We need to keep tension on the stock and to be able to run the drive unit at a speed slightly faster than the feed, so that when the feed stops, the clutch slips. When the feed restarts again, the stock tensioner takes up the slack and keeps tension on the stick as it moves forward."

The company had been using a mechanical clutch that was slipping all of the time, with unsatisfactory results, including feed problems, out-of-spec parts, and press down time. In addition to the production problems, the dry friction clutch would wear and require adjustment, maintenance or replacement, to the tune of four to five hours per week. According to Meinhardt, the production loss due to downtime was 20 percent.

Meinhardt estimates that the old style clutches were replaced every six months or so, in addition to the four to five hours per week of maintenance. At the time, the plant was working two 10-hour shifts, or nearly round-the-clock, so that level of weekly downtime for maintenance and adjustment was substantial and unacceptable. To top it off, the extra time to replace the clutch-brakes was even more troublesome, because the failures rarely occurred at convenient times. Then Meinhardt found help right around the corner, literally.

"My brother works for Force Control Industries (the manufacturer of the oil shear clutch brake) which is within a mile of our plant, so I knew all about their capabilities," he said.

Installing the oil shear clutch break was hassle free, and the little effort required reaped a significant return on investment. In the two years that the Posidyne 1.5 clutch brake has been installed on the ½ hp motor to tension the coil stock, there has been no unscheduled downtime for maintenance or repairs. At two years and running, the Posidyne 1.5 has already lasted 4-times longer than the dry clutch, and is still working fine.

How Oil Shear Technology Works

Normal dry clutch brakes employ a sacrificial surface — the brake disc or pad — to engage the load. Having no good way to remove the heat caused from engagement between the disk and plate, this material must absorb the heat. These extremely high temperatures will eventually degrade the friction material. As the friction surface wears away and begins to glaze, the ensuing torque fade causes positioning errors, which then require adjustment or replacement of the friction surface.

Oil-shear technology plays a major role in ensuring that the coil feeders at ART Technologies operates at peak efficiency — even at a much higher cycle rate. A fluid film flows between the friction surfaces, and is compressed as the brake is engaged. The Automatic Transmission Fluid (ATF) particles in shear transmit torque to the other side. This torque transmission causes the stationary surface to turn, bringing it up to the same relative speed as the moving surface. Since most of the work is done by the fluid particles in shear, by the time the surfaces actually meet or “lock up” wear is virtually eliminated.

In addition to transmitting torque, the ATF also helps to dissipate heat, thanks to a patented fluid recirculation system. Along with torque transmission and heat removal, the fluid also serves to continually lubricate all components — thus extending their service life. Oil Shear Technology also provides a “cushioned” stop that reduces shock to the drive system — further extending service life.

Unlike dry clutch brakes, the totally enclosed oil shear system is impervious to external elements such as wet, dusty or dirty environments, as are common in many manufacturing plants. Since the layer of oil eliminates wear, the Posidyne clutch brake provides a long service life. With elimination of wear comes elimination of adjustment — and increased “uptime” for ART Technologies.

The reliability and durability of oil shear technology helps plants with a critical pathway maintain high production. Oil shear technology has helped ART Technology’s plant increase precise control and stamp out downtime. Production is up 20 percent, with reduced scrap rates, fewer out-of-spec parts, and more parts per coil. The resulting efficiency and profitability keeps ART’s machines precise and reliable—giving them a competitive edge in a competitive industry.

For more information:

ART Metals Group
Phone: (513) 942-8800
www.artmetalsgroup.com

Force Control Industries
Phone: (800) 829-3244
www.forcecontrol.com

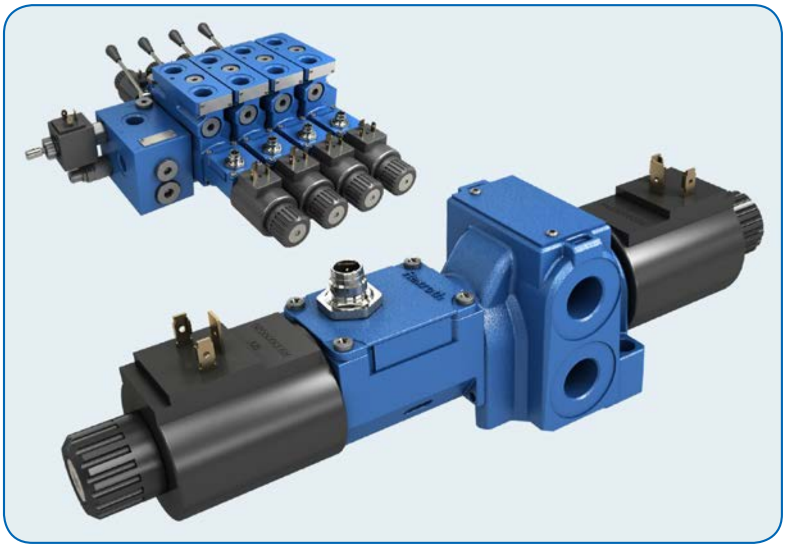
Bosch Rexroth

LAUNCHES NEW SENSOR FOR COMPACT DIRECTIONAL VALVES

Safety is vital when working with material handling equipment, especially when working with massive suspended loads. A confirmation of hydraulic directional on-off or proportional valve shifting is one step in improving the overall safety integrity level (SIL level) of the machine. Bosch Rexroth's new sensor provides reliable monitoring of spool position and satisfies all international standards in safety reliability engineering.

Bosch Rexroth's new compact directional control valve sensor monitors spool state of shifting in both mobile and industrial applications. The solid-state Hall-effect sensor, which is directly installed between the valve body and solenoid coil, reliably monitors any movement of the valve spool. They are available in digital for on-off directional valves and in analog and ratiometric for proportional directional valves.

This sensor offers improvements in both safety and reliability engineering, while having no negative effect on either valve switching times or hysteresis. In addition, the sensor is a plug and play component, using a standard M12 connection with 5 pins and IP69K protection rating for mobile applications.



For more information:
Bosch Rexroth
Phone: (800) 739-7684
www.boschrexroth.com

SEPAC

CLUTCH SERIES DELIVERS TORQUE IN OIL ATMOSPHERE

SEPAC designed the Multiple Disc Wet Clutch (ERD Series) to deliver a reliable, consistent torque within an oil atmosphere. This model offers a ball bearing-supported design to ease the installation process without relying solely on the customer for bearing support. The ERD Series is virtually maintenance-free under proper operating conditions as most of the components are hardened.

The ERD's design allows the clutch to automatically compensate for wear of the disc pack ensuring long life and the ease of installation make it ideal for a variety of applications. The unique disc design also reduces drag, as well as wear when operating de-energized. Additionally, the extraordinary reliability of the ERDSeries makes it a top choice for fan clutches used in heavy and/or armored vehicles, motion control for flight system actuators, critical door operating actuators in industrial plants, or heavy duty machinery.

The magnet body of the ERD is typically installed on a motor orgearbox shaft with the option for a hub, coupling, gear, pulley or sprocket mounted to the spider or clutch cup. When current is applied to the coil, magnetic flux flows through the discs to the armature, which causes the outer discs to be squeezed between the inner discs, which transmit torque from the rotor input to the spider output. The magnet body of the ERD is held stationary by the means of an anti-rotation screw or post anchored to a hole provided. Coils are fully encapsulated in epoxy protecting them from the oil and/or harsh environment the clutch may be subjected to and a sealed connector is standard.

For more information:
SEPAC
Phone: (800) 331-3207
www.sepac.com


BK
THE COUPLING. ABSOLUTE
PRECISION 0.1–100,000 NM.
R+W
Made in Germany
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A POPPE + POTTHOFF COMPANY

R+W

SAFETY COUPLINGS SUITABLE FOR WASTEWATER APPLICATIONS

By popular demand, R+W has developed a standard model of corrosion proof ball detent safety coupling for clarifier drives in water treatment plants. The industry requires a well sealed, high grade stainless steel mechanism to stand up to harsh conditions, resulting in the specific design criteria. The function of the ball detent torque limiter is to disengage the driveline to protect flights, chains and sprockets in the event of overload, which can result from the high gear reductions associated with the slow speed movements required in the settling tanks.

Two standard body sizes are available, with a maximum bore diameter of 2.5" and disengagement torque settings ranging from 1,600 to 21,000 in*lbs. Custom options are always available on request. The new R+W safety coupling design features: reliable, precise, and consistent overload protection, high grade stainless steel construction, adjustable and tamper proof options, optional switch plate and/or extension hub and torque overload protection for clarifier drives.

For more information:
R+W America
Phone: (630) 521-9911
www.rw-america.com

Miki Pulley

MICRO CLUTCHES DESIGNED FOR COMPACT PRECISION APPLICATIONS

Miki Pulley is introducing its CYT Micro Clutches for direct sale to OEM's in North America. These CYT Clutches are designed for compact precision applications and may be easily mounted to a driven shaft. Further custom variants can be realized by pairing a sprocket, timing pulley, V-belt pulley or shaft to the armature.

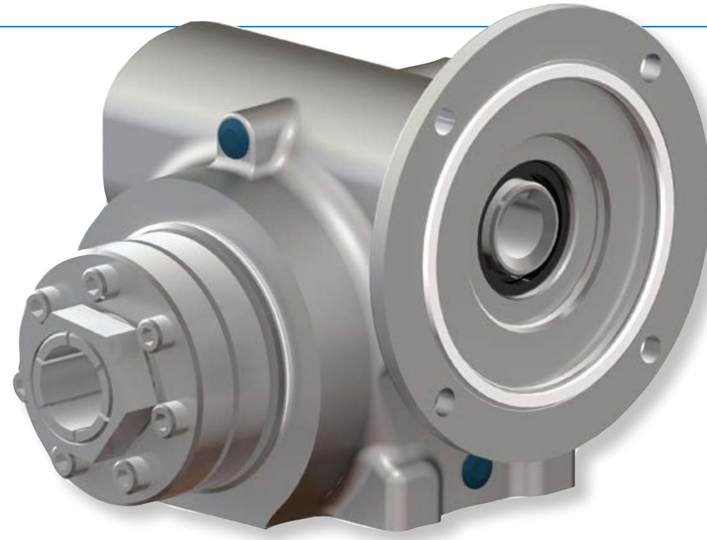
Miki Pulley CYT Clutches accurately connect and release power by being located between the input shaft and the load. The CYT stator is a bearing mounted type of clutch. It provides an efficient connection between a motor and a load with low inertia, minimal drag and long service life.

Two CYT Clutch models are available to accommodate different rpm ranges: a dry metal type and a ball bearing type. In addition, three types of armature configurations are available for pulleys, gears and for combining both on shafts.

Miki Pulley CYT Clutches are durable and versatile with a straightforward design, consisting of clutch stator, rotor and armature. The clutch assembly features an integrated bearing design making mounting fast and easy while ensuring application concentricity and

Winsmith

DISPLAYS STAINLESS STEEL CONVEYOR DRIVES



Winsmith recently participated in the International Production & Processing Expo (IPPE) in Atlanta, GA, January 31–February 2. A manufacturer of high performance industrial gearing technologies, Winsmith displayed an upgraded stainless conveyor drive that features FDA-compliant blue plugs and a new Keyless Shaft Locking Device.

Launched in 2015, Winsmith's new stainless conveyor drive is IP69K certified to withstand high pressure, high temperature wash down and contamination. It is the ideal speed reducer for food processing, and demanding industry applications including fruit, vegetable, and meat processing. The stainless conveyor drive is available in five case sizes and more than 15,000 configurations. It features a single-piece design and sealed housing, facilitating easy cleaning, continuous operation and safe wash down. All lubricant and drain plugs are easily accessible for maintenance checks and have been fitted with new FDA compliant blue plugs that are both magnetic and x-ray detectable. These plugs ensure a smooth, bolt-free housing and have been pressure wash tested to 1450 psi.

excellent system runout. CYT Clutches operate well in temperatures from +14°F to +104°F (–10°C to +40°C). They have a speed range up to 3,600 rpm.

Available in bores ranging from 6 mm to 10 mm, with clutch torques ranging from 0.3 ft. lbs. to 0.74 ft. lbs. (0.4 Nm–1.0 Nm). The CYT Clutch utilizes corrosion resistant materials, and is RoHS compliant like all Miki Pulley products.

"Miki Pulley's CYT Clutches are ideal for compact systems requiring precision operation including printing equipment, packaging machines, and web handling applications." reports Jon Davidson, Miki Pulley sales specialist. "They are very reliable making them a preferred choice of motion system designers throughout the manufacturing world."

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

The Winsmith KLD shaft locking device is an upgrade that benefits food and beverage processors who want to simplify equipment maintenance. The keyless locking system can be installed in minutes and reduces disassembly times by mitigating fretting corrosion, as well as downtime for field servicing.

For more information:
Winsmith Inc.
Phone: (716) 592-9310
www.winsmith.com

Ogura

INTRODUCES ACSB CLUTCH/BRAKE SERIES TO NORTH AMERICA

Ogura Industrial is pleased to announce a new addition to our product line. Although this product is not new to Ogura, it is new to North America. The ACSB series is finding new opportunities for machinery manufacturers in North America because of some of its unique features.

ACSBs are both a clutch and a brake combined into one unit. When the clutch is disengaged, the brake is on and when the brake is disengaged, the clutch is on. The ACSB series uses multiple discs to transmit high torque in a small diameter. All units have a built in adjustment for wear via adjustment nuts. Although primarily designed for industrial punch press applications, the ACSBs can be used wherever high torque pneumatic clutch/brake is required.

Ogura has been producing clutches and brakes since 1938. Over that time, we have developed over 5,000 different models of clutches and brakes. Although Ogura primarily produces electromagnetic clutches and brakes, we also produce magnet particle, mechanical, pneumatic, hydraulic and a variety of specialty products.

Ogura is the world's largest manufacturer of electromagnetic clutches and brakes. Current manufacturing capacity is over 30 million units per year. To provide localized support, we have manufacturing plants spread throughout the world in Asia, The Americas, and Europe. All manufacturing facilities are ISO recognized and conform to the ISO 9001;2008, ISO 140001, and ISO/TS 16949.

For more information:
Ogura Industrial Corp.
Phone: (732) 271-7361
www.ogura-clutch.com

Parker

OFFERS MINIATURE, PRECISE DIRECT-DRIVE ROTARY STAGE

Parker's Electromechanical and Drives Division - North America is pleased to announce the release of the mPR (miniature precision rotary) stage. The mPR series is a miniature, precision direct-drive rotary stage with integrated high-precision rotary bearing, high-resolution rotary encoder, and 3-phase AC servo motor. It has been engineered to deliver a combination of modularity, flexibility, and performance in an extremely compact package. "The mPR's many features make it the ideal positioner for a number of high-tech industries," says Travis Schneider, product marketing manager of precision mechanics. "Applications in metrology systems, laser processing/machining, electronics manufacturing, and semiconductor manufacturing, for example, will benefit from the stage's combination of size, low profile, and high precision." The mPR 80 has common mounting features to that of Parker's MX80 linear stage, as well as the mSR 80/100, and the mPR 100 size has common mounting features to that of our 404XR linear positioner so users will be able to quickly and easily create complete multi-axis systems using the mPR. Standard features of the mPR Series include: two form factors, mPR 80–80 mm dia. × 75 mm, mPR 100–104 mm dia. × 90 mm tall, continuous 360-degree operation, four encoder resolutions (3-digital incremental, 1-analog sine/cosine), dowel holes in top and base for repeatable pinning, through-hole aperture, direct mounting features to Parker MX, mSR, and XR stages, three-meter, high-flex cables and lightweight aluminum construction.

For more information:
Parker Hannifin
Phone: (704) 588-3246
www.parker.com

Bonfiglioli

DEVELOPS WHEEL-TRACTION DRIVES WITH FULL HYBRID TECHNOLOGY

Bonfiglioli has announced the development of wheel-traction drives of the 600 series specifically designed for Tigon Technology backhoe loaders manufactured by Huddig. Each of the special four-wheel traction drives is capable of reaching an output torque of 40.000 Nm, and is driven by a liquid cooled 30 kW electric motor. The independent control of each wheel gives the opportunity for the backhoe loaders to be driven better on slippery surfaces. The Bonfiglioli 610 drives also increase the precision of the machine, making the maneuverability easier in tight spaces, for example in building sites located at city centers. Following Huddig requirements, the four drives are perfectly quiet.

Tigon Technology by Huddig is a hybrid technology that combines diesel and electric power allowing energy generation and regeneration in a way that has never been done before for construction machinery: the concept machine achieves significantly lower fuel consumption than a traditional backhoe loader. The new Huddig hybrid backhoe loader (based on this technology and equipped with Bonfiglioli four-wheel traction drives) is more precise and more efficient without producing emissions.

"Our expertise in final drive technology and electromobility solutions made Bonfiglioli a desirable partner for Huddig to develop a drive technology that would meet its requirements for innovation and technical expertise for the construction industry," said Stefano Baldi, sales director for the Mobile Industries.

For more information:
Bonfiglioli
Phone: (859) 334-3333
www.bonfiglioli.com

Siemens Industry, Inc.

INTRODUCES SINAMICS V20 SMART ACCESS WEB SERVER MODULE

Siemens announces the launch of its Sinamics V20 Smart Access web server module, designed to mount directly onto the drive, transforming a mobile device or laptop into a virtual operator panel for drive control. By providing a wi-fi hot spot, the wireless connection on this module facilitates setup, programming, commissioning, production monitoring and maintenance on a variety of machines and production equipment.

A simple, embedded graphical user interface (GUI) enables easy use of the Sinamics V20 in every phase of operation. No separate app is required, nor is a written operator manual needed, making operation of this new server module and subsequent drive control highly intuitive and easy-to-learn.

Smart Access provides convenient access to the Sinamics V20, up to 100 meters away, even when the drive is located in difficult-to-access installations. Utilizing WPA2 security, the web server module offers full flexibility with both iOS and Android operating systems, along with commonly used HTML5-capable web browsers such as Chrome, Safari, Internet Explorer and others.

A built-in, multi-color LED provides quick communication status readout. Security features enable limit / restrict operator access and control functionality.

In use, the Sinamics V20 Smart Access module requires only a few steps to set-up and no installation or download of additional software is needed. The onboard Quick Set-up Wizard provides users a fast and easy commissioning procedure, enabling all the following: motor data can be entered and checked, connection macros for digital inputs/outputs can be activated, application macros can be selected and activated for pumps, fans, compressors and other devices plus the common and frequently used parameters on the drive can be set for motor start, acceleration, deceleration, min./max. speed, etc.

Smart Access allows monitoring of the drive status including speed, current, voltage, temperature and power, as well as drive servicing, with an overview of alarms, faults and individual values. Fault codes can be transferred via e-mail to a local service provider, while the immediate status of all digital and analog inputs and outputs can be checked at a glance. Parameter adjustment, motor test functions and full data back-up, storage and sharing with fast firmware downloads can all be accomplished via the web server.

For more information:

Siemens Industry, Inc.
Phone: (847) 640-1595
www.usa.siemens.com/sinamics-v20

Cooper

INTRODUCES QUICK-CHANGE BEARING PEDESTALS

New SAFQ inch-series Cooper Quick-Change angled bearing pedestals are now in stock for immediate delivery as quick change-out replacements for industry-standard pillow block housings. Their unique design consists of two split halves enabling easy assembly around a fixed shaft. In addition, the pedestal's innovative angled bottom-half simply slides under a shaft – even one with low clearance – without requiring a jack or hoist. Typical applications include fans and similar rotating machinery in the mining, marine, steel, power generation industries, and many others.

The SAFQ Series pedestals are available for shafts with diameters from $2\frac{3}{16}$" to $5\frac{5}{16}$" and join the SNQ and SDQ metric series of Cooper brand solutions as practical and time-saving alternatives to standard pillow-block housings (SAF500, SN500, and SD3100 series). All integrate standard Cooper 01, 01E, 02, or 02E split-to-the-shaft bearings and are interchangeable with other Cooper split roller bearing pedestals.

Among key pedestal advantages, the split-to-the-shaft roller bearings simplify disassembly and reduce time needed for installation, changeover, and inspection; an exclusive Aluminum Triple Labyrinth seal remains concentric and rotating on the shaft to prevent ingress of contaminants, even under water; spherical cartridge and pedestal design compensate for misalignment; a clamped inner race protects against shaft wear and the absence of set screws prevents shaft damage; and a polished outer race and rollers allow for dynamic axial expansion to minimize resistance and stress on other components.

For more information:

Cooper Split Roller Bearing Corp. (An SKF Brand)
Phone: (757) 460-0925
www.cooperbearings.com

FEATURE

Meet Norm Parker — Bearings Blogger

In case you missed them, following are three recent blog postings by our popular *PTE* bearings blogger — Norm Parker. We also felt that, should you not be a blog follower, this would be a good way to introduce you to Norm's bearings wisdom. Parker is currently the global senior specialist/roller bearings at Fiat Chrysler Automobiles (FCA). With his bachelor and master degrees in mechanical engineering from Oakland University (Rochester, Michigan), Parker has developed a keen interest in the academic, commercial and engineering aspects of the bearing industry. Prior to joining FCA, he rose through the ranks of traditional bearing companies and served as bearing technical specialist for the driveline division at General Motors. He is a regular contributor to *Power Transmission Engineering* Magazine, appearing often in the publication's popular Ask the Expert feature, as well as authoring a number of bearings-oriented feature articles.

Demystifying Bearing Fit Practices

Posted January 16, 2017…

I had a couple of recent conversations with different people regarding ball bearing fit practices. We have covered this before, but these issues reminded me that the topics of fit and clearance are always fair game. If you enjoy ambiguity and uncertainty, there are few better places to start than fitting a small ball bearing with tight clearance. Just to keep things simple, I'll use Koyo's (JTEKT) main Ball & Roller Bearings catalog for the technical information. All full-size bearing catalogs will have the same information.

The place to start when determining fit is with clearance. When dealing with ball bearings, the term "clearance" always means radial internal clearance, whereas when we preload tapered bearings, we are usually talking about axial preload/endplay (Fig. 1). For your reference, axial clearance is around 10× the distance of radial clearance.

Let's just walk through a quick example with my favorite bearing, i.e. — 6205. When ordering these you must know what clearance you are ordering; this will usually be indicated somewhere in the title. Even though there is a "normal" clearance designated with CN, C3 is by far the most common clearance.

Ok great — you now have a C3 6205; what does that mean? Fortunately, ISO ball bearings are very standardized, so you can find clearance tables similar to what you'll find in Table 1 just about anywhere online or in any catalog; and it will be the same for any brand. Since a 6205 has a 25 mm bore (multiply the last digit by 5), under the C3 column you see that we have a clearance range of 13–28 microns (µm). This is basically how much room we have to work with. Unlike tapers, we do not want to preload a ball bearing. There are a few situations with lightly loaded bearings where you can add a light axial preload via spring or similar method; but we will never try to radially preload a ball bearing.

For most normal ball bearing applications you will want an interference fit on one ring and leave the other ring loose

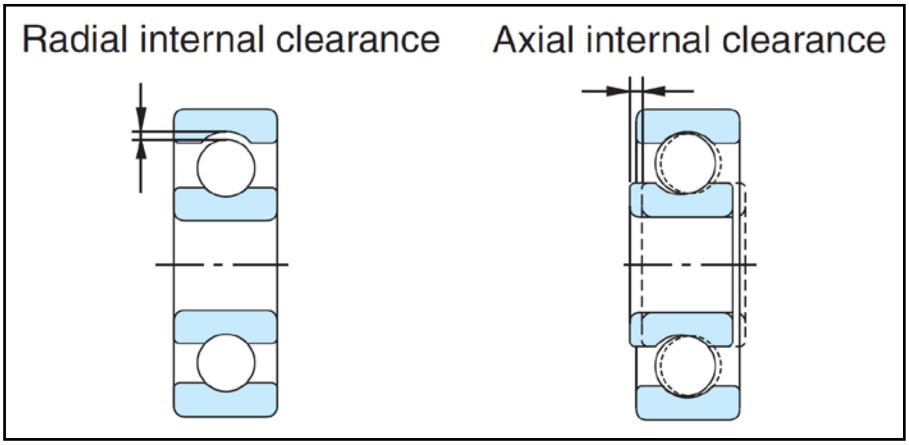


Figure 1 Distinguishing bearing radial internal clearance from preload tapered bearings (axial preload/endplay).

or leave the other ring as a loose fit (aka clearance fit). There are three reasons for this: 1) there is not enough internal clearance in the bearing to press both rings; 2) one slip ring prevents over-constraining the bearing system; and 3) without special tooling, you will need to press one of the rings through the bearing, which is a *huge mistake* (e.g., pressing the bearing into the housing with the shaft). You will damage the bearing by doing this. If you need

Table 1 Radial internal clearance of deep-groove ball bearings (cylindrical bores)

Nominal bore diameter d, mm		Clearance (in μm)									
		C2		CN		C3		C4		C5	
over	**up to**	**min.**	**max.**	**min.**	**max.**	**min.**	**max.**	**min.**	**max.**	**min.**	**max.**
2.5	6	0	7	2	13	8	23	14	29	20	37
6	10	0	7	2	13	8	23	14	29	20	37
10	18	0	9	3	18	11	25	18	33	25	45
18	24	0	10	5	20	13	28	20	36	28	48
24	30	1	11	5	20	13	28	23	41	30	53
30	40	1	11	6	20	15	33	28	46	40	64

Table 2 Recommended shaft fits for radial bearings (classes 0, 6x and 6)

Conditions [1]		Ball bearing		Cylindrical roller bearing Tapered roller bearing		Spherical roller bearing		Class of shaft tolerance range	Remarks	Applications (for reference)
		Shaft diameter (mm)								
		over	up to	over	up to	over	up to			
Cylindrical bore bearing (classes 0, 6×, 6)										
Rotating inner ring load or indeterminate direction load	Light load or fluctuating load $\left(\frac{P_r}{C_r} \leq 0.06\right)$	–	18	–	–	–	–	h5	For applications requiring high accuracy, js 5, k 5 and m 5 should be used in place of js 6, k 6 and m6.	Electric applicances, machine rools, pumps, blowers, carriers, etc.
		18	100	–	40	–	–	js6		
		100	200	40	140			k6		
		–	–	140	200	–	–	m6		
	Normal load $\left(0.06 < \frac{P_r}{C_r} \leq 0.10\right)$	–	18	–	–	–	–	js5	For single-row tapered roller bearings and angular contact ball bearings k 5 and m 5 may be replaced by k 6 and m 6, because internal clearance reduction due to fit need not be considered.	Electric motors, turbines, internal combustion engines, wood-working machines, etc.
		18	100	–	40	–	40	k5		
		100	140	40	100	40	65	m5		
		140	200	100	140	65	100	m6		
		200	280	140	200	100	140	n6		
		–	–	200	400	140	280	p6		
		–	–	–	–	280	500	r6		
	Heavy load or impact load $\left(\frac{P_r}{C_r} > 0.10\right)$	–	–	50	140	50	100	n6	Bearings with larger internal clearance than standard are required.	Railway rolling stock axle journals, traction motors
		–	–	140	200	100	140	p6		
		–	–	200	–	140	200	r6		

Table 3 Nominal shaft diameter (mm)

Nominal shaft dia. (mm)												
over	up to	k 5	k 6	k 7	m 5	m 6	m 7	n 5	n 6	p 6	r 6	r 7
3	6	+6	+9	+13	+9	+12	+16	+13	+16	+20	+23	+27
		+1	+1	+1	+4	+4	+4	+8	+8	+12	+15	+15
6	10	+7	+10	+16	+12	+15	+21	+16	+19	+24	+28	+34
		+1	+1	+1	+6	+6	+6	+10	+10	+15	+19	+19
10	18	+9	+12	+19	+15	+18	+25	+20	+23	+29	+34	+41
		+1	+1	+1	+7	+7	+7	+12	+12	+18	+23	+23
18	30	+11	+15	+23	+17	+21	+29	+24	+28	+35	+41	+49
		+2	+2	+2	+8	+8	+8	+15	+15	+22	+28	+28

to constrain the loose ring, you need a mechanical retainer — such as a snap ring.

When possible, the rotating ring should have the interference fit. The only reason for this is that the turning ring is most likely to try to walk around the shaft due to inertial effects. Some light walking or creeping is harmless as long as you aren't moving material or creating heat. For our 6205 example, let's say we are pressing onto a rotating shaft, which carries a stationary load.

Table 2 gives us a recommendation, but we still have more work to do. If we are working with a normal load (6–12 percent of the dynamic load rating), a k5 shaft fit is recommended. Usually hidden somewhere in the appendices there will be standard shaft and housing fit tables. As shown in Table 3, our 25 mm shaft diameter has a k5 fit of +2/+11 µm. These dimensions are applied to the diameter class of the bearing (not the average diameter). For instance, a 6205 has an inner ring diameter tolerance of -10 µm, leaving the true average around 24.995 mm, rather than 25.0. Fit tolerances are applied to 25.0 mm. For our 6205 the recommended shaft is 25 +2/+11. Sometimes unilateral tolerances can make for easy proofing (rather than 25.002/25.011); your choice.

You may quickly realize that you have been handed a 9-micrometer tolerance to work with. This is tighter than the bearing tolerance that you are buying and, for most places, unrealistic. A k7 or k8 is the more likely reality for most places. You can see that all of the fits in the k series have a minimum press of 2 µm. I will often use 5 µm as a minimum threshold for a "press fit," but I'm not going to argue about 3 µm. Where the hand-wringing starts is when we look at the stack-up for these fits. Again recalling that our inner ring has a –10 µm tolerance, a k5 leaves with a fit range up to 21-µm interference. All things considered, the residual clearance ranges from 11 µm interference to 26 µm clearance.

But what about the clearance?

Ok, there *is* a slight caveat; you can have a *little* radial preload before you fall off the edge. The reason I recommend not trying to design this in is, as you can see, trying to avoid any chance of preload would leave you with a very loose shaft on the other end; that will create problems for you. As we are threading this needle, the small amount of potential preload at the limits can be tolerated (more so than an excessively loose shaft).

Now let's see how this looks with a more realistic range. For me, I would target about +5/+30 for this application. That puts our effective clearance at 16 µm interference to 24 µm clearance. If you are looking for a rule of thumb for how far you can play this game, I like to have my clearance range about 2x the interference range. Statistically, this will rarely get you into the fringes of your clearance window. My +5/+30 would be a few microns on the tight side, but with a bearing that has a healthy life margin, there is nothing to worry about. If I were pushing the life limits of the bearing, I might back down to +2/+27. Many bearing suppliers will agree with this approach; some get nervous when relying on statistics. This is just the reality of bearing fits.

The housing fits are quite a bit easier. Start with a line to line fit and let your tolerance decide the upper end. So for my 6205 with a 52 mm outer diameter I am going to set my lower housing diameter at 52 and the upper end is going to be whatever I can hold. Easy-breezy.

Follow these simple rules and your fits will be perfect:

Know your bearing clearance, dimensions and tolerances up front.

Press fit one ring (preferably the turning ring) and slip fit the other. Follow the tables for guides, but also double-check your stacks to make sure you aren't running more than ~ 30% into potential preload.

Start line to line on the housing. Too loose of a housing can create alignment and/or noise problems. Sometimes too close of a housing fit can be difficult to install. Adding a little oil to the outer ring is common practice. Leave opening up the diameter as a last resort.

PRELOAD!

Posted December 8, 2016…

There is never a shortage of challenging and interesting bearing topics to discuss. Let's begin with everyone's favorite topic — *PRELOAD*!

I eventually will turn this into a full article, but I wanted to throw this out to see if anyone has had any experience with modeling differential housing preload in a somewhat flexible housing. I am working on trying to develop a true analytical model opposed to the usual method of start with some and then add more as needed.

In this application, we physically stretch the housing, insert the shims and bearings and then release the housing in hopes that we hit the correct preload — preferably the first time. The preload is verified by measuring the torque required to turn the differential with the preloaded bearings. This result is compared to a known bearing torque vs. preload relationship. In theory, our shim selection would give us the correct amount of housing stretch to achieve our desired preload. Often is the case, we find the analytical model and reality do not match and adjustments need to be made on the shop floor. What makes this problem trickier than it appears at first glance is the bearing compression vs. the housing stretch relationship and that the housing does not stretch uniformly. Keeping in mind that this entire process will take place within 0.2 mm, a few microns of lost deflection results in missing the preload window.

As we dive into this problem, we will need to have accurate housing and bearing stiffness measurements — preferably analytical and physical. Usually, the shim and differential contribute negligible amounts of deflection but is good practice to include them in the initial model.

The model shown in Figure 2 is where we begin. Some interesting conversations came up as we discussed where and how the differential is fixtured. We will get into all of this and more in a full article.

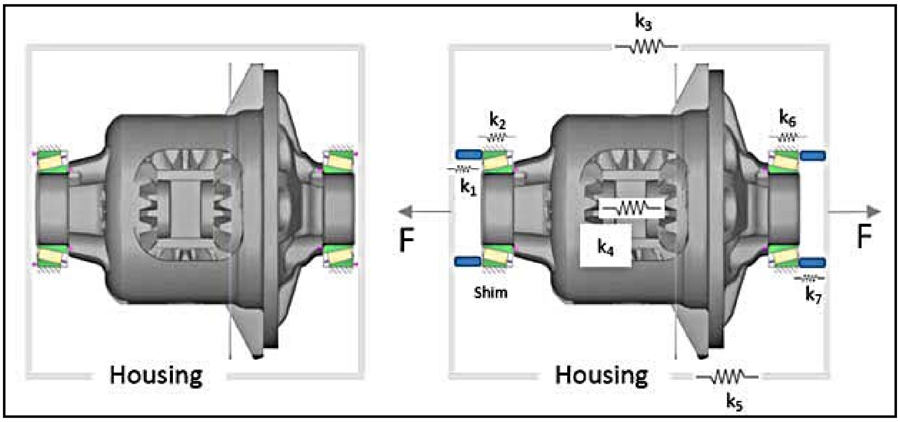


Figure 2 Where and how differential is fixtured.

Avoiding Bearings Made in "Wherever"

Posted February 21, 2017…

You can always tell when times are good in the automotive market due to the flurry of acquisitions, mergers and divestitures. As we are coming off a record 2016 in vehicle sales, many companies are flush with cash burning a hole in their billion-dollar pockets. There isn't enough front-page space for the number of companies reporting record profits for 2016. Others, however, realize that if they didn't see black in 2015–16, the writing is on the wall and it is likely time to start looking for a willing buyer ready to spend some new money on a discounted, struggling company.

As they say around here, "If you can't beat 'em, buy 'em."

In Detroit we saw some huge, industry-changing acquisitions, with AAM buying Metaldyne; Dana buying part of USM; and FCA looking for a buyer while GM is in THE process of dumping Opel. Many companies are doubling down on domestic manufacturing; meanwhile, Chinese companies continue to aggressively invest in U.S. markets.

All of this churn in the marketplace can make it tricky to differentiate the real competition from dotted line allies. As we deal with the Big 6 global bearing companies — Timken, SKF, NTN, NSK, JTEKT, Schaeffler — and an ever-increasing powerhouse — Nachi, along with the emerging Chinese behemoths ZWZ, LYC, HRB, TMB, Wanxiang, C&U and CW — you may be frequently left with the question of who exactly is doing what. Out of the thousands and thousands of bearing companies, (someone told me there were around 9,000 bearing companies in Ningbo China alone) only a small fraction is actually *producing* bearings. Indeed, the companies mentioned above produce well over 90% of the world's bearings. Of course, many, many smaller bearing companies do produce their own products. In global terms however, those numbers make up a very small part of the market. Supporting this ~ $70 billion roller bearing marketplace are what I would call assemblers, partial manufacturers, resellers, rebranders and distributors.

Now, more than ever before, it is common for bearing companies to purchase components that are in the least bit unfavorable to the bottom line. In this incredibly competitive marketplace the public companies need to show profitability to keep the stock price stable. This often means that if they are faced with the option of producing a custom ring at a single digit or flat rate of return, they are likely to shop out the business to another manufacturer. These deals are usually kept very close to the vest for obvious reasons, a major one being that if you spent tens of millions of dollars in marketing to tell people that your product is the best, people expect it to be your product.

The point I am making is the need to hold your suppliers accountable. Often a large supplier will not tell you whom they are buying your components from, but you absolutely have the right to know where your parts are being manufactured. If you are paying top dollar for what you believe to be top quality steel from a top quality producer, you don't need to be shy about asking questions. If you can't get a plant tour of where your bearings are being manufactured, there is a good chance it is because they are buying it somewhere else. If they are buying cheaper offshore rings — and then finishing them in a high-priced manufacturing country so that they can stamp them with Made in "Wherever" — you have some work to do, my friend. **PTE**

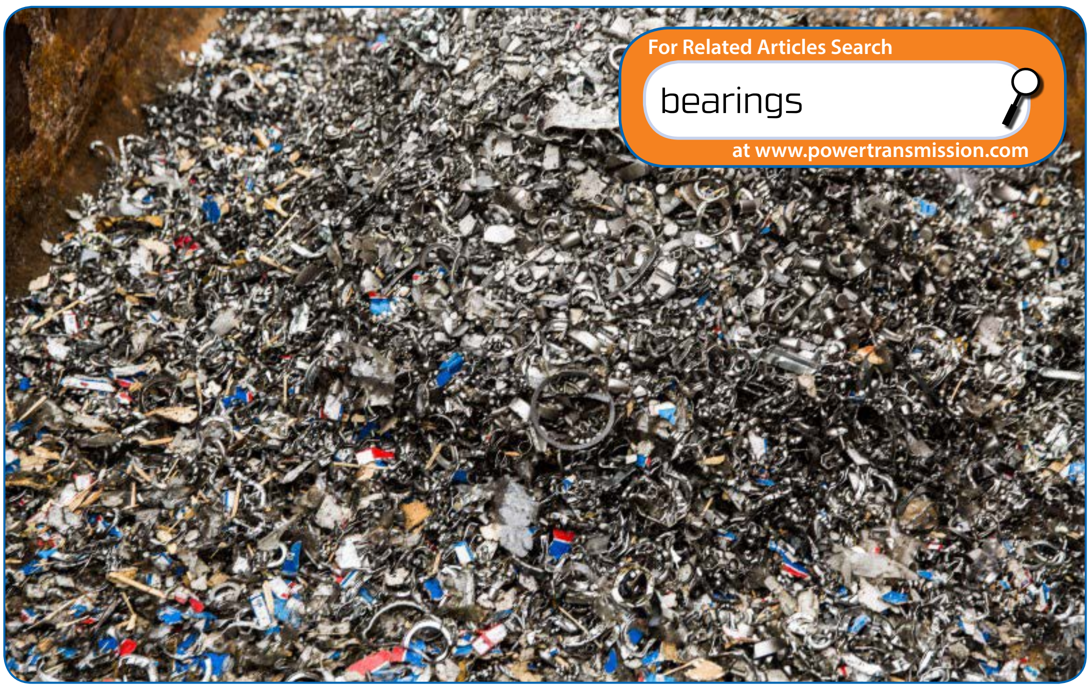


Photo courtesy of SKF

Norm Parker is currently the global senior specialist - roller bearings at Fiat Chrysler Automobiles (FCA). With his bachelor and master degrees in mechanical engineering from Oakland University (Rochester, Michigan), Parker has developed a keen interest in the academic, commercial and engineering aspects of the bearing industry. Prior to joining FCA, he rose through the ranks of traditional bearing companies and served as bearing technical specialist for the driveline division at General Motors. He is a regular contributor to *Power Transmission Engineering Magazine,* appearing often in the publication's popular Ask the Expert feature, as well as authoring a number of bearings-oriented feature articles and *The Bearing Blog*.

The Science of Radial Shaft Seals

SKF Product Investigation Center Troubleshoots Critical Rotating Equipment Applications with Analysis, Research and Testing Procedures

Matthew Jaster, Senior Editor

It's hardly a stretch of the imagination to walk into the SKF Product Investigation Center (42 miles northwest of Chicago) and think about a crime lab. Though the employees here have titles like material development engineer, investigation administrator or application engineer, they're all detectives in the grand scheme of things. Instead of solving crimes, they're tasked with determining why seals fail. They take this data and share it throughout SKF in order to make their products safer and more efficient. It's also a pivotal step in developing new seal technologies.

Bryan Uncapher, seal business development at SKF, says that most rotating equipment will work until bearing failure, which is greatly related to seal performance. "If it is a grease application, external contamination can enter the bearing and cause corrosion or wear, which will lead to system failure or stoppage. If oil is the main lubrication, the oil will eventually leak out, which causes metal to metal contact, which also leads to bearing failure. This can result in a great amount of repair and loss performance expense for the operator," he said.

The investigation center was created to determine the root cause of rotating equipment failure, analyze how the seal reacts under certain environmental conditions and build a database (or backlog) to classify typical failures for future reference. Here, in a maze of testing stations, analytical labs and manufacturing work cells, the former Chicago Rawhide headquarters (see sidebar) has the look and feel of a highly-technical research center where seal failure is literally put under the microscope.

The Investigation Process

Seals have a crucial impact on system performance. They need to provide a minimum amount of friction and wear while maintaining maximum protection from external environmental contamination and retain bearing lubrication under a variety of unique and hazardous conditions. When they fail,

The SKF Investigation Center was created to determine the root cause of rotating equipment failure, analyze seals under different environmental conditions and build a database to classify failures for future reference. Courtesy SKF.

machines, operators and components are all at risk.

The SKF Product Investigation Center has a very thorough (and organized) procedure to examine each seal case.

"The advantage of our new product investigation system is that it offers a systematic approach to define the root cause of failure and identify possible solutions. It ensures that our application engineers can collect the critical field data and work according to a scientific process with our analytical, chemical and validation labs to identity the causes of failure and offer the right solution for our customers," said Bouchra Le Hir, global product testing & investigation manager at SKF.

So, what exactly happens when a customer sends a seal into the investigation center? First, it's the job of application engineers like Mark Haughey to examine the seals, collect all the pertinent information (any damage analysis, application history, etc.), conduct initial inspections and report key findings.

This includes examining the packaging materials the seal was shipped in as well as checking for any contaminations. The engineers will also look to see if the damage was simply a case where the operator dropped the seal on the floor prior to installation (which happens more than you think).

Most of this initial research is considered a Level 0 investigation.

A Level 1 investigation includes taking photos of the seal (courtesy of a light box assembly created by Philip Sajor), and an initial comprehensive bearing seal inspection. Dave Zimmerman, investigation manager, is one of the first in the building to provide this detailed inspection.

Zimmerman begins with a small pile of plastic bags at his workstation. These bags hold various seals from industries like automotive, off-highway equipment, steel mills, rail, and mining applications (to name a few). Each seal tells a different story and provides Zimmerman with clues as to why the seal failed in the first place. There are several different causes of seal failure including manufacturing defects, assembly errors, compatibility with the lubrication or working fluid, and/or system design flaws.

Seal failure can sometimes be determined by simply taking the seal out of the plastic bag and examining it with the naked eye. Other cases might need a combination of microscopes, photographic images, hand tools and 38 years of bearing seal experience to help solve the mystery. Zimmerman has all four.

A seal customer may submit a performance claim (citing the seal product itself as the cause of failure) on many seals that come across Zimmerman's desk even though it shows obvious signs that it was damaged by the installer.

Zimmerman said one of the most common modes of failure is improper installation, where the seal case is hit with a metal socket which changes contact forces, backward installation, misalignment, and cut lips. What is very interesting is that operation errors are one of the easiest methods to detect if samples are provided.

If that's still not enough, the seal moves to Sajor and the SKF measurement team in a temperature-controlled room for a variety of evaluations. Then it is moved to the next levels of inspection (Level 2: the analytical lab with Albrecht Becker) and (Level 3: Terry Kirschbaum, test lab validation).

Once you get into Level 2 and Level 3, the seal is put through a series of intense chemical and environmental tests for further analysis. It truly is a team-based process where the slightest errors or miscommunication could hinder the outcome.

"We're in constant contact with everyone involved in a particular investigation," Kirschbaum said. "You have to have all the information in front of you in order to succeed and this information needs to be as accurate as possible. Communication is so important, it might be as simple as making sure metric and inch conversions are consistent."

These investigations come with a laundry list of challenges, according to Uncapher, though meeting these challenges is part of what makes the investigative work so interesting.

"Sometimes determining the mode

of seal failure is difficult when limited information is supplied, as it is critical to know the hours of operation, application temperature from oil and environment (min & max), operating speed, age of product installed being able to distinguish between damage from seal removal versus application issues and manufacturing defects," he said. "Also, when working with rubber seals it can be difficult to differentiate from manufacturing defects (process contamination and mishandling) versus application impact."

The key is having the right amount of experience and the right equipment to do the job.

"SKF meets these various challenges with equipment for visual inspection (microscopes), CMM dimensional measuring, and chemical analysis tools (FT-IR Fourier transform infrared spectroscopy and X-ray fluorescence), thermal analysis (thermal gravimetric analysis, and DSC Differential scanning calorimetry) and physical analysis tools (hardness, tension, compression, wear and fatigue testers)," Le Hir said.

The Benefits of Seal Investigation

The obvious benefit of product investigation is for SKF to develop a positive working relationship with its customer base. By providing quick, efficient solutions for their manufacturing challenges, they will continue to utilize SKF products and services in the future.

"We're working first and foremost for our customers," said Uncapher. "But there is plenty of research and analysis conducted here that benefits us internally as well."

Often, a customer will make upgrades to their machines including new additives to their lubrication, more horsepower, faster speeds and higher pressures and temperatures. "There are many cases where the customer fails to check with the seal manufacturer. Having the ability to understand the cause of failure and product limitations allows us to develop new seal configurations in the future," Uncapher said.

And this is why the work conducted here is so vital, according to Le Hir. "Each case involves a learning curve. We're taking this field research and applying it to future products that will make them better. It's an educational opportunity for our staff as well as our customers."

For example, Zimmerman discussed earlier that one of the most common modes of failure is improper installation. As a result, SKF has developed a line of installation tools for its Scot Seal Products to increase the robustness of the supporting structure and allow for easier installation in latter generations.

SKF is also currently working on the project to collect all typical failures and classify them, and at the same time develop a user-friendly mobile application. "This will be beneficial for our customers for understanding our investigations and visualizing the scenarios of failures," Uncapher said.

Working to Improve

One of the challenges mentioned regarding seal investigation was the lack of information available on the entire mechanical system. In many circumstances, the application engineer is working with very a limited amount of data and in most cases just the seal itself which begs a different question: Would it benefit the investigation center to look at the entire system and not just the seals?

Uncapher said that SKF would need a much larger receiving and tear down area as a bearing and seal might be part of large and heavy system. "We are able to measure and inspect up to 2.5 tons in Elgin and have other facilities in SKF which can accept much large items, but normally this is prohibited given the transportation costs and asset value. Additionally, we are able to perform level 0 & 1 inspections for bearings."

He continues, "Very often customers do not have spare housing and shafts and must reassemble new bearings and seals and continue production. This is why we rely on our field support team of application engineers and industrial specialists to collect the critical field data."

The plan in Elgin is to further implement computer simulation and modeling for seals. The investigation center and test lab allow for validation of existing models and creation of new modeling formulas. "We are planning to add a microscope with advanced features and also a Scanning Electron Microscope (SEM) to go in-depth in the investigations and analysis," Le Hir said. "Currently, we're using an SEM located in another SKF facility."

Without doubt, the success of the investigation team in Elgin will rest on the capable shoulders of the women and men behind the scenes, testing products, analyzing materials and providing data that can be used to make seal products and technologies better and more reliable in the future.

"It is very difficult to find experts in this field since it is not something that can solely come from education, but it is more of combination of higher education and years of experience," Le Hir added. "The more experts we have with investigation experience, the closer we get to the root cause of failures and can implement the corrective action." **PTE**

For more information:

SKF Sealing Solutions
900 N State Street, Elgin, IL
Phone: (847) 742-7840
www.skfusa.com

Each case involves a learning curve at the SKF Investigation Center where data can be applied to future seal design. Courtesy SKF.

SKF Seals

A MANUFACTURING HISTORY

During the Great Chicago Fire (1871) every establishment working with leather in the city of Chicago was destroyed. In the next few years, Chicago moved very rapidly to "Rise up Again" from the disaster.

Seven years after the Chicago Fire, in 1878, three men, William H. Preble (secretary), August C. Krueger (leather processor) and Andrew Spurling (president) met in a loft near Monroe and Clark in Chicago, hired 25 employees and began the manufacture and sale of rawhide products under the name of Chicago Rawhide Manufacturing Company.

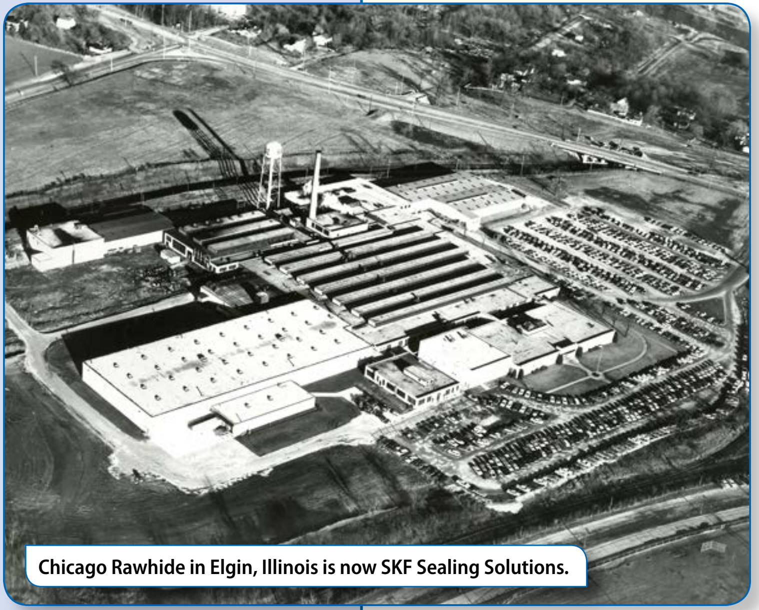
The main purpose of the company filed in the incorporation papers in 1879 was the "purchase and curing of hides, the manufacture and sale of rawhide leather, belting, lacing, ropes and all other articles of merchandise and utilities where rawhide leather can be used."

In 1882, William H. Emory, a banker, proposed to build for the original owners a building to manufacturer these products on Ohio Street in Chicago. By this time, the company was making rawhide fly nets for horses, rawhide ropes, rawhide shoe laces, leather belting and buggy whips.

Leather belting around steam engines and pulleys (in the late 1880s) was now driving the U.S. Industrial Revolution. In 1891, Emory was elected president of Chicago Rawhide and the company was the only manufacturer of rawhide belting in the world.

- In 1893 at the Chicago World Columbian Fair in Machinery Hall more than 200 exhibitors used Chicago Rawhide Belting and the company won an award.
- In 1897, Chicago Rawhide started a new product with the introduction of leather gears. In 1907, the Elston Avenue plant in Chicago was built. In 1918, 70 percent of the products made by Chicago Rawhide were for the U.S. war efforts.

- As early as 1914, Chicago Rawhide was selling leather products to the Ford Motor Company for the Model T.
- In 1928, the company patented the first Perfect Oil Seal made of leather.
- By 1938, 93 percent of the automobile equipment built was equipped with the Chicago Rawhide Perfect Oil Seal. During World War II, Chicago Rawhide was ordered by the U.S. Government to license other companies to make oil seals to ensure enough products were available for the war efforts. Chicago Rawhide had to share their trade secrets on seal manufacturing and design with the competitors.
- In 1949, Chicago Rawhide purchased the Majestic Radio and Television plant in Elgin, Illinois. This would later become its worldwide headquarters.
- In 1955, Chicago Rawhide began selling its products to the replacement markets. This included distributors, garages, repair shops and truck fleet operations.
- In 1964, the last of the Emory family to run the company died in Canada.
- In 1979, the original family and owners sold Chicago Rawhide to the IFNT Company. Also in the 1970's, SKF acquired the seals company Eurofigat S.p.A of Italy manufacturing seals for roller bearings, shock absorber seals and oil seals.
- In April of 1990, SKF acquired Chicago Rawhide. This was the largest SKF acquisition since the 1960s for SKF.
- In 1994, SKF Chicago Rawhide acquired Goetze Elastomere in Germany.
- Between 1995 and 2005, SKF started seals manufacturing units in India and China.
- In 2004, SKF formed a global seals business unit with operations in Europe, Asia, and North America.
- In 2006, SKF acquired Macrotech Polyseal Inc. in Salt Lake City, Utah and Economos, Austria GmbH which provide seals for the hydraulic and fluid handling markets.
- In 2007 SKF launched HMS5 and HMSA10 for metric gearbox seals.
- In 2012, SKF launched the next generation of SKF Speedi-Sleeves, which offered 30% less shaft wear then previous versions.
- In 2013, SKF acquired Blohm + Voss Industries, which specialized in marine solutions including Sterntube and bulkhead seals. Additionally, SKF purchased Kaydon Corporation, which included their Kaydon Ring & Seals.
- In 2017, SKF opened a new product investigation center to support customer field returns and product performance inquiries in Elgin, Il. **PTE**



Chicago Rawhide in Elgin, Illinois is now SKF Sealing Solutions.

SKF acquired Chicago Rawhide in April 1990.

Controlling Backlash in Mammography Systems

Express delivery program saves the day for mammography system designer

John Pieri, Sr. Product Line Manager, Thomson Deltran Clutch Brake

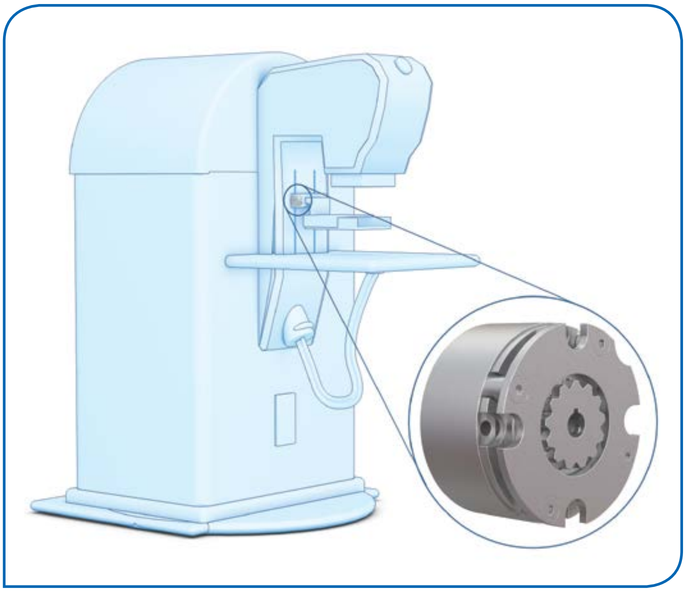
Medical imaging equipment, water handling systems, conveyors, robotic systems and rotary and linear actuators are among the many devices that may be fitted with electric friction brakes to hold their loads in place when the power is off or disrupted.

Because the inclusion of a brake has significant implications for the entire design, especially for determination of size and the selection of power supply, they must be designed in from the start. One mammography system designer learned this the hard way, not realizing the need for a braking system until he was testing the prototype. Fortunately, engineering support and an express fulfillment program from Deltran brake manufacturer Thomson Industries, Inc. enabled him to rescue a failing design and meet the schedule for the original design and prototype.

Why mammography systems need brakes

Mammography devices typically use a C-arm shaped apparatus in which an X-ray tube projects downward from the top of the C to scan the body generating a precise blur-free image that could reveal indications of breast cancer. A rotating ball screw bracketed to the tube assembly turns slowly - usually only a few hundred revolutions per minute - move the scanner evenly across the target area. Because the carriage must change direction many times in any session, any play resulting from gaps between components affects positioning precision and image quality. This loss of motion, commonly called backlash or backdrive, is also a potential problem when the system is at rest, when it can cause noise, vibration, and wear.

Electric brakes help control the backlash while the system is at rest and help bring things to a smooth stop if there is sudden loss of power from motor failure, power outage, or other event. Loss of electric power de-energizes the brake linings to grip a rotating plate (Figure 1), stopping it from turning or holding it in place once it is at rest. Re-energizing the system disengages the brake, allowing the shaft to rotate freely once again. However, the designer mentioned above did not discover the need for such a braking system until it was almost too late.



Thomson Industries, Inc. enabled a designer to meet the schedule for the original design and prototype of a mammography system utilizing a Deltran SB19 series spring set friction brake.

Tight specifications

Not considering the need for a braking system, the original motion control component design called for the following specifications:

- Backlash less than 0.5 degrees
- 2 Nm of holding torque
- Maximum diameter of 2 inches
- Voltage range: 16 VDC to 32 VDC range
- Ability to withstand 500 emergency stops
- Ability to operate in a radiation environment

When the designer discovered the need for the brake, he also realized that these specifications would limit brake options. Further, given the requirement to adhere to original production schedules, he needed to move quickly, adding further urgency to the situation.

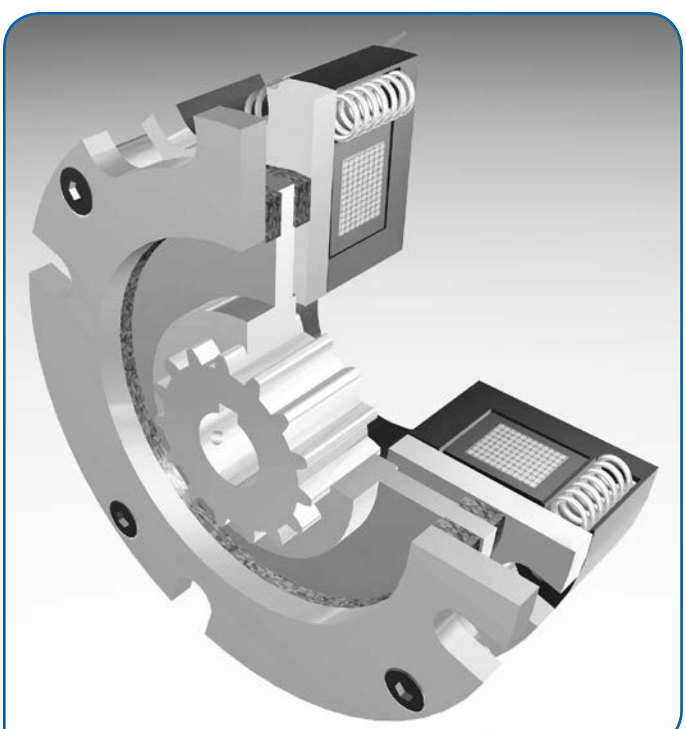
Meeting the spec

An Internet search led him to Thomson whose Deltran brake line was not only able to meet all technical specifications, but also offered a unique Servo Brake Express Program, which could ship a custom designed brake for prototyping within two days.

Thomson supplied Deltran SB19 series spring set friction brake, which met requirements in all of the following areas:

- **Low backlash:** Precision machining helped meet the backlash requirement. Play is all but undetectable when the system is at rest due to the tightly machined tolerances of the spline design.
- **Torque density:** In most cases, the available 2" × 1.2" available space would have limited the torque to around 10 inch pounds (approximately 1 Nm). The Deltran brake, however, provided 18 inch pounds (approximately 2 Nm). The increased torque density is the result of a high performance solenoid, which can overcome stronger spring force, and the use of a proprietary high-coefficient of friction brake pad.
- **Power:** The 24 VDC fell well within the 16–32 VDC available.
- **Emergency stopping:** The proprietary friction brake pad enables absorption of more than 500 hundred emergency stops.
- **Radiation protection:** The requirement was accomplished by expert adjustment of the lead wires.

After finalizing requirements with Thomson customer service and product specialists, which included a customer/supplier system data exchange confirming that the brake was fully capable of handling 500 emergency stops. Thomson was then able to ship a system for prototyping within 24 hours.



Thomson supplied the Deltran SB19 series spring set friction brake, which met requirements in power, emergency stopping, radiation protection, torque density and low backlash.

Thomson's Deltran brake line was not only able to meet all technical specifications, but also offered a unique Servo Brake Express Program which could ship a custom designed brake for prototyping within two days.

The prototype system had all the necessary adjustments other than the lead wire for radiation protection. This did not interfere with the initial prototyping and was completed the following week. In six weeks, Thomson shipped 20 final systems, to be used in the production of the first 20 mammography systems. The program is now in full production, with the manufacturer shipping about 300 systems per month. Plans are also underway for modified brake designs that will meet European power requirements as well.

This story has a happy ending. If the design engineer and his purchase team had not collaborated with the Thomson application engineering team on a Deltran brake solution, it could have had a very different outcome. The engineer might have had to make major design adjustments, which would have delayed time to market further and required additional budget. Additional resources may have needed to be invested to modify the initial design. All of these outcomes would have resulted in dealing with backlash of a different sort. (This article first appeared in *Product Design Technology* and *Medical Design Technology* in July 2016). **PTE**

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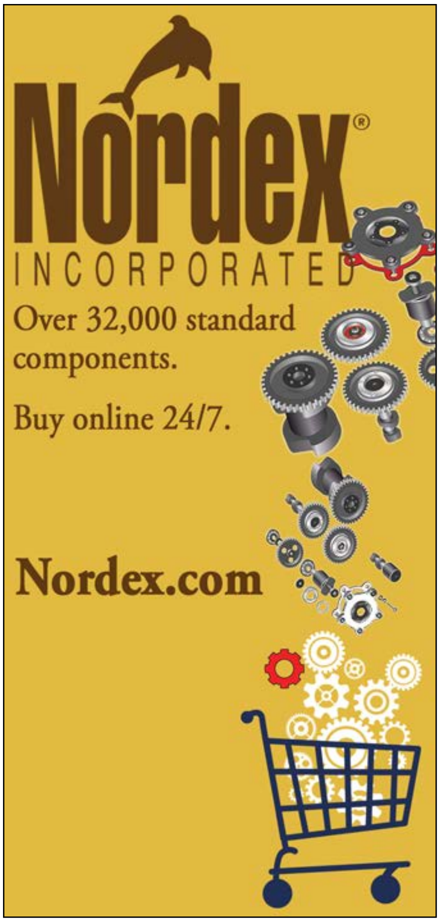
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The Hidden Cost of Incomplete Hydrodynamic Bearing Maintenance

Will Cannon, Application Engineer, Baldor Electric Company

Introduction

Reliability and maintenance engineers can improve uptime and save money on both long-term maintenance and downtime costs by properly diagnosing and correcting bearing vibration issues when they exceed their acceptable limits. This requires inspecting the housing as well as the liner for wear, and replacing them as a pair when the housing is worn, so that wear-in between the mating surfaces can occur. Without proper contact between the housing and liner, fretting damage will drastically reduce the life expectancy of the liner. Although the upfront cost is greater, the investment required for complete pillow block replacement versus the cost of replacing the liner alone, which is the most common maintenance practice, should be compared in order to expose the hidden cost of incomplete hydrodynamic bearing maintenance.

Vibration

Any industrial fan in operation has inherent vibrations that manifest themselves at the bearings. These vibrations and their amplitudes are indicative of the health of the fan assembly, and can indicate when servicing is required based upon the many standards that have been published concerning acceptable levels of vibration. Therefore, vibration monitoring is essential for proper operation and adequate understanding of the fan performance; however, even with all the vibration monitoring in place, there is still ambiguity as to the cause of the vibrations.

Before an entire fan refurbishing is planned, it is a common maintenance practice to service the unit and its components to optimize the fan performance and to complete corrective actions for the issues that have developed. Unfortunately, during the stress and time-sensitive nature of outages, planned or unplanned, the predominant corrective action of hydrodynamic bearing maintenance when vibration levels have exceeded their limits is to replace the liner (insert) of the bearing, and to leave the housing untouched and uninspected (Fig. 1).

The liner is frequently misdiagnosed as the sole contributor of fan bearing problems, as it is the component that not only supports the shaft but also requires replacement once the clearance within the liner increases beyond an acceptable limit, which can cause excessive vibration. However, this is an incomplete framework of the source of bearing problems. The amount of contact between the spherical seats of the housing and liner is an often overlooked source of bearing problems. When the contact between these two components is significantly worn, the result will be excessive vibration that can easily be misdiagnosed as excessive clearance in the liner (liner failure). However, replacing the liner will not fix this problem. It may provide a vibration reduction for a very short period of time, but the fan vibration will continue to reoccur, often more aggressively than before, since the other sources of vibration remain unchecked and uncorrected. The consequence of leaving unchecked vibrations in the system is the acceleration of fretting damage that will continue to occur to both the housing and liner, even after a new liner is replaced into the original housing.

Fretting Damage

When two properly machined surfaces in contact and are loaded against each other, there will be a period of run-in wear that occurs until they are suitably conformed to each other, similar to purchasing a new pair of shoes and letting them break-in for a few weeks before they fit most comfortably. However, over time, as proven with shoes, permanent wear damage occurs on the loaded and contacting surfaces, in a process of material removal. As the wear progresses, small amplitude oscillatory movements will begin to occur between the mating surfaces, which is the process of fretting, and is the result of external vibrations in the system (Waterhouse, 1992). Normally, this is a very slow process to develop in a new pillow block (liner and housing pair) since the components wear in together. However, when a new liner is placed into an older housing that has had years of deformation and material removal from operating in conjunction with the previous liner, there is no opportunity for the two surfaces to wear in together. Since the older housing surface cannot conform to the newly machined liner surface, the process of fretting will begin immediately and rapidly progress in severity on both the liner and housing; the only solution that will solve this problem is to replace the entire pillow

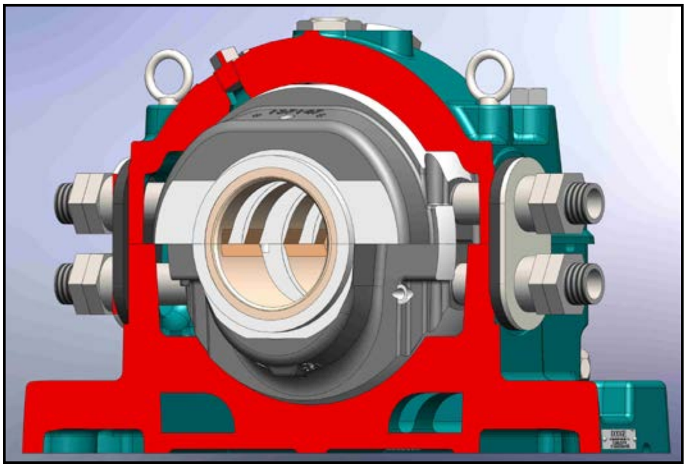

Figure 1 Cutaway view of a pillow block assembly showing the liner, which is commonly serviced, and the housing, which is commonly neglected of service. (Courtesy of Dodge)

Figure 2 A new liner installed into an old housing failed after ~1 to 1½ years of operation and caused an unexpected fan shutdown due to high vibration levels. The deep pitting and fretting corrosion caused by improper seating of the liner in the housing can be clearly seen.

block and allow the mating surfaces to wear in together as designed.

Performing bearing liner maintenance but neglecting the housing damage from wear and fretting corrosion could consign a new bearing liner to ~1–2 years of life before high vibration issues develop and potentially cause fan shutdown times, which is ~1⁄10th of the intended life of the liner with proper maintenance and servicing. In an actual example from the field (Fig. 2), a distributor reported: "The customer replaced the liner assembly with a new one approximately 1 to 1½ years ago because of high fan vibration. The fan is again out of service for high vibration". However, this customer is not unique in this situation of replacing a new liner into an old housing as a standard maintenance procedure

and achieving sub-satisfactory results that do not prolong equipment life or reduce unscheduled downtimes due to high vibration levels.

To further demonstrate the common occurrence of this problem, additional pictures from customers across the United States have been included that show the distinguishing marks of fretting corrosion (irregular dark-colored patterns and cavities) along both the liner sphere and spherical seats of the housing (Fig. 3-4).

The purpose of addressing this issue is to dispel the notion that liner repair or replacement is the single most important corrective action in hydrodynamic bearing maintenance when vibration levels are too high. Instead, the entire bearing pillow block assembly should be perceived as a unit of interdependent components that, for proper operation and life expectancy, must cohesively work together. When one of the components is significantly worn out, the entire pillow block should be replaced to alleviate the impending fretting that will occur from a liner that is improperly retained in the housing. However, the issue then becomes a matter of "unjustifiable cost", which is a more grievous misconception than the original problem of resolving fan vibration issues with liner replacement alone.

Figure 3 Three separate liners that have failed due to high fan vibration levels and show the fretting corrosion marks along their spherical seats.

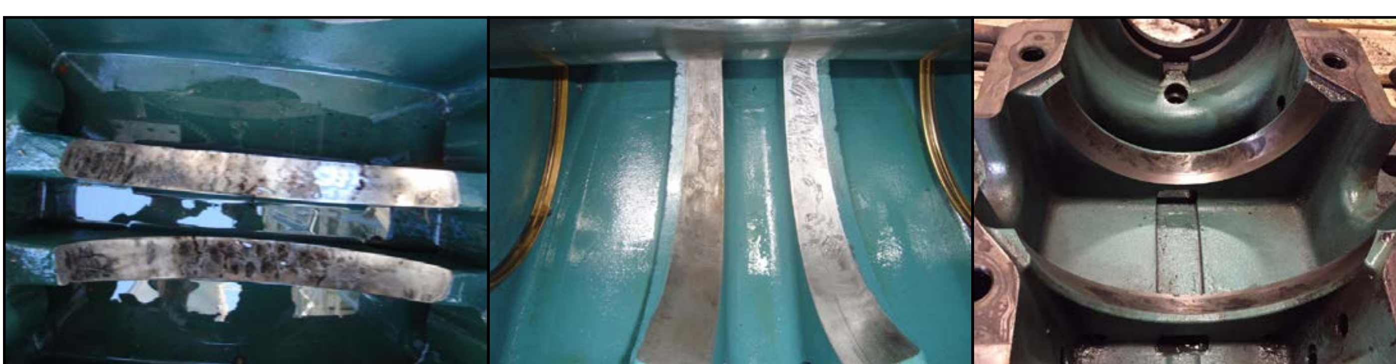

Figure 4 Three separate housings that have been pulled from service due to high fan vibration levels and exhibit the characteristic fretting marks caused by improper contact and retention of the liner within the housing.

Cost of Replacement vs. Expected Life

Durability is one of the many advantages of hydrodynamic bearings. Due to hydrodynamic lubrication, the only wear that should occur between the shaft and the liner takes place at start-up and shutdown. With proper oil care and maintenance, hydrodynamic bearings can theoretically run forever. It's for this reason, among many others, that end users and OEMs will make the financial investment for a hydrodynamic bearing rather than a spherical roller bearing. However, because of the rugged construction and many advantages of hydrodynamic bearings, it is well known that they are not inexpensive. That is why the financial case must be considered for proper bearing servicing versus low-cost servicing1 as it pertains to the life of the fan assembly.

In a pillow block assembly from a general manufacturer, the cost of the liner comprises approximately 72% of the entire cost of the pillow block. In a fictitious scenario2 where two customers (Customer A, Customer B) install a new fan with new hydrodynamic bearings that cost $10k at the same time, the cost comparison of two maintenance strategies over a twenty year period clearly demonstrates the value of proper maintenance and servicing (Table 1).

Although this scenario is fictitious, the general financial and life expectancy patterns it showcases is consistent with real applications, and the difference between the two methods becomes exceedingly more drastic as time continues, especially when downtime losses are considered. The reality is that the short-term financial investment required to replace the pillow block at the time of bearing servicing is more than compensated for by the longevity of the bearing. When considered from the perspective of the life of the fan, it is the frequency of servicing and the associated downtime costs that are significantly more expensive than the less frequent pillow block replacement costs. Therefore, the best financial investment that a customer can make is to take advantage of the life the bearings were designed for and save the money that would be spent on more frequent maintenance and liner replacement.

Conclusion

Improving uptime, reducing the frequency of required maintenance, and saving money are the target objectives of nearly every maintenance and reliability engineer. These targets can be most effectively accomplished by investing in the bearing equipment as designed by the manufacturer. Shortcutting proper housing maintenance will save marginal dollars at the time of servicing and cost exceedingly more in future repairs. The best method to properly keep hydrodynamic bearings running for the long durations that they're designed for is to consider the liner and housing as an interdependent pair. The ability of the housing and liner to wear-in together is critical to eliminating the fretting corrosion that will otherwise develop and gradually promote fan vibrations that are sure to exceed the limits. **PTE**

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William Cannon is an application engineer with Baldor Electric Company, a member of the ABB Group, with responsibility for Baldor-Dodge sleevoil bearings. Since joining the company in 2014, he has been involved in product development projects with hydrodynamic bearings, and he received a company-specific engineering achievement award for the Baldor-Dodge RTL-Spherical Bearing. William is a graduate of Clemson University.

Table 1 Fictitious 20-year financial comparison of two maintenance strategies

	Customer A				Customer B			
	Maintenance Course: Replace liner when fan vibration exceeds limits				**Maintenance Course:** Replace housing and liner pair when fan vibration exceeds limits			
	New Fan Installation - New Bearings				New Fan Installation - New Bearings			
	Housing Age	**Maintenance**	**Cost**	**Total investment**	**Housing Age**	**Maintenance**	**Cost**	**Total investment**
	0 years	None	$0	0	0 years	None	$0	0
	10 years of service				10 years of service			
	10 years	Replace Liner	$7,200	$7,200	10 years	Replace Pillow Block	$10,000	$10,000
Hidden Cost of Partial Bearing Replacment	5 years of service							
	15 years	Replace Liner	$7,200	$14,400				
	3 years of service							
	18 years	Replace Liner	$7,200	$21,600				
	2 years of service				10 years of service			
	20 years	Replace Pillow Block	$10,000	**$31,600**	10 years	Replace Pillow Block	$10,000	**$20,000**

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Gear Materials

THE QUESTION

We know the values for fatigue strength tooth root tension, σ_{Flim} and Hertzian stress, σ_{Hlim} for the steel 18CrNiMo7-6 according to DIN and others. But there is a new material — Ferrium (I found it searching in web) — and this is a high-performance material compared to traditional materials. I would like to know the values for σ_{Flim} and σ_{Hlim} for this material."

(Ed.'s Note: Newly developed ferrium *C64 (AMS 6509) is a high-strength, high-surface-hardness, good fracture toughness carburizable steel — steel that also has high temperature resistance, corrosion resistance and hardenability. C64 steel is a higher performance upgrade from 9310, X53 (AMS 6308), EN36A, EN36B, EN36C and 8620. It can achieve a surface hardness of 62-64 Rockwell C (HRC) via vacuum carburization. C64 steel is double-vacuum melted, i.e. — vacuum-induction -melted and then vacuum-arc-re-melted (or "VIM/VAR") for high purity, leading to much greater fatigue strength. Applications include: demanding Bell Helicopter and Sikorsky transmission gearboxes Source:* Questek.com.)

EXPERT RESPONSE PROVIDED BY CHUCK SCHULTZ PE: AGMA allowable stress values for tooth bending stress do not vary with the alloy selected. The allowables were negotiated based upon the committee members' experience. Relatively recently the standards were revised to provide a range of allowables for various levels of heat treat and metallurgical quality but these revisions still do not reference specific alloys. Only core hardness is considered and as each alloy has a unique hardenability profile, this could be considered "linkage" to a specific alloy. Put more directly, in the AGMA system there is no change in the allowable bending stress due only to an "upgrade" in the alloy.

Perhaps future research will provide a way to calculate a specific allowable bending stress for a specific material based upon traditional physical tests on samples. This would provide guidance to those seeking to develop better gear materials. At this time we simply do not have the "science" to provide such a formula.

ISO, DIN, and AGMA all allow designers to establish their own allowable stress levels based upon their experience and test results. It would be helpful if those with experience from making gears of unique materials would share their knowledge with the standards committees so the science can be nurtured.

Charles D. Schultz, PE is Chief Engineer for Beyta Gear Service (gearmanx52@gmail.com) in Winfield, Illinois, and a Technical Editor for Gear Technology and Power Transmission Engineering magazines. He is also a longtime AGMA member, having served on or chaired a number of its committees over the years. And now you can follow Chuck's new Gear Technology blog every Tuesday and Wednesday at geartechnology.com.

Performance Testing of Composite Bearing Materials for Large Hydraulic Cylinders

Leo Dupuis, Bosch-Rexroth Sr. Development Engineer

Introduction

Large hydraulic cylinders (LHCs) are integral components in the functioning of large machines in mechanically demanding, corrosive and abrasive environments, such as offshore drilling rigs. The materials utilized in these large-scale hydraulic systems must deliver reliable performance throughout their expected lifecycle.

One key LHC component is the radial bearing. Although a number of materials are utilized to create these bearings (such as aluminum-bronze, bronze and several thermoplastic materials including UHMWPE), the most commonly used materials are composite materials.

To effectively and reliably predict the longevity and operational performance of these composite radial bearings, a major LHC manufacturer developed a testing method that examines the layer structure of the bearing, as well as its friction behavior; the company also developed a method to investigate how the bearing deforms as the result of being placed under a load, and then further imaging to assess the bearing's response to being under a load.

The testing provides a basis for productive consulting with bearing manufacturers to help them improve key material characteristics of the utilized bearings. This unique investment will help the LHC manufacturer optimize the operational value of its customer's LHC system.

LHC radial bearing technology

Hydraulic cylinders convert hydraulic energy into mechanical movement. The hydraulic cylinder consists of a cylinder body, in which a piston connected to a piston rod moves back and forth. For the most common single rod cylinder, the barrel is closed on the cylinder cap end by the cylinder bottom and on the cylinder rod end by the cylinder head, where the piston rod comes out of the cylinder.

Both the piston and cylinder head have radial bearings and seals. Figure 1 shows where the seals and bearings are located. The piston divides the inside of the cylinder into two chambers: the cap end chamber and the rod end chamber.

Radial bearings are used for guiding the piston through the cylinder shell and the cylinder rod through the cylinder head. The radial bearings may be exposed to high loads due to:

- Side loads on the cylinder rod
- Gravitational force, depending on the orientation of the cylinder in the application
- Small misalignments (e.g., as the result of gravitational force) in combination with axial compressive external loads

High shear stresses can also be expected in operation, caused by the (dynamic) friction forces between respectively the cylinder shell (piston bearing), the cylinder rod (head bearing) and the bearing material.

Given its function, key bearing properties that merit consideration include:

- Compressive strength
- Shear strength
- Tensile strength
- Compressive modulus of elasticity
- Static and dynamic friction
- Temperature range of application
- Thermal expansion coefficient

The LHC manufacturer developed several testing procedures to evaluate some of these properties, with a goal of collaborating with suppliers to improve the performance of these bearings.

Friction response of composite bearing material

Understanding a bearing's response to friction is crucial, because friction means wear, negative frictional behavior (such as stick-slip), frictional heat and reduction of the cylinder force efficiency (in other words, energy loss).

Currently, the most commonly used materials for LHC radial bearings are

Figure 1 Location of radial bearings and seals in a hydraulic cylinder.

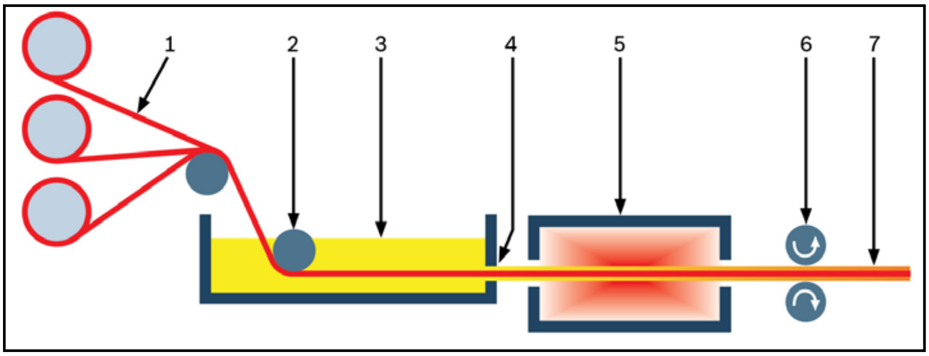


Figure 2 The key steps in the pultrusion process include: 1) continuous roll feed of reinforced fibers/woven fiber mat; 2) tension control roll; 3) resin impregnator bath; 4) resin-soaked fiber; 5) bearing die and heat source; and 6) fiber pull mechanism.

composite materials. These are technical fabrics impregnated with thermosetting resins, i.e., a polymer fabric reinforcement with a thermoset matrix.

The most commonly used fabric material is polyester; the most commonly used resins for the matrix are polyesters and phenols. For friction reduction, dedicated additives are used, mostly PTFE powder.

Bearing pultrusion production process

It is useful to understand how composite radial bearings are manufactured. While some bearings are manufactured via a pressing process, the most common method is the pultrusion process; the bearings discussed in this article were manufactured using this process.

In pultrusion, multiple layers of fabric are pulled through a resin bath, where the liquid resin and some additives are present as a mix. When the fabric has gone through the bath, the resin is heat cured in a mold and can be tempered afterward (Fig. 2).

In the pultrusion process the following process properties have an influence on the quality of the radial bearing:

- The fabric's speed through the process (particularly the curing process)
- Distance of the different layers of fabric in the resin bath
- Pulling forces on the fabric during the process
- Incorrect curing time and/or temperature
- Post-curing process errors

Bearing Performance Investigation

A standard testing methodology was developed in order to investigate key radial bearing properties. Composite bearing samples from several suppliers were tested and analyzed, along with a standard composite bearing material currently used by the LHC manufacturer and created to the company's specifications.

The following tests were carried out:

Microscopic imaging: A micrograph of the cross section of the composite bearing is taken to study the composite's layer structure and possible defects. Micrographs are taken pre- and post-friction testing.

Friction test: This test is conducted to generate a Stribeck curve (Figs. 3–10), which is made by measuring the friction on the bearing at different velocities and different loads. The magnitude of the load is varied, and two inclinations are used.

Thickness measurement: Before and after the friction test, the composite bearing's thickness is measured. Measuring the change in the bearing's thickness after putting the composite under loads provides an indication of the permanent deformation as a result of the load.

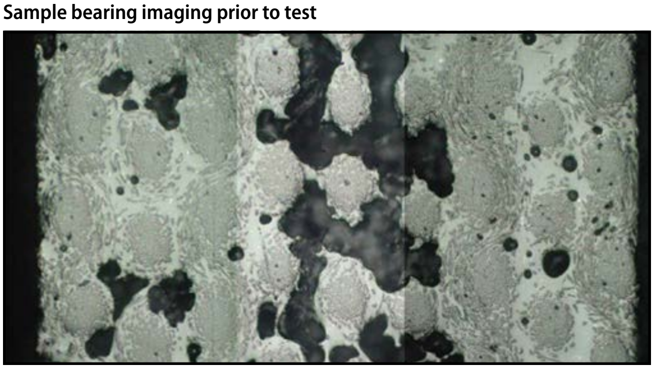
Step1 — microscopic imaging. Cross-sections were made both in the length and width direction of the radial bearing. The micrographs capture visual data related to the stresses that composite bearing material can undergo and the possible effects of those stresses, such as microscopic cracks, shearing of the different layers and deformation of the material.

Micrographs also image air enclosures in the material, which can contribute to weakening the integrity of the resin-impregnated material.

The radial bearings were cut with great care and cooling water was used to keep the material temperature low, so that the bearing structure was not

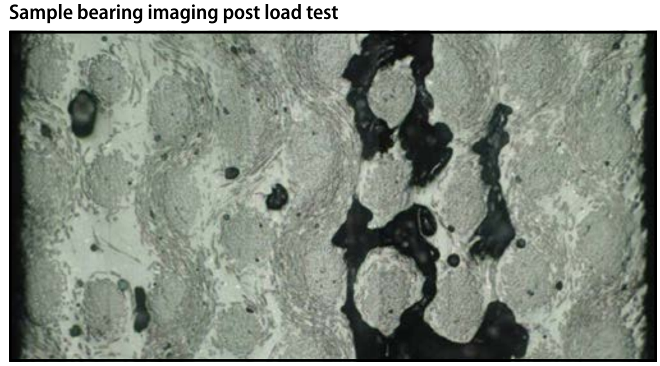
Results: Microscopic imaging—100× magnification

Sample bearing imaging prior to test



Transversal

Sample bearing imaging post load test

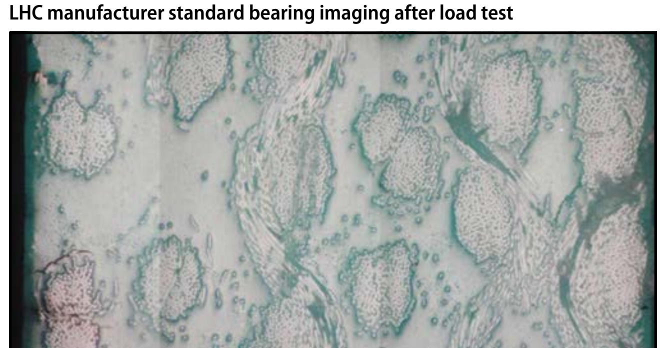


Transversal

LHC manufacturer standard bearing imaging prior to test

Transversal

LHC manufacturer standard bearing imaging after load test



Transversal

damaged. An image was made of the cross section with an optical microscope with a magnification of 100.

The microscopic investigation revealed that the radial cross section gives information on the quality of the fabrication process. The transversal cross section gives more information about the composite layer structure and incidence of and distribution of air inclusions.

Sample micrographs are presented below, comparing a sample of the LHC manufacturer's standard composite bearing material with bearing material provided by another manufacturer of composite bearings. Pre-friction testing and post-friction testing micrographs are shown for both.

Step 2—friction test. For determining the friction characteristics of the bearing strips, Stribeck curves were generated, as well as stick-slip curves. A Stribeck (master) curve is created by measuring the friction at different loads and velocities.

A special test rig was created, consisting of a cylinder rod which is driven by an electrical screw spindle. A bearing strip was placed in a bearing housing which can be put under load by a hydraulic cylinder perpendicular to the rod. The complete bearing housing can be put under an angle, which can be slightly varied. Thus, the influence of the deflection curve of a cylinder rod on the bearing can be simulated.

Three measurement series (at temperatures 25°C, 40°C and 55°C) were carried out to generate a Stribeck curve at two angles (0° and 1°). The friction was measured with a load cell at a sample rate of 1 kHz. The test program was programmed in *LabVIEW* and was executed fully automatically.

The following test series were performed:

0° angle, temperatures 25°C, 40°C and 55°C

Load [kN] Approx.:	Velocity:
8	0.1/-0.1
30	0.2/-0.2
60	0.5/-0.5
92	1/-1
123	2/-2
154	5/-5
184	20/-20
215	50/-50
	100/-100

1° angle, temperatures 25°C, 40°C and 55°C

Load [kN] Approx.:	Velocity:
10	0.1/-0.1
30	0.2/-0.2
45	0.5/-0.5
60	1/-1
70	2/-2
75	5/-5
	20/-20
	50/-50
	100/-100

Step 3—thickness measurement. Both prior to and after the friction test, measurements were made of the composite material's thickness to assess the change in thickness and permanent deformations.

Results—stick-slip curve measurements. Stick-slip curve measurements can show the presence or absence of stick-slip effects, and to what degree. The fluctuation in the friction line graph shows the presence of stick-slip phenomena.

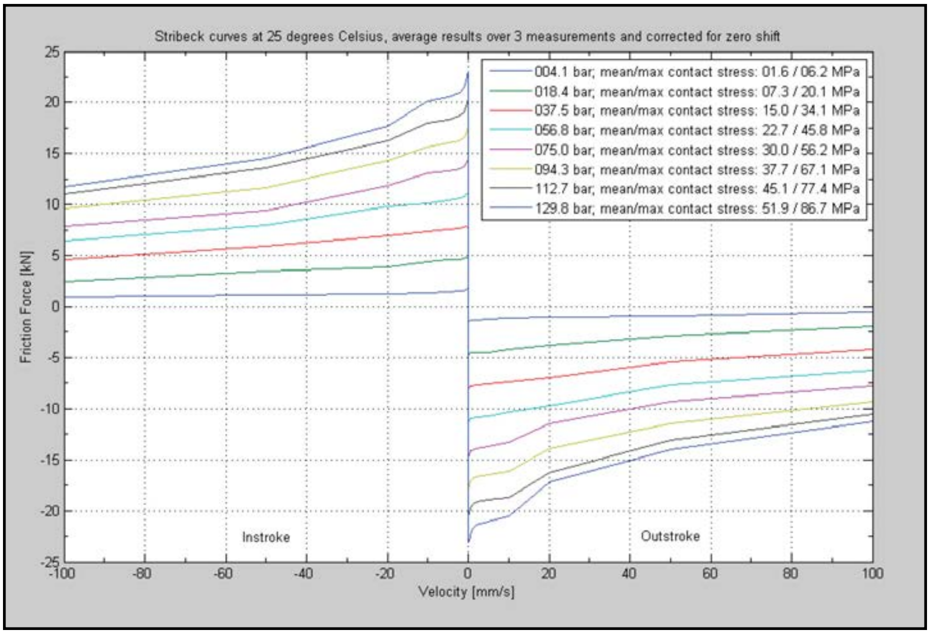


Figure 3 A Stribeck curve of supplier sample bearing material at 0° angle, general results at 25°C.

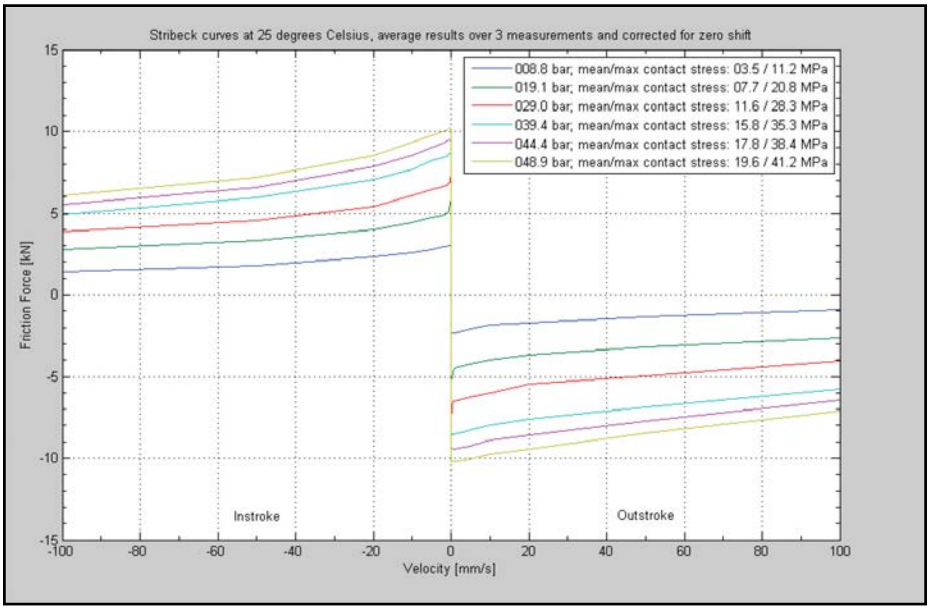


Figure 4 A Stribeck curve of supplier sample bearing material at 1° angle, general results at 25°C.

Friction test and stick-slip measurement findings:

- For the supplier-provided sample composite bearing material, a high level of friction was measured. Stick-slip was measured over the whole flat load test range. With the test conducted at a 1° angle, the sample bearing material showed stick-slip at the lowest load.
- Maximum friction 0° angle: 38kN, 1° angle: 16 kN
- For the LHC manufacturer's standard composite bearing material, a high level of friction was measured. Stick-slip was seen over most of the flat load test range — excepting the lowest load. With the test conducted at a 1° angle, this sample only demonstrated stick-slip at the highest load.
- Maximum friction 0° angle: 22 kN, 1° angle: 11 kN

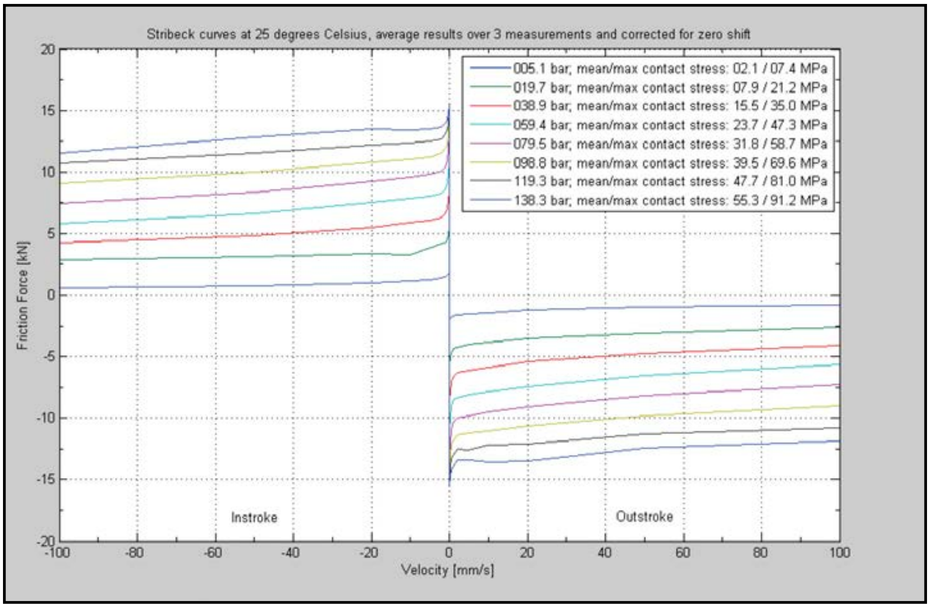


Figure 5 **A Stribeck curve of LHC manufacturer standard bearing material at 0° angle, general results at 25°C.**

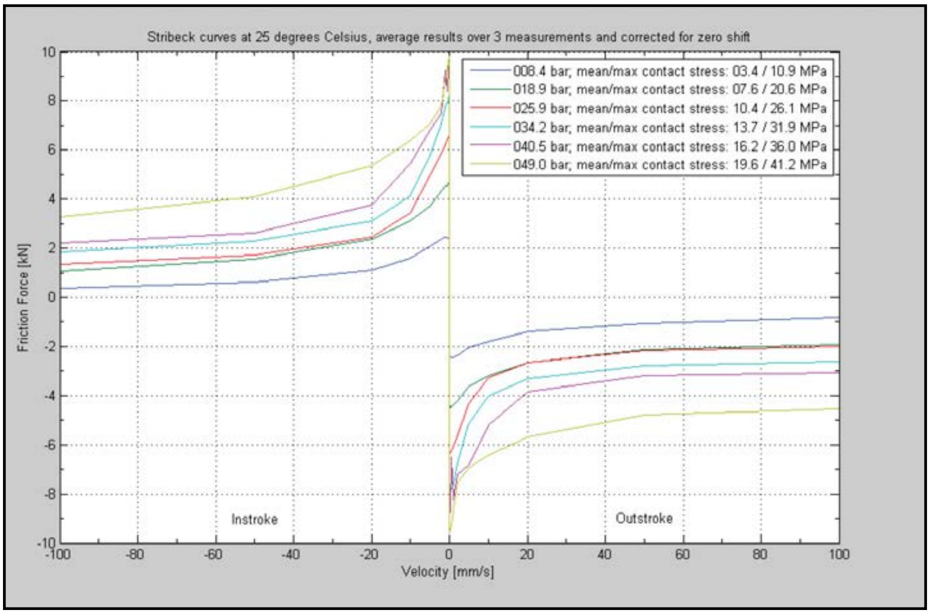


Figure 6 **A Stribeck curve of LHC manufacturer standard bearing material at 0° angle, general results at 25°C.**

Results: Stick-slip curve measurements

Stick-slip curve measurements can show the presence or absence of stick-slip effects, and to what degree. The fluctuation in the friction line graph shows the presence of stick-slip phenomena.

Supplier sample bearing stick-slip curves

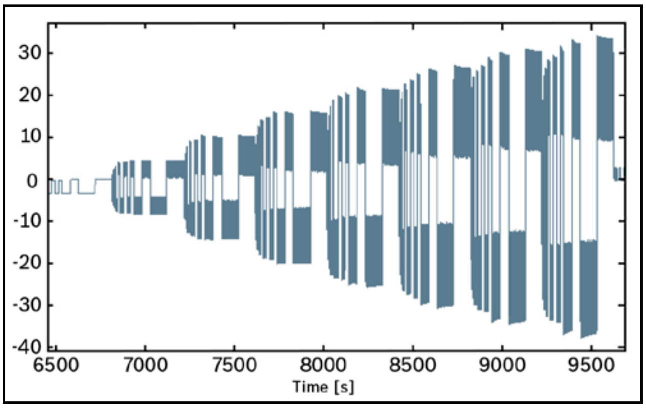


Figure 7 A stick-slip curve of BR non-standard bearing material at 0° angle.

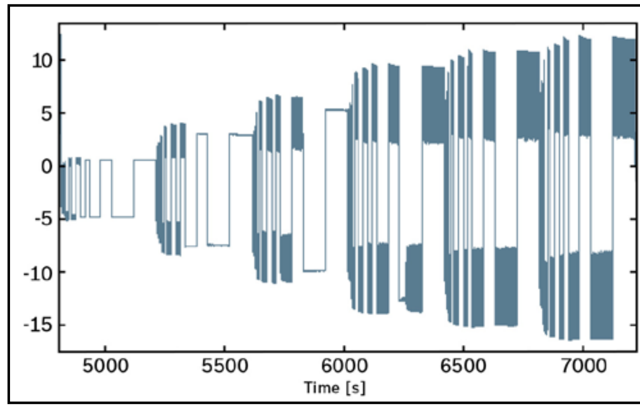


Figure 8 A stick-slip curve of BR non-standard bearing material at 1° angle.

LHC manufacturer standard stick-slip curves

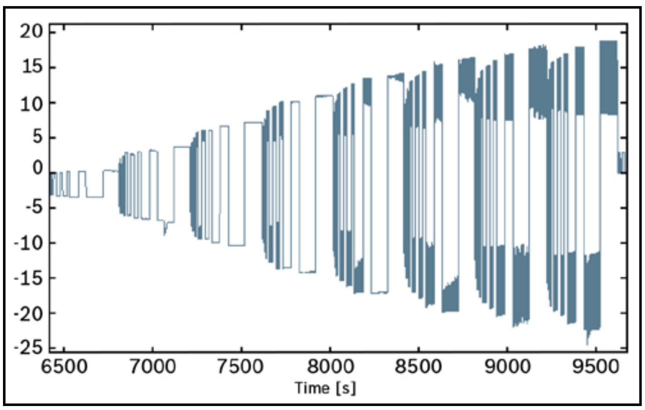


Figure 9 A stick-slip curve of BR standard bearing material at 0° angle.

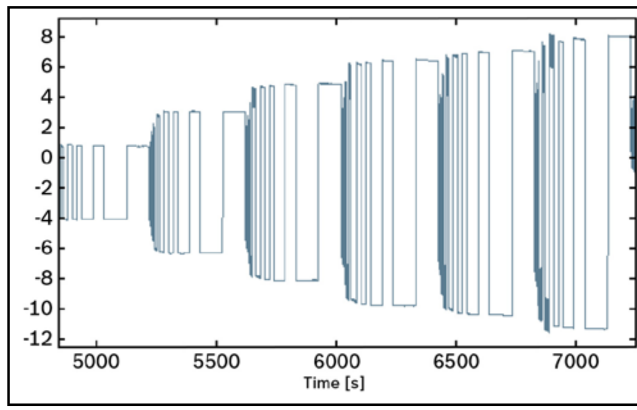
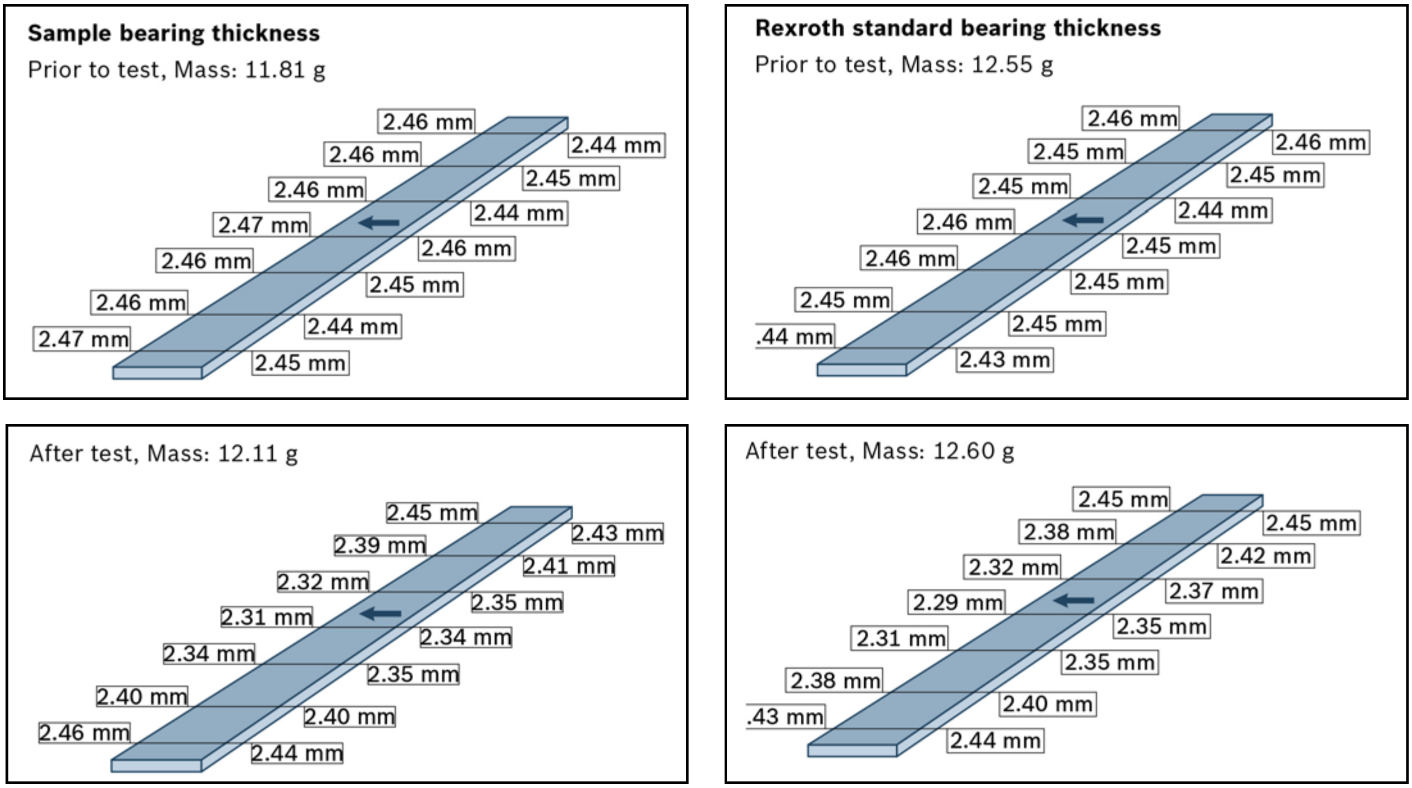


Figure 10 A stick-slip curve of BR standard bearing material at 1° angle.

Friction test and stick-slip measurement findings:

- For the supplier-provided sample composite bearing material, a high level of friction was measured. Stick-slip was measured over the whole flat load test range. With the test conducted at a 1° angle, the Sample Bearing material showed stick-slip at the lowest load.
- Maximum friction 0° angle: 38kN, 1° angle: 16kN
- For the LHC manufacturer's standard composite bearing material, a high level of friction was measured. Stick-slip was seen over most of the flat load test range, except the lowest load. With the test conducted at a 1° angle, this sample only demonstrated stick-slip at the highest load.
- Maximum friction 0° angle: 22 kN, 1° angle: 11kN

Results: Bearing thickness measurement pre- and post-friction test



Conclusions

During the investigation, significant differences were observed in friction test results and in layer buildup, i.e., air inclusions and cracks, between the LHC manufacturer's bearing material and the supplier's. There is an indication that air inclusions will give permanent deformation under load to the bearing composite material, depending on the load over time. In addition it can be seen that air inclusions and cracks will give rise to weak spots in the material.

Various friction levels and differences between static and dynamic friction (stick-slip) were observed. The LHC manufacturer's composite materials showed the lowest dynamic and static friction tested under an angle; also, layer buildup was the most stable when compared to the supplier's sample.

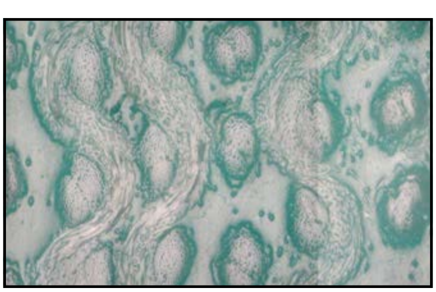


Figure 11

As a result, the LHC manufacturer undertook further refinements of its composite bearing material. The company also invested time and engineering resources to collaborate with the bearing supplier to optimize the pultrusion process. Improvements were made to help reduce cracks and air inclusions in the matrix and, therefore, the weak spots. This was done by adjusting the pultrusion process speed, yielding composite material matrix and layer structure that more closely aligns with the optimum structure for this type of material and application.

After optimization the LHC manufacturer performed a new test series including different hydraulic fluids and different life tests on a working LHC. Imaging results of a part of the new test series demonstrated improved performance.

It can be seen in (Fig. 12) after testing that only in the top, where the bearing strip is exposed to the highest loads (dynamic friction, shearing and stresses), are a few small cracks visible. In the material itself no air inclusions or cracks were present, which is a direct result of the optimization of the pultrusion process. **PTE**

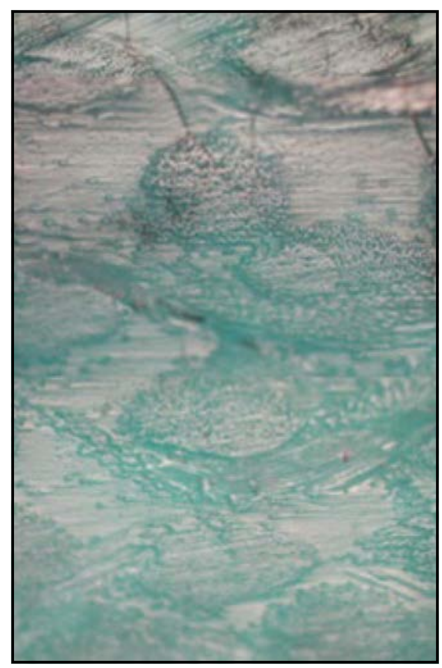


Figure 12

For more information:

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www.boschrexroth-us.com

Baldor Basics: Understanding Torque

Edward Cowern, P.E.

In the process of applying industrial drive products, we occasionally are misled into believing that we are applying horsepower. The real driving force is not horsepower — it is torque. This paper is developed to impart a deeper understanding of torque, its relationship to horsepower, and the types of loads we most frequently encounter.

Introduction

Torque is the twisting force supplied by a drive to the load. In most applications a substantial amount of torque must be applied to the driven shaft before it will even start to turn. In the English system the standard units of torque as used in the power transmission industry are pound inches (lb. in.) or pound feet (lb. ft.) and, in some cases for very low levels of torque, you will encounter ounce inches (oz. in.).

Torque Basics

At some time we all have had difficulty in removing the lid from a jar. The reason we have this trouble is simply that we are unable to supply adequate torque to the lid to break it loose. The solution to our dilemma may be to: 1) grit our teeth and try harder; 2) use a rubber pad or cloth to increase the ability to transmit torque without slippage; or 3) use a mechanical device to help multiply your torque producing capability. Failing on all of the above, we may pass the jar to someone stronger who can produce more torque.

If we were to wrap a cord around the lid and supply a force to the end of the cord through a scale (Fig. 1), we could get the exact measurement of the torque required to loosen the lid.

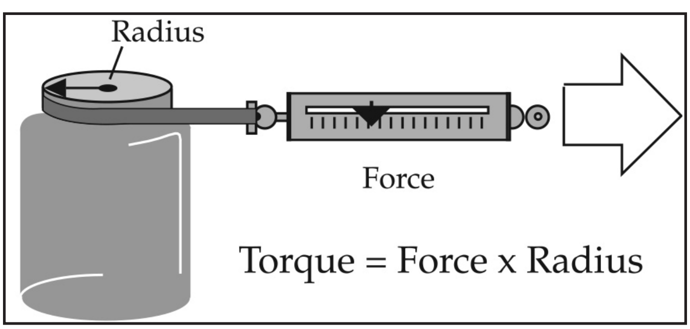


Figure 1 If we were to wrap a cord around the lid and supply a force to the end of the cord through a scale, we could get the exact measurement of the torque required to loosen the lid.

The torque required would be the force as indicated on the scale, multiplied by the radius of the lid.

For example, if the indicated force on the scale at the time of "breakaway" was 25 lbs., and the lid radius was 1.5 inches, the torque required would have been:

$$T = 25 \text{ lbs.} \times 1.5 \text{ in.} = 37.5 \text{ lb. inches}$$

Although this example does give a reasonable illustration of torque, it does not represent a very common example of requirements on industrial equipment.

There is, however, one additional, important point that can be derived from the jar and the lid example, namely — "sticksion." Sticksion is a term generated to indicate the amount of torque required to break a load loose on its way to making the first revolution.

In general the break-away torque requirement to start a machine will be substantially greater than that required to keep it running once it has started. The amount of sticksion present in a machine will be dependent on the characteristics of the machine, as well as the type of bearings that are used on the moving parts (Table 1).

Table 1 Typical values of breakaway torque for various general classifications of machinery.

Torque	% of Running Torque	Types of Machines
Breakaway Torque	120% to 130%	General machines with ball or roller bearings
Breakaway Torque	130% to 160%	General machines with sleeve bearings
Breakaway Torque	160% to 250%	Conveyors and machines with excessive sliding friction
Breakaway Torque	250% to 600%	Machines that have "high" load spots in their cycle, such as some printing and punch presses, and machines with "cam" or "crank" operated mechanisms.

Table 1 indicates typical values of break-away torque for various general classifications of machinery.

Assuming that the sticksion, or break-away torque, has been overcome and the load has started, a continuing amount of torque must be supplied to handle the running torque requirements of the machine.

In a high percentage of industrial applications the torque requirement of the load is independent of the speed at which the machine is driven. This type of load is generally called a "constant torque load."

Constant torque loads will be used to introduce the basic concepts of horsepower; additional load types will then be introduced.

Horsepower

Many years ago the invention of the steam engine made it necessary to establish a unit of measurement that could be used as a basis for comparison for how much work could be done by an engine. The unit that was chosen was related to the animal that was to be replaced by the new sources of power — the horse.

After a great deal of testing it was found that the average workhorse could accomplish work at a rate equal to 33,000 ft. lbs. in one minute — the equivalent to lifting 1 ton (2,000 lbs.) 16.5 feet, or 1,000 lbs., 33 feet in one minute.

This unit, once established, became the Western Hemi-

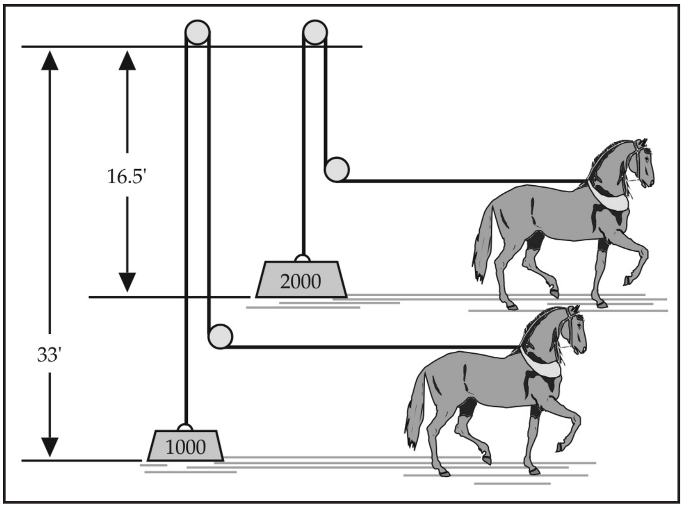


Figure 2 The average work-horse accomplishes work at a rate equal to 33,000 ft. lbs. in one minute; or equal to lifting 1 ton (2,000 lbs.) 16.5 feet, or 1,000 lbs., 33 feet in one minute.

sphere's standard for measuring the rate at which motors and other drives can produce work. For example, a 1 H.P. motor can produce 33,000 ft. lbs. of work in one minute.

Torque and horsepower are related to each other by a basic formula which states that:

$$\text{Horsepower} = \frac{\text{Torque} \times \text{Speed}}{\text{Constant}}$$

The value of the constant changes depending upon the units that are used for torque; the most frequently used combinations are:

$$HP = \frac{T \times S}{5252} \qquad T = \text{Torque in lb. ft.} \quad S = \text{Speed in RPM}$$

OR

$$HP = \frac{T \times S}{63{,}025} \qquad T = \text{Torque in lb. in.} \quad S = \text{Speed in RPM}$$

OR

$$HP = \frac{T \times S}{1{,}000{,}000} \qquad T = \text{Torque in in. ounces} \quad S = \text{Speed in RPM}$$

Rearranging these formulas to obtain torque, we can arrive at the equations:

$$T = \frac{HP \times 5252}{S} \qquad T = \text{Torque in lb. ft.} \quad S = \text{Speed in RPM}$$

OR

$$T = \frac{HP \times 63{,}025}{S} \qquad T = \text{Torque in lb. in.} \quad S = \text{Speed in RPM}$$

OR

$$T = \frac{HP \times 1{,}000{,}000}{S} \qquad T = \text{Torque in in. ounces} \quad S = \text{Speed in RPM}$$

In order to save time, graphs and tables are frequently used to show values of torque, speed and horsepower.

The previous discussion applies to calculations for *all* single-speed loads where the required torque and speed for a given operating condition are known.

Adjustable Speed Drives

When adjustable speed drives, such as DC SCR units, magnetic couplings, or variable frequency drives are to be utilized, a determination of *load type* must be made.

As previously mentioned, the most common type of load is the "constant torque" load. The relationships of torque and horsepower to speed for a "constant" torque load are shown (Fig. 3).

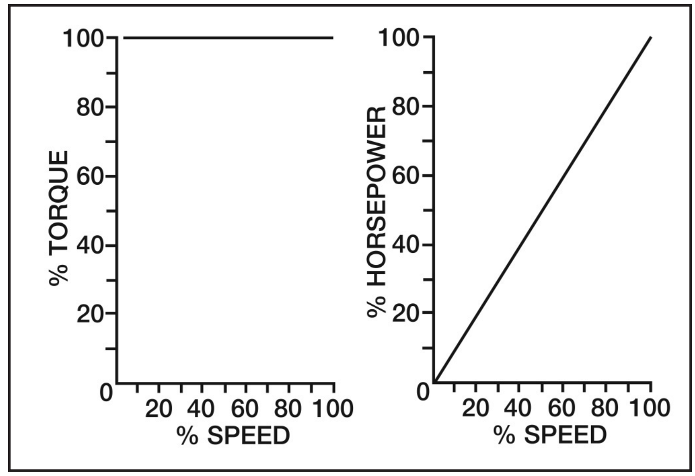


Figure 3 Constant Torque Speed-Torque Relationship

In the case of "constant torque" loads the drive must be sized to handle the following:

1. Torque required to break-away the load
2. Torque required to run load
3. Output *speed* required to operate machine at maximum required speed

Please note that only after the load has 1) been started, and 2) adequate torque is available to run it, does speed become a factor.

Only after these three items have been determined is it possible to calculate the required horsepower for the application.

Most adjustable speed drives are inherently "constant torque" devices; therefore no special considerations are involved in handling "constant torque" loads.

Constant Horsepower

A load type that occurs most frequently in metal working applications is the constant horsepower load.

On applications requiring constant horsepower the torque requirement is greatest at the lowest speed and diminishes at higher speeds. In order to visualize this requirement consider

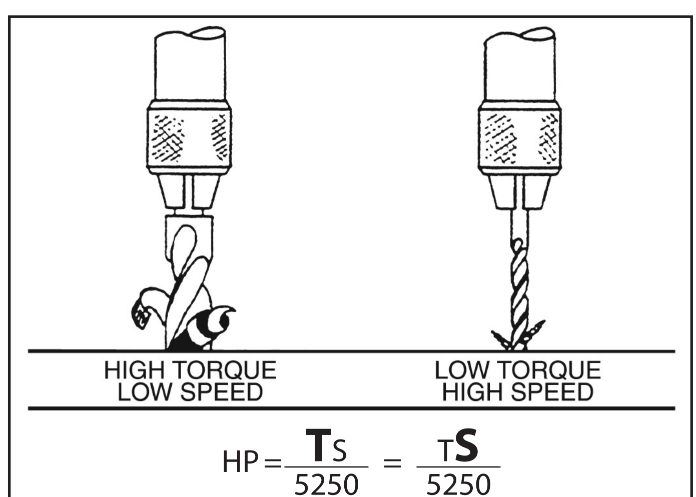


Figure 4 Constant HP Speed-Torque Relationships

the torque requirements of a drill press (Fig. 4).

When a *large hole* is being drilled, the drill is operated at a *low speed*; but it requires a very *high torque* to turn the large drill in the material.

When a *small hole* is being drilled, the drill is operated at a *high speed*, but it requires a very *low torque* to turn the small drill in the material.

A mathematical approach to this type of requirement would indicate that the HP requirement would be nearly constant, regardless of machine speed. Figure 5 shows the relationships of torque and horsepower to speed on constant horsepower loads.

As previously mentioned, this load type occurs most frequently on metalworking applications such as: drilling or boring; tapping; turning (lathes); planing; milling; grinding; wire drawing, etc. Center driven winders winding materials under constant tension also require constant horsepower. Constant horsepower can also be a requirement on some types of mixers.

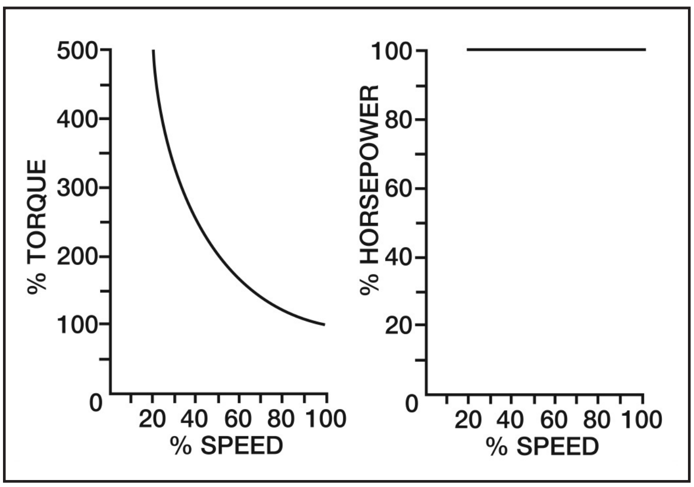


Figure 5 **Constant HP speed /torque relationships.**

An example of this might be a food mixer used to mix a variety of batters and dough. In this case dough would require *low speed* and *high torque*. Thin batters would require *high speed* and *low torque*; this is constant horsepower.

Spring coilers, four-slide machines, punch presses and eye-letting presses will frequently have torque requirements falling somewhere between the characteristics of constant horsepower and constant torque.

A general test for deciding if a machine might require constant horsepower would be to study the machine output. When a machine is designed to produce a fixed number of pounds-per-hour — regardless of whether it is making small parts at high speed, or large parts at a lower speed — the drive requirement is apt to be constant horsepower.

Although details of selecting drives for constant horsepower loads are beyond the scope of this presentation, some possibilities are cited.

For example, "constant horsepower" loads can be handled by oversizing drives such as standard SCR units or slip couplings. This is done by matching the drive's output torque with the machine's requirement at low speed. Depending upon the speed range that is required, this can result in gross oversizing at the high speed. More practical approaches involve using stepped pulleys, gearshift transmissions and metallic or rubber belt adjustable pitch pulley drives. Some additional and more sophisticated approaches are DC (SCR) drives operating with a combination of armature control at full field power up to base speed and field weakening above base speed. Some variable frequency drives can also be used at frequencies above 60 Hz, with voltage held constant to achieve a moderate amount of constant horsepower speed range.

Variable Torque

The final load type that is often encountered is the "variable torque" load. In general, variable torque loads are found only in centrifugal pumps, fans and blowers.

A cross-section of a centrifugal pump is shown (Fig. 6). The torque requirement for this load type can be thought of as being nearly opposite that of the constant horsepower load. For a variable torque load the torque required at *low speed* is very low, but the torque required at *high speed* is very high. Mathematically, the *torque* requirement is a function of the *speed squared*, and the *horsepower* is a function of the *speed cubed*.

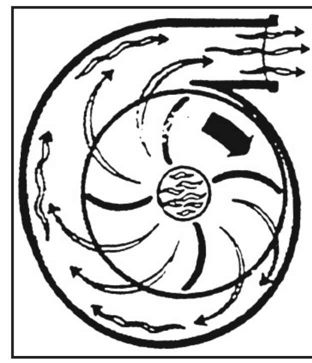


Figure 6 **Centrifugal pump — variable torque load.**

The relationships of torque and horsepower to speed on variable torque loads are shown (Fig. 7).

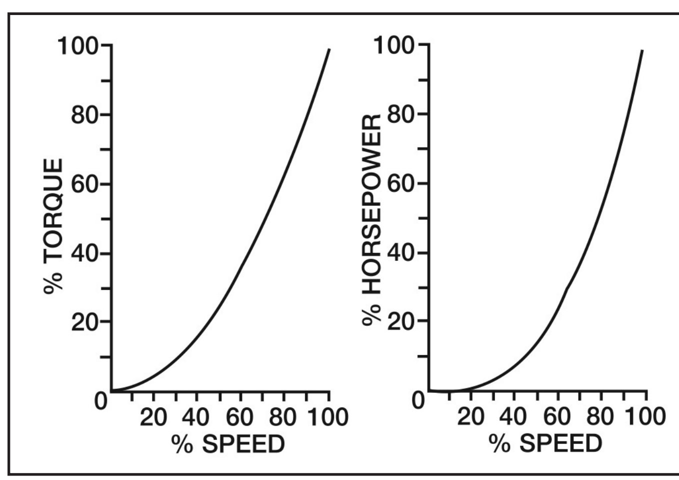


Figure 7 **Variable torque — speed-torque relationships.**

The key to drive sizing on variable torque loads is *strictly* related to providing adequate torque and horsepower at the *maximum* speed that will be required. *Maximum* must be emphasized, since a 9% increase in speed over the normal maximum will produce a 30% increase in the horsepower requirement.

It is impossible to speculate on the number of motors that have been burned out because people have unknowingly changed pulley ratios to obtain "more output" from their centrifugal pumps or blowers.

Table 2 illustrates the very dramatic changes in horsepower requirements for relatively small changes in speeds that occur with variable torque loads.

Table 2

% Speed Change	% Torque Change	% of Original HP	% HP Change
−20	−36	51	−49
−15	−28	61	−39
−10	−19	73	−27
− 5	−10	86	−14
0	0	100	0
+5	+10	116	+16
+10	+21	133	+33
+15	+32	152	+52
+20	+44	173	+73

Most variable speed drives are inherently capable of handling variable torque loads, provided that they are adequately sized to handle the horsepower requirement at *maximum* speed.

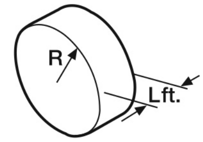
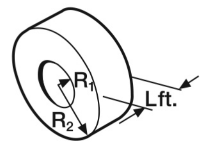
PART	RADIUS OF GYRATION (IN FEET)	SIZE OF LOAD WR^2 = POUNDS X $FEET^2$
CIRCULAR CYLINDER 	.71R	1.58 W LR^4 (R^4 = RxRxRxR) W = weight in pounds of one cubic foot of the material
HOLLOW CIRCULAR CYLINDER 	$.71\sqrt{R_2^2+R_1^2}$	1.58w L (R_2^4-R_1^4) W = weight in pounds of one cubic foot of the material

Figure 8 Formulas for determining radius of gyration and *WR*² of two frequently occurring cylindrical shapes: wt.: steel = 490; cast iron = 450; aluminum = 165.

High-Inertia Loads*

A discussion of load types would not be complete without including information on "high-inertia loads." **A load is generally considered to be "high inertia" when the reflected inertia at the motor shaft is greater than five times the motor rotor inertia.*

Inertia is the tendency of an object that is at rest to stay at rest, or an object that is moving to keep moving.

In the industrial drive business we tend to think immediately of flywheels as having high inertia; but, many other types of motor-driven equipment such as large fans, centrifuges, extractors, hammer mills, and some types of machine tools have inertias that have to be identified and analyzed in order to produce satisfactory applications.

The High-Inertia Problem

The high-inertia aspect of a load normally has to be considered only during acceleration and deceleration. For example, if a standard motor is applied to a large high-inertia blower, there is a possibility that the motor could be damaged or fail completely on its first attempt to start. This failure could occur even though the motor might have more than adequate torque and horsepower capacity to drive the load *after* it reaches the required running speed.

A good example of high inertia that most of us are familiar with is a jet plane taking off. In this case the maximum output of the engines is required to accelerate the weight of the plane and contents. Only when it has reached take-off speed and is nearly ready to leave the ground do the engines start doing the useful work of moving the plane to the final destination.

Similarly, when the plane lands the reversed thrust of the engines and brakes are used to slow down and stop the inertia of the plane.

In the motor and drive industry the inertia of a rotating body is referred to as the "WR^2" or "WK^2." In the English system "W" is the weight in pounds and "R" or "K" is the *radius of gyration* in feet. It is usually easy to obtain the weight of the body, but determining the radius of gyration can be a little more difficult. Figure 8 provides the formulas for determining the radius of gyration and WR^2 of two frequently occurring cylindrical shapes.

In most cases the WR^2 of flywheels can be determined by utilizing one or both of these normal shapes. In the case of flywheels having spokes, the contribution made by the spokes can generally be ignored and the inertia calculation based only on the formula for a hollow circular cylinder (Fig. 8); weight of the spokes should be included. If exact calculations are required, formulas are available to enable the calculation of WR^2 values of nearly any shape.

Similarly, equipment manufacturers will be able to provide the exact inertia values for a given application.

Motor manufacturers can be asked to supply the maximum WK^2 limits for any specific application requirement. (Please note WK^2 and WR^2 are used interchangeably and are the same).

The values shown in Table 3 are published in NEMA (National Electrical Manufacturers Association) standards MG 1. This table gives a listing of the normal maximum values of WK^2 that could be safely handled by standard motors. This table can be used as a guide. If the required WK^2 exceeds these values, the motor manufacturer should be consulted.

Why is High Inertia a Problem?

Prior to the time that a standard induction motor reaches operating speed, it will draw line current several times the rated nameplate value. The high current does not cause any problem if it is of short duration; but when the high currents persist for an extended period of time, the temperature within the motor can reach levels that can be damaging.

Figure 9 (a) presents typical plots of available torque from a standard motor vs. speed. Also plotted on curve (a) is the typical speed torque curve for a variable torque load. The values of A_1, A_2, A_3 and A_4 are the values of torque available to overcome the effect of the inertia and accelerate the load at different motor speeds as motor speed increases.

Table 3 Copyright NEMA MG 1							
	Speed, RPM						
	3600	1800	1200	900	720	600	514
HP	Load WK^2 (Exclusive of Motor WK^2), $Lb\text{-}Ft^2$						
1		5.8	15	31	53	82	118
1½	1.8	8.6	23	45	77	120	174
2	2.4	11	30	60	102	158	228
3	3.5	17	44	87	149	231	335
5	5.7	27	71	142	242	375	544
7½	8.3	39	104	208	355	551	799
10	11	51	137	273	467	723	1050
15	16	75	200	400	684	1060	1540
20	21	99	262	525	898	1390	2020
25	26	122	324	647	1110	1720	2490
30	31	144	384	769	1320	2040	2960
40	40	189	503	1010	1720	2680	3890
50	49	232	620	1240	2130	3300	4790
60	58	275	735	1470	2520	3820	5690
75	71	338	904	1810	3110	4830	7020
100	92	441	1180	2370	4070	6320	9190
125	113	542	1450	2920	5010	7790	11300
150	133	640	1720	3460	5940	9230	—
200	172	831	2240	4510	7750	—	—
250	210	1020	2740	5540	—	—	—
300	246	1200	3240	—	—	—	—
350	281	1370	3720	—	—	—	—
400	315	1550	—	—	—	—	—
450	349	1710	—	—	—	—	—
500	381	1880	—	—	—	—	—

Load WK^2 for Integral horsepower polyphase squirrel-cage induction motors. Table 3 lists the load WK^2 with integral-horsepower, polyphase squirrel-cage induction motors, and having performance characteristics in accordance with Part 12 (*i.e. — locked-rotor torque in accordance with 12.38.1, breakdown torque in accordance with 12.39.1, Class A or B insulation system with temperature rise in accordance with 12.43, and service factor in accordance with 12.51.2*), can accelerate without injurious heating under the following conditions:

1. Applied voltage and frequency in accordance with 12.44
2. During accelerating period a connected load torque equal to or less than a torque that varies as the square of the speed, and is equal to 100 percent of rated-load torque at rated speed.
3. Two starts in succession (coasting to rest between starts) with the motor initially at the ambient temperature, or one start with the motor initially at a temperature not exceeding its rated load operating temperature.

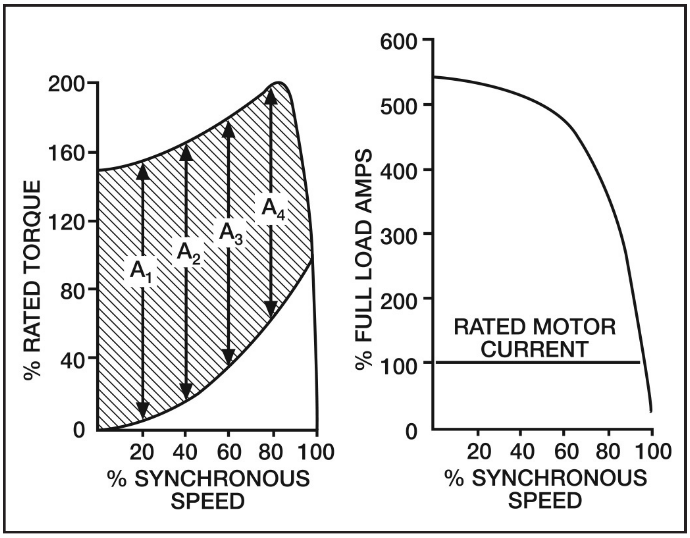


Figure 9 Acceleration period line currents (left and right).

Referring to Figure 9 (b), you will see that during the accelerating period this motor will draw line current that initially starts at 550% of rated current, and gradually drops off as the motor approaches rated speed. A great deal of heat is generated within the motor during this high-current interval. It is this heat build-up that is potentially damaging to the motor if the acceleration interval is overlong.

How Long Will it Take?

Calculating the time to accelerate a *direct-coupled* load can be determined quite easily by utilizing the following formula:

$$t = \frac{WR^2 \times N}{308T}$$

T = Average accelerating torque in lb. Ft.
N = Required change in speed
WR^2 = Inertia in lb. ft.^{2}
t = Time in seconds

The same formula can be rearranged to determine the average accelerating torque required to produce full speed in a given period of time.

$$T = \frac{WR^2 \times N}{308t}$$

Referring back to Figure 9(a), the accelerating torque would be the average value of the shaded area. In most cases, for standard motors through 100 HP, it is reasonable to assume that average accelerating torque available would be 150% of the motor full-load running torque and that accelerating times of 8–10 seconds, or less, would not be damaging, provided that starting is not repeated frequently. When load inertias exceed those shown in Table 4, the application should be referred to the motor supplier for complete analysis.

Reflected Inertias

Up to this point the only load inertias that have been considered have been rotating inertias *directly connected* to the motor shaft.

On many applications the load is connected to the motor by belts or a gear reducer. In these cases the "equivalent inertia" or "reflected inertia" seen at the motor shaft is the important consideration.

In the case of belted or geared loads, equivalent inertia is given by the following formula:

$$\text{Equivalent } WR^2 = WR^2_{LOAD} \left[\frac{N}{N_M}\right]^2 \times 1.1^*$$

WR^2_{LOAD} = Inertia of the rotating part
N = Speed of the rotating part
N_M = Speed of the driving motor

* Please note: the × 1.1 factor has been added as a safety factor to make an allowance for the inertia and efficiency of the pulleys (sheaves) or gears used in the speed change.

This formula will apply, regardless of whether the speed of the load is greater than, or less than, the motor speed.

Once the equivalent inertia has been calculated, the equations for accelerating time, or required torque, can be solved by substituting the equivalent WR^2 in the time or torque equation to be solved.

What Can Be Done?

When loads having high inertias are encountered, several approaches can be used. Some of the possibilities are:

1. Oversize the motor
2. Use reduced-voltage starting
3. Use special motor winding design
4. Use special slip couplings between the motor and load
5. Oversize the frame
6. Use an adjustable speed drive

Linear Motion

Occasionally applications arise where the load to be accelerated is traveling in a straight line, rather than rotating. In this case it is necessary to calculate an equivalent WR^2 for the body that is moving linearly. The equation for this conversion is as follows:

$$\text{Equilavent } WR^2 = \frac{W(V)^2}{39.5\,(S_M)^2}$$

W = Weight of load in pounds
V = Velocity of load in feet-per-minute
S_M = Speed of the motor in RPM when load moving at velocity V

Once the equivalent WR^2 has been calculated, the acceleration time, or required accelerating torque, is calculated by using the same equations for rotating loads.

Summary

- The turning force on machinery is *torque* — not horsepower.
- Horsepower blends *torque* with speed to determine the total amount of work that must be accomplished in a span of time.
- In all cases the horsepower required for single-speed application can be determined by utilizing the *torque* required at rated speed along with required speed.
- When variable speed drives are utilized, an additional determination of load type must be made. Most applications require either *constant torque* or *variable torque*. Metal cutting and metal forming applications frequently will require *constant horsepower*.
- High-inertia loads need to be approached with some caution due to high currents absorbed by the motors during the starting period. If there is any question regarding safe accelerating capabilities, the application should be referred to the motor manufacturer.

Note: An understanding of torque is essential for proper selection of any drive product. **PTE**

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ZF

OPENS EUROPEAN SERVICE CENTER

The opening of ZF's first European Service Center in Witten/Germany will focus on service and customer orientation in the industrial drives business. More than 30,000 gearboxes and 20,000 marine clutches of ZF Industrieantriebe Witten GmbH are in use worldwide. Many of them still had been delivered under the renowned brand of Lohmann & Stolterfoht.

In order to provide its customers with first class support in the aftermarket, ZF made a strategic decision bundling its service activities in Witten. In combination with the global service network of ZF Group, ZF Industrieantriebe positions itself as global strategic service partner for its customers. The company supplies products in several markets and for a broad variety of applications, such as mining excavators, offshore rigs, marine propulsions, recycling machinery, tunnel boring machines, cement mills or ropeways.

The new Service Center will offer ZF customers an extensive service portfolio, including multi-brand repairs. In close collaboration with ZF service hubs all over the world, the global service activities will be coordinated from Witten.

"We attribute great importance to a cooperative partnership with our customers, and this over the whole lifecycle of our gearboxes. Because of the broad variety of applications, it is essential to align our service activities to our customers' needs," said Christoph Kainzbauer, head of ZF's Industrial Drives business unit. (*www.zf.com*)

Bison Gear and Engineering

APPOINTS CHIEF EXECUTIVE OFFICER

The board of directors of Bison Gear and Engineering are pleased to announce the appointment of John Burch as chief executive officer of the company effective May 1, 2017. Burch succeeds Marcus J.S. Bours who has served as the interim CEO since June, 2016. The board greatly appreciates the service of Bours during this challenging transition period. Burch has over 25 years of general management, marketing, sales, business development and operations experience across a variety of industrial business to business markets. He joined Bison in 2008 as regional sales manager progressing to his current role in November, 2015, as the executive vice president, go to market. Burch earned an MBA from Kent State University and a B.S. in general engineering from the University of Illinois.

NFPA

FORMS STRATEGIC ALLIANCE WITH FIRST

The National Fluid Power Association (NFPA) recently started a strategic alliance collaboration with FIRST (For Inspiration and Recognition of Science and Technology), an international not-for-profit organization aimed at increasing interest and participation in science and technology among K-12 students. The partnership will harness the similarities between the mission of FIRST and those NFPA's workforce development to get more students aware of and interested in fluid power and engineering.

NFPA is excited to be a Strategic Alliance partner of FIRST as our members are very supportive of the FIRST mission as Mentors and competition Suppliers. We look to grow that participation and also make more students aware of the outstanding careers available in fluid power through the NFPA Robotics Challenge FIRST Scholarship," said Eric Lanke, president and CEO of NFPA.

The partnership will not only promote FIRST programs to NFPA's membership and education partners, but will also encourage them to become FIRST Mentors, Coaches, and Volunteers to extend the reach of NFPA's fluid power educational and career-building efforts to include FIRST students. NFPA will also offer a $40,000 merit-based scholarship to a high school senior who participates on a 2017 FIRST Robotics Competition team. The scholarship is to be used to study engineering at any accredited technical school or university in the United States.

"At FIRST, we work to broaden student exposure to a wide range of opportunities in STEM through our Mentors, Coaches, and Volunteers," said FIRST President Donald E. Bossi. "As a Strategic Alliance, NFPA will help give more FIRST students the opportunity to work side-by-side with professional engineers in the fluid power industry, while the generous NFPA Robotics Challenge FIRST Scholarship adds to the exclusive scholarship opportunities available for high school students who participate in FIRST." (*www.nfpa.com*)

Parker

RELEASES INTERACTIVE DRAWING TOOL

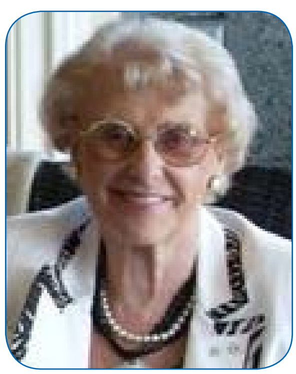
Parker Electromechanical and Drives Division has released a new interactive drawing tool. The same drawing tool that was launched for PS, PX, RS, and RX gearheads is now available for the BE, MPP, MPJ, and P- Series motors, and for PV gearheads. Drawings can be generated in a fraction of the time that it took before. Features include: 2D and 3D preview for immediate review of key dimensions, download options including the most popular 2D and 3D formats and the PDF download includes embedded 3D model for reference.

(*http://solutions.parker.com/LP=8335?cm_mmc=Eloqua-_-Email-_-LM_AUG_EMN_BE+Motor+Gearhead+Dwg+Tool-PU*)

PTDA

ACCEPTING NOMINATIONS FOR WENDY B. MCDONALD AWARD

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



A trailblazing woman business owner, Mrs. McDonald left many legacies through her long career in the industry. Her charm and grace are legendary as well as her philanthropy and commitment to give back to the industry and the communities that led to her success. The 2016 recipient of the award was Linda Miller of B&D Industrial. Previous recipients include Elizabeth (Liz) Moon of Kaman Industrial Technologies Corporation and Pat Wheeler of Motion Industries (Canada).

When merited, the Wendy B. McDonald Award will be presented annually. Nominations are now being accepted through June 15, 2017, and will be judged by the following criteria:

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

Nominees must exemplify leadership and integrity in all business relationships.

Download the criteria and nomination form at *www.ptda.org/wbmcdonaldaward*.

Hydraulic Institute

NAMES MEMBER OF THE YEAR

The Hydraulic Institute (HI) board of directors named Michael Coussens, director of development and chief engineer — Peerless Pump Company, as its 2016 Member of the Year. The award was conferred to Coussens during HI's 2017 Annual Conference & Centennial Celebration Gala held March 10th at the JW Marriott Grande Lakes, in Orlando, Florida.

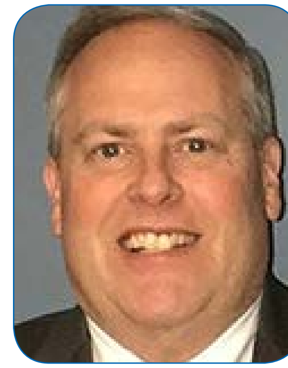
Coussens was selected for his tireless dedication to HI standards development, volunteerism, and his many significant technical contributions to the industry. He has held leadership positions on more than eight technical committees over the past eight years. Currently, he holds leadership positions as:

- Chair of Rotodynamic Pump Design & Operation Section
- Vice-Chair Standards Committee
- Chair 9.6.7 Effects of Liquid Viscosity on Rotodynamic Pump Performance
- Co-Chair 14.3 Rotodynamic Pump Design & Application
- Co-Chair 40.7 HI Program Guide for HI Pump Test Laboratory Approval
- Vice-Chair Fire Pump Committee
- Vice-Chair 14.6 Rotodynamic Pump Test

Pete Gaydon, technical director, HI, added, "Michael distinguishes himself by the level and extent of his leadership to advance the technical work of HI on many levels. With his calm demeanor and attention to detail, he has been a great mentor and role model for our young engineers. His tireless dedication to standards development and many significant technical contributions to the industry sets him apart." (*www.pumps.org*)

IDC-USA

APPOINTS EXECUTIVE VICE PRESIDENT



IDC-USA has announced the addition of **Bob Boyle** as executive vice president. Boyle is a graduate of Texas A&M University with an engineer's degree in industrial distribution. He has worked in the bearing and power transmission industry his entire career and has served in various positions including district sales engineer, aggregate and cement marketing manager, and director of power transmission and material handling.

"Bob brings a wealth of knowledge in managing supplier relationships for mutual growth and prosperity. He will serve our member companies well, and he will be a real asset to our preferred supplier partners. I am thrilled we have landed such a superior talent to join our team. His experience will be invaluable to our organization," stated George Graham, president and CEO of IDC-USA.

"I am extremely excited about the opportunities that lie ahead. IDC-USA allows independent business owners to truly partner with their customers, and not just be parts providers. The ability to fully understand and service customer needs is what sets independent distributors apart in the industrial market segment, and is why I chose to make this career move," commented Boyle.

Boyle's primary focus will be to grow existing supplier programs and to recruit new suppliers to the cooperative. "Bob will be instrumental in our continuous quest to add value to our independent distributors and our preferred supplier partners. Bob is a tremendous addition to the IDC family," Graham said. (*www.idc-usa.com*)

Nook

NAMES CHIEF OPERATIONS OFFICER

Nook Industries recently announced **Jim Rowell** as its new chief operations officer. In this position, Rowell is responsible for overseeing Nook's leadership initiatives, driving operational strategy and supporting employee development. Rowell has more than 25 years of industry experience. Prior to joining Nook, he held roles in Parker Hannifin's Fluid Connectors and Automation groups as territory manager, product manager, business unit manager, manufacturing manager, general manager and vice president of operations.

"Along with more than two decades of industry knowledge, Jim brings a global strategic perspective and leadership skills that will greatly help Nook achieve many of its personal and business objectives," said Joseph H. Nook III, president and CEO at Nook Industries. "We are confident in his abilities to accomplish complex initiatives that require quick execution as Nook moves forward as an industry leader."

Rowell earned a bachelor's degree in industrial engineering from the University of Cincinnati and completed the executive education program at the University of Michigan. (*www.nookindustries.com*)

Custom Machine & Tool

LAUNCHES NEW WEBSITE

Custom Machine & Tool Co., Inc., (CMT) a U.S. manufacturer in the power transmission and motion control products industry, recently announced they have launched a new interactive website with more viewing options and content.

"We are very pleased to see the results of our customers' feedback implemented into a new dynamic interactive website. Visitors to the new site now have more viewing options, enjoy easy navigation, and fast access to the detailed information in our product catalogs," says owner and president of Custom Machine & Tool Co., Inc., Robert Bennett.

Custom Machine & Tool Co., Inc. manufactures precision timing pulleys; the patented Concentric Maxi Torque keyless hub to shaft connection systems; drive systems; and components for the motion control and power transmission markets. CMT has been the preferred choice for pulley stock by OEM's and distributors for over 45 years. The company guarantees shipment of up to five pieces of pulley stock within 72 hours. See 'Timing Pulley Stock Shipping Program' for more details. (*www.cmtco.com*)

ABB

ACQUIRES B&R

ABB recently announced the acquisition of B&R, the largest independent provider focused on product- and software-based, open-architecture solutions for machine and factory automation worldwide. B&R, founded in 1979 by Erwin Bernecker and Josef Rainer is headquartered in Eggelsberg, Austria, employs more than 3,000 people, including about 1,000 R&D and application engineers. It operates across 70 countries, generating sales of more than $600 million (2015/16) in the $20 billion machine and factory automation market segment. The combination will result in an unmatched, comprehensive offering for customers of industrial automation, by pairing B&R's innovative products, software and solutions for modern machine and factory automation with ABB's world-leading offering in robotics, process automation, digitalization and electrification.

Through the acquisition, ABB expands its leadership in industrial automation and will be uniquely positioned to seize growth opportunities resulting from the Fourth Industrial Revolution. In addition, ABB takes a major step in expanding its digital offering by combining its industry-leading portfolio of digital solutions, ABB Ability, with B&R's strong application and software platforms, its large installed base, customer access and tailored automation solutions. (*www.abb.com*)

Bishop Wisecarver

CELEBRATES 10 YEARS OF FIRST ROBOTICS ROBOTICS SPONSORSHIP

In 2017, Bishop-Wisecarver Group (BWG) is a Diamond Supplier at the national level and supports four regional teams who are learning the importance of STEM-based academics, as well as the needed 21st century work skills of problem solving, critical thinking and collaboration.

"Supporting the FIRST program for ten years has been a rewarding experience on every level as we've watched students learn about STEM opportunities in a hands-on team experience that has provided inspiration and confidence for their futures," said Pamela Kan, president of Bishop-Wisecarver Group. "We talk often about the skills gap issues facing STEM-related fields, including manufacturing, and programs like FIRST are making a notable, positive difference in this area. FIRST has tracked that 75 percent of their alumni are in a STEM field as a student or professional—those are amazing results!"

FIRST (For Inspiration and Recognition of Science and Technology) was founded in 1989 to inspire an appreciation of science and technology in young people and now includes several different competitions for various age groups and interests. The 2017 FIRST programs will include more than 3,000 teams from 25 countries, more than 83,000 youth team members and 20,000 mentors and adult sponsors, as well as $50 million in scholarships. (*www.bwc.com*)

Doubling Down and Forging Ahead

After getting positive feedback on changes made to the show last year, AWEA Wind Power is doubling down and taking their ongoing transformation even further.

Alex Cannella, News Editor

Much like the industry it serves, AWEA Wind Power is a trade show in flux that's focused on self-improvement. And it shows. Going into their 2017 show, Wind Power is running with the motto: Brand New Attitude, and they're working to live up to it. According to the American Wind Energy Association (AWEA)'s Senior Vice President for Member Value and Experience, Jana Adams, the motto has a triple meaning to reflect both a changing industry and show.

On the industry side, big investor names are starting to become more and more common and the industry's public face is shifting. Last year, wind power became both the largest overall and fastest growing provider of renewable energy in the U.S. According to the Bureau of Labor Statistics, "Wind Turbine Technician" is the fastest growing job in the country. The industry now provides 5 percent of the country's electricity (that's over 17.5 million homes), and some individual states get up to 30 percent of their power from wind. Between all the new faces and growth, the wind industry is changing. Wind power isn't a nascent experiment to produce cleaner energy anymore. It's an established industry taking its place at the big boys table, and there's optimism both in the industry and at AWEA that it will keep growing. The AWEA believes that the wind power industry could provide 10 percent of the nation's energy — effectively double in size — by 2020.

"Wind the industry has a new attitude," Adams said. "We aren't this up-and-coming, new, emerging technology. We're established...We're kind of demonstrating that established 'we're here, we're a significant part of the U.S. economy, we're a great American resource.' And we really want to drive that home with this brand new attitude."

Wind Power the trade show started reflecting the industry's "new attitude" with a number of changes at their last show in 2016. Amongst the changes AWEA made, the most prominent were the introduction of "education stations" and the decision to change their registration to all-access.

The education stations were introduced last year as hubs for the show's scheduled educational programming. Seminars covering everything from finance to maintenance happened directly on the trade show floor. And with the all-access pass, all attendees had full access to the entire show, educational programming included, as opposed to having to choose between just walking the trade show floor or paying for an expensive ticket for the full conference.

"We really revolutionized the event last year where we did away with the distinction between companies or individuals coming to attend education programming or just going to the trade show or just going to these individual segments," Adams said. "It's now all-access, so if you come to the event, you can access anything you want with one simple registration, really streamlined the way people participate in the event, and more importantly, drove everything to the exhibit show floor, so whether you're going to have a private meeting with a company, visit the exhibit hall or attend a conference session, it's all right there on the trade show floor."

According to Adams, last year's changes have been met with a wide range of positive feedback, and both the education stations and the all-access passes are making a return this year.

"It makes it such a different experience to pop in and listen to a 25-minute presentation on a cool topic, and then walk next door and meet with your big customer," Adams said. "It really changes the dynamic of how people look at their schedules and plan their days at Wind Power, and we saw a massive increase in the consumption of the education."

In addition to returning changes from last year, AWEA is continuing to tinker with their trade show's formula to further centralize the show experience and save time for attendees. The main focus is on getting the entire show onto the same convention floor. Alongside the exhibitor booths and education stations, Wind Power will now also have meeting rooms both for individual business meetings as well as larger conferences to discuss industry policies. The idea is that the more Wind Power centralizes its functions, the more attendees will be able to focus on business or enjoy the show instead of wasting time in transit.

"The changes that we're making to the show make it so much of a different experience to be involved," Adams said. "I used to talk to people who would come to Wind Power. The amount of steps they would get on their little watches that track their fitness level would just be ridiculously astronomical because they're constantly running between the trade show floor and their company booth, then going two miles away to a hotel to have one meeting and then running back to go to a program in the convention center. And now that we've really driven all of that activity to one convention center trade show floor where you can do all of that right there makes it so much more of an interactive, networking experience."

And if you liked last year's educational offerings, you're in luck: AWEA is expanding their educational curriculum at the show this year.

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Wind
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"[We're] always looking for ways to grow that value that people get for their investment in the show, and expanding the hours that people are able to learn was definitely a goal," Adams said.

According to Adams, the goal was to expand the number of hours of education available without expanding the number of sessions. This translates into some changes in focus, such as focusing more on seminars with wide-reaching appeal instead of having a number of smaller sessions all running simultaneously.

This also means a focus on Wind Power's "general sessions," which will also be taking place on the main trade show floor this year. The general sessions are the show's mainstay panels and seminars. One session focused on the day's overall theme will be presented each morning.

Another small change the show is making is to open the trade show floor during the general sessions. However, the sessions will also be simulcast throughout the exhibit hall for those who can't see it in person, as well as online. If you can't make it out to Wind Power but are still interested in hearing some educational seminars from some of the show's biggest names, the general sessions can be watched online for free! The general sessions will take place 10 am PST on each morning of the show. Tuesday morning (May 23) will feature the Wind Industry Leaders Panel. Wednesday will focus on positioning the market competitively against other industries. Thursday will look at what market forces and trends might affect the industry in the future.

Between the changes to further centralize the show and a strong, growing industry to cover, Wind Power 2017 is expecting a strong year. According to Adams, every measurable sign of attendance is on-pace to outdo 2016. She believes wind power, both the industry and the show, have a bright future ahead.

"I think we will see and feel that in Anaheim," Adams said. **PTE**

For more information:

AWEA Wind Power
Phone: (202) 383-2500
www.windpowerexpo.org

April 30–May 3—CIM 2017 Convention Montreal, Quebec. Founded in 1898, the Canadian Institute of Mining, Metallurgy and Petroleum (CIM) is the leading technical society of professionals in the Canadian Minerals, Metals, Materials and Energy Industries. The CIM Expo features nearly 450 companies showcasing the latest in mining equipment, tools, technology, services and products. The event includes plenary sessions intended to bring focus and start dialogue around the conference theme of "New State of MINE." Leaders from all aspects of mining and some from unexpected tangential sectors are brought together in these thought-provoking discussions. For more information, visit *convention.cim.org*.

May 1–4—Offshore Technology Conference 2017 Houston, Texas. The Offshore Technology Conference (OTC) is where energy professionals meet to exchange ideas and opinions to advance scientific and technical knowledge for offshore resources and environmental matters. OTC is the largest event in the world for the oil and gas industry featuring more than 2,300 exhibitors and attendees representing 100 countries. Founded in 1969, OTC's flagship conference is held annually in Houston. The event provides excellent opportunities for global sharing of technology, expertise, products, and best practices. OTC brings together industry leaders, investors, buyers, and entrepreneurs to develop markets and business partnerships. Technical highlights include updates on world-class projects, offshore renewable energy, the digital revolution, safety and risk management and more. For more information, visit *http://2017.otcnet.org/Content/Welcome*.

May 2–4—AGMA Gearbox System Design Clearwater Beach, Florida. Learn the supporting elements of a gearbox that allows gears and bearings to do their jobs most efficiently. Gain a deeper understanding about seals, lubrication, lubricants, housings, breathers, and other details that are involved in the designing of gearbox systems. Gear design engineers; management involved with the design and manufacture of gearing type components; metallurgists and materials engineers; laboratory technicians; quality assurance technicians; furnace design engineers; and equipment suppliers should attend. Instructors include Raymond Drago and Steve Cymbala. For more information, visit *www.agma.org*.

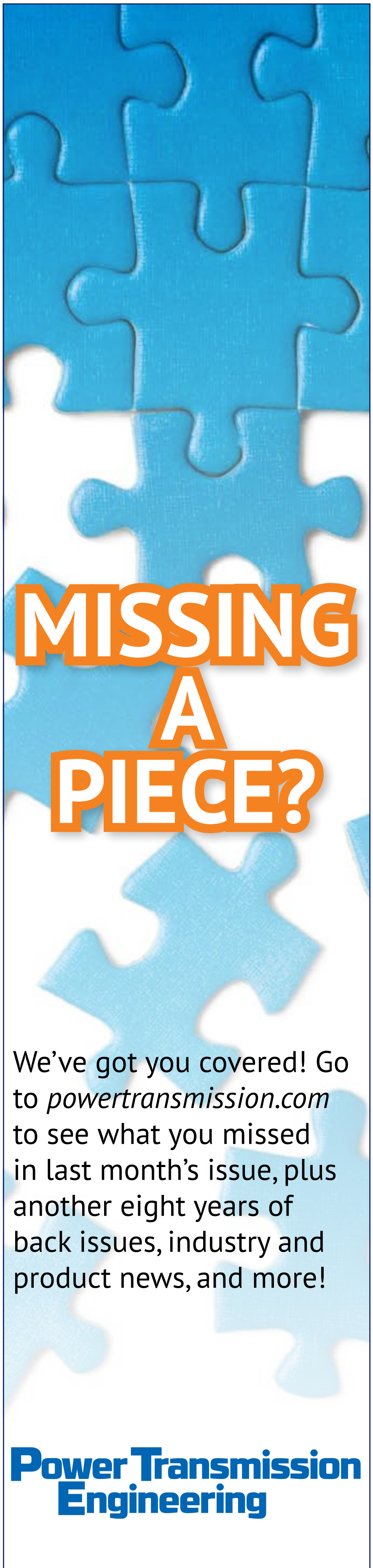
May 8–10—Industrial Design and Engineering Show Huntington Convention Center, Cleveland, OH. The three-day event brings together design engineers, industrial engineers, system integrators, plant operations and high-level decision makers, exhibitors and numerous conference sessions across three tracks in manufacturing operations and design engineering. The show is co-located with the IndustryWeek Manufacturing & Technology Conference & Expo and the MESA unConference that brings the global MES communities together to highlight what's new about Smart Manufacturing. Conference attendees can tour their choice of a world-class Cleveland-area facility for a behind-the-scenes look at continuous improvement initiatives, use of technology, and workforce best practices. The exhibit hall allows guests to keep up-to-date with emerging technologies and discover companies that have solutions for their business needs. Some of the activities for 2017 include viewing demonstrations of drones, robotics, virtual reality, live 3D printing, live cybersecurity hacks and a "demo drive" where attendees will rotate around the exhibition hall. For more information, visit *www.mfgtechshow.com*.

May 8–11—AISTech 2017 Nashville, TN. This event will feature technologies from all over the world that help steel producers to compete more effectively in today's global market. AISTech is a can't-miss event for anyone involved at any level of today's steel marketplace, providing perspective on the technology and engineering expertise necessary to power a sustainable steel industry. More than 8,000 people are expected to attend AISTech 2017. Along with over 500 exhibiting companies, AISTech 2017 allows attendees to meet face-to-face with key individuals involved in the production and processing of iron and steel. The AIST Conference programs are developed by technology committee members representing iron and steel producers, their allied suppliers and related academia. Committees focus on ironmaking, steelmaking, finishing processes, and various engineering and equipment technologies. For more information, visit *www.aist.org*.

May 8–11—Rapid + TCT 2017 RAPID + TCT is an additive manufacturing event that showcases product innovations and offers collaborative learning opportunities to ultimately accelerate the adoption and advancement of the technology. The two industry leaders in 3D technology events, SME and Rapid News Publications Ltd., are combining their nearly 30 years of insights and experience to produce the annual RAPID + TCT event starting in 2017. At RAPID + TCT, attendees will have the opportunity to engage with the most influential community in 3D technology. Explore the future of the industry through interactive experiences, 200+ hands-on exhibits, keynotes, and conference presentations from industry leaders. For more information, visit *www.rapid3devent.com*.

May 9–11—Railtex 2017 Birmingham, U.K. Railtex examines technological innovations across the entire rail supply market. It provides an opportunity in the U.K. for companies serving all aspects of the infrastructure and rolling stock sectors to present their capabilities, meet their customers and be part of the industry's networking event of the year. First staged in 1993, Railtex has long been established as Britain's leading showcase for railway equipment, products and services, with a strong reputation for attracting visiting managers, engineers and buyers at the highest level. With the exhibition as its centerpiece, Railtex additionally features a stimulating supporting program encompassing keynote speeches, seminars and discussion forums all devised to highlight industry trends and bring the industry together. For more information, visit *www.railtex.co.uk*

May 16–18—EASTEC 2017 West Springfield, Massachusetts. With more than 500 exhibitors, complimentary conference sessions, industry keynotes and much more, EASTEC is an event dedicated to keeping northeast manufacturers competitive. It's where manufacturing ideas, processes and products that make an impact in the northeast region, are highlighted through exhibits, education and networking events. For more information, visit *www.easteconline.com*.



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- ☐ Other (please describe) (15)____________________________

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

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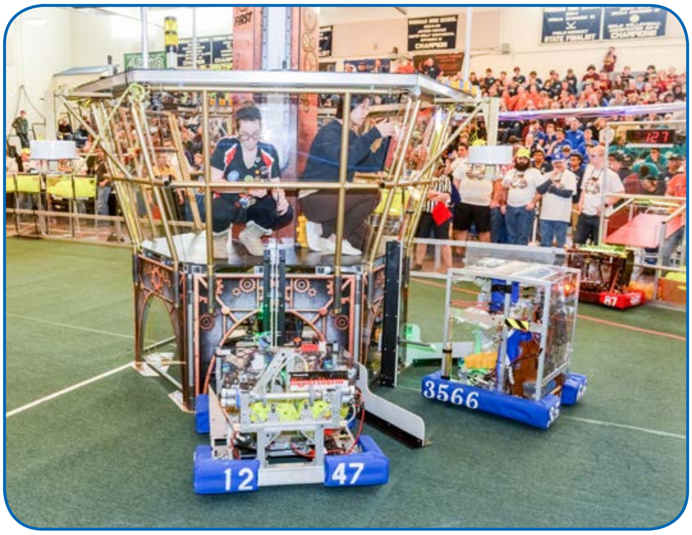
BATTLE BOTS Victorian Edition

Art, Science and Technology Mesh at FIRST Steamworks Competition

Matthew Jaster, Senior Editor

Airships, gears, robots, steam power and buzzer-beating theatrics highlight the 2017 FIRST (For Inspiration and Recognition of Science and Technology) Robotics Competition known as FIRST Steamworks.



For this year's event, organizers were interested in combining the steampunk aesthetic with a traditional robotics competition, according to Frank Merrick, director, FIRST Robotics Competition.

Students were given a kit of parts including motors, batteries, control system components, construction material and automation components. They had six weeks to work with adult mentors to design, build and test their robots to meet the engineering challenges of the competition.

The game itself consists of two alliances—three teams per alliance—facing-off in matches that last only two minutes and thirty-seconds. The concept of FIRST Steamworks is to prepare airships for a long distance race by fueling the ship, completing the geartrain and getting their robots on the craft prior to 'take-off.'

How exactly is all this accomplished?

First, the competitors utilize robots to collect fuel (whiffle balls) and build steam pressure (launch whiffle balls into high and low boilers for various points). Robots also deliver gears to the pilots on the airship for installation. Once the geartrain is complete, the pilots can turn the crank to start the rotors. Before time expires, the robots must also latch on to their respected airships by essentially climbing a rope to board the craft.

The alliance that is best prepared for 'take-off' when time expires wins the race and the match.

For the first time in the history of the competition, you'll find human players on the field right in the middle of the action.

"Pilots must make sure they get the gears delivered by their robots installed quickly and correctly to get those rotors turning, and must deploy the ropes the robots will eventually climb at the end of the match. It's a high-stakes, high-pressure position, and the students love it!" Merrick said.

The final moments of each match when six robots are trying to climb their ropes before the buzzer sounds can be heart-stopping, according to Merrick, and they add a level of excitement to this year's competition.

"We've never had a rope-climbing challenge in any of our games before, and had no historical information to use to estimate, but the teams have delivered the goods with climbing-capable robots!" Merrick said.

There are currently about 3,350 FIRST Robotics Competition teams worldwide. While most of those teams are school-based, they also come from Girl Scouts, 4H and other community organizations. There are even independent teams with no formal affiliation with other groups.

The competition has grown so large that they have scheduled two FIRST Championship locations this year. One will take place in Houston, Texas, on April 19-22, 2017 and the other in St. Louis, Missouri, on April 26-29, 2017.

Winning alliances at each event receive trophies, banners, medals for team members, and automatic invitations to the 2018 FIRST Championship. Additionally, the Chairman's Award is presented to the team that embodies the purpose and goals of FIRST competitions. This team is inducted in the FIRST Robotics Competition Hall of Fame and receives a lifetime invitation to the FIRST Championships.

Every time Merrick attends these competitions, he becomes more optimistic about the future of science and engineering.

"I see young people working so hard, in teams, on seriously challenging problems for which there is no 'right' answer! It honestly runs counter to the stereotypical image of lazy, over-indulged teenagers. *FIRST* isn't easy, and it's designed to not be easy, so when student participants complete a season they can look back with authentic pride in their accomplishments. This gives them a taste of their true capabilities."

(*www.firstinspires.org*) **PTE**