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SEPTEMBER 2015

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The SureGear series of high-precision servo gear reducers is an excellent choice for applications that require accuracy and reliability at an exceptional value.

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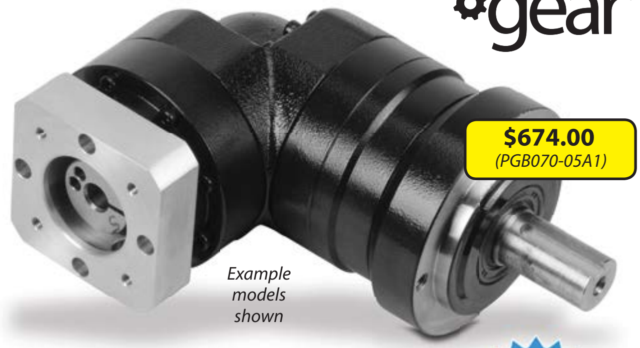

$674.00
(PGB070-05A1)

Example models shown

$398.00
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NEW!

$722.00
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Mates easily to SureServo motors

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*start under $1,000***

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*** All components sold separately.*

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www.automationdirect.com/motion-control

Servo Systems	AutomationDirect Price/Part Number	VS. Allen-Bradley Price/Part Number
Digital Servo Drive	$488.00 SVA-2040	$1,340.00 2098-DSD-005
100W Servo Motor with connectorized Leads	$325.00 SVL-201	$558.00 TLY-A130T-HK62AA
Breakout Board Kit for CN1 Control Interface	$94.00 ASD-BM-50A	$263.00 2090-U3BK-D4401
10' Motor Feedback Cable	$49.50 SVC-EFL-010	$90.00 2090-CFBM6DF-CBAA03
10' Motor Power Cable	$29.50 SVC-PFL-010	$101.00 2090-CPBM6DF-16AA03
Configuration Software	FREE SV-PRO*	$82.00 2098-UWCPRG
Complete 1-axis 100W System	$986.00	$2,434.00

*SureServo Pro software is FREE when downloaded and is also available for $9.00 on a CD

All prices are U.S. list prices, AutomationDirect prices as of 6/5/15.
The Allen-Bradley 100W system consists of part numbers shown in table above with prices from www.wernerelectric.com, www.rexelusa.com 5/25/2015.

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SEPTEMBER 2015

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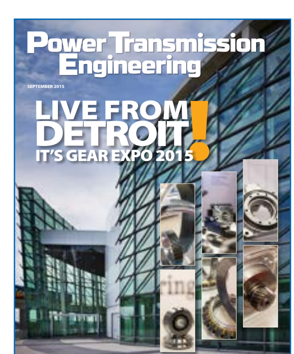
FEATURE ARTICLES

TECHNICAL ARTICLES

Vol. 9, No. 6. POWER TRANSMISSION ENGINEERING (ISSN 2331-2483) is published monthly except in January, May, July and November by Randall Publications LLC, 1840 Jarvis Ave., Elk Grove Village, IL 60007, (847) 437-6604. Cover price $7.00. U.S. Periodicals Postage Paid at Elk Grove Village IL and at additional mailing offices. Send address changes to POWER TRANSMISSION ENGINEERING, 1840 Jarvis Ave., Elk Grove Village, IL 60007.

Power Transmission Engineering

VOL. 9, NO. 6



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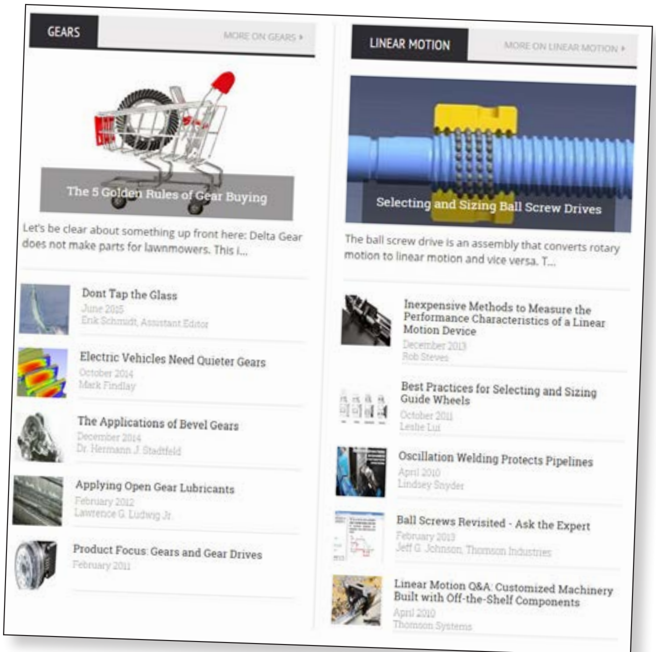
Inset photos by David Ropinski

www.powertransmission.com



This Month's Highlighted Topics

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

Gears
Linear Motion

Buyers Guide: Recently Added

The *Power Transmission Engineering* Buyers Guide is your fastest way to find information on suppliers of mechanical power transmission products. Check back often, because we're always improving the site and adding new companies to the listings.

AD Together, we win.

AmTech INTERNATIONAL

CUSTOM GEAR & MACHINE INC.

Ask the Expert!

We'll be hosting a LIVE edition of our popular Ask the Expert column at the upcoming Gear Expo (October 20–22) in Detroit. If you're coming to the show, please visit us at booth #2030 to tap into some of the brightest minds in gearing. Not coming to the show? No worries! You can ask your question NOW via our website, and we'll pose it to the experts during one of the panel sessions.

www.powertransmission.com/asktheexpert.php

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Top 5 Reasons To Go to Gear Expo

Gear Expo 2015 takes place October 20–22 in Detroit. If you haven't already made plans to attend the show, you might be might be missing out on a great chance to learn more about gears, find new potential suppliers and network with your peers. Gear Expo is the industry's premier trade show, and it provides plenty of opportunities for anyone whose work involves gears — from manufacturing to processing, purchasing and use.

Gear Expo used to be viewed as mostly a show for gear *manufacturers* to attend. Exhibitors included the major machine tool manufacturers, cutting tool suppliers and service providers aimed at supplying the gear manufacturing process. The show still includes all of those exhibitors, but over the years, Gear Expo's emphasis on the *entire* supply chain has grown — meaning that the show has become more and more important to gear *buyers*, because more and more gear and gear drive manufacturers are exhibiting.

For most of you who read *Power Transmission Engineering*, that's a very good thing, especially if you're involved with the specification, purchase and use of gears. Finding the right gear supplier can make all the difference in a successful product.

But there are many reasons to go to Gear Expo. Here we've laid out a few of the most important ones:

1. Learn more about how gears are made. Gears cannot be properly designed without a solid understanding of the manufacturing processes that will be used to make them. In recent years, some of those processes have been improved, and new processes are always being developed. In order to design and develop gears that cost less, run more quietly or handle more power, you need to learn more about those changes and the latest gear manufacturing technologies. Who knows, you might even consider bringing gear manufacturing in-house rather than outsourcing it (or vice-versa!).

2. Find new gear suppliers. There will be more than 70 gear and gear drive manufacturers exhibiting at the show, as well as vendors of gear software, lubrication, bearings and automation solutions. See our map and listings on pages 32–34 for a complete list of the Gear Expo exhibitors of most interest to *Power Transmission Engineering* readers.

3. There's always something new. Many of these exhibitors are interested in speaking with you. Our Gear Expo preview article, beginning on page 24, touches base with a number of them. Also, don't forget to read our special "Showstoppers" advertising section, beginning on page 29, to see what some of the Gear Expo exhibitors have to offer.

4. Educational opportunities. Visit *www.gearexpo.com* to learn more about the many educational programs put on by the AGMA in conjunction with the show. These include the Fall Technical Meeting (which takes place Oct 18–20, overlapping with the show), a number of gear-related seminars, and the Solutions Center, an educational theater right in the middle of the show floor.

5. Ask the Expert Live. In conjunction with *Gear Technology* magazine, we've put together some educational sessions of our own. For the first time ever, we're offering a live and in-person version of our popular "Ask the Expert" column. At Gear Expo, we're hosting four sessions of Ask the Expert Live, each one focused on a specific gear-related topic and featuring three or four renowned experts. See our ad on page 61 for a full schedule of topics and presenters who will be on hand to answer your questions.

Of course, we'll be at the show, and we'd love to see as many of you as possible. Please stop by Booth #2030 to meet our editors, participate in Ask the Expert Live and learn more about our print and electronic subscription options. Whether you can make it to the show or not, we're always interested in your feedback about how we can serve you better. Send me a note at *wrs@powertransmission.com* with any suggestions or comments.

Randy Stott

Heidenhain Renco Encoders

POSSESS MODULAR DESIGN WITHOUT AN INTEGRAL BEARING

Heidenhain's Renco brand of encoders is available for use within a variety of mobile robot applications. Specifically, the modular Remco R35i and RCML15 rotary encoders are designed for motion and speed feedback control in automated guided vehicle (AGV) projects used in distribution warehouses, manufacturing plants and medical facilities of note.

The Renco R35i rotary encoder is particularly noted for its modular design without an integral bearing. Its special properties are its compact design with a 35 mm outside diameter and only 14 mm height as well as its easy, self-centering mounting thanks to a patented slide lock. With its OPTO-ASIC technology, the Renco R35i offers improved functionality with the smallest possible dimensions.

The Renco RCML 15 rotary encoder offers similar reliability along with the OPTP-ASIC technology while having a height of only 8.9 mm, offering an alternative to the R35i with a low mounting profile.

AGVs are mobile wheel-based robots designed to carry a load through a facility without an onboard operator or driver. Due to new advancements of technologies used within the intelligent and flexible material handling field, many large warehouses and distribution centers have adapted to using AGVs. This switch has led to an increase in efficiency and a reduction in costs by automating some of the manufacturing facility or warehouse.

The Renco brand R35i and RCML15 rotary encoders currently provide motion and speed feedback in AGV motors which operate a multitude of the axes, with the most common axis being the motors used on the AGV drive wheels. When using the Renco brand encoders, an AGV system can accurately monitor and establish its speed with consistent repeatability. These aspects are crucial to the machines as they may need to interact directly with a person, and this level of quality is required in order to ensure the safety of the people involved.

The Renco encoders' slim and lightweight design allows motor designers more flexibility, especially in these applications where space and weight is critical. The Renco encoders combine brushless motor commutation pulses and incremental position feedback, which reduces the cost while improving the performance and reliability of the brushless motor/encoder package.

Typical AGV applications include transportation of materials (raw, work-in-process, and finished goods), storage/retrieval in support of picking in warehousing and distribution applications, carrying medical supplies and equipment within hospitals, and material handling within clean rooms in the semiconductor industry.

For more information:
Phone: (847) 519-4218
www.heidenhain.us

KISSsysWeb

ALLOWS COMPANIES SIMPLE CALCULATION OF KISSSYS MODELS

KISSsoft recently introduced the KISSsysWeb, a new product that allows companies to use the calculation of KISSsys models in an easy and simple way using the Internet. This application is designed for sales people who want to quickly recalculate already existing gearboxes with different operating values.

The KISSsys models are selected from the tree structure and the corresponding loads are defined. Not only the speed and torque parameters, but also load spectra as well as radial and axial loads can be specified.

With a single click, the gearbox is calculated and a report is created with service life, safeties, efficiency and other results. This report can then be further used in PDF format.

The models are located on the web server of the company's network and the application is password-protected so that all data is secure.

For more information:
Phone: +41 55 254 20 50
www.KISSsoft.AG

Maxon EC-4pole 30 Brushless DC Motor

PROVIDES NOMINAL TORQUE OF 106 MNM

Maxon motor recently developed a brushless DC motor for hand-held surgical tools: the EC-4pole 30. Featuring two pole pairs, this DC motor provides a nominal torque of 106 mNm and an output of 150 W. It has a hermetically sealed rotor, meaning that it can withstand over 1,000 autoclave cycles.

The EC-4pole 30 is equipped with ironless Maxon winding, which makes it more efficient. Another feature is that the torque and current behave linearly and the drive can be overloaded. It is available with an optional Hall sensor, as well as with a hollow shaft with a diameter of up to 4.1 mm.

For more information:
Phone: (508) 677-0520
www.maxonmotorusa.com

Miki Pulley BXR-LE Brakes

SPECIFICALLY DESIGNED FOR USE ON ROBOTIC ARMS

The recently introduced Miki Pulley BXR-LE brakes were specifically designed for use on robotic arms to reduce the cantilevered load. With accompanying voltage controller, power consumption is stepped down to 7VDC after a split second of 24VDC for brake actuation. When compared to the other BX brakes in the Miki lineup, this BXR-LE design provides just one-third power consumption and heat generation in one-half the overall size thickness. Specifications are: maximum RPM of 6,000; static friction torque range of 0.32 Nm – 1.32 Nm (0.236 ft-lbs - 0.973 ft-lbs); and ambient operating temperature of –20°C – –60°C (–4°F – –140°F).

Applications for the BXR-LE brakes include: robotic arm joints to stop movement during a catastrophic power failure; Z-axis ball screw brake on CNC machine centers; mounting on a servo motor face, the slim brake profile saves space.

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

Exsys

TO PRESENT GEARBOX LINE AT CMTS 2015

Alongside their collection of lathe tools, Exsys Canada will be presenting the line of gearboxes they offer from Eppinger at CMTS 2015 at booth 2022.

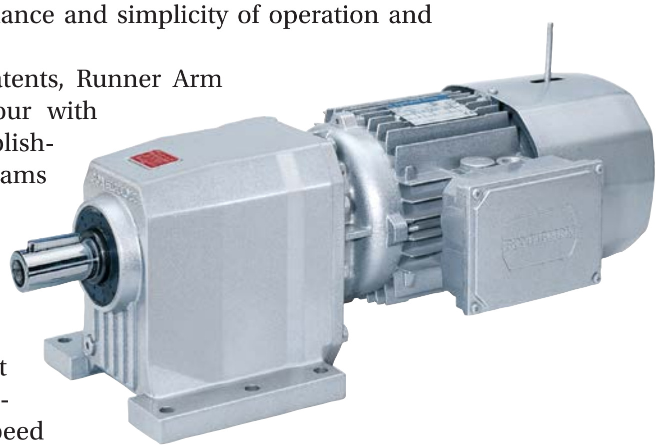
Suited for a range of applications, including machinery, automation and robotics, Eppinger's compact, high-transmission gearboxes are designed to meet demands for stiffness, efficiency and performance. These precision gearboxes come in a variety of types, including bevel, hypoid, planetary and cycloidal designs, to bring smooth, reliable operation to a variety of industry segments.

For more information:
Phone: (905) 792-8001
www.exsys-tool.com

Bonfiglioli Motor

HELPS TECHNOWRAPP CREATE NEW ARM PALLET WRAPPER

Through their close collaboration with Bonfiglioli, Technowrapp recently created Runner Arm, an arm pallet wrapper meant to combine high performance and simplicity of operation and maintenance.



Due to two specific patents, Runner Arm can wrap 136 pallets/hour with 10 turns per pallet, establishing a pallet with just 76 grams of stretch wrap film (value achieved with 15 µm film). The rotary arm with centripetal contrast ring contains and guides the arm as it moves and makes it possible to reach a rotation speed of 45 RPM, combining this with low structural stress. The cutting and welding pliers operate on the moving load, contributing to the increase in speed and efficiency of the work cycle.

For more information:
Phone: (859) 334-3333
www.bonfiglioliusa.com/en-us

Stafford Stainless Steel Shaft Collars and Mounting Devices

INCLUDE 18-8 OR 316 SST FASTENER FOR USE IN HARSH APPLICATIONS

Stafford Manufacturing Corp. recently introduced a full line of stainless steel shaft collars, couplings, and flange mounts designed for environments exposed to harsh chemicals and corrosives.

Stafford's stainless steel shaft collars and mounting devices are manufactured from 303 or 316 SST and include 18-8 or 316 SST fasteners for use in harsh environments where there is frequent exposure to chemicals and corrosives. Designed for building and maintaining equipment used in cleanrooms and laboratories, the collection includes shaft collars for use as stops, spacers, and mounting devices, flange mounts, and shaft couplings for drive systems.

Available in a wide range of sizes from one-quarter inch I.D. up, Stafford stainless steel shaft collars are available in one- and two-piece and hinged clamp styles with smooth bores. A Grip & Go handle can convert a standard shaft collar into an adjustable locating device. Accu-Flange collars are offered for component mounting, and rigid couplings come in one-, two- and three-piece styles with straight- or stepped bores.

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

Keyence Robot

INSTALLED ON FRESENIUS MEDICAL CARE'S DIALYSER PRODUCTION LINE

Fresenius Medical Care SMAD recently equipped its dialyser production line at its plant at L'Arbresle, France with a robot.

"In 2009, we decided to automate our dialyser handling unit, which is located between a conveyor system and a processing unit," said Mouloud Ifri of Fresenius Medical Care SMAD's manufacturing department. "Automating the unit was an obvious solution. The main challenge was that the dialysers are stacked unevenly and their positions vary along three dimensions. We had to find a system that could recognize their positions."

Fresenius automated its line for two reasons: it is installed in a controlled environment — problematic for human intervention — and it operated at high speed 24 hours a day, seven days a week.

The feed system consists of two conveyors, each with two rows of dialysers. The robot is equipped with a tool that allows it to pick up two dialysers per cycle. The tool is fitted with two Keyence LJ-G080 laser sensors, one per dialyser, so that the dialysers are correctly picked up one after the other.

The robot thus simultaneously removes dialysers from each stack on a conveyor. It takes dialysers from the two conveyors in turn. When a row (diagonal) is emptied, the conveyor moves forward until the stack reaches the unloading position.

The robot routinely 'scans' the theoretical positions depending on the type of product. It uses the Keyence sensors to detect products and correct its pick-up position accordingly. If no products are detected, the robot moves to a different stack. It also uses the sensors to correct its X- and Z-axes. To compensate for dispersion along the robot's Y-axis, the robot's tool is fitted with a homing cylinder for each type of dialyser. The maximum height and slope of the stacks are always the same for the same type of product. However, these stacks may be uneven.

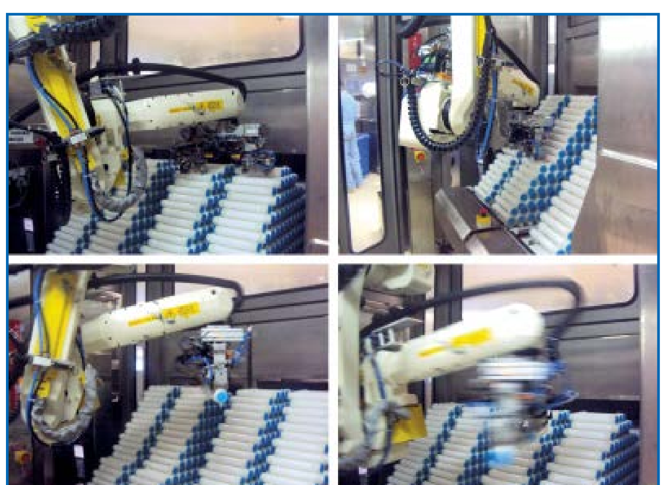
Ultimately, Fresenius Medical Care SMAD chose the LJ-G080 sensor by Keyence. Its user-friendly setting menu is meant to allow experienced and novice users to configure settings easily and quickly. The setting support software (LJ-H1W) supplied with the device is designed to make it easy to save and analyze data with a PC. Configuration is also made easy by several adjustment functions.

The position adjustment function is designed to provide stable measurements even when targets are not perfectly arranged or positioned. The tilt correction function simplifies installation of the sensor head and eliminates measurement errors. A number of functions make measurements to the surface properties of targets.

"We also place very high demands on the robustness and reliability of equipment," said Ifri.

The LJ-G sensor offers repeatability of 1 micron along the Z-axis and 10 microns along the X-axis. It is protected inside an IP67 housing and withstands vibrations (tested from 10 to 55 Hz, with an amplitude of 1.5 mm, for two hours along X, Y and Z). It weighs 350 g and has a detection range of 80 mm.

For more information:
Phone: (888) 539-3623
www.keyence.com



Haydon Kerk WGS Integrated Screw/Slide System

DESIGNED FOR STABILITY AND SPEED

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

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

Standard leads include 0.100-in, 0.200-in, 0.500-in and 1.00-in (2.54, 5.08, 12.7 and 25.4 mm) travel per revolution. There are short leads for non-backdriving vertical applications as well as longer leads capable of speeds of more than 60 inches per second (1.5 meters per second).

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

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

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

The Cult of the Maintenance Evangelist

Joel Leonard is out to solve the maintenance crisis one sermon at a time

Erik Schmidt, Assistant Editor

A man stands at a proverbial pulpit, dressed like a company executive but speaking like a fire and brimstone southern preacher.

Words shoot out of his mouth in rapid succession, powered by a heartfelt purpose and doused with an indeterminate southern twang — it's North Carolinian by way of California with a little Chicago edge thrown in for good measure. The syllables sway, bouncing up and down in a hypnotic, rhythmic cadence, daring you not to listen.

"Hello, I'm Joel Leonard," he says, "They call me the maintenance evangelist."

A hearty smile spreads across Leonard's face as he begins to preach, a sign of good fortune that belies the truth. Sure enough, as the sermon takes shape, the smile slowly fades into a solemn glower. Leonard gesticulates powerfully with his left hand and pounds out a series of bone crunching questions:

"What are we gonna do when the baby boomers retire? What are we gonna do when 35% of our skilled workers are no longer available? What are we gonna do when we have all these activities we need to do to perform maintenance but we don't have the resources or the capacity to deliver? What are we gonna do when we realize our machines and our equipment are beyond repair because we deferred it for so long?"

We face a maintenance crisis, he says. And that's the unfortunate Gospel truth.

But then his eyes soften at the creases, his fist becomes an open hand, and the questions give way to real, tangible answers.

Joel Leonard (left) and his nephew Baker Leonard pose with his "Game Changer Award" during a Greensboro Grasshoppers game.

"I really believe that if the United States became the reliability nation, and we build a surplus of skilled maintenance technicians and we have a ready resource 365 days a year to help us address our chronic problem, we will be able to not just compete in the global economy, we will actually thrive. "

Welcome to the Revival.

The Game Changer

Over the past 25 years, Leonard's business card has been in a constant state of upheaval.

Since graduating from Elon University in 1987, he's held dozens of different job titles. At various points he's been an editor, an author, an advisor, a consultant, a supporter and a community developer.

Of course, only one name has truly stayed with him after all this time.

"I was doing these workshops on how to turn maintenance from a cost center into a profit center," Leonard says, "and there was a professor from the University of Tennessee who attended this conference. He said that I was the first speaker he ever heard talk about maintenance that didn't put him to sleep.

"He said I talked with such energy and passion that I was the maintenance evangelist. I've been called other things, but that one stuck."

It's certainly a strange name at first glance, like two random words were pulled out of a hat and thrown together in sequential order. But despite the oddity of the title, there is no doubt that Leonard absolutely *is* one.

A broad shouldered, barrel chested southern transplant with faith in his heart and the fear of God in his voice, it would appear that Leonard was fated for this kind of work. But actually, he happened upon it by accident.

"I kind of fell my way into it," he says. "I started off going to orthodontistry school at the University of North Carolina. I got sidetracked and wound up transferring to Elon to get a degree in business and marketing. In order to pay my way there I ended up getting a job at a furniture factory at night. I worked on the machines and then they hired me as an industrial technician to work in the engineering department.

"They made me do every job in the factory. So I went from working with the wood to the saws to the shaping department to the assembly department to the finishing department, where they put on the lacquer, to the shipping department. I did that for about two years.

"While I was there, they assigned me to kind of tag along with the most interesting guy and the most hated guy there, whose name was on the loud speaker every ten minutes — the maintenance guy. The maintenance guy was responsible for keeping the water, the electricity, the power, all the machines, the whole facility running.

"It was very interesting to see all the things he did and I developed a great respect for him, although I didn't see anyone have respect for him because anytime a machine wasn't running and he wanted to take it down to fix it everyone would be chewing him out because they didn't want to lose their production counts.

"But I got a big dose of appreciation and respect for maintenance there."

In the fall of 1990, Leonard started work as a business developer at DPSI. He developed marketing and sales strategies to implement computerized maintenance management systems for major corporations, including Procter and Gamble, Burlington Industries and Coca-Cola.

For the next decade Leonard bounced around the maintenance industry, and during that time he began speaking at events for the Association of Facilities Engineering (AFE) — a decision that would ultimately push him towards developing the maintenance evangelist persona he still maintains today.

And it was at one of these speaking events that Leonard finally got his big break in the world of maintenance.

It was because of — and this is no metaphor or verbal trickery — "American Idol" and a bunch of crazy kids.

Seriously.

"In 2002, I attended a conference in Nashville, TN," Leonard says. "At this conference, the editor of *Maintenance Technology Magazine* got up before the crowd and asked, 'How many of you are going to retire in the next 10 years?' Over 90% of the audience raised their hand. These were people from Coors and Coca-Cola and other major corporations from around the country. These were the best of the best.

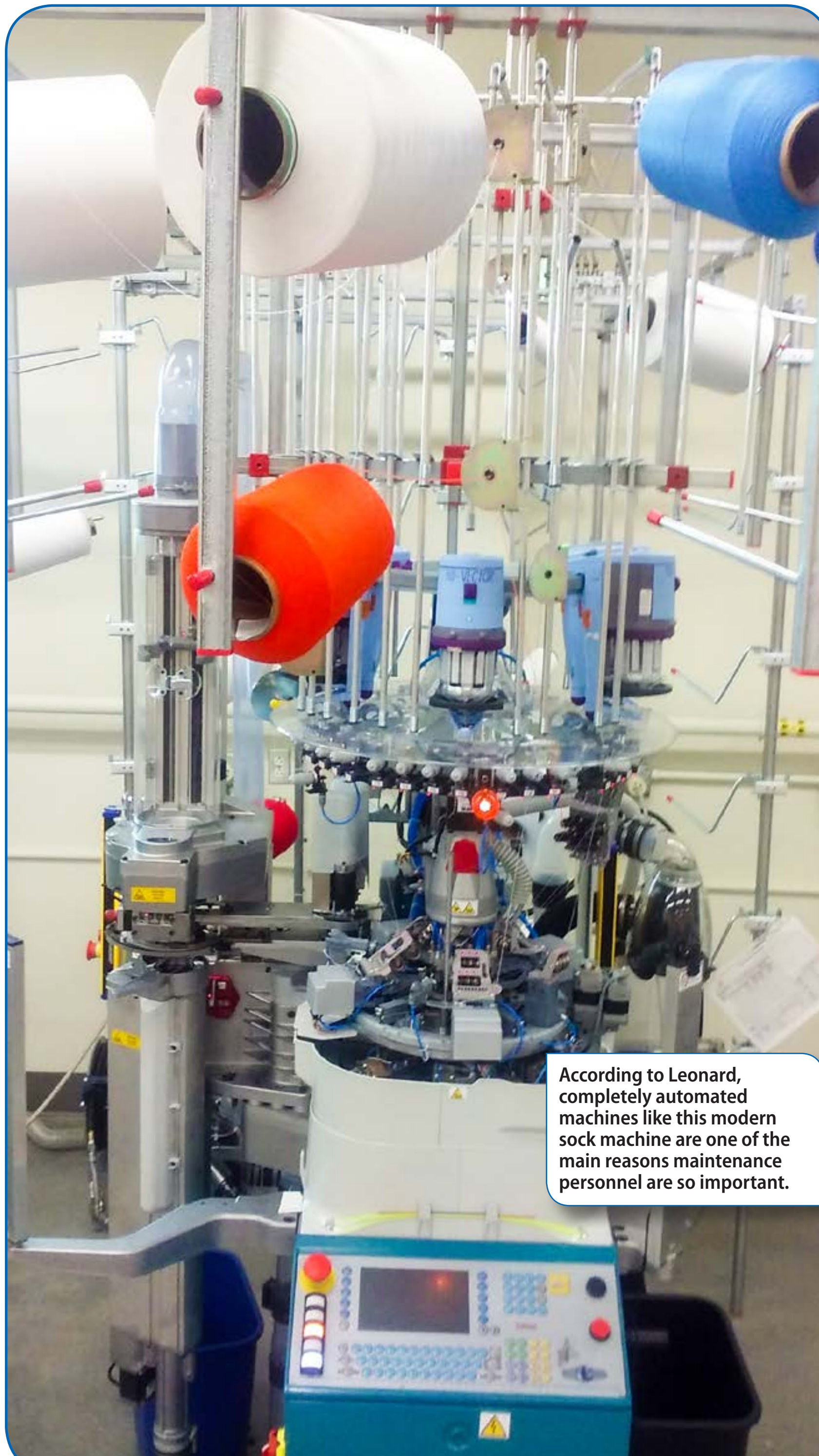
"It kind of woke me up. To see roughly 500 people of 600 raise their hands really hit me hard. That literally changed my life. From then on, I adopted the problem of building the next generation of skilled technicians as my life's chore.

"That was a big, momentous occasion, because that evening after I heard that I went outside to stretch my legs even though it was 30 degrees out, and standing outside were 5,000 kids trying to get on 'American Idol.' I realized that there were no kids [pursuing technical careers], but they were willing to sit in the cold air and try to sing. That evening a buddy and I were sitting around a table in Nashville and I said, 'Wait a minute, getting engineers to tell more engineers that we need more engineers is not going to generate the response that we need.'

"I said, 'What we really need to do is write a song. Then [my friend] said the magic words that really got me going: He said I couldn't do it."

Well, it turns out that Leonard *could* do it. He called up some musicians and 10,000 YouTube hits and nine different versions later, the maintenance evangelist had his own personal theme music.

Wherever Leonard goes, the aptly named — if a bit on-the-nose — "Maintenance Crisis Song" booms behind him, succinctly summarizing his mission statement with a bluegrass, toe-



According to Leonard, completely automated machines like this modern sock machine are one of the main reasons maintenance personnel are so important.

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maintenance
at www.powertransmission.com

tapping chorus.

No one wants to work in the boiler rooms
No one wants to work with the tools
Nation's youth are takin' the easy way out
There's no one left to fix our schools
Maintenance technicians are about to retire
Company executives have got no one to hire
How safe does it make you feel?

"That song really helped me propel my message forward," Leonard says. "If I wrote a book, nobody would read it. I've written thousands of articles — literally — and that gets some momentum. But the song has just been amazing."

The "Maintenance Crisis Song" has been played before Congress, at the Rock and Roll Hall of Fame, and in dozens of countries around the world, including Helsinki and Milan. It is, in a way, Leonard's universal communication device in his hopes of solving a worldwide problem.

"The problems of maintenance are universal," Leonard says. "We may have different cultures and different languages but the same challenges face everybody. Regardless of where we're at, we're facing the same issues worldwide."

In the wake of the success of the "Maintenance Crisis Song", Leonard wrote other tracks, including "Find Me a Maintenance Woman", which features this memorable line:

You can have your Britney Spears
Find me a woman who can work with gears.

It's all part of Leonard's grand plan to make maintenance more accessible and "sexy" to the nation's youth.

"I've been trying my best to break down stereotypes and stigmas and get businesses to hire based on performance," Leonard says.

Currently, Leonard works for The Forge in Greensboro, NC, where he is attempting to generate a higher level of skills and talent in the area. For his efforts, he was awarded "The Game Changer Award" by the city. Leonard received the award in front of a sellout crowd during a Greensboro Grasshoppers game (Class A affiliate of the Miami Marlins).

"It's kind of funny, I've gone my whole career helping maintenance guys and technicians and to show their appreciation for it they gave me a baseball bat," Leonard says. "People still know me as the maintenance evangelist, but I like this new title 'game changer.' That's what the people [in Greensboro] know me as.

"They're using it to get me more resources by saying 'He's a certified game changer, he got named that in front of a whole baseball stadium.'"

The Gateway Drug

Leonard is pounding his pulpit in Greensboro.

He's trying his damnedest to get the folks of central North Carolina (and beyond) to see his vision — that maintenance isn't just a cost sink, but in actuality, a way to make a lot of money.

"I like the idea that people are finally realizing that maintenance is actually a profit contributor," Leonard says. "If any area built a surplus of skilled technicians they would have a huge economic advantage. No area is trying to do that, and that's what I'm trying to get [Greensboro] to adopt."

According to Leonard, one of the key ways to accomplish this is by getting college kids hooked on drugs.

No, it's not what you think.

"3-D printing is the gateway drug to manufacturing," Leonard says. "When the kids learn how to use a 3-D printer they also learn quickly because they have to take care of it. If they don't, it'll break and fall apart and they have to do maintenance.

"The kids love 3-D printing and that's a great hook to get them interested in working on other machines and the whole world of manufacturing."

One of the other main issues of maintenance today is trying to get companies to see the big picture.

"We still have to get companies to look long term," Leonard says. "So many companies are focused on short term outputs that they don't look at maintenance as an investment. As a result they shortchange their entire business capabilities. We have to continue to upsell maintenance."

For example, when Leonard was in Dubai he encountered a man who had a $14 million paint budget for all the drill systems in the Persian Gulf that his company was trying to cut out. Leonard informed him that if they went through with it, all the pipes would rust within three years opposed to 15 years.

That's where Leonard has provided his most value to the maintenance community — not with his passionate speeches or catchy tunes — though that surely has gone a long way in establishing his footprint — but with his ability to see beyond the dollar signs that can cloud people's judgement.

"Seeing what [Leonard has done at The Forge] in just a year and a half is amazing," said Dan St. Louis, director of the manufacturing solutions center in Conover. "I never would have dreamed when he showed me the initial plans that all this stuff would be happening. It's a tough area of town and now there are all kinds of things going on. It's not smoke and mirrors.

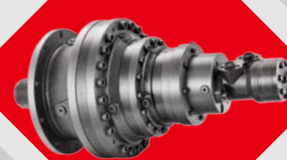
"He understands what manufacturers need. He's down in the trenches and a lot of times folks look at it from a high level and say, 'Oh, we just need more engineers.' Joel understands that we need technical people. He understands this from the ground floor."

Leonard, despite his now inseparable moniker, said he's a religious person who prefers not to talk about it out in the open. When pressed, he said his religion was simply "to make things better."

Well, he's toured the world, his song sweetening the air around him, and he's slowly built a devoted following — a cult of people who believe that his ideologies on maintenance are dogmatic truths. For 25 years he's done this, and any person of faith would have to believe that the maintenance evangelist's mission isn't over just yet.

"If more could join this cult," Leonard says, "our machines would be more reliable, businesses would become sustainable, our economy would be stronger and our technical skills gap would be solved." **PTE**

THOTH



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Pushing the Envelope(s)

Foresight and early warnings keep Encore Envelopes in the game

With advances in modern technology reducing demand for envelopes, manufacturers are finding it increasingly hard to justify investment. Matters became more complicated for Encore Envelopes when they were notified that their production line was at risk due to spares becoming obsolete. It was imperative that a strategy be put in place that would minimize machinery downtime and potential lost earnings.

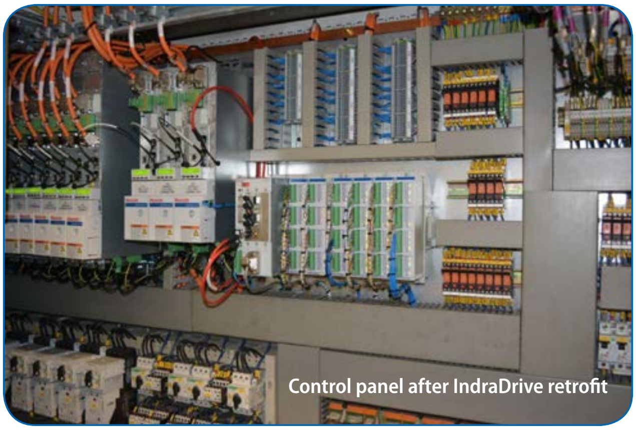
Encore Envelopes is the largest independent manufacturer of printed envelopes in the UK, operating 15 production lines printing up to 85 million envelopes per week. With 12 months remaining on the ten year service support guarantee of its Diax03 drives, Bosch Rexroth contacted the company to advise that with spares becoming obsolete, the maintenance and service of its drives could not be fully supported and production would be at risk.

"We are fully aware that investment in maintenance repair and overhaul (MRO) services for any production line can become very costly if unexpected," says Andrew Smith, service consultant at Bosch Rexroth. "This is why we make it a priority to notify manufacturers as soon as a risk is identified, so that we can help them upgrade machinery over time and avoid a sharp and unexpected impact on production."

First fully refurbished and retrofitted envelope printing machine

To minimize machinery downtime, a strategy was put in place to allow Encore Envelopes to retrofit a significant number of their envelope production lines with new drive technology. Encore chose the Rexroth IndraDrive for its versatility, compactness and multi-protocol support offering.

Commenting on the process, David English, electrical engineering manager at Encore Envelopes, said: "Taking Andrew's advice on board, I took one of the units out of production and, upon receipt of the IndraDrive system, upgraded the machine. Not only did this give me a machine with better operating capabilities, it also freed up spare parts for the remaining units, boosting our stock levels which were at a critical level.



Control panel after IndraDrive retrofit

"In addition, it was vitally important that the motion controller of the IndraDrive system be fully integrated with our current operational interface. Following consultation with Bosch Rexroth, we decided that the best course of action would be to install a Programmable Process Controller (PPC) as it utilized the same operating software as the Diax03. This ensured communications and protocols were easier and quicker to implement whilst keeping the time the machine was out of production down to a minimum."

David concludes: "Through the help of Bosch Rexroth, we knew 12 months ahead of anyone else in the industry that our envelope machines needed to be upgraded, which allowed us the necessary time to form a strategy to refurbish our equipment whilst maintaining full operational levels. Due to the success of the initial refurbishment, we have now finished the upgrade of our second machine and plans are in place to refurbish the remaining fleet over the coming months." **PTE**

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What makes Arrow's stock gear product line so unique? It's the level of precision. All Arrow stock spiral bevel gears are produced the same way we manufacture gears to meet the demands of the aerospace industry – with precision machining, precise inspection and high quality assurance.

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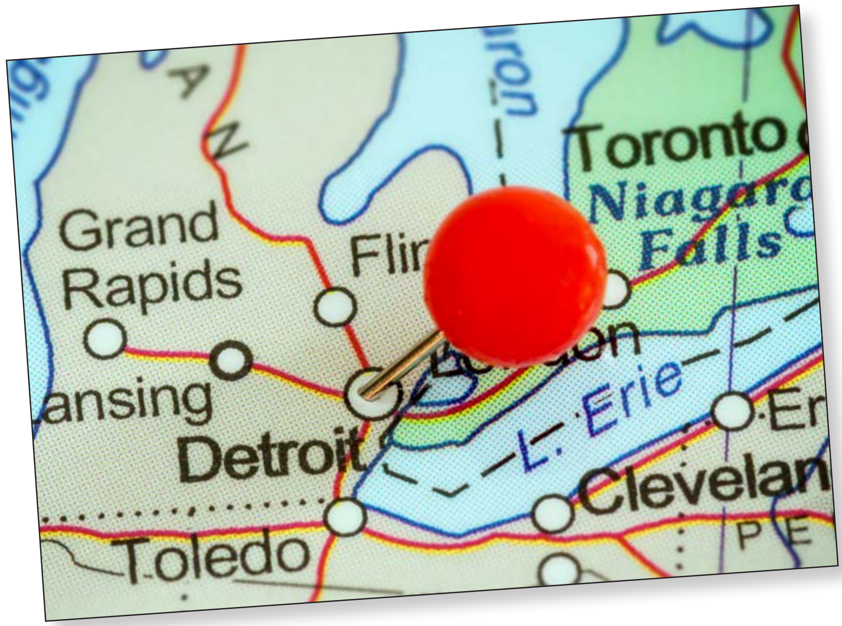
Gear Expo 2015: Gears Galore!

You'll see more gear exhibitors in the Motor City than ever before

Jack McGuinn, Senior Editor



You may well have already read the following Gear Expo description on the AGMA Website. But we figure it took a lot of years, people, and hard work to make these words reality—so they bear repeating. And besides, they describe the event perfectly:

"Gear Expo is the only trade show dedicated to the complete gear manufacturing process, and is one of the world's most affordable machinery shows for exhibitors.

"Gear Expo is a biennial event and the world's only conference and expo designed exclusively for the gear industry.

"For three days gear buyers and manufacturers network and build relationships that benefit their respective companies. Attendees see first-hand the latest technology on the market and discuss trends in the industry with experts. Exhibitors have the opportunity to meet face-to-face with attendees and other exhibitors and will display more than 750,000 pounds of machinery on the show floor.

"Thousands of professionals from around the United States, international manufacturing hubs, and emerging markets conduct profitable business transactions and collaborate on the innovations that make their operations more streamlined.

"Once again the ASM Heat Treating Society Conference & Exposition is co-locating with Gear Expo 2015. That means more access to more exhibitors and attendees, and emerging technologies and trends impacting the gear industry."

Enough said. Now, it's on with the show.

While things are improving, it is exceedingly premature to say the Motor City is back.

But Gear Expo is—and that's a fact—back in Detroit.

Yes indeed, the American Gear Manufacturers Association's premier event returns to Motown for the first time since 2007. And Gear Expo 2015 (Oct. 20–22, Cobo Hall) and Fall Technical Meeting (Oct. 20–22) promise to not disappoint; consider that in 2013 Gear Expo was recognized by Trade Show Executive magazine as one the 50 fastest-growing shows. This year's event is sub-headed "The Drive Technology Show"; but really—"Gear Expo" nails it—there are more gearmakers (60+) exhibiting this year; that is more than at any previous show (***gear-buyers' spoiler alert**: **but we don't see that affecting pricing***).

We couldn't talk to all of them, so we zeroed in on a few exhibitors—some long-time, some not so long. We touched on a number of issues. Depending on busy pre-show schedules, some had more to say than others, so we've attempted to include that which is of most interest to readers.

Excel Gear (Booth 1619)

N.K. (Chinn) Chinnusamy, President

Can you tell us more about your new bevel gear software design package?

Was it a goal to have it ready for roll-out at Gear Expo?

"Most software packages available in the market do not include bevel gears. There are significant differences in how bevel gears are manufactured. The Gleason, Klingelnberg, and Oerlikon systems are for spiral bevel gears. Our software is for the Gleason system only. Mating bevel gears must be manufactured to the same system, i.e.—a Gleason pinion cannot be matched to a Klingelnberg gear, and vice versa. Bevel gears are manufactured in sets

(i.e. — pinion is matched to gear) and cannot be interchanged from one set to another unless the gears are manufactured to a master. A recent development in machines is that they can cut universal bevel gears, eliminating the matched set requirement; but most bevel gear job shops do not have that capability. Still, one system cannot be matched to the other like spur or helical gears.

"Bevel gear capacity calculations are very complex and there are no standards, except AGMA; but even AGMA standards are not fully understood or followed by most designers. Yes, it was our intension to roll out the bevel gear software at the Gear Expo. Our software will help designers to design, analyze, and create one-page-gear dimension data for bevel gears like spur or helical gears."

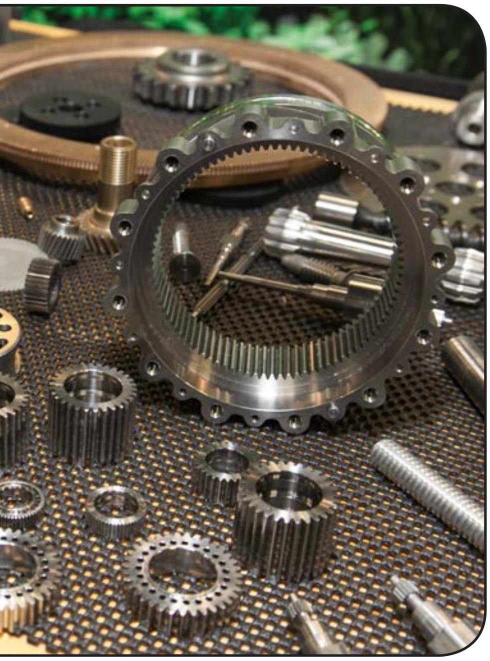
Rave Gears and Machining LLC (Booth 1339)

"Rave will be introducing its Gen3 line of spiral bevel gears which also has the option of adding our Ravecoat. The coat that covers the teeth improves efficiency and durability of the gear," says Diana Martinez, contract administrator.

KISSsoft AG (Booth 1830)

"We will be highlighting the newest release of KISSsoft, Version 03-2015," says Stefan Beermann, KISSsoft CEO and partner. "Looking back, Gear Expo was always a chance to show new features to our customers in a hands-on way, discussing pros and cons while demonstrating the software directly in the booth. This is typically not a one-way road; we also learn a lot from the feedback or our customers. Since Dr. Kissling, the 'brain' behind the software is present, this is a perfect opportunity.

"The new release of KISSsoft has several new features that will be demonstrated," says Beermann. "Why? Our customers have made request for added functionality and KISSsoft has responded. New features are added standards, like the AGMA 6001/6101 shaft calculation, or some Russian GOST standards for spur and helical gears. The contact analysis for planetary gearsets now determines the load distribution on the gear flanks taking the results of finite element calculations of the planet carrier into account; for this we included an open-source FE



core into the installation. And we can now read in a full stiffness matrix form various FE programs to model the resilience of the housing of a gearbox. In addition, the determination of critical speeds on a system level is now available. And, of course, many other more specific improvements."

Gleason Corp. (Booth 1017)

Gleason is a leading source of education for gear technology and it will be promoting its customer training classes that offer a full range of courses and education forums, ranging from the beginner to the most advanced user covering all aspects of gear design, production and inspection.

Offering plastic gear design and injection molded plastic gears, including helical gears, spur gears, planetary gears and internal gears, Gleason plastic gears provide customers with the benefit of a plastic gear with no weld-line for a stronger, more accurate and economical drivetrain — eliminating the additional expense of secondary machining. Gleason will display some of its most recent innovations and have experts available to answer any plastic gear-related question.

EXSYS Tool, Inc. (Booth 1731)

Florida-based EXSYS Tool, Inc. will showcase its new line of Eppinger gearboxes and custom gear-making services for a wide variety of industries. EXSYS, known as the North American supplier of high-precision, German-made Eppinger live and fixed tool-holders, as well as modular adapter systems — both for CNC turning centers — just recently expanded its line of productivity–enhancing systems into the gear sector. Therefore Gear Expo attendees can now enjoy some personal time with the latest Eppinger spiral bevel-, planetary-, planetary-bevel-, hypoid- and cycloid-type gearboxes. They can also learn how to acquire specialty, custom-made gears produced to their specifications.

EXSYS Gearbox Types

Spiral bevel. BT (bevel torque) and BM (bevel maximum torque) compact spiral bevel gears bring high torque and maximum efficiency to gear applications that require a high degree of reliability and variability. Each of these bevel gearbox types offers minimized tooth clearance and optimal transmission properties via precision axes and bearing seats, combined with Gleason bevel gears that can withstand high loads. The single-component steel housings for these bevel gearboxes feature mounting threads on all sides to ensure stable attachment in a variety of installation positions. The heavy-duty bevel gears inside these housings offer high transmission precision and reduced stress on the bearings. A friction-locked, zero backlash connection of the crown gears on the drive shaft reduces the mass of the gearing component. Both BT and BM gearboxes come in solid or hollow shafts, in standard and custom designs. BT-type models are available in seven sizes with a transmission ratio of $i = 1:1$ to 5:1, while BM-type models are available in five sizes with a ratio of $i = 1:1$.

Planetary. PE (planetary eco) and PP (planetary precision) planetary gearboxes are for applications that require low backlash, high efficiency, shock resistance and a high-torque-to-weight ratio. With a modular design that combines ground gears and precision gear components, these gearboxes ensure performance efficiency and maximum uptime. They are also energy-efficient and easily mount to a variety of motors. The present range of planetary gearboxes comprises five sizes, with each size offered as

a single- dual-, or triple-stage design. Each gearbox variant is also available as a precision design with reduced backlash. The wide range of sizes and designs allows users to achieve overall transmission ratios from $i = 3:1$ to $i = 512:1$ in a variety of applications.

Bevel planetary. BP (bevel planetary) gearboxes combine features of the company's BT (bevel torque) series bevel gearboxes with the pre-stages of its PE (planetary eco) planetary gearboxes, creating an innovative solution for various applications. The stable housing design and hardened, super-finished gear components of these gearboxes help ensure smooth running and constant backlash control. These planetary bevel gearboxes are efficient and achieve high-output torque and extremely high transmission ratios up to $i = 320$. Currently offered in eight sizes, the planetary bevel gearboxes easily mount to a wide range of motors.

Hypoid. The Eppinger HT-type hypoid gearboxes feature a compact, robust design suitable for both specific and highly dynamic applications. With a specially developed, aluminum mono-bloc housing with high-precision bearing seats and an integrated input shank, each hypoid gearbox provides unmatched stability, accuracy and efficiency. These hypoid gearboxes easily connect to a variety of servo motors. With solid steel alloy and hollow shafts for shrink disc connection, users can install these gearboxes in various positions with a choice of the output side. Currently, the gearboxes are available in four sizes in the ratio range, from $i = 5:1$ to $i = 15:1$.

Cycloidal. To round out its product portfolio, Eppinger develops and manufactures cycloidal gearboxes. These compact, high-transmission gearboxes are an excellent choice for tool machinery, automation and robotics. With integrated support bearings and a high-overload capacity, they excel in applications that require the utmost stiffness, performance and efficiency.

The cycloidal gearboxes are available in six sizes—with ratio ranges from $i = 57:1$ to $i = 175:1$—and in solid and hollow shaft designs. They can also be adapted to meet specific customer requirements.

Custom gear manufacturing. In terms of custom gear-making services, Eppinger is virtually without limits. They can for example develop and manufacture crown gear diameters ranging from 0.4 mm to 330 mm—depending on the transmission ratio. Examples include professional-quality, high-performance bevel gearboxes, as well as ring and pinion gearsets. Gear customers include Mercedes, Bugatti and Airbus.

The company uses modern Gleason milling and grinding centers to machine its gears from a wide variety of workpiece materials. Gleason and Zeiss measuring machines, along with Gleason test equipment, ensure each gear complies with the quality requirements of DIN 3965 and the American Gear Manufacturers Association (AGMA) at all times.

Final Thoughts

We close with some exhibitor observations about the value of the show.

Says Wendy Young, President/COO, Forest City Gear (**Booth 1826**): "We get leads and some are brought to fruition. We do have exposure to an enormous amount of networking, and that is very valuable in itself."

Offers Dan Kondritz, General Manager, KISSsoft USA: "Yes, (Gear Expo) always had (been good) — for KISSsoft, at least. We get a chance to meet face-to-face — something that is hard to put a dollar figure against."

(Rave Gear's) Martinez: "The costs of attending Gear Expo are justified because it's the place to meet individuals from the aerospace to automotive industries that require our services. And the best part of it is everyone is under one roof. This is the perfect opportunity for any company in this industry to get their name out there."

And from Kerry Klein, Arrow Gear (**Booth 1626**) Vice President, sales and marketing, "Certainly (the show is worth the expense)! Arrow Gear has long been a supporter of Gear Expo and will continue to support the industry. We feel that it is important to have a forum where all companies in the industry can display their products and gather together to share ideas and business experiences." **PTE**

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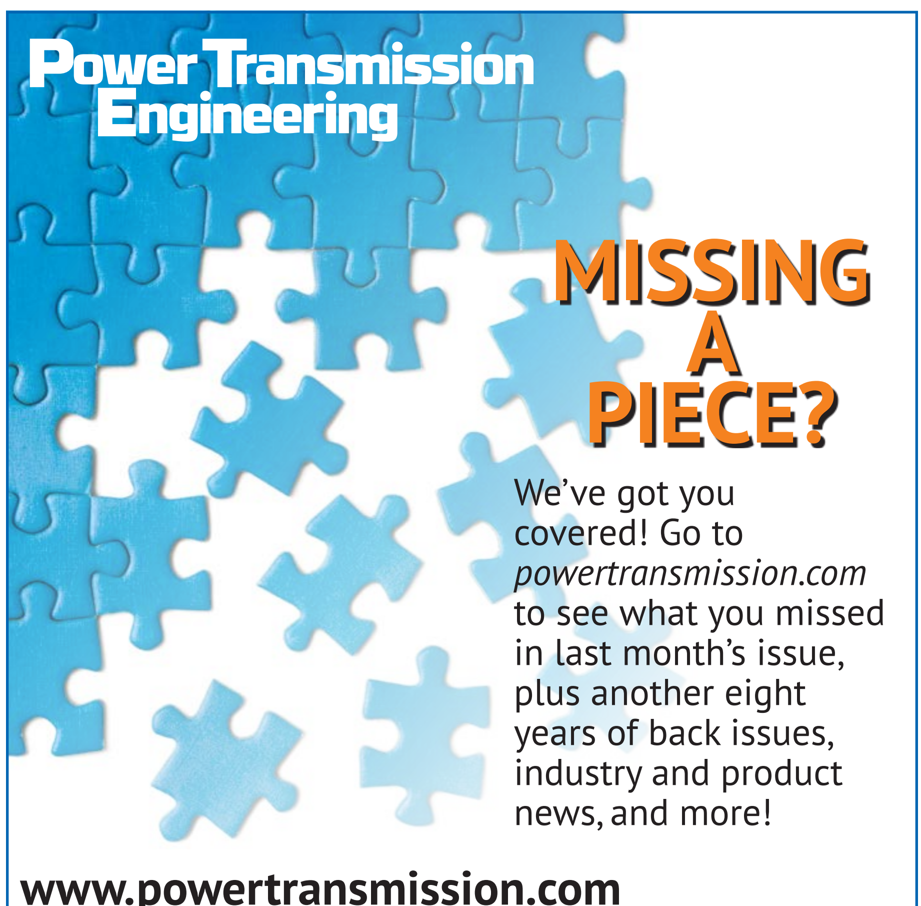
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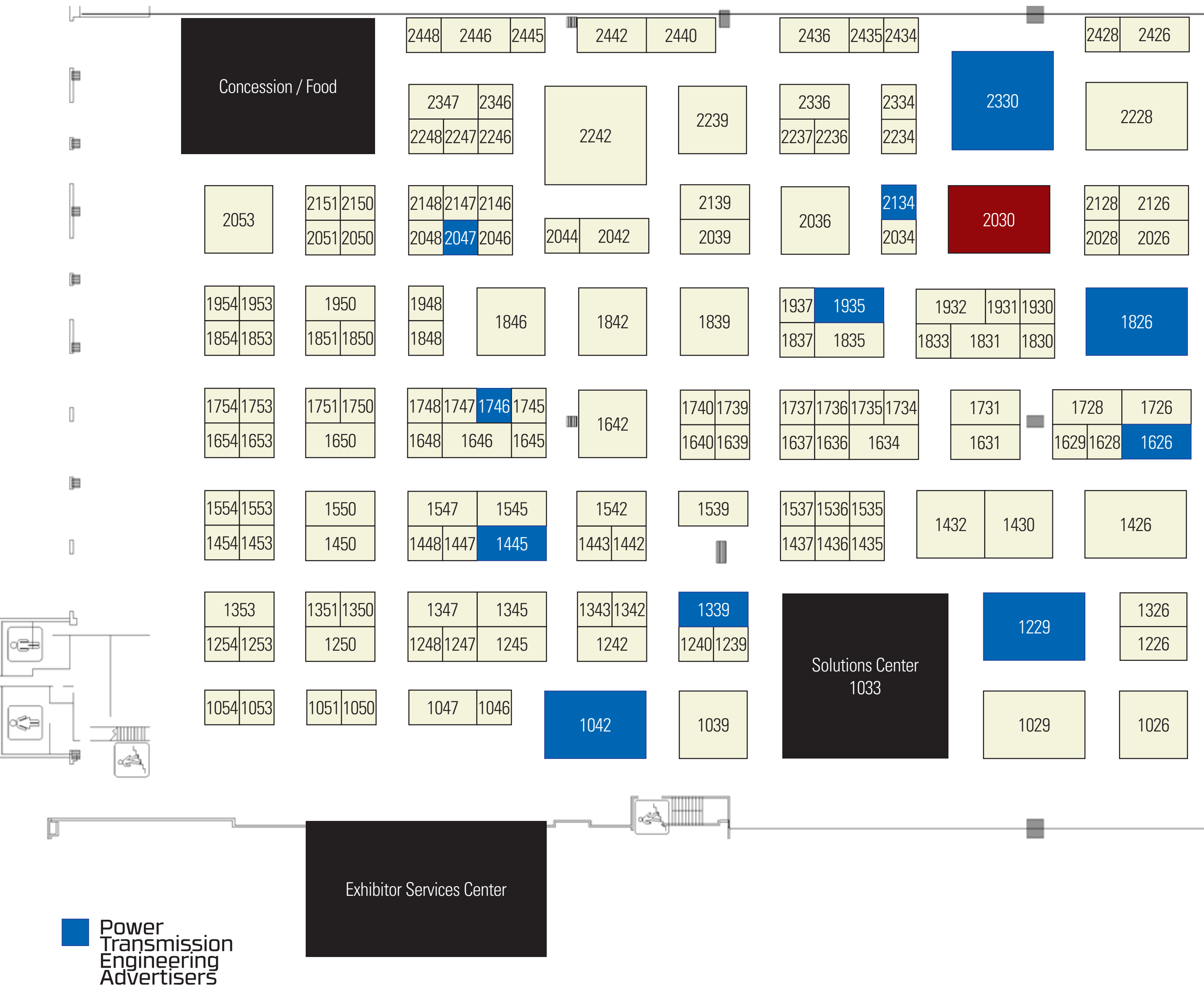
Gear, Gear Drive and Power Transmission Component Suppliers at Gear Expo 2015



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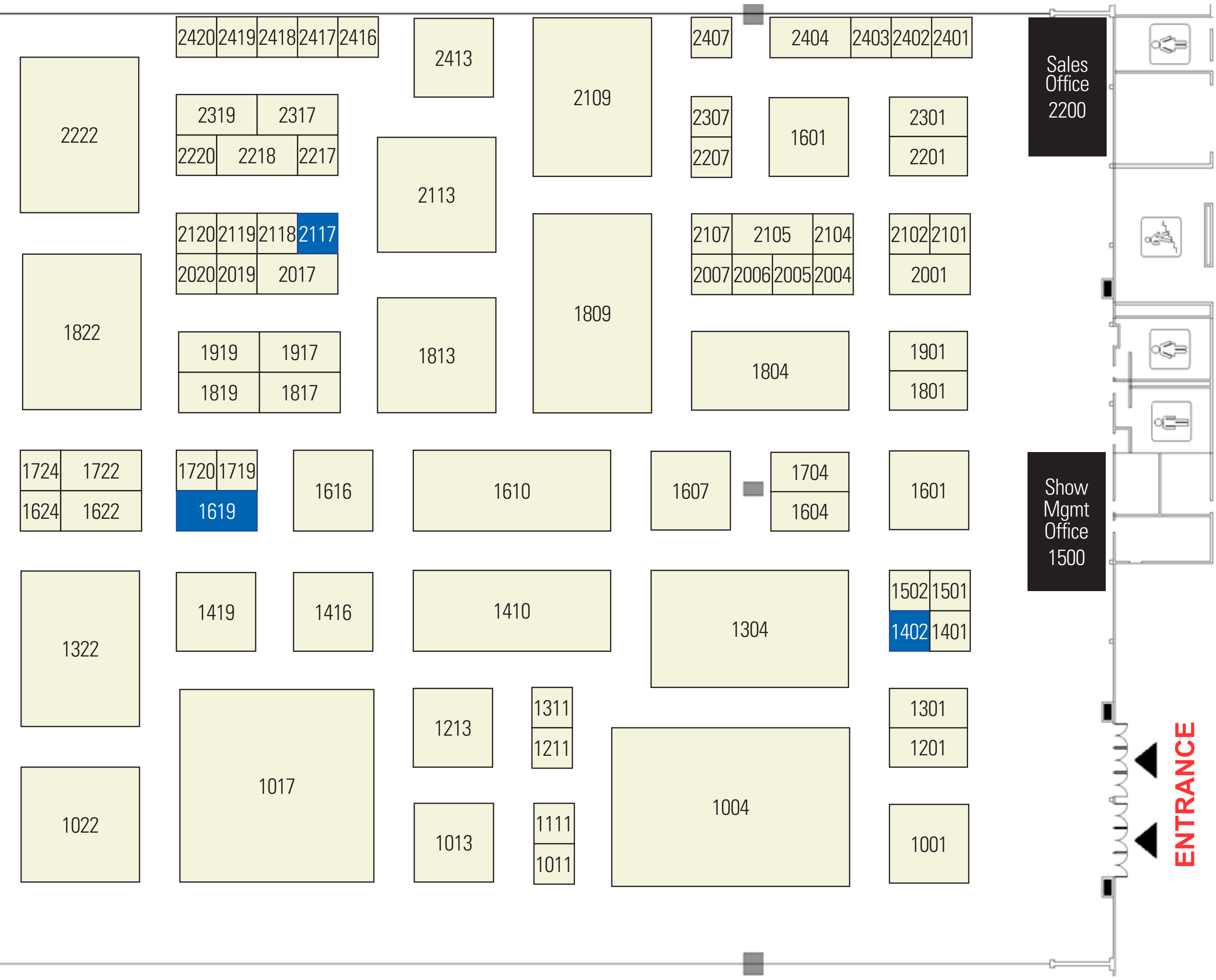

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Riverside Spline & Gear, Inc.	http://splienandgear.com	1254	
Romax Technology Inc.	**www.romaxtech.com**	**1402**	**p29**
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Shanghai Superoll Bearing Co., Ltd	www.superoll-bearing.com	1435	
Shivam AutoTech Ltd.	www.shivamautotech.com	1351	
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Smart Manufacturing Technology Ltd.	www.smartmt.com	2207	
SMC Powder Metallurgy, Inc.	www.smcpowdermetallurgy.com	2247	
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Syn-Tech Ltd.	www.syn-techlube.com	1442	
Taiwan United Gear Co., Ltd.	http://tugear.com.tw	1542	
Tecsia Lubricants USA	www.tacbeconlube.com	2128	
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Mutiple Gearset-Type Calculation Software

THE QUESTION

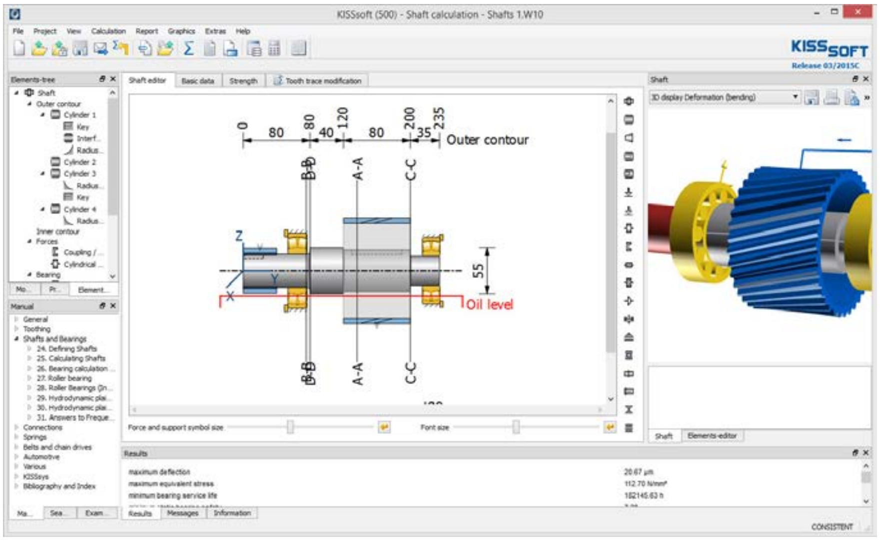
Is there a gear software package out there that will calculate the design of spur, helical, worm, and planetary gearsets? Also, we would like a program that calculates stresses and material selection. Finally, we would like to have the program calculate bearings loads, too. Thank you for your help.

Expert response provided by: Dr. Stefan Beermann, KISSsoft CEO: Yes, there are some software packages available. In the U.S. the important commercial ones are *KISSsoft*, *IGS*, *ROMAX* and *MASTA*; each has its own strengths. I will only write about *KISSsoft* — for obvious reasons. So the first brief answer is: *KISSsoft* can do the tasks above — and more.

Some details: *KISSsoft* is a program that calculates machine elements. This is generally what an engineer learns at University when it comes to assessment of strength or life time of machine components. Most of *KISSsoft* is dedicated to the various gear types and configurations, including the abovementioned ones; calculating stresses is a basic functionality of all the respective software packages. Let me at this point say a few words about the meaning of stress values: if you compare stress values from gear software with those from an FEM package, you will often find significant deviations. Also, among the different software packages available, results will differ. That is because the calculated "stress" in a machine calculation software is compared to a permissible stress value. The calculation of both is defined by the method applied. As long as you stay within the method, this is perfectly fine. But don't mix methods!

Since all the common gear software is implementing more or less the same methods, the main difference is in how the input process is tailored. In *KISSsoft* we focus on making this process as flexible — yet still convenient — as possible. Thus *KISSsoft* has a monolithic approach, i.e. — all tasks are performed using the same user interface. Depending on what the topic is you need to address — say, new design or assessment of given design — you will enter different data. However, trust the software to always keep your gear definition consistent and make proposals for suitable input data. For material selection, *KISSsoft* provides a data base with the relevant data of some hundred steels and other material. This data base can be extended by the user if some special data set is requested. The definition of the macro geometry for a given gearset is the hardest part of the job. If you have a mistake in the input — typically due to a misunderstanding of the meaning of the input, regarding either the drawing or the software — you are subsequently calculating the wrong gears. So be careful and cross-check all available control data; again, trust the software to keep you informed whenever something looks strange.

For the design of a gearset, *KISSsoft* has some special functionalities that make first proposals; refine and optimize a given design; and, finally, optimize the micro geometry, i.e. — the profile and flank modifications. Other functions are for the in-depth analysis, beginning with the contact analysis. This module finds the real contact points on the flanks under load, with flexible teeth, shafts, and bearings. Based on this approach, more realistic stresses and transmission error — key phrase here: noise vibration harshness (NVH) — can be determined.



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software

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What is more, power losses and heat generation, wear, oil film thickness and other negatives can be calculated. The bearing loads are also calculated here; however, I recommend using the *KISSsoft* shaft calculation for this topic, as it is more precise. Other parts of the gear software treat operational backlash, summarize data for the shop floor, and so on.

Let's summarize: with *KISSsoft* you get an easy-to-use tool for calculating spur, helical, worm and planetary gearsets, and also bearing loads with stress calculation and material selection supported by a data base. If you want to try it, contact us. (*www.kisssoft.ch*)

Dr. Stefan Beermann studied mathematics and computer science in Karlsruhe. His Doctoral thesis — "Simulation of Vibrations in Gearboxes Applying Spectral Simulation" — was written in collaboration with the FZG from the TU Munich. In 1996 Beermann joined Zurich-based gearbox company, L. Kissling & Co. AG, as product manager for the calculation software KISSsoft. In 1998 he switched to KISSsoft AG as one of its first employees. Today Beermann is company CEO and partner at KISSsoft AG — together with the company's founder, Dr. U. Kissling.

Setting Techniques for Tapered Roller Bearings

David Novak, Timken Director, Service Engineering

Introduction

Tapered roller bearings can be set at initial machine assembly to any desired axial or radial clearance. This unique feature enables a designer to control bearings to meet anticipated application operating conditions, and thereby provide optimum bearing and system performance.

Some advantages of tapered roller bearings pertaining to setting include:

- Longer bearing life, achieved by optimizing bearing settings while meeting application performance requirements
- Increased mounting stiffness, achieved by properly set tapered roller bearings resulting, for example, in better gear contact and longer gear life
- Easier assembly because cone and cup are separable
- The bearings can be set at the time of machine assembly, allowing wider shaft and housing tolerances

The setting of tapered roller bearings can be readily accomplished by a wide variety of viable methods. These bearings can be set manually, supplied as pre-set assemblies, or set by automated techniques. There are a number of approaches, considerations and advantages for each, with special focus on five popular automated techniques, i.e. — set-right; acro-set; projecta-set; torque-set; and clamp-set (Table 1).

Bearing Setting

With tapered roller bearings, the term "setting" simply indicates the specific amount of end-play (axial clearance) or pre-load (axial interference) within a mounted bearing. The flexibility to easily adjust and optimize setting at the time of assembly is an inherent advantage of tapered roller bearings. Unlike other types of anti-friction bearings, tapered roller bearings do not require tight control of shaft or housing fits to obtain setting. Because tapered roller bearings are mounted in pairs (Fig. 1), their setting is primarily dependent upon the axial location of one bearing row relative to the opposite row.

The three primary conditions of bearing setting are defined as:

1. ***End-play.*** Axial clearance between rollers and races producing a measurable axial shaft movement when a small axial force is applied — first in one direction and then in the other — while oscillating or rotating the shaft (the reference bearing load zone less than 180°).
2. ***Pre-load.*** Axial interference between rollers and races such that there is no discernible axial shaft movement when measured as described above. A rolling resistance to shaft rotation results, which may be measured (load zone greater than 180°).
3. ***Line-to-line.*** A zero setting condition; the transitional point between end-play and pre-load.

A bearing setting obtained during initial assembly and adjustment is the "cold" or "ambient" bearing setting, and is established before the equipment is subjected to service.

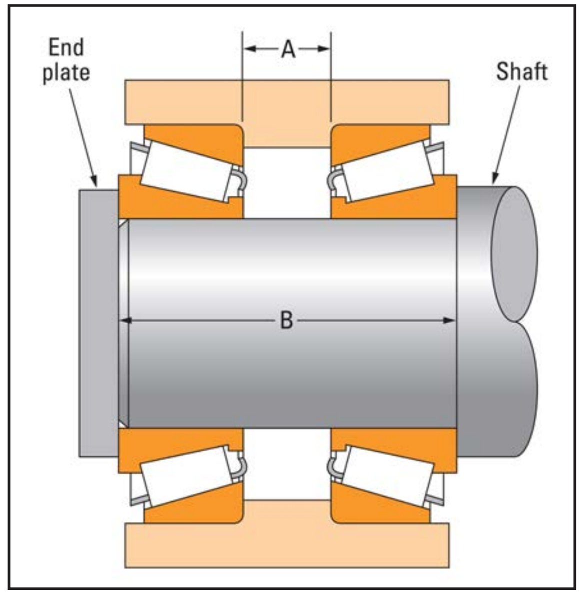

Figure 1 Simplified machine assembly showing a typical tapered roller bearing (indirect) mounting

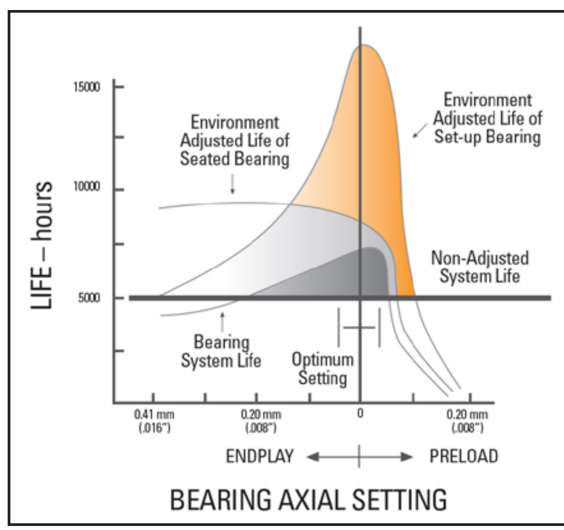

Figure 2 Calculated Bearing L10 Life vs Operating Setting

Bearing setting during operation is known as the "operating bearing setting," and is a result of changes in the ambient environment bearing setting due to thermal expansion and deflections encountered during service. The ambient bearing setting necessary to produce the optimum operating bearing setting varies with the application. Application experience, or testing, generally leads to the determination of optimum settings. Frequently, however, the exact relationship of ambient to operating bearing settings is unknown, and an educated estimate has to be made. To determine a suggested ambient

Table 1 Comparison of Tapered Roller Bearing Setting Methods

Requirements	Setting Method						
	Manual	**Preset assembly**	**Set-Right**	**Acro-Set**	**Projecta-Set**	**Torque-Set**	**Clamp-Set**
Mounted bearing setting range, in. (Typical Min to Max)	0.004 – 0.010	0.006 – 0.012	0.008 – 0.014 (probable)	0.004 – 0.006	0.002 – 0.004	0.005 – 0.007	0.003 – 0.005
Mounted setting region	End play	End play or preload	End play or preload	End play or preload	End play or preload	End play or preload	End play
Loose fitted member for adjustment?	No	No	No	Yes	No	No	Yes
Apply set-up load?	Yes	No	No	Yes (constant)	Yes Gauge spring	Yes	Yes (constant)
Special gauges, fixtures; components?	No	No	No	No	Yes Special gauge & LVDT*	Yes Rolling torque gauge	Yes Compensating ring
Special bearing codes or assemblies?	No	Yes "Matched" assembly	Yes Codes & spacers	No	No	No	No
Pretesting needed to develop method?	No	No	Yes (limited)	Yes	Yes	Yes	Yes
Typical production volume	Low to Moderate	Low to High	Moderate to High	Low to High	Moderate to High	Low to High	Low to High
Assembler skill or training level	High	Low	Low	Low-Med Chart reading	Low LVDT reading	Low-Med Chart/ gauge reading	Low
Shim pack applied?	Yes	No	No	Yes	Yes	Yes	Yes constant size
Shim gap measurement?	Yes	No	No	Yes	No (LVDT)	No (brg torque)	No
Possible use of wider bearing system mounting tolerances?	Yes	No Control fit tolerances	No Need tighter tolerances	Yes	Yes	Yes	Yes
Bearing rotation or oscillation?	Yes	No	No	Yes	Yes	Yes	Yes
Applicable for field service?	Yes	Yes	Yes	Yes (with service manual)	No	Yes (with manual and new brgs)	Yes
Readily applicable to large, heavy units?	No	Yes	Yes	Yes	No	No	No

Note: All bearing setting methods above require proper backing and positive clamping of bearing components.
* Linear Variable Differential Transformer (LVDT)

bearing setting for a specific application, contact your bearing representative.

Generally, the ideal operating bearing setting is near-zero, to maximize bearing life. Most bearings are set with a cold setting of end-play at assembly. This comes as close as possible to the desired near-zero setting when the unit reaches its stabilized operating temperature.

Some applications are set with cold pre-load to increase rigidity and axial positioning of highly stressed parts that would otherwise be dramatically affected by excessive deflection and misalignment.

Excessive operating pre-load must be avoided, as bearing fatigue life can be drastically reduced. Also, excessive operating pre-load can lead to lubrication problems and premature bearing damage due to high heat generation.

Load zone is a physical measure of the raceway-loaded arc and is a direct indication of how many rollers share the applied load. For a single-row tapered roller bearing, maximum life is obtained with a load zone of approximately 225°.

Figure 2 shows the graphical representation of bearing L10 life vs. the operating bearing setting for a typical (overhung) pinion bearing mounting.

The ideal operating setting that will maximize bearing system life is generally near-zero to slight pre-load.

Manual Bearing Setting

Manual methods are frequently used to set bearings on a variety of equipment with low-to-moderate volume production requirements, whereby a less-than-exact, primarily end-play setting range variation is acceptable. No special tooling, gauges, charts or fixtures are typically required, but assembler skill and judgment are necessary. For example, in the case of a conventional truck non-driven wheel with a single adjusting nut design (Fig. 3), manual setting involves tightening the adjusting nut while rotating the wheel until a slight bind is felt. Then the adjusting nut is backed off ⅙-to-¼ turn to the nearest locking hole — or sufficiently to allow the wheel to rotate freely with some minimal end-play. The adjusting nut is then locked in this position. Skill and judgment are required to determine when the wheel binds slightly in rotation. The more complicated the equipment, and the bulkier and heavier it is, so is a greater degree of skill and judgment required.

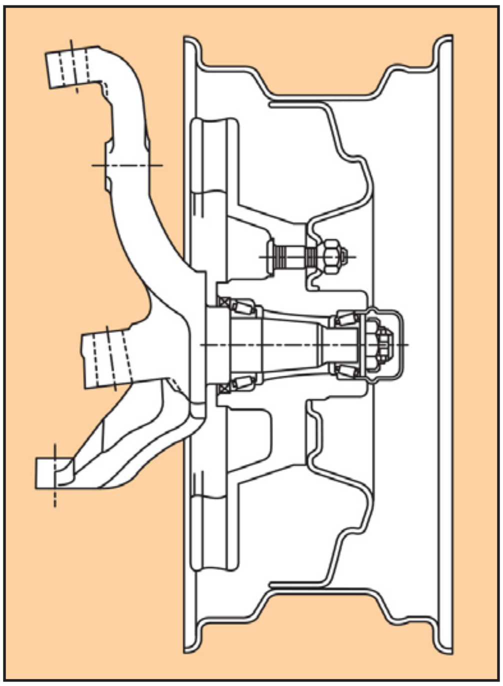

Figure 3 Truck Nondriven Wheel

For certain complex designs, large equipment, or high-production applications, manual setting may be too troublesome, of inappropriate accuracy and reliability, or too time consuming. The Timken Company has devised pre-set bearing assemblies and automated setting techniques as alternatives to manual setting.

Pre-set Bearing Assemblies

Many applications utilize or require the use of two-row or close-coupled bearing assemblies. This will depend upon the design and operating characteristics of the machine (e.g., thermal growth effects, high loads, etc.). To facilitate bearing settings of this type of design, pre-set bearing assemblies are frequently used. Pre-set bearing assemblies are available in a variety of forms, styles and arrangements, but for the most part are typically referred to as spacer bearings (Fig. 4). The majority of pre-set bearings are manufactured and supplied

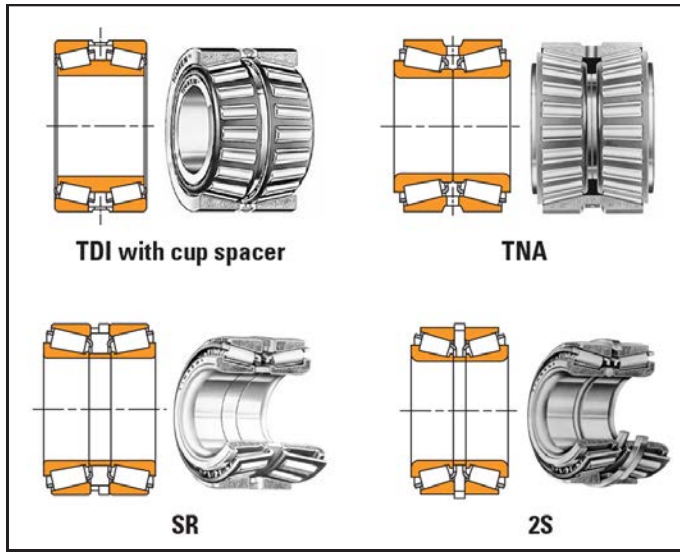

Figure 4 Example of Typical Preset Assemblies

with spacer rings "custom-fitted" between the bearing rows to control the internal clearances (Ref. "2S"- and "TDI"-types). As such, these customized or "matched" spacers cannot be interchanged with any other bearing assembly. Other pre-set assemblies such as "SR"- or "TNA"-types may apply interchangeable spacers and/or bearing components. Such interchangeable assembly components are designed to hold closer control of the critical tolerances that affect bearing setting; as a result, they can be randomly selected.

Each pre-set bearing is supplied from the manufacturer with a specified (unmounted) internal clearance or bench end-play (BEP). This BEP is chosen to provide the desired mounted setting range for the given application requirements. The mounted bearing setting range is determined from this BEP, based strictly on the effect of shaft and housing fits. Typically only one tight-fitted (shaft or housing) requirement is applied (i.e., on rotating member). This results in expected mounted setting ranges of less than 0.008". The mounted setting range of interchangeable component assemblies is typically wider than that for "matched" spacer assemblies. To apply pre-set assemblies in an application, simply mount and ensure proper clamping of the bearing components through the spacers.

Typical Pre-Set Bearing Assembly Applications

Pre-set bearing assemblies are widely and frequently used in many industrial applications. Typically this includes application in: planet pinions; hitch or linkage positions; transmission idler gears; fan hub shafts; water pump and idler pulley shafts; sheaves; conveyor idlers; winch drums, at the fixed and float positions of mining equipment; propel and swing drives; and in larger gear box drives.

Automated Bearing Setting Techniques

In addition to pre-set bearing assemblies, Timken has developed five popular automated bearing setting techniques (set-right; acro-set; projecta-set; torque-set; and clamp-set) as alternatives to manual adjustment.

Table 1provides a matrix format of various features of these

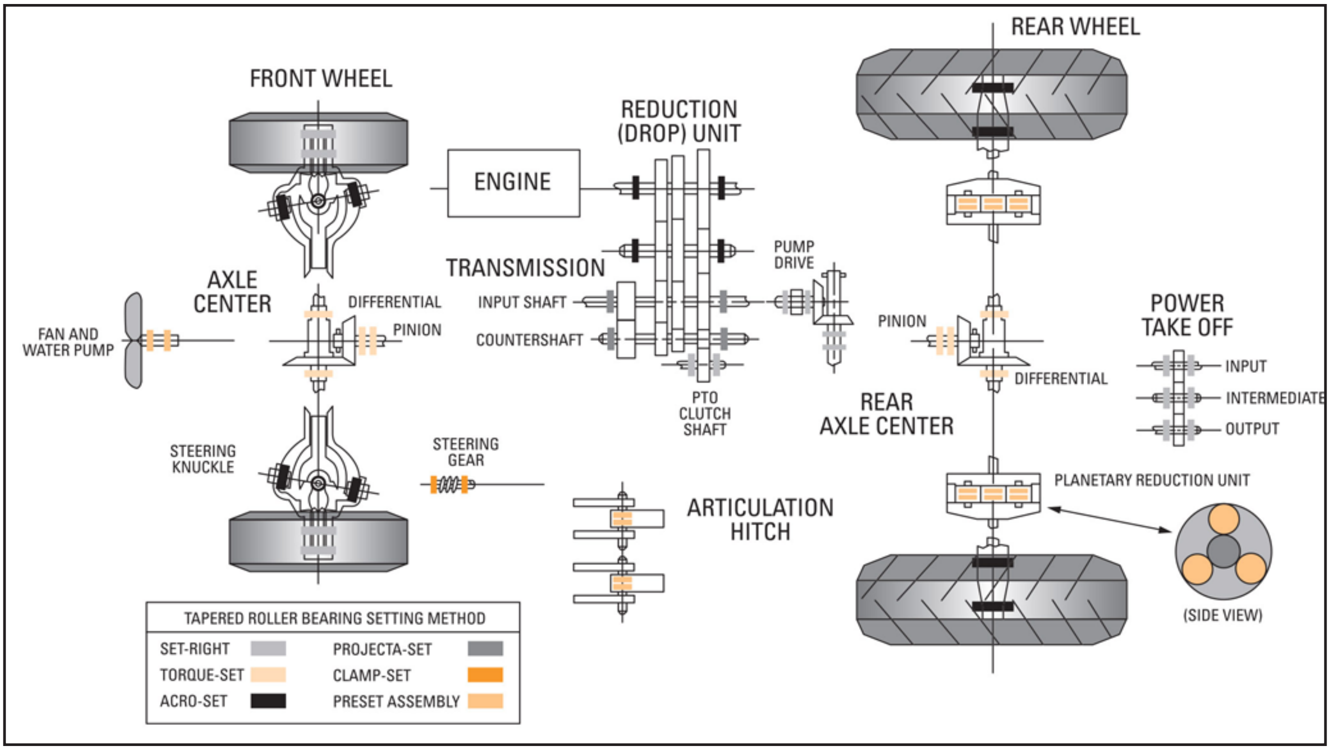

Figure 5 **Application of Automated Bearing Setting Techniques**

techniques. Each method's ability to hold a reasonably controlled mounted bearing setting "range" is compared on the first line of this table. These values are simply an indication of overall variability in setting for each method and have nothing to do with the "pre-load" or "end-play" setting target. For example, under the set-right column the expected (probable or 6-sigma) setting variation — due to control of certain bearing and housing/shaft tolerances — could range from a typical minimum of 0.008"–0.014". This range of setting can then be apportioned between end-play and pre-load to best optimize the bearing/application performance.

Figure 5 utilizes a typical four-wheel-drive farm tractor design to demonstrate examples of the common application of tapered roller bearing setting methods.

The specific definition, theory and formal process for the application of each technique is discussed in detail in the following sections.

Set-Right

Set-right eliminates manual setting adjustment of tapered roller bearings by controlling certain bearing and mounting system tolerances. The statistical laws of probability are applied to predict the effect of these tolerances on the bearing setting. Generally, the set-right method requires closer control of some shaft/housing machining tolerances, as well as closer control (with special class and code) of critical bearing tolerances.

The method considers that each component involved in the final assembly of a machine has a controllable tolerance range for critical dimensions. The laws of probability indicate that combinations of all low tolerances or all high tolerances will rarely occur in such an assembly. It then follows that for a "normal tolerance distribution" (Fig. 6), the overall dimen-

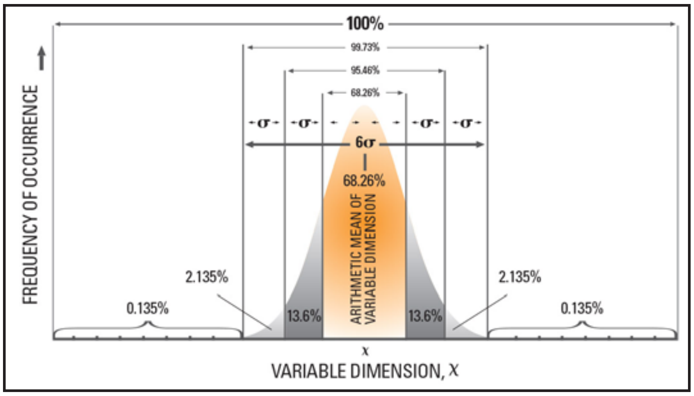

Figure 6 **Frequency Curve for a Normal Distribution**

sional stack-up of all parts will statistically tend to be somewhere in the middle of the total possible tolerance range.

The goal of the set-right method is to control only the most critical tolerances affecting bearing setting. These tolerances may be completely contained within the bearing or may involve certain mounting components (i.e., widths A and B of Fig. 1 or 7, plus shaft OD and housing ID). The result is an acceptable bearing setting that will occur within a desired range, with a defined statistical probability/reliability for all assemblies. (A probable reliability of 99.73% or 6-sigma is typical, but in higher-volume production a 99.994%, or 8-sigma reliability, is sometimes required). There are no adjustment steps required to use the set-right concept; the components of the machine are simply assembled and clamped.

All dimensions affecting the bearing setting in a machine assembly — such as certain bearing tolerances, shaft OD, shaft length, housing lengths, and housing bores — are considered as independent variables when calculating the probable range. In Figure 7 both cones and cups are mounted with

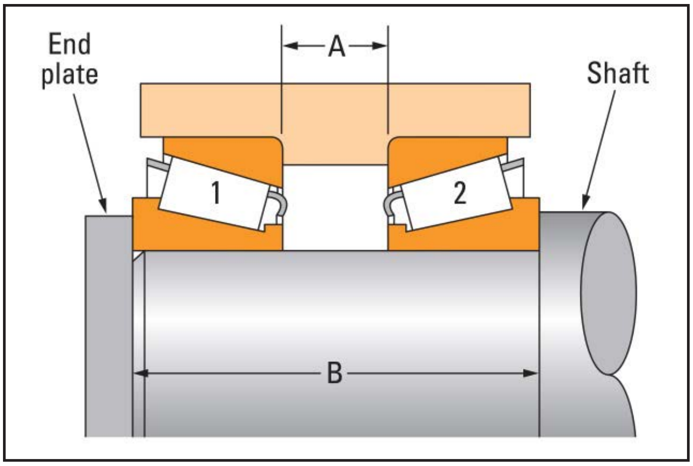

Figure 7 Machine Assembly

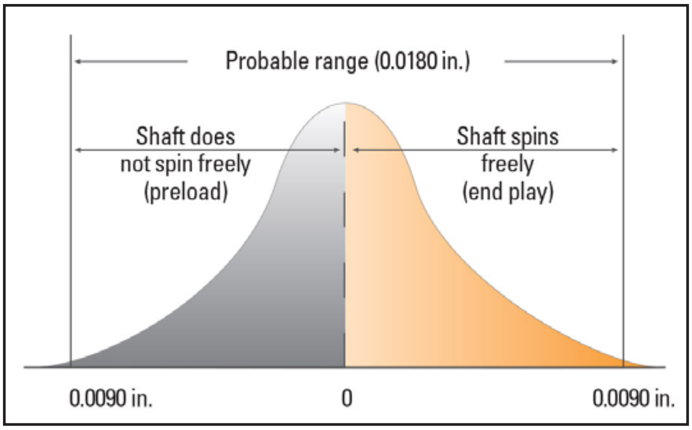

Figure 8 1st Spin Check

conventional tight fits and the end-plate is simply clamped against the end of the shaft.

***Special set-right considerations*:**

1. The overall calculated bearing setting range can vary significantly, depending on the bearing K-factor, its equivalent axial tolerance, and the number of tight-fitted components applied (i.e., larger with both tight cones and cups). A review of the application during the design stage will enable selection of special tolerance-controlled bearings and assist in optimizing the mounting design for the smallest probable setting range.
2. To control bearing mounting dimensions, the tolerances of the bearing system mounting dimensions used in establishing the probable setting range must be consistently maintained and, in some cases, more tightly controlled.
3. If the probable bearing setting range cannot be tolerated by the application, and attempts to reduce the larger tolerances are not practical or successful, then consider the spin-right variation to set-right.
4. The same class and code of bearing must be used for field service replacement as was used for the initial production.

Spin-right variation of the set-right technique. In some cases the probable bearing setting range with set-right can be too large for the application. To reduce this range and still apply the laws of probability, a technique called spin-right is used. This technique can be applied to applications that could also lend themselves to "shim-pack" adjustment (Fig. 16). To apply this method, the existing probable range is simply divided by a factor of two or three, depending on which is necessary to result in an acceptable setting range for the application.

For example, assume the probable bearing setting range for the design in Figure 1 is calculated to be 0.018" and the application demands a setting of 0.000" – 0.009" end-play. The current range must be divided by a factor of two. Thus with spin-right the desired shim increment (apply shim between endplate and shaft) would be equal to 0.009" and the following technique applied:

1. Assemble the gearbox without a shim and "spin check" — without the seal in place — to determine if the bearings are set with end-play or pre-load. Under the first spin check, if the shaft spins freely, end-play is present (Fig. 8) and the bearings are properly set.
2. If the shaft does not spin freely, the bearings are pre-loaded. Then a 0.009" shim must be installed. A second "spin check" should result in a freely rotating assembly indicative of end-play.
3. If pre-load is the desired setting, the spin-right procedure would be applied in reverse of the above example: if the shaft rotates freely, the bearings are not properly set and a shim of 0.009" would need to be removed.

Typical set-right applications. The set-right technique has been used for a wide variety of bearing setting applications, which include: tractor PTO assemblies — especially with blind end or split housing designs; automotive front-drive wheels; gear reducer shafts; planet pinions; and sprockets and torque hub units, as used on construction and mining equipment.

Acro-Set

This widely used setting technique is based on Hooke's law, which states: within the elastic material limit, component deflections are proportional to the load applied (i.e. $F = kx$, where k = system spring rate). This method assumes that total system deflection of an assembly will be consistent and repeatable for a given applied load (Fig. 9) in an application where parts and sections of parts are reasonably uniform throughout a group of units.

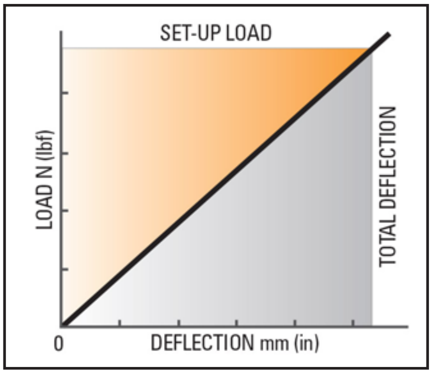

Figure 9 System Deflection

To establish the method for a given machine configuration, a dimensional reference condition known as the "deflection constant" must first be determined. The deflection constant is simply the (averaged) system deflection, resulting from a known "set-up" load applied through the bearings, as determined from the testing of several preproduction units. This system deflection is typically gauged by measuring a shim gap (Fig. 10).

The acro-set system constant is then developed; it equals the deflection constant for a given applied "set-up" load, plus the desired bearing setting. In production this constant is added to the measured shim gap to determine the final shim pack thickness for each unit.

The selection of the final shim pack thickness for each unit is simplified by use of a shim chart (Fig. 11). The shim pack thickness indicated on the chart includes the effect of the previously established acro-set constant. Note that the shim chart facilitates proper shim pack determination based on shim gap measurements taken at two positions, 180° apart. The planetary drive wheel assembly (Fig. 10) will be used to illustrate the acro-set technique.

1. The "set-up" load "P" was established by pretesting and is applied by 2 cap-screws (180° apart). The applied load is proportional to the bolt torque. (Commonly, a much larger "seating" force is first applied and the bearings rotated to ensure proper assembled positioning of the components prior to the acro-set shim gap measurement.)
2. Rotate or oscillate the bearings while applying the "set-up" load "P" and measure the shim gap — first at 0° and again at 180°.
3. Select the proper shim pack thickness (from the shim chart), equal to the measured gap plus the predetermined acro-set system constant (that was established from pre-tested assemblies). The Figure 10 chart averages the two readings and provides the final shim pack thickness. In this case, 0.66 at 180° and 0.61 at 0° gives a 0.97-thick shim pack.
4. Install the final shim pack and torque up all cap-screws to their clamp-up torque.

***Special acro-set considerations*:**

1. Bearing "seating" and "set-up" loading is typically applied with multiple cap-screws (i.e. Load = $NT/(d\mu)$ where: N = # of cap-screws, T = cap-screw torque, d = cap-screw diameter, and μ = coefficient of thread friction; where typical $\mu = 0.17$). Commonly, the applied seating force should be 2–3× Ca (90) and the applied set-up force chosen as ¾–1× Ca (90) of the lowest capacity bearing in the system.
2. Loose-fitted adjustable component; a loose-fitted member at the adjustable position is preferred. However, variations to tight-fitted cups and cones can be made as described below:
   A. The use of tight cups in carriers that are loose fitted in the housing.
   B. The use of a loose fitted "master" cup or cone for the bearing setting operation (mean fit compensated for in acro-set constant).
   C. Fixture designs with built-in compensation for the tight fitted member (projecta-set).
3. All components essential to the acro-set concept, such as housing walls and cover plates, must have a fairly uniform section size in successive production units.
4. The design must lend itself to applying a set-up load to the movable bearing race for adjustment.
5. The bearings must be rotated or oscillated while applying the set-up load.
6. The design must also lend itself to gap measurement of the movable member.
7. The thickness of the actual shim pack used should be verified.

It may be seen from this simplified schematic that projecta-set is basically a method of 'projecting' the two faces essential to measuring the spacer size from an otherwise inaccessible position to a point where gauging is possible.

A. The tapered roller bearing assembly with lower cone pressed into position on the shaft
B. Also lower and upper cups in the housing but without upper cone
C. Spacing element projects lower cone abutment face a known distance (x) (clear of shaft end)
D. Gauging element projects upper cup track the same distance (x)
E. Upper cone in gauging position
F. NO F??
G. Gauging point (e.g. for low volume, a dial indicator, or for high volume, an electronic transformer – LVDT) arranged and pre-set to indicate spacer size (S)

In this schematic, a movable base (H) is shown which, by the application of a known force, seats the upper cone in its 'projected' track, i.e. — the gauging element then giving a direct reading of the necessary spacer size. In practice, alternative methods may be used to suit any particular production

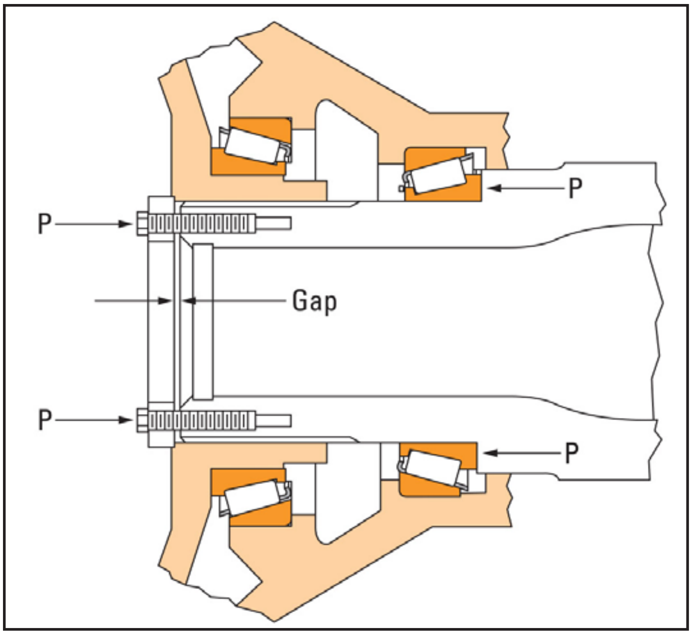

Figure 10 Planetary Drive Wheel

Second measured gap (at 180 deg), mm	First measured gap (at 0 deg), mm 0.56	0.58	0.61	0.64	0.66
0.56	0.89	0.91	0.91	0.94	0.94
0.58	0.91	0.91	0.94	0.94	0.97
0.61	0.91	0.94	0.94	0.97	0.97
0.64	0.94	0.94	0.97	0.97	0.99
0.66	0.94	0.97	0.97	0.99	0.99
0.69	0.97	0.97	0.99	0.99	1.02
0.71	0.97	0.99	0.99	1.02	1.02

Figure 11 TYPICAL ACRO-SET SHIM CHART

facilities and requirements; (for instance, a static base with the gauging load applied by a moving head from the top).

Typical acro-set applications. Examples where the acro-set method has been successfully applied to tapered roller bearings include: manual transaxles; drop box transmissions; farm tractor axle assemblies; power take-off units; planetary pinions; differential and pinion shafts; gear reducers; and off-highway truck and tractor wheels.

Projecta-Set

The projecta-set technique is similar in concept and application to acro-set, but adds additional versatility and sophistication through utilization of a special gauging fixture. This gauge enables one to "project" an inaccessible shim, spacer gap, or reference surface, to a position where it can be easily measured. This gauge typically incorporates the use of a dial indicator or an LVDT for measured readings. It is also readily applied in designs where the adjusting component (cone or cup) is tight-fitted without sacrificing assembly speed or accuracy. The method (Fig. 12) consists of two key gauging elements: a spacer sleeve (Ref. C) and a tapered gauging sleeve (Ref. D), of known (typically equal) design lengths (Ref. X). These sleeves will project the inaccessible spacer gap beyond the shaft end.

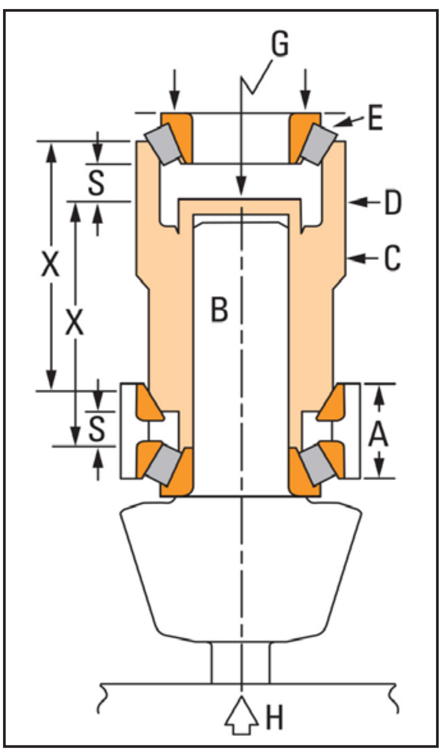

Figure 12 PROJECTA-SET Concept

To illustrate the projecta-set method, reference a typical spiral bevel pinion shaft assembly (Fig. 13). In this indirect-mounted, cone-adjusted design the bearing setting is achieved through the use of a spacer located between the two cone front faces. The cups and cones are tight-fitted in this application; the required gauging steps are:

1. Place the assembled pinion shaft, except for the upper cone and spacer, on the press table. Position the gauge

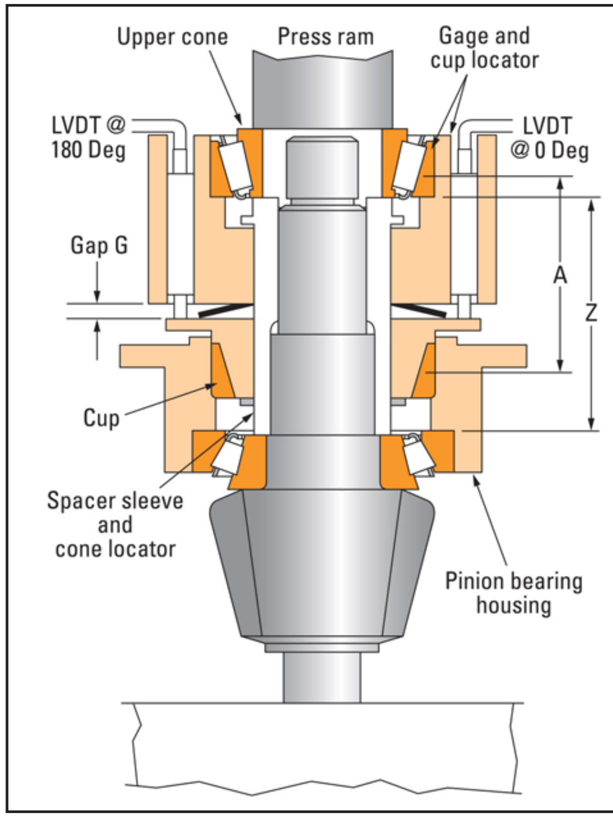

Figure 13 PROJECTA-SET Gauging Example With Pinion Shaft

onto the upper bearing cup and apply the upper bearing cone (Fig. 13).

2. Activate the press to clamp the gauge through the two bearing cones. A known axial load is applied through the bearing cups at this time by the Belleville spring internal to the gauge. (Note that the press is required simply to clamp the upper cone in place, against the spacer sleeve, for proper seating; some gauge designs accomplish this with a threaded nut design.)
3. Oscillate the gauge (handles) to seat the bearing rollers. The LVDT probes then measure the axial displacement between the two gauging members and the required spacer size is displayed on the digital readout.
4. The spacer size is determined by the gauge based on the formula (Fig. 13): $S = Z - A + K$

Where

S = Spacer size required

Z = Sleeve length (fixed)

A = Variable distance between corresponding diameters on the tapers of cone and cup locator ("zeroed" dimension is known)

K = Constant to compensate for system deflection due to gauge spring load, mean tight cone fit effect (loss of clearance), and the desired bearing setting

G = Measured gap, which represents the change in distance "A"; distance "A" includes "G"

***Special projecta-set considerations*:**

1. The size, weight, cost and design of the projecta-set gauge should be reviewed for specific application viability. Typical gauging cost for an industrial application, incorporating LVDT and internal gauge springs, is approximately $10,000 each. To increase set-up efficiency (for higher volumes over 30,000 assemblies/year), the designer should consider design/usage of a special automated press and press fixture to apply the gauge.

2. Separate gauges or interchangeable components (i.e. dual tapers) would be required if various models or shafts of the same application use a different bearing series.
3. An alternate method of field servicing would be required to set the tapered roller bearings (acro-set is similar and should be given primary consideration).

***Advantages of projecta-set*:**

- Prevents time-consuming teardowns to change shims or spacers in an application with tight cone or cup fits.
- It can be readily applied to automated assembly processes.
- Human judgment is minimized when compared to past traditional manual methods.
- The use of projecta-set gauges requires minimum training time.
- The projecta-set method provides consistent and reliable settings.

Torque-Set

The torque-set method is based on the principle that the rolling torque in a pre-loaded bearing directly increases as a function of the applied pre-load force (typically measured by dimensional pre-load). Laboratory tests have shown that the torque variation of a new bearing is small enough to effectively use bearing rolling torque as a basis for predicting/gauging a consistent dimensional pre-load setting. This relationship (Fig. 14) is established during pre-testing of several units and loads. Shims are added or subtracted after initial bearing rolling torque is measured to satisfy the desired bearing setting — either end-play or pre-load. A shim chart is normally used to assist selecting the final shim pack for each unit (Fig. 15).

The steps required to perform the torque-set technique are outlined below:

1. Assemble the unit with a reference (constant thickness) shim pack that assures a pre-load in the system (Fig. 16). Note the resulting bearing pre-load will actually differ for each assembly, depending on the variations in the accumulated tolerances of the component parts.
2. Measure the bearing rolling torque (Fig. 17).
3. Select the final shim pack thickness based on the pre-constructed shim chart (Fig. 15).
4. Install final shim pack and complete the assembly by installing all cap screws (Fig. 18).

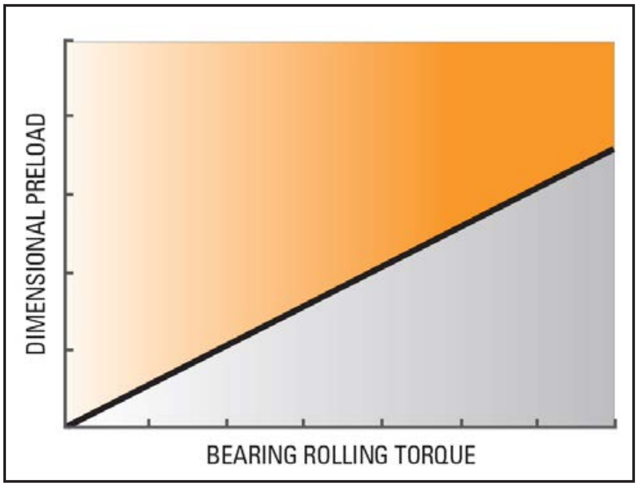

Figure 14 Dimensional Preload vs Bearing Rolling Torque

Measured Bearing Rolling Torque (N•m)	Shim Pack Addition (mm)
1.13	.20
1.36	.23
1.58	.23
1.81	.25
2.03	.25
2.26	.28
2.49	.28
2.71	.30
2.94	.30
3.16	.33
3.39	.33
3.62	.36
3.84	.38
4.07	.38
4.29	.41

Figure 15 TORQUE-SET Shim Pack Determination Chart

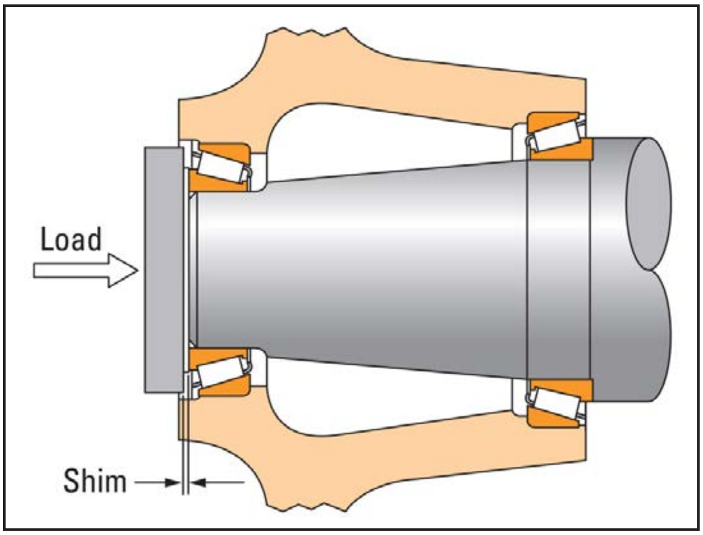

Figure 16 Assemble With Reference Shim Pack

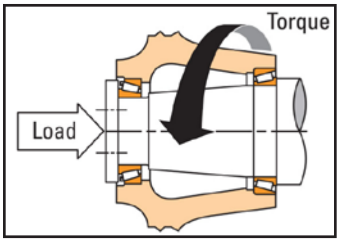

Figure 17 Measure Bearing Rolling Torque

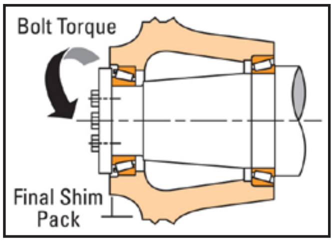

Figure 18 Assembly Complete

Bearing rolling torque is influenced by rotational speed and lubricant used. In any application using the torque-set approach, the lubricant and speed should remain constant.

The most common method of measuring bearing rolling torque is with a torque wrench. Sometimes a socket can be used, which fits over a nut on the end of the shaft or, if this is not possible, a special adapter can be made that fits the end of the shaft. In cases where the housing can be rotated, the torque wrench is adapted to the housing to measure rolling torque.

If a torque wrench cannot be used, a spring scale may be substituted to measure bearing rolling torque. Using a string wound around a gear or wheel and a scale, record the pull force needed to keep the assembly turning. Rolling torque is calculated by multiplying the radius of the gear or wheel around which the string has been wound by the pull force. This step could be avoided by a shim chart, which indicates pull force vs. shim pack size.

When measuring bearing rolling torque, turn the shaft as slowly as possible while maintaining smooth rotation.

Special Torque-Set considerations:

1. ***Ability to measure rolling torque.*** The design must lend itself to measuring the rolling torque of the bearings. Where other components such as seals, piston rings, etc. contribute to the torque, provisions must be made to recognize and isolate these torque values from the bearing rolling torque. An example would be to record the drag torque of a shaft and seal in an end-play condition, then "add-on" the required bearing rolling torque for the pre-load condition.
2. ***Ability to "reseat" tight-fitted members.*** When tight-fitted members are used for bearing adjustment, provisions must be made to "re-seat" or back-press that member after the torque-set load has been applied and the final shim pack determined.
3. ***Bearing rolling torque is influenced by rotational speed and applied lubrication***. These should be kept constant between units. The most common method of measuring rolling torque is with a torque wrench. When measuring rolling torque, turn the shaft as slowly as possible (estimated at 3.5 rpm) while maintaining smooth rotation.
4. ***Torque-set.*** *Should not be used if there is an unbalanced load* (*created, for example, by heavy parts, clutch plates or caliper brakes*); this would cause the torque to vary during rotation.
5. ***For field servicing,*** the torque-set method should not be used to reapply previously operated (e.g., run-in) bearings; a new set of bearings or alternate technique must be used.

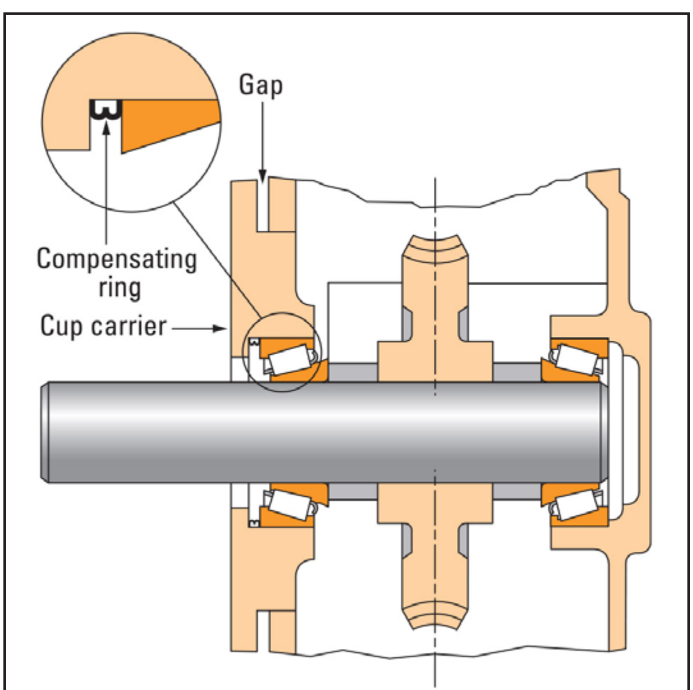

Figure 19 Worm Gear Reducer

Typical torque-set applications. Torque-set has been used successfully on various industrial and automotive applications. Typical applications include pinion and differential shafts, transmission shafts, and gearbox shafts.

Advantages of torque-set:

- Usually, no special fixtures or tooling are required; a torque wrench or simple spring scale and cord are all that are needed.
- No shim gap measurements are required; the shim pack is simply changed to obtain the correct setting.
- This method is useful in equipment where manual methods are physically impractical or difficult. However, it may not be practical in very large equipment.
- Torque-set can be applied to field servicing when new bearings are installed.

Mounting Designs and Setting Devices

Tapered roller bearings can be mounted in various configurations, and there are a variety of devices available to set the bearings to the desired end-play or pre-load in an application.

Cone-setting devices. In an indirect mounting, generally one cone is backed against a fixed shoulder while the other cone is movable and backed by some setting device.

A slotted nut (Fig. 20) can be used for obtaining the bearing setting. The nut is locked in place with a cotter pin. Both the nut and washer should be of sufficient size to give adequate backing to the cone. Two cotter pin holes in the shaft, spaced 90° apart, are used to obtain twice as many locking-positions-per-revolution of the nut, and a corresponding, closer bearing setting. A locknut, tongued washer, and lock-washer can be used instead of a slotted nut (Fig. 21). (See the *Auxiliary Parts* in the *Tapered Roller Bearing Guide* for other locknut arrangements.)

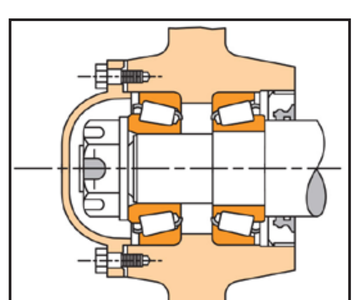

Figure 20 Slotted Nut

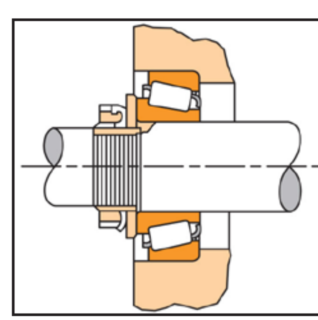

Figure 21 Locknut

A stake nut (Fig. 22) can be used for setting the bearings; and peening the thin section into a keyway slot locks it in place.

The setting in Figure 23 is obtained with shims and an end-plate held in place by cap screws in the end of the shaft. A slot may be provided in the end-plate to measure the shim gap.

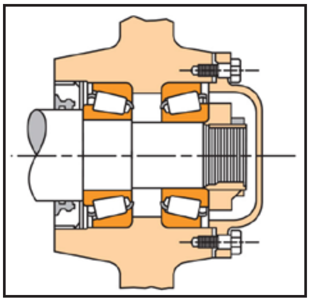

Figure 22 Stake Nut

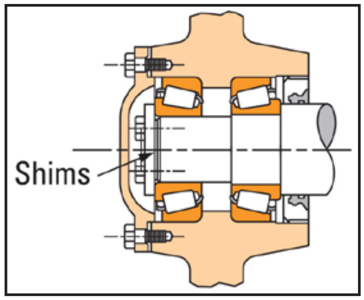

Figure 23 End Plate

A TDO-type bearing with cone spacer, above the centerline (Fig. 24) and a TNA-type bearing, below centerline, are manufactured with a fixed internal setting built into the bearing. The TDO-type bearing is shown assembled on the shaft with a cone spacer and clamped against a shoulder by an end-plate. The TNA-type bearing is assembled on the shaft with the cones butted together and similarly clamped against the shoulder. No further set-up provision is required in either case.

Cup-setting devices. In a direct mounting, generally one cup is backed against a fixed shoulder, with the movable cup positioned by some satisfactory setting device.

A cup carrier (Fig. 25, above centerline) and cup follower (below centerline) use shims for setting, and the carrier or follower is held in place by cap-screws.

Bearings can be set by the use of a selected cup spacer with the end-plate secured to the housing by cap-screws (Fig. 26).

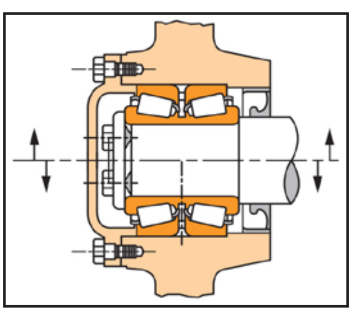

Figure 24 TDO and TDA

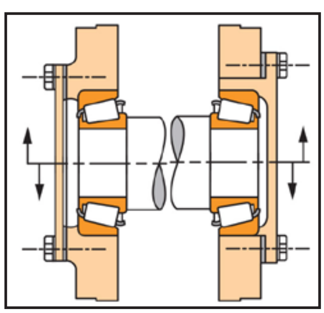

Figure 25 Cup Carrier and Cup Follower

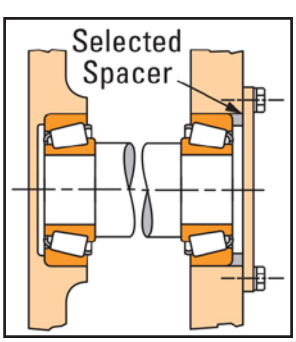

Figure 26 Selected Cup Spacer

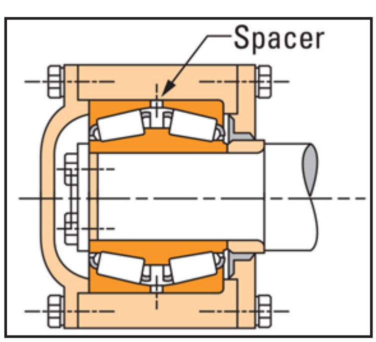

Figure 27 TDI

A TDI-type bearing with a cup spacer (Fig. 27) is supplied with the cup spacer providing a specific, fixed internal setting. The bearing is clamped by a cup follower through the cups and spacer against the housing shoulder. No further setup provision is required.

Summary

The fact that tapered roller bearings can be set is an advantage over other types of bearings. Manual setting has been considered an acceptable approach by many manufacturers and will continue to be used. The trend, however, is toward automated bearing setting procedures because of cost and more exacting performance requirements.

The selection of an automated bearing setting technique is best made early in the design stage.

But, if the equipment is already designed and built, one or possibly a combination of setting techniques could be incorporated to improve setting reliability and reduce assembly time. **PTE**

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Recalibrated Equations for Determining Effect of Oil Filtration on Rolling Bearing Life

William M. Needelman and Erwin V. Zaretsky

Summary

In 1991, Needelman and Zaretsky presented a set of empirically derived equations for bearing fatigue life (adjustment) factors (*LF*s) as a function of oil filter ratings. These equations for life factors were incorporated into the reference book, *STLE Life Factors for Rolling Bearings*. These equations were normalized ($LF = 1$) to a 10 μm filter rating at $\beta_x = 200$ (normal cleanliness) as it was then defined. Over the past 20 years, these life factors based on oil filtration have been used in conjunction with ANSI (American National Standards Institute)/ABMA (American Bearings Manufacturers Association) standards and bearing computer codes to predict rolling bearing life. Also, additional experimental studies have been made by other investigators into the relationship between rolling bearing life and the size, number, and type of particle contamination. During this time period, filter ratings have also been revised and improved, and they now use particle counting calibrated to a new National Institute of Standards and Technology (NIST) reference material, NIST SRM 2806, 1997. This paper reviews the relevant bearing life studies and describes the new filter ratings. New filter ratings, $\beta_{x(c)} = 200$ and $\beta_{x(c)} = 1{,}000$, are benchmarked to old filter ratings, $\beta_x = 200$, and vice versa. Two separate sets of filter *LF* values were derived based on the new filter ratings for roller bearings and ball bearings, respectively. Bearing *LF*s can be calculated for the new filter ratings.

Key Words: oil filter ratings; oil filtration; rolling-element-bearings; life prediction

Nomenclature

C_1 = Empirically determined constant
C_2 = Empirically determined exponent
C_u = Fatigue load or stress endurance limit of the bearing material
D_p = Bearing pitch diameter, mm
E_x = Particle removal efficiency, $(1 - 1/\beta_x) \times 100$ (%)
e_C = Lubricant contamination or oil cleanliness factor
FR = Filter rating, micron
h = Elastohydrodynamic (EHD) film thickness, μm
h_c = Central or average elastohydrodynamic (EHD) film thickness, μm
L_{10} = Bearing 10% or catalog life or the operating time that is exceeded by 90% of a group of bearings of a given type, hours, or millions of inner-ring revolutions
L_{50} = Bearing 50% or median life, or the time that is exceeded by 50% of a group of bearings of a given type, hours or millions of inner-ring revolutions
LF = Life adjustment factor for effect of filtration on bearing L_{10} life
M = Weibull slope or modulus
ND_x = Average number of particles downstream of the filter whose particle sizes are greater than x μm
NU_x = Average number of particles upstream of the filter whose particle sizes are greater than x μm
P = Applied dynamic equivalent (applied) load to the bearing, (N)
x = Particle size, μm
β = Filtration factor or filter rating, NU_x/ND_x
β_x = Old filter rating, (μm)
$\beta_{x(c)}$ = New filter rating, (μ)
κ = Ratio of the actual viscosity of the lubricant in the bearing to a reference viscosity where $\kappa \approx \Lambda^{1.3}$
Λ = Lubricant film parameter, h_c/σ
σ = Composite surface roughness, (σ 2 1 22) (μm)
σ_1, σ_2 = Root mean square (rms) surface roughness of contacting bodies (μm)
$\omega_{1,2}$ = Rotational speed (rpm)

Subscripts

1 = Body 1 or ball or roller
2 = Body 2 or bearing raceway

Introduction

It has long been recognized that lubricant contamination can affect bearing life, reliability, and performance. In 1976, T.E. Tallian (Refs. 1–2) was the first to publish a systematic study of the effect of contaminated lubrication on rolling-element fatigue life. He presented a probabilistic model (Ref. 1) to predict rolling-element fatigue life under conditions where the elemental surfaces in contact (raceways and balls or rollers) incur progressive damage during stress cycling from contamination in the lubricant. He subsequently correlated his analysis to surface density damage from experimental fatigue life data for ball bearing inner raceways (Ref. 2).

Hirano and Yamamoto (Ref. 3) reported that contaminants added to various lubricants could initiate scuffing in rubbing contacts. Dalal et al (Refs. 4–5) reported that ball bearing lives in excess of 50 times ANSI/ABMA (American National Standards Institute/American Bearing Manufacturers Association) bearing manufacturers' catalogue calculations (Refs. 6–7) were achieved by operating with pre-filtered, ultraclean lubricant, in which the only source of metallic contamination was the test bearing itself. However, Dalal et al did not run a control test lot of bearings without filtration to de-

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termine the exact life improvement attributable to filtration. They later induced raceway damage in their test bearings with a hardness indenter in an attempt to simulate in-service, contaminant-caused indentations; although reduced from the ultraclean values, the fatigue lives were still longer than the ANSI/ABMA calculations.

Fitzsimmons and Clevenger (Ref. 8) carried out an extensive test program with tapered roller bearings using a variety of lubricants and contaminants with controlled particle sizes, types, and concentrations. They noted that two-body wear occurs when a hard, rough surface plows a series of grooves in an opposing, softer surface. They stated that solid contaminants in lubricants are conducive to three-body abrasive wear, which occurs when particles are introduced between sliding surfaces and abrade material off both surfaces. They added the caveat that a certain amount of abrasive wear may be tolerable, depending on the application. As an example, noise in gears that results from looseness in the bearing system rather than from surface failure may constitute cause for "failure" of the application.

In tapered roller bearings, wear normally occurs on the surfaces where there is combined rolling and sliding contact; for example, between the roller ends and the cone large-end flange. Fitzsimmons and Clevenger found that wear of these surfaces increased linearly with the concentration of hard contaminant particles: In a discussion to the Fitzsimmons and Clevenger paper (Ref. 8) Kirnbauer and Ferris reported that 3 μm filtration prevents circulation of the hard particles that cause abrasive wear.

There were two independent investigations to determine the effect of oil filtration on rolling-element bearing life. These were those of Loewenthal et al (Refs. 9–11) from 1978 to 1982 for ball bearings at NASA Lewis (now Glenn) Research Center in Cleveland, Ohio, and Bhachu et al (Refs. 12–13) for roller bearings in 1981 at the Imperial College in London, England. In general, the results reported by both Loewenthal et al and Bhachu et al verified the trends of Tallian's analysis (Refs. 1–2).

Subsequent to the research reported above, additional experimental and analytical studies have been made by other investigators into the relationship between rolling bearing life and the size, number, and types of particle contamination (Refs. 14–20). Gabelli, Morales-Espejel, and Ioannides (Ref. 20) provide a comprehensive review of the literature, analysis, and a limited database related to particle damage in Hertzian contacts and rolling bearing life ratings. The reported research results from these references were similar to that reported by Tallian (Refs. 1–2), Fitzsimmons and Clevenger (Ref. 8), Loewenthal et al (Refs. 9–11), and Bhachu et al (Refs. 12–13). For a defined operating condition and bearing size and type, bearing life was a function of lubricant cleanliness and the number, size, and material properties of particles entering the Hertzian contact of the rolling-element and raceway. However, there is conflicting opinion as to whether the elastohydrodynamic (EHD) film thickness to surface composite roughness, or the Λ ratio, mitigates the negative effect of lubricant cleanliness on rolling-element fatigue life. That is, is the effect of contamination on bearing life less severe with increasing film thickness?

There are numerous published papers that have studied the relation of debris dents on the EHD film thickness and rolling-element fatigue. Most of these papers are summarized and discussed by Gabelli, Morales-Espejel, and Ioannides (Ref. 20). Wedeven and Cusano (Ref. 21) and Kaneta, Kanada, and Nishikawa (Ref. 22) experimentally studied the effects of moving dents and grooves on the EHD film thickness. Numerical solutions to the effect of simple debris dents on the EHD film thickness and surface and subsurface stresses are given by Venner (Ref. 23); Ai et al (Refs. 24–26); Nelias and Ville (Ref. 27); Ville et al (Refs. 28–29); and Chapkov Colin, and Lubrecht (Ref. 30). The results of these analyses and experiments suggest that the debris dents reduce the EHD film thickness at the dent site, increasing the contact and subsurface stresses, thereby reducing rolling-element bearing fatigue life.

In 1991, based on the research of Loewenthal et al (Refs. 9–11) and Bhachu et al (Refs. 12–13), Needelman and Zaretsky (Ref. 31) presented a set of empirically derived equations for bearing fatigue life adjustment factors (*LF*s) as a function of the oil filter ratings (*FR*), where $\beta_x = 200$. These equations for *LF* were incorporated into the reference book, *STLE Life Factors for Rolling Bearings* (Ref. 32). These equations (Ref. 31) were normalized ($LF = 1$) to a *FR* of 10 μm at $\beta_x = 200$ (normal cleanliness) as it was then defined by ISO Standard 4572 (Ref. 33) and a ratio of the EHD film thickness to surface composite roughness, Λ, of 1.1.

For over two decades these life factors based on oil filtration (Refs. 31–32) have been used in conjunction with ANSI/ABMA standards (Refs. 6–7) and bearing computer codes (Ref. 34) to predict rolling-element bearing life. However, in 1999 filter ratings underwent a revision and improvement (Ref. 35). They are now based on ISO 16889 (Ref. 36), replacing the older and now disavowed ISO 4572:1981 (Ref. 33).

A primary difference between these filter ratings is how particle size is specified and measured. The older filter test method used particle counters calibrated per ISO 4402:1991 (Ref. 37), with a variable material comprising irregularly shaped particles — AC Fine Test Dust (AC FTD). The new filter test method employs particle counters calibration per ISO 11171:1999 (Ref. 38), based on spherical particles traceable to a National Institute of Standards and Technology (NIST) reference material (SRM 2806, 1997) (Ref. 39), providing more accurate and verifiable results (Refs. 35 and 40). In addition, the new filter rating method uses a somewhat different test dust (ISO Medium Test Dust, ISO MTD) (Ref. 41), replacing the no-longer-available dust previously used, AC FTD.

The Needelman-Zaretsky life factors published in the early 1990s (Refs. 31–32) do not correspond to this revised and improved system of filter ratings. This paper focuses on the improved methods for measuring filter performance (improved filter ratings), and recalibrated bearing life factors incorporating these new ratings. The improved system of filter rating provides more accurate measurement of industrial filter performance, including performance improvements achieved by advancements in the design, materials, and manufacture of filters over the past 20 years.

In 2007 the ISO Standard 281, "Rolling Bearings — Dynamic Load Ratings and Rating Life" (Ref. 42), was radically changed. It is based on the inclusion of a "fatigue limit" in the bearing life calculations and the addition of a "contamination factor for circulating oil lubrication with online filters." This "contamination factor" is based on the analytical work of Ioannides et al (Ref. 17) and on a "cleanliness code according to ISO 4406" (Refs. 43–44). ISO 4406:1987 (Ref. 43) is the cleanliness code based on particle counters calibrated to the now obsolete ISO 4402 calibration (Ref. 37) that used irregularly shaped AC FTD. ISO 4406:1999 (Ref. 44) uses the approved ISO 11171 calibration method (Ref. 38) based on the NIST spherical-particle reference material. ISO 281:2007 (Ref. 42) incorporates the new ISO 4406:1999 cleanliness code (Ref. 44) but does not specify filter ratings to the levels of contamination.

Based on the above discussion it became the objective of the work reported herein to 1) review methods and data for determining the effects of lubrication oil particle quantity and size for calculating the bearing L_{10}; 2) experimentally calibrate older filter ratings that used AC FTD to new filter ratings using the NIST traceable particle counting calibration and ISO MTD; 3) re-calibrate the Needelman-Zaretsky equations for determining the effect of oil filtration on rolling-element bearing life to new filter ratings per ISO 16889 (Ref. 36); 4) determine filter ratings and related life factors based on new cleanliness codes per ISO 4406:1999 (Ref. 44) for ISO 281:2007 (Ref. 42); and 5) compare recalibrated filter life adjustment factors to cleanliness ratings presented in ISO 281:2007.

Failure Morphology

It is generally accepted that if a rolling-element bearing is properly designed, manufactured, installed, lubricated, and maintained, "classical" rolling-element fatigue is the failure phenomenon that limits bearing life (Ref. 45). Rolling-element fatigue is extremely variable but also statistically predictable — depending on the steel type, steel processing, heat treatment, bearing manufacturing/type, and operating conditions. This type of fatigue is a cycle-dependent phenomenon resulting from repeated stress under rolling-contact conditions and is considered high-cycle fatigue; Sadeghi et al provide an excellent review of this failure mode (Ref. 46).

Rolling-element fatigue can be simply categorized as either surface- or subsurface-initiated (Fig. 1). The subsurface-initiated fatigue failure is referred to as "classical" rolling-element fatigue. The fatigue failure manifests initially as a pit or spall that is generally limited in depth to the zone of the resolved maximum shearing stresses and in diameter to the width of the contact area (Fig. 1) (Ref. 45). Figure 1(a) illustrates the sequential progression of a subsurface-initiated crack from a non-metallic inclusion such as a hard oxide inclusion that acts as a stress raiser. With repeated stress cycles the crack propagates to form a crack network that reaches the surface, resulting in a spall shown sequentially in the bearing race track. At this point the bearing is no longer fit for its intended purpose and should be removed from service. Bearing rating life L_{10}, as defined by the standards ANSI 9–1990 (Ref. 6), ANSI 11–1990 (Ref. 7), and ISO 281:2007 (Ref. 42), is based on subsurface origin (classical)rolling-element fatigue (Ref. 45).

When the bearing is operated under conditions that deviate from the rating or reference condition, the fatigue origin can be of "surface" origin as illustrated in Figure 1(b). In this instance the spall can initiate from a defect or stress raiser on and/or near the surface of the bearing raceways and/or rolling-elements (Refs. 45, 47–48). In the instance of surface-initiated fatigue spall from hard-particle contamination, the load zone of the race and/or rolling-element is indented by the contaminant (Fig. 2). The indent acts as a stress raiser from which the crack initiates and then propagates to form a crack network into the subsurface region of resolved maximum shearing stresses (Fig. 1(b)). A spall similar in appearance to that of the surface-initiated spall is formed. The characteristic difference between the surface- and subsur-

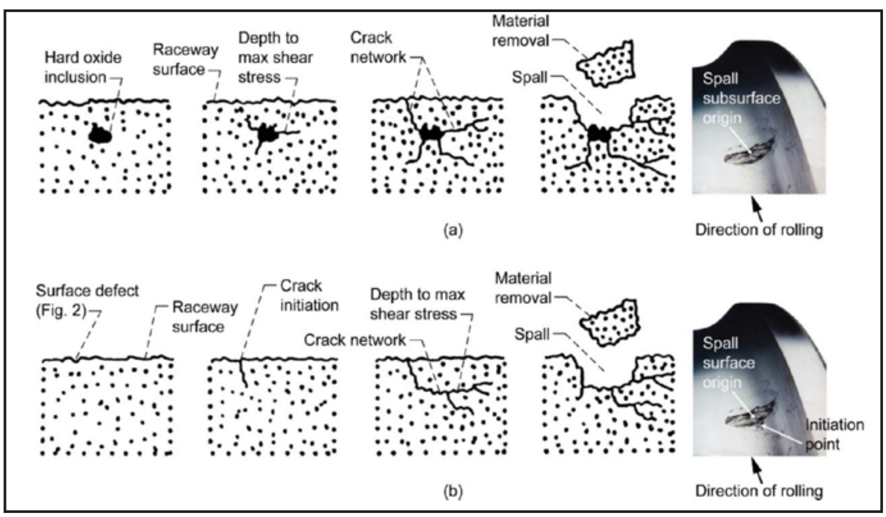

Figure 1 Subsurface- and surface-initiated rolling-element fatigue failures. For subsurface-initiated failure, spall begins in subsurface region of maximum resolved shearing stresses. For surface-initiated failure, spall begins at surface defect or discontinuity that acts as a stress raiser. (a) Subsurface- initiated spall from hard oxide inclusion; and (b) surface-initiated spall from debris indentation.

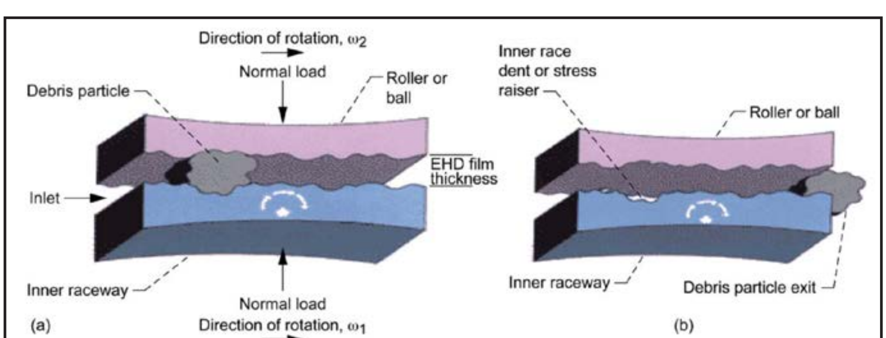

Figure 2 Stress raiser from indentation of debris particle entering Hertzian contact zone between rolling-element and raceway.

face-initiated spalls is that surface-initiated spalls result in “arrowhead-type” geometry at the leading edge or point of origin on the rolling-element or raceway surface (Fig. 1(b)).

The number, size, and material properties of particles entering the Hertzian contact of the rolling-element and raceway impact bearing life. The nature of the particles in the oil is a function of several processes:

1. Manufacturing processes (swarf, chips, and grit)
2. Internal generation, including wear debris and chemical attack on surfaces
3. Ingression from the environment (sand and dust)
4. Maintenance activities (making/breaking fittings and new oil)
5. Lubricant breakdown products (sludges, precipitates, and coke)

Typical particle size distributions from a variety of mechanical systems are shown in Figure 3 (Refs. 31 and 32). The greater number of smaller particles in each of the lubrication systems is due in part to wear mechanisms that generate smaller particles and, in part, to removal processes that tend to remove more large particles than small ones (Ref. 31). Steele (Ref. 49) reported a wide range of particulate levels in unused turbine oils.

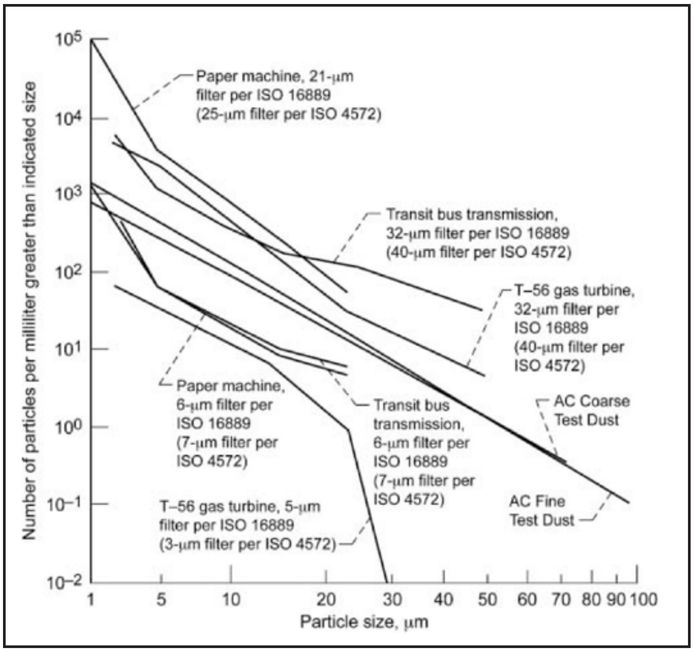

Figure 3 Comparing effect of oil filtration on particle contamination for mechanical systems according to particle size calibration, per ISO 4402 (Needelman and Zaretsky (Ref. 31); Zaretsky (Ref. 32)).

Work reported by Tonicello et al (Ref. 47) showed that for pairs of disks in rolling contact where one pair comprises silicon nitride (Si3N4) against AISI M–50 bearing steel, and the second pair comprises AISI M–50 against AISI M–50, and the oil particle contamination is AISI M–50 wear debris, the dent indentation on the raceway of the respective AISI M–50 disk was three times deeper with the Si3N4 disk than with the AISI M–50 steel disk on a AISI M–50 steel disk (Fig. 4). Greater stress concentration in the Hertzian contact results from deeper dents. It can be reasonably concluded that ceramic balls or rollers can lead to deeper dents in a mating steel raceway (Ref. 47).

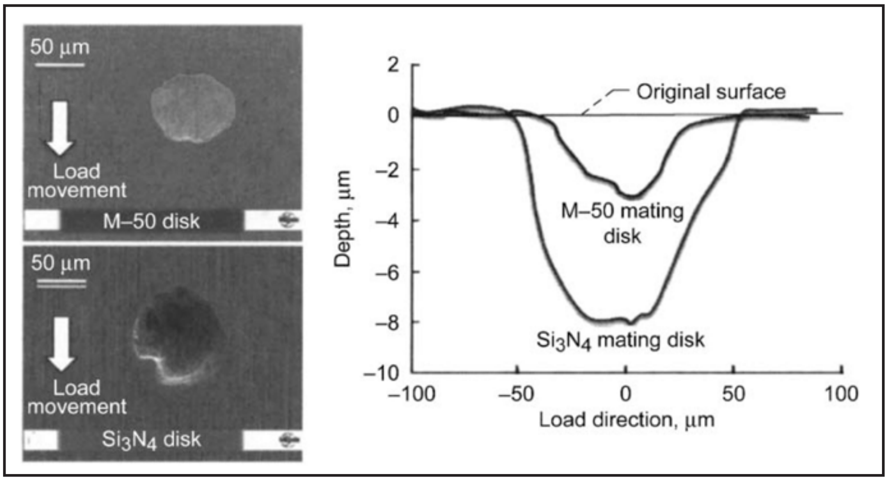

Figure 4 Dent profiles made on AISI M-50 steel disks by AISI M-50 particles. Mating disk (counterface) is Si3N4 or AISI M-50 steel. Figure from Tonicello, et al (Ref. 47) (Courtesy of Maney Publishing.)

Morales-Espejel and Gabelli (Ref. 50) discuss “the different hypotheses available to explain the interaction of sliding (and rolling) with the indentation marks in both gears and rolling bearings” under EHD lubrication. They present theory and analysis that are qualitatively verified by experiment. The micropitting phenomenon occurring around indentation marks is described by them with the same physical model of Morales-Espejel and Brizmer (Ref. 51) that takes into account the progression of surface fatigue induced by locally reduced lubrication conditions.

Filter Rating Procedure

The basic procedure for rating filters is shown in the flow diagram of Figure 5. During the multi-pass test, slurry of silica particles is continuously fed into a recirculating system.

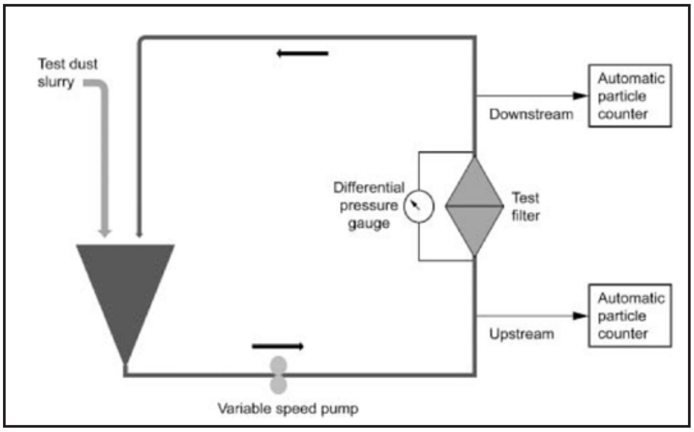

Figure 5 Basic multi-pass test procedure for filter rating using silica-based test dust and in-line particle counting.

Particles flow into the filter, where some are captured, and others return to the reservoir where they continue to recirculate. Throughout the test, particles upstream and downstream of the filter are quantified with electronic, automatic particle counters. The filter factor, or filter rating, for particle size x is defined as the ratio of upstream to downstream counts recorded during the test, denoted β_x and termed the "beta ratio":

$$\beta_x = (\text{Upstream counts} \geq x\,\mu\text{m}) \div (\text{Downstream counts} \geq x\,\mu\text{m}) = \frac{NU_x}{ND_x} \quad (1)$$

where NU_x and ND_x are the average number of particles upstream and downstream of the filter, respectively, whose particle size is greater than $x\,\mu$m.

Originally, automatic particle counters were calibrated using AC FTD, per ISO 4402:1991 (Ref. 37). This material comprises small, irregularly shaped particles of silica sand. Although widely used, AC FTD lacked traceability, had batch-to-batch variations, and reported size distributions of dubious accuracy especially below 10 µm. In order to make testing more reproducible, calibration methods were developed traceable to an NIST standard reference material, NIST SRM 2806 (Ref. 39). However, the new calibration method (per ISO 11171:1999) (Ref. 38) reports the size of particles (in micrometers) as the diameter of an equivalent sphere, rather than the longest dimension as in the original method. These alterations, in effect, changed the "micron ruler." As shown in Table I (Ref. 38), the particle sizes originally below 10 µm are reported larger, and sizes above 10 µm are reported smaller. For example, a silica particle reported to be 3 µm in size by

Table 1 Experimental Comparison of Contaminant Particle Size Classification Based on ISO Standards

Old Size, ISO 4572:1981 Method (Ref. 33), AC FTD Calibration $\beta_x = 200$ (µm)	New Size, ISO 16889:2008 Method (Ref. 36), ISO MTD Calibration $\beta_{x(c)} = 200$ (µm)
1	4.2
2	4.6
3	5.1
5	6.4
7	7.7
10	9.8
15	13.6
20	17.5
25	21.2
30	29.4
40	31.7

a particle counter calibrated to the old standard is now reported as 5.1 µm in size using the new calibration standard.

Along with calibrating automatic particle counters, AC FTD was also used as the test contaminant in the previous version of the multi-pass test. When it became unavailable in the 1990s, an alternative test material with similar chemical composition and size distributions was adopted — ISO MTD. Although using a slightly different test contaminant influences test results, the changes are less significant than those produced by the new particle size calibration described above. The β_x values obtained via the revised multi-pass test (ISO 16889:2008) (Ref. 36), using ISO MTD and particle counters calibrated to the NIST standard, are now reported as $\beta_{x(c)}$ values, where x is the micron size, per Equation 1, and c emphasizes the new calibration method.

Twenty years ago, leading filter manufacturers rated filters at the particle size x where $\beta_x = 200$ (Eq. 1). For example, a filter rated at $\beta_5 = 200$ has one out of every 200 particles equal to or greater than 5 µm pass through the filter during testing. However, most engineers and end-users prefer to think of filter ratings as the size where essentially no particles pass through the filter. In an attempt to reach this ideal, many manufacturers now also rate filters at the particle size x where $\beta_x = 1{,}000$. At this higher rating, only one particle in 1,000 passes through the filter during testing. Using the new ISO standard, modern

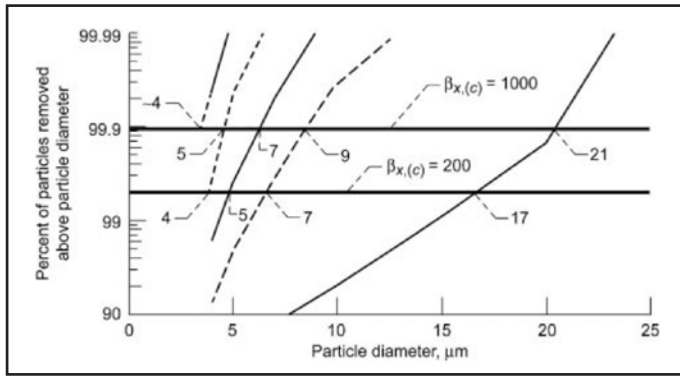

Figure 6 Representative results from multi-pass filter testing per ISO 16889:2008.

filter ratings are $\beta_{x(c)} = 200$ and $\beta_{x(c)} = 1{,}000$. Typical removal efficiency results are shown in Figure 6.

The removal efficiency E_x of any particle size x can be related to the β factor as follows:

$$E_x = (1 - 1/\beta_x) \times 100 \quad (2)$$

For any filter there is a large particle size above which essentially nothing passes, $\beta_x \geq 10^4$ and $E_x \rightarrow 100\%$. In contrast, there is also a small particle size for which $\beta_x \geq 1$ and $E_x \rightarrow 0\%$, so that nearly all particles this size and smaller freely pass through the filter and accumulate to copious amounts in recirculating systems. For intermediate sizes, a fraction of the particles are captured and the rest pass downstream. As an example, a filter with $\beta_{10(c)} = 1{,}000$ removes 99.9% of all particles ≥10 µm in size during a multi-pass test.

In summary, the changes to the ISO filter rating standard were as follows:

1. Particle counter calibration
   a. Changed from AC FTD to NIST calibration
   b. Increased accuracy and reproducibility
   c. Changed the "micron ruler"
2. Test contaminant
   a. Changed from AC FTD to ISO MTD
   b. Increased accuracy and reproducibility
3. Highest filter rating changed from $\beta_x = 200$ to $\beta_{x(c)} = 1{,}000$
   a. Closer to concept of "absolute rating"

Examples:

1. For $\beta_{5(c)} = 200$, 1 out of every 200 particles or 5 out of every 1,000 particles greater than 5 µm passes through filter during test.
2. For $\beta_{5(c)} = 1{,}000$, 1 out of every 1,000 particles greater than

5 μm passes through filter during test.

Results and Discussion

In 1991, based on the experimental research of Bhachu et al (Refs. 12–13) and Loewenthal et al (Refs. 9–11), Needelman and Zaretsky (Ref. 31) presented a set of empirically derived equations for bearing fatigue life (adjustment) factors (*LF*) as a function of oil filter ratings (*FR*). These equations were normalized (*LF* = 1) to a 10 μm filter rating at normal cleanliness (as it was then defined, where $\beta_x = 200$ per ISO 4572:1981 (Ref. 33). The life factor equations were incorporated into the reference book, *STLE Life Factors for Rolling Bearings* (Ref. 32).

The Needelman and Zaretsky oil filtration life factors (Ref. 31) have been used in conjunction with ANSI/ABMA standards (Refs. 6–7) and with bearing computer codes (Ref. 34). Experimental studies made by other investigators verify the relationship between rolling-element bearing life and the size, number, and types of particle contamination (Refs. 1–2; 4–5; 8, 14–20; 47–48). The ISO 281:2007 (Ref. 42) incorporates a rolling-element bearing life factor based on lubricant cleanliness and EHD film thickness based on the work of Ioannides et al (Ref. 17) but does not relate the lubricant cleanliness to the filter ratings.

Filter ratings have been revised and improved (Ref. 35). They are now based on an upgraded filter rating method per ISO 16889:2008 (Ref. 36), employing particle counts calibrated to an NIST standard (Refs. 36 and 39). The work reported here was undertaken to calibrate the "old" and obsolete filter ratings to the "new" filter ratings, and to recalibrate the previously published bearing life factors based on the old filter ratings to life factors based on the new filter ratings. The revised rolling-element bearing life factors were then compared to the life factors in the ISO 281:2007 Standard (Ref. 42) that are based solely on lubricant cleanliness levels.

Recalibration of Filter Ratings

Old $\beta_x = 200$ filter ratings (AC FTD calibration) were converted to new $\beta_{x(c)} = 200$ filter ratings using Table I and plotted in Figure 7. Although this transformation does not take into account the change in test contaminant to ISO MTD, alterations in counter calibration dominate over the change in test contaminant. The equation relating old $\beta_x = 200$ filter ratings (FR_{OLD200}) with new $\beta_{x(c)} = 200$ filter ratings (FR_{NEW200}) is:

$$FR_{NEW200} = 0.722\,(FR_{OLD200}) + 2.97 \tag{3a}$$

or

$$FR_{\beta_{x(c)}200} = 0.722\,(FR_{\beta_x 200}) + 2.97 \tag{3b}$$

and

$$FR_{OLD200} = 1.39\,(FR_{NEW200}) - 4.11 \tag{4a}$$

or

$$FR_{\beta_x 200} = 1.39\,(FR_{\beta_{x(c)}200}) - 4.11 \tag{4b}$$

Multi-pass filter tests were then performed according to the new ISO 16889:2008 (Ref. 36) method using 25 different filters over a wide range of filter efficiencies from five different manufacturers. This allowed plotting $\beta_{x(c)} = 1{,}000$ filter ratings with $\beta_{x(c)} = 200$ ratings. The relationship is plotted in Figure 8, and approximated by the equations:

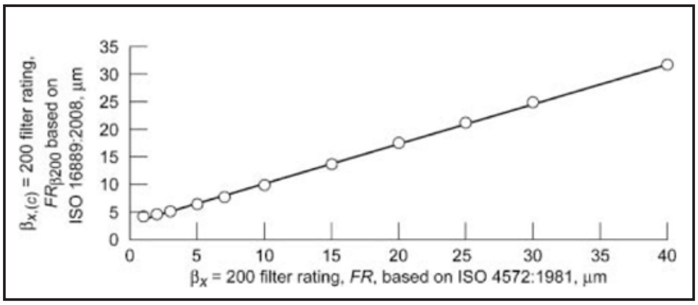

Figure 7 Comparison of new filter ratings — based on ISO 16889:2008 using spherical particle calibration per ISO 11171:1999 — with obsolete filter ratings based on ISO 4572:1981 using AC FDT calibration, per ISO 4402:1991.

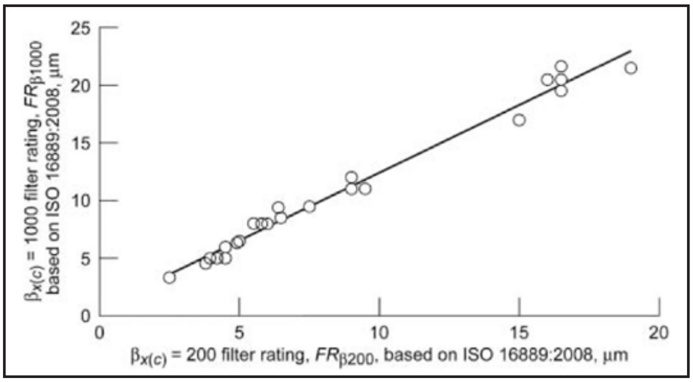

Figure 8 Comparison of $\beta_{x(c)} = 1{,}000$ filter ratings with $\beta_{x(c)} = 200$ filter ratings; filters tested, per ISO 16889:2008.

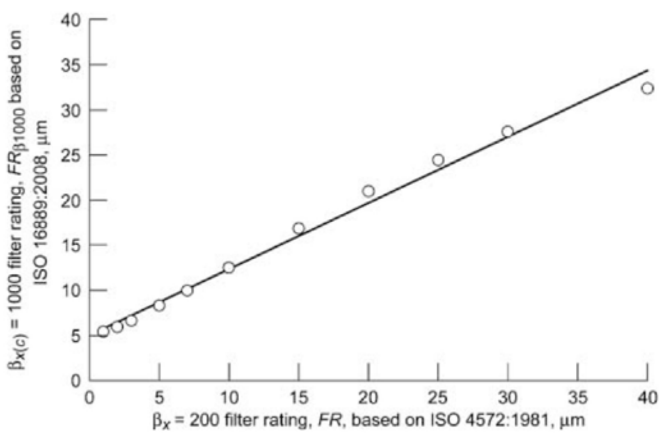

Figure 9 Comparison between $\beta_{x(c)} = 1{,}000$ filter ratings based on ISO 16889:2008 and $\beta_x = 200$ filter ratings based on obsolete ISO 4572:1981.

$$FR_{NEW1000} = 1.77\,(FR_{NEW200}) + 0.650 \tag{5a}$$

or

$$FR_{\beta_{x(c)}1000} = 1.77\,(FR_{\beta_{x(c)}200}) + 0.650 \tag{5b}$$

and

$$FR_{NEW200} = 0.855\,(FR_{NEW1000}) - 0.556 \tag{6a}$$

or

$$FR_{\beta_{x(c)}200} = 0.855\,(FR_{\beta_{x(c)}1000}) - 0.556 \tag{6b}$$

By cross-plotting the data of Figure 7 with that of Figure 8, a relation between old and obsolete $\beta_x = 200$ filter rating and new filter rating $\beta_{x(c)} = 1{,}000$ is obtained (Fig. 9), as described by the equations:

$$FR_{NEW1000} = 0.848\,(FR_{OLD200}) + 4.14 \tag{7a}$$
or
$$FR_{\beta_{x(c)}1000} = 0.848\,(FR_{\beta_x 200}) + 4.14 \tag{7b}$$
and
$$FR_{OLD200} = 1.18\,(FR_{NEW1000}) - 4.88 \tag{8a}$$
or
$$FR_{\beta_x 200} = 1.18\,(FR_{\beta_{x(c)}1000}) - 4.88 \tag{8b}$$

Rolling Bearing Fatigue Life Factors

There were two independent investigations to specifically determine the effect of oil filtration on rolling-element bearing life. These were the studies of Loewenthal et al. (Refs. 9–11) and Bhachu et al (Refs. 12–13).

Roller bearings. Bhachu et al (Refs. 12–13) used a gear test machine to generate wear debris. The gear wear debris, verified by ferrography to be representative of that found in helicopter gearboxes, was used as the contaminant. Rolling-element fatigue tests were conducted with 25 μm bore roller bearings having a 2,957 N radial load. For each test series, gear oil flow was passed through one of four possible filters of different ratings from 2.5–40 μm or through an electromagnetic separator and continuously supplied to a parallel roller-bearing fatigue tester. EHD film thickness and Λ values during testing are shown in Table 2.

Significantly, tests run with 40 μm filtration for only 30 min before switching to 3 μm filtration showed substantially the same lives as if all running had been with a 40 μm filter. Apparently the early damage could not be healed, at least in these small roller bearings. The test results are also shown in Table 2. These results show that life increased with improved filtration.

The original filter ratings were based on the old filter rating method, ISO 4572:1981 (Ref. 33), with $\beta_x = 200$. From these roller bearing life data (Table 2), it was assumed (Ref. 31) that the filter life factor takes the following form:

$$LF \approx C_1 (FR)^{-C_2} \tag{9a}$$

C_1 is an empirically determined constant and C_2 is an empirically determined exponent. The experimentally determined L_{10} lives from Table 2 for filter ratings $\beta_{x\,200} = 3$ and 25 μm are 8 and 2.5 million inner-race revolutions. Solving for C_1 and C_2 and normalizing $LF = 1$ when $FR = 10$ μm at $\beta_x = 200$ (normal cleanliness), the following empirical relation was obtained:

$$LF \approx 3.5\,(FR_{OLD200})^{-0.55} \tag{9b}$$
or
$$LF \approx 3.5\,(FR_{\beta_x 200})^{-0.55} \tag{9c}$$

Substituting the new filter ratings into Equation 9, from Equations 4 and 8, respectively, allows calculating bearing life factors for new filter ratings. For $\beta_{x(c)} = 200$:

$$LF \approx 3.5\,[1.39\,FR_{NEW200} - 4.11]^{-0.55} \tag{10a}$$
or
$$LF \approx 3.5\,[1.39\,FR_{\beta x(c)200} - 4.11]^{-0.55} \tag{10b}$$

For Equation 10, where $FR_{\beta x(c)200} \leq 4$, $LF \approx 2.8$.
For $\beta_{x(c)} = 1{,}000$:

$$LF \approx 3.5\,[1.18\,FR_{NEW1000} - 4.88]^{-0.55} \tag{11a}$$
or
$$LF \approx 3.5\,[1.18\,FR_{\beta x(c)200} - 4.88]^{-0.55} \tag{11b}$$

For Equation 11, where $FR_{\beta x(c)1000} \leq 5$, $LF \approx 3.5$.

Using Equation 9, the L_{10} lives for roller bearings were predicted by normalizing Equation 9 to the experimentally obtained L_{10} life, using the 3 μm-rated ($\beta_x = 200$) filter. These results are presented in Table 2 for comparison purposes.

The post-test inner-raceway measurements for 40 μm ($\beta_x = 200$) filtration showed greater out-of-roundness than in the untested bearing. Less out-of-roundness was observed with finer filtration down to the 8 μm ($\beta_x = 200$) rating. Virtually no out-of-roundness was observed when the 3 μm ($\beta_x = 200$) filter was used. Below the 3 μm ($\beta_x = 200$) level the measurement was similar to that of the un-used bearing. Bhachu et al (Refs. 12–13) suggested that particles smaller than 3 μm ($\beta_x = 200$) were too small to have any effect on roundness and merely passed through the contacts of the rollers and raceways.

There is a strong suggestion from the data of Table 2 that the lack of contamination contributes to improvement in bearing raceway surface finish during operation. There appears to be a correlation between the lubricant film parameter Λ after testing and rolling-element fatigue life as evidenced by the

Table 2 Effect of Oil Filtration on the Rolling-Element Fatigue Life of 25-mm-Bore Roller Bearings[a]

				Experimental Life, Millions of Inner-Race Revolutions						
Test Filter Rating ($\beta_x \geq 200$) (μm)	Composite Surface Roughness, σ after testing (μm)	Calculated Film Thickness (μm)	Film Parameter after Testing, Λ	10% Life, L_{10}	90% Confidence Limits	50% Life, L_{50}	90% Confidence Limits	Weibull Slope, *m*	Failure Index[b]	Predicted L_{10} Life from Eq. [9], 10^6 Inner-Race Revolutions
40	0.41	0.58	1.4	1.5	1.0–2.0	2.1	1.9–2.3	5.8	10 out of 10	1.9
25	0.36	0.58	1.6	2.5	2.0–3.0	3.0	2.9–3.1	9.4	10 out of 10	2.5
6	0.32	0.58	1.8	4.5	3.4–5.8	5.9	5.4–6.3	7.0	10 out of 10	5.5
3	0.26	0.58	2.2	8.0	5.5–11.6	12.0	10.7–13.5	4.7	10 out of 10	8.0[c]
2.5	0.22	0.58	2.6	6.5	3.6–11.8	12.4	9.9–15.5	2.9	10 out of 10	8.8
Magnetic	0.20	0.58	2.9	5.0	2.9–8.5	10.0	8.3–12.0	3.2	10 out of 10	–
40	0.41	0.99	2.4	1.8	–	3.3	–	3.1	10 out of 10	2.2
3	0.26	0.99	3.8	9.3	–	16.2	–	3.4	10 out of 10	9.3[c]
40/3[d]	0.41	0.99	2.4	1.7	–	3.1	–	3.3	10 out of 10	2.2

a) Radial load, 2,975 N; original surface composite roughness (rms) 0.33 μm. From Bhachu, et al. (12) and Sayles and Macphearson (13).
b) Number of fatigue failures out of number of bearings tested.
c) Life prediction normalized to 3-μm filter, $\beta_x = 200$.
d) Test run with 40-μm filter for 30 min before switching to 3-μm filter.

data.

Ball bearings. Loewenthal et al (Refs. 9–11) performed a series of tests to measure the quantitative effects of filtration on rolling-element fatigue life. Four levels of filtration were investigated using full-flow ($\beta_x = 200$) filters rated at 3, 30, 49, and 105 µm. The 3 µm ($\beta_x = 200$) filter used for these tests had been developed to replace the original 40 µm ($\beta_x = 200$) filter for a helicopter gas turbine lubrication system. During service these new filter elements were not only found to provide a much cleaner lubricant — with less component wear — but contrary to prior belief, to also greatly extend the time between filter and oil changes — as discussed by Loewenthal et al.

The test bearings were 65 µm deep-groove ball bearings run at 15,000 rpm under a radial load of 4,580 N, which produced a maximum Hertz stress of 2,410 MPa. The lubricant contaminant rate was 0.125-g/hr-per-bearing. The test environment was designed to simulate an aircraft lubrication system containing multiple bearings, pumps, and other components commonly found in such systems. Test temperature was 347 K. The test lubricant was a MIL–L–23699-type, which produced a Λ value of 3.3, based on race and ball pre-test surface finish measurements. The test contaminant was similar to the particulate matter found in the lubricant filters of 50 IT8D commercial engines (Jones and Loewenthal (Ref. 52)). Because this engine has a number of carbon-graphite bearing sump seals, replication of oil contaminants in engines with "windback-type" labyrinth seals demanded the use of a contaminant made of 88 percent carbon-graphite dust, 11 percent Arizona test dust, and 1 percent stainless steel particles.

The results of these tests are summarized in Table 3. As with the work of Bhachu et al (Refs. 12–13), improved filtration increased bearing life. However, for the contaminated tests there appears to be no statistical difference in life obtained between the 3- and 30 µm filters. Because of the severe wear obtained, the contaminated 105 µm filter test series was suspended after 448 hours on each bearing. No fatigue failures were encountered due to the gross wear of the bearing races. Based upon the test results between the 3- and 49 µm filters, the following life relation is suggested from the data for $\beta_x = 200$ filter rating for ball bearings:

$$LF \approx 1.8\,(FR_{OLD200})^{-0.25} \tag{12a}$$

or

$$LF \approx 1.8\,(FR_{\beta_x 200})^{-0.25} \tag{12b}$$

Using Equation 12, the L_{10} lives of ball bearings were predicted by normalizing Equation 12 to the experimentally obtained L_{10} life using the 3 µm ($\beta_x = 200$) rated filter. These results are presented in Table 3 for comparison purposes.

Substituting the filter rating for FR ($\beta_x = 200$) from Equations 4 and 8, respectively, into Equation 12, filter life factors can be calculated for the new filer ratings where:

1. For $\beta_{x(c)} = 200$ filter rating:

$$LF \approx 1.8\,[1.39\,(FR_{NEW200}) - 4.11]^{-0.25} \tag{13a}$$

or

$$LF \approx 1.8\,[1.39\,(FR_{\beta x(c)200}) - 4.11]^{-0.25} \tag{13b}$$

for Equation 13, where $FR_{\beta x(c)200} \leq 4$, $LF \approx 1.6$.

$$LF \approx 1.8\,[1.18\,(FR_{NEW1000}) - 4.88]^{-0.25} \tag{14a}$$

or

$$LF \approx 1.8\,[1.18\,(FR_{\beta x(c)1000}) - 4.88]^{-0.25} \tag{14b}$$

For Equation 14, where $FR_{\beta\,x(c)\,1,000} \leq 5$, $LF \approx 1.8$.

Table 4 provides results from Equations 6, 9, and 12 for various filter ratings. The resultant life adjustment factors can be used to adjust the calculated bearing L_{10} or catalog life to account for filtration level in the lubricant system. These LF values are normalized to filter ratings (FR) of 10 µm at $\beta_{x(c)} = 200$ and 13 µm at $\beta_{x(c)} = 1{,}000$ (normal cleanliness) and are independent of the Λ and/or κ values, loading conditions, and bearing size. Based upon the data of Loewenthal et al (Refs. 9–11), it is not recommended to use a life adjustment factor less than 0.5 — even when no filter is used. Further, Equations 9 and 12 may reflect the differences in the effect of particle damage between roller and ball bearings.

Technology for improved oil filtration is commercially available. By minimizing the number of harmful particles entering a rolling-element bearing, oil filtration can substantially extend bearing life. In addition to machine-generated wear debris and ambient mineral dusts, all-too-frequent high contamination levels in new oil also requires good filtration.

No reported testing has been performed comparing grease lubrication, which entraps wear debris, with oil lubrication

Table 3 Comparison of Ball Bearing Fatigue Life Results with Ultra-Clean Lubricant and Different Levels of Filtration in a Contaminated Lubricant[a]

		Experimental Life (h)				Confidence Number (%)[c]		
Test Series (Lubricant Condition)	Test filer Rating ($\beta_x \geq 200$) (µm)	10% Life, L_{10}	50% Life, L_{50}	Weibull Slope, m	Failure Index[b]	L_{10}	L_{50}	Predicted L10 Life from Eq. [12] (h)
Ultraclean	3	1,099	1,741	4.1	5 out of 9	–	–	1,099[d]
Clean (Baseline)	49	672	2,276	1.5	9 out of 32	76	–	547
Contaminated	3	505	993	2.8	10 out of 16	93	99	505[f]
Contaminated	30	594	857	5.1	11 out of 16	96	99	284
Contaminated	49	367	533	5.1	20 out of 32	99	99	251
Contaminated	105[e]	–	–	–	–	–	–	208

a) Radial load, 4,580 N; speed, 15,000 rpm; temperature, 347 K; test lubricant, MIL-L-23699 type; film parameter Λ, 3.3. From Loewenthal, et al. (9)–(11).
b) Number of fatigue failures out of number of bearings tested.
c) Probability (expressed as a percentage) that bearing fatigue life in a given test series will be inferior to the life obtained with ultraclean lubrication. A 90% or greater confidence number is considered statistically significant.
d) Life prediction normalized to 3-µm filter, $\beta_x = 200$ and $L_{10} = 1{,}099$ h.
e) Test series was suspended after 448 test hours on each of the test bearings due to excessive bearing wear. No fatigue failures were encountered.
f) Life prediction normalized to 3-µm filter, $\beta_x = 200$ and $L_{10} = 505$ h.

Figure 4 Needelman-Zaretsky Oil Cleanliness (Filter) Life Factors (*Lfs*) Based on Oil Filter Rating (*Fr*)

Filter Rating, FR (µm)			Life Factor, LF	
(Old) $\beta_x = 200$	(New) $\beta_{x(c)} = 200$ Eq. [3]	(New) $\beta_{x(c)} = 1{,}000$ Eq. [7]	Roller Bearing, Eqs. [9]–[11]	Ball Bearing, Eqs. [12]–[14]
3	5	7	2.0	1.4
6	7	9	1.4	1.2
8	9	11	1.1	1.1
10[a]	10	13	1.0	1.0
12	12	14	0.9	1.0
25	21	25	0.6	0.8
40	32	38	0.5	0.7
49[b]	38	46	0.4[b]	0.7
60[b]	46	55	0.4[b]	0.6
105[b]	79	93	0.3[b]	0.6

a) Normalized to FR = 10 µm at ($\beta_x = 200$).
b) For filter ratings at $\beta_x = 200$ exceeding 40 µm; with no filtration it is not recommended to use LF < 0.5.

for the same bearings, with or without oil filtration. It is suggested that for long-term application a $LF = 0.5$ for grease lubrication be considered the same as oil lubrication for bearings without filtration where no periodic re-greasing of the bearing occurs. Where the bearing is continuously or periodically re-greased, a $LF = 1$ should be considered.

Comparison of Filter Life Factors (*FLFs*) to ISO Standard 281:2007

In 2000 the International Organization for Standardization (ISO) modified the standard ISO 281:1990 — "Rolling Bearings: Dynamic Load Ratings and Rating Life" — to include a fatigue limit (Ref. 53). The endurance or fatigue limit as applied to rolling-element bearings is based on the theoretical work presented in 1985 by Ioannides and Harris (Ref. 54). It is a theoretical load or shearing stress (based on a Hertzian contact stress) below which no fatigue failure is assumed to occur, and therefore where fatigue life is infinite. ISO 281:2007 replaced ISO 281:1990, as modified in 2000. The 2007 standard adopted the 1999 approach to bearing life calculations presented by Ioannides, Bergling, and Gabelli (Ref. 55), and includes the effects of lubricant contamination on bearing life.

The ISO 281:2007 Standard (Ref. 42) incorporates a new service life formula that integrates all life adjustment factors *LF* in what is now called a_{ISO}. The life factor a_{ISO} includes four interdependent factors: 1) lubrication regime, κ; 2) lubricant contamination or oil cleanliness, e_C; 3) applied dynamic equivalent load (applied load) to the bearing, *P*; and 4) fatigue load or stress endurance limit of the bearing material, C_u.

$$a_{ISO} = f\left[\frac{e_C C_u}{\mathrm{P}}, \kappa\right]. \tag{15}$$

From the ISO 281:2007 Standard (Ref. 42) p. 24 —

"...when the lubricant is contaminated with solid particles, permanent indentations in the raceway(s) (and rolling-elements) can be generated when these particles are overrolled. At these indentations local stress risers are generated which will lead to a reduced life of the rolling bearing. This life reduction due to contamination in the lubricant film is taken into account by the contamination (life) factor e_C."

In the ISO 281:2007 Standard (Ref. 42), the contamination factor e_C is given in a table based upon levels of contamination that are not tied to specific filter ratings.

The standard states that the contamination life factor is dependent on the following:

- Type, size hardness and quantity of the (contaminant) particles
- EHD lubricant film thickness (viscosity ratio, κ)
- Bearing size (bearing pitch diameter, D_p)

The lubrication regime is defined by EHD theory; in the standard it is defined by the parameter κ, the ratio of the actual viscosity of the lubricant in the bearing at operating temperature to a reference viscosity. The reference viscosity is that which would produce a lubricant film thickness equal to the composite surface roughness of the rolling-element and the raceway, or $\Lambda = 1$. If $\kappa < 1$, the contact is in a boundary lubrication regime where the surface asperities of the rolling-element and the raceway are in contact. It is preferable to operate the bearing in a lubrication regime where $\kappa \geq 1$. As κ increases, bearing life increases. The κ value in the standard is based on what is termed in EHD theory as the "lubricant factor," or Λ. The Λ is equal to the EHD film thickness *h* divided by the composite surface finish σ of the rolling-elements in contact with the raceway:

$$\Lambda = \frac{h}{\sigma} \tag{16a}$$

where

$$\sigma = (\sigma_1^2 + \sigma_2^2)^{1/2} \tag{16b}$$

and σ_1 and σ_2 are the root mean square (rms) surface roughness of contacting bodies. Unfortunately, in the ISO 281:2007 Standard (Ref. 42), κ is based on an undefined lubricant and lubricant properties, and an undisclosed composite surface finish, σ. However, from the standard where Λ can be calculated:

$$\kappa \approx \Lambda^{1.3} \tag{17}$$

An approximate correlation can be established between filter ratings and the contamination levels. For example, BFPA/P5:1999 (Ref. 56) correlates ISO contamination levels with $\beta_x = 75$ and $\beta_x = 200$ filter ratings. Using a similar approach, we correlated filter ratings at $\beta_{x(c)} = 1{,}000$ per ISO 16889:2008 (Ref. 36) to the contamination levels listed in ISO 281:2007 (Ref. 42). These contamination levels and filter ratings, together with the life factor e_C, are shown in Table 5. Also listed are the Needelman-Zaretsky filter life factors at $\beta_{x(c)} = 1{,}000$,

Table 5 Comparison of Effect of Oil Filter Rating on Rolling-Element Bearing Fatigue Life Factor (*Lf*) Between ISO 281–2007 (Ref. 42) and Needelman- Zaretsky, Equations 11 and 14

Level of Contamination[a]	Filter Rating (µm), ($\beta_{x(c)}=1{,}000$)	Contamination Life Factor, e_C[a]		Needelman-Zaretsky Filter Life Factor, LF	
		$D_p<100$ mm	$D_p>100$ mm	Roller Bearing	Ball Bearing
Extreme cleanliness (particle size of the order of lubricant film thickness; laboratory conditions)	≤4	1	1	3.5	1.8
High cleanliness (oil filtered through extremely fine filter; conditions typical of bearing greased for life and sealed)	5–9	0.8–0.6	0.9–0.8	3.5–1.3	1.8–1.2
Normal cleanliness (oil filtered through fine filter; conditions typical of bearings greased for life and shielded)	10–14	0.6–0.5	0.8–0.6	1.2–0.9	1.1–1.0
Slight contamination (slight contamination in lubricant)	15–24	0.5–0.3	0.6–0.4	0.9–0.6	1.0–0.8
Typical contamination (conditions typical of bearings without integral seals; coarse filtering; wear particles and ingress from surroundings)	25–34	0.3–0.1	0.4–0.2	0.6–0.5	0.8–0.7
Severe contamination (bearing environment heavily contaminated and bearing arrangement with inadequate sealing)	>35	0.1–0	0.1–0	0.5	0.7–0.5
Very severe contamination	None	0	0	0.5	0.5

from Equations 11 and 14 for comparison purposes.

The contamination life factors e_C from ISO 281:2007 (Ref. 42) are differentiated by both filter rating and bearing pitch diameter. The Needelman-Zaretsky life factor equations do not differentiate based on bearing size but do distinguish between ball and roller bearing types (Ref. 31).

Although we consider the effect of Λ on rolling-element fatigue life independent of and separate from the filter life factors, the Needelman-Zaretsky life factors (Ref. 31) are normalized at $\Lambda \approx 1.1$ and $LF=1$.

The contamination life factor e_C from Table 5 can be used to adjust the calculated bearing L_{10} or catalogue life to account for filtration level in the lubricant system; these results are shown in Table 6. The values of e_C for $D_p<100$ mm were used. For the data of Bhachu et al (Refs. 12–13), $\kappa=2.8$. For the data Loewenthal et al (Refs. 9–11), $\kappa=4.7$. For each respective set of data the EHD film thickness was assumed by us to remain unchanged. Hence the effect of Λ and/or κ was not factored into the predicted lives shown in Table 6.

The results from the Bhachu et al (Refs. 12–13), and Loewenthal et al (Refs. 9 and 11), suggest the following: 1) for filtration levels between 4 and 34 µm at $\beta_{x(c)}=1{,}000$, representing "extreme cleanliness" to "typical contamination," the ISO 281:2007 Standard (Ref. 42) provides a reasonable qualitative estimate of the effect of particle damage on rolling bearing life; and 2) at conditions of severe contamination and above where the filter ratings are ≥35 µm at $\beta_{x(c)}=1{,}000$, the ISO 281:2007 Standard (Ref. 32) correlates with the Bhachu et al results. In contrast, the ISO 281 is conservative compared to the Loewenthal et al test results. This may be attributed to the use in the Loewenthal et al tests of a contaminant based on carbon-graphite particles that may act as a solid lubricant

Table 6 Comparison of Predicted Rolling Bearing Fatigue Lives Based on Contaminant Life Factor, ec, From Table 5

Level of Contamination (see Table 5)	Test Filter Rating $\beta_{x(c)}=1{,}000$ ($\beta_x=200$) (µm)	L_{10} Life, Inner-Race Revolutions	Predicted L_{10} Life Based on Contaminant Life Factor e_C 10^6 Inner-Race Revolutions
Roller bearing data from Bhachu, et al. (12) and Sayres and Macphearson (13), Table 2			
High cleanliness	6.5 (2.5)	6.5×10^6	$\sim9\times10^6$
High cleanliness	7[a] (3)	8.0[a]	8[a]
Normal cleanliness	10 (6)	4.5	~6
Typical contamination	32 (25)	2.5	~3
Severe contamination	46 (40)	1.5	~1
Ball bearing data from Loewenthal, et al. (9)–(11), Table 3			
High cleanliness	7[a] (3)	455×10^6[a]	455×10^6[a]
Severe contamination	36 (30)	535	~56
Severe contamination	56 (49)	330	~56
Severe contamination	116 (105)	403[b]	~56

a) Normalized to 7-µm filter rating where $\beta_{x(c)}=1{,}000$ for high cleanliness level of contamination.
b) See Table 3. Test series was suspended after 448 test hours on each of the test bearings because of excessive wear. No fatigue failures were encountered.

while the ISO 281 life ratings are primarily concerned with common hard steel contamination that can be found in industrial gearboxes and used by Bhachu et al in their tests.

Gabelli, Morales-Espejel, and Ioannides (Ref. 20) provide a discussion of the theoretical basis for the calculation of the contamination factor e_C that correlates with the curves of contaminant life factors vs. κ values presented in the ISO 281:2007 Standard (Ref. 42) and the contaminant life factors presented in Table 5 that are from the standard. According to Gabelli, Morales-Espejel, and Ioannides (Ref. 20), the following variables should apply in determining a contamination factor e_C: (Ref. 1) mean bearing (pitch) diameter; (Ref. 2) level of contamination (filter size); and (Ref. 3) lubrication rating of the bearing (κ value). Gabelli, Morales-Espejel, and Ioannides reduce the variables by the elimination of the fatigue limit. Their theoretical results were similar to those in Annex A of the ISO 281:2007 Standard (Ref. 42).

There is an issue as to whether the EHD film thickness or Λ (κ value) mitigates the negative effect of lubricant contamination on rolling-element fatigue life. That is, is the effect of contamination on bearing life less severe with increasing film thickness? In order to benchmark their analysis, Gabelli, Morales-Espejel, and Ioannides (Ref. 20) presented endurance data of 172 bearing population samples obtained over several years comprising 14 types and sizes of rolling-element bearings. It was reported by them that "Each bearing sample is normally formed of a group of 30 bearings; several thousand bearings were endurance tested for this set of experimental results."

The Gabelli, Morales-Espejel, and Ioannides data for Λ values varying from 0.4–2.9 (κ = 0.3–4) comprises three contamination levels. They classified their contamination conditions as follows:

1. The first contamination condition was classified as their "standard cleanliness tests." The filtration level at $\beta_{x(c)} = 1{,}000$ was ≤ 7 µm and their range for e_C varied from 0.8–1, or equivalent to "high cleanliness" in Table 5.
2. The second contamination condition was classified as "slight contamination," or equivalent to $\beta_{x(c)} = 1{,}000$ filter range of 15–24 µm in Table 5. They reported that under the given test conditions the expected contamination (life) factors e_C can range from 0.3–0.5; their actual life data showed e_C values that ranged from 0.1–0.5.
3. The third contamination condition was classified as "typical to severe contamination." From Table 5 this would comprise $\beta_{x(c)} = 1{,}000$ filter range of 25 µm or greater — although no filters appear to have been used in this test series. Their actual life data showed e_C values that ranged from 0.01–0.3.

The Gabelli, Morales-Espejel, and Ioannides experimental data, if statistically significant, shows a relation between the contaminant life factors e_C and κ — as presented in Annex A of the ISO 281:2007 Standard (Ref. 42).

In 1985 Lorosch (Ref. 14) reported that "The influence of contaminants is great with small bearings, and decreases with increasing bearing size. Consequently large bearings have a larger capacity than calculated." In other words, for a particular contamination level or oil cleanliness the effect of lubricant contamination is less severe for larger-pitch diameter bearings than for smaller-pitch diameter bearings. The effect of bearing size on the contamination factor e_C is incorporated in Annex A and Table 13 (Table 5 of this paper) of ISO 281:2007 Standard (Ref. 42). For pitch diameters ranging from 25–2,000 mm the contamination (life) factors e_C increased with increasing pitch diameter or bearing size. That is, the larger the bearing the less effect of contamination on the life of the bearing.

The pitch diameters for the bearing tests reported by Gabelli, Morales-Espejel, and Ioannides (Ref. 20) were between 25–200 mm. Their data did not show a statistical relation between the contamination levels and bearing size. They explained, "The range of bearing sizes limited the range for comparison with bearing size." However, the sizes and types of bearings were reasonably representative of those used in most rotating machinery applications.

A comparison of the data in Table 2 for the roller bearings from Bhachu et al (Refs. 12–13) — and the data for ball bearings in Table 3 from Loewenthal et al (Refs. 9–11) — suggests that the roller bearing lives are more sensitive to changes in contamination level than those of the ball bearing. This is reflected in the Needelman-Zaretsky contamination life factors in Table 5.

Summary of Results

In 1991 Needelman and Zaretsky (Ref. 31) presented a set of empirically derived equations for bearing fatigue life (adjustment) factors (*LF*) as a function of oil filter ratings (*FR*). These equations for life factors were incorporated into the reference book, *STLE Life Factors for Rolling Bearings* (Ref. 32). These equations were normalized ($LF = 1$) to a 10 µm filter rating at $\beta_x = 200$ (normal cleanliness) as it was then defined and Λ of 1.1. Over the past 20 years these life factors based on oil filtration have been used in conjunction with ANSI/ABMA (American Bearing Manufacturers Association) standards and bearing computer codes to predict rolling bearing life. Also, additional experimental studies have been made by other investigators into the relationship between rolling bearing life and the size, number, and types of particle contamination. During this time period filter ratings have also been revised and improved and are now based on particle counts calibrated to a NIST standard reference material in the ISO 11171:1999 Standard (Ref. 38). It was the objective of the work reported herein to 1) review methods and data for determining the effects of lubrication oil particle size for calculating the bearing L_{10} or catalog life; 2) experimentally correlate older and/or obsolete filter ratings and the new ISO filter ratings; 3) re-calibrate the Needelman-Zaretsky equations for determining effect of oil filtration on rolling-element bearing life to the new filter ratings; 4) relate the new filter ratings to contamination levels listed in the ISO 281:2007 Standard; and 5) compare recalibrated filter life adjustment factors to those cleanliness ratings presented in ISO 281:2007 (Ref. 42).

The following results were obtained:

1. Using two transformations, obsolete filter ratings can be converted to new ISO filter ratings and vice versa. Approximate equations relating the old $\beta_x = 200$ *FR* values with the new $\beta_{x(c)} = 200$ and $\beta_{x(c)} = 1{,}000$ *FR* values are:

   For new $\beta_{x(c)} = 200$ filter rating;

   $$FR_{\beta x(c)\,200} = 0.722\,(FR_{\beta x\,200}) + 2.97$$

   For new $\beta_{x(c)} = 1{,}000$ filter rating;

$$FR_{\beta x(c)\,1{,}000} = 0.848\,(FR_{\beta x\,200}) + 4.14$$

2. Two separate sets of life factors (LF) based on lubricant cleanliness for roller bearings and ball bearings, respectively, were derived based on the new $\beta_{x(c)} = 200$ and $\beta_{x(c)} = 1{,}000$ ISO filter ratings. These LF values are normalized to FR values of 10 µm at $\beta_{x(c)} = 200$ and 13 µm at $\beta_{x(c)} = 1{,}000$ and are independent of Λ and/or κ, loading conditions, and bearing size. These are:

   For roller bearings and new $\beta_{x(c)} = 200$ filter rating;
   $$LF \approx 3.5\,[1.39\,(FR_{\beta x(c)\,200}) - 4.11]^{-0.55}$$
   For roller bearings and new $\beta_{x(c)} = 1{,}000$ filter rating;
   $$LF \approx 3.5\,[1.18\,(FR_{\beta x(c)\,1{,}000}) - 4.88]^{-0.55}$$
   For ball bearings and new $\beta_{x(c)} = 200$ filter rating;
   $$LF \approx 1.8\,[1.39\,(FR_{\beta x(c)\,200}) - 4.11]^{-0.25}$$
   For ball bearings and new $\beta_{x(c)} = 1{,}000$ filter rating;
   $$LF \approx 1.8\,[1.18\,(FR_{\beta x(c)\,1{,}000}) - 4.88]^{-0.25}$$

3. ISO 281:2007 Standard provides a reasonable qualitative estimate of the effect of particle damage on rolling-element bearing fatigue life for filtration ratings ranging from ≤ 4 µm at $\beta_{x(c)} = 1{,}000$ (extreme cleanliness) up to $\beta_{34(c)} = 1{,}000$ (typical contamination). At conditions of severe contamination and above, where the filter ratings are ≥35 µm at $\beta_{x(c)} = 1{,}000$, the ISO 281:2007 Standard correlated with test results obtained with common hard steel contamination that can be found in industrial gearboxes. **PTE**

Acknowledgments. *The authors would like to thank the Donaldson Company for supplying the multi-pass test data used to create Figures 6 and 8 and Equations 5 and 6.*

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William (Bill) Needelman received his B.Sc. from McGill University and M.S. from Princeton in Physical Chemistry. His work career began at the Pall corporation (26 years) — his last six years there as Principal Scientist. After eight years at the Donaldson Corporation as Chief Science Advisor, Needelman began his own company — Filtration Science Solutions, Inc. His business was devoted to the R&D, design, build, modeling, laboratory testing, field testing, and evaluation of filters for a countless variety of aerospace and industrial applications. Bill's work has appeared in some 47 publications, including international standards and handbook chapters. He was also the lead author for AWEA-Recommended *Practices for Wind Turbine Gearbox Filtration*. In addition, he is an Adjunct Professor of Chemistry at SUNY-Suffolk and at the U.S. Merchant Marine Academy.

Erwin V. Zaretsky, PE is an engineering consultant to industry and government, a noted speaker and teacher, and author of more than 200 technical papers and two books. He retired as Chief Engineer/Materials and Structures at the NASA Glenn Research Center where he retains an emeritus position as Distinguished Research Associate. A 1957 graduate of the Illinois Institute of Technology in Chicago — and with a 1963 doctorate from Cleveland State University — Zaretsky is a former head of the NASA Bearing, Gearing and Transmission Section, where he was responsible for most of the NASA mechanical component research for air-breathing engines and helicopter transmissions. With over half a century of experience in mechanical engineering related to rotating machinery and tribology, Zaretsky has performed pioneering research in rolling-element fatigue, lubrication and probabilistic life prediction; his work resulted in the first successful 3 million DN bearing. Zaretsky is an Adjunct Professor at Case Western Reserve University. In 1992 he edited and co-authored the STLE (Society of Tribologists and Lubrication Engineers) book — *STLE Life Factors for Rolling Bearings* — as he had done previously, in 1997 — *Tribology for Aerospace Applications*. Zaretsky is the recipient of numerous NASA awards for his contributions to the Space Program, among which are the NASA Medal for Exceptional Engineering Achievement, the NESC Director's Award and the Astronaut's Silver Snoopy Award. In both 1999 and 2013 the STLE presented Zaretsky with the Wilber E. Deutsch Memorial Award, which honors the most outstanding paper written on the practical aspects of lubrication. In 2012 the STLE presented Zaretsky with their International Award — STLE's highest technical honor for his lifetime of contributions to the field of tribology research. He has also received four IR-00 awards. Zaretsky is a Life Fellow of both the ASME and STLE.

Global Industrial Outlook: Game Changer

Brian Langenberg

The number one question today is: are the current low oil prices a near-term or structural development?

We don't know. And neither does the U.S. Oil Sector.

- Saudi Arabia is continuing to pump; this is a geopolitical move.
- Technology has reduced cost and increased energy access — a structural fact.
- Low prices + crushing debt = bankruptcies among small producers.

Add it up and you have lower confidence and lower spending.

July 2013 — Oil @ $100: Prince Alwaweed announces that oil price structurally going down; nobody is listening.

May 2014 — Oil @ $100: Russia, Iran, ISIS behaving badly, while U.S. doing nothing. Saudis notice.

Nov 2014 — Oil @ $75: Saudi's deploy Weapon Alpha — i.e. oil production — to (take your pick) fight an economic battle against the above and — possibly — U.S. fracking.

Feb 2015 — Oil @ $50: Oil has traded more or less between $40–$60-per-barrel all year, and a 50% price cut should lead to 50% (or more) capex cuts; thus far we are seeing 25%–35% cuts. It will get worse; producer bankruptcies are coming.

We are beginning to reconsider our long-term position on oil prices. At the very least, your construction equipment, power generation and mining customers face a continuing cyclical challenge.

The structural challenge is something else. With respect to natural gas, the question is what is the true, fully loaded cost to explore, produce and transport energy sourced through hydraulic fracturing? Whatever that true cost is will likely prove the "high-end" price cap on oil longer-term and, with that, an important factor with respect to the energy sector's demand for heavy machinery over time.

Additionally, to the extent the energy boom of the last few years was a "bubble" — history shows that bubbles do not reflate.

July 28, 2013: Billionaire investor Prince Alwaleed bin Talal is reported to have written an open letter to Saudi oil minister Ali al Naimi, warning that Saudi Arabia must diversify its revenue sources because of the impact of shale gas extraction on the oil industry globally. Oil was trading between $100-$105. Again, nobody (at least whom I know of) was listening.

May 2014, Oil ~ $100: Frog Legs, Bratwurst and the Bear

Capital spending will remain stable this year, with particular strength in downstream (refining) and midstream (pipeline infrastructure), while upstream will decline perhaps (1%–3%) overall.

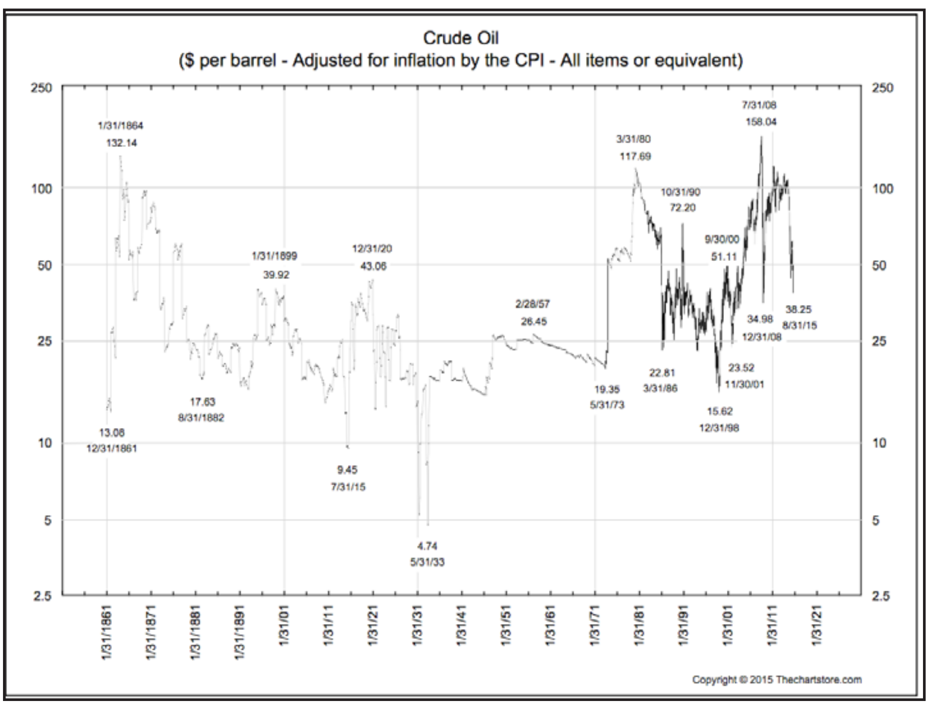
August 2014, Oil ~ $100: Das (Human) Kapital

We were fortunate to meet with the management teams of 16 major companies from offshore drillers like Transocean, to oil field services provider Dresser-Rand. We also spoke directly to Exploration & Production companies. In a nutshell this is what we are communicating:

- **Off-shore.** Poised to accelerate in 2015.
- **Mid-stream** (pipelines). Strong expansion continues to get upstream energy supplies to market.
- **Refineries**. Because of condensate export approvals, expect rising demand for new, modern, LNG/LPG tankers and infrastructure.

Our writing through August pointed out rising geopolitical issues — Russia, Ukraine in particular — and how it will drive rising European defense spending over time.

We did not anticipate that Saudi Arabia would deal with ISIS, Iran, and Russian challenges with its No. 1 weapon: oil production



Extended timeline for Oil & Gas

November 2014, Oil ~ $75: Houston, we have a problem

At this juncture it became clear the Saudis would maintain production to crush oil price. The only debate was whether they were going after Putin / Iran (our view) or U.S. fracking (a la Prince Alwaweed). Either way, we were early in saying lower oil price is bad for the sector and that 2015 oil related capital spending would be significantly lower with this passage:

"Netting it all out though, oil price down (25%) = (25%) lower industry revenue = bad for you. *Most of your customer base is claiming it will have little impact. They are wrong."*

February, 2015, Oil ~ $50: Oil slick, currency headwinds challenge growth

The general consensus is about a (25%) reduction in 2015 capital spending, which seems right ***but can worsen in 2016*** should oil price not recover.

March, 2015, Oil ~ $50: Oil slick, currency headwinds worsen

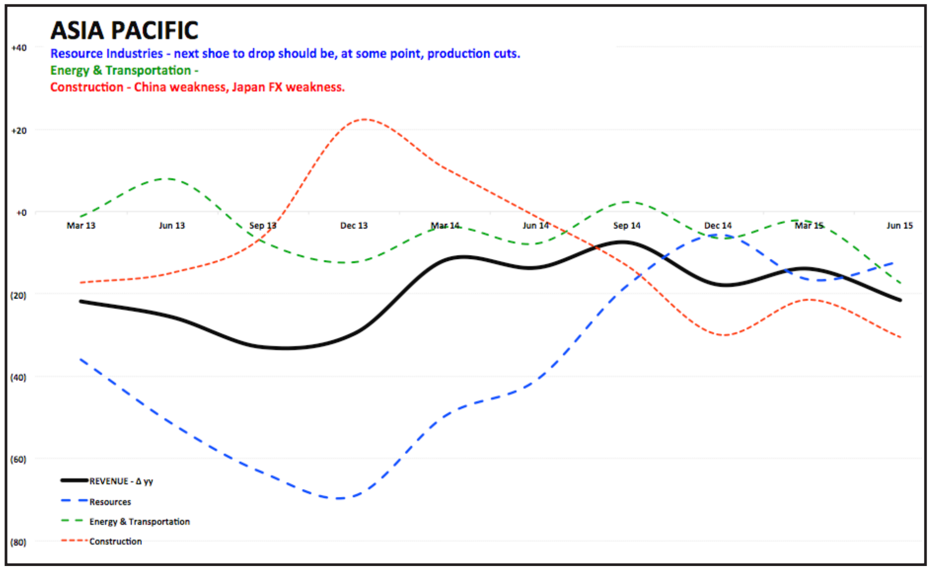
The general consensus remains for a (25%) reduction in 2015 capital spending by global oil companies, but those forecasts implicitly assume at least some recovery in oil price from curtailed exploration activity. Unfortunately, cuts in natural gas fracturing — even 4%–6% in a week — do not boost oil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

One month ago WTI (West Texas Intermediate) was at a "depressed" $52. Now we are looking at $45; expect more capital spending cuts.

May, 2015, Oil ~ $60: Slow growth ahead; farm belt can't help

The Saudis continue to step on the gas, driving and keeping prices low; and as a result North American capital spending continues to decline.

Huge capex cuts in upstream exploration and production drove a number of weak first quarter results for industrial companies, and we see no respite for the next two to four quarters before stabilizing at a lower spending level.



July, 2015, Oil ~ $50: Kind of sluggish

Oil had rallied from about $50 to $60 over the past month (I like round numbers) but have since round tripped. Expect further, deeper capital spending cuts in the U.S. oil sector to continue affecting demand for large capital equipment.

Assuming Iranian oil comes back into the global market, prices will be further pressured. We continue to see negative comps for the next two to four quarters before stabilizing at lower spending levels.

Opportunities remain. Low commodity prices hurt commodity producers and their equipment suppliers — but also benefit these major sectors:

- Aerospace
- Auto production
- Non-residential construction
- Residential construction
- Consumer discretionary

Construction equipment is getting hit near-term (equipment flowing out of L&G sector), but on the flip side lower raw material prices make large construction projects more affordable.

We anticipate no substantive improvement in manufacturing activity. Not in the U.S., nor internationally. Headwinds include oil price and commodities in general (down), and the U.S. dollar (up). Expect further, deeper capital spending cuts in the U.S. oil sector to continue affecting demand for large capital equipment.

China's stock market meltdown is their problem, not our problem. China's stock market rocketed upward on fundamentals — not economic acceleration — and the opposite is now true. In fact, China has proven a weak market for commodities and capital equipment for some time. Shown here is Caterpillar's "core" revenue trend for Asia Pacific over the past two and a half years:

Construction equipment weakness reflects excess supply and possibly market share loss in China; Resource Industries ties more closely to Australia/NZ but indirectly also reflects China.

The weakness is broad-based; Rockwell Automation core Asia Pacific revenue has remained at or below 5 percent since June 2014 and United Technologies' Otis Elevator unit has reported flat or down China elevator orders for the last five quarters. The point is that the weakness is old news, while the headlines are about their capital market excesses.

The rest of the world is hardly doing great — updated outlook for key geographic regions:

U.S. remains the safe, modest growth bet. Weakness abounds in commodity-related sectors — e.g. oil, coal, and farm equipment — but fundamentals remain positive for non-residential construction, consumer durables (auto, housing) boosted by gradually improving employment.

Europe. The weaker Euro benefits exports, lower commodity prices and slowing China growth are headwinds. Life will go on. Modest growth will continue.

Middle East. Right now, Saudis are investing to keep their mature fields working; incremental growth in defense spending is likely.

Latin America. Mexico continues to grow, and capital investment in the auto and aerospace sectors remains strong. Brazil, Argentina — much of the region — is toast.

China. No longer a huge growth market for outside players. Costs, business risk, and military belligerence are up and the mask is off. I do expect the nation's economy will grow 5–7 percent — but with greater wealth capture from domestic players. It will still be a good place to do business, but not a *great* place to do business.

The End Market Picture is Likewise Mixed

Oil & Gas. We believe it is going to get worse.

Mining. Awful, plunging toward hideous.

Power generation. Sounds like GE will be able to close on Alstom deal. Devil is in the details regarding what they must to concede. Supplier memo: Have your helmet on and the chinstrap fastened for that upcoming "partnership" discussion.

Transportation infrastructure. More stability through 2016–2017 with, perhaps, modest growth. I remain convinced that lower oil prices will lessen the growth profile for oil shipped by rail. Conversely, a new President in 2017, low commodity prices (steel, cement, energy), and dilapidated infrastructure would, you think, be a growth catalyst. Let us hope.

Machinery. Everything ex-truck stinks. Construction equipment is soft, agriculture will remain weak and mining is hideous. Only silver lining — and not enough to off-set — is growth in non-residential and residential construction.

Consumer (auto, appliances). Same story; auto benefitting from old cars, improving employment and capital investment in Mexico. Residential construction growth should help appliances.

Aerospace / Defense. Global commercial aircraft demand is rock solid driven by economic growth, low fuel prices and strong capital markets. Cargo is also picking up. One offset is that Boeing 747 orders remain weak and there is chatter about the program's future as the world demands more narrow body and the A380 makes inroads. Defense spending has troughed in the U.S. and international growth strikes us as likely though with little benefit to the U.S. industrial base.

Focus Company: AGCO (ACGO)

AGCO is a distant No. 2 in North American farm equipment behind John Deere and is far more leveraged to global conditions. As such weaker U.S. demand is more of an annoyance than huge challenge. Operationally, the company has succeeded in driving North American margin to 10 percent or better in two of the last three years.

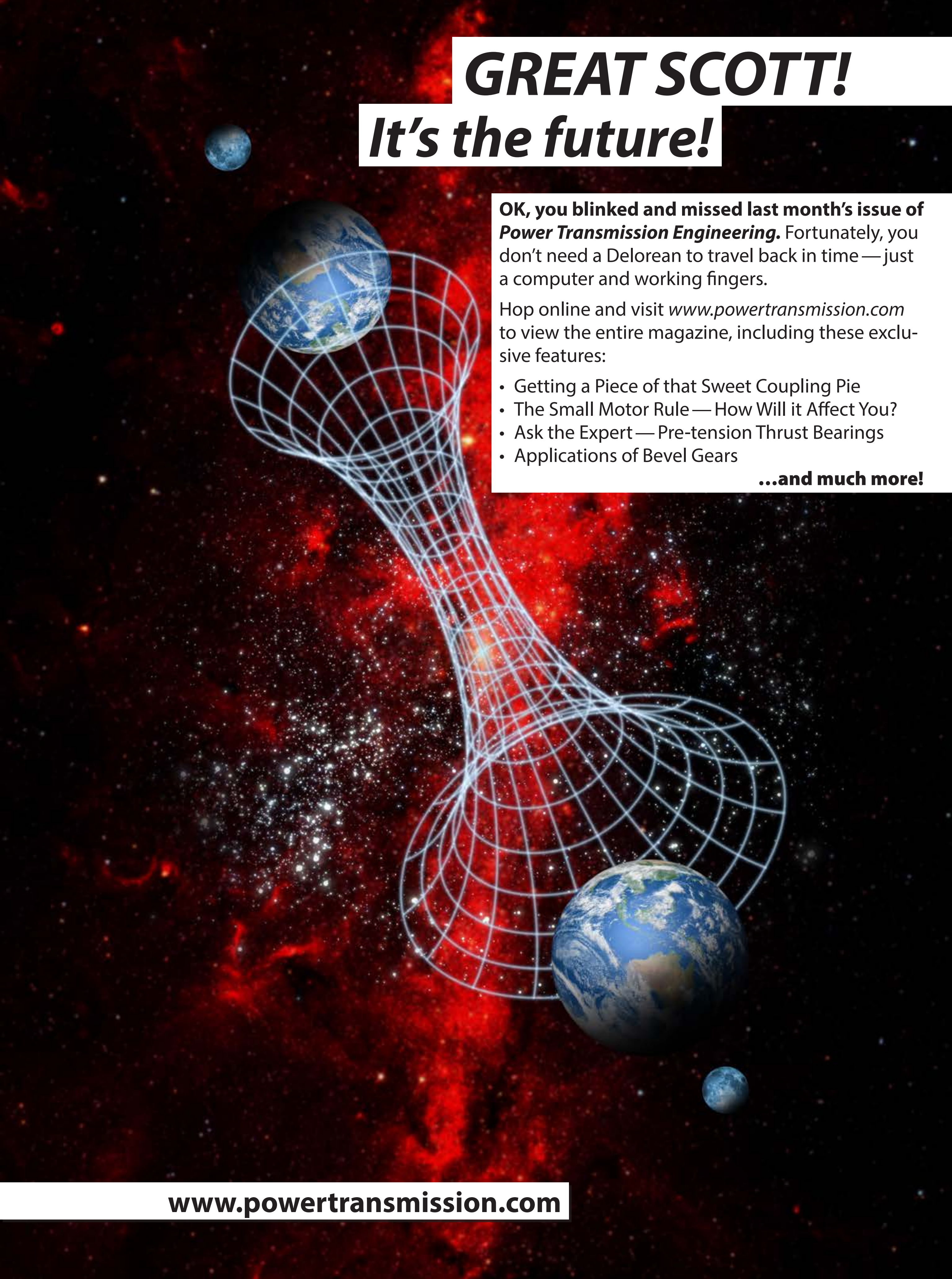
	2012	2013	2014
EAME	$5,074	$5,482	$5,158
	475	558	500
N. America	$2,584	$2,758	$2,414
	260	326	219
S. America	$1,856	$2,040	$1,663
	162	213	134

Still, weak conditions are also a challenge for AGCO, particularly as S. American operations are suffering and the strong dollar has hurt currency translation back to the company.

We just returned from meetings at the Farm Progress Show in Decatur, IL, including a presentation and booth tour by Bob Crain, SVP & general manager for the Americas. There was silver lining, given low crop prices and excess dealer inventory. The key takeaway was that while dealer inventory was down (12 percent) y/y as of end of July with production cuts and targeted marketing programs planned to make further reduction.

Finally, whole farm sector is under the weather — and in North America that includes Deere, AGCO and CNH Industrial — all of which cite excess inventory and lower used prices. None see a rebound in 2016. And neither do we. **PTE**


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

BECOMES NEW EXECUTIVE BOARD MEMBER OF RENK AG

Renk AG's supervisory board recently appointed **Christian Hammel** as executive board member for production and administration. Hammel is currently a member of the executive board responsible for finance at MAN Truck & Bus Österreich AG. After studying business administration at the University of Regensburg, he held various management positions at Sixt AG and at MAN Truck & Bus AG. He has been working at MAN since 2004.

Hammel will succeed Ulrich Sauter, who left company for health reasons. He will still be assisting Renk in an advisory capacity for a transition period. Sauter has been on the executive board of Renk AG for almost 20 years.

"The supervisory board would like to thank Mr. Sauter for his excellent work at Renk AG and for his great service to the company. He joined Renk in 1995 amidst difficult times and has since then decisively contributed to Renk's extremely successful development. Renk's success story is not least based on Mr. Sauter's ability of taking farsighted decisions," said Dr. Bartölke, chair of the supervisory board.

Pacamor Kubar Beatings

PURCHASED BY TWO COMPANY EXECUTIVES

Pacamor Kubar Bearings (PKB) was recently been purchased by Edward M. Osta, the company's current executive vice president, and Stephen A. Angrisano, the company's current chief financial officer.

The agreement calls for the sale of the entire business, including all the factory capabilities and certifications. The company will continue to reside in its 13,250 square foot factory, clean room assembly area, and office area. The transition will be seamless, as all employees, machinery, certifications, as well as current skills and capabilities remain in place.

The purchase price was not disclosed.

"We were very pleased with the strong interest in Pacamor Kubar Bearings from highly reputable manufactures and other interested investors and received several highly competitive proposals," said Augustine Sperrazza, president and CEO of S/N Precision Enterprises.

PKB
PACAMOR KUBAR BEARINGS

With the sale of the company, Sperrazza will retire. Sperrazza said that Osta and Angrisano are in the best position to protect and enhance the value and reputation of Pacamor Kubar Bearings (PKB).

"The future growth and expansion capabilities for this American owned and operated company have always been important considerations for us and we believe that this is the best decision for the customers, employees, and creditors," he said.

Haydon Kerk

INTRODUCES CATALYST MOTION GROUP AS A SUPPLIER OF INTEGRATED MOTION SYSTEMS

Haydon Kerk Motion Solutions recently introduced the Catalyst Motion Group as a supplier of integrated motion systems. Catalyst offers custom high-precision linear and rotary motion systems at any level of integration.

According to Haydon Kerk, Catalyst understands the many facets of motion system technology and is focused on harnessing that knowledge to develop next-level application specific solutions. With global resources and an expertise in manufacturing and engineering that is related specifically to linear and rotary motion systems, Catalyst is positioned to provide full-service integrated motion system development, design, test and manufacture.

Motion Industries Florida Location

ACHIEVES TIMKEN BEARING CERTIFIED SHOP GOLD LEVEL

Motion Industries recently announced that its service center in Pensacola, FL has recently achieved Timken Bearing Certified Shop "Gold" Level for 2015. Associates from The Timken Company audited the shop and presented certification signifying the shop adheres to Timken standards for bearing removal and handling, cleaning and inspection, analysis of bearing damage, bearing setting and adjustment, and bearing storage.

"We congratulate Motion Industries Pensacola Service Shop for its commitment to meeting the high standards necessary to achieve the status of Timken Bearing Certification," said David A. Novak, Jr., Timken director of service engineering and strategic projects.

As part of this certification process, Timken trained Motion Pensacola shop personnel in the proper cleaning, inspection and storage of wheel bearings. By training its associates in the proper techniques and then auditing and certifying the process, Motion has indicated its confidence that the bearings returned to service are properly cleaned and inspected and will deliver their full life potential.

"The outstanding efforts of our Pensacola team reflect Motion's ongoing commitment to quality service," said Randy Breaux, Motion Industries' senior vice president of southern U.S. operations, corporate marketing and strategic planning. "Their hard work, focus, and dedication resulted in this significant achievement."

NFPA

REPORTS FLUID POWER INDUSTRY FOR MAY DOWN 8.5 PERCENT FROM PREVIOUS MONTH

The latest data published by the National Fluid Power Association shows industry shipments of fluid power products for May 2015 decreased 14.2 percent compared to May 2014, and decreased 8.5 percent when compared to the previous month.

Mobile hydraulic, industrial hydraulic, and pneumatic shipments decreased in May 2015 when compared to May 2014. Mobile hydraulic, industrial hydraulic, and pneumatic shipments also decreased when compared to last month. These charts are drawn from data collected from more than 80 manufacturers of fluid power products by NFPA's Confidential Shipment Statistics (CSS) program. Much more information is available to NFPA members, which allows them to better understand trends and anticipate change in their market and the customer markets they serve.

October 4–7—Euro PM2015 Congress & Exhibition Reims Congress Centre, Reims, France. Europe's annual powder metallurgy congress and exhibition, organized and sponsored by the European Powder Metallurgy Association, will return to France in 2015. The combination of a world class technical program and state-of-the art exhibition will provide the ideal networking opportunity for suppliers, producers and end-users. The program of plenary and keynote addresses, oral and poster presentations and special interest seminars will focus on: additive Manufacturing; hard materials and diamond tools; hot isostatic pressing; new materials and applications; and more. Alongside the technical sessions the Euro PM2015 Exhibition will be an excellent opportunity for international suppliers to the PM industry to network with new and existing customers from the powder metallurgy and associated sectors. For more information, visit *www.europm2015.com*.

October 20–22—Gear Expo 2015 Cobo Center, Detroit, MI. For more than two decades, power transmission professionals—including CEOs, owners, presidents, engineers, marketing and sales managers, consultants and other executives have come to Gear Expo to learn the latest industry information and see firsthand technology, products, and services that help them expand and streamline their business. Attendees represent a variety of industries including off-highway, industrial applications, automotive, and oil and gas as well as aerospace, agriculture and construction. They come from around the United States, international manufacturing hubs, and emerging markets to conduct profitable business transactions and collaborate on the innovations that make their operations more streamlined. Exhibitors have the opportunity to meet face-to-face with attendees and other exhibitors and will display more than 750,000 pounds of machinery on the show floor. For more information, visit *www.gearexpo.com*.

October 21–24—PTDA 2015 Industry Summit Hilton Chicago, Chicago, IL. More than 550 delegates, representing 240 PTDA distributor and manufacturer member companies in the power transmission/motion control industry are expected for cross-channel networking, shared learning and collaborative experiences at this year's Industry Summit, titled "Spark A Movement: Inspire. Motivate. Lead." The PTDA 2015 Industry Summit is starting a movement of its own by offering three keynotes in one with a session called IML Talks, playing off the theme, Inspire. Motivate. Lead. Speakers include: Phil Hansen, an artist with a debilitating physical limitation; Ryan Estis, a former chief strategy officer for a global advertising agency; and Scott Klososky, a technologist who returns to the Industry Summit after receiving in 2013 the highest ratings of any other speaker. Another returning speaker is Alan Beaulieu, who delivers his uncannily accurate economic forecast as the Closing Keynote.The signature event of the PTDA Industry Summit is the Manufacturer-Distributor Idea Exchange (MD-IDEX). MD-IDEX is a time- and cost-effective forum to bring together distributor and manufacturer executives for high-level discussions on market strategies and issues to mutually benefit each other and end users. Distributor and manufacturer members alike laud MD-IDEX as one of the most organized face-to-face cross channel business meeting programs.Additional cross-channel networking opportunities abound at the PTDA 2015 Industry Summit. From receptions and networking lunches, to a Roaring 20's theme party at Chicago's Navy Pier, participants will find takeaways that can be put to use right away to achieve success going forward. For more information visit *ptda.org/IndustrySummit*.

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

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

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

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

ADVERTISING INDEX

The Museum of Unworkable Devices

The strange place where broken is beautiful

Jack Mc Guinn, Senior Editor

Albert Einstein, it is said, defined insanity as "doing the same thing over and over again, expecting a different result." My favorite variation (*Anon*) on those words of wisdom is "someone who hits one's self in the head with a hammer until it stops hurting."

Somewhere between those two worldviews resides the plucky spirit behind The Museum of Unworkable Devices (*www.lhup.edu/~dsimanek/museum/unwork.htm*).

This museum, quite simply (quoting from its Website) — "is a celebration of fascinating devices that don't work." Continuing, in downright humorous, if snarky, fashion — it "houses" "diverse examples of the perverse genius of inventors who (*I love this part*) refused to let their thinking become intimidated by the laws of nature — remaining optimistic in the face of repeated failures."

While *PTE* is indeed a magazine devoted to power transmission, we're thrilled to report that there is nevertheless no shortage of ill-fated, once-and-no-future gizmos "on display" (it's virtual, people) at the "museum" whose creator (more later) seriously intended to help move earth if not heaven as well — *if only they had worked*. (And somewhere, Rube Goldberg must be grinning his ass off.) Popular at this repository are "intricate perpetual motion machines that have remained *steadfastly unmoving* (my ital) since their inception." It is also good to know that, like all museums, the Museum of Unworkable Devices considers itself a "work in progress" — signaling a recognition that Man's talent for the stupid continues to evolve, unchecked.

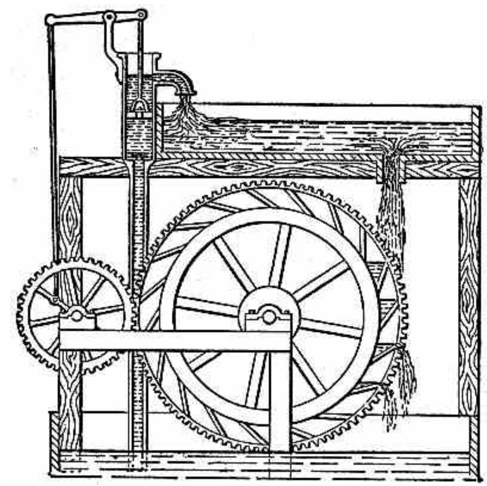
Let us read, for example, the museum listing for this water wheel/pump "system", cited from a 1927 tome by engineer/inventor of some note Gardner D. Hiscox — *Mechanical Appliances and Novelties of Construction*:

*"**Water wheel and pump** —* a principle so often employed for the production of self-moving machines that it ranks next to that of perpetual eccentric weights in its delusive power upon the minds of inventors. The attempt to compel a water wheel to raise the water that drives it is in one form or other perpetually recurring in devices upon which our counsel and opinion are sought.

"The worst of the matter is that in most cases our advice to drop such absurd projects is received as evidence of want of sagacity and knowledge, *a*nd our would-be client becomes the dupe of some not over-conscientious patent agent, who pockets his fees and laughs in his sleeve at the greenness of the applicant.

"The device illustrated is one submitted by one of those enthusiastic individuals, who, without understanding the first principles of mechanics, believes he is about to revolutionize the industry of the world by his grand discovery; and as honor, and not pecuniary reward, is his object, he seeks to make public his invention through the wide circulation of some journal. He is quite willing we should adversely criticize the device, because its merits are so great that no amount of skepticism resulting from our blind prejudice can, he thinks, influence candid minds against a principle so obviously sound and sublimely simple.

"Even if you could completely eliminate friction and viscosity in the pump and gears, this device requires the pump to lift water above the open reservoir, that is, higher than the pressure head of the reservoir, requiring more work than one can gain from the falling water. *Even if that minor (!) flaw were fixed, the work done in carrying the water around a closed loop is zero.* Exercise for the student: If viscosity and friction were zero, and this device were primed and started running, how and when would it come to a stop?"



"The device illustrated is one submitted by one of those enthusiastic individuals, who, without understanding the first principles of mechanics, believes he is about to revolutionize the industry of the world by his grand discovery." (Text, illustration from Mechanical Appliances and Novelties of Construction, Gardner D. Hiscox, M.E., 1927, Norman W. Henley Publ. Co.)

Okay then! In closing, let me mention the list (below; go to the site for the links) of various "galleries" available for viewing — don't touch! — at the museum. Just the titles for some of these tell you all you need to know as to what they're all about, such as: "Whatever Were They Thinking?" — (*deep* thoughts, obviously); "The Gallery of Ingenious, but Impractical Devices" — (*just* what our economy needs); "The Basement Mechanic's Guide to Building Perpetual Motion Machines" — (there is *no way* I'm hanging out with some "*basement mechanic*" who is *building* "perpetual motion machines, *no way, no how;* "The Basement Mechanic's Guide to Testing Perpetual Motion Machines" — (*see above*!!)

So take a day trip in your PJs or BVDs. Pack a lunch. Chill some beverage. Pull up a chair to your favorite desktop or laptop, and prepare to immerse your brain in a totally useless but decidedly diverting pastime. *(Note: The Museum of Unworkable Devices is the 1994 brainchild of Dr. Donald E. Simanek, Professor Emeritus, Lock Haven University of Pennsylvania. Space did not permit a fair telling of his story. We hope to address that with the next Power Play in the October issue of PTE.)* **PTE**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