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DECEMBER 2017

Buyer's Guide

- Linear Motion for Large-Scale CNC Cutters
- Putting the Brakes on a Transmission Test Stand
- Gear Expo Highlights



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DECEMBER 2017

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Inset photo by David Ropinski





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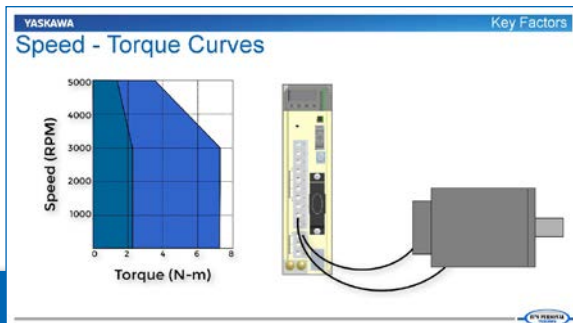
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**PTE Videos
Yaskawa**

This video from Yaskawa examines servomotor sizing concepts using peak torque, RMS torque, inertia ratio and speed.

www.powertransmission.com/videos/Servo-Motor-Sizing-Basics-Part-1---Core-Concepts-/

Schaeffler Takes on the Skills Gap

Schaeffler USA has about 100 apprentices at its Wooster plant. The “skills gap” hasn’t been a problem for the company. With ApprenticeOhio, they can design training programs that meet their needs. Learn more at:

www.powertransmission.com/videos/ApprenticeOhio-at-Schaeffler-/

Editor’s Choice

Check out our Editor’s Choice blog to find the latest case studies, product features and technical article involving mechanical power transmission. Here are some of the latest entries:

A Look at Belt, Chain and Gear Drive Technology

[\(www.powertransmission.com/blog/a-look-at-belt-chain-and-gear-drive-technology/\)](http://www.powertransmission.com/blog/a-look-at-belt-chain-and-gear-drive-technology/)

Amsted Rail Taps Jorgensen for Conveyor Belt Upgrades

[\(www.powertransmission.com/blog/amsted-rail-taps-jorgensen-for-conveyor-belt-upgrades/\)](http://www.powertransmission.com/blog/amsted-rail-taps-jorgensen-for-conveyor-belt-upgrades/)



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OFFSET SHAFTS. SOLVED.

Offset Couplings from Zero-Max reduce space requirements for parallel offset shafts in large system applications. These specialized couplings provide machine designers with an important option for reducing overall machine size and footprint.

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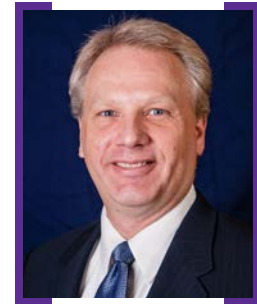
Schmidt Offset Couplings can be mounted to shaft hubs or directly to existing machine flanges. They are available for shaft displacements of 0.156 inches to 17.29 inches and torque capacities from 55 to 459,000 inch-pounds. Many design configurations are available including specials.



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10 Years and Counting



Just over 10 years ago, we started publishing *Power Transmission Engineering*, and I'm proud to say we're still going strong.

When we founded this magazine, it was designed to serve an unfilled niche. Our goal was to be the premier information resource for designers, buyers and users of machinery with power transmission components. We wanted a magazine that focused on gears and gearboxes, bearings and clutches, electric motors and brakes, along with all the other critical parts and systems that make things go.

It hasn't always been easy. Our writers and editors have to prepare content for design engineers working with the latest software as well as maintenance technicians working with screwdrivers and wrenches. We cover the latest R&D right alongside the basics. And we cover virtually every industry with moving parts, from steel mills to dental drills.

But despite those challenges, we think we've found our niche. Nobody else does what we do.

But we couldn't do it without some help. I'd like to extend a special thanks to all of our advertisers, both current and past, for their support. Without them, we would have never made it this far. In fact, I'd like to call special attention to B&R Machine and Gear, Circle Gear, Designatronics, Forest City Gear and R+W America, all of whom advertised in the very first issue, all of whom are advertising in *this* issue, and all of whom have consistently supported us throughout the past 10 years. Thanks, guys.

I'd also like to thank you, the readers. For those of you who have been reading *Power Transmission Engineering* since the beginning, thank you for sticking with us. For those of you who have only recently subscribed, we welcome you and hope you've found enough value to keep reading for another 10 years.



After 10 years of publishing the magazine, we've compiled quite a lot of content. And whether you've been with us since the beginning or you've just recently subscribed, you can always go back and revisit our past issues. Most of the articles in our archive are just as valid today as they were when they were first published, and they're freely available in the PTE library at www.powertransmission.com.

Speaking of the website, I should mention that although *Power Transmission Engineering* has existed since 2007, we actually began serving the marketplace way back in 1997 when we first launched powertransmission.com. So when you think about it, our 10th anniversary is actually our 20th.

In the early days, powertransmission.com was mainly a buyers guide. We took pride in providing the most comprehensive online directory of suppliers available anywhere. We provided detailed information about suppliers of gears, bearings, motors, and related products that couldn't be found anywhere else.

And that's still true today. Powertransmission.com remains your best resource for finding suppliers of gears, bearings, motors, couplings, clutches, and other power transmission and motion control components. Connecting our audience with potential suppliers remains a core of what we do here. That's why we're also proud to present, in this issue, the printed version of our 2017 Buyers Guide (beginning on page 34). Although there's still a *lot* more information available on the website, we provided this handy guide to help you find what you need, quickly and easily.

We hope you enjoy the issue, just as we hope you've enjoyed reading *Power Transmission Engineering* over the past 10 years. But most of all, we hope you keep reading.

Randy Stott

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SKF

LAUNCHES RE-ENGINEERED MOUNTED BALL BEARING UNITS

SKF has launched a newly re-engineered product line of mounted ball bearing units with industry-standard inch series cast iron housings in several designs, high-performance SKF inch or metric insert bearings, and several shaft-locking methods. These mounted bearing units will suit a wide range of light- and medium-duty industrial applications, including conveyor systems, fans and air-handling equipment, and similar machinery across industries.

Four robust cast iron housing types have been introduced: pillow block units (P2B, P2BL, and P2BM series), 4-bolt square flange units (F4B and F4BM series), 2-bolt oval flange units (F2B series), and tapped base units (P2BT series). All housings conform to ABMA (American Bearing Manufacturers Association) inch series dimensional standards and their solid construction adds strength and promotes stability in service. The housings in an assortment of bore sizes are supplied pre-lubricated and with grease fittings for re-lubrication. Housings easily slip fit onto a shaft. Set screw, eccentric, or concentric shaft-locking mechanisms can be specified.

Unlike competing versions in the marketplace, these mounted units exclusively integrate highly engineered, high-quality SKF brand insert ball bearings (inch or metric). Their race-



ways have been ground and honed to provide precision, quiet running, and higher speed capability. A molded glass fiber reinforced polyamide cage adds durability. In addition, the outside diameter of the bearing and the inside diameter of the housing have been uniquely sphered, allowing the bearing to swivel within the housing and compensate for any errors in initial alignment during installation.

These mounted ball bearing units

ultimately offer ideal solutions accommodating diverse mounting surfaces, load requirements, shaft sizes, and dimensional requirements, whether in OEM or aftermarket applications. Custom mounted bearing products can be developed.

For more information:

SKF USA Inc.
Phone: (800) 440-4753
www.skfusa.com

R+W

OFFERS MID-SIZE HEAVY-DUTY SAFETY COUPLINGS

R+W has recently introduced two new smaller STN series safety couplings with conical clamping hubs, to bring heavy duty industrial style overload



protection into midsize applications. The two new sizes, 2 and 5, cover disengagement torques ranging from 200 to 5,000 Nm, and shaft diameters from 45 to 80 mm (1.750" to 3.125").

Taking advantage of the well-proven ball-detent safety element system, the driving and driven ends separate within milliseconds of a torque overload, releasing the motor inertia from the driveline, and reducing expensive repairs and costly downtime. The driveline is free to coast to a stop after disengagement, and re-engagement simply requires that a force be applied to the back sides of the plunger modules,

either with a mallet or pry bar. Disengagement torque values are adjustable in the field and multiple ranges are available depending on the module configuration.

These safety couplings are designed with an output flange that can connect to a pulley or sprocket. Shaft to shaft connection methods are also available with either disc, elastomer, bellows, or gear coupling designs.

For more information:

R+W America
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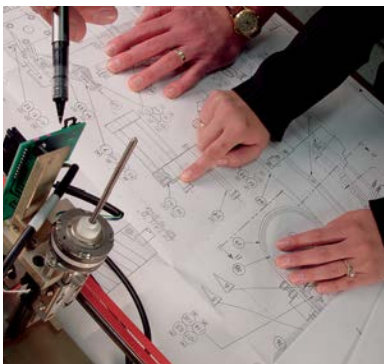
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Brecoflex

INTRODUCES OPEN-ENDED TIMING BELT

Brecoflex CO., L.L.C. is pleased to introduce the newest belt option in its move-series, the AT10move-M. This open-ended version joins the successful BFXmove truly endless timing belt that was launched in 2015. The move-series represents the next generation of timing belts, offering up to 75 percent higher stiffness and 30 percent higher transmittable force than the standard AT10 belt.

The advantages of AT10move open-ended belts include:

- 75% higher stiffness and tensile strength
- 30% higher transmittable force
- permits reducing the belt width by one standard size
- narrower drive reduces drive inertia and noise
- reduced wear and increased service life

In linear drive applications, the high tensile strength and high stiffness of the “move tension members” ensure faster settling and more accurate positioning with a very high degree of preci-

sion. The high load-bearing capacity of the tension members, optimized tooth shape and low-friction coefficient of the belt coating provides advantages in applications where long service life and low friction are important. These properties proved superior for BFXmove, and will make AT10move-M the most outstanding option for linear drives.

Move-series AT10-M is currently available in 25 mm, 32 mm, 50 mm, 75 mm and 100 mm widths with galvanized steel tension members.

The diagram below demonstrates the belt stiffness of the AT10move compared to other common belt sizes. One can clearly see that the AT10move opened ended belt provides a higher stiffness in a smaller belt, enabling the machinery to run efficiently with a more compact belt.

For more information:

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Nexen

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Nexen Group, Inc. offers three models of BD (brute disk) caliper brakes: air actuated, spring actuated/air released, and spring actuated/hydraulically released. The innovative BD design utilizes arms to provide a mechanical advantage, allowing higher clamping force. (Stopping power can be multiplied by installing additional caliper brakes on each disc.) Torque ranges from 8,000 to 23,000 in.-lbs., with disc diameters from 12 to 24 inches. Actuators mount on either side of the brake. Connections can be rotated 360° around actuator axis, and all pivot points have life lubricated bearings.

Brake shoes are mounted with detent pins for quick replacement. Larger, curved brake shoes with more contact area are available for longer life and higher peak input rate.

Highlights of the BD caliper brake (features vary by model) include static brake torque (8,000 to 23,000 in.-lbs.), a maximum disc speed of 3,200 rpm, optional discs: 12 to 24 in., a disc thickness of 0.50 to 1.00 in. and QD Bushing compatible for shaft sizes up to 3.875 in.

For more information:

Nexen Group
Phone: (800) 843-7445
www.nexengroup.com



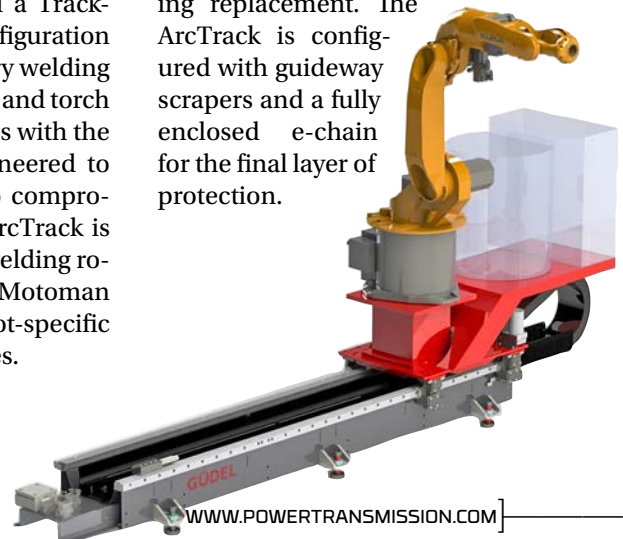
Güdel

INTRODUCES TRACK MOTION MODULE FOR ARC WELDING APPLICATIONS

Güdel, the global manufacturer of linear motion modules, robot track motion units, gantry robots and components, introduces ArcTrack, a pre-configured robot track motion module for arc welding applications that delivers the highest flexibility and reduces lead time by 40 percent.

Güdel has pre-engineered a TrackMotion Floor (TMF-2) configuration with an auxiliary shelf to carry welding equipment, large wire spools and torch cleaning and reaming devices with the robot. The shelf is pre-engineered to support over 750 kg with no compromise in performance. The ArcTrack is available for all leading arc welding robots, including Fanuc, ABB, Motoman and KUKA, and includes robot-specific gear boxes, motors and cables.

All Güdel TrackMotion units are designed and engineered for harsh environments, including welding, painting, die cast, foundry, sealing, machining and grinding. Güdel's unique cam follower and cartridge bearing design handle the worst environments, and enable 15-minute MTTR for bearing replacement. The ArcTrack is configured with guideway scrapers and a fully enclosed e-chain for the final layer of protection.



Güdel TrackMotion units, including the ArcTrack, are available in lengths from 3–100 meters, and can be equipped with robot risers from 50–600 mm. Because ArcTrack is a standard engineered solution, lead times are reduced to 8–10 weeks from order.

For more information:

Güdel Inc.
Phone: (734) 214-0000
www.gudel.com

Renold Gears

DISPLAYS RANGE OF GEARING SOLUTIONS AT IAAPA

Renold displayed its range of gearing solutions for theme park rides at the IAAPA Attractions Expo in November. IAAPA is the International Association of Amusement Parks and Attractions and represents nearly 5,000 amusement-industry members in 99 countries worldwide. It operates several global amusement-industry trade shows with Orlando being the largest. Renold is a long established and trusted supplier to the global theme park industry and offers a wide range of solutions, including custom designed gear units, with the choice of helical, bevel or worm gear types. Renold's experienced engineers were available throughout the duration of the show to offer advice on gearing for new rides as well as replacement gear units, maintenance and refurbishment. Renold's booth also displayed the new RBI coupling range, fluid couplings, sprag clutches and trapped roller free-wheels. The latter are precision devices that are fitted to provide absolute safety in the event of drive failure. If such a failure were to occur a backstopping safety feature engages immediately to protect the riders and the ride itself from back running.

For more information:

Renold Gears
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Renold Gears supplies many gearing solutions for the amusement park industry.



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Hansford Sensors

RELEASES HIGH-PERFORMANCE CABLE OPTION FOR INDUSTRIAL ACCELEROMETERS

Hansford Sensors has released a new FEP cable option for its high-performance vibration sensors. The new cable and connector design meets the needs of several industries, but is specifically suited for the extreme demands of mining and heavy applications.

Hansford Sensors said: "In developing this new premium design cable option for our vibration sensors, we've met the specialist requirements of machinery operating in even the most extreme of applications. The FEP cable with its protective conduit is available to order now and can be paired with several of



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The new FEP cable features a protective conduit made from stainless steel, which is highly resistant to oil, rust and corrosion and ideal for withstanding the demands of harsh industrial environments without compromising performance. Highly flexible, this cable and conduit combination also offers impressive compression, impact and tensile strength. With notable resistance to abrasion and tearing, the FEP cable is able to withstand temperatures ranging from -80°C to 200°C.

Commenting on the new launch, Chris Hansford, managing director of

our popular HS-100 and HS-150, top and side entry AC accelerometers."

Available with both dual acceleration and temperature outputs, the FEP cable and protective conduit is ideal for a broad range of industries including building services, pulp and paper, mining, metals, utilities, automotive, water and pharmaceutical.

For more information:

Hansford Sensors U.S.
Phone: (888) 450-8490
www.hansfordsensors.com

Ogura Industrial

IMPROVES AMC AND AMB SERIES DESIGN

Ogura Industrial is pleased to announce a new addition to our product line. Although the AMC and AMB series are not new, they have gone through significant design improvements. Because of these improvements, the AMC/AMB-E series is finding new opportunities for machinery manufacturers in North America because of some of its unique features.

The new E series is available in four

sizes, 2.5, 5, 10, and 20. These have been redesigned to have approximately a 25% increase in torque, a 5% reduction in power consumption, approximately a 15% lighter weight while also being roHs compliant and having a lower cost. These increases in performance and lower cost have been achieved by optimizing the coil, housing and rotor design to achieve a more efficient flux path.



For more information:

Ogura Industrial Corp.
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Igus

REDUCES AUTOMATION COSTS WITH
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Mechanical engineers often need a basic, space-saving linear guide when designing technology that performs simple tasks, such as vending machines. Since such technology does not handle high loads, high speeds or high positioning accuracy, intricate linear guides are not necessary. However, most solutions on the market are very advanced, leaving manufacturers forced to either pay for features they do not need or develop their own linear guide.

Igus is now offering the drylin ZLW eco, a ready-to-install entry-level series that is making simple position-



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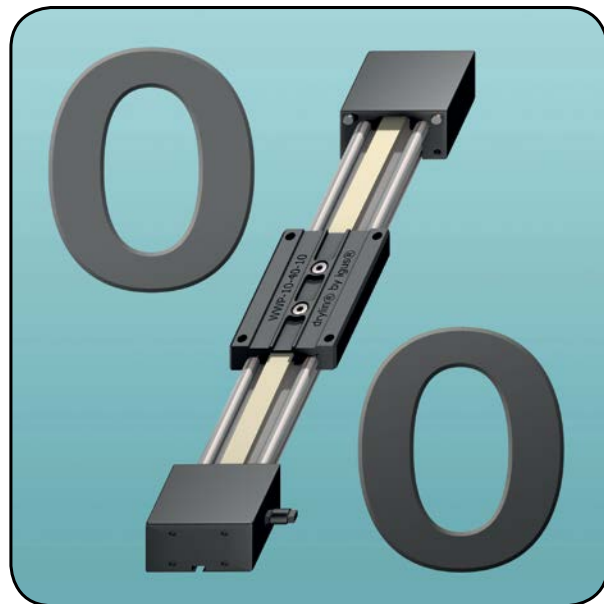


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ing and adjustment tasks extremely efficient and, above all, cost-effective. “A toothed belt axis of this entry-level series with a stroke length of 100 millimeters starts as low as 150 dollars,” said Stefan Niermann, head of Igus’ drylin linear and drive technology division. “In comparison, a toothed belt axis from the standard series, which has high-performance features and is therefore unnecessary to use for simple operations, costs almost three times more.” The carriage and shaft end supports are produced by injection molding, which is more cost-effective than mechanical filling used for metal component production. “This also reduces the number of components and thus the installation efforts for every eco linear axis, which in the end is reflected in the low prices of this entry-level series,” explained Niermann. A further cost-saving element of the drylin ZLW eco is the plain bearings used in the sliding carriage, which are made of iglide high-performance plastics. “Iglide bearings are forty percent more cost-effective than conventional rolling bearings and 100 percent maintenance-free in operation,” Niermann said. Without compromising the smooth-running operation and durability of the standard series, users can simply install the eco axes and save time and money with the maintenance-free, dry-running triboplastic bearings.



The entry-level drylin ZLW eco has two installation sizes: 0630 and 1040. The base is an anodized drylin W profile made of clear anodized aluminum. At the ends of the profile are plastic shaft end supports for drive technology. A neoprene toothed belt is tensioned between the shaft end supports, which pushes and pulls a solid plastic carriage with a positioning accuracy of 0.3 millimeters. The stroke lengths can be individually adjusted by the user. Due to its lightweight construction, the toothed belt axes weigh only 0.3 kg and 0.7 kg, and can move loads up to 3 kg or 10 kg respectively. Matching motor kits also are available.

For more information:

Igus
Phone: (800) 521-2747
www.igus.com

DESTACO

INTRODUCES VACUUM CUPS FOR PICK-AND-PLACE PARTS HANDLING

DESTACO, a provider of precision movement, positioning and control solutions in industrial automation, is leading the way in vacuum-cup innovation with the introduction of its new Deepdish Series Vacuum Cups, which are designed to maximize grip performance on contoured and oily surfaces.

The new DESTACO Deepdish Series Vacuum Cups feature an all new bell-shaped body design, available in three sizes, 65, 90 and 110 mm. Unique, molded-in gussets prevent the cup edges from rolling up on convex surfaces while the thin, pliable design and special inner tread pattern allow the cups to effectively adhere to flat, convex, concave, domed and oily surfaces for maximum grip performance.

“The addition of the Deepdish Series to our vacuum cup line instantly expands the number of applications and uses for our cups. These new cups establish a whole new standard of operational reliability, longevity and productivity for our customer’s pick-and-place parts handling needs,” stated Matt Girand, vice president, global research, development and engineering of DESTACO. “We’re constantly striving to innovate new ways to help manufacturers improve their production processes, and the addition of the Deepdish Vacuum Cups illustrates this commitment.”

Another key attribute that sets the new DESTACO Deepdish cups apart from all others is the polyurethane materials-of-construction. Polyurethane provides superior wear-resistance when compared to rubber to promote longer service life and reduced maintenance. In addition, polyurethane will not leave any marks on handled objects and has fantastic elastic memory, even after hundreds of thousands of cycles. To top off this robust feature set, the new DESTACO Deepdish cups have a durometer of 60 (PU60, translucent red in color). The 60 durometer makes it easier to pick up highly contoured panels, while also being capable of withstanding the elevated shear forces that are created by increased acceleration and deceleration rates.

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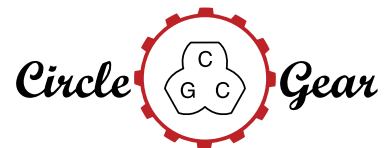
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Röhmm

OFFERS LUBRICATION SYSTEM, CLAMPING SYSTEM AND DRONE GRIPPER

Röhmm exhibited several of the company's latest product developments at EMO 2017 including the Lubritool lubrication system, an intelligent clamping system, a drone gripper and several industry-specific solutions.

The Lubritool system is the most intelligent lubricating tool on the market and enables automatic lubrication of tool clamping systems in machine tools within seconds instead of minutes. It reduces costly and time-consuming maintenance work associated with the manual lubrication process and prevents machine downtime. At



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EMO, the Lubritool received the MM Award for innovation presented by the trade journal *MaschinenMarkt*.

Röhmm also showcased an intelligent clamping system that opens new possibilities in workpiece clamping by supporting manufacturers in Industry 4.0 initiatives. The system registers clamping force during rotation and during the course of machining, logging status data and prompting users to undertake proactive maintenance when necessary.

To meet the needs of the growing drone market, Röhmm debuted a gripper that makes it possible for drones to grip, move and release objects automatically. Additionally, visitors to the company's booth experienced several industrial solutions for railways and the oil and gas industries, as well as an all-around package for the machining of artificial hip and hip socket implants—a market that is steadily growing as life expectancies increase.

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Iwis

OFFERS GRIP CHAINS WITH WEAR-RESISTANT CLAMPING ELEMENTS

Chain manufacturer Iwis presents grip chains that have wear- and corrosion-resistant clamping elements that ensure safe and reliable feeding, transport and positioning of thin-walled materials with a large surface area. Grip chains are used, for example, in packaging, medical technology, electronics, PCB production and metal-working industry applications. For use in food processing, all chains are available with food-grade lubrication.

The clamping elements allow the chain to grip and hold thin-walled materials with large surface area, such as films. Different levels of spring force allow a wide range of materials to be gently gripped and securely held. Conventional grip chains often have the drawback that the clamping elements do not provide enough space to insert the film. They also apply a point load to the film, which can cause the film to deform at the gripper or even to rupture. They are also noisy in operation. The gripper of the new grip chains from Iwis fits into its groove very accurately and therefore offers a better retention force. More free space in the foil insertion area allows an improved foil feed and the foils do not twist or deform at the edge of the gripper element. This design also reduces noise emissions. The grip chains also feature burred plates, which ensure a reliable operation and optimum hygiene.



High-performance grip chains from Iwis have an outstanding wear resistance. Being optimally pre-stretched, the chains exhibit only a minimal initial elongation. Being highly rigid, they can also be used in long machines. Identical chain lengths within the selected tolerance range ensure excellent running characteristics in both synchronous and parallel operation.

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CNC System Upgrades

Thomson Cuts Costs and Production Time for Autoscale Inc.

Jack Kang, Global Product Manager – Linear Bearings and Guides, Thomson Industries, Inc.

Automated production of large objects such as auto body prototypes, boat hulls and surfboards traditionally requires computerized numerical control (CNC) systems costing nearly a million dollars. But now, through an innovative integration of advanced composite materials and high-precision motion control technology, Thomson Industries has reduced the cost of high-performance CNC systems for cutting light grade materials such as wood, foam, concrete or aluminum by about 90 percent.

Redefining CNC Price/Performance

Building on experience gained through custom surfboard production, Santa Clara, California-based Autoscale, Inc. has developed cutting-edge CNC technology, which can produce objects with lengths up to 16 feet (Figure 1).



Figure 1 Autoscale's high-performance CNC technology can cut objects up to 16 feet long.

On the business end of a CNC system used for softer materials is a router, hotwire or other cutting technology appropriate for materials such as light woods or Expanded Polystyrene (EPS) foams. Following advanced algorithms, which product designers create for mainstream CAD/CAM software, frame components known as gantries guide the cutting tools along the X-Y axes, while an arm on the Z axis moves the tools vertically to add the third dimension to the product. For most mainstream systems today, the frame and all of the moving parts are composed of steel.

"When we started building larger routers, we were kind of like a dog chasing his tail," said Autoscale Founder and Owner Dan Bolfing. "We wanted rigidity and speed. But adding rigidity to the moving parts also added weight, which slows the system down and adds cost. Our first models were

weighing in at around 3,500 pounds, including the gantry, carriage and all the gear boxes, profile rails ball screws and other mechanisms. This is not far from what the rest of the industry was doing with steel, but we wanted to do better."

Replacing Steel with Carbon Fiber

After much experimentation with lighter material alternatives, Bolfing and his engineering team concluded that replacing steel with carbon fiber would offer many advantages. For one, it would be easier and faster to produce the gantries. Steel expands and contracts, and it was taking up to two weeks to get the steel gantries straight. After being welded together, they would have to be straightened, assembled, and after it was determined that they were straight enough, pulled apart, powder coated and reassembled. Carbon fiber, on the other hand, allowed creation of patterns and molds in advance, and when the design was laid out in the carbon fiber, it would always be faithful to the pattern, unlike steel, which twisted, turned and flexed as the temperature changed.

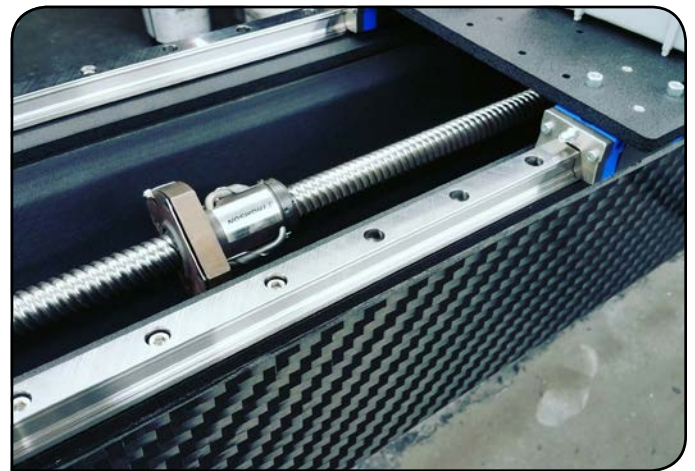


Figure 2 Thomson precision ball screws assist the gantry in moving smoothly and accurately along the X and Y axes.

Most dramatic, however, was that in addition to nearly halving production time — from two weeks to about one — carbon fiber was much lighter than steel, reducing weight of all the moving parts from 3500 pounds to 350 pounds, virtually eliminating the industrywide rigidity/speed tradeoff. Where a steel gantry based system might cut at 300 to 400 inches per minute, the carbon fiber-based system could cut at 800 inches per minute without vibration. Moreover, the speed advantage comes not only in top-end speed but in ramp time as well, shortening the time it takes a system to reach its top speed and thus further reducing processing time.

"You never want to accelerate at full throttle," said Bolfing. "With a heavy gantry, you have to ramp slowly – accelerating

and decelerating at the beginning and end of every cut. The lighter the gantry, the faster you can accomplish that. With a steel gantry, you are never going to get up to 300 inches a minute on a detailed part because you are going to be constantly accelerating and decelerating. With carbon, however, you can shorten the acceleration distance by 90 percent, which is especially valuable in finishing parts in foam, clay or soft plastics. You run faster without having drastic acceleration requirements cutting into your production time.”

Precision Guidance

Key to meticulous operation of this system is effective use of linear guides, which control the X-Y axes and ball screws. Autoscale chose Thomson linear guides, which are known for their high precision, reliability and adaptability.

“Thomson is the only company that provides a 16-foot profile rail,” said Bolfing. “With any other vendor we would have to splice two shorter pieces together, which challenges accuracy and durability.” Thomson provided complete, fully machined screw assemblies with bearing mounts ready to bolt on.

For router-based operations, which tend to have a higher load, Thomson provided profile rail linear guides. These ensure smooth, precise guidance of the milling head. Thomson also supplied its Super Ball Bushing bearings, which have 27 times the travel life of conventional linear bearings.

Besides the increase in load capacity, the Thomson Super Ball Bushing bearing is self-aligning, lightweight and adjustable with a low coefficient of friction.



Figure 3 The Thomson 500 Series profile rail provided the length, durability and accuracy Autoscale required for its system.

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The Thomson technology attaches to the carbon fiber-based gantries shown in Figure 4. Exactly how it connects, however, is highly proprietary and is a key component of Autoscale's success.

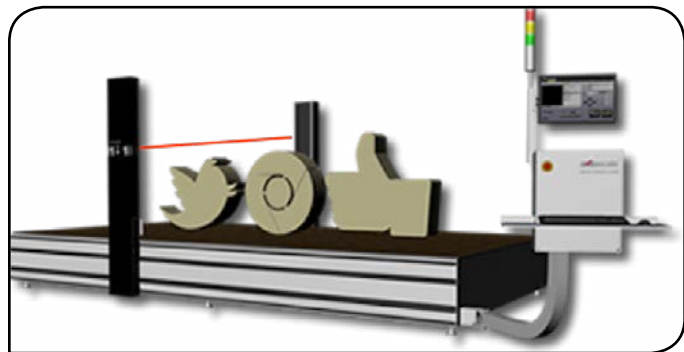


Figure 4 Autoscale's new Monster CNS system with carbon fiber gantries, guided by Thomson technology.

For hotwire applications such as those that might be used for the softest substances like foam (Figure 4), Autoscale uses Thomson ball screws and round rail linear bearings to guide the hotwire cutting portion of the machine. Round rail guidance is well-suited for the hotwire portion of the machine because the rail can be end-supported or intermittently supported depending on application requirements. It is less prone to jamming because of misalignment and less demanding to assemble and align than a profile rail, and component costs are typically less than profile rail equivalents.

A New Standard in Large-Scale Modeling and Production

For precision cutting of soft materials, including softer metals such as aluminum, Autoscale systems have set a new price performance standard for cutting of large-scale products, models and prototypes. An Autoscale system costs \$89,000, where a conventional system costs more than \$800,000. The Autoscale cuts as fast as a conventional system, but does so with a smaller footprint. Application areas for the new system include:

- Aerospace, including prototypes for wings and fuselages
- Automotive market, including race cars, smart cars, electric cars, interior consoles and exterior bodies
- Architecture, including pre-cut concrete pool patterns, concrete staircase patterns and concrete for pools and spas

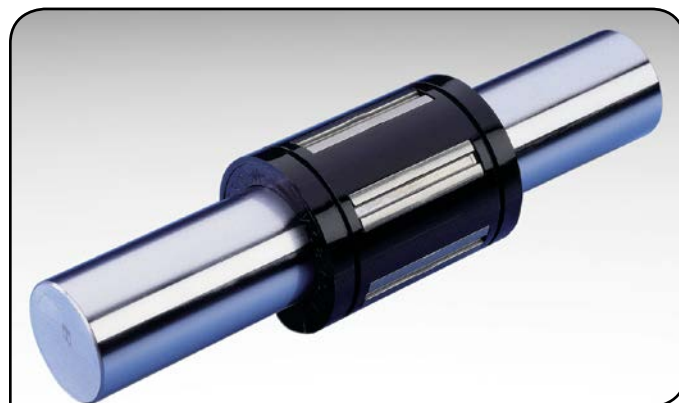


Figure 6 Thomson Super Ball Bushing bearings are used on a 60 Case round rail to guide hot wire cutting.

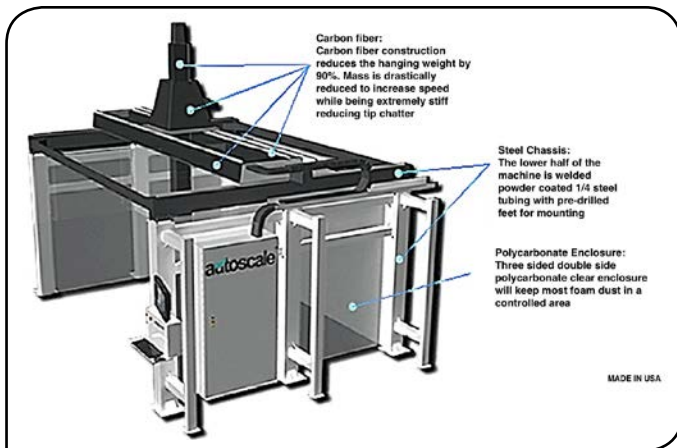


Figure 5 Full-scale CNC hotwire cutter for Expanded Polystyrene (EPS) foam.



Figure 7 A concept vehicle cut on an Autoscale router.

- Marine, including surfboards and standup boards, outrigger canoes and large boat hulls
- Furniture, including kiosks and point-of-purchase displays
- Custom packaging

Future Growth Swell

Although business is going quite well, Bolfling is looking to even bigger and better things for the future. In addition to marketing his equipment, he also applies it in a separate business called Contactscale, which offers competitive job shop services to companies that need large-scale products and prototypes. Bolfling also envisions a franchising operation in which localized job shops would purchase his system and provide local services. The little company that started out providing custom boards to surfers is now clearly riding a big wave. **PTE**

For more information:

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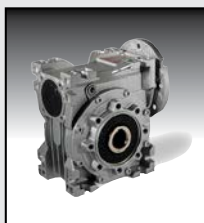
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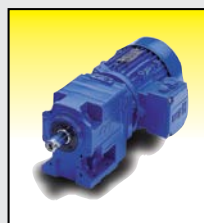
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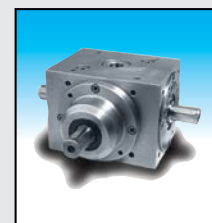
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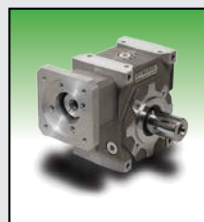


Spiral Bevel Gearboxes

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Force Control

Assists in Helicopter Transmission Overhaul

Force Control Industries

Professional Aircraft Accessories, a Greenwich Aero-Group company based in Titusville, Florida, is a Federal Aviation Administration (FAA) and European Aviation Part 145 Repair Station specializing in the repair and overhaul of landing gear, accessories, instrumentation, pressurization, radio, radar, avionics and airframe components. They offer component capabilities on aircraft such as Bombardier, Boeing, Lockheed, Gulfstream and more.

As an approved Licensed Repair Facility for Gulfstream Aerospace, Electromech, Goodrich, Hawker Beechcraft and Lockheed, the 19-year old company is well entrenched in fixed wing aircraft. Expanding into the helicopter market was a natural extension, and the request for quote for overhaul and recertification of LH58 Kiowa helicopter transmissions provided a perfect opportunity. There was only one caveat: a very condensed timeframe.

Although it was a case of “be careful what you wish for,” PAA officials found that oil shear braking technology helped them meet not only the technical requirements of the project but also the fast response needed for initial certification testing.

A Short-Fused Project Takes Off

Once the RFQ was approved, PAA officials had essentially four months to design and build a dynamometer test stand, and overhaul three transmissions to get their project verification audit from the Army. Given the short timeframe, the project engineering team opted to use non-regenerative technology. Jerry Leach, director of production engineering and planning led a team of people to begin to design the system, and race against the clock.

“Designing a system to do what we wanted with regenerative technology would have been more efficient, but it would have taken four times longer and cost at least twice as much,” said Leach of his tight timeframe and

fixed-cost project. “We contracted out key components of a system we could build in-house and decided to dump power into the system and then load it via braking. This is very effective, but it builds a lot of heat which must be exhausted.”

Power is supplied via a General Electric (GE) electric motor and variable frequency drive (VFD) combination with 700 horsepower and 3,600 rotations per minute. The specification calls for testing at 40- 60- 80- 100- and 112 percent of load for various timeframes.

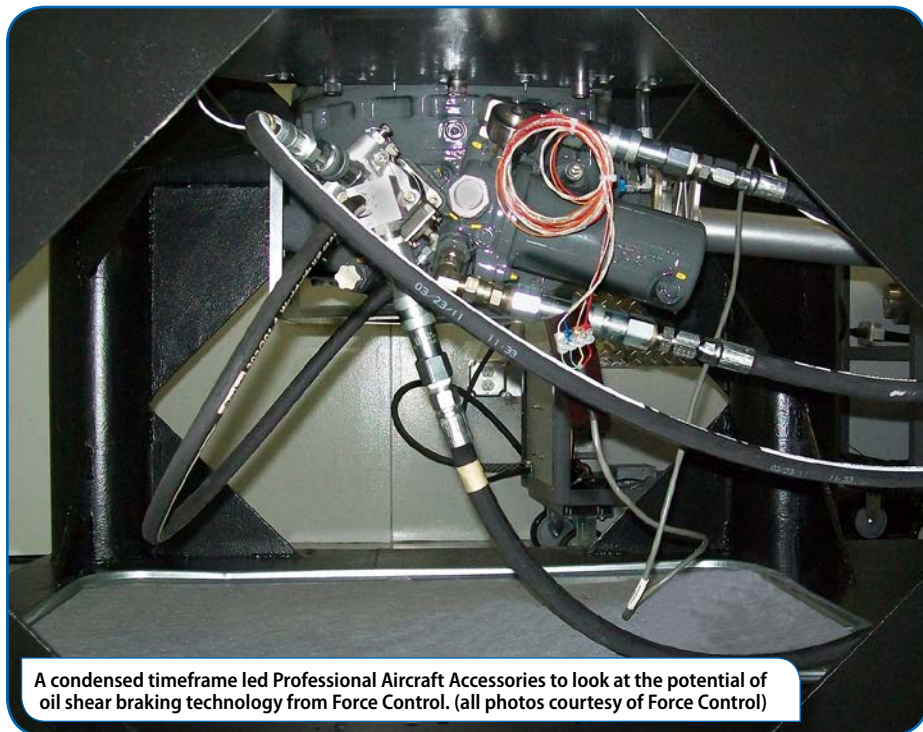
Load is supplied by a TB 83 oil shear brake from Force Control Industries which can be precisely controlled to meet the various spec points.

At 112 percent of load, Leach cited 8,200 foot pounds of torque. The test takes about an hour, Leach said, during which there is very little temperature rise.

“They definitely sized the cooling unit correctly,” he said.

To dissipate the heat generated by that amount of torque, the oil shear brake came equipped with a patented forced lubrication and cooling system. In addition, the heat exchanger circuitry also has components to safeguard against viscosity changes as the oil changes temperature. The cooler the oil, the thicker it gets. To safeguard against overloading the pump, which moves 250 gallons of oil per minute across the brake, heating elements are installed to keep the oil at a fairly consistent temperature. A programmable logic controller (PLC) monitors the temperature and controls the heating elements.

Leach and the team specified components from various sources, the GE motor and VFD, the skid mounted brake and cooling system from Fairfield, Ohio based Force Control Industries, and a 1.7:1 gearbox to reach the 6000 rpm needed. They used LabView software from National Instruments



A condensed timeframe led Professional Aircraft Accessories to look at the potential of oil shear braking technology from Force Control. (all photos courtesy of Force Control)

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oil shear brakes

at www.powertransmission.com

for data acquisition and testing. Once the major components were installed, though, there was still plenty of work to do.

“The plumbing required to move that much fluid, at temperatures which range from 50 to 175 degrees to accommodate the expansion and contraction was challenging,” said Leach.

Additionally, 1,200-amp electrical service had to be run into the facility before everything could be wired and tested. Getting it all in place was the first hurdle, but it wasn’t the last challenge for the project.

While the system was designed for vertical operation the brake was mounted horizontally to allow direct torque measurement. Force Control officials didn’t foresee a problem doing so, but an air pocket was created

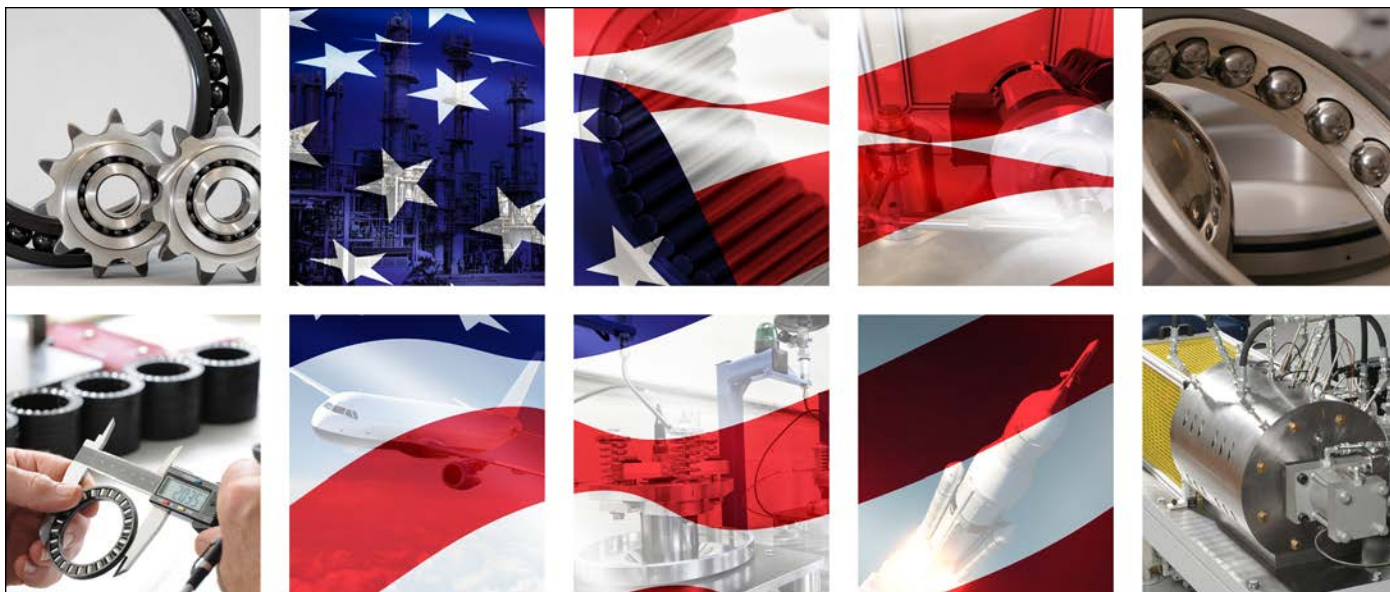


Close-up of the motor used in the application.

which caused the friction disc to burn up, since it was not fully wetted. A fast and effective solution was suggested and implemented, allowing PAA to

produce overhauls for the initial validation audit.

After passing the validation audit, PAA was off and running. The test



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stand was performing well, but the brake needed to be rebuilt after every 10-15 transmissions.

Leach, whose company, Professional Aircraft Accessories, was footing the bill, thought that rebuilding the brake so frequently would be a normal course of business until Force Control engineers devised a solution.

“Apparently their engineers were seeing similar issues with another installation and determined a need for additional surface area. We were very pleased to hear that it could be accomplished by changing out internal components,” Leach said.

The alternative, changing external components, would likely change the location of virtually all of the hoses, power connections and positioning of the brake relative to the helicopter transmission. In short, it might mean relocating everything.

Leach reported that once the new components were installed, they haven’t experienced any issues with the system.

“The level of support was outstanding” said Leach of his experience with Force Control. “We would work with them again without hesitation.”

How Oil Shear Technology Works

Normal dry brakes employ a sacrificial surface, usually a disc or pad, to engage the load. Having no good way to remove the heat caused from engagement between the disk and plate, these surfaces must absorb the heat. The extremely high temperatures will eventually degrade the friction material. As the friction surface wears away and begins to glaze, the coefficient of friction is also reduced, causing torque fade. This causes positioning errors which require adjustment or replacement of the friction surface.

Oil shear technology plays a major role in ensuring that the helicopter transmission test stand can operate continuously without adjustment and still achieve precise positioning for the desired percentages of load. Since

a fluid film flows between the friction surfaces, the fluid is compressed as the brake is engaged. The fluid particles in shear transmit torque to the other side. Since most of the work is done by the fluid particles in shear, wear is virtually eliminated.

In addition to transmitting torque, the oil also helps to dissipate heat due to a patented fluid re-circulation system. Given the size of the system and the temperature rise, a cooling system was also provided.

Along with torque transmission and heat removal, the fluid also serves to continually lubricate all components, thus extending their service life. Oil shear technology also provides a “cushioned” engagement that reduces shock to the drive system – further ex-

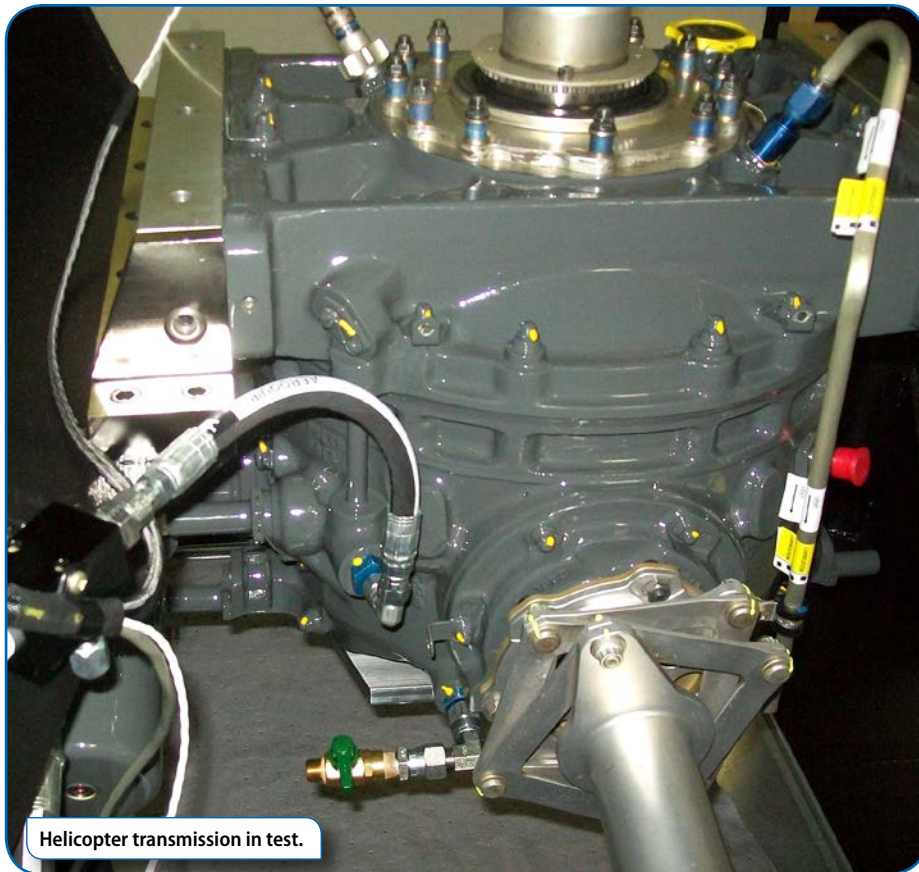
tending service life. Unlike dry brakes and clutches, the totally enclosed oil shear system is impervious to external elements such as wet, dusty or dirty environments. Since the layer of oil eliminates wear, the Force Control brake provides a long service life. With elimination of wear comes elimination of adjustment, and ultimately increased “uptime” for the helicopter gearbox test stand.

Clear Skies and Smooth Sailing

Despite a short timeframe for development and a rocky start, the LH58 Kiowa transmission overhaul test stand is operating smoothly and has been for the past two years. Following this successful project, PAA is also planning addi-



Close-up of the oil-shear brake used in the application.



tional helicopter components, which can be accomplished with minor modifications to the existing test stand.

While the success of the project was important to both PAA and Force Control, it is vital for the brave men and women who rely on the Kiowa reconnaissance aircraft day-in and day-out, or, more appropriately, night-in and night-out. Rebuilding the transmissions quickly yet proficiently and getting these armed scout aircraft back into service is a top priority for the Army. Thanks to cooperation between customer and vendor as well as the robust and reliable oil shear braking technology employed in the custom dynamometer test stand at Professional Aircraft Accessories can tell the Army, "mission accomplished." **PTE**

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Gear Expo 2017

Gears, Bearings and Software Highlight Columbus Event

The following recap looks at some of the exhibitors from Gear Expo that manufacture mechanical power transmission components or provide resources for these components.

Check www.powertransmission.com for additional material from the show.

How to Qualify a Bearing Supplier

Managing Editor Randy Stott sat down with Chris Napoleon, President of Napoleon Engineering Services, to discuss the importance of qualifying your bearing supplier, especially in consideration of today's global supply chain.



To see the full 12-minute interview, visit www.powertransmission.com/tv/

The highlights of the interview can be summarized by these eight steps:

1. Acknowledging that there is risk and allocating the necessary resources.
2. Defining the suppliers you will work with and understanding their structure: buying direct, through distribution, brokers, etc.
3. Defining the level of engineering support you need from the bearing supplier and evaluating how they will service this need.
4. Perform a quality system audit by your own quality department
5. Perform a bearing design and manufacturing audit at the plant using a bearing specialist.
6. Product inspection to determine design intention, actual manufacturing capability to carry out the design intention and overall quality of workmanship.
7. Use of modeling to determine stress distribution and theoretical product life based on actual inspection characteristics.
8. Physical testing based on expected failure mode sitting the application. Typically related to static/impact testing, environmental test to evaluate seal efficiency, or life testing under accelerated or application conditions.

The entire process is detailed in Figure 1. According to Napoleon, steps 5-8 are typically carried out by an independent

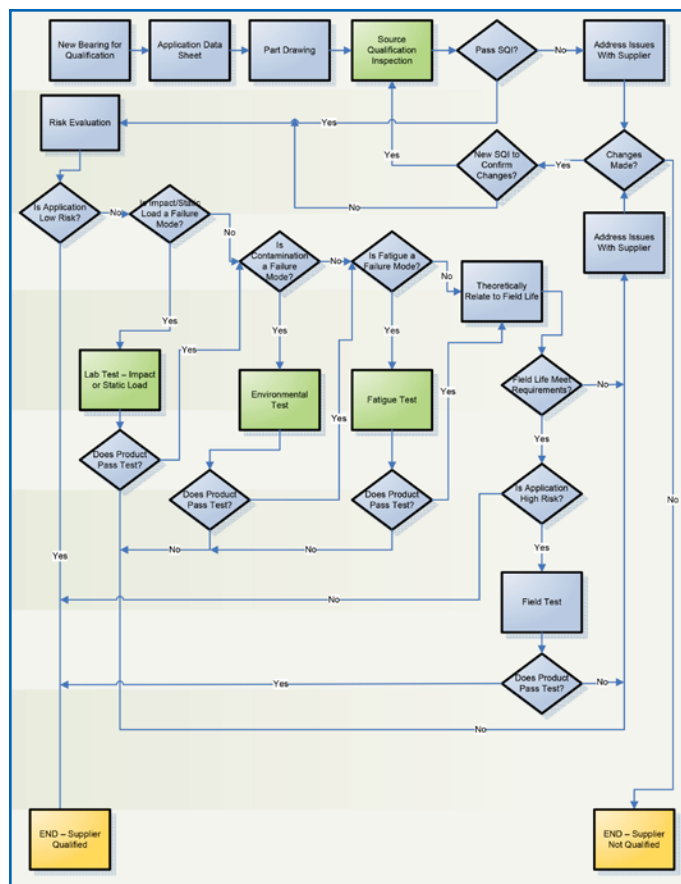


Figure 1 Flowchart of the bearing supplier qualification process.

bearing lab, such as Napoleon Engineering Services.

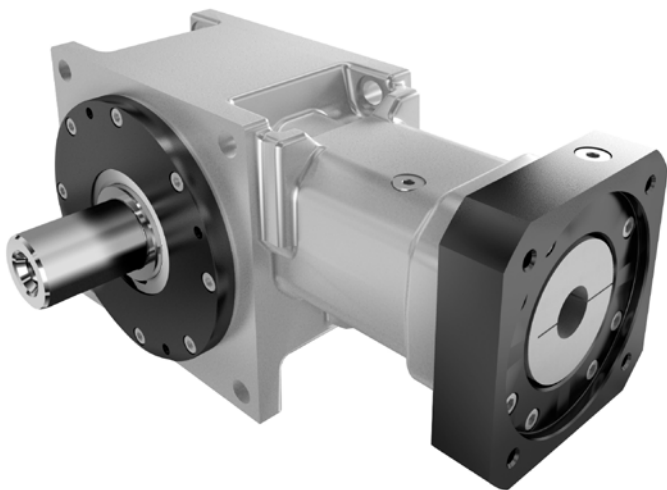
For more information:

Napoleon Engineering Services
www.nesbearings.com/technical-information/steps-for-risk-mitigations/

Exsys Upgrades Gearbox Capabilities

Exsys Tool offered a line of high-quality Eppinger Spiral Bevel Gearboxes during Gear Expo 2017. Eppinger's BT (bevel torque) and BM (bevel maximum torque) compact spiral bevel gears deliver high torque and maximum efficiency for gear applications that require extreme reliability and variability at speeds over 1,000 rotations per minute, as is the case for vehicle differentials.

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 $i = 1:1$.

After the show, Exsys also announced the expansion of Eppinger gearbox offerings to include the HT-type hypoid gearboxes that feature compact, robust designs suitable for both specific and dynamic applications.

The Eppinger HT-type hypoid gearboxes have mono-bloc housings that distinguish this series with extreme stability and offers maximum precision and efficiency. A highly flexible flange and coupling system enables the gearboxes to be connected to a host of servo motors without difficulty.

With solid steel alloy and hollow shafts for shrink disc connection, users can install the 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 $i = 15:1$.



Heavy-duty bevel gears, designed and manufactured according to the Gleason process provide optimal gearing efficiency, high transmission precision and reduced load on bearings. Users also gain extremely secure torque transmission through a friction-locked, zero backlash connection of the crown gears on the drive shaft. The tooth flanks are



ground to handle heavy operating demands on transmission performance at minimal tooth clearance. Such precise gear settings are achieved through constant measuring of the gear components and 100 percent test running during assembly.

In addition to extreme stability and precision, the hypoid gearbox housing offers exact positioning of the bearing seats and an integrated reinforced input neck that ensures a secure motor connection. Screw holes in the housing edges also enable a stable connection of the gearbox for various installation positions.

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Exsys Tool, Inc.
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www.exsys-tool.com

McInnes Rolled Rings

FOCUSES ON SPEED AT GEAR EXPO

“Lead time is an opportunity to differentiate ourselves from the competition,” explained Shawn O’Brien, vice president sales and marketing at McInnes Rolled Rings. The company has established 1 to 2 week lead times as standard. In many



cases, they are shipping in a matter of days. "Our mission is to offer industry leading cycles in every market. When our customers realize that they can consistently rely on this it gives them an advantage over their competition and enables them to avoid carrying excess inventory," he added.



Speed has been the cornerstone of the McInnes brand. While the company is always looking for new efficiencies, lead times have remained consistent. There are no premiums or special programs. McInnes continues to invest in both its team as well as its equipment. The new \$8 million heat treat investment enables the company to process significantly more tonnage. "Maintaining a full staff of experienced associates in our plant, office and in the field enables us to meet any demand surges in stride," O'Brien said.

Gear Expo gave the organization the opportunity to spread the word on everything they've been doing to enhance capabilities and increase lead times. The company sees an increased value on service in the coming years and plans to focus on being the best value option for the gear market.

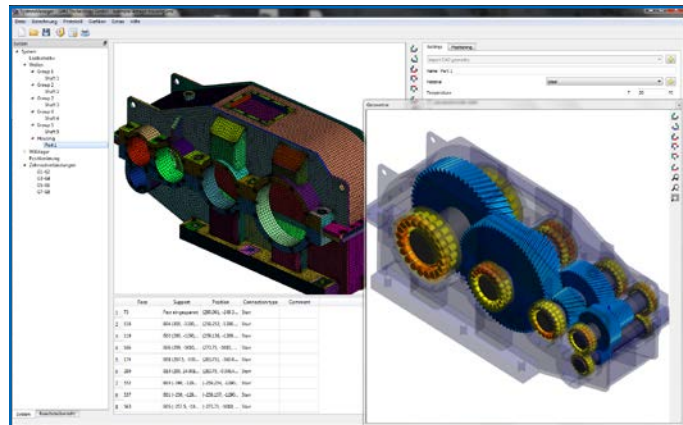
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GWJ

EXAMINES GEARBOX CALCULATIONS AT GEAR EXPO

GWJ presented a system calculation using its *SystemManager* software during Gear Expo. The presentation at the Solutions Center took a closer look at the combination of FEM and analytical methods with just one calculation system and which interactions can occur (*SystemManager* to do the FEM calculations together with the analytical calculations e.g. for gears according to different standards like DIN 3990, ISO 6336 or ANSI/AGMA 2101/2001). The influence of the bearing and housing stiffness were also considered.



GWJ released new developments with *SystemManager* earlier this year. The software can be utilized for complete systems of machine elements, i.e., the software is a coupled FE calculation of multi-shaft systems with gears as non-linear coupling elements. *SystemManager* runs as a desktop application, making it possible to configure and calculate entire systems with just a few mouse clicks. *SystemManager* also allows the import of 3D housings as STEP files. The software meshes the parts automatically to consider deformation and stiffness of the housing throughout the system. A further extension of the 3D elastic parts function is the support of planet carriers and imported shafts. Planet carriers can be imported as CAD models or be defined parametrically; various basic designs are available for the parametric planet carriers.

For more information:

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Cincinnati Gearing Systems

DISCUSSES WHAT QUESTIONS A GEAR BUYER SHOULD ASK A POTENTIAL SUPPLIER

Matthew Jaster, senior editor at *Power Transmission Engineering*, sat down with Patrick Potter, director of sales at Cincinnati Gearing Systems during Gear Expo to discuss some of the questions a gear buyer should ask of a potential supplier. Here's a quick overview of that discussion.

What kind of gears do you specialize in?

While this seems obvious, it's important to know that the supplier has the knowledge and experience with the type of gears the buyer needs. If you're buying a small automotive spur gear they should be able to show they have made plenty of these types of gears. If they're buying turbomachinery (a two-meter, high-speed, high-powered bull gear, for example) you want to be sure they've made these before and have the experience and the knowhow necessary to be successful.

Can you audit my design and make suggestions?

One feature you would like to have is some engineering expertise, both from a design standpoint and a manufacturing standpoint, according to Potter. You may have a design that you're comfortable with, but it wouldn't hurt to have the gear manufacturer look at the design to make it more economical and possibly function better in the application. Added value is so important to look for when shopping for a supplier.

Do you have your own heat treat facilities?

Ideally, they should own a heat treat facility, though this is not common. The benefits include that they are accustomed to dealing with distortion issues and the various problems that can occur during the heat treat process. This should also make things more economical from a cost perspective. Scrap should be lower and lead times and delivery times should be better.

Does the company offer engineering services?

This goes back to the design side of things. A good gear manufacturer should be able to provide 3D models that can match up with the buyer's software. If they're doing high-speed gearing, they should be able to offer vibration analysis. They should also be able to manufacture to different standards, not just AGMA. (API, DNV, ABS, for example).

How important is quality and the quality system you have in place?

It is not a complete requirement that your gear manufacturer be ISO-certified, but it does give you a certain comfort level. In lieu of maybe having many, many references for a specific design, if they can demonstrate that they are ISO-certified it gives you a feeling that they're going to be reliable in terms of quality.

A really good gear shop would be ISO 9001. Those involved in automotive gearing, you'd expect them to be TS16949.



Going into next year, you'd also expect them to be IATF certified (International Automotive Task Force).

Do you offer any field services?

You would really like to have field service support from your supplier whether you're working with single gears or complete gear drives. Knowing they can come out tear down a unit, inspect it and support that equipment is a nice feature to have.

What about rebuild and remanufacture capabilities?

This fits in with service. If you have units out in the field and your technician goes out and finds issues that cannot be addressed, you'll want to be able to send the unit back to the gear manufacturer and have them tear it down completely and rebuild it to new specifications.

More importantly, if you have many different models of gearboxes, a refinery for example that may have a lot of different products onsite, you'd like to have a gear manufacturer that can service all those units in their own shop.

What kind of gearbox testing can you do?

Whether you rebuild a unit or build a new one from scratch, you want to be able to spin test it according to the latest standards. These are both AGMA and API standards, but a no-load, 4-hour, full-speed spin test with complete vibration and temperature data you'd expect to get from a top tier gear manufacturer.

How long has your company been in business?

This relates to the stability of a company. These days there are so many mergers and acquisitions and it's hard to tell who is going to be there a year from now. If you buy millions of dollars of equipment and three years from now you can't get it serviced that's not a great scenario. 10 years of stability is a great starting point.

Is your company privately or publically held?

This is more of a personal preference. A family-owned company can give you that more direct level of service and a large, publically held company might have deeper pockets and might be more stable, though we've seen lately that this is not the case every time. The buyer should always know what he or she is getting into. **PTE**

For more information:

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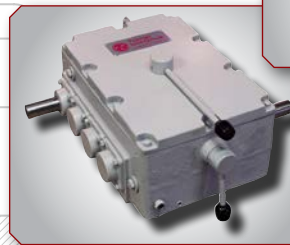
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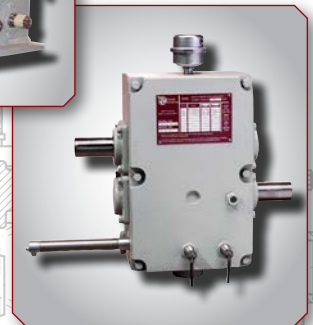
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2017 PTE Buyers Guide

About This Directory

The 2017 *Power Transmission Engineering* Buyers Guide was compiled to provide you with a handy resource containing the contact information for significant suppliers of power transmission components.

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Bold Listings throughout the Buyers Guide indicate that a company has an advertisement in this issue of *Power Transmission Engineering*.

How to Get Listed in the Buyers Guide

Although every effort has been made to ensure that this Buyers Guide is as comprehensive, complete and accurate as possible, some companies may have been inadvertently omitted. If you'd like to add your company to the directory, we welcome you. Please visit www.powertransmission.com/getlisted.php to fill out a short form with your company information and Buyers Guide categories. These listings will appear online at www.powertransmission.com, and those listed online will automatically appear in next year's printed Buyers Guide.

Handy Online Resources

The *Power Transmission Engineering* Buyers Guide – The listings printed here are just the basics. Visit our online buyers guide for the most comprehensive directory of suppliers of gears, bearings, motors, clutches, couplings, gear drives and other mechanical power transmission components, broken down into sub-category by type of product manufactured:

www.powertransmission.com/directory/



The Gear Industry Buyers Guide – If you manufacture gears or need information on suppliers of machine tools, tooling and services for gear manufacturers, please visit the buyers guide on *Gear Technology's* website:

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Steelmans Broaches Pvt Ltd
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Neugart USA Corp
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Pragati Transmission Pvt Ltd
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Precision Pump and Gear Works
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Precision Technologies Group (PTG) Ltd.
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PSL of America
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PTD Outlet: Power Transmission Distributors
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Gear Design

PLEASE NOTE—ASK THE EXPERT is intended—and will certainly remain so—to be a reader-driven platform where engineers, designers, specifiers, shop managers, machinists and more can ask a question that needs answering—complex or otherwise. But just to keep the line moving, so to speak, we occasionally ask our “Experts” to pose both an expert question—and answer—informed by the invaluable back-and-forth that occurs between client and, to cite a few examples, consultant, or job shop, or OEM...

Following is an experienced-based Q&A scenario regarding gear design, presented by **Dr. Alex Kapelevich**, principal, AK Gears:

THE QUESTION

From a practical gear design point of view, can one find an acceptable “happy medium” design approach between specifying either conventional standard gears—and their inherent, modest performance characteristics regarding, for example, size, weight, load carrying capacity, noise and vibrations, etc.; or going with a non-standard (involute or non-involute), deeply optimized high-performance gear solutions—and their increased cost in development time, custom tooling, etc.?

The advantages of the conventional gears include availability of well-described methodology, gear design software, rating standards, off-shelf tooling, etc. They can be designed by an experienced mechanical engineer who might not be an expert in gearing. These are reasons why conventional standard gears should be considered as a first possible option of a gear drive design. If it is determined that conventional gearing cannot deliver the required gear drive performance, non-standard gear geometry options should be considered. Non-standard gears should be application-specific and optimized and designed by a gear expert familiar with specialized gear design optimization software. Non-standard gears require customized—and typically more expensive—tooling that will require extended time for design and fabrication. It is important to understand that any deviation from the standard tooth proportion requires such customized tooling. Gear drives that utilize non-standard optimized gears must pass comprehensive testing to confirm their suitability and conformance. It is for these reasons that it makes no sense to do “baby steps” in gear tooth geometry optimization, in that the result will be limited performance improvement offset by significant effort and expense. Non-standard gears should be *totally optimized* to achieve maximized gear drive performance. Only then will additional design, fabrication and testing efforts and expense be justified. In the end, it just might be that a “happy medium” cannot exist in the optimization of non-standard gears.

THE QUESTION

High contact ratio (HCR) spur involute gears have long been considered an option for increasing gear drive load carrying capacity, and for reducing noise and vibration by sharing load between two and three pairs of contacting teeth. What’s more, this type of gear is addressed by most gear design and rating standards. Why don’t we see many—hardly any, in fact—applications of these promising, high-performance gears?

There are some cases of application of the high contact ratio (HCR) spur involute gears for high-performance gear drives, but they are usually considered to be non-standard gears. Unfortunately, “over-standardization” of a gear tooth geometry resulted in a “tunnel vision” that limits potential gear drive performance improvements that could be achieved by application of the HCR gears.

In addition to running AK Gears (akgears.com) **Dr. Alex Kapelevich** has produced or co-produced numerous technical articles. He is also the author of Direct Gear Design.

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Progress in Rolling Bearing Technology for Refrigerant Compressors

Guillermo E Morales-Espejel, Hans H Wallin, Rudolf Hauleitner and Magnus Arvidsson

This paper describes the latest technological solutions in rolling bearings (ball and roller) used in refrigerant compressors. First, the numerous tribological challenges faced by rolling contacts in a lubricant environment made of oil and refrigerant mixture are discussed. It is followed by a description of the even tougher conditions derived by the replacement of the more chemically stable pre-Montreal and pre-Kyoto Protocol refrigerants by the new generation of more environmental friendly refrigerants. In these conditions, rolling bearings are expected to suffer from surface distress and sometimes corrosion fatigue. Thus, attempts to model these conditions by using advanced tribological models are described. Finally, descriptions of different solutions in rolling bearings in refrigerant compressors facing challenges in lubrication and bearing life are described, all the way from traditional oil-refrigerant mixture lubrication up to the latest innovation related to oil-free lubrication, namely the pure refrigerant lubrication.

Introduction

Since the early 1990s, researchers have been developing technologies for rolling bearings in refrigerant compressors (Refs. 1–2). In this application rolling bearings generally are lubricated with a mixture of oil and refrigerant. The latest ones are not considered to be good lubricants for being very thin liquids, where its dilution reduces the viscosity of the mixture — thus increasing the compressibility and reducing the increase of viscosity with pressure (piezo-viscosity) — as compared with pure oils. At the time, considerable effort has been invested by researchers in the study of elasto-hydrodynamic lubrication (EHL) in oil-refrigerant mixtures (Refs. 3–12). Also, a considerable amount of experimental work has been carried out (Refs. 1–2, 6 and 13) to understand the effects of refrigerant dilution in lubricating oils and its consequences on bearing performance and life. It was found that conventional, all-steel bearings started to exhibit signs of inadequate lubrication at refrigerant dilution levels of 20% to 30% (Refs. 2 and 6).

In general, the main problems that rolling bearings are facing in refrigerant compressors are the effects from poor lubrication and high particle contamination (i.e. surface life). Sub-surface fatigue is very rare because the loads are moderate. Thus, for poor lubrication or high contamination conditions more robust solutions can be found in conventional rolling bearings with the help of one or more ceramic (Si_3N_4) rolling elements (Ref. 13) and/or the use of special surface steel heat treatments (e.g. nitriding and carbonitriding) (Ref. 14) and some coatings (black oxidizing and others). Even before the Montreal and Kyoto protocols (Refs. 15–16) refrigerants imposed harsh operating conditions (lower lubricant viscosities and piezo-viscosities and higher surface corrosion potential) to

rolling bearings. This led, in the past, to the research of alternative bearing designs and materials to improve the bearing operation and life under these poor-lubrication conditions. Studies showed that it was difficult to find a limiting dilution ratio for hybrid bearings, having steel rings and ceramic balls made of silicon nitride (Si_3N_4) (Ref. 17). Finally, in 1996, hybrid bearings were run in pure refrigerant with no traces of oil, and the bearings after the feasibility test were in as-new condition. This was a critical test result, which opened up the possibility to use pure refrigerant as a lubricant for special rolling bearings (PRL technology). Since then, research and application development have continued and lead to several

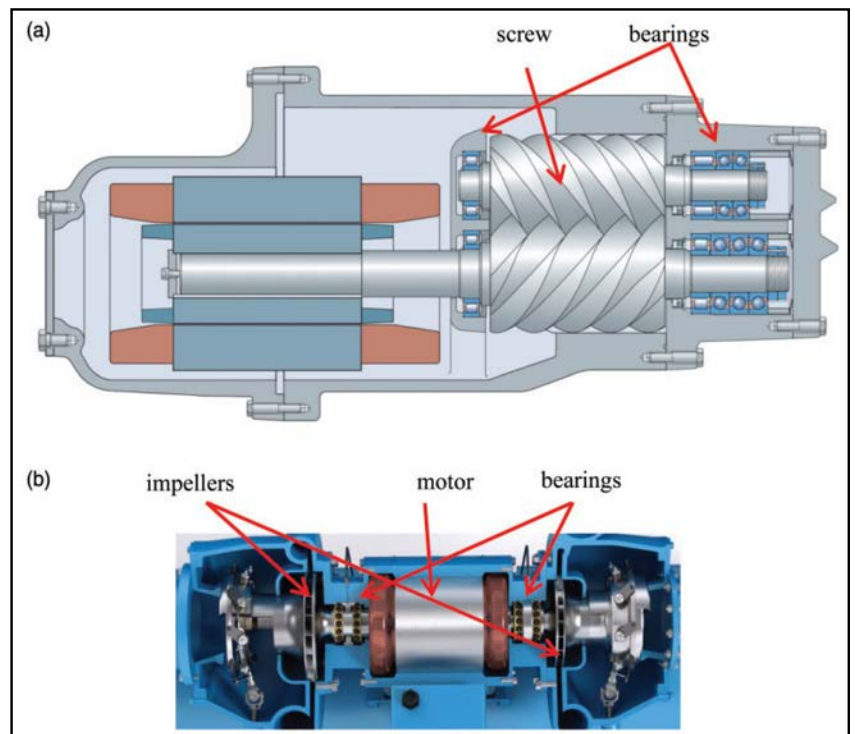


Figure 1 Refrigerant compressor types: (a) semi-hermetic screw compressor; (b) centrifugal compressor.

additional product features enabling reliable long-term operation (Refs. 17–20) currently in industrial use.

This paper summarizes the tribological challenges faced by rolling bearings in refrigerant compressors. It provides an account of modeling and experimental aspects, which prompt explanations for proven technological solutions making rolling bearings one of the most critical machine elements in the positive evolution of the refrigerant compressor technology. It will be impossible to have a complete review of the topic, since there is a very large and diverse literature database in the area of refrigerant compressors. However, the current paper will focus on the tribological aspects of rolling bearings in these conditions. The desired objective is twofold, to show the reader 1) brief application aspects of rolling bearings in these conditions, like different bearing arrangements and potential rolling bearing technologies for this application; and 2) more extensive scientific aspects on lubricant properties of oil-refrigerant mixtures and pure refrigerants and modeling aspects of surface life in rolling bearings. The described dual objective is difficult to find in the existing publications.

Description of compressor designs. There are two types of refrigerant compressors where rolling bearings play an important role for compressor function, reliability, and efficiency. These are both industrial-size compressors with a motor power range typically in excess of 50 kW.

The first type is the screw compressor (Fig. 1-a). The function of bearings in twin-screw compressors is to provide accurate radial and axial positioning of the rotors and to support rotor load. These functions are to be performed reliably, with low friction and low noise generation. With accurate positioning of the rotors, it is possible to design the compressor with small clearances for high efficiency. Radial positioning accuracy of the rotors is accomplished by using bearings with small operating clearances and high running accuracy (low run-out). Axial positioning accuracy is accomplished by small axial bearing clearance or preload.

The rotors can be supported on rolling bearings or on a combination of hydrodynamic and rolling bearings. The main advantage with rolling bearings is their small operating clearances. Rolling bearings also have lower friction than hydrodynamic bearings, require less oil for lubrication and cooling, and are less sensitive to momentary loss of lubricant and refrigerant flooding than hydrodynamic bearings. It is important that the rolling elements are separated from bearing rings by a lubricant film. If there is partial contact, then the surfaces must have high resistance to surface distress, in order to maintain bearing function over time.

The second type is the centrifugal compressor (Fig. 1-b). This compressor type is used in higher power ranges than screw compressors. The compression is accomplished by adding kinetic energy to the refrigerant in an impeller. The impeller shaft is either direct coupled to the drive motor or driven through gears. The bearings have traditionally been hydro-dynamic bearings, but a relatively recent development is to apply rolling bearings because of lower friction and improved impeller positioning accuracy. The newest technology is to use rolling bearing lubricated by the refrigerant itself, without the addition of oil.

Description of bearing types and arrangements. In screw compressors, the bearing loads are produced by gas pressure on the rotors, gear forces from input and timing gears, rotor forces from transmission of torque from one rotor to the other, and induced loads from the inertia of the rotors at startup. The loads in refrigerant screw compressors can be very high and the bearing size is limited by the space available between the two rotors, this makes the selection of bearings and bearing arrangement critical.

By using a combination of cylindrical roller and angular contact ball bearings, the loads can be separated such that the radial loads are taken by the cylindrical roller bearing and the axial loads by the angular contact ball bearing. The load sharing is accomplished by a radial gap between the outer ring of the angular contact ball bearing and the housing. This way it is not possible for the angular contact ball bearing to take radial load and the cylindrical roller bearing cannot take axial load since it is axially compliant.

There are several advantages with this arrangement. The angular contact ball bearing operates with axial load only and all balls have the same contact loads and contact angles. In this way, cage forces are minimized and load capacity is maximized. With the separation of axial and radial loads into two bearings, the load capacity of the arrangement is opti-

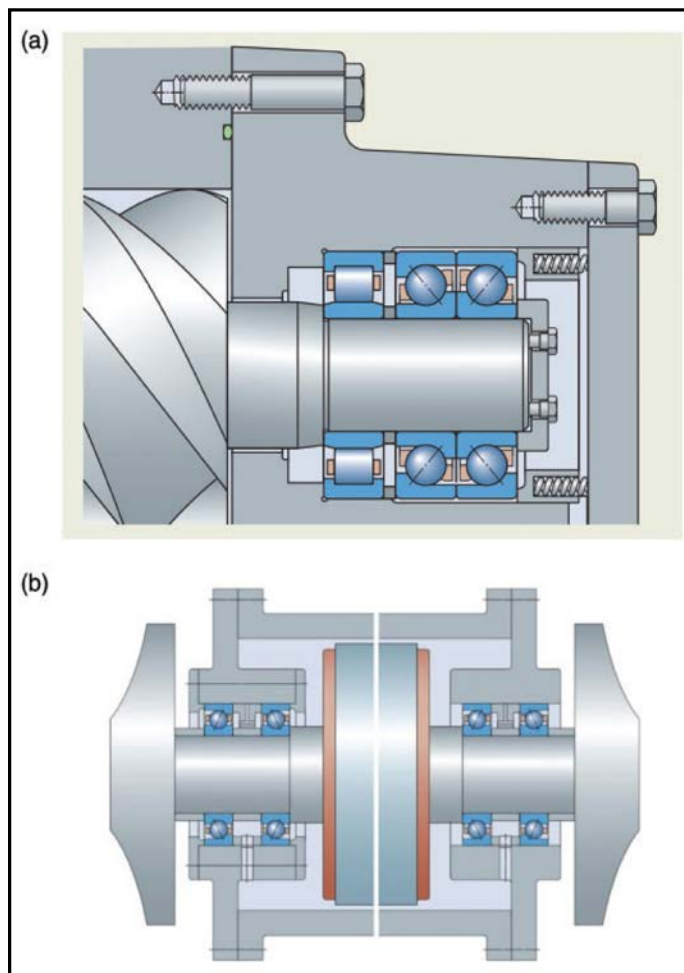


Figure 2 (a) Typical bearing arrangement for compressors; (b) two-stage compressor with back-to-back impellers positioned at opposite ends of the motor containing a bearing arrangement of two back-to-back pairs of high-speed angular contact ball bearings.

mized. With the angular contact ball bearing mounted with a light fit, it is easy to set the axial position of the rotor and the rotor end clearance. There are many ways to do this. It is also easy to dismount the angular contact ball bearing in case the rotor end clearance needs to be adjusted. A typical bearing arrangement is shown (Fig. 2-a).

In centrifugal compressors, the loads are primarily produced from pressure on the impeller and in gear-driven designs, from gear forces. Rotor weights contribute to radial bearing forces. The impeller shaft speeds are typically higher than speeds in screw compressors. The speed of compressors using low-pressure refrigerants is lower than designs using medium-pressure refrigerants. The driveline and the impeller arrangement play a significant role in bearing selection and arrangement.

In direct drive designs with two impellers back to back, the axial pressure forces from the impellers are balanced and the bearings only see the difference between the two. The radial forces are from the impeller shaft and motor rotor weights and magnetic forces on the motor rotor. The speed is typically high and the net forces are moderate. The preferred bearing arrangement is two back to back pairs of high speed angular contact ball bearings (Fig. 2-b). In direct drive designs with one or two impellers on one side (Fig. 3-a), the axial loads can be balanced with balance pistons. The radial loads are uneven, with the heavier load at the impeller end. The bearing set at this end is the locating set, which defines the axial rotor position.

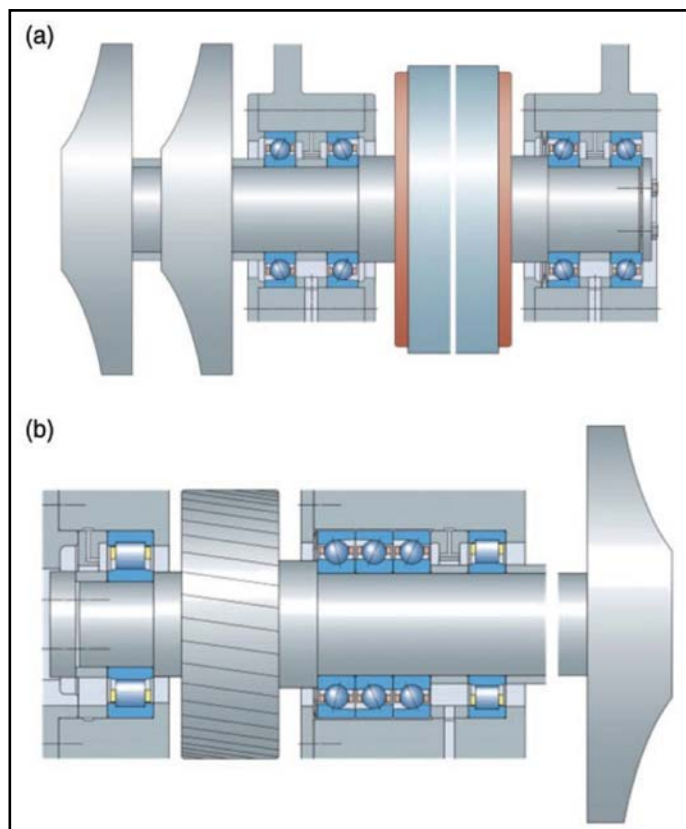


Figure 3 (a) A two-stage compressor with impeller positioned at one side of the extended motor shaft. The bearing arrangement made of two back-to-back pairs of high-speed angular contact ball bearings; (b) high-speed shaft, gear drive, tandem thrust bearings.

In gear drive designs there are two shafts, one low-speed motor shaft and one high-speed impeller shaft. In the low-speed motor shaft the gear is positioned at one end. There are axial and radial forces from the gear and an additional radial force from the motor weight. The forces on the high-speed impeller shaft can be significant, produced by the gears and pressures on the impeller. A combination of cylindrical roller and angular contact ball bearings is preferred in the arrangement (Fig. 3-b).

Description of oils and refrigerants. As refrigeration and later air conditioning were being used in the early to mid-1900s, there were many accidents with the refrigerants, such as ammonia and chloromethane, which are both toxic and flammable. When DuPont developed industrially made refrigerants branded Freon such accidents were eliminated, as the new refrigerants were much safer. The commonly used refrigerants for compressors were the low-pressure refrigerant trichlorofluoromethane — also known as CFC-11 — and medium-pressure refrigerants dichlorodifluoromethane — also known as CFC-12 — and chlorodifluoromethane — also known as HCFC-22. The reader should notice that another use of refrigerants is in foam blowing.

In the 1980s researchers found that a hole in the Earth's ozone layer was developing and the ozone depletion potential (ODP) of refrigerants started to be measured. In 1989 a global ban of ozone-depleting refrigerants was agreed upon in the Montreal Protocol.¹⁵ New HCFC and HFC refrigerants were developed having low or no ODP. Low-pressure refrigerant HCFC-123 replaced CFC-11 and medium-pressure refrigerant HFC-134a became a substitute for CFC-12 and CFC-22.

As global warming started to be a concern in the 1990s, another unintended characteristic of refrigerants got more attention—their high global warming potential (GWP). For example, HFC-134a has a GWP of 1,300 times that of CO₂. At a conference in Kyoto in 1997, an agreement on reduction of global greenhouse gases was proposed, the Kyoto Protocol (Ref. 16).

The Kyoto Protocol was not ratified by major countries and never fully implemented. At a conference in Kigali, Rwanda in 2016, it was decided to use the more successful format of the Montreal Protocol to control phase down of HFCs. This is referred to as the Kigali Amendment of the Montreal Protocol.

The newer refrigerants were not necessarily simple substitutes for CFCs. HFC-134a refrigerant, for example, is incompatible with the natural mineral oils that were normally used with CFCs. An important property of any refrigerant and oil combination is the ability to dissolve with one another. HFC-134a does not dissolve in mineral oils because of differences in molecular polarity. Polyol ester (POE) or polyalkene glycol (PAG) synthetic lubricants are necessary with this particular refrigerant. Refrigerant HFC-134a is currently used in screw compressors and centrifugal compressors of high speeds. This is one of the so-called medium pressure refrigerants, whose phase down is governed by the Kigali Amendment. The refrigerant HCFC-123 is a low-pressure refrigerant and it is still used mainly in centrifugal compressors. It is being phased out in accordance with the Montreal Protocol.

Later-generation refrigerants for compressors. New refrigerants with low GWP and zero ODP are now being de-

veloped and phased in by the global refrigeration and air-conditioning industries. The most promising new refrigerants are low-pressure refrigerant HCFO-1233zd and medium-pressure refrigerants HFO-1234ze and HFO-1234yf and various blends containing these refrigerants. The GWP of the new refrigerants is less than five and the ODP is zero. Natural refrigerants such as ammonia and CO₂ are also used increasingly but have limitations with toxicity, flammability, and high pressure.

The new refrigerants have a very short atmospheric lifetime, thus they are chemically very active, which can bring corrosion or material incompatibility with the rolling bearings, adding an extra element in the equation. Besides all this, the thermodynamics of the new refrigerants make the oil-refrigerant mixtures more prone to having high concentrations of refrigerants (30% or higher) during important times of the operation cycle.

Lubrication with Refrigerants

In all refrigerant compressors (except with magnetic bearings) the refrigerant is part of the lubrication of the bearings; it can be as a oil-refrigerant mixture or as pure refrigerant lubrication (PRL). In any case, the knowledge of the lubricating properties of the refrigerants is needed. Lubrication and tribological aspects are, of course, an important element in the estimation of the bearing life.

Refrigerant lubricating properties. Any EHL film thickness calculation formula or procedure requires of the following lubricant properties: 1) dynamic viscosity, η_0 at atmospheric conditions (for refrigerants: saturation conditions); 2) the piezo-viscosity coefficient a or its integrated form α^* , and 3) the compressibility or density function with pressure, $p=f(p)$. Then, the tribological performance of the lubricant is measured by other important parameters besides film thickness, not always exclusive of the lubricant: 1) boundary friction coefficient, μ_{bl} , and 2) lubricant rheology (or fluid film friction), $\mu=f(j)$. There are certainly other properties that might have some effect in the performance, such as thermal transport properties, but they will not be discussed here.

The film thickness properties, can in general, be measured by a high pressure viscometer and a density meter. Jacobson and Morales-Espejel (Ref. 21) Laesecke and Bair (Ref. 22) and Vergne et al (Ref. 23) gave film thickness properties for some of the currently used refrigerants in refrigerant compressors. It can be seen that the refrigerants show relatively low piezo-viscosity values, which does not favor the build-up of thick lubricating films. A summary of literature values of some film thickness properties of known refrigerants is given in Table 1.

For comparison consider typical oil values of $\eta_0 \approx 0.02$ Pas (20 cSt oil) and $\alpha^* \approx 20$ GPa⁻¹.

Friction properties are difficult to measure in pure refrigerants and there is no extensive literature on the subject. However, in Morales-Espejel et al (Ref. 19) boundary friction coefficients of $\mu_{bl} \approx 0.07$ were reported for HFO-1233zd in a hybrid contact (steel-Si₃N₄). Rheological (non-Newtonian) properties and fluid film friction have not been reported to

Refrigerant	Temp. (°C)	η_0 (mPas)	α^* (GPa ⁻¹)	$\bar{p} = f(p)$ (-)	Reference
HCFC-123	25	0.4	6.4	Equation (4) in reference	Vergne et al. ²³
HFC-134a	40	0.2	5.0	Equation in reference [22]	Laesecke and Bair ²² and Bair ²⁴

the knowledge of the authors. Full EHL numerical solutions for film thickness using refrigerants have been reported in Morales-Espejel et al (Ref. 19) and Vergne et al (Ref. 23) where central and minimum film thickness have been approximated to well-known or specialized film thickness equations.

Oil-refrigerant mixtures. Many studies have been carried out in the past to measure or to estimate the film thickness in EHL contacts lubricated with oil-refrigerant mixtures (Refs. 3-5,7-12). More recent studies of oil-refrigerant mixtures in EHL are in Bair et al. (Refs. 24-25) and Tuomas (Ref. 26). In the calculation of film thickness in EHL contacts lubricated with oil-refrigerant mixtures the estimation of the mixture properties is a key intermediate step. Diverse equations have been suggested to estimate viscosity and piezo-viscosity of oil-refrigerant mixtures. For example, Akei and Mizuhara (Ref. 9) uses the Eyring theory to derive equations for the piezo-viscosity coefficient and viscosity:

$$\alpha_{mix} = \frac{m s_{ref} (\alpha_{ref} - \alpha_{lub})}{s_{ref} (m - 1) + 1} + \alpha_{lub} \quad (1)$$

$$\log(\eta_{mix}) = (\log \eta_{ref} - \log \eta_{lub}) \left(\frac{m s_{ref}}{(m - 1) s_{ref} + 1} \right) + \log \eta_{lub} \quad (2)$$

where $m = M_{lub}/M_{ref}$, with M being the molecular weight of the component.

In an example (Ref. 24) with polyol ester oil (POE) and HFC-134a, a value of $m = 4.5$ is reported. In Bair (Ref. 24) a different mixture law is used for the viscosity; however, it also depends on the ratio m . The molecular weight of oil varies with oil viscosity and oil type. For example, the molecular weight varies between 700 and 1,100 g/mol for typical POE compressor oils of different viscosities. The molecular weight of refrigerants is typically 100 to 150 g/mol. As explained in Bair (Ref. 24), the molecular weight of the oil can be calculated backwards with Equation 2 if measurements of viscosity of the mixture at different dilution rates are known. Equations 1 and 2 can be plotted for different refrigerant dilution rates and molecular weight ratios (Fig. 4).

Now, in rolling bearings the lubrication quality parameter (κ) is used as a measure of the lubrication condition in the bearing. This parameter is defined as the ratio between the actual viscosity used in the bearing at the working temperature and the required viscosity recommended by the manufacturer; it is properly defined and explained in ISO 281:2007. The required viscosity is a parameter given by the bearing manufacturer and will not be changed at this point (next, when bearing life is discussed, this assumption can be relaxed). Thus the only parameter that remains to be estimated for a proper calculation of κ in oil refrigerant mixture conditions is the actual viscosity of the lubricant. Thus, Equation 2 can be used for this purpose.

For bearing life calculations, Meyers (Ref. 2) recommends

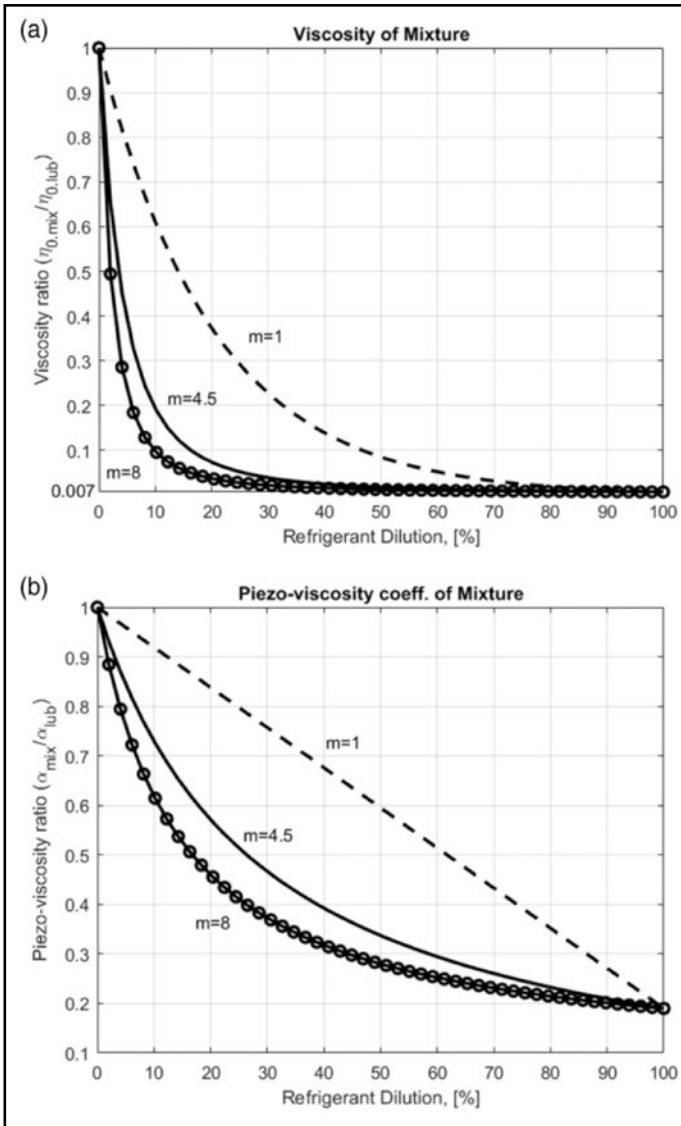


Figure 4 Variation of (a) viscosity and (b) piezo-viscosity coefficient in a mixture of oil and refrigerant as a function of the refrigerant dilution and the molecular weight ratio (m) between the oil and the refrigerant, following equations (1) and (2).

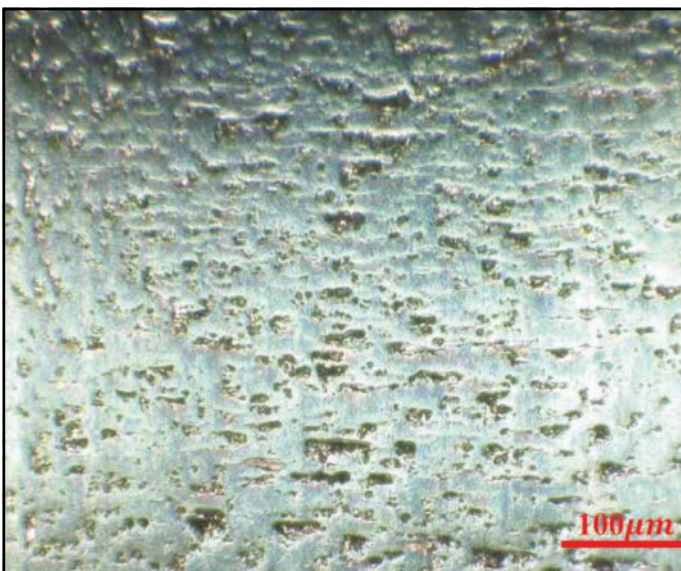


Figure 5 Surface distress in a raceway bearing surface.

to increase the required kinematic viscosity of the bearing (ν_1) calculated as oil by multiplying it with a factor $f=3$, for HFC-134a and POE oil. This is to account for the reduction in piezo-viscosity plus other un-accounted effects (e.g. lubricity reduction and chemical aggressiveness) in the actual viscosity estimated with Equation 2. For extremely compressible refrigerants an extra increase factor should be derived for correction in the final calculated κ . Then the bearing life can be calculated with this final adjusted value of κ .

Pure refrigerant lubrication. For centrifugal compressors, the newest trend is to remove oil completely from the lubrication system, either by using magnetic bearings or special hybrid bearings lubricated with refrigerant, this latest one is the so-called pure refrigerant lubrication or PRL technology. For screw compressors the oil will still be required to lubricate the screws. In PRL, the rolling bearings are lubricated only with the refrigerant, 17–20 so there is no oil or its contents is so low that it does not account in the lubricant film building-up properties of the refrigerant (e.g. contaminant). For rolling bearings (ball and roller) to work in these conditions, special materials and design are required; a full description of these is given in Morales-Espejel et al (Ref.20). In summary, the main required features are:

PRL hybrid ball and roller bearings are made of:

- The highest quality bearing grade silicon nitride (Si_3N_4) rolling elements with the most stringent defect inspection procedure.
- Cage with a strong design shape and fiber-reinforced PEEK material.
- Rings made of stainless high-nitrogen, through-hardened steel with chemical composition as described in Ragen et al. (Ref. 27), heat treated according to in-house specifications and ground to super-finished raceways in processes developed in-house.

Apart from this, in the bearings the refrigerant flow has to be directed in a particular way with thermodynamic conditions that ensures refrigerant liquid at all times with sufficient amount.

The work described in Morales-Espejel et al (Refs.19–20) shows that with the use of the refrigerant viscosity and piezo-viscosity values, the film thickness can be approximated with use of the well-known Hamrock and Dowson film thickness equation (Ref.28) — at least for the following low-pressure refrigerants: HFO-1233zd and HCFC-123. Laesecke and Bair (Ref.22) gave recommendations for film thickness estimations with the medium-pressure refrigerant HFC-134a. Other refrigerants remain to be characterized.

Rolling Bearing Surface Distress

The typical limitation in refrigerant compressor bearings, where usual maximum Hertzian pressures are below 2 GPa, is rather surface-related problems, like poor lubrication and contamination, which can bring wear and/or surface distress (known as well as micropitting). Other surface failure modes are also possible e.g. smearing (or adhesive wear); however, it is limited in these applications since in general no large accelerations are experienced. Thus, most of the effort invested to increase the life of the bearings goes to preventing or delaying surface distress from poor lubrication conditions. This

is why the authors have included a special section regarding this topic in this paper. The authors also included a modeling example for indications of surface distress occurrence and ways of mitigation. Surface distress is thus strongly influenced by high dilution rates of refrigerant, low viscosities in the mixture, low piezo-viscosity, and high compressibility of the refrigerants, plus sometimes some corrosion due to the chemical activity of the refrigerants or the combination with moisture. In Morales-Espejel et al (Refs. 29–30) a bearing life model that separates surface from subsurface is described; in future this model should lead to much better handling of life calculations in these types of applications.

Surface distress phenomenon. Micropitting is a term widely used by the gear industry to describe microsurface spalls and cracks, which sometimes appear on the surface of rolling-sliding contacts. ISO 15243 (Ref. 31) refers to this damage or failure mode as surface distress or surface initiated fatigue, i.e. the failure of the rolling contact metal surface asperities under a reduced lubrication regime and a certain amount of sliding motion causing the formation of 1) burnished areas (glazed; grey stained); 2) asperity microcracks, and 3); asperity microspalls (Fig. 5). All of this will be described herein using the term surface distress. It is well-known that surface distress is the result of the competition between surface fatigue and mild-wear (Refs. 32–33). Mild-wear (wear at asperity level) modifies the initial topography reducing the asperity heights and also removes fatigued surface layers refreshing the exposed material, thus surface fatigue is delayed.

Surface distress modeling. Before testing solutions for a bearing application, often modeling is required to shorten the development time and study the influence of different parameters in the application, lubricant, and bearing. One of the current authors has developed a numerical model for surface distress (Ref. 34) in rolling bearings that can be applied for the present case of oil-refrigerant mixtures. The flowchart of it is shown in Figure 6; the model is described in detail in the reference and will not be presented here; only a short summary will be given. The model requires 3-D digitized roughness samples of the two contacting surfaces (inner- or outer-ring raceway and roller or ball). In general several samples are obtained, so several simulations with the different roughness samples representing different raceway locations are required to obtain a fair estimation of life or damage under given conditions. Here, since the focus is only an illustrative case, only one measurement for the inner ring raceway and one measurement for the roller are considered and used at all times.

The modeling of the interactions surface-fluid (pressure and surface shear stress) is resolved via a 2-D transient Reynolds equation for the fluid and half-space elasticity for the solid. The scheme simulates the relative movement of the microgeometry inside the contact in time for ev-

ery load cycle. The simulation process begins with the virgin surfaces with no accumulated damage. Once both rough surfaces enter into the contact in the first cycle, the partial transient EHL problem is solved (Ref. 35) for the different time steps (typically 10) until exit from the contact. In that time the local pressure on the roughness follows the Hertzian parabola as it travels within the contact, as explained (Ref. 34). The topography is updated only after a certain number of load cycles (in this case 40 times during the whole simulation time) to reduce the computational time (instead of every single load cycle). Thus the fatigue damage criterion, e.g. — Dang Van (Refs. 36–37) is applied for the current cycle in the points of the domain where the criterion is not exceeded ($d < 1$) the damage is accumulated according to a damage accumulation law (e.g. Palmgren–Miner) for the current accumulated number of cycles. After this, if the local accumulated damage parameter d exceeds 1, a micropit will be created locally by simply removing the material where $d > 1$ and above this location. Basically what happens here is that locally the fatigue strength of the material has been exceeded and it cannot withstand more fatigue cycles. The actual morphology of a crack is more complex than this, and there is a phase of generation and a phase of propagation, as explained in Morales-Espejel and Brizmer (Ref. 34). The surface topographies

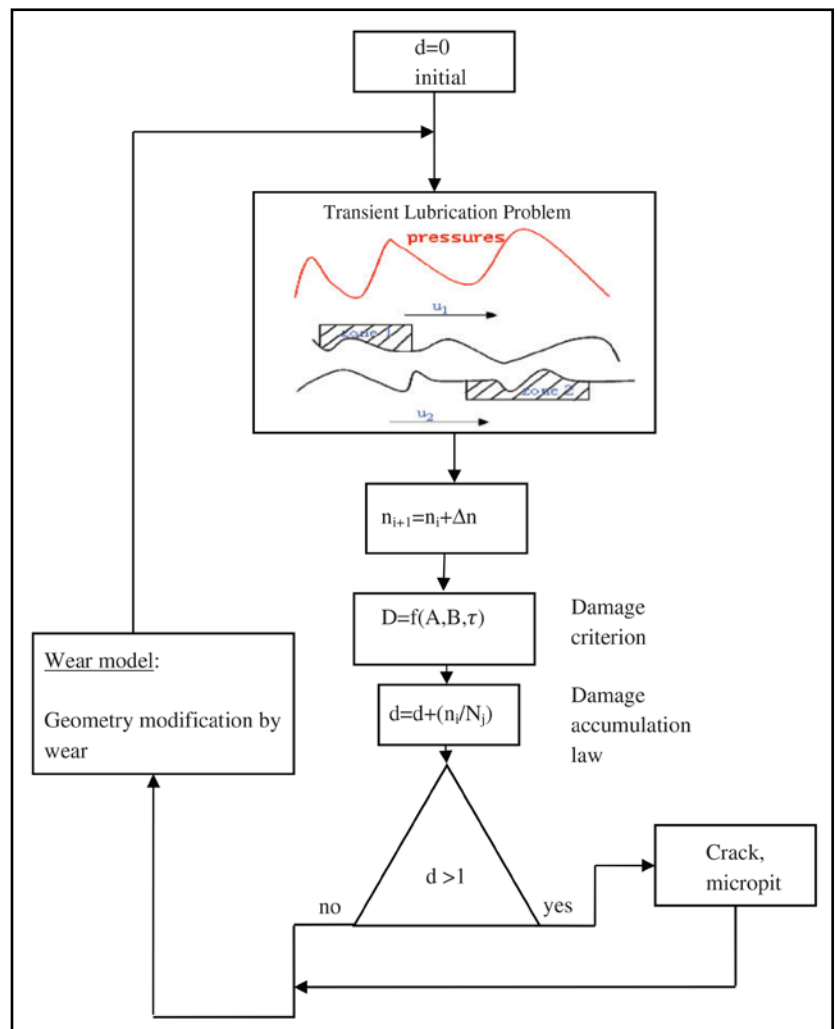


Figure 6 Flowchart of the surface distress model (Ref. 34).

then are fed into a lubricated wear model, which will modify the asperities. The calculation process restarts and continues until the number of desired simulated load cycles is reached. At the end of the simulation, the modified surface topography is obtained together with the accumulated damage map on the surface and below.

Calculation case for enhanced surface material.

As an application example of detailed modeling aspects in rolling bearing lubrication and surface strength a case is presented next. Considering a screw compressor application working at low speed, with the operating conditions summarized in Table 2.

The Dang Van (Refs. 36–37) fatigue criterion used by the model requires the two Wöhler curve constants (Ref. 38) for the surface material used in the bearing. Here the bulk values provided in Shimizu et al. (Ref. 39) are used corresponding to hardened bearing steel ASTM 52100, $A = -43.0$ MPa, and $B = 1,220$ MPa. The results of the simulation with the data of Table 2 and these fatigue properties, together with the roughness samples of Figure 7, are shown in Figure 8 with the areas marked in red as surface distress — or about 2.32% of the total area. By comparing Figures 7-b and 8 it can be seen that the surface distress on the raceway surface develops faster in the areas closer to roughness depressions, more precisely in the areas of higher pressure slopes like in the borders of grinding mark valleys (Ref. 40). Figure 9 shows some intermediate results in a given simulation time (snap shot) of deformed raceway roughness and hydrodynamic pressure fluctuations generated when the roughness sample is inside the rolling bearing most heavily loaded contact.

Bearing bore diameter	40 mm
Bearing type	Cylindrical roller bearing
Mixture viscosity at temp.	10 cSt
Max. Hertzian pressure (most loaded contact)	1.2 G Pa
Lambda ratio	0.88
Rotating speed (inner ring)	800 r/min
Running time	30 years
Load cycles	8×10^{11}

Next to consider is that the standard bearing is replaced by a bearing with the HN (carbonitriding) heat treatment, as suggested in Gabelli et al (Ref. 13), showing higher surface hardness after running-in (which will reduce the wear by up to 45% of the previous value) and will increase the bearing surface fatigue strength sufficient to double the life of the bearing. Thus, emulating the experimental results, by running the present model it is found that an increase in the Wöhler constant B of 22% will double the life of the bearing. Therefore the selected constants for the new surface treatment HN are $A = -43.0$ MPa, $B = 1,488$ MPa, together with a reduction of the wear coefficient by a multiplication factor of 0.45. All other parameters of Table 2 remain unchanged; thus the new results from the model are summarized in Figure 10. This figure shows a very substantial reduction of the surface distress area to negligible values (only 0.38%), despite the reduction of mild wear of the harder surfaces. It can be seen that with the use of advanced tribological modeling the conditions of poor lubrication in rolling bearings can be simulated, including the screening of alternative solutions.

For the case of contamination, the same model can be used to determine the sensitivity of the number and geometry of the indentations, and substantial work has been carried out in this direction by some of the current authors (Refs. 30, 40). Another effective solution for poor lubrication problems is the use of full or semi-hybrid bearings (steel rings and all or a few rolling elements made of ceramic material, Si_3N_4). The same model has been used (Ref. 41) to explain how a hybrid bearing, by reducing its boundary friction coefficient, can reach much better performance under poor lubrication conditions than all-steel bearings. The predictions of the model were also confirmed by experiments.

Corrosion-enhanced fatigue.

It was discussed above that the most recent refrigerants being explored as possible substitutes of the current ones, for reducing even further the environmental impact, have the tendency to exhibit shorter atmospheric lifetime.

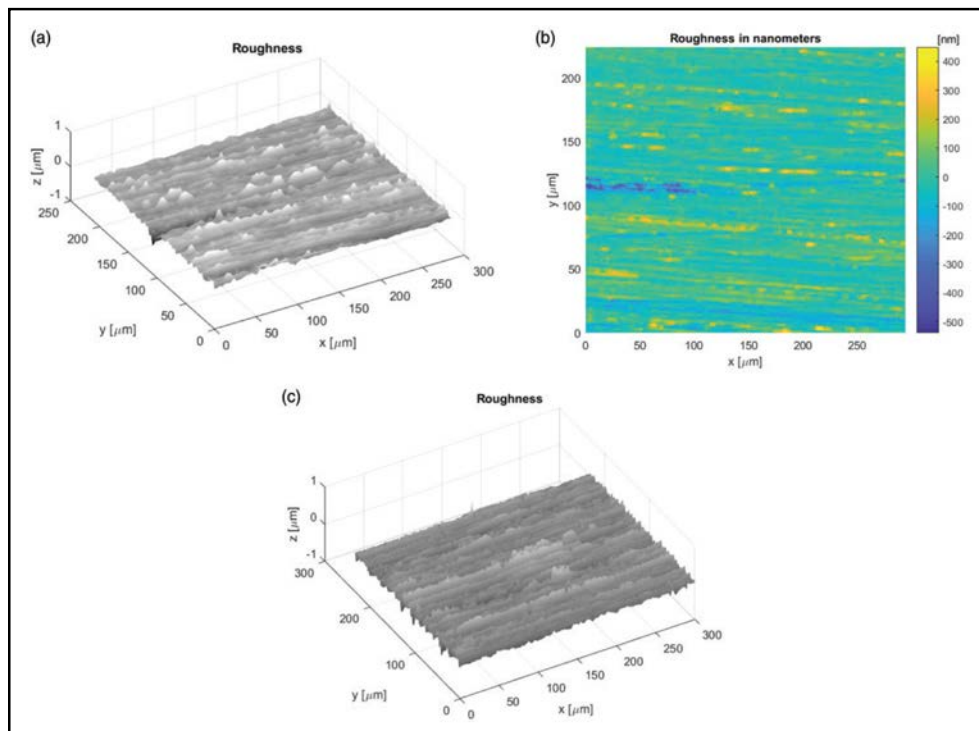


Figure 7 Roughness sample of the rolling bearing inner ring raceway and roller: (a) 3D view of a spot on the inner-ring raceway $R_q = 0.068 \mu\text{m}$; (b) normal view of a spot on the inner-ring raceway $R_q = 0.068 \mu\text{m}$; (c) 3D view of a spot on the roller $R_q = 0.090 \mu\text{m}$.

In some cases it means more chemically reactive compounds and often more corrosion of steel or bearing organic components (e.g. seals, cage, etc.) and even ceramic material. In addition to this, frequent changes of temperature experienced by some parts of the compressors could favor the ingress of moisture in the bearings, and the corrosion environment in the bearings can then be worsened. Therefore it is expected that a new challenge for rolling bearings in refrigerant compressors is the exposure to some degree of corrosion. Very mild corrosion will manifest only as a noticeable increase of wear or mild wear in the bearing. If corrosion is important, then some corrosion pitting can appear, which can be distinguished from the “mechanical” micropitting or surface distress discussed above by careful examination of the features under a microscope. It is believed that this corrosion pitting not only will impact the local stress concentration, but also could weaken the surface fatigue strength of the surface. One semi-empirical way to model it using the above surface distress model is by reducing the value of the Wöhler constant B to a value that matches the observations. Otherwise, the actual corrosion phenomenon would need to be modeled.

In PRL conditions with the presence of hybrid bearings, in order to reduce the risk of corrosion some additive or preservative can be used in the bearing or lubrication system to protect the bearing surfaces from the chemical attacks. Special stainless bearing material (as described in the PRL section) already provides some good protection against corrosion. Besides, proper earthing systems for the area where the bearings are located in the compressor avoids electrostatic potentials from the friction between the refrigerant and the moving parts of the bearings. Otherwise, the build-up of electrostatic charges might favor corrosion attack on the bearing surfaces. Finally, to reduce the risk of galvanic corrosion (bimetallic corrosion) the use of brass and other metallic cages should be minimized or used with caution (Ref.42).

Technological Solutions

Surface treatments in conventional rolling bearings. For oil-refrigerant mixtures under poor lubrication and/or high contamination, a heat treatment that has large potential for rolling bearings is nitriding and carbonitriding. The latest one is an added surface heat treatment with the addition of ammonia to the standard gas carburization process. This introduces N and C atoms diffusing into the surface steel, improving the fatigue strength and reducing wear. Carbonitrided (HN code) (Ref. 13) bearings have been tested and compared with normal heat-treated bearings in harsh contaminated conditions. Wear rate was reduced to about 45% of the normal bearings and bearing life was roughly doubled for those particular test conditions. Also, the surface distress model was used to simulate the behavior and provide understanding.

As explained in Gabelli et al. (Ref. 13) for standard (ASTM 52100) hardened and tempered steel bearings, the permitted maximum content of retained austenite is 15 vol.% and the diameter of the residual carbides is usually 1 μm or less. The hardness is in general above 62 HRC. The carbonitriding surface hardening of the SAE 52100 rings produces a structure at the ground surface that consists of martensite strengthened

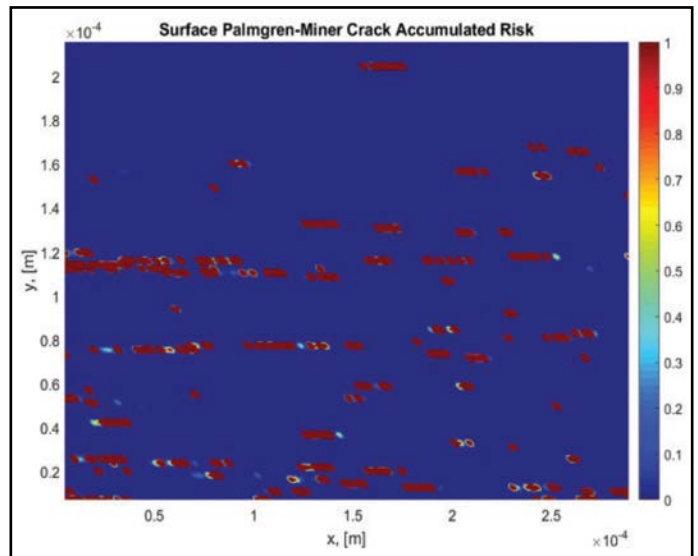


Figure 8 Surface distress area, as predicted by the model ($A_p = 2.32\%$) on the bearing raceway for hardened bearing steel ASTM 52100.

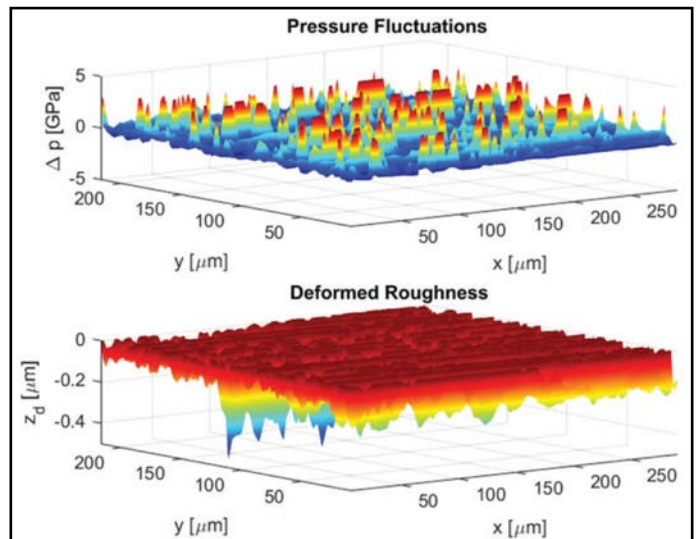


Figure 9 Example of deformed bearing raceway roughness (bottom) and EHL pressure fluctuations (top) in a given simulation time, as calculated by the model.

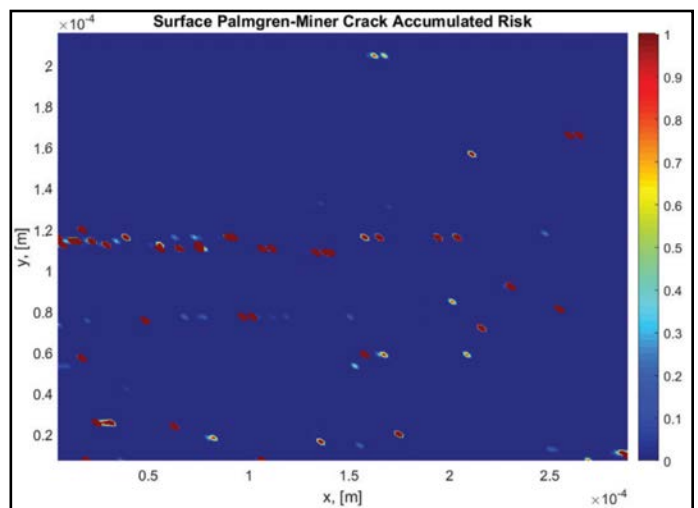


Figure 10 Surface distress area, as predicted by the model ($A_p = 0.38\%$) for the surface treatment HN (carbonitriding).

by carbon and nitrogen, but with a higher retained austenite content than in a standard through hardened bearing ring, and a proportion of enlarged carbides with diameters of more than 2 to 3 μm. Along this, there is a positive modification of the surface residual stresses. All this substantially improves the strength of the surface by providing toughness and higher wear resistance.

Semi-hybrid rolling bearings. It has been verified by testing (Ref. 14) that steel rolling bearings containing at least one rolling element made of ceramic (Si_3N_4) improve substantially their performance under poor lubrication and high contamination conditions. Therefore the next line of defense after nitriding or carbonitriding for lubrication with oil-refrigerant mixtures should be this type of rolling bearing, i.e. — so-called “semi-hybrid.” The working mechanism is similar to hybrid bearings as already explained in Brizmer et al (Ref. 41) and Vieillard et al (Ref. 43).

Coatings. In-house experiments have proven that most of the porous coatings flake-off in the presence of high concentrations of refrigerant within the lubricant (for sure in PRL conditions). Since the refrigerant tends to penetrate the coating through the pores to reach the steel surface, sometimes it brings about corrosion on the steel surface below the coating. Thus, if a coating has to be chosen in harsh oil-refrigerant lubrication conditions, it is recommended to select a sacrificial coating that will mainly help to run-in the surfaces, thereby improving the tribology and smearing itself onto the steel surfaces. In this way it will also protect them from mild chemical attacks (mild corrosion) rather than pretending to be staying untouched and covering uniformly the steel surface. This is the case for black oxide coating (Ref. 44), which could be a rolling bearing performance improver in mild severity conditions of poor lubrication with oil-refrigerant mixture. Note that zinc-coated bearings are an alternative to avoid corrosion. In milder refrigerant dilution rates other coatings can be used, providing the bearings with some protection. This is the case for special DLC (NoWear) with and without porosity seal, made to endure poor lubrication conditions and to fight adhesive wear. Other anti-wear solutions include manganese phosphate coating and titan molybdenum disulfide coating (or triple M) with very low friction coefficients.

Discussion

This paper attempts to summarize the current rolling bearing technology status for the application of refrigerant compressors, where the lubrication with an oil-refrigerant mixture is common practice. It is observed that all-steel bearing technologies applied up to date for the present refrigerants and typical dilution rates up to 20% refrigerant content are being challenged by more chemically active post-Kyoto protocol new refrigerants and dilution rates higher than 30% in parts of the duty cycle. Besides this, a new technology pioneered by the current authors which allows

the rolling bearings to be lubricated solely with refrigerant (the so-called PRL technology) is observed to be developing well in the industry.

As a graphical summary of the application of the different rolling bearing technologies, the authors propose the schematics of Figure 11 to describe the possible usage ranges and technology sophistication level of the different rolling bearing technologies discussed in the current paper as a function of the refrigerant dilution rate in the oil.

Notice that the ordinate in Figure 11 (bearing technology level) represents a qualitative scale where no values are given. But it shows, in a relative level and according to the experience of the authors, the technology degree that is required for every product to be successful and reliable.

Conclusions

Different compressor designs and bearing arrangements were briefly discussed. The main failure mode of rolling bearings lubricated with refrigerant-oil mixture (surface distress) was discussed in detail. A model to predict surface distress from poor lubrication conditions was explained with the application to carbonitrided bearings. Some aspects of corrosion fatigue were mentioned and lubrication aspects of oil-refrigerant mixtures were discussed in detail with emphasis on film thickness estimation and effects on bearing life. Additionally, PRL technology is summarized and different technological solutions to mitigate the old and new challenges to rolling bearings lubricated with oil-refrigerant mixtures are discussed.

The following conclusions can be drawn from this study:

- Current all-steel rolling bearing technology is now challenged by the arrival of new, more environmentally friendly refrigerants in the industrial compressor sector that deliver higher refrigerant dilution rates and more chemically active compounds.
- As a response to this challenge, new rolling bearing technologies are proposed, such as the use of special heat

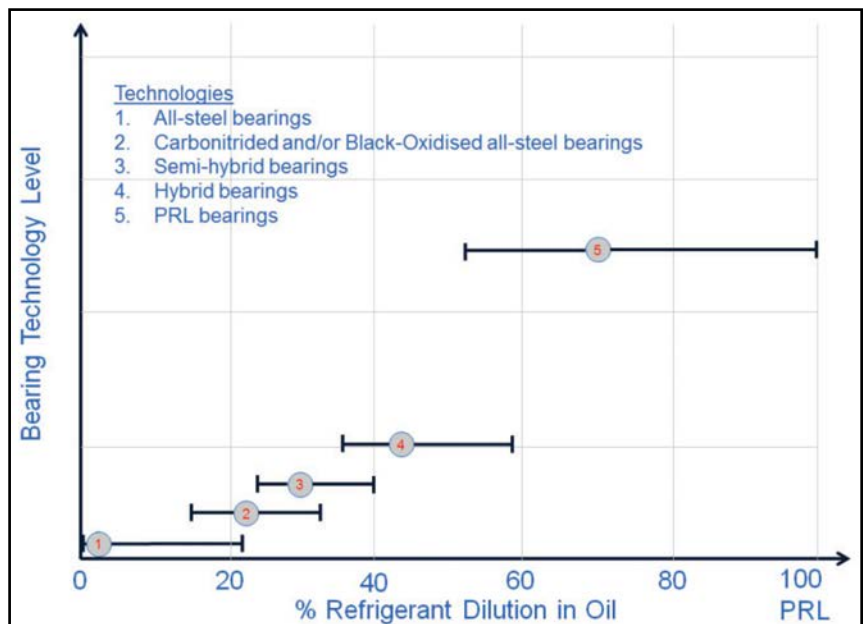


Figure 11 Proposed rolling bearing technology applicability as a function of the refrigerant dilution rate in the oil. PRL: pure refrigerant lubrication.

treatments, semi-hybrid, and hybrid rolling bearings.

- For the more advanced technological challenge of PRL, special hybrid bearings have proved successful in industry.
- Better understanding of the lubrication mechanisms and failure inception via modeling, experiments, and testing is proposed to face the new challenges of rolling bearings in refrigerant compressors. **PTÉ**

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Appendix

Notation

- A* Wöhler constant for the S-N curve slope (Pa)
- A_p* area ratio with surface distress (%)
- B* Wöhler constant for the S-N curve intercept (Pa)
- d* accumulated damage parameter for fatigue, $0 \leq d \leq 1$
- D* current cycle damage parameter for fatigue, $0 \leq d \leq 1$
- f* safety multiplication factor for the required viscosity in oil-refrigerant mixture
- p* pressure (Pa)
- n_i* accumulated number of load cycles
- N_i* number of load cycles to failure under current load
- m* molecular weight ratio between oil and refrigerant
- M* molecular weight of a substance (kg/kmol)
- u₁* contact surface 1 velocity (m/s)
- u₂* contact surface 2 velocity (m/s)
- R_q* r.m.s. of the roughness (μm)
- S* fraction content in the mixture
- x* coordinate, rolling direction (m)
- γ* coordinate, transverse to rolling direction (m)
- z* coordinate, normal direction to contact (m)
- z_d* contact clearance (μm)
- α* Piezo-viscosity coefficient of lubricant (GPa⁻¹)
- α** integrated piezo-viscosity coefficient,

$$\alpha^* = \left[\int_0^\infty \frac{\eta(p=0)}{\eta(p)} dp \right]^{-1} (\text{GPa}^{-1})$$
- $\bar{\rho}$ density ratio, $\bar{\rho} = \rho / \rho_o$
- ρ* density (kg/m³)
- Δn load cycle increment
- η* dynamic viscosity (Pas)
- η_o* dynamic viscosity at atmospheric conditions (oil) or at saturation conditions (refrigerant) (Pas)
- $\dot{\gamma}$ sliding rate (s⁻¹)
- κ* lubrication quality parameter for rolling bearings, $\kappa = v/v_1$
- η* friction coefficient
- v* actual kinematic viscosity used in the bearing (cSt)
- v₁* required kinematic viscosity of the lubricant (cSt)
- ρ_o* lubricant density at atmospheric conditions (oil) or at saturation conditions (refrigerant) (kg/m³)
- τ* Dang Van stress field (Pa)

Subscript

- bl* boundary lubrication
- lub* lubricant (oil)
- ref* refrigerant
- 1 contact surface 1
- 2 contact surface 2

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Baldor Motor Basics — Part 9

Amps, Watts, Power Factor and Efficiency

Approximate Load Data from Amperage Readings

Power Factor Correction on Single-Induction Motors

Edward H. Cowern, P.E.

(A continuing series of articles, courtesy of the Baldor Electric Co., dedicated primarily to motor basics; e.g. — how to specify them; how to operate them; how — and when — to repair or replace them, and considerably more.)

Introduction

There seems to be a great deal of confusion among the users of electric motors regarding the relative importance of power factor, efficiency and amperage, as related to operating cost. The following information should help to put these terms into proper perspective.

At the risk of treating these items in reverse order, it might be helpful to understand that in an electric bill, commercial, industrial or residential, the basic unit of measurement is the kilowatt hour. This is a measure of the amount of energy that is delivered. In many respects, the kilowatt hour could be compared to a ton of coal, a cubic foot of natural gas, or a gallon of gasoline, in that it is a basic energy unit. The kilowatt hour is not directly related to amperes, and at no place on an electric bill will you find any reference to the amperes that have been utilized. It is vitally important to note this distinction. You are billed for kilowatt hours: you do not necessarily pay for amperes.

Power factor. Perhaps the greatest confusion arises due to the fact that early in our science educations, we were told that the formula for watts was amps times volts. This formula, $\text{watts} = \text{amps} \times \text{volts}$, is perfectly true for direct current circuits. It also works on some AC loads such as incandescent light bulbs, quartz heaters, electric range heating elements, and other equipment of this general nature. However, when the loads involve a characteristic called inductance, the formula has to be altered to include a new term called power factor. Thus, the new formula for single phase loads becomes, $\text{watts} = \text{amps} \times \text{volts} \times \text{power factor}$. The new term, power factor, is always involved in applications where AC power is used and inductive magnetic elements exist in the circuit. Inductive elements are magnetic devices such as solenoid coils, motor windings, transformer windings, fluorescent lamp ballasts, and similar equipment that have magnetic components as part of their design.

Looking at the electrical flow into this type of device, we would find that there are, in essence, two components. One portion is absorbed and utilized to do useful work. This portion is called the real power. The second portion is literally borrowed from the power company and used to magnetize the magnetic portion of the circuit. Due to the reversing

nature of AC power, this borrowed power is subsequently returned to the power system when the AC cycle reverses. This borrowing and returning occurs on a continuous basis. Power factor then becomes a measurement of the amount of real power that is used, divided by the total amount of power, both borrowed and used. Values for power factor will range from zero to 1.0. If all the power is borrowed and returned with none being used, the power factor would be zero. If on the other hand, all of the power drawn from the power line is utilized and none is returned, the power factor becomes 1.0. In the case of electric heating elements, incandescent light bulbs, etc., the power factor is 1.0. In the case of electric motors, the power factor is variable and changes with the amount of load that is applied to the motor. Thus, a motor running on a work bench, with no load applied to the shaft, will have a low power factor (perhaps .1 or 10%), and a motor running at full load, connected to a pump or a fan might have a relatively high power factor (perhaps .88 or 88%). Between the no load point and the full load point, the power factor increases steadily with the horsepower loading that is applied to the motor. These trends can be seen on the typical motor performance data plots which are shown (Fig. 1).

Efficiency. Now, let's consider one of the most critical elements involved in motor operating cost. This is efficiency. Efficiency is the measure of how well the electric motor converts the power that is purchased into useful work. For example, an electric heater such as the element in an electric stove, converts 100% of the power delivered into heat. In other devices such as motors, not all of the purchased energy is converted into usable energy. A certain portion is lost and is not recoverable because it is expended in the losses associated with operating the device. In an electric motor, these typical losses are the copper losses, the iron losses, and the so-called friction and windage losses associated with spinning the rotor and the bearings and moving the cooling air through the motor.

In an energy efficient motor, the losses are reduced by using designs that employ better grades of material, more material and better designs, to minimize the various items that contribute to the losses in the motor.

For example, on a 10 hp motor, a Super-E energy efficient design might have a full load efficiency of 92.4%, meaning that, at full load (10 hp), it converts 92.4% of the energy it receives into useful work. A less efficient motor might have an efficiency of 82%, which would indicate that it only converts 82% of the power into useful work.

Typical performance - not guaranteed values.

15 HP 3 PH 60 HZ 1765 RPM 460 V 0936M
 TORQUES (LB-FT): P0=165 PU=68.2 LR=79.1 LRA=122.9

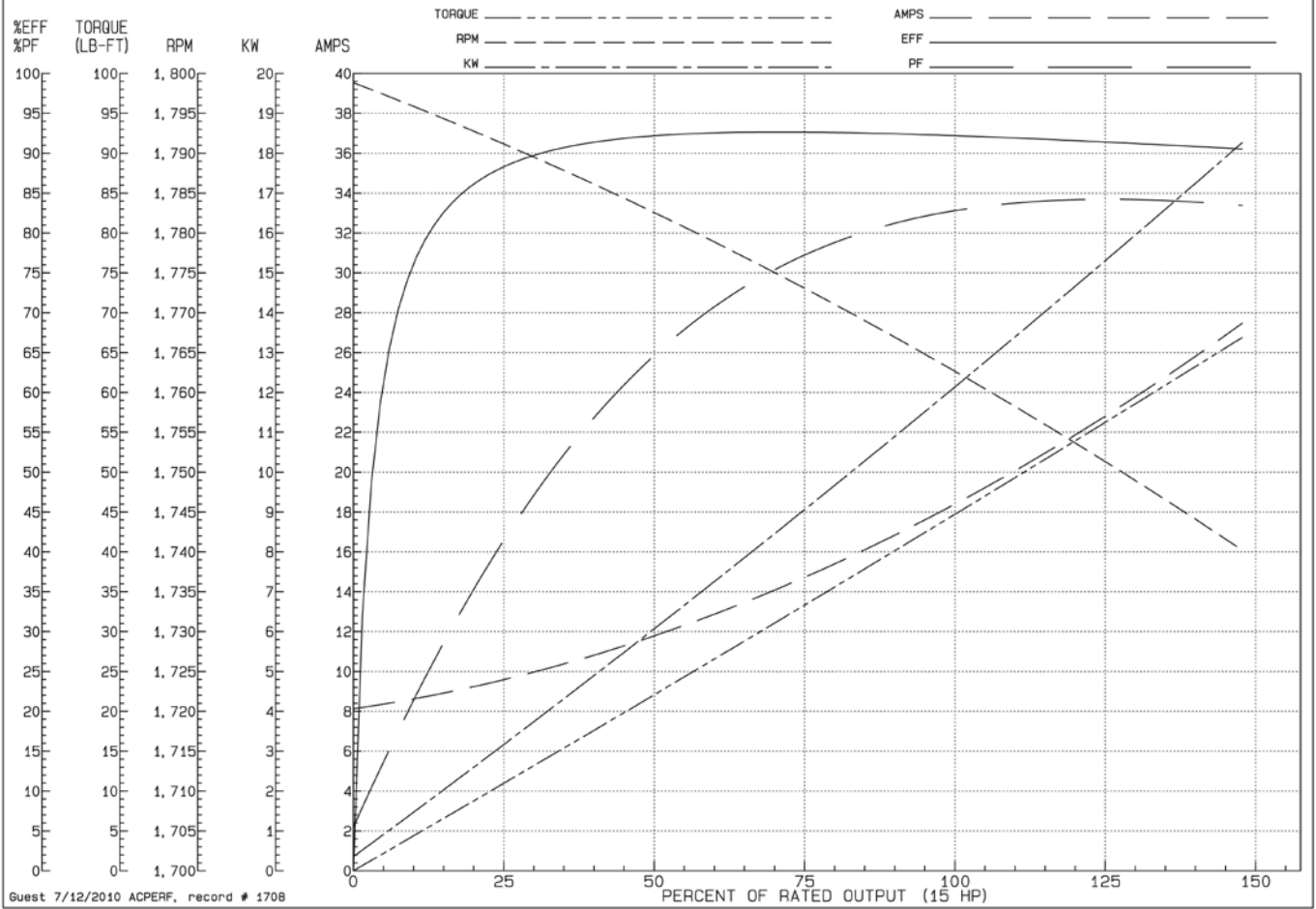


Figure 1 Typical — not guaranteed — motor performance data plots.

In general, the efficiency of motors will be relatively constant from 50% to 100% of rated load.

Amperes. Now, let's discuss amperes. Amperes are an indication of the flow of electric current into the motor. This flow includes both the borrowed as well as the used power. At low load levels, the borrowed power is a high percentage of the total power. As the load increases on the motor, the borrowed power becomes less and less of a factor and the used power becomes greater. Thus, there is an increase in the power factor as the load on the motor increases. As the load continues to increase beyond 50% of the rating of the motor, the amperage starts to increase in a nearly straight line relationship (Fig. 1).

Summary

Figure 1 shows significant items that have been discussed as plots of efficiency, power factor and watts, as they relate to horsepower. The most significant factor of all these is the watts requirement of the motor for the various load levels because it is the watts that will determine the operating cost of the motor, not the amperage.

The customer that has an extremely low power factor in

the total plant electrical system, may be penalized by his utility company because he is effectively borrowing a great deal of power without paying for it. When this type of charge is levied on the customer, it is generally called a power factor penalty. In general, power factor penalties are levied only on large industrial customers and rarely on smaller customers regardless of their power factor. In addition, there are a great many types of power customers such as commercial establishments, hospitals, and some industrial plants that inherently run at very high power factors. Thus, the power factor of individual small motors that are added to the system will not have any significant effect on the total plant power factor.

It is for this reason that the blanket statement can be made, that increasing motor efficiency will reduce the kilowatt hour consumption and the power cost for all classes of power users, regardless of their particular rate structure or power factor situation. This same type of statement cannot be made relative to power factor.

The following basic equations are useful in understanding and calculating the factors that determine the operating costs of motors and other types of electrical equipment. **PTE**

Operating Cost Calculations

Motors

$$\text{Kilowatt Hours} = \frac{\text{hp}^{**} \times .746 \times \text{Hours of Operation}}{\text{Motor Efficiency}}$$

** Average Load hp (May be lower than Motor Nameplate hp)

General Formula All Loads

$$\text{Kilowatt Hours} = \frac{\text{Watts} \times \text{Hours of Operation}}{1000}$$

Approximate Operating Cost* = Kilowatt hours × average cost-per-kilowatt-hour

* Does not include power factor penalty or demand charges which may be applicable in some areas.

Useful Constants

Average Hours per Month = 730

Average Hours per Year = 8,760

Average Hours of Darkness per Year = 4,000

Approximate Average Hours per Month (single shift operation) = 200

APPROXIMATE LOAD DATA FROM AMPERAGE READINGS

Conditions

1. Applied voltage must be within 5% of nameplate rating.
2. You must be able to disconnect the motor from the load (by removing V-belts or disconnecting a coupling).
3. Motor must be 7½ hp or larger — 3,450; 1,725; or 1,140 rpm.
4. The indicated line amperage must be below the full load nameplate rating.

Procedure

1. Measure and record line amperage with load connected and running.
2. Disconnect motor from load. Measure and record the line amperage when the motor is running without load.
3. Read and record the motor's nameplate amperage for the voltage being used.
4. Insert the recorded values in the following formula and solve.

$$\% \text{ Rated hp} = \frac{(2 \times \text{LLA}) - \text{NLA}}{(2 \times \text{NPA}) - \text{NLA}} \times 100$$

Where:

LLA = Loaded Line Amps

NLA = No Load Line Amps (Motor disconnected from

load)

NPA = Nameplate Amperage (For operating voltage)

Note: This procedure will generally yield reasonably accurate results when motor load is in the 40 to 100% range and deteriorating results at loads below 40%.

Example and Calculation:

A 20 hp motor driving a pump is operating on 460 volts and has a loaded line amperage of 16.5. When the coupling is disconnected and the motor is operated at no load the amperage is 9.3. The motor nameplate amperage for 460 volts is 19.4.

Therefore we have:

Loaded line amps (LLA) = 16.5

No load amps (NLA) = 9.3

Nameplate amps (NPA) = 19.4

$$\% \text{ Rated hp} = \frac{(2 \times 16.5) - 9.3}{(2 \times 19.4) - 9.3} \times 100 = \frac{23.7\%}{29.5\%} = 80.3\%$$

Approximate load on motor slightly over 16 hp.

POWER FACTOR CORRECTION ON SINGLE-INDUCTION MOTORS

Introduction

Occasionally we are asked to size power factor correction capacitors to improve the power factor of a single motor. Usually, the requested improved power factor level is 90 or 95%. The necessary calculations to get the proper capacitor KVAR (kilovolt ampere reactive) value are straightforward, but since we don't do it often it is nice to have the method in writing.

Procedure. The first thing needed is the full load power factor and efficiency information for the motor. On Baldor motors this can be found on the internet at www.Baldor.com. Next, since most power factor tables are worked in terms of kilowatts, it is necessary to convert the motor output rating into kilowatts. The procedure for doing this is to take the motor hp multiplied by the constant for kW per hp (0.746). This will give Output kW. Then it is necessary to divide this by the efficiency of the motor (as a decimal) to get the input kW at full load. Next, refer to power factor correction (Table I) going in from the left with the existing power factor and coming down from the top with the desired power factor. Where they intersect find the multiplier needed.

Next, multiply the motor input kilowatts by the appropriate

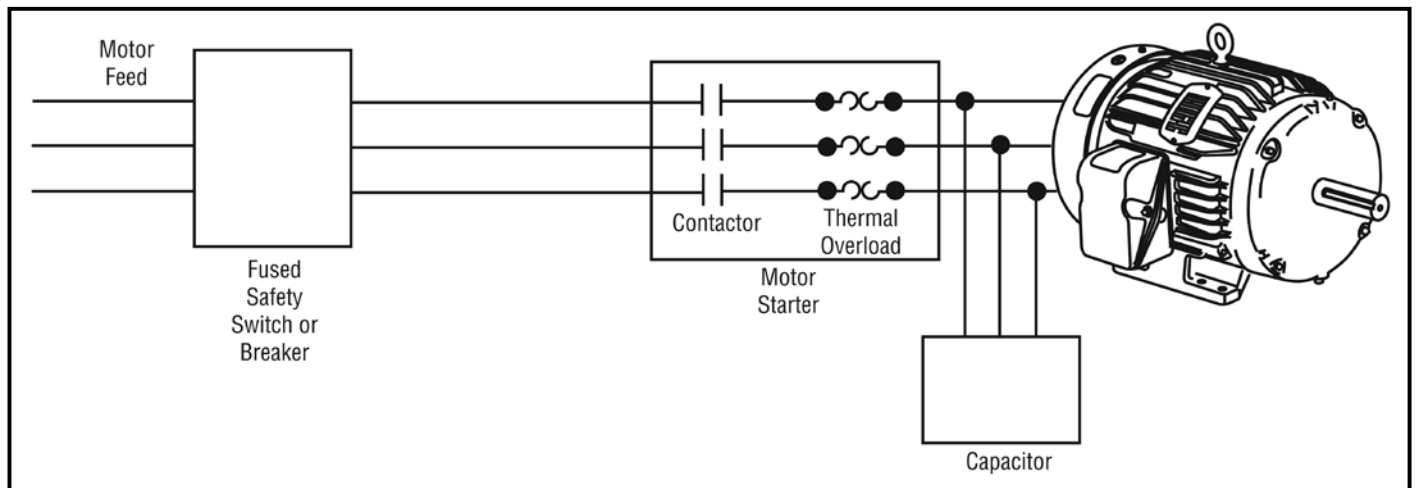


Figure 2 For single-motor correction, capacitors are connected between motor starter and motor at the motor terminals.

ORIGINAL FACTOR %	DESIRED POWER FACTOR		
	85	90	95
60	0.713	0.849	1.004
62	0.646	0.782	0.937
64	0.581	0.717	0.872
66	0.518	0.654	0.809
68	0.458	0.594	0.749
70	0.400	0.536	0.691
72	0.344	0.480	0.635
74	0.289	0.425	0.580
76	0.235	0.371	0.526
77	0.209	0.345	0.500
78	0.182	0.318	0.473
79	0.156	0.292	0.447
80	0.130	0.266	0.421
81	0.104	0.240	0.395
82	0.078	0.214	0.369
83	0.052	0.188	0.343
84	0.026	0.162	0.317
85	0.000	0.136	0.291
86		0.109	0.264
87		0.083	0.238
88		0.056	0.211
89		0.028	0.183
90		0.000	0.155
91			0.127
92			0.097
93			0.066
94			0.034
95			0.000

3-PHASE STANDARD CAPACITOR RATINGS KVAR (Kilovolt Amperes Reactive)		
1.0	20.0	70.0
1.5	22.5	75.0
2.0	25.0	80.0
2.5	27.5	85.0
3.0	30.0	90.0
4.0	32.5	100.0
5.0	35.0	120.0
6.0	37.5	140.0
7.5	40.0	150.0
8.0	42.5	160.0
9.0	45.0	180.0
10.0	50.0	200.0
11.0	52.5	225.0
12.5	55.0	250.0
15.0	60.0	300.0
17.5	65.0	350.0

multiplier from Table 1 to get the required KVAR of power factor correction. This value would be rounded out to match commercially available power factor correction capacitor ratings shown in Table 2.

Example. To illustrate the procedure an example is worked as follows:

What is the KVAR of power factor correction capacitors needed to improve the power factor of a catalog number M2555T, 100 hp motor, to 95% at full load?

Step 1: Look up the existing power factor and efficiency.

$$\text{Efficiency} = 94.1\%$$

$$\text{Power factor} = 85\%$$

Step 2: Convert the hp to kilowatts *output*. $100 \text{ hp} \times 0.746 = 74.6 \text{ kW}$

Step 3: Convert kilowatts *output* to kilowatts *input* by dividing by the full load efficiency.

$$\frac{74.6}{.941} = 79.3 \text{ kW input}$$

Step 4: Look in Table 1 to find the multiplier to achieve the desired 95% corrected power factor.

The multiplier is 0.291.

Step 5: Multiply input kW by this multiplier. $79.3 \times 0.291 = 23.1 \text{ KVAR}$

This gives the required capacitor KVAR.

Step 6: Select closest value from Table 2. 22.5 KVAR

The voltage of the capacitor would also have to be specified; in this case it would be 480 volts.

Current correction. In many cases when a single motor is being corrected, the capacitors are connected between the motor starter and the motor at the motor terminals (Fig. 2). With this being the case, the effect of the correction is to reduce the current flowing through the starter and overload relay. Since the overload heaters are selected (or adjusted) on the basis of the motor at full load current, this means that the overloads will not correctly protect the motor unless the ampacity is reduced to reflect the reduced current now flowing as a result of the power factor improvement.

The motor itself will draw the same number of amps at full load as it would without the power factor correction. However, the power factor correction capacitors will be supplying a portion of the current, and the balance will be coming through the starter from the power line.

The new value of current passing through the overloads is given by the following formula:

$$\text{Current}_{\text{new}} = \text{Motor full load (nameplate) amps} \times \frac{\text{Power factor original}}{\text{Power factor corrected}}$$

For example, in the case of the 100 hp motor in the example, the heater size — which would normally be selected from the motor nameplate current at 118 amps — would have to be adjusted as follows:

$$\text{Current}_{\text{new}} = 118 \times \frac{.85}{.95} = 118 \times .895 = 105.6 \text{ or approximately } 106 \text{ amps } .95$$

Summary

A few words of caution might be appropriate here. Usually it is desirable to “under correct” rather than “over correct.” If the capacitors chosen are too large, there can be a number of problems — including high transient torques and overvoltage. Thus it is usually not desirable to attempt to improve power factor beyond 95%; it also usually becomes uneconomical to attempt improvements beyond 95%.

Please note: This type of power factor improvement *should not be used* in any situation where the motor is being controlled by a solid state device such as a soft start control or a variable frequency drive.

(See Figure 2 for Convenient Motor and Energy Formulas.) **PTE**



Vibration and Operational Characteristics of a Composite-Steel (Hybrid) Gear

Robert F. Handschuh, Kelsen E. LaBerge, Samuel DeLuca and Ryan Pelagalli

Hybrid gears have been tested consisting of metallic gear teeth and shafting connected by composite web. Both free vibration and dynamic operation tests were completed at the NASA Glenn Spur Gear Fatigue Test Facility, comparing these hybrid gears to their steel counterparts. The free vibration tests indicated that the natural frequency of the hybrid gear was approximately 800 Hz lower than the steel test gear. The dynamic vibration tests were conducted at five different rotational speeds and three levels of torque in a four square test configuration. The hybrid gears were tested both as fabricated (machined, composite layup, then composite cure) and after regrinding the gear teeth to the required aerospace tolerance. The dynamic vibration tests indicated that the level of vibration for either type of gearing was sensitive to the level of load and rotational speed.

(This work was sponsored by the Revolutionary Vertical Lift Technologies (RVLT) Program at the NASA Glenn Research Center.)

Introduction

Drive systems of the future will be required to have increased power-to-weight ratios in addition to reduced maintenance, noise, and cost. These requirements have been under constant evolution, resulting in improvements over the last several decades. Materials, manufacturing, processing, lubricants, and advanced drive system analysis have led to drive system designers pushing the technology and steadily making improvements (Refs. 1–5). Typically these technology improvements are implemented when drive system upgrades occur and when new vehicles are designed. The implementation is timed in this manner to reduce costs, as qualification of a drive system for rotorcraft is an expensive process.

Power-to-weight ratio is one of the most important drive system attributes. Parametrically the resultant gearbox and lubrication system weight, w_t , for rotorcraft of all types, sizes, number of engines, and number of main rotors is shown (Fig. 1) (Ref. 6). Here, the parametric value, γ , is a function of the input and output speeds v_{in} and v_{out} , respectively (both in revolutions-per-minute), of the drive system and the transmitted power P (in horsepower). Once this calculation is made, the anticipated weight of the gearbox and lubrication system, w_t , can be found. Using advanced technologies results in drive systems that are lighter weight for a given gear ratio and power level such that they fall below the curve shown (Fig. 1). This improved capability offers the rotorcraft

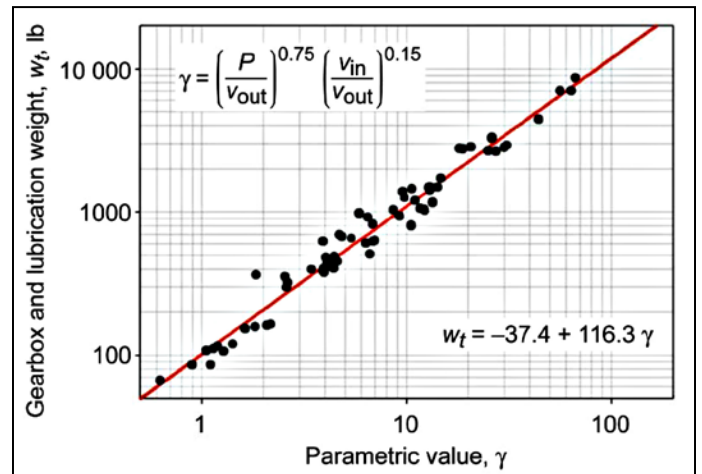


Figure 1 Parametric curve fit for rotary-wing aircraft gearbox and lubrication system weight (Ref. 6).

extended range and/or payload.

Materials play an important role in improving the power-to-weight ratio. Currently rotorcraft drive systems utilize light-weight structure materials (aluminum and magnesium) for the housing and minimize gear weight via careful analysis and machining. Generally, rotorcraft gears have only enough mass for load carrying ability. Minimizing the mass of a gear leads to a lack of heat storage capability. This attribute can cause problems during a primary lubrication system failure in which gears would operate under starved or dry conditions. Therefore all operation scenarios need to be carefully considered.

Composite materials have been considered for some rotorcraft drive system housings, such as the work contained in Reference 7.

Table 1 Composite and aerospace steel material properties.

Property	Composite material	AISI 9310 gear steel
Modulus of elasticity, Pa (psi)	Tension: 44×10^9 (6.4×10^6) Compression: 42×10^9 (6.1×10^6)	200×10^9 (29×10^6)
Poisson's ratio	0.3	0.29
Density, kg/m ³ (lb/ft ³)	1800 (112)	7861 (491)
Thermal conductivity, W/m·°C (Btu/h.ft.°F)	T700 fiber-axial: 9.4 (5.43)	55 (32)
Useful maximum temperature as gear material, °C (°F)	150 (302)	175 (347)
Coefficient of thermal expansion, 10^{-6} m/m-K ⁻¹ (10^{-6} in/in-°F ⁻¹)	In plane: 2 (1.1)	13 (7.3)
Failure strain, percent	Tension: 1.9 Compression: 0.94	
Elongation, percent		15

Many issues were worked out to attain project success. However, cost and some other technical issues yet to be resolved have been a roadblock to incorporating this technology into production.

In addition to structural components, there have also been recent efforts to incorporate lower-density composites in dynamic components, such as shafts and gears. This report focuses on the potential application of composite material in rotorcraft drive system gears. The web of the test gear was replaced with composite material. The material properties of the composite material used in this study are compared with those of a typical aerospace gear material in Table 1. One property that is of real importance is the density. The density of the composite material used in this study is approximately 25 percent of that of typical gear steel. Also, there was an anticipated benefit expected that the material change should help with mesh-generated vibration and noise that is transmitted from the gears to the shafts and bearings.

The objective of this study is to describe how composite webbed gears (referred to here as “hybrid” gears) were fabricated and the resultant effect on the gear natural frequency, transmitted vibration, and noise.

Hybrid Gear Manufacture

The hybrid gears manufactured in this study followed the process as described (Reference 8); a brief description follows.

The test gear design used for this study has the design shown (Table 2). Figure 2 provides a pictorial explanation of the hybrid gear assembly process. A hexagonal region was machined out of a steel gear leaving two steel gear components: a gear rim with the teeth and a hub region for attachment to the facility shafting. The braided pre-preg composite material (*fibrous material pre-impregnated with a particular synthetic resin used in making reinforced plastics*) was built up in a fixture around the steel hub and rim as shown (Fig. 2, steps 3–8). A total of 36 layers of composite material were built up and then cured in the fixture (step 9) at a final temperature of 177°C (350°F). The fixture for fabrication and curing used the inner diameter of the hub and the gear measurement over pins in an attempt to keep the gear teeth aligned with the axis of rotation. An example of the cured gear is shown (Fig. 3).

Table 2 Gear design for this study.	
Number of teeth	42
Module, mm (Diametral pitch 1/in)	2.12 (12)
Circular pitch, mm (in.)	6.650 (0.2618)
Whole depth, mm (in.)	4.98 (0.196)
Addendum, mm (in.)	2.11 (0.083)
Chordal tooth thickness, mm (in.)	3.249 (0.1279)
Pressure angle, deg	25
Pitch diameter, mm (in.)	88.9 (3.5)
Outside diameter, mm (in.)	93.14 (3.667)
Measurement over pins, mm (in.)	93.87 (3.6956)
Pin diameter, mm (in.)	3.66 (0.144)
Backlash ref., mm (in.)	0.15 (0.006)
Tip relief, mm (in.)	0.013-0.018 (0.0005-0.0007)
All-steel gear weight, kg (lbf)	0.3799 (0.8375)
Hybrid gear weight, kg (lbf)	0.3242 (0.7147)

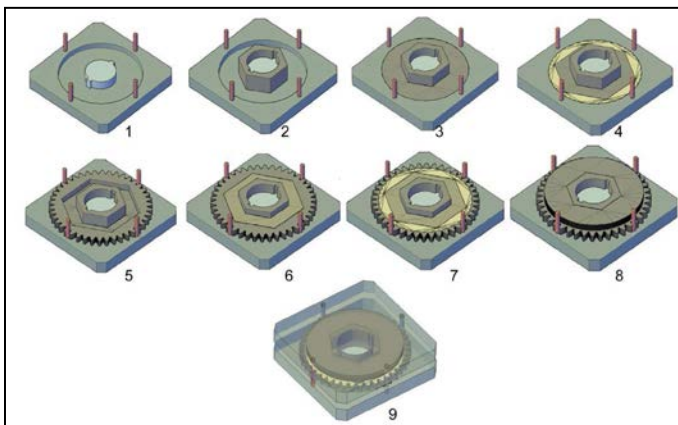


Figure 2 Hybrid gear assembly process.

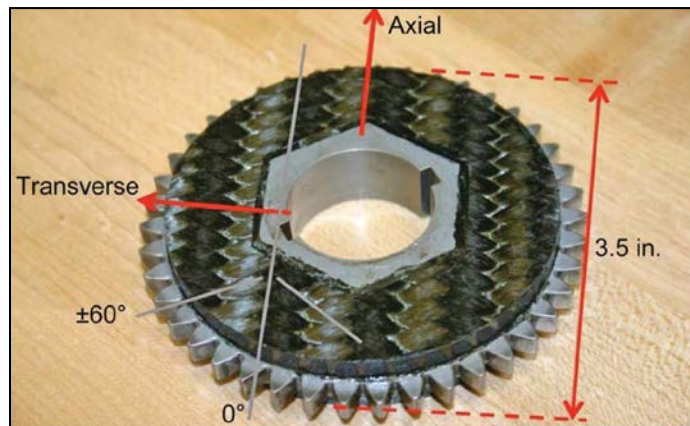


Figure 3 Assembled and cured hybrid gear.

A total of four gears (two pairs) were manufactured and used in this study. The first gear pair (A) was used for fatigue testing (to be described later), while the second pair (B) was used for both static and dynamic vibration tests. Because of material differences, i.e. — coefficient of thermal expansion — the initial gear geometry was not preserved after the curing process, thus leading to significant pitch variation and other anomalies compared with the original steel aerospace gear. Figure 4 provides example pitch variation measurements for two of the four hybrid gears after curing. These pitch variations were greater in gear set A than in set B. This

resulted in a reduction in backlash, normally around 0.15 mm (0.006 in.), of 0 to 0.038 mm (0 to 0.0015 in.). Even with the reduced backlash, gear set A withstood fatigue testing without damage to the contact surfaces.

To reduce the pitch variations and return the tooth geometry to its original form, the teeth on gear set B were re-ground after performing initial vibration tests. The geometry of one of the re-ground gears is shown in Figure 5. Although the tooth geometry was restored, the re-grind process resulted in excess backlash of 0.58 to 0.64 mm (0.023 to 0.025 in.), which is nearly 4 times the normal steel gear backlash. Some

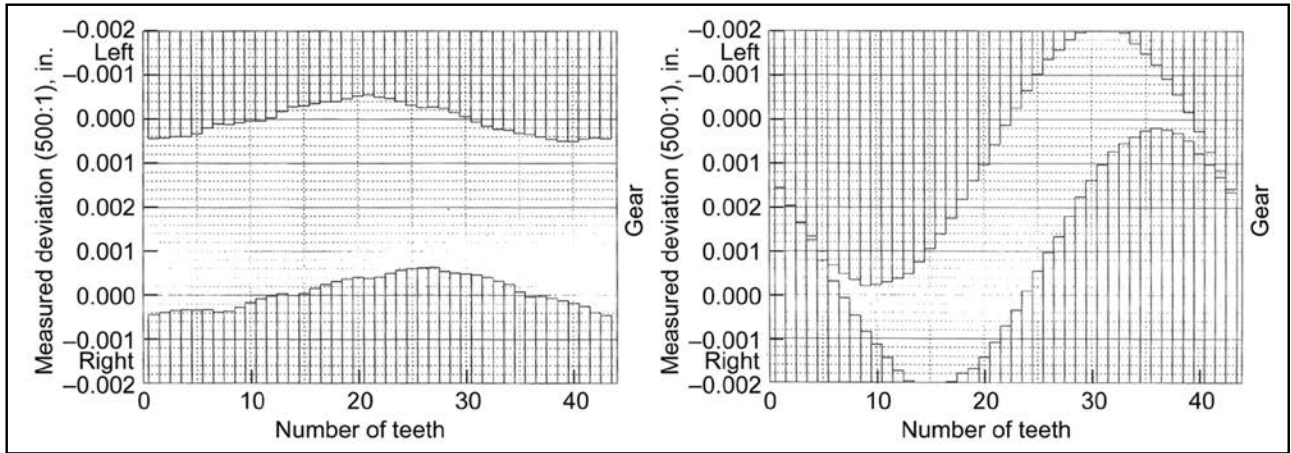


Figure 4 Measured pitch variation (index) of two assembled hybrid gears.

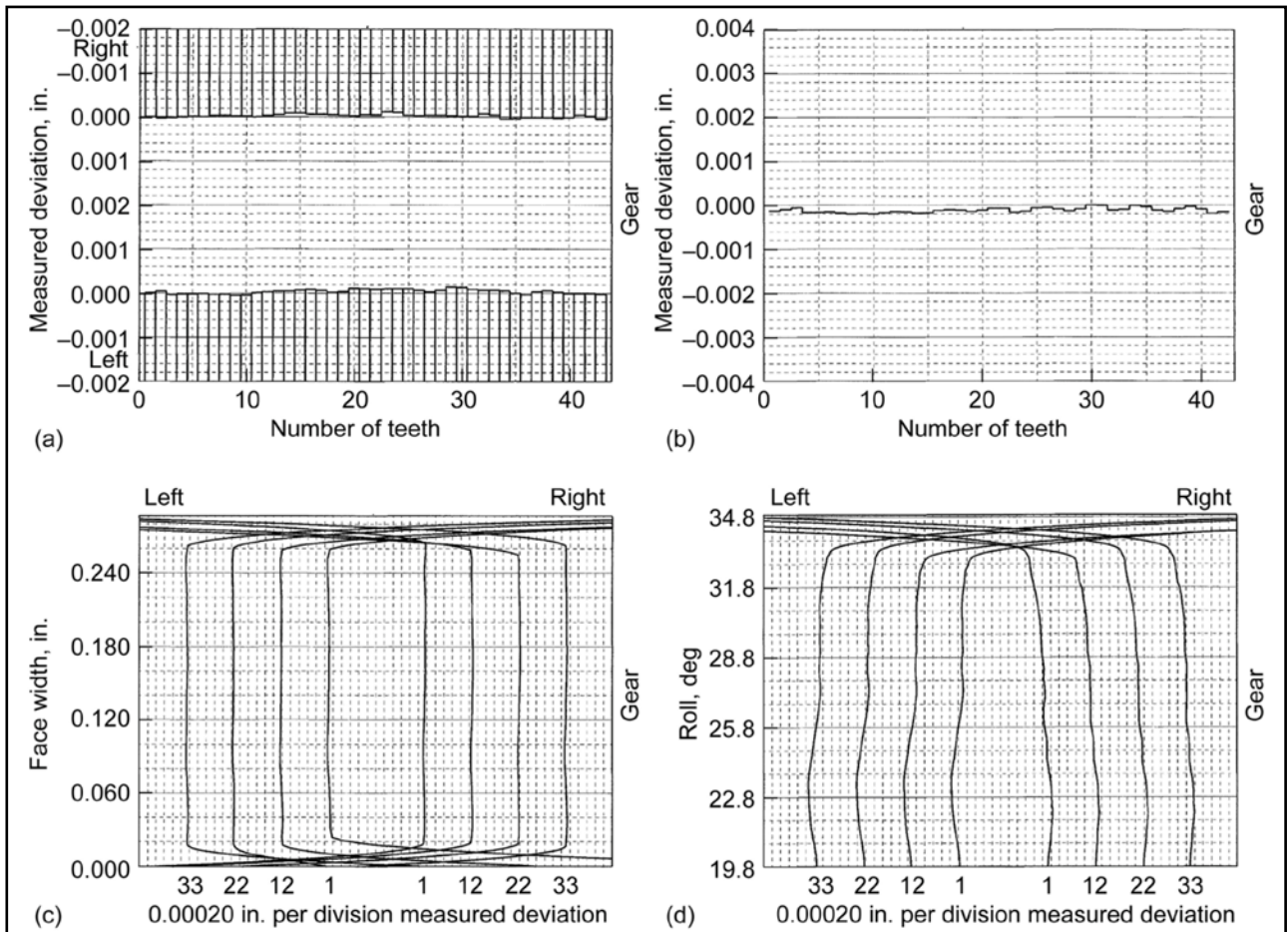


Figure 5 Hybrid gear geometry measurements after regrind; (a) pitch variation (index); (b) pitch line runout; (c) lead; (d) involute.

modifications would be needed in adapting this process to production, such as waiting until after composite curing to perform the final tooth grind to minimize repeat effort.

Free Vibration Modes

In an effort to understand how adding a composite web to a spur gear affects the vibration and noise generated and/or transmitted, first free-free vibration tests were performed in the form of impact tests. Impact tests were performed on both the composite hybrid gear and its steel counterpart. The vibration response was collected with an accelerometer connected to the gear's inner bore via a small aluminum bracket affixed (glued) to the bore to provide a rigid mount that was easily removed. The gear was suspended by an elastic band. Using an instrumented modal hammer, the gear was impacted radially in line with the accelerometer. This is slightly different from the setup previously reported, that placed the accelerometer in the axial direction adjacent to the bore (Ref.8). Figure 6 shows the test setup and accelerometer bracket. Data acquisition was triggered by the instrumented hammer and was collected at a sample rate of 50 kHz. Figure 7 shows representative vibration plots of both a hybrid and a steel gear. Although the majority of the vibration has diminished in the composite gear after 5 ms, the steel gear vibrates out past 10 ms.

The accelerance for the steel gear impact tests is shown (Fig. 8); the first natural frequency for the steel gear is around 7,000 Hz. This is consistent with a previous finite element analysis (FEA) that provided estimated natural frequencies at 7,187 and 7,270 Hz (Ref.8). Similarly, the natural frequencies seen at 12.1 and 12.7 kHz are also close to FEA approximations. Those seen at 15.2, 16.5 and 17.2 kHz were not seen in previously published axial impact tests—likely due to the fact that the mode shapes around 15 and 16 kHz involve only radial motion, as was seen in the FEA-estimated mode shapes. Since damping for any particular mode is related to the width of the peak in the frequency spectrum, the sharp spikes at 12.1, 12.7, 16.5 and 17.2 kHz represent the modes with limited damping, whereas the modes at 7 and 15.2 kHz exhibit a higher level of damping.

Hybrid gear impact tests displayed (Fig.9) show the first natural frequency around 6,400 Hz; this is consistent with

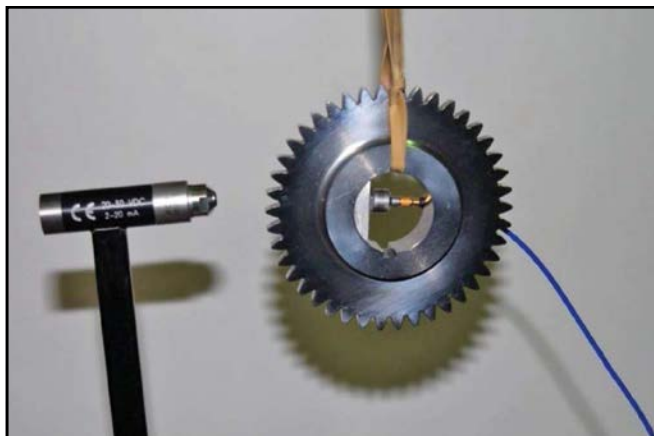


Figure 6 Impact test setup (with steel gear).

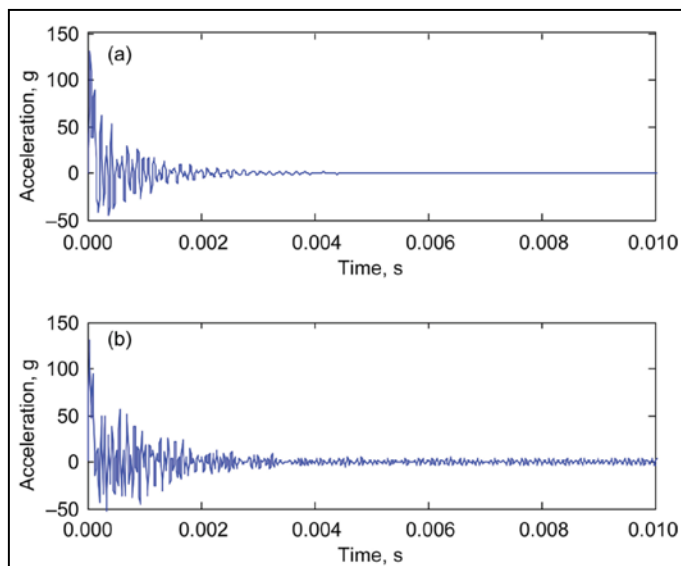


Figure 7 Vibration response of typical impact tests; a) hybrid gear; b) steel gear.

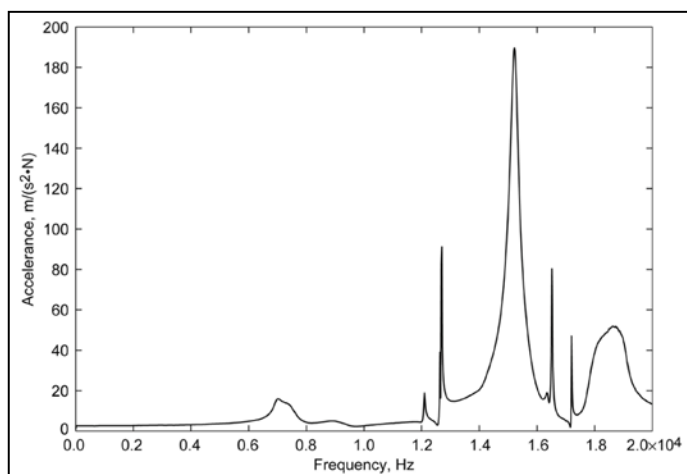


Figure 8 Steel gear accelerance magnitude (average of 10 impacts).

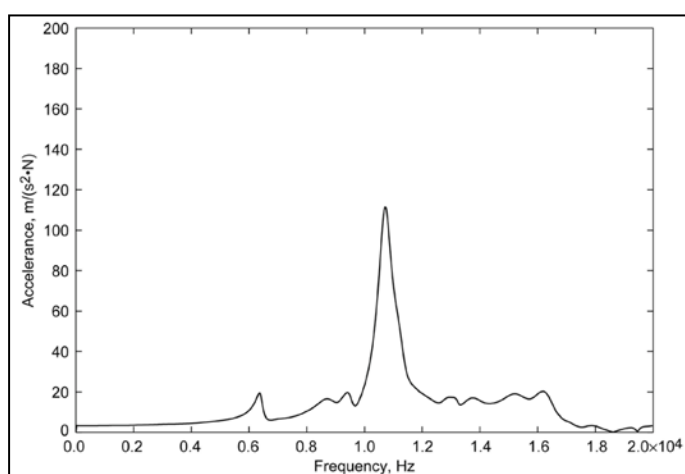


Figure 9 Hybrid gear accelerance magnitude (average of 10 impacts).

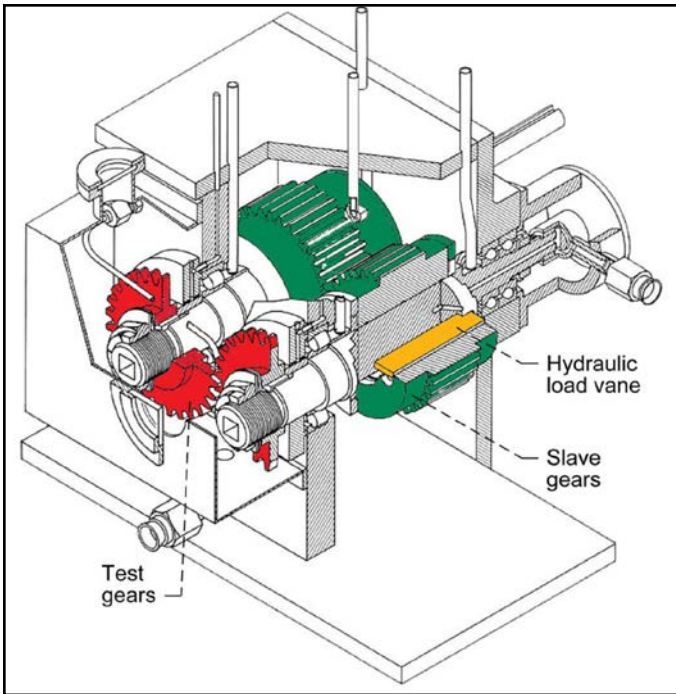


Figure 10 NASA Glenn spur gear contact fatigue facility test rig.

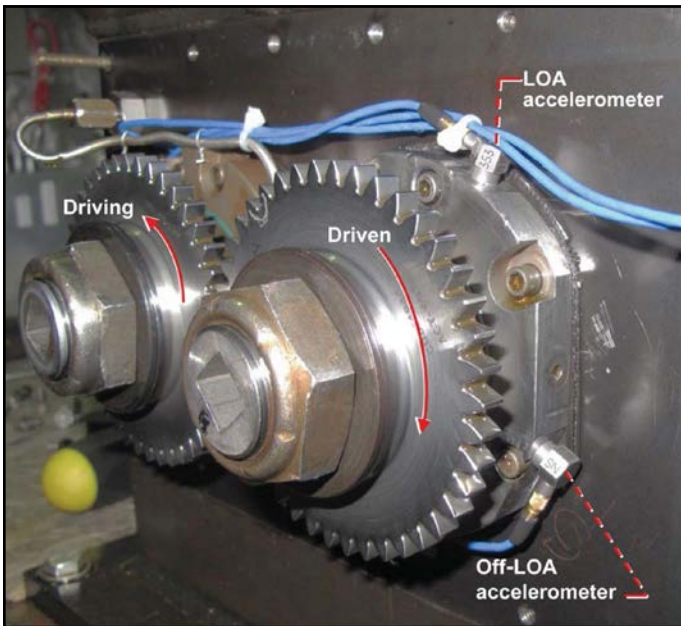


Figure 11 Location of accelerometers on driven shaft of test rig.

previous testing (Ref. 8). These results also show a strong natural frequency at 10.7 kHz. It is difficult to separate the other resonant frequencies, as they are highly damped and overlap. Neither of these natural frequencies is consistent with those identified by the FEA that predicted the first frequency to be higher than that of the steel gear — at 7,780 Hz (Ref. 8). Further modeling efforts are needed to obtain useful hybrid gear FEA predictions, with a focus on ply arrangement effects and a better model of the interface between the steel and composite material.

Test Facility

The dynamic tests reported herein were conducted on the NASA Glenn spur gear fatigue test rig (Fig. 10), a closed-loop test rig that operates at speeds up to 10,000 rpm and 57.9 N·m (513 in·lb) torque. Speed and load are independently adjustable; load is adjusted by varying the supply pressure to the rotating torque actuator located in one of the slave gears. From FEA the torque mentioned above induces a bending stress of 0.212 GPa (30.8 ksi) and a contact stress of 1.17 GPa (170 ksi). These stresses were calculated using the full three-dimensional FEA method described in Reference 9.

Vibration data were attained using accelerometers with a bandwidth of 10 kHz with ± 5 percent accuracy (18 kHz at ± 10 percent). The accelerometers were mounted as shown (Fig. 11), such that one was parallel to the line of action (LOA) of the gears and the other perpendicular (off-LOA). Data were captured via laboratory analog-to-digital converters and read into a computer at a rate of 50 kHz.

Experimental Results

The experimental results of the three types of tests performed are described in the following sections.

1. Vibration. Dynamic vibration data was taken with test gears installed in four different configurations: steel driving steel, steel driving hybrid, hybrid driving steel, and hybrid driving hybrid. It is important to note that the hybrid gears used for the dynamic tests were set B that were reground to aerospace tolerances, as discussed previously. The tests were performed at five different shaft speeds and at three levels of torque (load pressure). The rotational speeds were 2,500, 5,000, 7,500, 8,750 and 10,000 rpm and the torque levels were at 20.5, 39.2, and 57.9 N·m (182, 347, and 513 in·lb, respectively). From Figure 11, the driving gear is the one on the left and the driven gear is on the right. All vibration data were taken from the right-side seal housing that is in direct contact with the shaft support bearing.

Root-mean-square (RMS) vibration levels for each test are shown (Fig. 12) for both the LOA and off-LOA accelerometers. Results show that the highest vibration was experienced when the hybrid gear was driving the steel gear. There exists — when comparing the hybrid driving steel results with the steel driving hybrid results — a significant reduction in vibration. With the driving gear mounted on the left and the driven gear mounted on the right, adjacent to the accelerometers, we see that lower vibration levels are experienced when the hybrid gear is mounted next to the sensors. The hybrid gear pair shows some improvement over the steel pair at

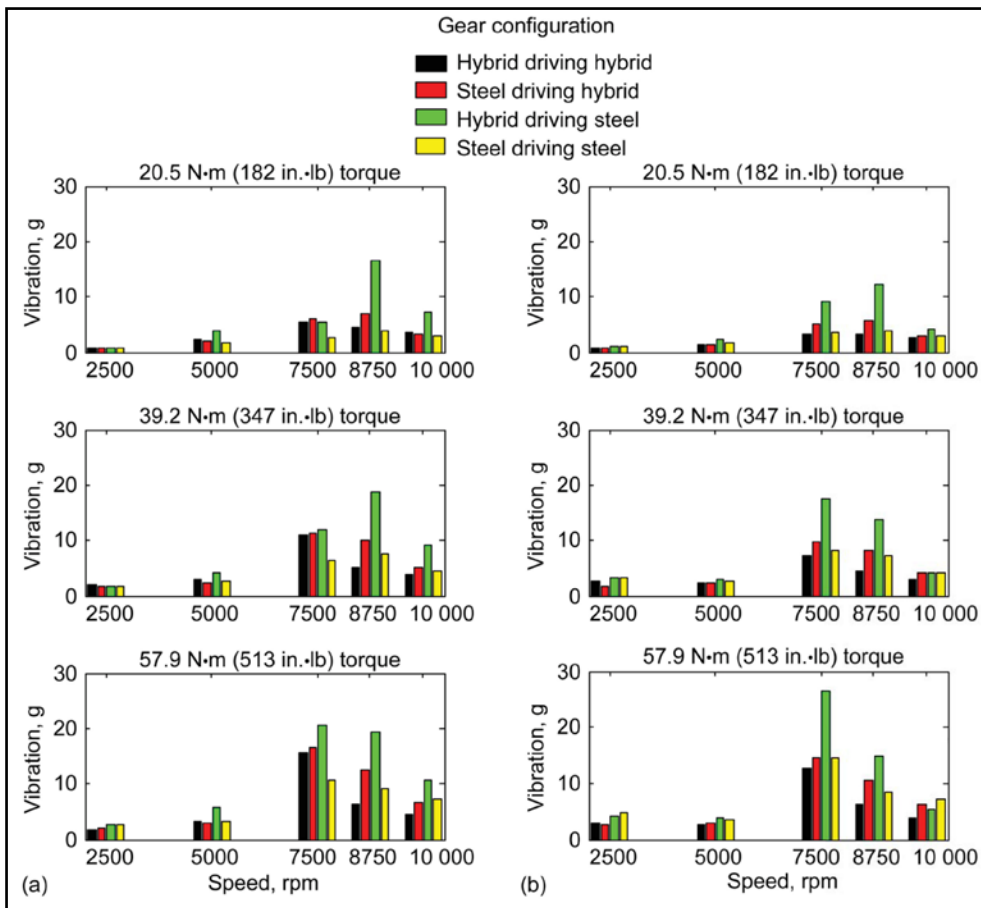


Figure 12 Comparison of root-mean-square (RMS) vibration signal of gears in different configurations; a) along the line of action (LOA); b) perpendicular to the line of action (off-LOA).

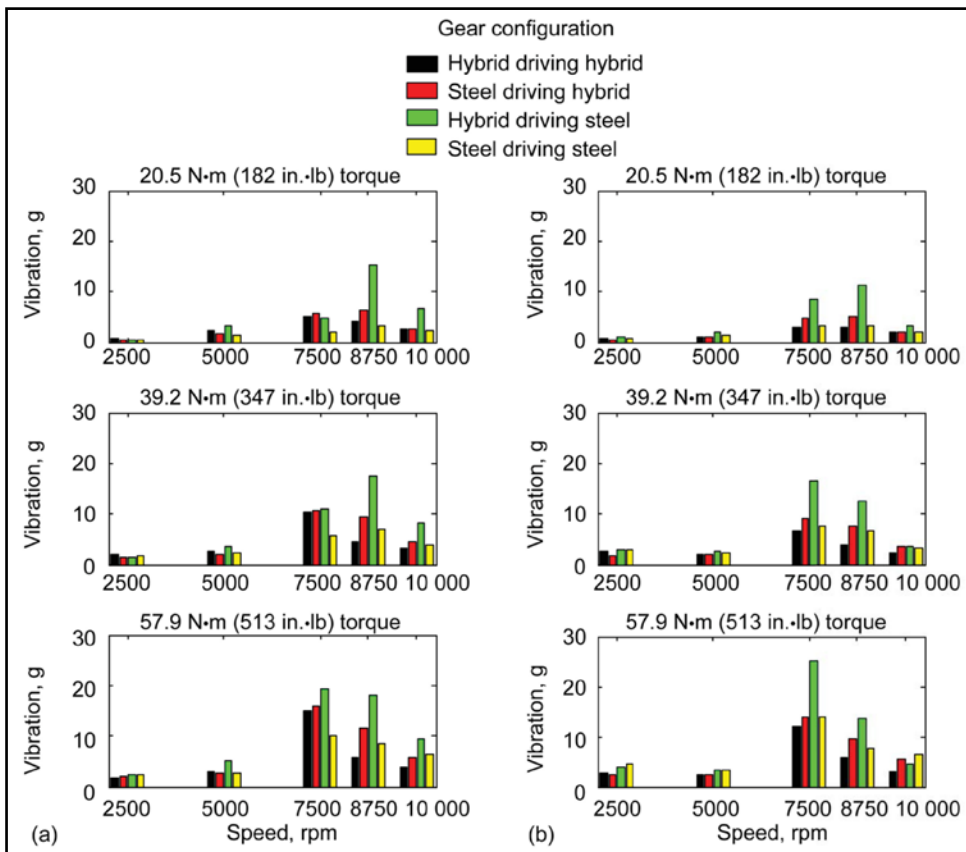


Figure 13 Comparison of RMS synchronous signal average of dynamic vibration of gears in different configurations; a) along the line of action (LOA); b) perpendicular to the line of action (off-LOA).

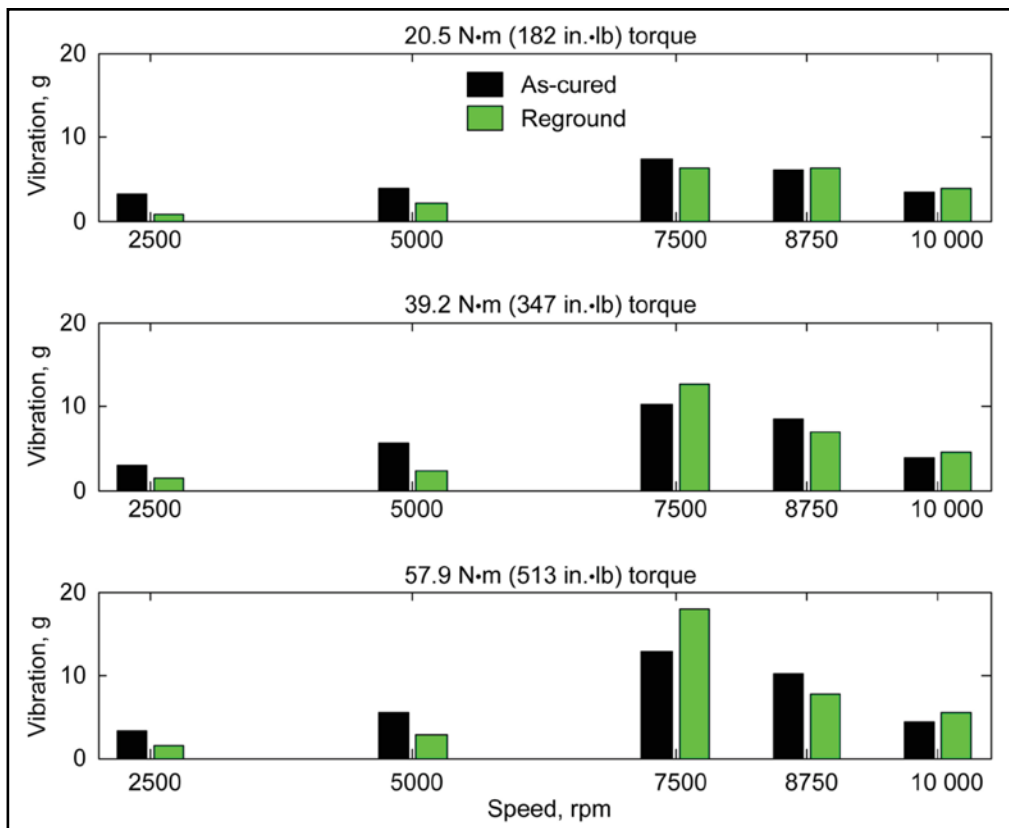


Figure 14 Comparison of RMS vibration level along the line of action (LOA) between a hybrid gear pair before and after regrind.

the higher rotational speeds and loads for the LOA accelerometer, and shows a small overall improvement when comparing to off-LOA results.

The raw data were also averaged over 20 shaft rotations to obtain the time-synchronous average. This process is performed to remove nonsynchronous noise from the data. The RMS of the average is shown (Fig. 13) and provides similar results to those shown in Figure 12.

Dynamic tests were performed both before and after gear set B was re-ground, providing a comparison between the two geometries; these vibration results are shown (Fig. 14). The re-grind process is shown to reduce the vibration by as much as half at lower speeds. The results presented could be improved by optimizing the composite curing process to minimize the distortion that occurs. The need for re-grinding after the composite cure process may not be eliminated, but the amount of material removal required to bring the geometry back to aerospace quality could be reduced.

2. Noise. While the vibration data was being taken, a sound pressure level measurement was made. The measurement was taken using a single, hand-held acoustic probe that uses the A-weighted scale. The probe was held at a distance of 0.4 m (~16 in.) at the height of the centerline of the meshing gears, pointed directly at the test gear cover. The RMS-averaged data are shown (Fig. 15). The noise data produced similar trends, as did the vibration data discussed previously.

3. Fatigue. Hybrid gear set A was used for endurance testing as a proof of concept. The gear set used in this test was put into the facility after composite curing and without any re-

grinding of the gear tooth profiles. A single long-term test was conducted to examine the durability of the composite-steel bond. The test was run for one billion cycles at 10,000 rpm and 48.6 N-m (430 in.-lb) at 49°C (120°F) oil inlet temperature. This corresponds to a calculated contact stress of approximately 1.35 GPa (195 ksi) and a bending stress of 0.20 GPa (29.2 ksi). The gears survived without any visual signs of de-bonding or degradation of the composite material.

Conclusions

Based on the results attained in this study, the following conclusions can be made:

- A hybrid gear mesh has shown promise for operation in a simulated aerospace environment with the possibility of reducing component weight.
- Hybrid gears need to be further processed to maintain aerospace gear quality.
- The hybrid gears were fatigue-tested to one billion stress cycles at representative loads and speeds, with no resulting degradation.
- The hybrid gears exhibited a lower natural frequency than the standard steel gears of the same size and dimension.
- Dynamic vibration results indicated the strong dependence of operational speed on the measured results. The hybrid driving hybrid configuration exhibited the lowest vibrations — but only at the higher speeds and loads.
- Vibration tests performed with the hybrid gear driving the steel gear typically exhibited the highest RMS vibration levels — particularly at the higher rotational speeds.
- The A-weighted, RMS sound-level measurements trended similarly to those of the vibration results. **PTE**

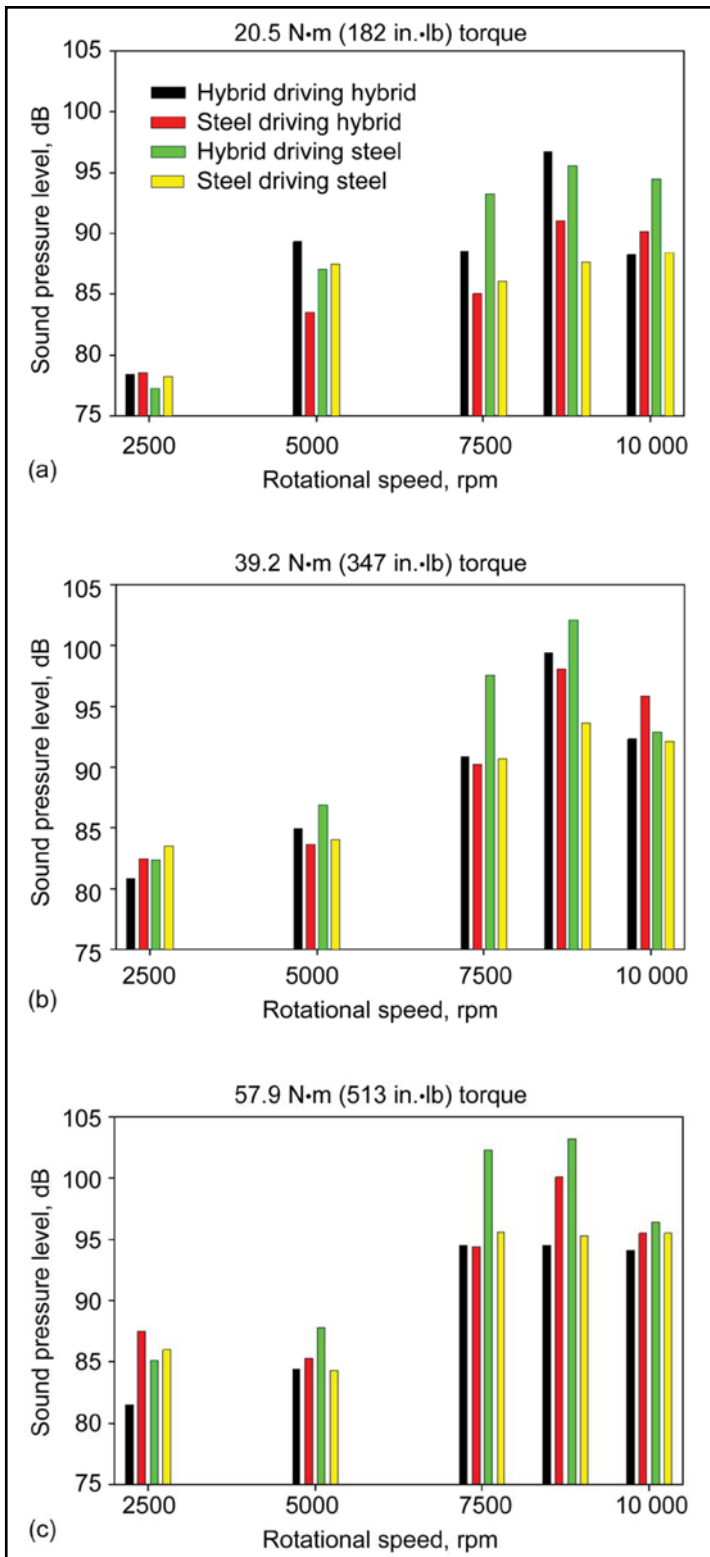


Figure 15 Effect of rotational load and speed on RMS A-weighted sound pressure level for different meshing conditions.

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Dr. Robert F. Handschuh — National Aeronautics and Space Administration, NASA Glenn Research Center, Cleveland, Ohio — possessing over 30 years' experience with NASA and the Department of Defense working on rotorcraft drive system analysis and experimental methods — is chief for the Rotating and Drive Systems branch at the NASA Glenn Research Center. Among his numerous research projects at NASA Glenn are high-speed gearing (including windage), loss-of-lubrication technology, and hybrid gearing. Handschuh has successfully developed and utilized a number of experimental research facilities at NASA Glenn, thus enabling his R&D work in high-temperature, ceramic seal erosion; planetary and high-speed, helical geartrains; spiral bevel gears and face gears; single-tooth-bending fatigue. Handschuh is a masters graduate in mechanical engineering from the University of Toledo; he received his Ph.D from Case Western Reserve University.



Dr. Kelsen LaBerge received her Ph.D. in mechanical engineering from Case Western Reserve University, where she worked on health monitoring of rotor shafts. She joined the U.S. Army Research Laboratory in the spring of 2009 as part of the Vehicle Technology Directorate, stationed at NASA Glenn Research Center in Cleveland, OH. While there, she has worked closely with NASA colleagues on vertical lift powertrain research — of interest to both the commercial and military sectors. She has experience in the area of vibration-based mechanical component diagnostics — particularly planetary gears and rolling element bearings. Currently, LaBerge's research focuses on hybrid (composite/steel) gear technology and methods for increasing gear life under starved oil conditions.



Samuel Deluca and **Ryan Pelagalli** were interns working in Lewis Educational and Research Collaborative Internship Project.

PTDA

ANNOUNCES 2018 BOARD OF DIRECTORS AND MANUFACTURER COUNCIL

The Power Transmission Distributors Association (PTDA) elected its 2018 Board of Directors and Manufacturer Council at the annual business meeting during the recent NIBA/PTDA Joint Industry Summit in Hollywood, Fla.

Jim Williams, vice president corporate purchasing & supplier relations, Motion Industries Inc. (Birmingham, Ala.) will become PTDA's president in 2018. He succeeds Tom Clawser. Williams has been active in PTDA since 2005, when he joined the Motion Control Task Force. A past chair of the Programs & Products Committee, Jim has served on the PTDA Board of Directors since 2015.



Following his election, Williams said, "This is a great honor and I look forward to serving PTDA in 2018. We have successfully begun to implement the new four-goal strategic plan. This new plan will ensure the Board is in a strong position to further the Association's mission to advance our industry and empower members to be successful, profitable and competitive in this changing environment."

Joining Williams on the 2018 PTDA Board of Directors will be:

- First Vice President Jim Halverson, manager power products, Van Meter Inc. (Cedar Rapids, Iowa)
- Second Vice President Brian Davis, vice president, B & D Industrial (Norcross, Ga.)
- Treasurer LeRoy Burcroff, vice president sales, Bearing Service Inc. (Livonia, Mich.)
- Manufacturer Council Chair Chris Curran, president, Climax Metal Products Company, (Mentor, Ohio)
- Manufacturer Council Vice Chair Sandy Sullivan, national account director, Nidec Motor Corporation/U.S. Motors (St. Louis, Mo.)
- PTDA Foundation President Brian Short, corporate account manager, NSK Americas (Ann Arbor, Mich.)
- EPTDA President Jan Friman, director business development, YTM-Industrial Oy (Vantaa, Finland)

Directors:

- J.P. Bouchard, General Bearing Service Inc. (Ottawa, Ontario, Canada)
- Steven W. DuComb, operations manager, W.C. DuComb Co., Inc. (Detroit, Mich.)
- Brian Kolman, president, Brewer Machine & Gear Co. (St. Louis, Mo.)
- Mike McLain, vice president, Allied Bearing & Supply, Inc. (Harahan, La.)
- Bill Shepard, vice president SW region & marketing, BDI (Cleveland, Ohio)
- Tom Weihsmann, senior VP & GM, Kaman Industrial Technologies Corporation (Bloomfield, Conn.)

Chris Curran, president, Climax Metal Products Company (Mentor, Ohio) assumes the duties of the PTDA Manufacturer Council chair in 2018, succeeding Michael Cinquemani, president & CEO, Master Power Transmission, Inc. (Greenville, S.C.). Curran has been active in PTDA governance since 2004 when he joined the Employee Development Committee. Since then, Chris has served on several committees and chaired both the Employee Development Committee and the 2012 Industry Summit Planning Committee. He joined the Manufacturer Council in 2014 where he served as vice chair in 2017.



"I am honored to continue my service to PTDA and its members. Working closely together with the distributors leading our industry, I am confident we can help our fellow members build relationships, solve problems and keep the world moving ahead through the sales and service of power transmission products," Curran said.

Joining Curran on the Manufacturer Council for 2018 will be:

Vice Chair Sandy Sullivan, national account director, Nidec Motor Corporation/U.S. Motors (St. Louis, Mo.)

Immediate Past Chair Michael Cinquemani, president & CEO, Master Power Transmission, Inc. (Greenville, S.C.)

Council Members:

- Tammy Balogh, vice president human resources, Flexco (Downers Grove, Ill.)
- Andrew A.O. Brown, vice president, Whittet-Higgins Company (Providence, R.I.)
- Chester Collier, senior VP & general manager Bio Circle div., Walter Service Technologies (Windsor, Conn.)
- Randy Disharoon, vice president global accounts, Rexnord Industries, LLC (Milwaukee, Wis.)
- Jeff Moore, executive vice president marketing mechanical PT, Baldor Electric Company (Greenville, S.C.)
- Michael Nisenbaum, director industrial sales & marketing, Carlisle Belts by Timken (Franklin, Tenn.)
- Jos Sueters, vice president, Tsubaki of Canada Limited (Mississauga, Ontario, Canada)

The Power Transmission Distributors Association (PTDA) is the leading global association for the industrial power transmission/motion control (PT/MC) distribution channel. Headquartered in Chicago, PTDA represents power transmission/motion control distribution firms that generate more than \$16 billion in sales and span over 2,500 locations. PTDA members also include manufacturers that supply the PT/MC industry. (www.ptda.org)

Rexnord

ANNOUNCES COMPREHENSIVE U.S. SERVICE NETWORK

Rexnord is proud to announce the Rexnord Industrial Service Network (RIS), formerly Falk Renew. Offering the largest network of gear drive service centers and certified repair shops, RIS provides fast repair solutions throughout the life-cycle of your Falk Gear Drive or system.



Rexnord Industrial Services are available in two alternative forms.

Rexnord Service Centers: Rexnord is now offering the high quality, Rexnord repair experience for Falk industrial gear drives in cooperation with Falk gear drive repair shops in the United States. These new Service Centers repair Falk drives as exclusive partners after passing an extensive facility and process audit. This includes Falk gearbox repair services, largest network in the United States, Rexnord original OEM Warranty, OEM trained personnel and authentic Falk Gear parts.

Rexnord Certified Falk Repair Shops: Certified by Rexnord, Falk Repair Shops are some of the local partners that you already know and trust. In addition to offering the Rexnord products and services that you expect, Certified Repair Shops can now receive and repair Falk Gear Drives. These shops have passed extensive facility and process audits, always use Authentic Falk Gear Parts and back their work with an exceptional warranty. This includes certified gear drive repair services, authentic Falk parts and Rexnord repair endorsement. (www.rexnord.com)

Schaeffler

CELEBRATES ANNIVERSARY AND CAPITAL INVESTMENTS IN OHIO

For 40 years, Schaeffler Group USA Inc. — a global automotive and industrial supplier — has invested in the United States, with a significant portion of those investments taking place in Wooster, Ohio. Schaeffler commemorated the facility's 40th anniversary with a grand opening celebration and unveiling of the company's latest capital investment.

"Schaeffler's \$60 million capital investment in its Wooster facility advances our automotive business and strongly secures our position as the community's largest employer," said Marc McGrath, president, Automotive Americas, Schaeffler. "From 2007 to 2016, our automotive business has outpaced market growth by nearly double and we have expanded our manufacturing capacity in Wooster to keep pace with growth in our automotive transmission business. This latest investment, which builds upon our recent \$36.5 million expansion of our Fort Mill, S.C. facility, prepares us for the production of future transmission technologies and further optimizes the

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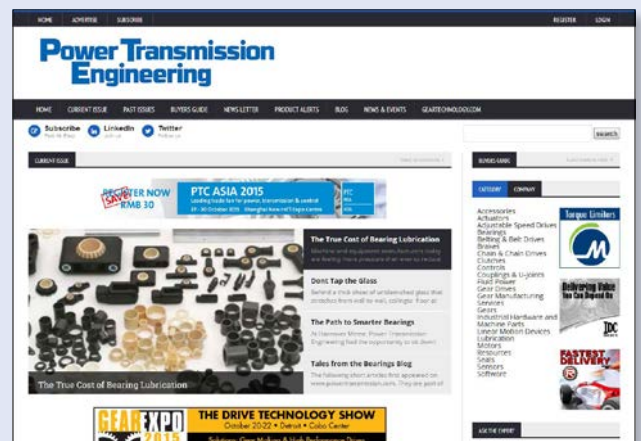
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facility's vertical integration."

"Schaeffler's growth in Wooster from a small team to a manufacturing campus recognized worldwide has engineered the template for international business development," Lt. Governor Mary Taylor said, adding: "Schaeffler operates at the highest quality whether in product, people, or community leadership. We are delighted to call them Buckeyes."

Developing Tomorrow's Technologies, Today

Throughout the company's 40-year history in Ohio, Schaeffler has expanded the Wooster campus 20 times, growing from 27,000 square feet to over 800,000 square feet. What began as a team of six employees assembling manual clutches has evolved into nearly 2,000 highly skilled employees. Today, the Wooster facility serves as the global center of competency for Schaeffler's torque converter product line, and is the leading facility for the development and production of Schaeffler's E-Mobility solutions in the United States.

"The expansion has allowed us to streamline our current operations and has prepared us for our future development and production of tomorrow's mobility solutions," said McGrath. "It is a testament to the success of the team in Wooster and positions them to carry forward the tradition of delivering the best solutions for today's vehicles, while also leading the way in future electrification."

The newly renovated campus equips the Wooster team with the capacity for the production of hybrid modules, the first of many Schaeffler electrification solutions, and a critical part of the company's E-Mobility strategy. To meet the high comfort demands of its customers, Schaeffler's hybrid module combines the electric motor with hydrodynamic converter to yield a P2 hybrid solution. Beginning in 2018, the Wooster facility will produce the module for a major U.S. automotive manufacturer. The team is also engaged in pure electric vehicle technology development, including the production of Schaeffler's e-Axles.

Schaeffler Group CEO Klaus Rosenfeld said, "Our investment in Wooster, Ohio is another milestone in implementing our strategy 'Mobility for tomorrow.' Wooster will become our U.S. center for advanced E-Mobility development and will continue shaping mobility through innovation and employee development. This new, state-of-the-art manufactur-

ing facility is a testament to our commitment to attract and retain the best and brightest talent."

People Come First: Wooster Apprenticeship Program

In addition to its high-tech facility, Schaeffler's nationally recognized apprenticeship program also helps the global supplier to attract and cultivate top talent in Wooster. Launched in 1980, the program is now the largest program of its kind in Ohio and currently partners with the University of Akron (UA) to offer a variety of courses, including Tool and Die, Industrial Design, Maintenance Electrician and more. The 3.5-year program consists of both classroom and on-the-job training, and currently has an 83 percent retention rate after graduation, providing the Wooster facility with skilled employees that ensure its long-term competitiveness.

Through partnerships with the UA and other Ohio universities and colleges, the Wooster campus develops employees through a number of programs, including:

A Co-Op Program that teaches Schaeffler-specific skillsets while students complete their degree in business or engineering.

An Advanced Manufacturing Degree Program, a partnership with the UA that allows skilled workforce to earn a degree in advanced manufacturing while working fulltime.

Additional training and development programs that foster leadership development, project management, foreign language learning, quality and safety skills. (www.schaeffler.us)

Regal Beloit Corporation

ANNOUNCES DOE GRANT FOR MARATHON MOTORS DIVISION

Regal Beloit Corporation has announced that its Marathon Motors division has been awarded a U.S. Department of Energy (DOE) grant for research and development aimed at increasing energy efficiency in electric motors.

The DOE announced a total of \$25 million in grants to enable innovative electrical motors used in manufacturing, of which Marathon Motors received \$1 million to develop a rare-earth element-free motor.

"The Marathon Motors team will direct its research toward developing a motor that uses soft magnetic composites and grain-oriented electrical steel in a state-of-the-art, con-



cal air gap motor architecture,” said Paul Knauer, advanced technology manager, Regal Beloit. “The project aims to improve the efficiency of a 5-kilowatt standard industrial motor from 92 percent to greater than 96 percent.”

According to the U.S. Office of Energy Efficiency and Renewable Energy, industrial electric motors account for 70 percent of the overall electricity usage by U.S. manufacturers and almost 25 percent of all electricity used nationally. The goal of the grant program is to increase the efficiency of these motors, thereby saving energy and increasing competitiveness for American manufacturers.

“The opportunity to further develop this motor technology came with Regal’s acquisition of NovaTorque’s technology in 2016 and our ability to now leverage that technology,” said Howard Richardson, business leader, Regal Beloit. “We are excited to be a part of this grant, along with the research group of Dr. Hamid Toliyat of Texas A&M, which is assisting in model and testing development.”

NovaMAX motors are ultra-efficient, electronically commutated permanent magnet motors. These energy-efficient motors are designed for use in air handling, pumps, data centers, conveyors, general industrial and other applications. Additional information about the DOE grant program is available at energy.gov. (RegalBeloit.com)

BSA

ANNOUNCES 2017 EXCELLENCE AWARD WINNERS

On October 16, 2017, BSA President, Brian Davis, B&D Industrial, presented the Annual BSA Excellence Awards for outstanding service by bearing distributors and bearing manufacturers to the end use customer.

The Bearing Manufacturer Excellence of Innovation in Product Design Award recognizes companies for innovation and excellence in product design or technology. BSA distributor members reviewed and ranked manufacturer innovation submissions. Among the abundance of innovative product designs submitted, three submissions were chosen to be recognized for their outstanding service to the end use customer.

The first Manufacturer Excellence Award was presented to NTN Bearing Corporation of America for the NTN SPAW Mounted Roller Bearing. The NTN SPAW is a one piece solid ductile iron housed bearing that dimensionally interchanges to a standard split block SAF style bearing unit.

The second Manufacturer Excellence Award was presented to SKF USA, Inc. for the SKF Multilog Online System IMx-8. The SKF Multilog Online System IMx-8 provides online monitoring in a compact footprint that offers you more flexibility. SKF Multilog IMx-8 provides a complete system for early fault detection.

The third Manufacturer Excellence Award for outstanding service to the end use customer was presented to The Timken Company for the Timken Tapered Double Inner (TDI) Roller Bearing for Wind Turbine Main Shafts. This is The Timken Company’s second Manufacturer Excellence Award, they are the first company to be a recipient of this award more than once.



SKF won an award from BSA for its online monitoring system IMx-8.

The 2017 BSA CBS Excellence Award recognized two BSA Distributor Companies, one company with the highest percentage of their inside and outside sales force having attained CBS status, the other company with the greatest percentage increase of its inside and outside sales force having attained Certified Bearing Specialist status. BSA’s Certified Bearing Specialist (CBS) program is the only bearing industry-specific program that identifies and quantifies the specific skill sets to certify an industry professional as a bearing specialist.

BSA recognized two companies with Honorable Mentions for this Award. BDI USA and Bearing Headquarters Company received special recognition for the number of Certified Bearing Specialists within their sales force. This award recognizes the esteemed regard of the CBS designation by these two companies specifically, and the industry as a whole.

The first 2017 CBS Excellence Award for the highest percentage of its inside and outside sales force having attained CBS status was presented to BDI Canada, Inc. The second 2017 CBS Excellence Award for the greatest percentage increase of its inside and outside sales force having attained CBS Status was presented to B&D Industrial.

(www.bsahome.org)

January 8–12—SciTech 2018 Kissimmee, FL. From its creation in 1963, the American Institute of Aeronautics and Astronautics (AIAA) has organized conferences to serve the aerospace profession as part of its core mission. Spanning over 70 technical discipline areas, AIAA's conferences provide scientists, engineers, and technologists the opportunity to present and disseminate their work in structured technical paper and poster sessions, learn about new technologies and advances from other presenters, further their professional development, and expand their professional networks. The AIAA Science and Technology Forum and Exposition (AIAA SciTech) has continued to grow in each succeeding year, drawing participants from around the globe. SciTech participants tackle the most pressing issues impacting the future of aerospace, while the technical program presents innovative research and technologies that offer solutions. For more information, visit www.aiaa-scitech.org.

January 17–19—A3 Business Forum Orlando, FL. The A3 Business Forum is the world's leading annual networking event for robotics, vision & imaging, motion control, and motor professionals. Over 550 global automation leaders attended the 2017 show. The 2018 agenda includes TED Talker Mike Rayburn presenting his "What If...?" Experience. This transformational, hilarious keynote presentation uses guitar work and comedy as a metaphor, to illustrate three tools designed to turn your team into an army of innovators; Alan Beaulieu will examine the 2018 Economic Outlook and Forecast and Attorney Eric O'Neill will examine cybersecurity issues. Nine breakout sessions will cover the latest best practices in automation. For more information, visit www.a3automate.org.

January 22–26—World of Concrete 2018 Las Vegas, Nevada. Original equipment manufacturers from around the world and exclusive U.S. distributors of equipment, tools, products and services for the commercial construction, concrete and masonry industries attend World of Concrete. The show attracts approximately 1,500 exhibitors and occupies more than 700,000 net square feet of indoor and outdoor exhibit space. World of Concrete is the premier event for the commercial construction trades. Education tracks include engineering, safety and risk management, general business, business and project management and concrete 101. Interactive workshops include trainer training, construction boot camp, sales and more. For more information, visit www.worldofconcrete.com.

January 23–25—AGMA Gearbox System Design Las Vegas, Nevada. Attendees will explore the supporting elements of a gearbox that allows gears and bearings to do their jobs most efficiently. Learn about seals, lubrication, lubricants, housings, breathers, and other details that go into designing gearbox systems. Gear design engineers; management involved with the design and manufacture of gearing type components; metallurgists and materials engineers; laboratory technicians; quality assurance technicians; furnace design engineers; and equipment suppliers should attend. Instructors include Ray Drago and Steve Cymbala. For more information, visit www.agma.org.

January 30–February 1—IPPE 2018 Atlanta, Georgia. The International Production & Processing Expo is the world's largest annual poultry, meat and feed industry event of its kind. A wide range of international decision-makers attend this annual event to network and become informed on the latest technological developments and issues facing the industry. The 2017 IPPE featured more than 8,018 international visitors from over 129 countries. Mexico and Latin American/Caribbean countries represent the largest region of international visitors, but there has been continued growth in numbers coming from Europe. Canada represents the largest single country outside the United States with regards to number of attendees. For more information, visit www.ippexpo.org.

February 5–7—AGMA Gear Materials Clearwater Beach, FL. Learn what is required for the design of an optimum gear set and the importance of the coordinated effort of the gear design engineer, the gear metallurgist, and the bearing system engineer. Investigate gear-related problems, failures and improved processing procedures. Gear Engineers, gear designers, application engineers, people who are responsible for interpreting gear designs, technicians and managers that want to better understand all aspects of gear design should attend. Instructors include Ray Drago and Roy Cunningham. For more information, visit www.agma.org.

February 22–24—IPTEx 2018 Mumbai, India. Designed to meet the growing need for excellence in all aspects of the gears and power transmission to an imperative for all players to stay competitive. IPTEx 2018 is an important event for all relevant stakeholders in automobile, aerospace, or energy as well as manufacturers, buyers, partners, and consultants. IPTEx will provide a consistent channel of communication to the members of this industry to come together under one roof and participate in technical seminars, share knowledge and expertise with industry leaders and to be a part of discussion on policy codes, standards and challenges faced by the industry. Grindex 2018 is co-located with IPTEx 2018. For more information, visit www.iptexpo.com.

March 7–10—The MFG Meeting 2018 Miami, Florida. Hosted by two major manufacturing trade associations, AMT – The Association For Manufacturing Technology and National Tooling and Machining Association (NTMA), The MFG Meeting brings together the complete manufacturing chain for a unique conference experience. This event provides unparalleled opportunities to network with industry leaders and the agenda topics are designed to address key business challenges and provide actionable solutions. The event is intended for senior leadership, executives, vice presidents, senior sales directors, manufacturing technology's builders, distributors and end users. Learn about the future challenges and opportunities facing the American manufacturing industry, discover new ideas and participate in interactive discussions. For more information, visit www.mfgmeeting.org.

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

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History of a Forgotten Engine

Alex Cannella, News Editor

In 2017, there's more variety to be found under the hood of a car than ever. Electric, hybrid and internal combustion engines all sit next to a range of transmission types, creating an ever-increasingly complex evolutionary web of technology choices for what we put into our automobiles.

But every evolutionary tree has a few dead end branches that ended up never going anywhere. One such branch has an interesting and somewhat storied history, but it's a history that's been largely forgotten outside of columns describing quirky engineering marvels like this one. The sleeve-valve engine was an invention that came at the turn of the 20th century and saw scattered use between its inception and World War II. But afterwards, it fell into obscurity, outpaced by the poppet valves we use in engines today that, ironically, it was initially developed to replace.

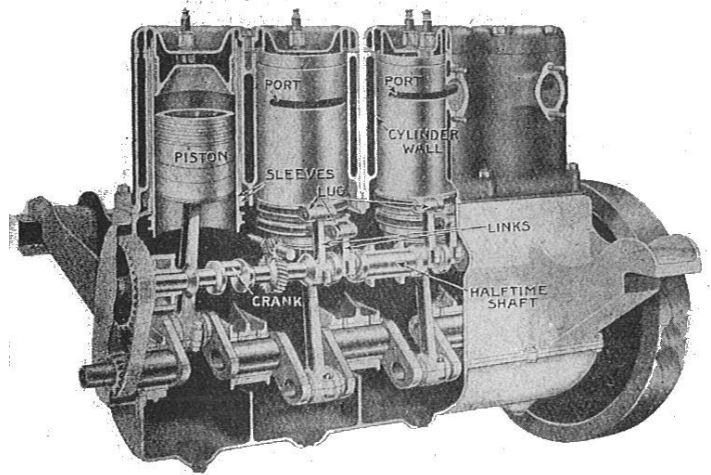
Back when the sleeve-valve engine was first developed, the poppet valves in internal combustion engines were extremely noisy contraptions, a concern that likely sounds familiar to anyone in the automotive industry today. Charles Yale Knight, the entrepreneur behind the design, took a note from steam engines to develop a contraption that replaced the poppet valves with metal sleeves that would slide around an engine's pistons. And Knight's initial design called for not one, but two sleeves around each piston. The idea was that the sleeves had holes in them that, as the sleeves slid around the piston, would line up periodically with the intake and exhaust ports on the piston, managing the entry and release of fuel, air and exhaust during the entire process.

The engine worked. It was indeed vastly quieter than other engines of the day, but it was also more efficient than other engines: It was better at getting air in and out of the engine and eliminated valve float and bounce. The tradeoff, of course, was that it was expensive and complicated to manufacture and required more oil than other models to ensure the sleeves continued to move smoothly over the pistons. In fact, the "Silent Knight" sleeve-valve engine is almost as well-remembered for its hunger for oil as for its quiet footprint.

Despite being an American invention, the sleeve-valve engine found its first success in Europe, and by 1913, was being utilized by six car manufacturers across four countries, including Daimler, the first manufacturer to give the design a chance, and Mercedes.

Soon enough, even American companies that had at first turned their nose up at the invention were incorporating it in their car designs. The engine reached its peak soon after when John North Willys built an automobile manufacturer almost entirely around the Silent Knight engine design, the most popular of which became the Willys-Knight, a mid-priced vehicle that sold well through the '20s.

The sleeve-valve engine, however, was one of many casualties of the Great Depression. Willys' company limped



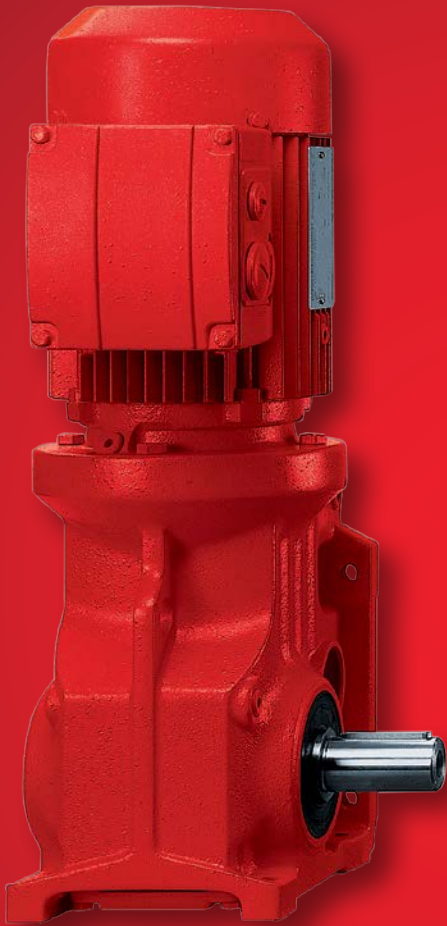
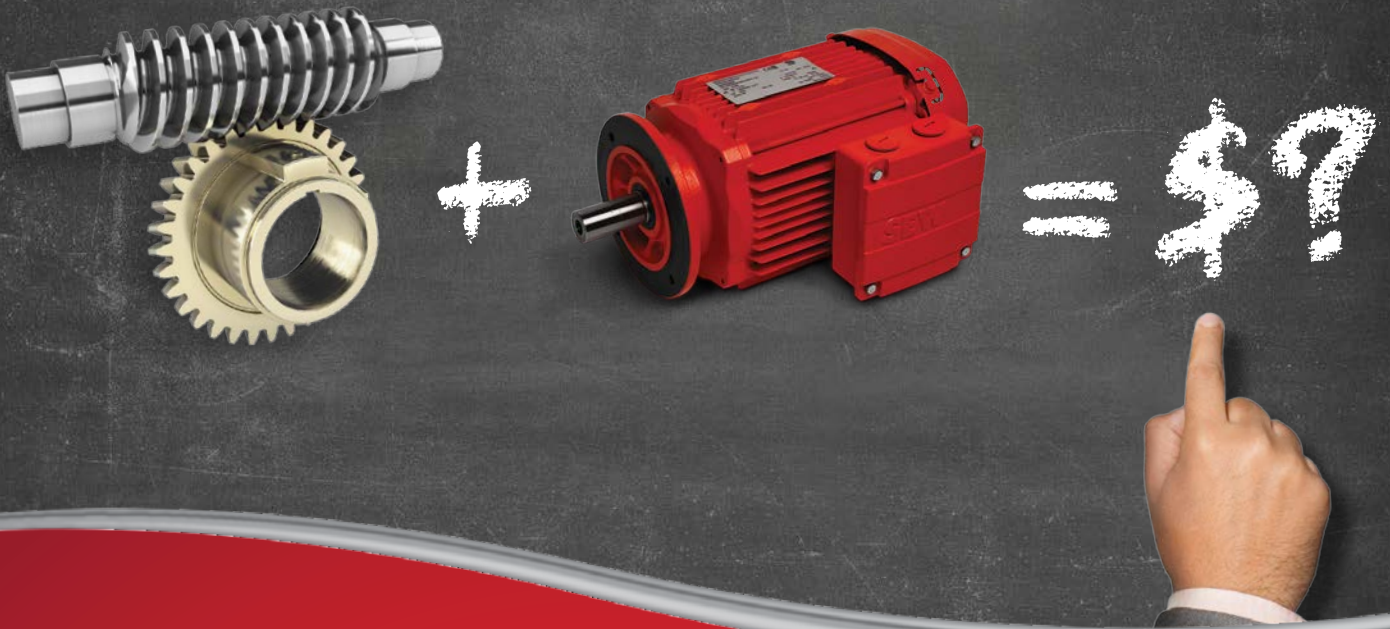
(By Andy Dingley (scanner) - Scan from The Autocar (Ninth edition, circa 1919) Autocar Handbook, London: Iliffe & Sons., pp. p. 38, fig. 21, Public Domain, <https://commons.wikimedia.org/w/index.php?curid=8771152>)

through the economic downturn, and by the time the economy was looking up again, poppet valve engines had caught up to the sleeve-valve and were quickly becoming just as quiet and efficient.

The sleeve-valve never saw another iteration in automobiles, no attempt to innovate and regain an advantage over other technologies, but it did still have one last moment to shine during World War II. Instead of in an automobile, however, the sleeve-valve was instead implemented in the Hawker Typhoon, a British fighter-bomber that became a mainstay in the RAF during the later years of the war. The Typhoon became Britain's answer to Germany's Focke-Wulf Fw 190, a fighter that had previously been able to outrun almost anything else in the air. And once the Luftwaffe, and by extension, the Fw 190, had been largely taken out of the war, the fighter-bomber found another use as a ground attack aircraft, bombing both infrastructure targets such as bridges and supply trains and enemy tanks alike.

Both of the Typhoon's primary functions were in part made possible by the sleeve-valve engine design it carried, which provided the extra horsepower needed for the fighter to keep up with the Fw 190 and carry heavier munitions like bombs.

Again, however, the sleeve-valve was eventually replaced by the jet engine, and it's rested in obsolescence ever since. Or at least, until very recently. Pinnacle Engines, a fresh contender on the market, has taken a second look at the sleeve-valve engine. Designed to produce reduced CO2 emissions at a higher fuel efficiency at comparable cost to other automotive engines, Pinnacle Engines is marketing their product to markets such as China and India that they view as hungry for those exact advantages. It's too early to tell whether the engine will catch on or not, but the company did receive a fair amount of attention in 2014-15, even drawing the attention of media outlets such as *Forbes*. It may be that the sleeve-valve is about to get a third day in the sun. **PTE**



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