GEAR TECHNOLOGY

MAY/JUNE 2001
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- LOAD DISTRIBUTION IN PLANETARY GEARS
- Q&A: CALCULATING LOADS, SIZING GEARS & FIGURING FACTORS
- EVALUATING CONTACT STRESS IN GENERALIZED GEAR PAIRS

ALSO IN THIS ISSUE:
- GEARBOX FIELD PERFORMANCE
- GEAR COMPANIES AND THEIR END-PRODUCTS

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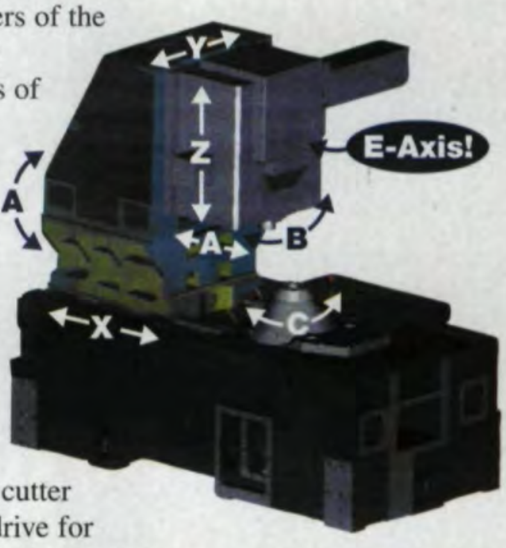

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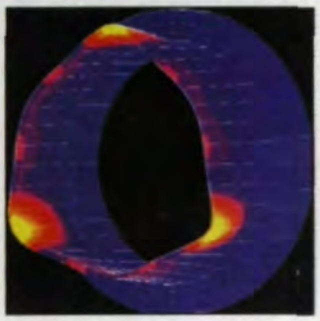
FEATURES

FOCUS ON GEAR DESIGN

27

DEPARTMENTS



11

56

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The Journal of Gear Manufacturing

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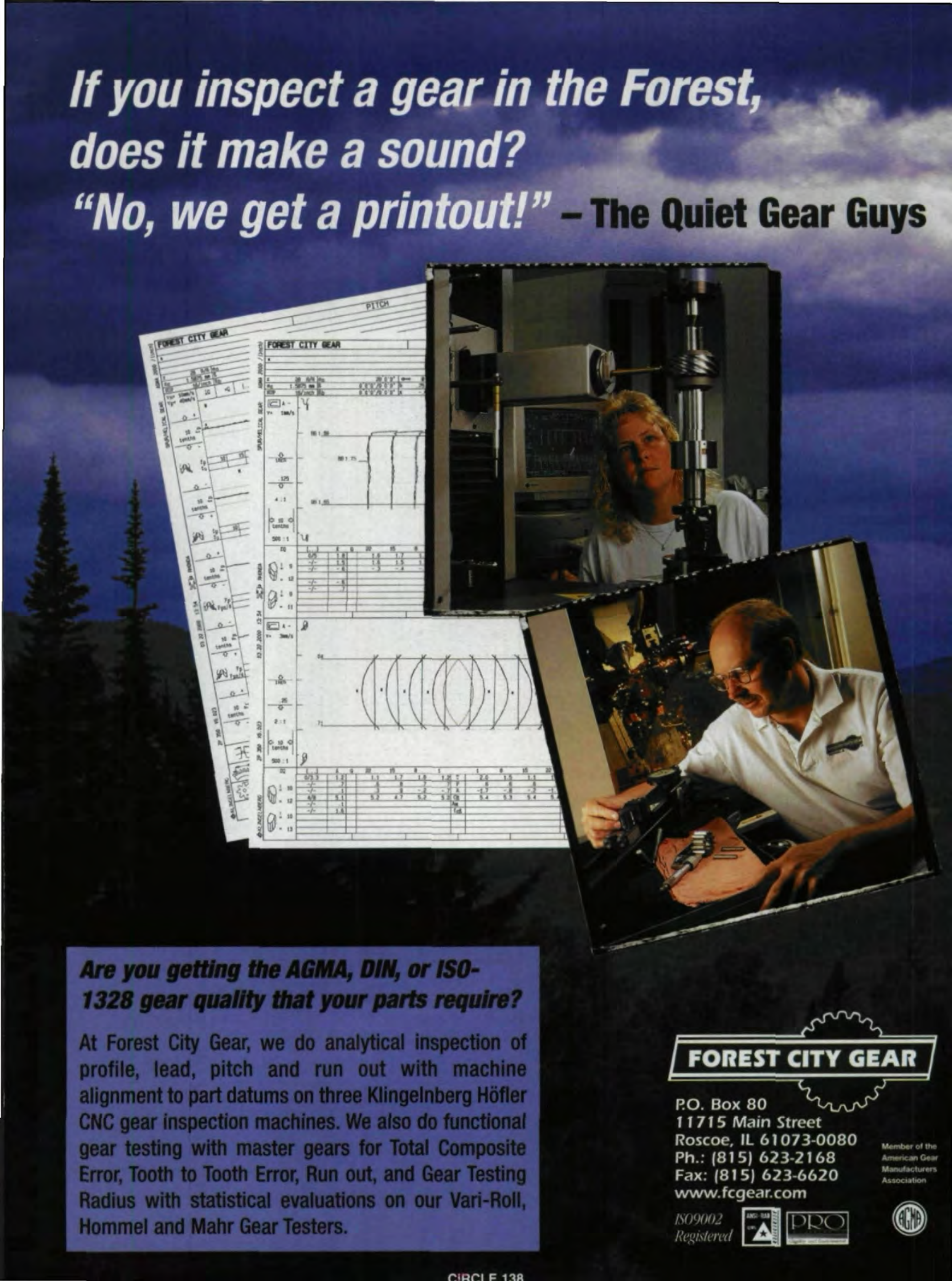
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Core Competency

Concentrating on the Crucial Components Can Keep Your Company Competitive

Rodgers and Hammerstein produced some of America's most memorable and lasting songs in musical theater. Lyricist Oscar Hammerstein II once said of composer Richard Rodgers, "I hand him a lyric and get out of his way." Hammerstein knew what Rodgers was good at, and vice versa, and each trusted his partner. Their partnership was so successful that you can scarcely think of one man without the other.

Musical composers aren't the only ones who form successful partnerships. In today's manufacturing world, partnerships are almost a necessity. Original equipment manufacturers have competition not only from other manufacturers of similar equipment, but also from manufacturers of the components that go into their equipment and the service providers who process some of those components along the way.

Ron Davis of Caterpillar Inc. made this point at the recent annual meeting of the American Gear Manufacturers Association. Davis works in Caterpillar's business administration division, which helps decide when to manufacture components in-house and when to outsource them.

Caterpillar equipment requires a broad range of gear products, including transmission shafts, straight and spiral bevel gears, planet gears, sun gears and final drive ring gears. They use large quantities of both machined and powder metal gears.

But Caterpillar can't hope to stay competitive by manufacturing all of its components when other companies specialize in manufacturing some of those components, Davis says.

The challenge is to decide which components and manufacturing processes give Caterpillar the greatest competitive advantage when manufactured in-house, and which give them the greatest competitive advantage when outsourced. Of course, the company examines factors such as cost, quality and customer satisfaction when considering competitive advantage. Another factor is whether Caterpillar has special expertise in manufacturing a particular component. The company also examines various risk criteria: whether a part requires proprietary technology, whether it's an aftermarket-critical component or whether its time-to-market is critical.

Those components that give them the greatest competitive advantage when manufactured in-house are part of Caterpillar's core competency. Outsourcing those components could lower the quality of the end-product, make it more expensive or otherwise worsen the company's position in the marketplace. Components that are not a part of Caterpillar's core competency might be better manufactured by others, specialists who can lower the costs, improve the quality or both. Outsourcing those non-core components also affords Caterpillar the opportunity to devote its own resources to improving those processes and products that are part of its core competency.

Many of Caterpillar's less critical, high-volume components, such as engine gears, are generally outsourced. Critical components, such as large transmissions or final drive gears, are generally manufactured in-house. Medium-duty components, such as small axles and small

transmissions, are mixed between in-house manufacturing and outsourcing.

Some of the outsourced gears are manufactured by Avon Gear Co., a division of Okubo Gear Co. Ltd. in Rochester Hills, MI. At the AGMA meeting, I had the chance to talk with Avon Gear's president, Aaron Remsing.

According to Remsing, there was a time not long ago when Avon Gear had a near-death experience. Business was down, the company had just moved into a new, expensive building, and the future was uncertain. Avon Gear had to do some soul-searching about its business. Management was forced to think about the company's capabilities and reinvent the way that it obtained new business, Remsing says. The answer lay in examining the company's own core competencies.

Avon Gear knew it had the technical expertise, equipment and experience to manufacture certain sizes, types and quantities of gears of excellent quality and at competitive prices. Instead of waiting for orders to come in, Remsing says, the company began approaching those customers who could best take advantage of Avon Gear's core competencies. One of those companies was Caterpillar.

By making a commitment to specialize in a certain type of gear for a certain market segment, Avon Gear was able to capture Caterpillar's business. That business, in turn, allowed the company an opportunity to invest in new equipment and training to provide even greater competitive advantage.

Today, according to Davis, Avon Gear is one of the most important suppliers for Caterpillar's transmission business unit. Working together, the companies are able to provide the world-class manufacturing, quality and pricing on gear components that help keep Caterpillar's equipment competitive in the world market. Furthermore, the processes, machines and expertise developed by Avon Gear have given the company the ability to sell gears of world-class quality and price to other customers with similar needs.

In fact, Davis says, the partnership between these two companies has developed into one of mutual trust and planning. Like most successful partnerships, it's now in the best interest of each party that the other doesn't fail.

Like Caterpillar and Avon Gear, each of our own companies has core competencies—things that are central to our business, which years of working in certain areas allow us to do better than other companies without the same focus. Finding those things, concentrating on them and expanding on them can help make us more successful.

Not finding and developing your core competency can have the opposite effect. It's true that some companies will be more focused than others. But even the most generalist of companies has core competencies. For example, a general-purpose gear job shop with a broad range of equipment and tooling might not seem to have a specific niche. Nevertheless, that same company might specialize in fast turnaround, design assistance or extremely high quality, each of which could be developed for greater competitive advantage in certain marketplaces.

You can learn more about your core competencies by choice or by necessity. If you do so by choice, perhaps you'll be able to make intelligent decisions about your company's operations, as Caterpillar did. However, if you wait, you may be forced to learn about your core competencies the hard way. Perhaps you'll be able to act quickly enough to save your company, as Avon Gear did. But then again . . .

Michael Goldstein,
Publisher & Editor-in-Chief


We're our most demanding customer...
but want you to prove us wrong.

Harmonic Concepts Jazz Up Planetary Design

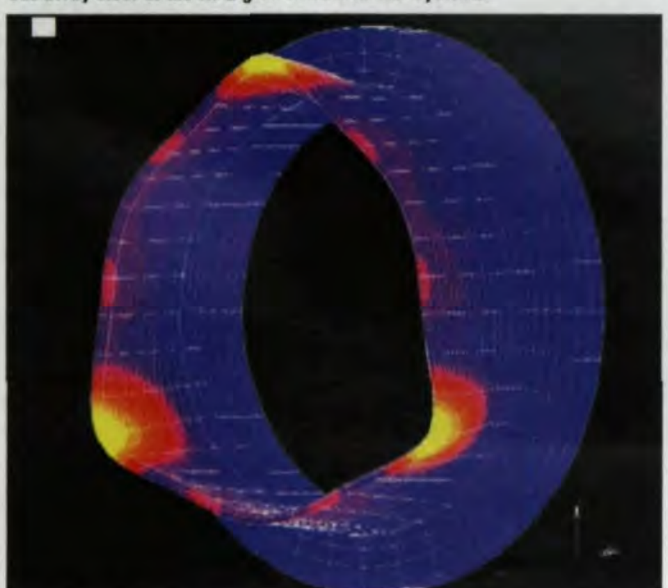
In the world of planetary servo drives, precision is everything. Advertisements abound for servo gear drives with extremely low or even zero backlash. With that type of drive in high demand, manufacturers have done everything they can to tighten their designs—through more exacting tolerances, tighter assembly and better materials.

Instead of tightening, one manufacturer of planetary servo drives has tried loosening in order to provide the low backlash that manufacturers of robotics and other precise positioning equipment require.

That loosening has nothing to do with the manufacturing tolerances or assembly. Precision manufacturing is still required. It has to do with a type of flexibility borrowed from a related technology: harmonic drive gearing.

Harmonic drive gearing has long been known for its low backlash, high precision and compact design. Harmonic drive systems use a circular spline, a flex spline and wave generator to create extremely high ratios in a single reduction. One of the advantages of the harmonic drive system is that the flex spline deforms as it rotates to keep its teeth fully loaded against the circular spline at all times.

Cut-away view of the HPG gear drive from HD Systems.



The HPG drive includes a flexible outer ring gear to control backlash (deformation exaggerated for illustrative purposes).

Engineers at HD Systems of Hauppauge, NY, have figured out how to apply the deformation concept of harmonic drive systems to planetary gears as well. Instead of a flexible inner spline, the HPG "harmonic planetary" gearheads use a flexible outer ring gear to preload the gearset.

The flexible outer ring gear of the HPG gearhead creates a continually adjusting backlash compensation method. The ring gear is thinner in the radial direction than a typical ring gear. This makes it radially flexible, yet torsionally stiff, says HD Systems' marketing manager, Brian St. Denis. "As normal gear wear occurs, any increase in space between the teeth is immediately compensated by the preload in the ring gear. The concept is as if the ring gear is a spring in the radial direction."

The system employed by HD Systems has a huge advantage over those commonly employed by other manufacturers, says St. Denis. The backlash of the HPG drive won't deteriorate over time.

One method commonly employed to provide "low" or "zero" backlash in planetary gear systems is to create a "tight fit" between all of the gears. However, this method is subject to gear tooth wear, and over the life of the gearbox, the backlash can increase significantly.

Welcome to* Revolutions, *the column that brings you the latest, most up-to-date and easy-to-read information about the people and technology of the gear industry.* Revolutions *welcomes your submissions. Please send them to* Gear Technology, *P.O. Box 1426, Elk Grove Village, IL 60009 or fax (847) 437-6618. If you'd like more information about any of the articles that appear, please circle the appropriate number on the Reader Service Card.

Another method is to use preloaded bevel gears as part of the mechanism. Shims force the bevel gears together axially, which forces a radial preload in the planetary gears. Backlash in this type of system also increases over time because of gear tooth wear.

Because of the flexible deformation of the planetary ring gear, the HPG gearhead automatically compensates for gear tooth wear, St. Denis says. This helps ensure consistent backlash throughout the rated life of the gearbox.

The HPG Series planetary gearheads have standard backlash of less than 3 arc-minutes, and they can be ordered with 1 arc-minute of backlash as an option. They are available in ratios from 5:1 to 33:1.

Circle 301

Microgears: For Use In The Human Machine?

In a Hungarian lab, using a microscope's laser, researchers are creating gears and rotors thinner than a human hair. And, while hair grows out of the body, the researchers want to put their gears and rotors into the body, in microscopic machines for conducting experiments.

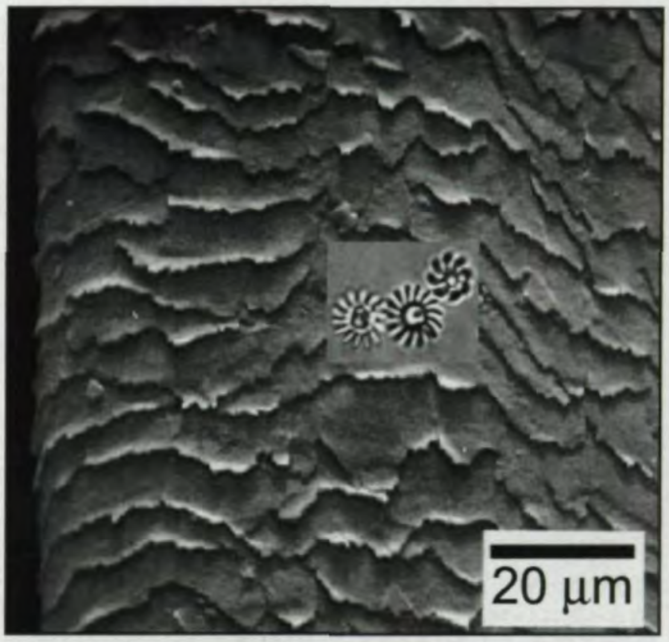
"It's like the movie *Inner Space* with Dennis Quaid," says researcher Pál Ormos. The machines could be built so small that they would fit into blood vessels, for analyzing blood.

To create the gears and rotors, Ormos uses hardened liquid photopolymer. But, unlike the microparts and their possible use, the material isn't nearly as exotic.

"Dentists use it a lot," Ormos says. The material is the same as what's used for tooth fillings hardened with ultraviolet light.

Ormos developed the technique with Péter Galajda in a little more than a year—"Just now, we are perfecting it." Ormos adds that he'd like to use the technique in less than a year to build several types of machines.

With a doctorate in physics, Ormos belongs to the Hungarian Academy of Sciences and works at its Biological Research Centre, located in Szeged. Ormos is director of the centre's Institute of Biophysics. Galajda is his doctoral student.



Created under a microscope, three rotors appear in front of the oak-like trunk of . . . a human hair.

"We study the physics of living systems," Ormos says. He adds that there's a strong drive in biology toward miniaturizing, which has "a lot of promise in it."

Ormos and Galajda's gears and rotors could be used in that study as parts in microscopic biochemical devices. The micromachines would be used to investigate living systems at a cellular level, permitting the movement of individual cells—"even individual large molecules," Ormos says. "It's a very precise tool that can be used in biology."

With such machines, scientists could introduce chemicals into cells, manipulate their reactions, and maintain those cells' environments, pumping different solutions to different regions. They could measure cell properties to select certain cells, create a channel for transporting those cells and transport the cells individually. They could also remove compounds from cells, even destroy cells.

The gears' and rotors' diameters can be as small as two microns. A human hair's diameter is about 100 microns. At two microns, the parts can weigh as little as 10^{-10} grams.

Ormos and Galajda make their microparts by placing liquid photopolymer under a microscope, then moving a computer-controlled laser focus into the microscope. Wherever the focus moves, the liquid becomes hardened. If it moves in a gear shape, the focus makes a hardened gear.

"We can create a three-dimensional object that is as complicated as we want," Ormos says.

But, there's a wrinkle in the technology that needs to be ironed out.

The microgears are rotated by the microrotors. The microrotors are rotated by light.

"It's not easy to introduce light into the blood vessels," Ormos says.

But, he adds there may be a solution to that problem. Ormos explains that the machines in the body could be powered by light, if it were introduced effectively—via a thin optic fiber, for example.

In similar research, Sandia National Laboratories in New Mexico has used chemical etching to create microgears and experimental microengines, electric motors no larger than a grain of sand.

Ormos started his research to understand how light causes different trapped particles to rotate. But, he later realized such light-induced rotation could have useful applications.

"It was not really the area in which we were deliberately moving," he says.

Ormos is unsure about the technique's future—"Whether this can be commercialized or not, isn't clear." Still, he and Galajda are testing resins to decide which works best for creating their gears and rotors.

"I think the possibilities are really unlimited," Ormos says. "Anything that can be done with larger machines can be done with smaller ones, too."

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T E N T H
International Induction Heating Seminar

AGMA Elects New Officers

The American Gear Manufacturers Association elected its 2001–2002 officers at the association's annual meeting in March. The new officers will help direct the organization through March 2002.

The newly elected officers are:

- Chairman: Arlin Perry, president of Comer Inc. in Charlotte, NC. Perry was formerly president of Gear Products Inc. in Tulsa, OK.
- Vice Chairman: Roger Pennycook, vice president of sales for Textron Power Transmission, North America in Traverse City, MI.
- Treasurer: J. Cameron Drecoll, president of Brad Foote Gear Works Inc. in Cicero, IL.
- Chairman of the Administrative Division Executive Committee: Martin Woodhouse, vice president of sales for Star Cutter Co. in Farmington Hills, MI.
- Chairman of the Technical Division Executive Committee: David McCarthy, vice president of engineering for Dorris Co. Inc. in St. Louis, MO.

Outgoing chairman Frederic M. Young, president of Forest City Gear Co. in Roscoe, IL, assumes the title of chairman emeritus.

Ian Shearing Joins Mitsubishi

Ian Shearing

Ian Shearing has been named vice president of sales for the Mitsubishi Gear Technology Center, headquartered in Wixom, MI, a Detroit suburb.

According to an announcement from Thomas P. Kelly, the center's president, Shearing will be responsible for all U.S. and Canadian sales of the company's line of gear-cutting machine tools and cutters.

Regal-Beloit Buys Spiral Bevel Line from Philadelphia Gear

Regal-Beloit Corp. has purchased the spiral bevel gear product line of the Philadelphia Gear Co., according to a statement released earlier this year. The purchased assets included inventory and selected machinery, equipment and tooling relating to the spiral bevel gear line.

The line, with annual sales of about $4 million, includes design and manufacturing capabilities for Klingelnberg bevel gearing. The business will be moved to Chicago, to the Foote-Jones/Illinois Gear division of Regal-Beloit.

Derlan Acquires Windsor Gear & Drive

Derlan Industries Ltd., a Toronto-based manufacturer of products for the aerospace and pump industries, acquired substantially all the assets of Windsor Gear & Drive Inc. of Windsor, Ontario in March.

Windsor Gear & Drive manufactures gears and drives for aerospace and automotive applications. The acquisition helps Derlan compete in the commodity segment of the aerospace gear market, according to a release issued by Derlan.

Gary L. Forbes Appointed GM of UQM Power Products

UQM Technologies Inc. (formerly Unique Mobility Inc.), a developer of alternative energy technologies in Golden, CO, named Gary L. Forbes as general manager of its UQM Power Products unit in January. The Power Products unit manufactures gears, gearboxes and brushless DC motors at the company's facility in Frederick, CO.

Forbes joined UQM after more than 35 years with Hamilton Sundstrand, where he was most recently plant manager of the company's Denver manufacturing facility. Prior to that, Forbes was managing director of Sundstrand Pacific Aerospace, located in Singapore. Hamilton Sundstrand is a subsidiary of United Technologies Corp.

United Gear & Assembly Adds Heat Treating Capacity

United Gear & Assembly Inc. of Hudson, WI, has installed a large-capacity pusher furnace to carburize, quench and temper gears and shafts. The new furnace has a capacity of 1,000 lbs./hr.

The system has been integrated into United Gear & Assembly's manufacturing operations, so high-volume components can move right from manufacturing cells to the furnace and directly into press quenching, according to a statement released by the company.

United Gear & Assembly manufactures gears, shafts and assemblies for the automotive, agriculture, construction and marine markets.

ASME Codes and Standards Available Online

The ASME Digital Store, which opened in March, allows engineers to browse and obtain electronic versions of ASME's codes and standards via the society's Website at *www.asme.org*.

The online store includes electronic files of 400 ASME publications, including codes and standards for tools, fasteners, flow measurement, pumps, turbines, dimensioning and tolerancing, as well as other categories. Visitors can search the store's database by keyword or category.

Fairfield Expands to India

Fairfield Manufacturing Co. Inc. of Lafayette, IN, acquired Atlas Gear Ltd. in October.

Located just outside Belgaum, India, Atlas Gear is a major supplier of gears for agricultural and off-road vehicles in the Indian market. The division will be known as Fairfield Atlas Ltd. ⭘

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Barit International Corp.	136	4
Becker Gearmeisters Inc.	183	55
Bodine Gear Manufacturing	109	9
Bourn & Koch Machine Tool Co.	144	53
The Cincinnati Gear Co.	170	52
Crown Power Train s.r.l.	146,171	43,52
Dr. Kaiser (S.L. Munson)	122	50
Dura-Bar	158	10
Fairfield Manufacturing Co. Inc.	172	52
Fässler Corp.	125	35
Forest City Gear	138	6
Gear Technology	298	53
Gear Technology CD-ROM Buyers Guide	295	53
Gear Technology Center, Div. of Mitsubishi International Corp.	175	28-29
Gleason Cutting Tools Corp.	105,154,173	BC,55,52
The Gleason Works	110,174	IFC-1,52
Holroyd	142	32
Index Technologies Inc.	185	54
Inductoheat	161	14
Insco Corp.	147	17
Interstate Tool Corp.	155	54
ITW Heartland	119,188	8,53
Klingelnberg Söhne GmbH	143	12-13
Koepfer America L.L.C.	156	55
Koro Sharpening Service	157	55
Kreiter Geartech	192	55
LeCount Inc.	137	51
Metal Media Clocks & More	299	37
MicroGear	148	43
Midwest Gear & Tool Inc.	149	37
Midwest Gear Corp.	160	54
Nachi Machining Technology Co.	114,176	5,52
Niagara Gear Corp.	162	54
Noria Gear Failure Seminar	127	51
On-Line Services Inc.	141	4
Parker Industries Inc.	151	33
Perry Technology Corp.	134	IBC
powertransmission.com	297	53
Precision Gage Co. Inc.	189	16
Presrite Corp.	108, 182	30, 52
Process Equipment Co.	130	38
Profile Engineering Inc.	184	55
Pro-Gear Co. Inc.	163	54
The Purdy Corp.	178	15
Raycar Gear & Machine Co.	177	55
Show Central on The Gear Industry Home Page	290	45
Star Cutter Co.	128,164	2,54
SU America Inc.	107	26
Toolink Engineering	152	17
Transatlantic Connection Inc.	166	54
United Tool Supply	131	45

Taking On Nature: Three Gear Companies And Their End-Products

Joseph L. Hazelton

Many gear companies make parts, build assemblies . . . and stop there. But, some don't stop; they go a step further and create end-products. Three gear companies have taken that step, and taken on nature with their creations.

Ontario Drive & Gear rides across land and water with its amphibious vehicles. ATA Gears funnels water through its turbines for electricity. Amarillo Gear raises temperatures through its wind machines.

Finding Their Niches

Amarillo Gear. Started in 1917, Amarillo Gear Co. of Amarillo, TX, manufactures spiral-bevel, right-angle gear drives and fan drives. The gear drives are used for agricultural pumps, bow thrusters for docking ships, cooling tower fans, drilling rotary table gears, fire control, flood control, sewage treatment plants and water desalination plants.

In 1934, the company started building and servicing gearboxes in agricultural applications.

In 1974, it turned its expertise to upper and lower gearboxes for wind machines.

An employee of ATA Gears Ltd. works on one of the company's water turbines. The turbines are usually used by energy-producing companies and private companies that own river rapids.

Years later, Amarillo Gear used its know-how to develop whole wind machines. In 1989, the company formed Amarillo Wind Machine Co.

Located in Exeter, CA, Amarillo Wind manufactures and services wind machines used to protect crops susceptible to frost, with a relatively high value per acre. The crops include avocados, cherries, flowers, strawberries and citrus fruits, like grapefruits, lemons and oranges.

ATA Gears. Started in 1937, ATA Gears Ltd. of Tampere, Finland, manufactures spiral bevel gears and custom-made gearboxes. The parts and assemblies are for machines that handle heavy or shock loads and operate in extreme temperatures or dusty environments.

ATA Gears' spiral bevel gears are in its water turbines. Pentti Hallila, general manager of sales and technical services, says the company started making water turbines to extend those gears to an end-product.

The turbines are usually provided to energy-producing companies and private companies that own river rapids. Hallila says a company buys a turbine if it happens to be on a river and wants to make electricity from it. He adds that businesses may be in very remote places where they can't get electricity from a public network.

Ontario Drive & Gear. Started in 1962, Ontario Drive & Gear Ltd. (ODG) developed and manufactured a special transmission for a six-wheel-drive, amphibious all-terrain vehicle.

"They were looking for a stable market for their products," says Michael Eckardt, ODG's vice president of finance.

In 1967, the company entered the market with its own amphibious vehicles, called Argos.

Located in New Hamburg, Ontario, Canada, ODG manufactures spur, helical, internal and worm gears. It also manufactures spur, helical and worm gearboxes. Meanwhile, its amphibious vehicles use helical and spur gears. The parts and assemblies are used in agriculture, automation, material-handling, mobile lift equipment, municipal tractors, and transportation.

The vehicles are used for disaster response, emergency medical service, search and rescue, resource exploration, and recreation—like hunting and fishing.

"And anything else where you have to be off the beaten track," Eckardt says.

The Machines In Their Niches

Amarillo Gear. Amarillo Gear's spiral bevel gears are in Amarillo Wind's machines. Amarillo Wind receives the gears, assembles the engines, then creates their frames and the wind machines' towers.

Each machine has two sets of spiral bevel gears: one set in the lower gearbox, one in the upper gearbox. The lower box receives power from the engine and sends it up the pipe's drive line to the upper box, which is connected to the fan blade.

Also, each machine has a set of worm-and-worm-wheel reduction gears, allowing the upper gearbox—and the blade—to rotate 360° around the tower.

As an example, the standard machine for citrus crops is about 35 feet tall, from the ground to the blade's center—tall enough to clear treetops. The blade is 18 feet long.

Tilted 6° down, the blade draws warm air from above, from the inversion layer, and blows it down on the trees, raising the temperature at ground level to protect the crops from frost.

A machine is placed in the center of every 10 acres of citrus trees in an orchard. To affect all 10 acres, the machine blows air at nearly 30 mph.

"It stirs the air quite a lot," says Roger Hein, Amarillo Wind's general manager.

The wind machines can't hold nature at bay. As Hein says, they don't prevent all frost, but they allow farmers to change Mother Nature by a few degrees.

Those few degrees can make the difference, though.

For example, America's largest orchards for eating oranges are in southern California, in the Central Valley. Despite its warm climate, California can have freezing temperatures in November and December.

An employee of Ontario Drive & Gear Ltd. bends over an Argo amphibious vehicle on the assembly line. ODG's amphibious vehicles are built in New Hamburg, Ontario, Canada.

The best temperatures for growing oranges range from a high of 95° to a low of about 30°. At about 27°, oranges can start to have problems, says farm manager Edward Lorenzi of Sun Pacific. Located in Bakersfield, CA, Sun Pacific grows citrus fruits in the Central Valley and uses machines from Amarillo Wind.

The oranges' problems are dissipation of their juice and damage to their appearance.

Oranges have small cells that hold their juice. If the temperature is too low, the oranges freeze, and the frozen juice breaks through the cells. When the oranges later thaw, their juice evaporates. Also, California eating oranges must look appetizing—no wind scars, no fungus.

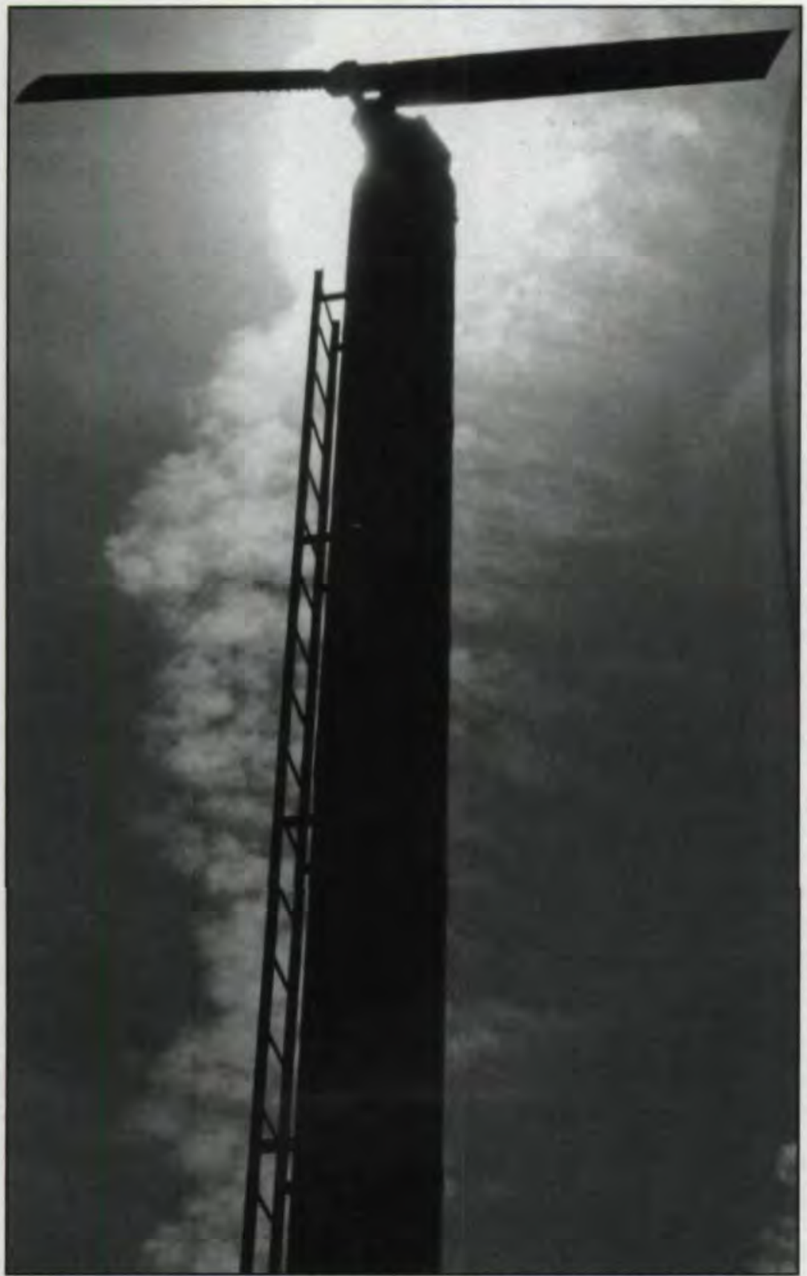
Once damaged, the eating oranges would be used for any remaining juice. Their value would be limited to their juice value.

"The difference between a fresh market orange and a juice orange in California is the difference between making money and losing money," Lorenzi says.

He explains that depending on the market, eating oranges of export quality can sell for as much as $14 per carton. As for a carton of eating oranges used for juice: "You might be able to get $4."

Each wind machine usually runs about 100 hours during winter and protects probably $20,000–$30,000 in citrus fruit. As a farm manager, Lorenzi himself is responsible for 3,000 acres of citrus fruits.

Joseph L. Hazelton
is associate editor of Gear Technology. *Trained in journalism, he was a reporter for two weekly newspapers in Michigan before joining* Gear Technology*'s staff.*



Made by Amarillo Wind Machine Co., a wind machine for citrus crops can protect 10 acres of trees by blowing higher, warmer air down on them. The machines blow the air at almost 30 mph.

Each of a wind machine's gearboxes includes a set of spiral bevel gears, as in this cross section. Amarillo Wind gets those gears from its parent company, Amarillo Gear Co.

A standard machine from Amarillo Wind costs $15,000–$20,000.

ATA Gears. ATA Gears' water turbines create electrical energy from flowing water. The turbine's power depends on the water's net head and flow volume. The turbines usually need water with a head of 6–70 feet, flowing at a rate of 30–1,800 cubic feet per second.

Depending on those factors, the turbines usually generate 20–2,500 kilowatts.

To generate their power, the turbines must be part of a dam. When the dam is opened, water flows through the turbine and rotates the runner blades, driving the shafts and spiral bevel gears that power the generator.

Hallila explains a water turbine's cost depends on size and conditions, but it's roughly $1,000 per rated kilowatt. Also, there may be additional costs for other items, like the dam.

4751 *Ontario Drive & Gear.* The Argos are small vehicles that look like rugged, open automobiles. Depending on the model, they have six wheels and can carry four people or have eight wheels and can carry six people. They have 16-hp to 20-hp engines and bodies made of polyethylene, a type of plastic. The all-wheel-drive Argos can travel at 20–22 mph.

Weighing 765–1,025 pounds, the vehicles might not be expected to float, but they do.

"They can go through any depth of water," Eckardt says. "They're fully amphibious."

In the water, the six-wheelers can carry two people, the eight-wheelers can carry four people. With their wheels as propellers, the Argos can go 2.5 mph on water.

"We don't actually drive the vehicle," Eckardt adds, "we propel the vehicle through the water."

An Argo typically costs $6,000–$11,000.

Gear Companies With End-Products?

According to Hallila, it's unusual for gear companies to make end-products. As he explains, gear companies don't want to compete with their own customers.

Hallila knows of no other gear company that makes water turbines. At ODG, Eckardt says there is one other manufacturer of amphibious vehicles, but it isn't a gear company. Also, Hein isn't aware of any other gear companies with wind machine operations.

The Machines' Markets

Hallila describes the water turbine market as a relatively small market. According to Hein, the domestic market for wind machines is probably either shrinking or staying the same, but the foreign market appears to be growing.

As for amphibious vehicles, Eckardt says the market's size is difficult to determine. He describes it as wide but shallow, explaining there are many niche uses for amphibious vehicles.

But, he adds the market has been growing rapidly during the past few years. "It mirrors—in a way—the ATV market, for the four-wheelers."

Niches in Production?

Having gone a step further, the three gear companies have made their end-products part of their operations. But, the step hasn't had the same effect on all of them.

ATA Gears makes its small-scale water turbines only when customers specially order them. And, it designs them to suit customers' local conditions. Amarillo Gear's subsidiary makes its end-products as part of its daily operations. Amarillo Wind makes 200–300 wind machines each year. Likewise, ODG makes its amphibious vehicles in its daily operations—about 2,000 vehicles each year.

Hallila says the water turbines are a specialized part of ATA Gears, not in the company's mainstream operations. For example, ATA Gears has 170 employees, but the number of employees making water turbines varies depending on workload.

Hein also describes Amarillo Wind as a specialized part of Amarillo Gear, not in its mainstream operations. Amarillo Gear has more than 130 employees, Amarillo Wind has 21—and only eight of them build wind machines.

But, ODG's amphibious vehicles are its mainstream. Out of 110 employees, 30 work in sales and administration, 34 in the gear division, and 46 in the vehicle division.

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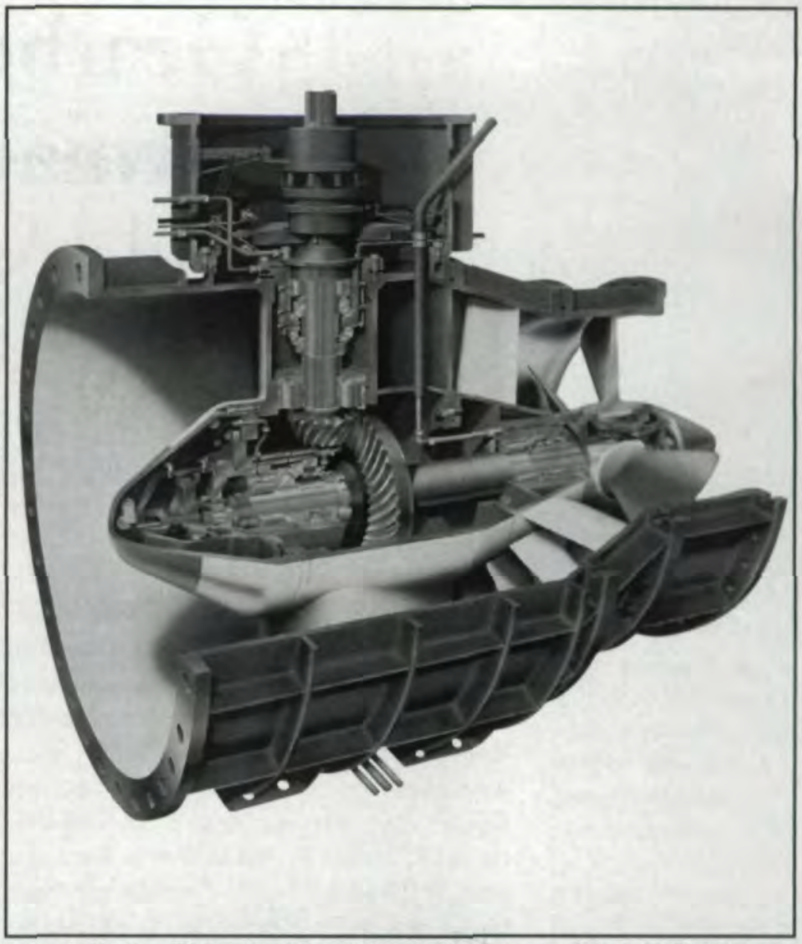
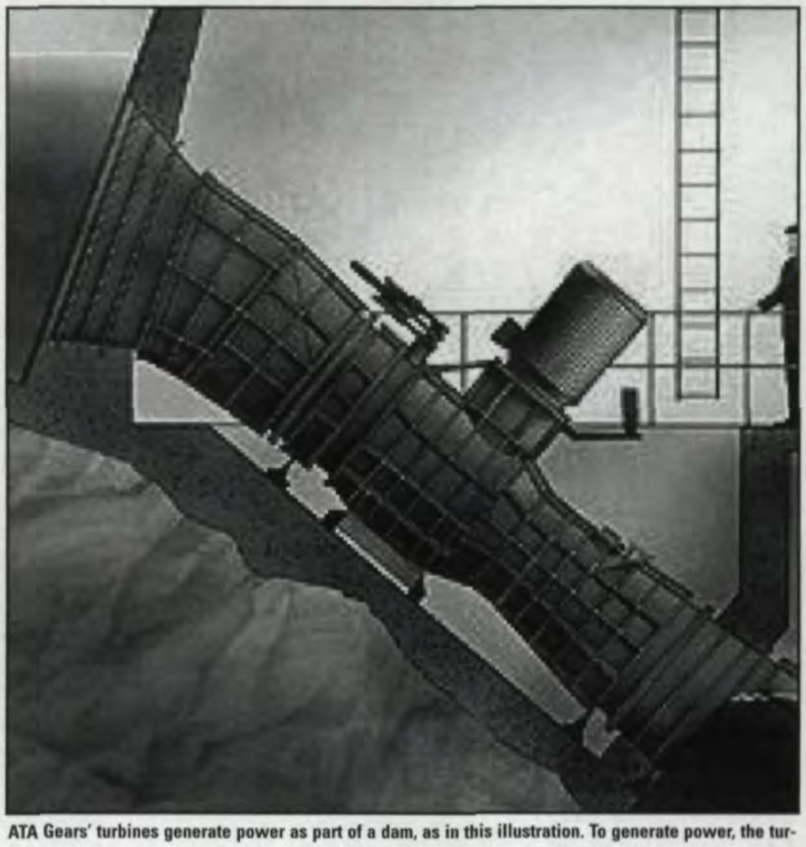


A water turbine from ATA Gears includes a set of its spiral bevel gears, as shown in this illustrated cross section. Entering the turbine, water turns the runner blades (right), driving the shafts and spiral bevel gears that power the generator.



ATA Gears' turbines generate power as part of a dam, as in this illustration. To generate power, the turbines usually need water with a head of 6–70 feet, flowing at a rate of 30–1,800 cubic feet per second.

Gearbox Field Performance From a Rebuilder's Perspective

Charles D. Schultz

This paper was presented at the 1999 Fall Technical Meeting of the American Gear Manufacturers Association.

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Introduction

The major focus of the American Gear Manufacturers Association standards activity has been the accurate determination of a gearbox's ability to transmit a specified amount of power for a given amount of time. The need for a "level playing field" in that critical arena was one of the reasons the association was formed in the first place. Over the past 85 years, AGMA committees have spent countless hours "discussing" the best ways to calculate the rating of a gear set, often arguing vigorously over factors that varied the resulting answers by fractions of a percentage point. While all that "science" was being debated in test labs and conference rooms all over the country, our industry's customers were conducting their own experiments through the daily operation of gear-driven equipment of all types.

Unfortunately, the results of those "test programs" are usually unavailable to the design engineer unless failure occurs during the relatively short warranty period offered on new equipment. My employer, the Pittsburgh Gear Co., has been engaged in the repair, rebuilding and field servicing of gearboxes for many years. The record of that activity provides some interesting insights into what happens to gearboxes long after their warranties run out. (See Appendix A.) While the majority of our customers are in the steel business, equipment from the chemical, mining and rock-quarrying industries has also been repaired. At one time or another, products from most domestic and foreign suppliers have been serviced. The gearboxes cover much of AGMA's history and include most of the designs popular today. One of the drives operated by a local customer was put on-line in 1921 and hasn't missed a day of work yet. That type of performance is exceptional, of course, but it certainly inspires respect for the designers who labored in our trade long before the advent of the computer.

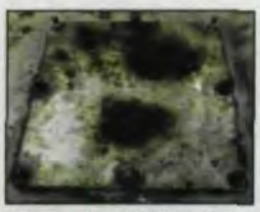

Figure 1—Traditional inspection cover.

Figure 2—Threaded pipe plug inspection port.

It is in deference to those creative engineers that I encourage today's gear designers to avail themselves of any opportunity to study long-term performance. Having spent most of the last 25 years engaged in gearbox design and development, exposure to the far less glamorous side of the business has been extremely beneficial to my design work. In this paper, I'll try to share some of the things I've learned and how they've affected my design philosophy.

Failure Definition

Our customers have their own definition of "gear failure," and it has nothing to do with bending stress or durability rating. The average mill superintendent cares about only one thing: *Can the equipment work today?* If a little pitting or a small crack appears, the user couldn't care less if production can continue. While some of the more sophisticated plants are rapidly moving towards a "predictive maintenance" environment, the vast majority of mills react only to catastrophic breakdowns. We've seen some incredible performances by gearboxes run completely without oil for months or missing sizable tooth fragments due to bearing-related misalignment. We have seen very few "failures" caused by overrating or misapplication, although overloads due to process line "crashes" and field modifications remain a significant problem. AGMA's standards writers and the application engineers can be justifiably proud of their work.

The same pride cannot be shared by the plant maintenance crews, however. The most common causes of failure recorded in our database are lack of lubrication, poor lubricant quality and debris damage. Tooth breakage is rarely seen unless bearing damage, extreme tooth wear or debris is involved. Pitted teeth are usually left untreated until the drive becomes noisy enough to attract attention or someone becomes alarmed at the metal seen during an oil change. While there are certainly design deficiencies that contribute to those problems, most customers would enjoy lower overall operating costs if they did a better job of monitoring the equipment's oil quality. For the designer, knowing that even the most state-of-the-art gearbox will operate in an environment full of soot, sand, water, blunt instruments and mechanics struggling to keep everything moving can help reduce the anxiety of determining the contact stress to the third decimal place.

Design Practice Changes

Observing the maintenance problems first-hand has resulted in some changes to my design philosophy. For example, unless a customer insists, we no longer use the traditional bolt-on inspection covers. (See Figure 1.) We've found the thin plates almost impossible to seal against moisture and have noticed that many times the debris that inflicted the final damage to the gears was a small capscrew used to attach the plate. Our technicians have found an incredible amount of "stuff" inside gearboxes with large covers, including wrenches, screwdrivers, files and flashlights. We've often joked that the bigger the inspection cover, the bigger the debris. While handy for gear inspection, the covers ought to have a warning label affixed reading: "No user serviceable parts in here." We've switched to large threaded pipe plugs for inspection ports. (See Figure 2.) They seal tightly against water, permit quick inspection of tooth condition and are impossible to drop into the oil sump. The pipe wrenches required to remove them are too big to drop though the hole, although the tiny flashlights favored by some mechanics do present a continuing hazard.

We've also become believers in "low oil level" sensors. (See Figure 3.) Originally wary of putting delicate electrical devices in the rough-and-tumble mill environment, observing a plant start-up where the only gearboxes run "dry" were the only ones without the sensors was convincing evidence of their value. We've also been increasing the "robustness" of the external lubrication plumbing used on both new designs and rebuilds. Lube lines apparently make excellent ladders for climbing, serve as emergency crane hooks and are easily snagged by passing loads. One of our customers now has us fitting his more susceptible gearboxes with guards made of 0.25-in. plate to protect them from damage. (See Figure 4.) For most pressure lines, we've switched to high-quality hydraulic hose after discovering that even the bravest mechanic won't use them for a step or a lift point. (See Figure 5.) So far, the customers have been very accepting of that change, especially when they realize it is much cheaper to make a new hose than it is to repair a damaged pipe.

Fast turnaround on repairs is increasingly important to the mills, many of which have greatly reduced both their maintenance staffs and their spares inventories. Among the most distressing problems we have to deal with on a regular basis is the housing bore damaged by a "spun bearing." (See Figure 6a.) Repairing a damaged housing is very time consuming and costly, especially on special units with multiple split lines or a divided power path. The AISE heavy-duty crane specification (see AISE Technical Report 6, June 1996) requires that bearings be mounted in replaceable cartridges (see Figure 6b) due to the difficulty in removing the housings from their lofty perches for re-machining. Superintendents faced with expensive in-place machining of gearbox housings grouted into the floor quickly make sure future purchase specifications follow the AISE's example. Except for bearings that must float axially, it might be worth studying the effects of tighter outside diameter fits on most housings. Where possible, anti-rotation pins (see Figures 7a and 7b) seem to be quite effective.

Some of the lubricant loss experienced in the mill can be traced to worn seal diameters. Many output shafts are made of material that has not been heat treated. Consequently, those shafts have soft seal diameters, which wear out rather quickly. After-market wear sleeves have proven to be quite effective in fixing that problem, although the units have to be removed from their mountings and stripped of their couplings to

Figure 3—Oil level sensor.

Figure 4—Guarded lubrication system.

Figure 5—External lubrication system with hydraulic hose.

Charles D. Schultz

is vice president of engineering at Pittsburgh Gear Co. A registered professional engineer in Pennsylvania and Wisconsin, he has worked in the gear industry for 30 years.

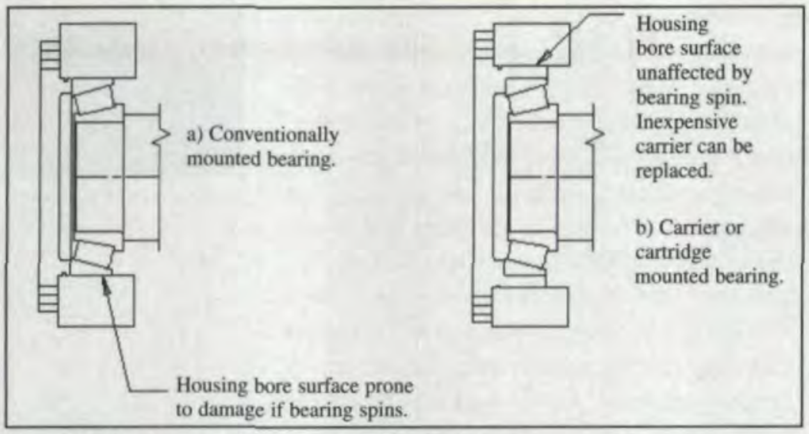

Figure 6—Bearing mounting for extended service life.

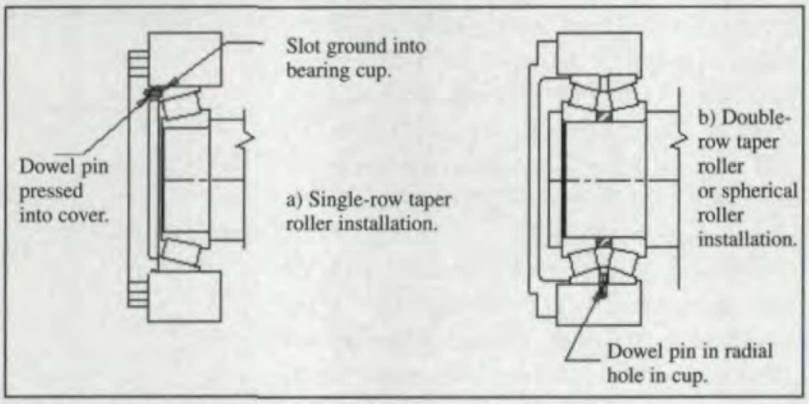

Figure 7—Common anti-rotation pin designs.

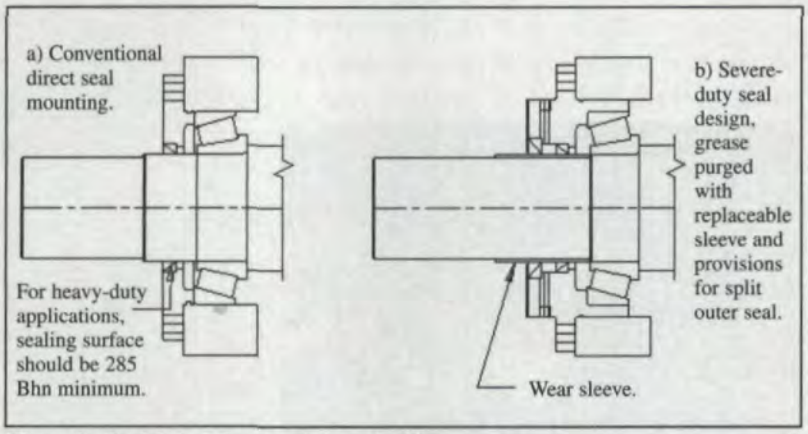

Figure 8—Seal designs for extended service life.

Figure 9—Example of miscellaneous damage.

install them. Coupling removal occasionally causes enough additional damage to the shafts to necessitate their replacement anyway. We recommend that seal diameters be at least 285 Bhn on all shafts unless wear sleeves are fitted. (See Figures 8a and 8b.) Chrome plating or induction hardening is preferred by some customers, although those processes add to the turnaround time. For severe-duty "wash down" environments, special seals, such as packing glands, are well worth the extra expense.

We are still trying to develop better ways to keep water out of the sump. Continuous caster gearboxes are particularly prone to water contamination, even when fitted with expansion chambers rather than breathers. As a result of that problem, the mills have reluctantly gone to strict preventive maintenance programs on those critical drives. Through careful attention to details, such as chrome plating, seal diameters and regularly repainting the interior surfaces of the housing, we have been able to lengthen the time between rebuilds by 50%. The most amazing thing to me is how well the carburized bevel gears have held up after being lubricated with watered-down oil. In similar situations, through-hardened helical gears destroy themselves in a matter of weeks.

Wear is a much more common "gear failure" mode than tooth breakage on the drives that we repair. As noted earlier, customers seldom react to the appearance of pitting, even on carburized gears. Severe wear frequently produces a noticeable change in how the gearbox sounds and draws the interest of the mechanics. Carburized gears typically outlast several sets of bearings, unless the bearings are allowed to deteriorate so far that misalignment results in severe pitting or tooth breakage. In similar applications, through-hardened gears do not seem to hold up as well and are more frequently replaced at the same time as the bearings. The longevity of through-hardened gear sets is adversely affected by the high face-to-diameter ratios used in some designs, which makes the sets prone to deflection-related load distribution problems. In redesigning those gear sets, we often use the additional capacity afforded by changing to carburized gearing to reduce the face width, thereby reducing the face/diameter ratio to more conservative levels.

Basic design problems are relatively rare, but when they occur, the customer has to live with the results for years. One installation we service has a 49:1 ratio, double-reduction gearbox in a very tight location. The 7:1 gear sets have face-to-diameter ratios of over 2.0 and require replacement every 18 to 24 months. We have other "frequent visitors" with multiple input-multiple output configurations that suffer from recirculating power problems. Improved part quality or increased hardness has lengthened the time between rebuilds, but the lack of redesign options prevents effective, long-term resolution. When designing our own "specials," we try to give ourselves room for future upgrades by taking a conservative approach to ratio selection, face-to-diameter proportions and internal housing clearance.

Design for Extended Life

While recognizing the commercial limitations imposed upon us by the global marketplace, I believe that we can "design for extended life" by considering the problems likely to be encountered during the 15- to 25-year operating life of the typical process line and addressing them at the original equipment level. Not every gearbox needs "cartridge bearings," but all deserve a water-free sump. Following the AISE's example of graduated service classes to account for duty cycle, reliability expectations and life requirement would go a long way towards improving customer satisfaction with our industry. Within the context of those service classes, it would be possible to address detail design issues in a way that provides a consistent, level playing field for all competitors without forcing customers to develop their own in-house specifications. I realize the difficulty of changing from a catalog selection system based upon a list of applications and service factors to a system that includes non-gear related factors, such as housing design. It was just a few years ago, however, that we considered 10,000,000 cycles to be "infinite life" despite knowing that that was less than 100 hours of use for the typical high-speed pinion. A proposal for service classes is shown in Appendix B.

Acknowledgments

The author would like to acknowledge the assistance of Will Willman and Ed Besong of Pittsburgh Gear Co. in preparing some of the photographs and illustrations used in this paper.

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Appendix A—Cause of repair summary.

Cause of Repair	Description	# of Orders	% Total
Inadequate Lubricant	Contaminated oil	353	16.1
Low Oil Level	Insufficient oil	449	20.5
Bearings	Worn or failed bearings	211	9.64
Miscellaneous	Non-Gear, Bearing, Lube	198	9.05
Overload	Gear or bearing damage	381	17.4
Preventive Maintenance/ Precautionary Repairs		596	27.2
Total		2,188	

Comments

1. Bearings and seals were changed on all gearboxes unless the customer specifically requested their re-use.
2. Worn or pitted gears were not changed unless authorized by the customer.
3. The "Overload" category includes all tooth breakage and indications of plastic flow. Frequently, the problem occurs when the process line "crashes," putting unforeseen loads on the gears, bearings and shafts.
4. The "Inadequate Lube" category includes units with evidence of contamination and general bearing/gear wear. The "Low Oil Level" category was reserved for cases of relatively sudden failure due to temporary lack of oil rather than long-term damage.
5. The "Miscellaneous" category includes physical damage to the gearbox by external sources, leaks not related to seals, hand-of-assembly changes, coupling changes and modification requests (See Figure 9).
6. The "Preventive Maintenance" repairs are made based upon previous experience that failure to do so on a regular basis will result in unplanned shutdowns.

Appendix B—Design for extended service life classes.

Class	1	2	3	4
Description	Regular Duty	Heavy Duty	Severe Duty	Critical Duty
Abbreviation	RD	HD	SD	CD
Minimum Gear Strength SF	1.25	1.75	2.50	3.00
Lube System	Splash	Splash	Pressure with filter	Pressure with filter
Min. Gear Design Life	10,000 hrs.	15,000 hrs.	20,000 hrs.	30,000 hrs.
Min. Bearing L-10 Life	10,000 hrs.	20,000 hrs.	40,000 hrs.	60,000 hrs.
Bearing Mounting	Conventional	Conventional	Cartridge	Cartridge
Seal Type	Single Lip	Single Lip	Grease purged with provisions for split replacements	Grease purged with provisions for split replacements
Shaft Surface	220 Bhn min.	285 Bhn min.	Replaceable sleeve required	Replaceable sleeve required

Gearboxes repaired included:	Alten Engineering Division, Westerman Companies Amarillo Gear Co. Browning, a division of Emerson Power Transmission Chicago Gear-D.O. James Corp. Cleveland Gear Co. Cone Drive, a division of Textron Power Transmission David Brown, a division of Textron Power Transmission Delroyd Worm Gear Products, a division of Nuttall Gear L.L.C. Dodge, a division of Rockwell Automation Dorris Co. Falk Corp. Fessler Machine Co. Flender Corp. Foote-Jones/Illinois Gear, a division of Regal-Beloit Corp. Hansen Corp., a Minebea Group company Harnischfeger Industries Inc. Hitachi Zosen Corp. Horsburgh & Scott Co. Lohmann + Stolterfoht GmbH Morse, a division of Emerson Power Transmission Nord Gear Nuttall Gear L.L.C. Philadelphia Gear Prager Inc. Renold plc Santasalo Gears (now Metso Drives) SEW Eurodrive Sumitomo Machinery Corporation of America Xtek Inc.


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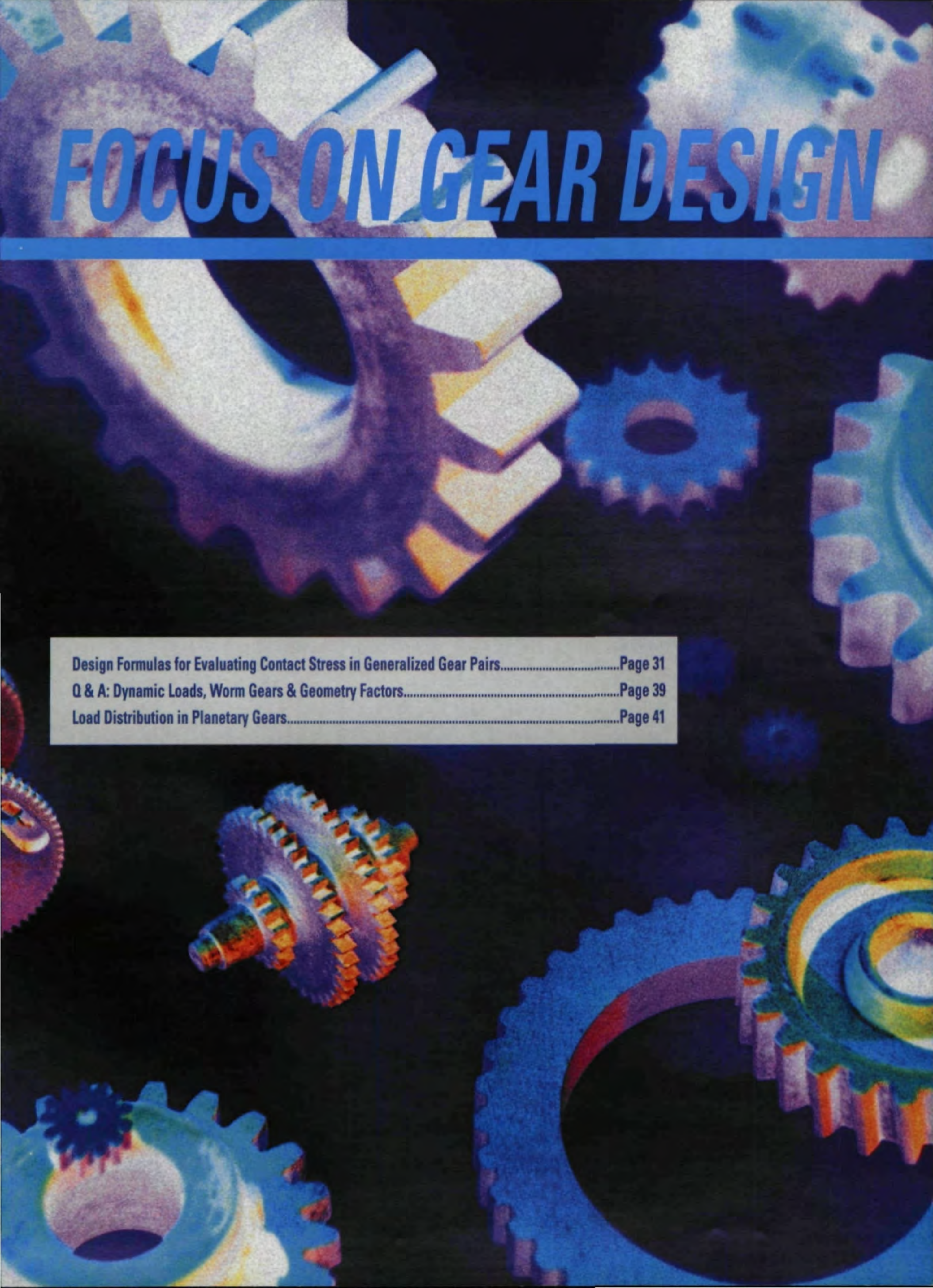

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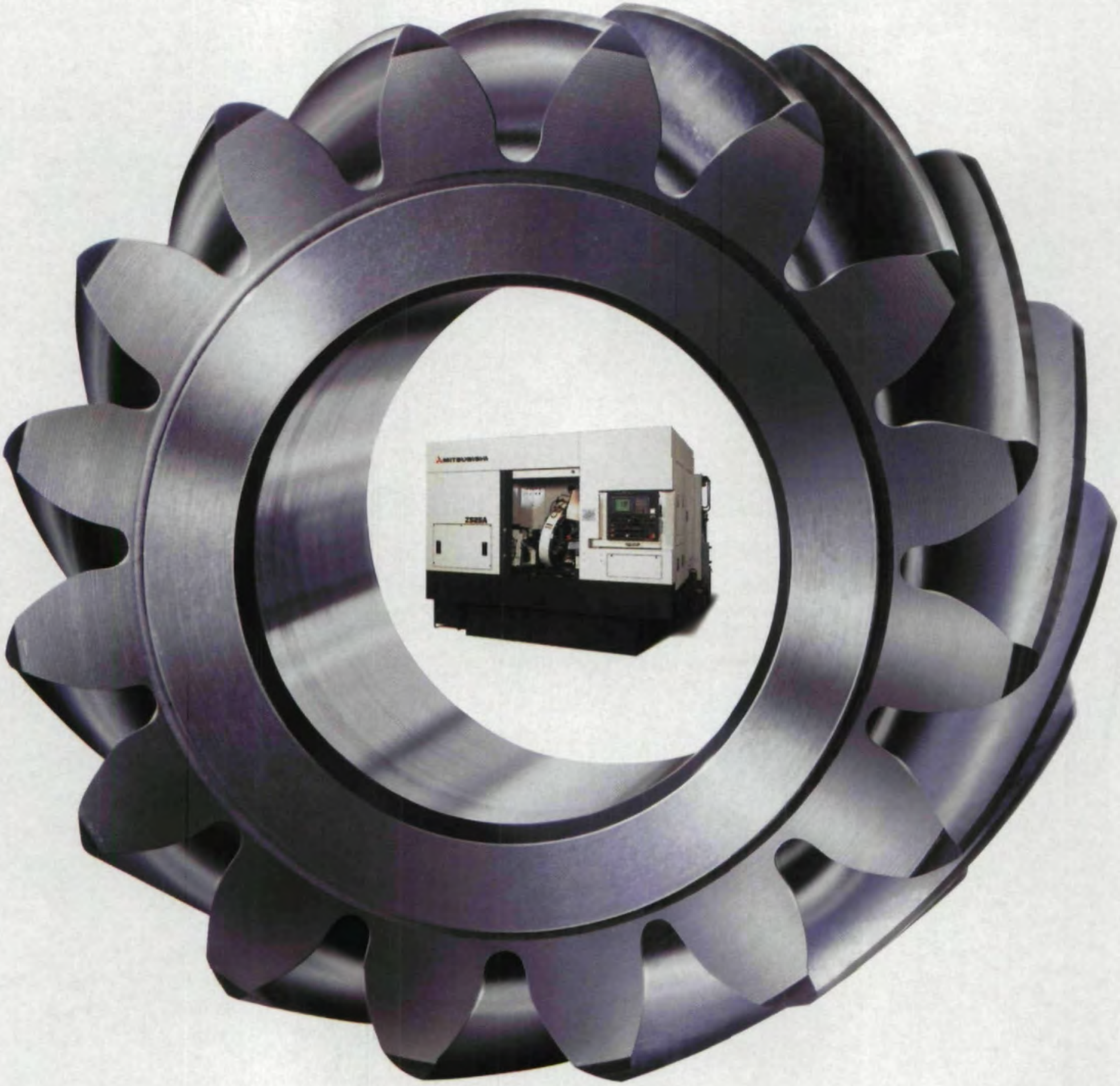

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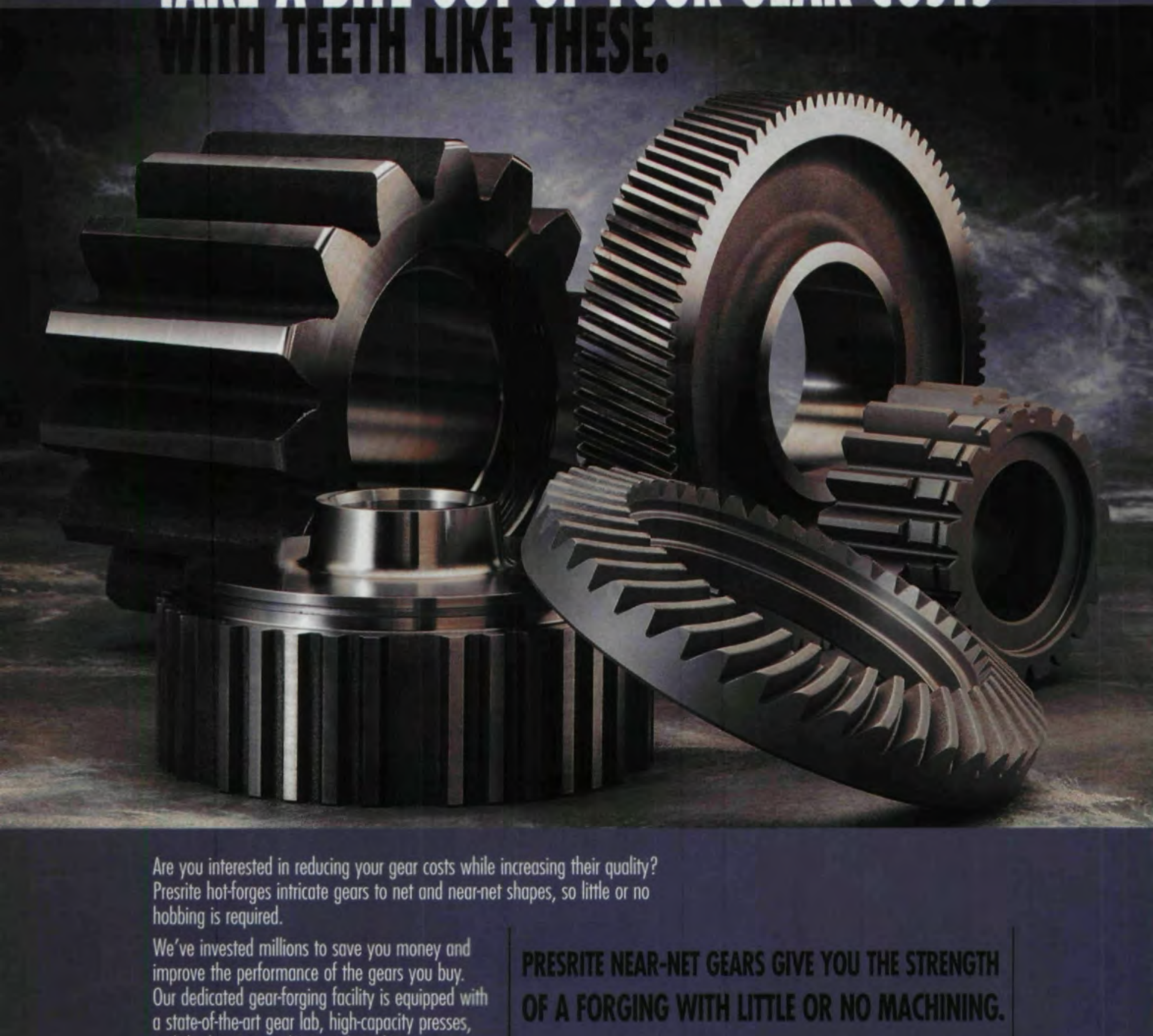

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CIRCLE 108

Design Formulas for Evaluating Contact Stress in Generalized Gear Pairs

David B. Dooner

Introduction

A very important parameter when designing a gear pair is the maximum surface contact stress that exists between two gear teeth in mesh, as it affects surface fatigue (namely, pitting and wear) along with gear mesh losses. A lot of attention has been targeted to the determination of the maximum contact stress between gear teeth in mesh, resulting in many "different" formulas. Moreover, each of those formulas is applicable to a particular class of gears (e.g., hypoid, worm, spiroid, spiral bevel, or cylindrical—spur and helical). More recently, FEM (the finite element method) has been introduced to evaluate the contact stress between gear teeth. Presented below is a single methodology for evaluating the maximum contact stress that exists between gear teeth in mesh. The approach is independent of the gear tooth geometry (involute or cycloid) and valid for any gear type (i.e., hypoid, worm, spiroid, bevel and cylindrical).

Relative Curvature

The contact stress between two gear teeth in mesh depends on the relative gear tooth curvature, material properties of the gear teeth, and the transmitted load between the gear teeth. Determination of the relative gear tooth curvature can be problematic for certain gear types. The relative gear tooth curvature between two gear teeth in mesh results in a contact that is either point-contact or line-contact. In general, the transverse contact ratio for two gears in mesh is greater than zero, and line-contact exists between the two gear teeth in mesh. Helical or spiral gears with line-contact experience both axial and transverse displacement during mesh. Such conditions are inherent for any tooth profile type (namely, involute or cycloid). Point-contact is the alternative scenario for two conjugate surfaces in mesh. That condition occurs when the transverse contact ratio is zero. That type of contact applies to circular-arc type profiles (namely, Novikov-Wildhaber or BBC).

Determination of the relative gear tooth curvature $\Delta\kappa$ between two planar involute gear teeth is demonstrated prior to presenting the relative gear tooth curvature between two generalized gear teeth. Depicted in Figure 1 are two involute gear teeth in mesh. The radius of the input pitch circle is R_i, whereas R_o is the radius of the output pitch circle. ρ_i and ρ_o are the radii of curvature for the input and output gear teeth respectively. Projecting the pitch radii R_i and R_o onto the contact normal

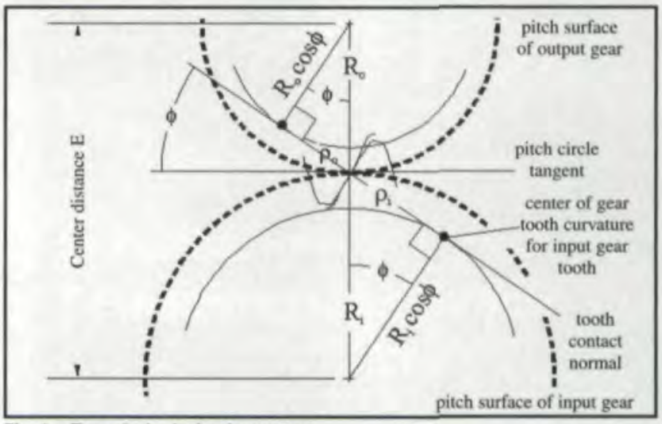

Fig. 1—Two pitch circles in contact.

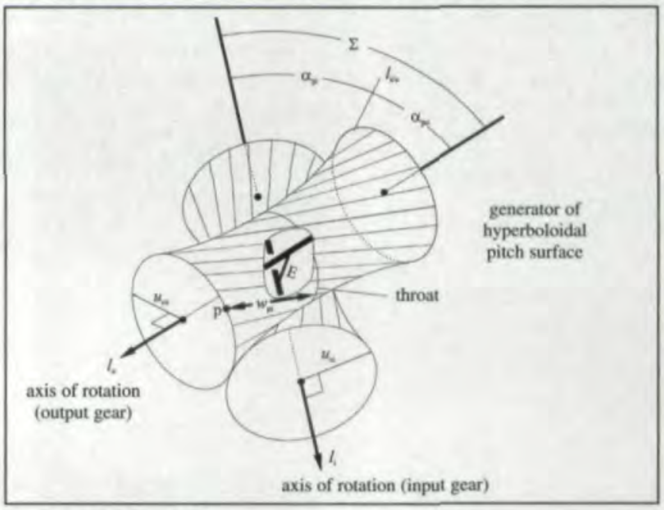

Fig. 2—Gear nomenclature.

David B. Dooner

is an associate professor in the department of mechanical engineering at the University of Puerto Rico-Mayagüez. His research involves a universal approach for the concurrent design and manufacture of gears. That generalized approach can be used to produce gears with any number of teeth, any face width and any spiral angle. The approach applies to helical, bevel, worm, spiroid, hypoid and non-circular gears.

yields

$$\rho_i = R_i \sin\phi \tag{1a}$$

$$\rho_o = R_o \sin\phi \tag{1b}$$

where ϕ is the angle between the pitch circle tangency and the tooth contact normal. For planar curves, the curvature κ and radius of curvature ρ are reciprocals (i.e., $\kappa = 1/\rho$). Thus, relative gear tooth curvature $\Delta\kappa$ can be expressed as follows:

$$\Delta\kappa = \left(\frac{1}{\rho_i} + \frac{1}{\rho_o}\right) \tag{2a}$$

or

$$\Delta\kappa = \left(\frac{1}{R_i} + \frac{1}{R_o}\right)\frac{1}{\sin\phi} \tag{2b}$$

where

R_i radius of input pitch circle
R_o radius of output pitch circle
ϕ pressure angle.

The above expression establishes a unique relation between the pressure angle ϕ, the pitch radii R_i and R_o and the relative gear tooth curvature $\Delta\kappa$. Regardless of the radii of tooth curvature ρ_i and ρ_o, the relative gear tooth curvature $\Delta\kappa$ depends solely on pitch radii R_i and R_o and pressure angle ϕ. The above relation for relative gear tooth curvature is for cylindrical gears with spur-type gear teeth. Furthermore, the relation is valid only for contact at the pitch point.

Prior to presenting a generalized relation for the relative gear tooth curvature, it is necessary to establish some nomenclature and introduce certain expressions. Depicted in Figure 2 are two pitch surfaces in mesh, two axes of rotation l_i and l_o, the perpendicular distance E between the two axes l_i and l_o, and the included angle Σ between the two axes l_i and l_o. The pitch surfaces in Figure 2 are hyperboloids. Notice in Figure 2 that each hyperboloidal pitch surface is determined by a series of straight lines. Hyperboloids can be envisioned as the surface produced by rotating a line or generator about a central axis. For example, rotating the common generator $l_{i/o}$ between the input and output body about the axis of rotation l_i produces the input hyperboloidal pitch surface. The shape of the hyperboloidal pitch surface depends on an angle α and distance u. The angle α is the cone angle of the generator, and u is the radius of the hyperboloidal pitch surface at the throat. Introducing u_{pi} as the radius of the input hyperboloidal pitch surface and u_{po} as the radius of the output hyperboloidal pitch surface, then $u_{pi} + u_{po} = E$ for two hyperboloidal surfaces in mesh. Similarly, defining α_{pi} as the cone angle for the input hyperboloidal pitch surface and α_{po} as the cone angle for the output hyperboloidal pitch surface, then $\alpha_{pi} + \alpha_{po} = \Sigma$ for two hyperboloidal pitch surfaces in mesh. Also shown in Figure 2 is the distance w_{pi} between the throat and point p. As the two hyperboloids rotate, they are always in contact along the common generator.

Cylindrical gearing occurs when the angle Σ between the input axis of rotation and the output axis of rotation is zero (i.e.,

$\Sigma = 0$ and hence the cone angles α_{pi} and α_{po} are also zero). In general, a gear type depends on both the center distance E (offset) and angle Σ between the input and output axes of rotation. When the distance E between two axes of rotation is zero, then the pitch surfaces become cones and the throat radii u_{pi} and u_{po} are zero. Alternately, when neither E nor Σ is zero, then the two pitch surfaces are hyperboloids. Equation 2b for relative curvature $\Delta\kappa$ was derived in terms of cylindrical pitch surfaces, and consequently it is not valid for conical or hyperboloidal pitch surfaces.

An important parameter for specifying relative gear tooth curvature is the effective radius. The effective radius is the distance between the point p on the pitch surface and the axis of rotation, as shown in Figure 2. For generalized gear pairs with a constant input/output gear ratio g, the effective radii u_{ei} and u_{eo} can be expressed

$$u_{ei} = \sqrt{u_{pi}^2 + w_{pi}^2 \sin^2\alpha_{pi}} \qquad (3a)$$

$$u_{eo} = \sqrt{u_{po}^2 + w_{po}^2 \sin^2\alpha_{po}} \qquad (3b)$$

where

- u_{pi} radius of the input pitch surface (at the throat)
- u_{po} radius of the output pitch surface (at the throat)
- E shaft center distance between the two axes of rotation ($u_{pi} + u_{po} = E$)
- w_{pi} axial position of tangent point on input pitch surface
- w_{po} axial position of tangent point on output pitch surface ($w_{pi} = -w_{po}$)
- α_{pi} cone angle of input pitch surface
- α_{po} cone angle of output pitch surface
- Σ the included shaft angle between the two axes of rotation ($\alpha_{pi} + \alpha_{po} = \Sigma$).

It is also convenient to introduce the following relations

$$R_i = u_{ei}\cos(\gamma_{pi} + \psi_{pi}) \qquad (4a)$$

$$R_o = u_{eo}\cos(\gamma_{po} + \psi_{po}) \qquad (4b)$$

where

$$\tan\gamma_{pi} = \frac{u_{pi}\sin\alpha_{pi}}{\sqrt{u_{pi}^2\cos^2\alpha_{pi} + w_{pi}^2\sin^2\alpha_{pi}}} \qquad (5a)$$

$$\tan\gamma_{po} = \frac{u_{po}\sin\alpha_{po}}{\sqrt{u_{po}^2\cos^2\alpha_{po} + w_{po}^2\sin^2\alpha_{po}}} \qquad (5b)$$

to determine relative gear tooth curvature. It is of central importance to know that the gear ratio g is equal to the ratio of radii R_i and R_o (i.e., $g = R_i/R_o$). Cone angles α_{pi} and α_{po} are zero for cylindrical gears, and consequently γ_{pi} and γ_{po} are also zero. For bevel gears, the pitch radii u_{pi} and u_{po} are zero such that γ_{pi} and γ_{po} reduce to zero. For spur gears, ψ is zero.

In general, the extreme relative curvature between two gear teeth in mesh can be determined with the following expressions:

$$\Delta\kappa_{min} = 0 \qquad (6a)$$

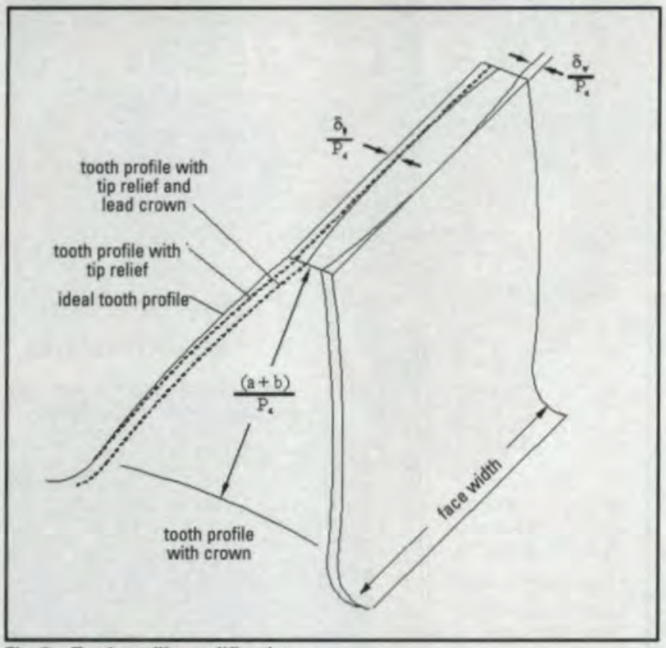

Fig. 3—Tooth profile modification.

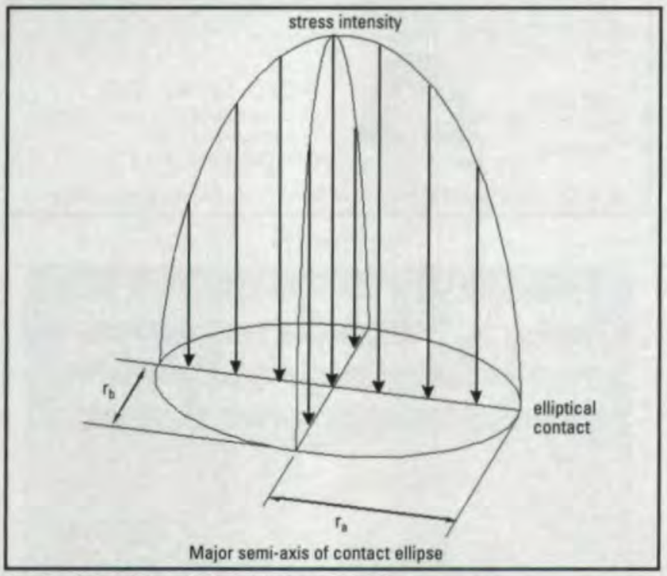

Fig. 4—Stress intensity for elliptical contact.

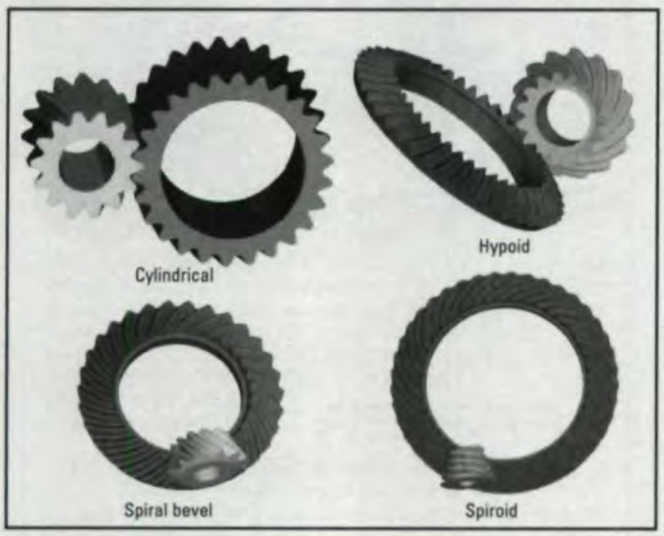

Fig. 5—Gear pairs.

$$\Delta\kappa_{max} = \sqrt{\left[\frac{1}{R_i^2} + \frac{2\cos\Sigma}{R_i R_o} + \frac{1}{R_o^2}\right](\cos^2\psi_{pi} + \tan^2\phi_n)\frac{\cos^2\phi_n}{\sin\phi_n}} \quad (6b)$$

where

$\Delta\kappa_{min}$ minimum relative curvature between gear teeth
$\Delta\kappa_{max}$ maximum relative curvature between gear teeth
R_i virtual radius of the input hyperboloidal pitch surface
R_o virtual radius of the output hyperboloidal pitch surface
ψ_{pi} instantaneous spiral angle of the input gear ($\psi_{pi} = -\psi_{po}$)
ϕ_n normal pressure angle
Σ included shaft angle.

The above formulas for extreme relative gear tooth curvature are applicable for hypoid, spiroid, worm, bevel and cylindrical gear pairs. Furthermore, those formulas are independent of the type of gear tooth profile. Recognize that when $\Sigma = 0$ (i.e., planar gearing) and $\psi = 0$ (i.e., spur gears), Equation 2b and the above relation for maximum relative gear tooth curvature, $\Delta\kappa_{max}$, are identical. The mathematical development of the above expressions involves many mathematical relations, and only the results are presented. Additional insight into the mathematical derivations is provided in Reference 1.

Profile Modification

Ideally, two gear teeth in mesh are in line-contact for gear pairs with involute-type tooth profiles. However, gear designers introduce both profile relief and lead crown to accommodate errors in tooth spacing, runout, misalignment and deflections. Gear teeth with profile or tip relief have a reduction in tooth thickness in a particular transverse plane. The magnitude of the tip relief is usually restricted to micrometers (μm) or a few thousandths of an inch. Crowned teeth have a reduction in tooth thickness in the lengthwise direction of the gear tooth. The magnitude of crowning is restricted to a few micrometers across the tooth face. Depicted in Figure 3 is a tooth profile with tip relief and lead crown. Such profile modification reduces theoretical line-contact to point-contact. Consequently, the above relations for extreme relative gear tooth curvature (i.e., $\Delta\kappa_{max}$ and $\Delta\kappa_{min}$) must be modified to account for crown and profile relief.

There is no established standard for specifying tooth profile modification. Here, the deviation in ideal tooth profile is quadratic. Tip relief and lead crown are specified here in a manner analogous to the specification of addendum and dedendum. That is achieved by introducing a tip relief constant δ_ϕ and a lead crown constant δ_ψ. Given the following tip relief constant δ_ϕ and lead crown constant δ_ψ, the changes in curvature $\delta\kappa_\psi$ and $\delta\kappa_\phi$ are

$$\delta\kappa_\psi = \frac{8}{F_i^2 + F_o^2}\,\frac{\delta_\psi}{P_d} \quad (7a)$$

$$\delta\kappa_\phi = \left(\frac{2P_d}{a + b}\right)^2 \frac{\delta_\phi}{P_d} \quad (7b)$$

where

δ_ψ lead crown constant
δ_ϕ tip relief constant
a addendum constant
b dedendum constant

F_i, F_o face width

P_d normal diametral pitch.

δ_ψ and δ_ϕ are the same for both the input and output gear elements such that the modified curvatures become

$$\delta\kappa_{min} = \frac{\delta\kappa_\psi + \delta\kappa_\phi\,(\sin^2\phi_n \tan^2\psi_{pi})}{1 + \sin^2\phi_n \tan^2\psi_{pi}} \tag{8a}$$

$$\delta\kappa_{max} = \frac{\delta\kappa_\phi + \delta\kappa_\psi\,(\sin^2\phi_n \tan^2\psi_{pi})}{1 + \sin^2\phi_n \tan^2\psi_{pi}} \tag{8b}$$

where

$\delta\kappa_{min}$ change in minimum relative tooth curvature

$\delta\kappa_{max}$ change in maximum relative tooth curvature

$\delta\kappa_\phi$ change in relative tooth curvature in profile direction

$\delta\kappa_\psi$ change in relative tooth curvature in lengthwise direction

ϕ_n normal pressure angle

ψ_{pi} spiral angle.

For spur gears (i.e., $\psi = 0$), the face widths F_i and F_o are identical and equal to the distance between the heel and toe. The above change in relative gear tooth curvature for modified gear teeth must be added to the theoretical value. Thus, extreme gear tooth curvature can be expressed

$$\underline{\kappa}_{min} = \delta\kappa_{min} \tag{9a}$$

$$\underline{\kappa}_{max} = \Delta\kappa_{max} + \delta\kappa_{max}. \tag{9b}$$

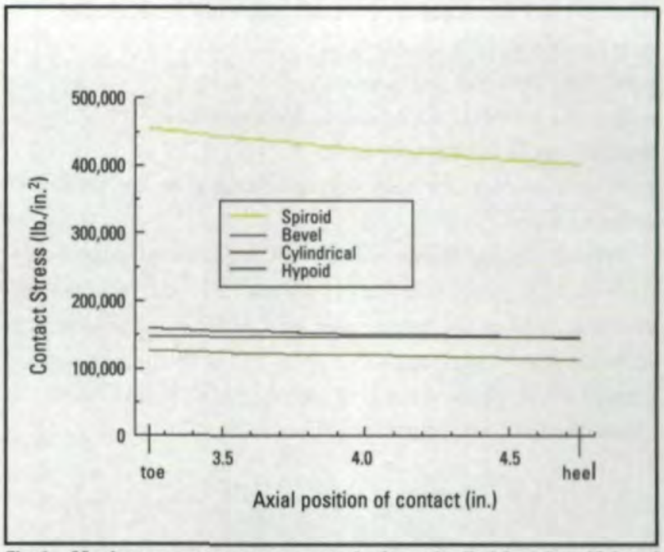

Fig. 6—Maximum contact stress across the face of cylindrical, hypoid, bevel and spiroid gear pairs.

Contact Stress

The transmitted load between two gear teeth is non-uniformly distributed over the surface area of contact. Depicted in Figure 4 is an elliptical contact area with semi-axes r_a and r_b along with a parabolic stress intensity. The sum of the pressure distribution over the area of contact results in the net force applied to the gear mesh interface. Determination of the maxi-

mum contact stress is based on the following assumptions:
gear tooth materials are elastic,
gear tooth materials are isotropic,
gear tooth material properties are homogeneous,
contact area is frictionless, and
radii of curvature are very big compared with the semi-axes contact ellipse.

The maximum compressive stress is evaluated using a mattress-based formula (see Ref 2). Hence, it is advisable that the elastic moduli of the foundations are similar in magnitude. It is recommended that Hertz's formulas for predicting maximum contact stress are used for gear elements with highly dissimilar elastic foundations. Introducing the constant

$$C = P\left(\frac{4\sqrt{6}}{3\pi^3}(5\pi-4)\right)\left(\frac{1-\mu_i^2}{E_i}+\frac{1-\mu_o^2}{E_o}\right) \quad (10)$$

where

P	normal contact force
E_i	modulus of elasticity for input gear
E_o	modulus of elasticity for output gear
μ_i	Poisson's ratio for input gear
μ_o	Poisson's ratio for output gear,

the semi-axes of the contact ellipse become

$$r_a = \left[C\left(\frac{\underline{K}_{max}}{\underline{K}_{min}}\right)^{1/4}\frac{1}{\underline{K}_{min}}\right]^{1/3} \quad (11a)$$

$$r_b = \left[C\left(\frac{\underline{K}_{max}}{\underline{K}_{min}}\right)^{1/4}\frac{1}{\underline{K}_{min}}\right]^{1/3} \quad (11b)$$

where

r_a	major semi-axis of contact ellipse
r_b	minor semi-axis of contact ellipse
C	constant
$\underline{K}_{max}$	maximum relative curvature with profile modification
$\underline{K}_{min}$	minimum relative curvature with profile modification.

The maximum contact stress σ_c is obtained by integrating the pressure distribution over the area of contact A_c and equating to the transmitted load P. The maximum stress σ_c is $3\pi/4$ times the average stress for an elliptical contact area and a proportional stress intensity, hence

$$\sigma_c = \frac{3\pi}{4}\,\frac{P}{\pi r_a r_b} \quad (12)$$

That formula neglects common loading factors that instantaneously increase the transmitted load P. One such loading factor that results in an instantaneous increase in transmitted load is the dynamic load that results from transmission error. A second loading factor that increases the transmitted load is a distribution factor. Shaft misalignment can result in a concentrated load for gears with high contact ratios. A third type of loading factor that gives an instantaneous increase in transmitted load is an application factor. Such factors are inherent in "rough" operating machinery, like internal combustion engines and crushing mechanisms. Additional insight into those factors is provided by AGMA. Contact stress is further affected by tractive or shear forces that result from the relative sliding and friction at the mesh and residual stresses in the tooth sub-surface. The magnitude of the shear load on the gear surface depends on the type of lubricant and its thickness. Simultaneously, the gear designer should be aware that relative gear tooth sliding at the contact

Table 1—Gear Pair Parameters.

	Cylindrical	Hypoid	Bevel	Spiroid
Shaft center distance E (in./mm)	2.00 / 50.8	2.00 / 50.8	0.00 / 0.00	2.00 / 50.8
Included shaft angle Σ (deg./rad.)	0.0 / 0.0	90 / 1.571	90 / 1.571	90 / 1.571
Axial position of toe w_{toe} (in./mm)	3.25 / 82.55	3.25 / 82.55	3.25 / 82.55	3.25 / 82.55
Axial position of heel w_{heel} (in./mm)	4.75 / 120.65	4.75 / 120.65	4.75 / 120.65	4.75 / 120.65
Number of teeth on input gear N (integer)	14	15	12	2
Number of teeth on output gear N (integer)	26	41	31	41
Nominal spiral angle ψ_p (deg./rad.)	17.42 / 0.304	35.54 / 0.620	37.13 / 0.648	81.21 / 1.417
Nominal pressure angle (normal) ϕ_n (deg./rad.)	30.00 / 0.524	55.94 / 0.976	65.07 / 1.136	47.80 / 0.834
Input shaft torque T ($\frac{lb\text{-}in}{N\text{-}m}$)	900 / 101.7	900 / 101.7	900 / 101.7	120 / 14.36
Axial contact ratio m_a (dimensionless)	1.5	1.5	1.5	4
Addendum constant (dimensionless)	1	1	1	1
Dedendum constant (dimensionless)	1	1	1	1
Tip relief constant δ_ϕ (dimensionless)	0.01	0.01	0.01	0.01
Lead crown constant δ_ψ (dimensionless)	0.02	0.02	0.02	0.02

Table 2—Contact Stress Calculations.

	Cylindrical	Hypoid	Bevel	Spiroid
$E_i = E_o$ (lb./in.^{2}/N/m^2)	30×10^6 / 207×10^9	30×10^6 / 207×10^9	30×10^6 / 207×10^9	30×10^6 / 207×10^9
$\mu_i = \mu_o$ (dimensionless)	0.287	0.287	0.287	0.287
R_i (in./mm)	0.6672 / 16.95	1.388 / 35.25	1.151 / 29.24	0.1180 / 3.00
R_o (in./mm)	1.239 / 31.47	3.793 / 96.34	2.974 / 75.54	2.418 / 61.42
Σ (deg./rad.)	0 / 0	90 / 1.571	90 / 1.571	90 / 1.571
P_d (in./mm)	10.49	5.405	5.212	8.477
α_{pi} (deg./rad.)	0 / 0	26.74 / 0.467	21.16 / 0.369	13.4 / 0.234
α_{po} (deg./rad.)	0 / 0	63.26 / 1.104	68.84 / 1.201	76.6 / 1.337
γ_i (deg./rad.)	0 / 0	5.666 / 0.099	0 / 0	1.53 / 0.027
γ_o (deg./rad.)	0 / 0	21.35 / 0.373	0 / 0	25.18 / 0.439
$\delta\kappa_{min}$ (1/in./1/m)	0.0063 / 0.248	0.0225 / 0.886	0.025 / 0.984	0.1061 / 4.177
$\delta\kappa_{max}$ (1/in./1/m)	0.1334 / 5.252	0.0538 / 2.118	0.0477 / 1.878	0.005 / 0.189
$\underline{K}_{min}$ (1/in./1/m)	0.0063 / 0.248	0.0225 / 0.886	0.0250 / 0.984	0.1061 / 4.177
$\underline{K}_{max}$ (1/in./1/m)	4.425 / 174.21	0.8818 / 34.72	1.008 / 39.69	6.413 / 252.5
P (lb./N)	1,552 / 6,903	1,158 / 5,151	1,855 / 8,247	1,514 / 6,734
C (in.^{2}/mm^2)	1.171×10^{-4} / 0.076	0.874×10^{-4} / 0.056	1.399×10^{-4} / 0.0903	1.143×10^{-4} / 0.074
r_a (in./mm)	0.4570 / 11.607	0.2133 / 5.418	0.2418 / 6.142	0.1443 / 3.665
r_b (in./mm)	0.0173 / 0.439	0.0341 / 0.866	0.0380 / 0.965	0.0186 / 0.472
σ_c (ksi/MPa)	148 / 1,018	119 / 823	151 / 1,043	424 / 2,925

zone can cause a rise in temperature at the mesh, resulting in a temperature gradient in the gear tooth and thus further affecting localized tooth contact stress.

Examples

Four examples are presented to illustrate the determination of contact stress between gear teeth in mesh. The first example is a helical cylindrical gear pair, the second example is a hypoid gear pair with non-zero spiral angle, the third example is a spiral bevel gear pair, and the last example is a spiroid gear pair (i.e., a hypoid gear pair with high spiral angle). Each gear pair has a 1.5-inch face width. The nominal gear parameters for each gear pair are provided in Table 1. Graphical illustrations of the gear pairs are provided in Figure 5. Intermediate calculations and final contact stress are presented in Table 2 for the face midpoint. Values for maximum contact stress are based on a single concentrated load and neglect load sharing resulting from high contact ratios, tooth deflections or wheel body deflections. A computer program has been written, and the variation in contact stress across the face of the gear pairs is depicted in Figure 6.

Summary

Simplified design formulas for evaluating the maximum contact stress between two gears in mesh are presented. The methodology is summarized as follows:

- demonstrated that relative tooth curvature for planar gears depends on pitch radii and pressure angle,
- presented a generalized formula for extreme relative curvature between gear teeth in mesh that is valid for any tooth type (involute or cycloid) and any gear type (cylindrical, bevel, hypoid, spiroid or worm),
- presented a generalized formula for relative gear tooth curvature for arbitrary tooth profile modification (tip relief and lead crown),
- presented explicit expressions for semi-axes of elliptical contact based on mattress formula,
- presented formula for maximum contact stress between gear teeth, and
- presented four examples to illustrate the use of formulas to determine maximum contact stress.

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2. Dooner, D.B. "On the Generalized Contact Stress," *The International Conference on Mechanical Transmissions*, Chongqing, P.R. China, April 5-10, 2001.

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CIRCLE 130

CIRCLE 123

Calculating Dynamic Loads, Sizing Worm Gears and Figuring Geometry Factors

Robert Errichello

Answers submitted by Robert Errichello, Gear Consultant, GEARTECH, Townsend, MT, and Technical Editor, Gear Technology

Questions submitted by Mark H. Swalley, U.S. Navy, Puget Sound Naval Shipyard, Bremerton, WA

Q: What is the best method to measure or calculate dynamic loads in a low-speed spur gear subject to load reversals?

A crane's gear train is subject to sudden load reversal when an emergency brake suddenly applies on the hoist drum. As the drum comes to a sudden stop, thanks to large caliper brakes, the electric-powered drive train continues to rotate. The backlash is taken up, and the pinion gear contacts the opposite face of the drum's bull gear. The pinion gear is on the output shaft of the speed reducer. The impact is so violent, the pinion gear stand bolts snap the head off, ricocheting around the machinery house.

The gear train has about 10 times the inertia of the hoist drum. To complicate matters, the electric motor has time to produce regenerative braking prior to impact.

A: Strain gages are commonly used to measure dynamic loads. They are a good choice because they are readily available, relatively inexpensive, and accurate for capturing transient loads such as you describe. Computer modeling, on the other hand, is accurate only if mass, stiffness and damping are precisely known. Damping is especially difficult to estimate and usually requires strain measurement on the actual system to determine accurate values. Therefore, strain measurement is the best choice for determining dynamic loads in your application. Additionally, the measurements can help determine system damping, which you could use in a computer model to explore alternative design changes to reduce dynamic loads.

Q: How should the strain gages be placed, and how should the signal be transmitted?

I am currently working out a full Wheatstone bridge arrangement. Strain gages will be placed on the ends of the 19 gear teeth, one on either side of the gear tooth base (the high tension/compression areas). The 38 gages and two dead gages will be divided into four legs of 10 gages in series. The signal will be transmitted from the rotating shaft via one of the telemetry-type strain transmitters.

That arrangement will reduce the signal about five times over the single gage in a 1/4 bridge circuit, but will transmit a constant signal. The transmitter sends 500 signals a second, and I believe that to be fast enough to catch the dynamic event. I can correlate the results to strain gages reading from a steady state load.

A: Your transducer and method of transmitting signals are state-of-the-art. To capture tooth bending stresses, the gages should be placed on the root fillet at the point of maximum bending stress. That point can be found by painting the root fillets with brittle lacquer and applying loads to the gears. The first place to crack the brittle lacquer is the point of maximum stress. However, it is seldom practical to mount gages at the critical point because of interference with the mating gear's tips. You need to decide what it is you want. Is it maximum tooth bending stress, or is it the maximum dynamic torque applied to the gears? It may be sufficient to measure the torque and calculate tooth stresses.

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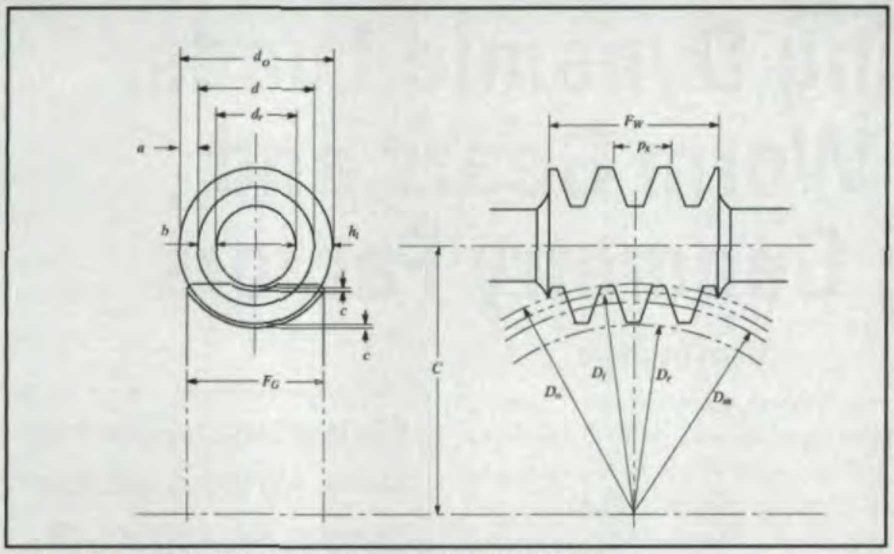

Fig. 1 — Worm gear basic dimensions. Source: ANSI/AGMA 6022-C93.

Table 1—Definition of symbols.

a	Worm and worm gear addendum
b	Worm and worm gear dedendum
C	Center distance
c	Clearance
D_m	Worm gear mean diameter
D_o	Worm gear outside diameter
D_r	Worm gear root diameter
D_t	Worm gear throat diameter
d	Worm pitch diameter
d_o	Worm outside diameter
d_r	Worm root diameter
F_G	Worm gear face width
F_W	Worm face width (Thread length of worm)
h_t	Full depth of worm thread
p_x	Worm axial pitch

Source: ANSI/AGMA 6022-C93.

Question submitted by Christian Williams, Texas A&M University, College Station, TX

Q: How does the desired gear ratio for a worm and worm gear relate to the actual size of the gears? For example, if I want a gear ratio of 30:1, what diameter constraints are placed on the gears for that ratio?

A: Worm wheel diameter is set by required load capacity. ANSI/AGMA 6034-B92, "Practice for Enclosed Cylindrical Wormgear Speed Reducers and Gearmotors," relates worm wheel diameter to tangential load as follows:

$$Dm = [Wt/(Cs\ Cm\ Cv\ Fe)]^{1.25}$$

Where:

Dm = mean diameter of the worm wheel
Cs = materials factor (accounts for the worm wheel material's strength)
Cm = ratio correction factor (accounts for the gear ratio)
Cv = velocity factor (accounts for sliding velocity and varies with diameter, lead and speed of the worm)
Fe = effective face width of the worm wheel ($Fe = F_G$ or $Fe = (2/3)d$, whichever is less.)
Wt = tangential load

The diameter of the worm can vary somewhat without significantly influencing load capacity. ANSI/AGMA 6022-C93, "Design Manual for Cylindrical Wormgearing," recommends a worm diameter in the following range:

$$\frac{C^{0.875}}{3.0} \le d \le \frac{C^{0.875}}{1.6}$$

Where:

C = center distance
d = worm pitch diameter

ANSI/AGMA 6022-C93 provides guidelines that give a practical overview of worm gear design.

You should audit the gearset to ensure it meets the design guidelines of ANSI/AGMA 6022-C93 and rate the load capacity in accordance with ANSI/AGMA 6034-B92 to ensure it has an adequate service factor for the application.

Question submitted by Richard Friedman, Nichols Aircraft Division, Parker Hannifin Corp., Ayer, MA

Q: Is there a closed-form solution for the Lewis form factor?

A: No, there is no known closed-form solution. However, an efficient algorithm for numerically solving the problem is given in AGMA 908-B89 (Ref. 1). The algorithm is readily programmed using the flow chart given in AGMA 918-A93 (Ref. 2). Benchmark examples useful for validating software are given in Reference 2 and tables of J factors are given in Reference 1. The algorithm derives from References 3 and 4.

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Load Distribution in Planetary Gears

Wolfgang Predki, Friedrich Jarchow and Christof Lamparski

This paper was presented at the 4th World Congress on Gearing and Power Transmission in Paris, March 1999.

Introduction

Two-shaft planetary gear drives are power-branching transmissions, which lead the power from input to output shaft on several parallel ways. A part of the power is transferred loss-free as clutch power. That results in high efficiency and high power density. Those advantages can be used optimally only if an even distribution of load on the individual branches of power is ensured. Static overconstraint, manufacturing deviations and the internal dynamics of those transmission gears obstruct the load balance. With the help of complex simulation programs, it is possible today to predict the dynamic behavior of such gears. The results of those investigations consolidate the approximation equations for the calculation of the load factors $K_\gamma K_v$ and $K_{H\beta}$.

Parameter Study on Load Factors

The calculation of the operational dynamic behavior of a planetary gear stage is done with the 3-dimensional simulation program SIMPLEX, which was developed in an earlier research project (Ref. 2). The program is able to describe all masses within the transmission with six degrees of freedom each and considers the influence of the mentioned disturbances on the masses' dynamics. The dynamic model underlying the program is shown in Figure 1.

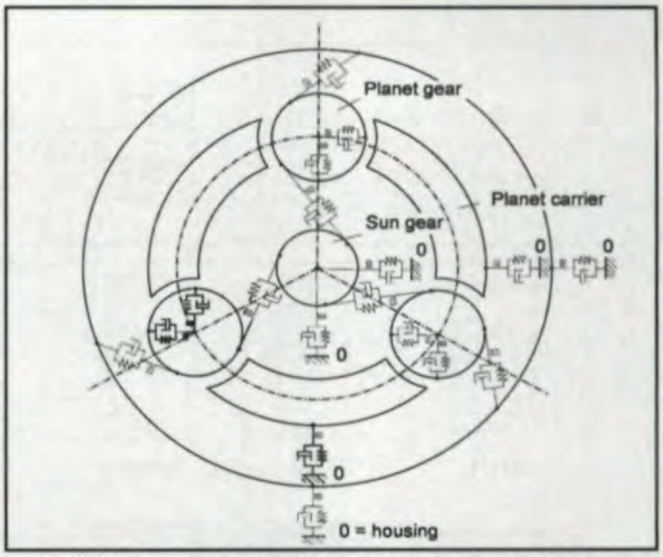

Fig. 1—Dynamic model of a planetary gear stage in transverse section.

Afterwards, the masses are linked to the bearings and to the foundation by discrete, elastic shock-absorbing elements. Table 1 quotes the parameters, which were considered on the dynamic simulation.

Table 1—Parameters considered on the dynamic simulation.

Environmental conditions: Attached mass inertia Stiffness of attachment Line load Speed of sun wheel Relative circumferential velocity	**Size and geometry of the gearing:** Center distance Gear ratio Number of planet gears Sum of the addendum modification Face width/diameter coefficients Number of teeth of sun Base helix angle
Tooth correction: Crowning Tip modification Long addendum teeth	**Gear tooth quality:** QF_p, Qf_p, $Qf_{H\beta}$ Pitch tolerance of the planet carrier Bearing eccentricity Bearing clearance of planet gears Angle of position of the planet gears

The single elements are combined in nine different types of planetary gears, for which the parameter study gives the required load factors. The planetary gears are categorized according to the designations of the first column of Table 2. Thus, for example, a transmission with a sun gear shaft and a housing-fixed internal gear receives the designation "S1H1." That transmission version forms the basis for the parameter variation. All other transmissions provide the

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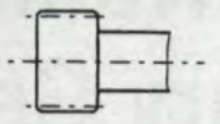
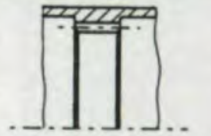
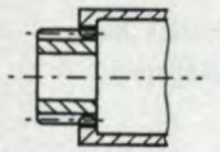
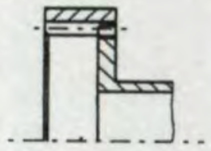
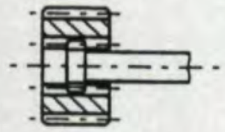
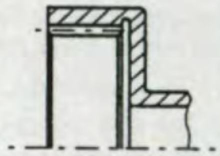
Prof. em. Dr.-Ing. Friedrich Jarchow
is the former head of the chair of mechanical components, industrial and automotive power transmission at the Ruhr University, Bochum, located in Germany.

Dr.-Ing. Christof Lamparski
is the head of the department for research and development of hydraulic components of Schwaebische Huettenwerke GmbH in Bad Schussenried, Germany. This paper is based on his doctoral dissertation.

Table 2—Construction design of sun and ring.					
Var.	**Description**	**Scheme**	**Var.**	**Description**	**Scheme**
S1	Sun gear		H1	Housing-fixed ring gear	
S2	Sun wheel with circumferential engaging curved-tooth gear coupling		H2	Ring gear with internal engaging curved-tooth gear coupling	
S3	Sun wheel with internal engaging curved-tooth gear coupling		H3	Ring gear with flange connection	

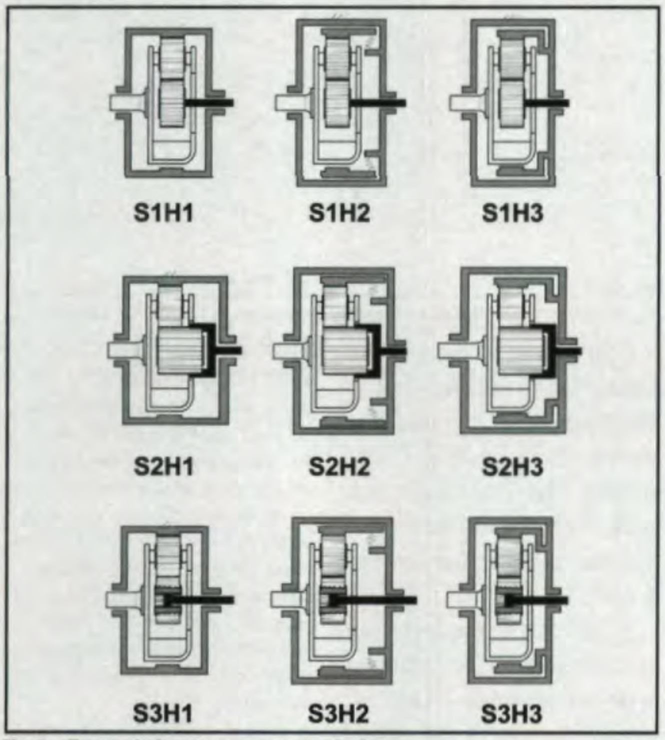


Fig. 2—Types of planetary gear transmissions.

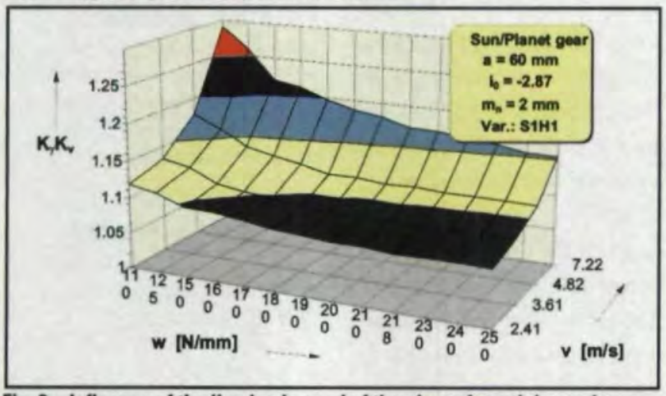


Fig. 3—Influence of the line load w and of the circumferential speed v on the factor product $K_I K_v$ in the engagement of sun and planet gear.

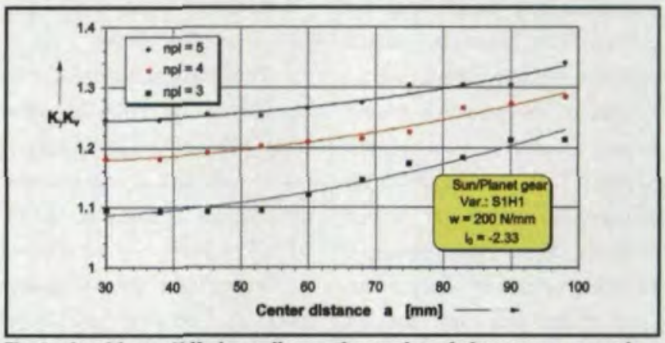


Fig. 4—Load factor $K_I K_v$ depending on the number of planet gears n_{pl} and on the center distance a, shown for the engagement of sun and planet gear.

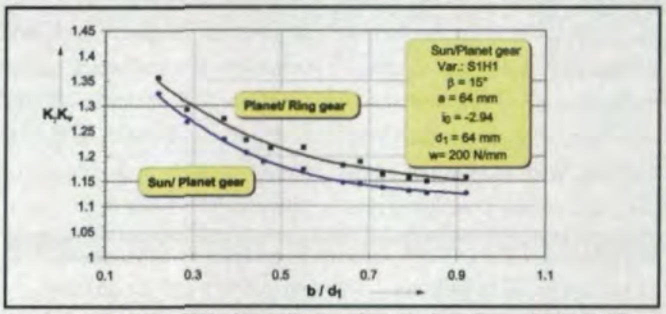


Fig. 5—Load factors $K_I K_v$, depending on the ratio from face width b to diameter d_1 of the sun wheel.

influences of the examined parameter field.

Influences on the load factors $K_I K_v$. The following results were achieved by a parameter study with the program SIMPLEX. It shows the generally valid tendencies that a modification of the single parameters according to Table 1 has on the tooth load conditions as mentioned above. Those tendencies are represented for a basic S1H1 transmission, according to Figure 2. The indicated values of the load factors apply to the transmission specified in the respective picture. The load factors, achieved by transmissions of other geometry or type of construction, can be determined by approximating equations, contained in Reference 1. The results apply to gears of DIN quality class 6.

All rolling bearings indicate a bearing clearance of class CN.

Figure 3 shows the influence of the specific load w for different peripheral speeds v on the dynamic behavior of the transmission.

The distribution of the load on several planet gears permits a higher throughput of power. Figure 4 shows the influence of the associated increasing static overconstraint and the influence of the phase position of the engagements at the circumference of the coaxial gears, dependent on the number of planet gears.

Figure 5 shows the effect of a changed face width b on the value of the load factors $K_\gamma K_v$. The basis of all calculations is a constant line load w, which corresponds to an increase of the outside torque with rising face width. The improvement of the dynamic behavior with an increasing face width does not result from the modified geometry, but from the risen load level.

The quality of gear production is crucial for the level of the dynamic excitations in the transmission. Both the total pitch variation F_p and the individual pitch variation f_p indicate a clear influence on the load factors.

Influences on the load factor $K_{H\beta}$. A misalignment of the axles causes a system-dependent inclination of the gears. For that reason, an increase of the effective tooth trace variation $f_{H\beta}$ can generally be expected. Figure 7 shows the dependence of the factor $K_{H\beta}$ on the quality of the tooth trace angle variation $f_{H\beta}$. In both engagements of the planet gear, a significant rise of the factor $K_{H\beta}$ is to be observed, already starting from DIN quality class 4. The housing-fixed ring gear of the basic version S1H1 causes a partial adjustment of the planet gear in reference to the ring gear, which results in a more favorable face width load distribution. Accordingly, the values of the factor $K_{H\beta}$ of the engagement planet/ring gear are lower than those of the engagement of sun/planet gear.

Figure 8 shows an increase of the factor $K_{H\beta}$ in the S1H1 version, which depends on the number of the engagements of sun/planet gear.

Approximation equations for the factor product $K_\gamma K_v$ ***and the factor*** $K_{H\beta}$. An important result of this research project is approximation equations for the determination of the load factors $K_\gamma K_v$ and $K_{H\beta}$. Detailed information is given in Reference 1.

Testing

A system of coupled, inhomogeneous, non-linear, second-degree differential equations describes the dynamic behavior of an idealized planetary gear. The solution of that system of differential equations provides the displacements of the geometrical centers of the sun/planet gears and of the planet carriers in the plane of transverse section. A comparison of the calculated and the measured axis shift allows for the control of the simulation program.

The measurements are taken at the self-adjusting transmission elements. Their movements can be registered with contact-free working displacement transducers. Two horizontally and two vertically attached displacement transducers determine the misalignment of the transmission elements. The mounting

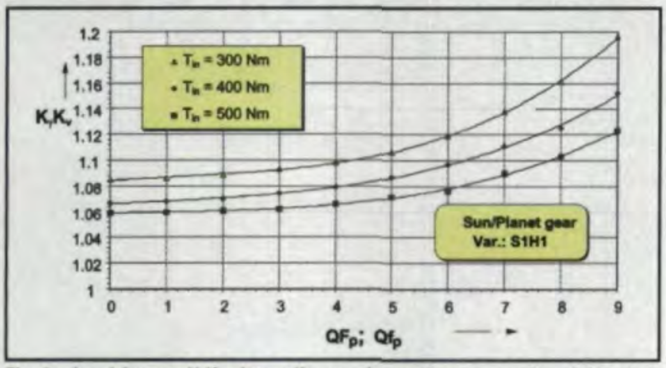


Fig. 6—Load factors $K_\gamma K_v$, depending on the production quality of the pitch variations.

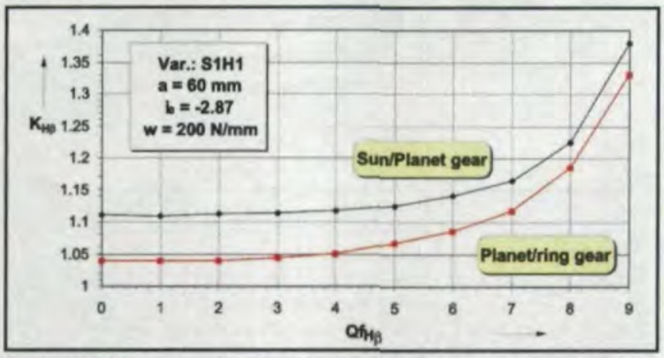


Fig. 7—Load factor $K_{H\beta}$ dependent on the quality of tooth trace angle variation $Qf_{H\beta}$, shown for the engagement of sun/planet gear and planet/ring gear.

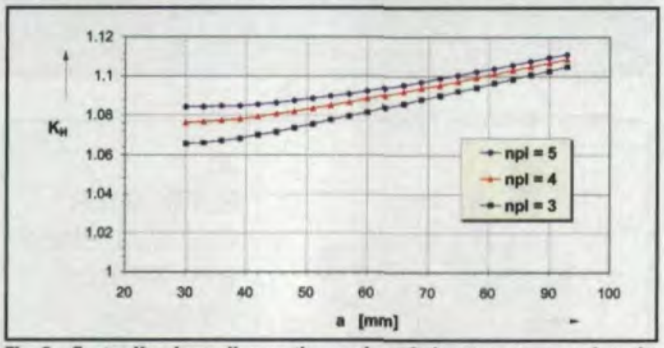


Fig. 8—Factor $K_{H\beta}$ depending on the number of planet gears n_{pl} and on the center distance a, shown for the engagement of sun and planet gear.

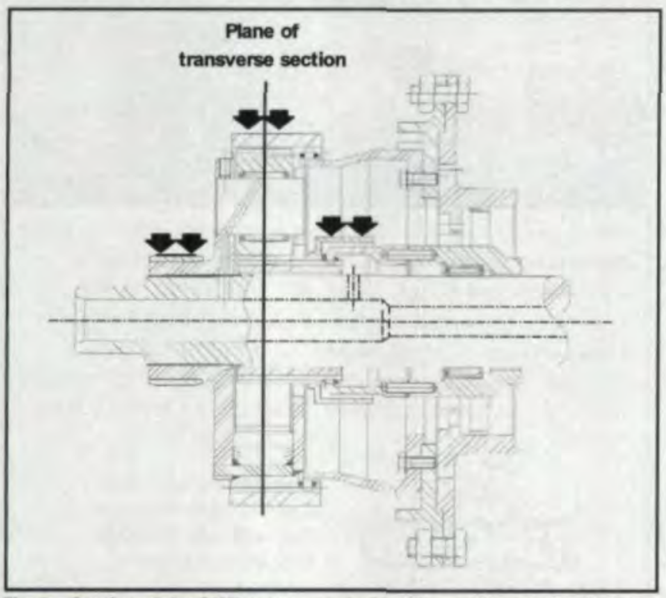


Fig. 9—Section of the S2H2 test transmission. The position of the displacement transducers is marked by arrows.

arrangement of the displacement transducers can be seen in Figure 9.

Reference 1 compares calculation and measurement and discovers a good correspondence between the respective values.

Notes for the Technical Designer

Based on recent findings of projects concerning load distribution in planetary gears, the following notes for design can be recorded:

1. With the rising extent of utilization of the transmissions, the dynamic effects lose some of their importance. Caused by a high line load, the ability of self-adjustment of the gears improves due to existing degrees of freedom and due to elasticity. The factors $K_\gamma K_v$ and $K_{H\beta}$ decrease with rising load.
2. An increasing circumferential velocity, on the other hand, shows opposite tendencies regarding the load balance. High-speed transmissions usually show very high load factors, especially when operated on low load levels.
3. Gearing deviations, such as F_p, f_p and $f_{H\beta}$, if toleranced according to DIN quality class 6 or better, only exert a small influence on the dynamic behavior of the transmission. Starting from DIN quality class 6, it is advisable to provide crowning or end relief.
4. Center distance deviations, tolerated according to DIN 3964, remain without influence on the load factors, as long as there is still some backlash existing. However, if the backlash is consumed, the loads can achieve several times the nominal values.
5. Large numbers of planet gears increase the degree of static overconstraint of the transmission. In order to select the number of planet gears, the gain at load carrying-capacity as a result of power branching has to be compared to the losses resulting from increased dynamic excitation.
6. The load factors $K_\gamma K_v$ of spur gear transmissions can be up to 10% higher than with comparable helical gearing transmissions.
7. For center distances less than 100 mm, no influence of size on the load distribution—or more precisely on the load factors $K_\gamma K_v$ and $K_{H\beta}$—could be determined.

References

1. Lamparski, C. *Simple equations for determining the load increase in light weight planetary gear transmissions,* Final report on research project 51/IV, Forschungsvereinigung Antriebstechnik, Frankfurt/Main, 1994.
2. Winkelmann, L. *Load distribution in helical planetary gear transmissions,* series of institute for construction, issue 87.3, Ruhr-Universität Bochum 1987.

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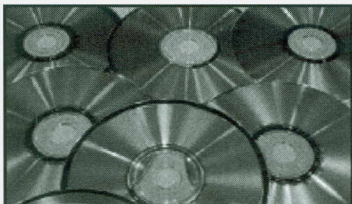

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New Name for Gear Software from Hornet GmbH

KISSsoft gear software has been renamed *NORA Transmission Software* and is being marketed by Hornet GmbH, a German company.

KISSsoft was originally developed for internal use by Kissling & Co., a Swiss gear manufacturer. In 1999, the program was combined with another software package, *Hirnware*. Except for the name, *NORA* is identical to that *KISSsoft/Hirnware* package.

NORA Transmission Software includes calculation modules for gears, springs, tie rods, transmission elements, shafts and bearings. The gearing can be calculated according to the most current AGMA, ISO or DIN standards for spur gears, bevel gears, worm gears and more.

For more information, contact Hornet GmbH by mail at Seewiesenstr. 90, D-73329 Kuchen, Germany; by telephone at (49) 7331-98-16-46; by fax at (49) 7331-98-16-47 or by e-mail at *info@hornet-gmbh.com*.

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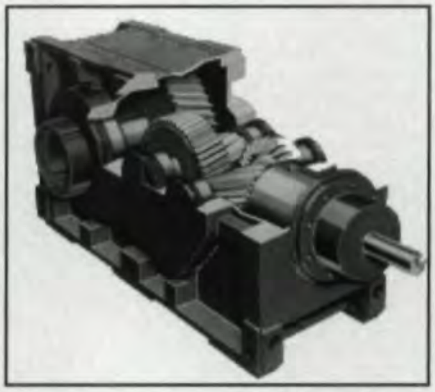

Falk Introduces Drive Line with Flexible Configurations

The Falk Corp. of Milwaukee, WI, introduced Drive One, a line of medium-torque gear drives with multiple mounting options and drive configurations.

The new line of drives, introduced in March, can be mounted from any of the housing's six sides. The drives can be configured for use as general-purpose parallel shaft drives and as right-angle drives, or they can be used as shaft-mounted drives, mixer drives or alignment-free drives.

Drive One gear drives come in four sizes, with torques ranging from 150,000 lb.-in. to 500,000 lb.-in. (17,000 Nm to 56,000 Nm).

For more information, visit *www.falkcorp.com.*

circle 328

New Small-Footprint Induction Machine

Welduction Corp. of Novi, MI, introduced its Flexscan induction hardening machine, a compact heat treating system, complete with controls and tooling for less than $100,000.

The new model can induction harden and temper parts up to 13" long.

The Flexscan comes with a built-in, solid-state power supply in a variety of power and frequency combinations. It also includes a quench cooling system and heat potentiometer.

For more information, contact Welduction by telephone at (800) 798-3042, by fax at (248) 735-2821, by e-mail at *weldn@welduction.com*, or on the Internet at *www.welduction.com*.

circle 329

New Wind Turbine Unit from Cincinnati Gear

The Cincinnati Gear Co. of Cincinnati introduced a 1.5 megawatt wind turbine gear unit for use in wind power generation applications.

The new unit is rated at 1.66 megawatts power and has a calculated design life of 20 years. Features include an integrated oil sump, an electric motor-driven lubrication pump, high capacity filtration and visual inspection ports for all gear meshes.

The gearbox has an input speed of 20 rpm, an output speed of 1,440 rpm and a ratio of 72:1. All gearing meets AGMA/AWEA 921-A97 and Germanischer Lloyd standards for wind turbine gearboxes.

For more information, contact The Cincinnati Gear Co. by mail at 5657 Wooster Pike, Cincinnati, OH 45227; by telephone at (513) 271-7700; by fax at (513) 271-0049 or by e-mail at *industrialsales@cintigear.com*. The company's Website is at *www.cintigear.com*.

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If you did not care for this column, **circle 326**.

If you would like to respond to this or any other article in this edition of *Gear Technology*, please fax your response to the attention of Randy Stott, managing editor, at 847-437-6618 or send e-mail messages to *people@geartechnology.com*.

COMPONENT GEAR MANUFACTURING

Literature available from The Cincinnati Gear Company provides information covering its facilities and expertise in component gear manufacturing, including hobbing and cutting, grinding, turning, boring, milling and inspection. The brochure includes product photos and specifications, as well as manufacturing capabilities.

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Presrite's 8-page color brochure discusses the company's forging capabilities, equipment and forging plants. The literature also highlights Presrite's Bessemer division, a near-net gear manufacturing facility, and identifies industries served and customer benefits.

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LITERATURE MART

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CIRCLE 184

GEAR *fashion*

parts together to create small swatches of "fabric" to see how they would hang on a woman's body as dress material. The parts were held together by jump rings, small nickel-plated, brass rings that are used in jewelry making.

Combining involute curves and body curves, merging factory and fashion, Winzeler Gear has transformed one of its products into gear *haute couture*: Winzeler Gear has created a plastic gear dress.

"I was trying to come up with something unusual to create a theme for our 60th anniversary party," says John Winzeler, the company's owner, "and that related to our already successful gear jewelry campaign."

Started in 1997, that campaign was a series of elegant, black-and-white photographs of a fashion model wearing rings, earrings and bracelets—all made from plastic gears. The understated ads graced the pages of *Auto World, Design News* and *Machine Design*.

The new gear dress consists of many, many, many copies of one plastic piece, a small, square part with a star-shaped internal gear spline. The part is used in a gear-clutch assembly, which itself is used in an actuator motor, for providing lumbar support in automobile seats.

Located in Harwood Heights, IL, a Chicago suburb, Winzeler Gear manufactures precision-molded plastic gears for the appliance, automotive and business machine industries.

Winzeler Gear's latest piece of gear fashion was created by a Chicago fashion artist, Cat Chow. Winzeler describes Chow as an artist who's trying to decide whether she's a sculptor or a fashion designer—"She's working in wearable art."

Commissioned last summer to create the dress, Chow toured Winzeler Gear's factory and took home a variety of plastic parts, including the part with the internal gear spline. As she recalls, the square part's star-shaped space intrigued her—"It added another dimension."

Chow obtained about 1,500 copies of the part to make the dress. She linked the parts together to create small swatches of "fabric" to see how they would hang on a woman's body as dress material.

One of the parts weighs almost nothing in a person's hand, but a dress of them weighs a lot on a woman's body. Winzeler estimates the dress weighs about 10 pounds—"It's a little bit on the heavy side."

Worn by fashion model Tiffany (right), Winzeler Gear's plastic gear dress is made up of small, plastic parts that feature star-shaped internal gear splines (above).

In addition, the dress's weight limited its length. Chow originally talked about it being a long dress. But, a long dress would have had structural problems. Chow estimates that a long dress might have weighed as much as 20 pounds or more; the dress would have been too heavy for its rings to hold together, and for its wearer to stand up.

"It became a short dress," Winzeler says.

In creating the dress, Chow turned the square parts so they looked diamond-shaped. Turning them allowed her to create the dress's V-neckline, its front and back V-hemlines and its V-shaped sleeves. She explains that creating the V-cuts "accentuates the part itself."

"It made it more interesting to have the part turned the diamond way," Chow says about the dress. She adds that turning the parts also broke the fabric's rigidity, gave the parts more depth and made the dress more exciting.

Winzeler says he commissioned Chow to create a dress because of his "interest in fashion." His interest was what acquainted him with Chow's work. He'd seen her work in Chicago, at the Museum of Contemporary Art, where he serves on the museum's permanent collection committee. In spring 2000, he saw three dresses created by Chow in a museum exhibit.

Months later, Chow's new gear dress was featured at a fashion show—at Winzeler Gear. The show was part of the company's anniversary party, held Oct. 20.

That day, the main aisle in Winzeler Gear's factory became the runway for the fashion show. Professional models wore the gear dress and other dresses created by Chow. The dresses included one made of o-rings and another made of a single 100-yard-long zipper.

"When we do a commission, we do not want to stifle the artist's creativity," Winzeler says. "We commissioned her to do whatever she wanted."

Since the party, the dress has been shown at some trade meetings. Also, Winzeler expects to display the dress at upcoming plastic shows. It's currently displayed in Winzeler Gear's lobby, along with pictures of the gear jewelry.

Like the jewelry, the dress may become part of a Winzeler Gear ad campaign. Winzeler has already started to work on plans for such a campaign.

The campaign's purpose? "To make sure people remember who we are," Winzeler says. He adds that whether people like the campaign or not, they will be talking about it. "A business-writer friend of mine calls it 'womp': word-of-mouth potential." ○

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