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JANUARY/FEBRUARY 2003

The Journal of Gear Manufacturing

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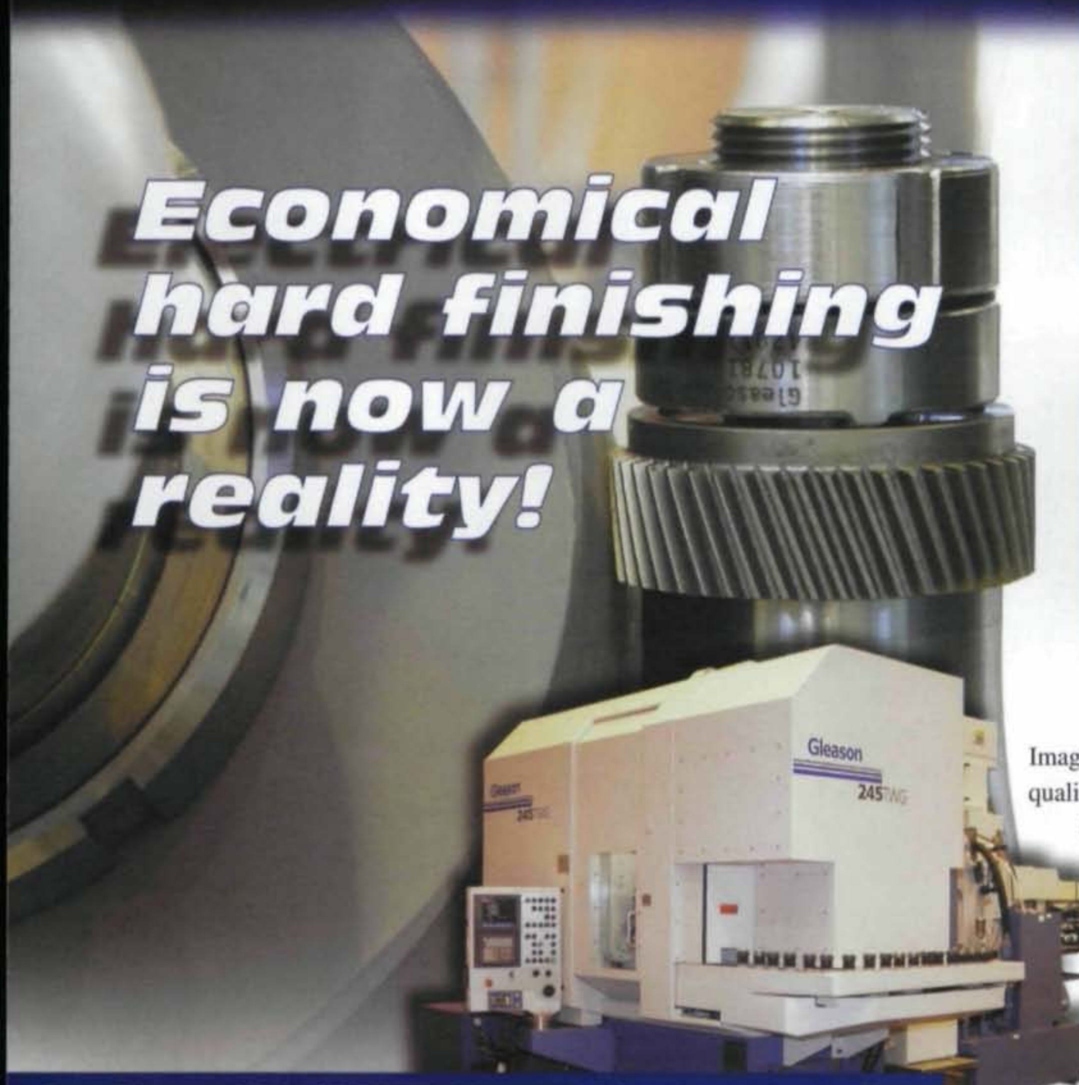

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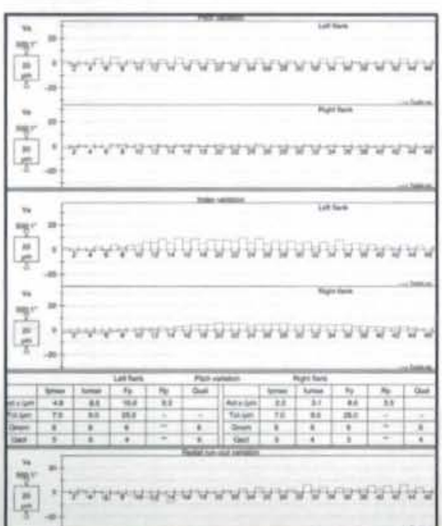

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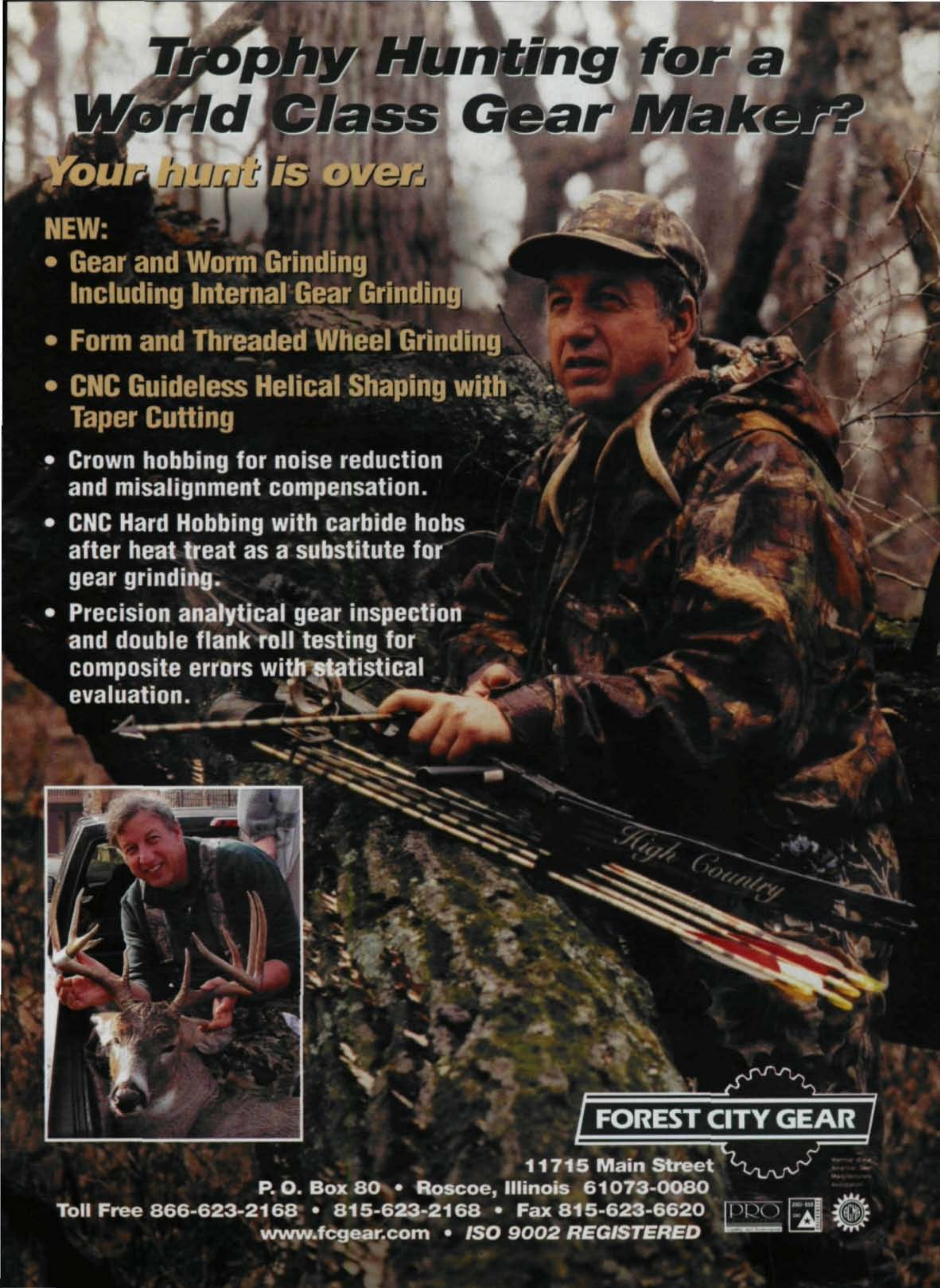
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Phone: 847-437-6604
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A Packed House

"Ahem. Could everybody please scootch over? We need to make as much room as possible, because we're expecting a full house."

As many of you know, we're being joined this issue by more than 1,200 subscribers who are reading the magazine for the first time via *E-GT*, the completely electronic version of *Gear Technology*.

The launch of *E-GT* places us at the forefront of the publishing world. We've worked very hard over the past year to make an exact duplicate of the printed version, including all articles and advertisements in full color, just as they appear in the magazine.

E-GT allows us to serve the gear industry better than ever before. Our mission has always been to provide vital technical information and news that helps make the gear industry more productive and competitive. As the gear industry has become more global, we've found the need to reach out to a wider, more global audience. *E-GT* lets us better reach all the places where gears are being designed, made, bought and sold.

Before *E-GT*, subscribers outside the United States had to pay for *Gear Technology*. Now, with *E-GT*, qualified subscribers anywhere in the world have free access to the same information that appears in our printed version. Previously, where an overseas gear manufacturer might have subscribed to one copy of the magazine, now five or even ten people subscribe in the same location.

Already, the response has been overwhelming. Many of those who have viewed our sample pages online and subscribed to *E-GT* have written to tell us what they thought about the new format.

For example, the chief engineer of a major gear cutting tools factory in China told us, "I am glad to be one of the subscribers of *E-GT*. I think I just found a shortcut to contact the advanced gear world."

A mechanical engineer at a gear manufacturing company in Argentina wrote, "I just want to congratulate you for the idea of having an electronic version of your magazine free on the Net."

A new *E-GT* subscriber from Malaysia said, "This provides updated gear technology know-how and related information—appreciated."

"I am happy to be able to read your magazine again, now as a .PDF instead of printed, and free of charge instead of by payment," said an engineer who works for a manufacturer of gearmotors in the Netherlands.

Others have written just to wish us well and to congratulate us on the new format. "Go on and good luck," wrote the director of a gear manufacturing company in Brazil, after he filled out the subscription form for *E-GT*.

Up to now, *Gear Technology* had subscribers in 35 countries outside the United States. Today, thanks to *E-GT*, our subscribers are in more than 50 countries, and the number of people we reach outside the United States has more than doubled.

In some places, like India, the number of subscriptions has nearly tripled. Also, we've expanded our readership to places the magazine has never gone before. Today, our magazine is read in: Argentina, Australia, Austria, Belgium, Brazil, Bulgaria, Canada, Chile, China, Colombia, the Czech Republic, Denmark, France, Germany, Greece, Hong Kong, Hungary, India, Indonesia, Italy, Japan, Korea, Latvia, Luxembourg, Malaysia, Mexico, the Netherlands, New Zealand, Oman, Pakistan, Peru, the Philippines, Poland, Portugal, Romania, Russia, Singapore, Slovakia, Slovenia, South Africa, Spain, Sweden, Switzerland, Taiwan, Thailand, Turkey, the United Kingdom, Venezuela, Yugoslavia and Zimbabwe. This list will continue to grow with *E-GT*.

But *E-GT* isn't just for those outside the United States. We're pleasantly surprised that *E-GT* is just as attractive to our readers here in America, many of whom now subscribe to both the print and electronic versions.

"I am impressed with your electronic approach to *Gear Technology* magazine," said the product development manager of a Midwest gear manufacturer.

"I look forward to seeing and receiving *Gear Technology* online," said the general manager of a Midwest gear manufacturing job shop.

All of this hubbub about *E-GT* doesn't mean we're going to change anything in our regular printed version. We'll continue to bring you the same high-quality product, including the best technical articles, feature articles, news and other information related to gear design, manufacturing, inspection, processing and buying. That quality applies to both our printed and electronic versions, but now you have a choice in how you want to receive the information.

For those of you who haven't yet seen *E-GT*, I encourage you to log on to *www.geartechnology.com* and try the sample pages. If you think *E-GT* could be useful to you or your company, then sign up for a free subscription, and let us know what you think about our latest endeavor. We can always squeeze in a few more subscribers.

Sincerely,

Michael Goldstein

Michael Goldstein, Publisher & Editor-in-Chief

"Hey, buddy! Can you PLEASE make some room down there?"

New Gear Software

GearTrax 2003 to Model Spiral Bevel Gears

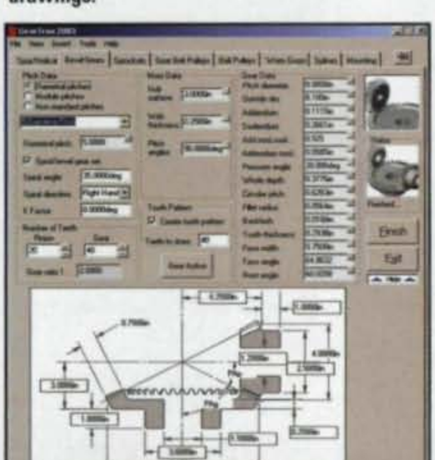
Gear designers using GearTrax software to model gears can expand that software this new year by obtaining its new feature: the ability to model spiral bevel gears.

Created by Camnetics Inc., GearTrax is used to model gears in SolidWorks.

Gregory Hottman, Camnetics' founder and president, designs GearTrax to be compatible with that modeling program, made by SolidWorks Corp.

Gear designers can obtain the new feature as part of GearTrax's new 2003

GearTrax 2003 has a new feature that can create 3-D models of spiral bevel gears for assembly drawings.



GearTrax 2003 creates models of spiral bevel gears by operating with SolidWorks. The new GearTrax program creates those models from various gear data.

version. If they bought GearTrax's support plan, then they can download the new version free from Camnetics' website. If not, then they can buy GearTrax 2003 itself.

GearTrax 2003 creates solid models of gears and other drive parts. Hottman recommends his new program for modeling of spiral bevel gears and worm gears and for rapid prototyping of straight bevel gears, involute splines, internal and external spur gears, and internal and external helical gears. He also recommends his software for plastic injection molding, powder-metal compacting and wirecutting/EDM of involute splines and internal and external spur gears.

Hottman explains the modeling of spiral bevel gears would be useful for assembly drawings to check the fit of gear and pinion in a gearbox housing and their fit and function in relation to the gearbox's other parts. The electronic images could also be used in customer presentations.

However, he doesn't recommend his latest version for creating the geometry needed to manufacture a spiral bevel gear by traditional metal-cutting methods.

Hottman expects GearTrax 2003 to be available for $595 in January 2003.

The new feature will create 3-D models of spiral bevel gears from various gear data, including the number of teeth on gear and pinion; the diametral pitch; geometry parameters—like face width and backlash; the hub diameter, which must be given to the program; the pitch diameter, which the program calculates itself; and the spiral angle and direction of the gear's teeth.

These models are created according to Gleason Corp.'s spiral bevel gear standards. Hottman obtained those standards from the 24th edition of *Machinery's Handbook,* published in 1992 by Industrial Press Inc., located in New York, NY.

***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, fax (847) 437-6618 or e-mail* people@geartechnology.com. *If you'd like more information about any of the articles that appear, please use Rapid Reader Response at* www.geartechnology.com/rrr.htm.**

Like earlier versions, GearTrax 2003 only creates models of gears; it doesn't rate gears, take material data into account or provide power capacities.

GearTrax 2003 will operate as either an add-in or an add-on to SolidWorks. Gear designers can use GearTrax 2003 as an add-in by starting and operating it through SolidWorks. They can use it as an add-on by starting GearTrax 2003 itself and operating SolidWorks through it. Hottman says that GearTrax 2003 runs faster as an add-in.

Besides Camnetics, IronCAD L.L.C. of Atlanta, GA, offers software that can create 3-D models of gears, including spiral bevel gears. Universal Technical Systems Inc. of Rockford, IL, also offers gear modeling software. UTS models are restricted to internal and external spur gears and external helical gears, but they're accurate for creating molds for making plastic or powder-metal gears or dies for making forged gears.

Hottman is timing the addition of GearTrax's new feature with the introduction of SolidWorks 2003. The revised SolidWorks program became available in November.

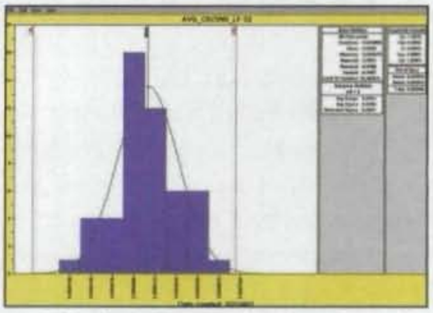
Hottman explains that his software is tied to SolidWorks software—"The vast majority of my customers are SolidWorks users." Consequently, each time SolidWorks is updated, Hottman usually has to update GearTrax to be sure they remain compatible.

Based in Oregon, WI, Hottman describes his main customers as people who design products that use a fair amount of gears. Those customers include automation/machine designers and gear manufacturers. His gear manufacturers tend to be smaller companies.

Hottman created the spiral bevel gear feature because it was requested by some of his customers, including a gear manufacturer and a gearbox designer for motorcycle transmissions and all-terrain vehicles.

"I've had people specifically looking for that function," he says.

G.E.A.R.S. Software—SPC for Gear Manufacturers



Nicholson Gear Technology has software to make it faster and easier for a gear manufacturer to statistically track its processes to be sure they're manufacturing gears within design tolerances. Specifically designed for gear data, the software is an interface between the data collected from metrology equipment and a connected SPC program.

The software comes in two versions: a network version available since 1996 for large companies that manufacture gears and a new non-network version for small to intermediate-sized companies.

The network version is called Gear Engineering Analysis Retrieval System—G.E.A.R.S. for short. The non-network version is called G.E.A.R.S. Lite.

G.E.A.R.S. Lite was created because G.E.A.R.S.—starting at $25,000—was too expensive for some potential customers. G.E.A.R.S. Lite, however, starts at $6,700.

"We felt that we weren't reaching the customers," Nicholson says.

Nicholson is the founder and president of Nicholson Gear Technology, based in Medina, OH. He's also product manager for the spiral bevel gear division of SU America Inc., located in Hoffman Estates, IL.

As a network program, G.E.A.R.S. uses a dedicated server for storing data and can operate on multiple personal computers in a network.

As a non-network program, G.E.A.R.S. Lite is meant to be installed on only one personal computer and uses only the computer's hard drive for storing data. Normally, the program operates with the statistical-process-control (SPC) program CHARTrunner, made by PQ Systems of Dayton, OH.

Bought as a package, G.E.A.R.S. Lite includes the gear program itself, CHARTrunner and the installation CD-ROM.

G.E.A.R.S. Lite can also operate with other SPC programs, such as SPC for Excel, made by SQC Development Corp. of Newark, DE.

According to Nicholson, any SPC program on the market can retrieve and analyze gear data, but those programs may be difficult and time-consuming to use. He explains that G.E.A.R.S. and G.E.A.R.S. Lite were designed to be easy to use and that people don't need to know much about statistical process control to use them. Nicholson himself has trained groups of people to use G.E.A.R.S. in half-day classes. He adds that learning to use G.E.A.R.S. Lite is no different from learning to use G.E.A.R.S. "The basic structure itself is identical," he says of his two programs.

G.E.A.R.S. and G.E.A.R.S. Lite collect and query data on all gear characteristics from metrology equipment via a company's Intranet.

Because it lacks a server, G.E.A.R.S. Lite doesn't handle as much data as G.E.A.R.S. or return queries as quickly as G.E.A.R.S. Still, Nicholson says G.E.A.R.S. Lite can be connected to any number of measuring or inspection machines.

G.E.A.R.S. and G.E.A.R.S. Lite import data from metrology equipment as CSV (comma-separated variable) data. The data can be reviewed one gear characteristic at a time and is printed on standard inspection reports. The characteristics include average lead slope, left and right flank; average slope, left and right flank; average involute slope/profile variation; average index variation; and average crown on lead, left and right flank. The data is also transferred to CHARTrunner, so engineers can review the data in either run charts or histograms.

"With G.E.A.R.S. Lite, we keep it as simple as possible," Nicholson says. He adds that the whole basis of the program was to make it easy to use.

As Nicholson explains, people don't have time today to write importing code for SPC programs. They only have time to select their part number, process, date range, and analysis types, then create their SPC charts.

G.E.A.R.S. Lite can be customized by its buyers as needed. Nicholson describes the program's possible customers as any non-automotive company that manufactures gears. In contrast, he says G.E.A.R.S. is suitable for automotive companies and has been bought by several automotive plants.

Nicholson adds he has already sold his first copy of G.E.A.R.S. Lite to a company, a North American aerospace business. ⭘

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Gear Damage Detection Using Oil Debris Analysis

Paula Dempsey

Abstract

The purpose of this paper was to verify, when using an oil debris sensor, that accumulated mass predicts gear pitting damage and to identify a method to set threshold limits for damaged gears. Oil debris data was collected from eight experiments with no damage and eight with pitting damage in the NASA Glenn Research Center's spur gear fatigue rig. Oil debris feature analysis was performed on this data. Video images of damage progression were also collected from six of the experiments with pitting damage. During each test, data from an oil debris sensor was monitored and recorded for the occurrence of pitting damage. The data measured from the oil debris sensor during experiments with no damage was used to identify membership functions, which are required to build a simple fuzzy-logic model. Using fuzzy-logic techniques and the oil debris data, threshold limits were defined that discriminate between stages of pitting wear. Results indicate that accumulated mass combined with fuzzy-logic analysis techniques is a good predictor of pitting damage on spur gears.

Introduction

One of NASA's current goals, the National Aviation Safety Goal, is to reduce the aircraft accident rate by a factor of five within 10 years and a factor of 10 within 25 years. One of the leading factors in fatal aircraft accidents is loss of control in flight, which can occur due to flying in severe weather conditions, pilot error and vehicle/system failure. Focusing on helicopters' system failures, an investigation in 1989 found that 32% of helicopter accidents due to fatigue failures were caused by damaged engine and transmission components (Ref. 1).

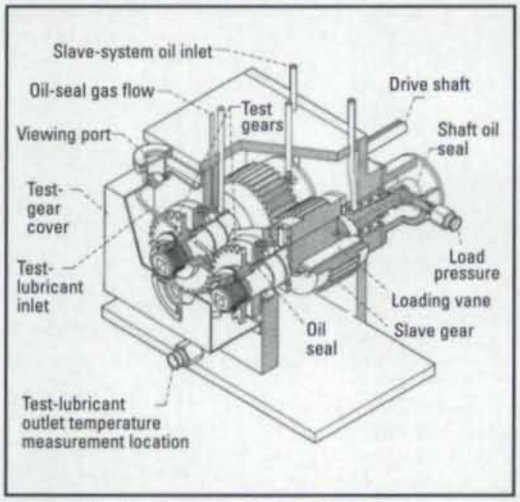

Figure 1—Spur gear fatigue test rig.

In more recent statistics, of the world total of 192 turbine helicopter accidents in 1999, 28 were directly due to mechanical failures with the most common failure in the drive trains of gearboxes (Ref. 11).

A study published in July 1998, in support of the National Aviation Safety Goal, recommended areas most likely to reduce rotorcraft fatalities in the next 10 years. The study of 1,168 fatal and nonfatal accidents that occurred from 1990–1996 found that, after human factor-related causes of accidents, the next most frequent cause of accidents was due to various system and structural failures (Ref. 2). Loss of power in flight caused 26% of this type of accident and loss of control in flight caused 18% of this type of accident. The technology area recommended by this study for helicopter accident reduction was helicopter health and usage monitoring systems (HUMS) capable of predicting imminent equipment failure for on-condition maintenance and more advanced systems capable of warning pilots of impending equipment failures.

Helicopter transmission diagnostics are an important part of a helicopter health monitoring system because helicopters depend on the powertrain for propulsion, lift and flight maneuvering. In order to predict transmission failures, the diagnostic tools used in the HUMS must provide real-time performance monitoring of aircraft operating parameters and must demonstrate a high level of reliability to minimize false alarms. Various tools exist for diagnosing damage in helicopter transmissions, the most common being vibration tools. Using vibration data collected from gearbox accelerometers, algorithms are developed to detect when gear damage has occurred (Refs. 16 and 20). Oil debris is also used to identify abnormal wear-related conditions of transmissions. Oil debris monitoring for gearboxes consists mainly of off-line oil analysis or plug-type chip detectors. And, although not commonly used for gear damage detection, many engines have on-line oil debris sensors for detecting the failure of rolling element bearings. These on-line, inductance-type sensors count the number of particles, measure their approximate sizes, then calculate an accumulated mass (Ref. 10).

The goal of future HUMS is to increase reliability and decrease false alarms. HUMS are not yet capable of real-time, on-line health monitoring. Current data collected by HUMS is processed after the flight and is plagued with high false alarm rates and undetected faults. The current fault detection rate of commercially available HUMS through vibration analysis is 60%. False warning rates average one per 100 flight hours (Ref. 17). This is due to a variety of reasons. Vibration-based systems require extensive interpretation by trained diagnosticians. Operational effects can adversely impact the performance of vibration diagnostic parameters and result in false alarms (Refs. 5 and 3). Oil debris sensors also require expert analysis of data. False alarms of oil debris technologies are often caused by nonfailure debris. This debris can bridge the gap of plug-type chip detectors. Inductance-type oil debris sensors cannot differentiate between fault and no-fault sourced data (Ref. 8).

Several companies manufacture on-line, inductance-type oil debris sensors that measure debris size and count particles (Ref. 10). New oil debris sensors are also being developed that measure debris shape and size, and the shape is used to classify the failure mechanism (Ref. 8). The oil debris sensor used in this analysis was selected for several reasons. The first three reasons were sensor capabilities, availability and researcher experience with this sensor. Results from preliminary research indicate the debris mass measured by the oil debris sensor showed a significant increase when pitting damage began to occur (Ref. 4).

This sensor has also been used in aerospace applications for detecting bearing failures in aerospace turbine engines. From the manufacturers' experience with rolling element bearing failures, an equation was developed to set warning and alarm transmissions. A modified version of this

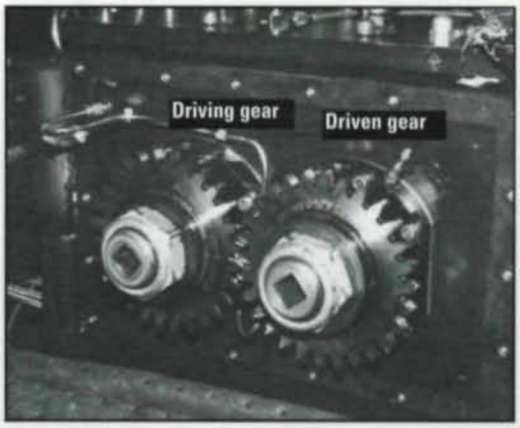

Figure 2—Spur gear fatigue rig gearbox.

Table 1–Oil debris particle size ranges.

Bin	Bin range, µm	Average size, µm	Bin	Bin range, µm	Average size, µm
1	125–175	150	9	525–575	550
2	175–225	200	10	575–625	600
3	225–275	250	11	625–675	650
4	275–325	300	12	675–725	700
5	325–375	350	13	725–775	750
6	375–425	400	14	775–825	800
7	425–475	450	15	825–900	862.5
8	475–525	500	16	900–1,016	958

sensor has been developed and installed in an engine's nose gearbox and is currently being evaluated for an operational AH-64 helicopter (Ref. 10), which is Boeing Co.'s Apache attack helicopter. Due to limited access to oil debris data collected by this type of sensor from gear failures, no such equation is available that defines oil debris threshold limits for damaged gears.

The objective of the work reported herein is to first identify the best feature for detecting gear pitting damage from a commercially available on-line oil debris sensor. Then, once the feature is defined, the objective is to identify a method to set threshold limits for different levels of pitting damage to gears. The oil debris data analysis was performed on gear damage data collected from an oil debris monitor in the NASA Glenn Research Center's spur gear fatigue rig.

Test Procedure

Experimental data was recorded from tests performed in the NASA Glenn rig (Ref. 16). This rig is capable of loading gears, then running them until pitting failure is detected. A sketch of the test rig is shown in Figure 1. Torque is applied by a hydraulic loading mechanism that twists one slave gear relative to its shaft. The power required to drive the system is only enough to overcome friction losses in the system (Ref. 13). The test gears are standard spur gears having 28 teeth, 8.89 cm pitch diameters and 0.64 cm face widths. The test gears are run offset to provide a narrow

Paula Dempsey, Ph.D.,
is an aerospace research engineer employed by the NASA Glenn Research Center in Cleveland, OH. She joined NASA in 1991, after performing experimental and analytical research work in the area of cryogenic fluid management systems for a government contractor from 1987–1991. She continued her cryogenic research until 1995, then served as lead project engineer for several aeronautics programs. She is currently performing research in the area of monitoring and diagnostics for helicopter drive systems and propulsion systems for transport aircraft.

effective face width to maximize gear contact stress while maintaining an acceptable bending stress. Offset testing also allows four tests on one pair of gears. Two filters are located downstream of the oil debris monitor to capture the debris after the sensor measures it.

Fatigue tests were run in a manner that allows damage to be correlated to the oil debris sensor data. For these tests, run speed was 10,000 rpm and applied torque was 72 N-m and 96 N-m. Prior to collecting test data, the gears were run-in for one hour at a torque of 14 N-m. The data measured during this run-in was stored, then the oil debris sensor was reset to zero at the start of the loaded test. Test gears were inspected periodically for damage either manually or using a microcamera connected to a videocassette recorder and monitor. The video inspection did not require gearbox cover removal. When damage was found, it was documented and correlated to the test data based on a reading number. Reading numbers are equivalent to minutes and can also be interpreted as mesh cycles equal to reading numbers multiplied by 10^4. In order to document tooth damage, reference marks were made on the driving and driven gears during installation to identify tooth 1. The mating teeth numbers on the driving and driven gears were then numbered from this reference. Figure 2 identifies the driving and driven gears with the gearbox cover removed.

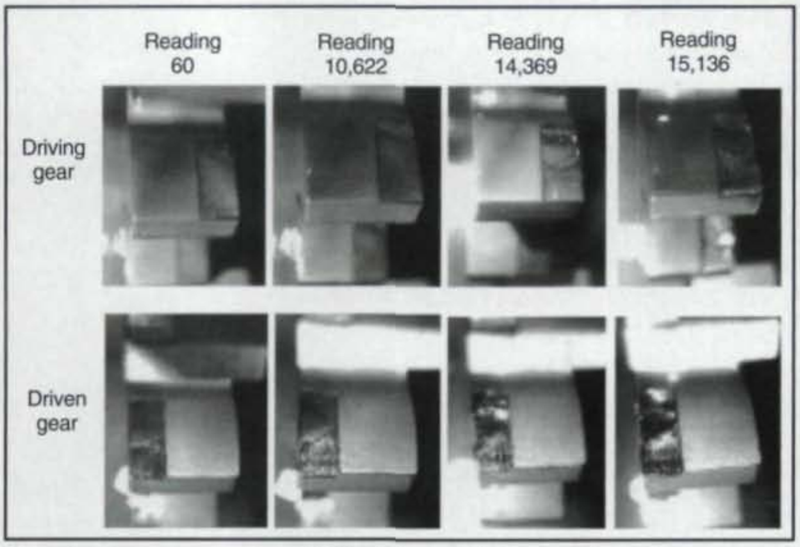

Figure 3—Damage progression of driving/driven tooth 6 for experiment 1.

Data was collected once per minute from oil debris, speed and pressure sensors installed on the test rig using the programs ALBERT, Ames-Lewis Basic Experimentation in Real Time, co-developed by NASA Glenn and NASA Ames Research Center. Oil debris data was collected using a commercially available oil debris sensor that measures the change in a magnetic field caused by passage of a metal particle where the amplitude of the sensor output signal is proportional to the particle mass. The sensor counts the number of particles, measures their approximate sizes (125–1,016 μm) and calculates an accumulated mass (Ref. 9). Shaft speed was measured by an optical sensor once per shaft revolution. Load pressure was measured using a capacitance pressure transducer.

The principal focus of this research is detection of pitting damage on spur gears. Pitting is a fatigue failure caused by exceeding the surface fatigue limit of the gear material. Pitting occurs when small pieces of material break off from the gear surface, producing pits on the contacting surfaces (Ref. 19). Gears are run until pitting occurs on several teeth. Pitting was detected by visual observation through periodic inspections on two of the experiments with pitting damage. Pitting was detected by a video inspection system on six of the experiments with pitting damage. Two levels of pitting were monitored, initial and destructive pitting. Initial pitting is defined as pits less than 0.04 cm in diameter and covering less than

Table 2—Experiments with video inspection.

Experiment 1		Experiment 2		Experiment 3		Experiment 4		Experiment 5		Experiment 6	
Reading #	Mass, mg	Reading #	Mass, mg	Reading #	Mass, mg	Reading #	Mass, mg	Reading #	Mass, mg	Reading #	Mass, mg
60	1.003	1,573	3.285	58	0	64	0	62	0	60	0
120	1.418	2,199	8.934	2,669	8.69	150	2.233	1,405	4.214	2,810	3.192
1,581	5.113	2,296	16.267	2,857	11.889	378	8.297	2,566	7.413	2,885	6.396
10,622	12.533	2,444	26.268	3,029	14.148	518	9.462	4,425	10.811	2,957	8.704
14,369	15.475					2,065	12.132			9,328	11.692
14,430	22.468					2,366	13.977			12,061	14.365
14,512	24.586					3,671	17.361			12,368	22.851
14,688	28.451					4,655	23.12				
14,846	30.686					4,863	26.227				
15,136	36.108										

*Note: Highlighted cells identify reading and mass when destructive pitting was first observed.

approximately 25% of tooth contact area. Destructive pitting is more severe and is defined as pits greater than 0.04 cm in diameter and covering more than approximately 25% of tooth contact area. If not detected in time, destructive pitting can lead to catastrophic transmission failure if the gear teeth crack.

Discussion of Results

The analysis discussed in this section is based on oil debris data collected during 16 experiments, 8 of which resulted in pitting damage. The oil debris sensor records counts of particles in bins set at particle size ranges measured in microns. The particle size ranges and average particle size are shown in Table 1. The average particle size for each bin is used to calculate the cumulative mass of debris for the experiment. The shape of the average particle is assumed to be a sphere with a density of approximately 7,922 kg/m^3.

Experiments 1–6 were performed with the video inspection system installed on the rig. Table 2 lists the reading numbers and the measured oil debris masses at those readings. The highlighted cells for each experiment identify the reading number and the mass measured when destructive pitting was first observed on one or more teeth. As this table shows, the amount of mass varied significantly for each experiment. A representative sample of the images obtained from the video damage progression system is shown in Figure 3. The damage progression of tooth 6 on the driving and driven gears for experiment 1 for selected readings is shown in this figure. The damage is shown on less than half of the tooth because the test gears are run offset to provide a narrow effective face width to maximize gear contact stress.

Experiments 7 and 8 were performed with visual inspection. Table 3 lists the reading numbers when inspection was performed and the measured oil debris masses at these readings. Only initial pitting occurred during experiment 7. During experiment 8, initial pitting was observed at reading 5,181 and destructive pitting at reading 5,314.

No gear damage occurred during experiments 9–16. Oil debris mass measured at test completion is listed in Table 4. At the completion of experiment 10, 5.453 mg of debris was measured, yet no damage occurred. This result is more than the debris measured during experiment 7 (3.381 mg) when initial pitting was observed. This result and observations made from the data collected during experiments when damage occurred made it obvious that simple linear correlations could not be used to obtain the features for damage levels from the oil debris data.

Table 3—Experiments with visual inspection.

Experiment 7		Experiment 8		Pitting Damage
Reading #	Mass, mg	Reading #	Mass, mg	
13,716	3.381	5,181	6.012	Initial
		5,314	19.101	Destructive

Table 4–Oil debris masses at completion of experiments with no damage.

Experiment	Reading #	Mass, mg	Experiment	Reading #	Mass, mg
9	29,866	2.359	13	25,259	3.159
10	20,452	5.453	14	5,322	0
11	204	0.418	15	21,016	0.125
12	15,654	2.276	16	21,446	0.163

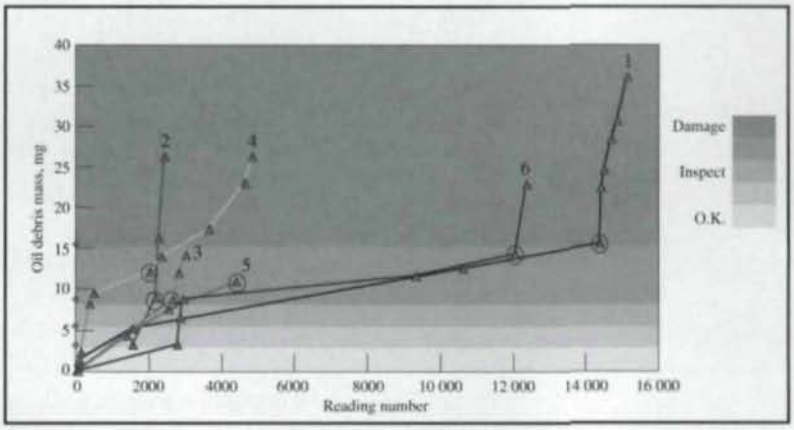

Figure 4—Oil debris masses at different damage levels.

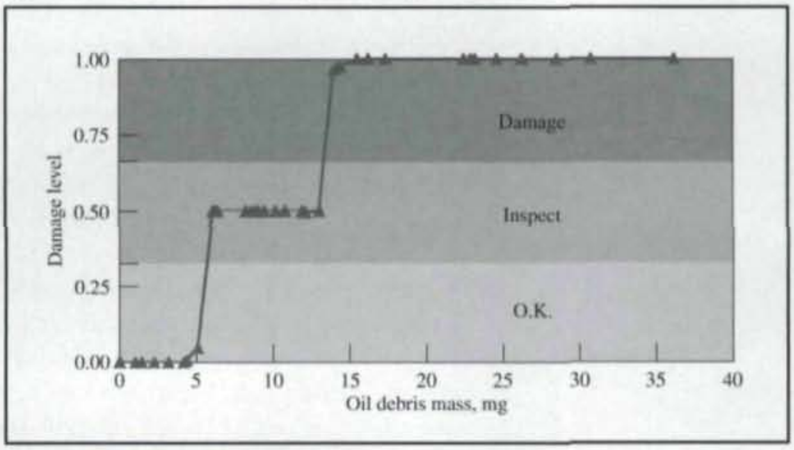

Figure 5—Output of fuzzy-logic model.

Prior to discussing methods for feature extraction, it may be beneficial for the reader to get a feel for the amount of debris measured by the oil debris sensor and the amount of damage to one tooth. Applying the definition of destructive pitting, 25% of tooth surface contact area for one tooth for these experiments is approximately 0.043 cm^2. A 0.04 cm diameter pit, assumed spherical in shape, is equivalent to 0.26 mg of oil debris mass. This mass is calculated based on the density used by the sensor software for calculating mass. If 0.04 cm diameter pits densely covered 25% of the surface area of one tooth, it would be equivalent to approximately 9 mg. Unfortunately, damage is not always densely distributed on 25% of a single tooth, but is distributed across many teeth, making accurate measures of material removed per tooth extremely difficult.

Several predictive analysis techniques were reviewed to obtain the best feature to predict damage levels from the oil debris sensor. One technique for detecting wear conditions in gear systems is by applying statistical distribution methods to particles collected from lubrication systems (Ref. 15). In this reference, mean particle size, variance, kurtosis and skewness distribution characteristics were calculated from oil debris data collected off-line. The wear activity was determined by the calculated size distribution characteristics. In order to apply this data to on-line debris data, calculations were made for each reading number for each bin. Mean particle size, relative kurtosis and relative skewness were calculated for each reading for six of the experiments with pitting damage. It was not possible, however, to extract a consistent feature that increased in value from the data for all experiments. This may be due to the random nonlinear distribution of the damage progression across all 56 teeth. For this reason, a more intelligent feature extraction system was analyzed and will be discussed in the following paragraphs.

When defining an intelligent feature extraction system, the gear states that a person plans to predict must be defined. Due to the overlap of the accumulated mass features, three primary states of the gears were identified: OK (no gear damage), inspect (initial pitting) and damage (destructive pitting). The data from Table 2 was plotted in Figure 4. Each plot is labeled with experiment numbers 1–6. The triangles on each plot identify the inspection reading numbers. The triangles circled indicate the reading number when destructive pitting was first observed. The background color indicates the OK, inspect and damage states. The overlap between the states is also identified with a different background color. The changes in states for each color were defined based on data shown in Tables 2–4. The minimum and maximum debris masses measured during experiments 1–6 when destructive pitting was first observed were used to define the upper limit of the inspect scale and the lower limit of the damage scale, respectively. The maximum amount of debris measured when no damage occurred (experiment 10) was above the minimum amount of debris measured when initial pitting occurred (experiment 7). The former was used as the lower limit of the inspect state. The next largest mass measured when no damage occurred (experiment 13) was used as the upper limit of the OK scale.

Fuzzy logic was used to extract an intelligent feature from the accumulated mass measured by the oil debris sensor. Fuzzy logic was chosen based on the results of several studies to compare the capability of production rules, fuzzy logic and neural nets. One study found fuzzy logic the most robust when monitoring transitional failure data on a gearbox (Ref. 7). Another study comparing automated reasoning techniques for condition-based maintenance found fuzzy logic more flexible than standard logic because it made allowances for unanticipated behavior (Ref. 14). Fuzzy logic applies fuzzy set theory to data, where fuzzy set theory is a theory of classes with unsharp boundaries and the data belongs in a set based on its degree of membership (Ref. 20). The degree of membership can be any value between 0 and 1.

Defining the fuzzy logic model requires inputs (damage detection features), outputs (state of gear), and rules. Inputs are the levels of damage, and outputs are the states of the gears. Membership values were based on the accumulated mass and the amount of damage observed during inspection. Membership values are defined for the three levels of damage: damage low, damage medium and damage high. Using the mean-of-the-maximum (MOM), fuzzy-logic defuzzification method, the oil debris mass measured during the six experiments with pitting damage was entered into a simple fuzzy-logic model created using commercially available software (Ref. 6). The output of this model is shown on Figure 5. Threshold limits for the accumulated mass are identified for future tests in the spur gear fatigue test rig. Results indicate accumulated mass is a good predictor of pitting damage on spur gears and fuzzy logic is a good technique for setting threshold limits that discriminate between states of pitting wear.

Conclusions

The purpose of this research was to first verify that accumulated mass predicts gear pitting damage when using an inductance-type, on-line oil debris sensor. Then, using accumulated mass as the damage feature, the purpose was to identify a method to set threshold limits for damaged gears that discriminate between different levels of pitting damage. In this process, the membership functions for each feature state were defined based on the level of damage. From this data, and a simple fuzzy-logic model, accumulated mass measured by an oil debris sensor combined with fuzzy-logic analysis techniques can be used to predict transmission health. Applying fuzzy logic incorporates decision making into the diagnostic process that improves fault detection and decreases false alarms.

This approach has several benefits compared with using the accumulated mass and an arbitrary threshold limit for determining if damage has occurred. One benefit is that it eliminates the need for an expert diagnostician to analyze and interpret the data since the output would be one of three states: OK, inspect and shutdown. Since benign debris may be introduced into the system due to periodic inspections, setting the lower limit above this debris level will minimize false alarms. In addition to these benefits, a more advanced system can be designed with logic built in to minimize these operational effects. Future tests are planned to collect data from gears with initial pitting to better define the inspect region of the model and the severity of gear damage. Tests are planned for gears of different sizes to determine if a relationship can be developed between damage levels and tooth surface contact area to minimize the need for extensive tests to develop the membership functions for the threshold levels.

Update

Due to the success of oil debris analysis in predicting damage on the spur gear fatigue rigs, an oil debris sensor was installed on the NASA spiral bevel gear test facility, and further tests were run. Details of that research are found in the report "Spiral Bevel Gear Damage Detection Using Decision Fusion Analysis," available at *www.grc.nasa.gov*. ○

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The Alignment of High Speed Gears

Kevan Whittle

Synopsis

This paper reviews the necessity for detailed specification, design and manufacture to achieve required performance in service. The precise definition of duty rating and a thorough understanding of the environmental conditions, whether it is in a marine or industrial application, is required to predict reliable performance of a gearbox through its service life. A case study relating to complex marine gears and other general practice is presented to review the techniques used by Allen Gears to design and develop a gearbox that integrates with the requirements of the whole machinery installation. Allen Gears has considerable experience in the design of a variety of industrial and marine gears (Ref. 1, 2). The requirements of different types of installations are reviewed to study the implications on gear alignment. Particular types of gearboxes have been developed in recent years to achieve more accurate alignment while also reducing the size and cost of a drive system.

Photo 1—Fast patrol boat for Royal Navy of Norway.

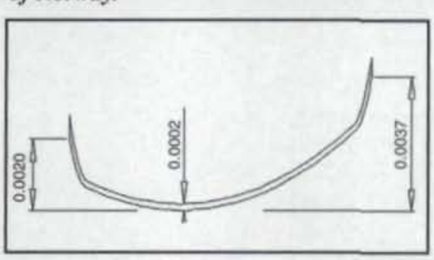

Figure 1—An overlay chart and the level of accuracy required from the form gear grinder.

Introduction

The design of gearboxes for marine and industrial applications requires exhaustive techniques to assure the mechanical integrity of rotating shafts and bearings for transmitting the duty power. The same magnitude of resources is now applied to understanding structural displacements and the consequent effect on gear mesh misalignment and stress overload factors.

Project design activity has increased to allow more detailed understanding of how the gearbox performance is affected by the supporting and connecting equipment and the applied external loading. Modern analysis techniques allow for very detailed modeling of the gearbox and propulsion system by including all relevant equipment from the gas turbine to the water jet using finite element analysis. FEA models give displacements of all gearbox flanges and bearing support blocks, as well as stress distribution and system natural frequencies. This information is an advantage to the gear designer because of the increased understanding of various factors internal and external to the gearbox that can influence gear alignment. Confidence that the gear will operate satisfactorily at the rated power for the required life reduces risk for the gear manufacturer and the end user.

Gearboxes for High Speed Craft

The experience of marine gearing within the author's company in recent years has been related to high-speed, light craft for a variety of applications, including naval patrol boats and high-speed, luxury yachts. Gearboxes for high-speed patrol boats in monohull, surface effect ship and hovercraft have been designed by Allen Gears for combinations of gas turbines (Rolls-Royce Allison, Pratt & Whitney and General Electric Co.), diesel engines and hydraulic motors driving water jets or propellers for main propulsion. Installations have used gas turbines with rated powers up to 10,000 kW and input speeds up to 16,000 rpm, with vessel speeds up to 60–70 knots.

The general configurations of gearbox design have included CODOG (COmbined Diesel Or Gas turbine) and single-input, single-output in "c" or "z" layouts. Recent gearbox designs have been carried out to satisfy the requirements of Det Norske Veritas (DNV) rules for high speed, light craft. DNV is an organization that produces rules for design of equipment, including ship propulsion systems.

The gear teeth are rated in accordance with DNV Calculation Note 41.2 (generally based on the requirements of ISO 6336). The design rules extend to the mechanical strength at full-load torque, shock loading, fatigue loading due to water jet aeration and the assessment of gear alignment in the extremes of operation. The classification rules give nominal magnitudes of shock for a specific duration for passenger, cargo or patrol-and-rescue craft, the highest being 69 m.s-2, or an acceleration of 60 m/s/s, for a duration of 0.050 seconds. This shock level is attributed to vertical slamming relative to the type of vessel in sea conditions.

Loading on the propulsion system can also include vertical, transverse and longitudinal accelerations for a particular number of cycles as deemed appropriate to the design of a vessel.

Gear Design and Alignment

Naval marine gearboxes are generally designed to be lightweight while having onerous requirements for optimizing tooth loading and gear alignment. It is not proposed to give extensive details of the design of such a gearbox, but to give an overview and discuss areas that influence gear tooth alignment.

A general configuration uses an aluminium-fabricated gearcase with horizontal and vertical joints to allow for ease of separation and support lines of gas turbine input, intermediate shafts, diesel input and water jet output shafts. The gearshafts can be supported by a combination of rolling element and hydrodynamic bearings, both of which can be designed to provide a means of adjusting gear alignment. The CODOG gearboxes have automatic changeover from gas turbine to diesel drive, which is achieved by self-synchronizing clutches and a multiplate clutch with two-stage pressure engagement (Ref. 3). The gearboxes designed for these applications generally have integral lubricating oil systems and become complex with the extent of pipework, pumping, cooling and control hardware to satisfy space restrictions.

Gear elements are designed and manufactured in either case-carburized or gas-nitrided steels for the primary and secondary reduction-gear meshes. A review of a particular gearbox is provided later where the gears have been designed to DNV rules, and a number of specific observations will be presented relating to this process. Rating of the gear teeth to DNV complies with limits that Allen Gears have historically specified in terms of surface load and specific bending, but with some exceptions. The main differences observed using DNV is in the scuffing capacity and the requirement for a greater effective case depth (at 400 HV) than would be required when rating a nitrided gear to other rules.

As is common with all manufacturers of high-speed and loaded gears, the predictions of structural displacements are used in the design of gear tooth grinding reliefs. As stated, the magnitude of distortions can be predicted by finite element modeling of the gearcase assembly, with loads applied to represent the full range of operating conditions. The predicted movements of gear-shaft bearing housings can be directly related to gear alignment, and helix-angle corrections can be calculated.

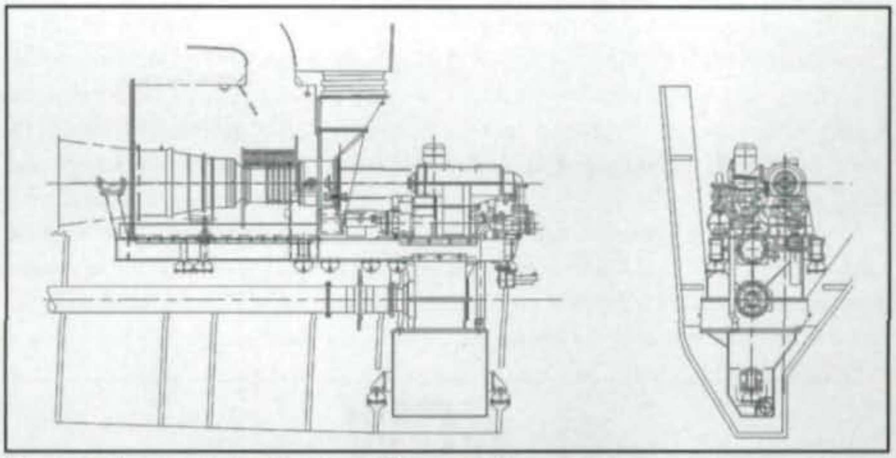

Figure 2—A general elevation view of the propulsion system.

The finite element analysis of the gearbox and support system is carried out for a number of combinations of the loading conditions. At one end of the range would be self-weight plus transmitted torque, while at the other end would be these two cases plus all the maximum sea conditions, including slamming. This ensures the gearcase and support system are subjected to the range of loading likely to be experienced under service conditions.

From each of the loading cases, the gearcase deflections at all of the bearing positions are extracted from the finite element results. From these deflections, the slope of each shaft line along the line of action of each gear mesh is established. These slope values indicate the misalignment due to loading of each mesh within the gearbox. Without helix modification to the gear teeth, these misalignments will cause high overloads at the ends of the teeth and hence premature failure in these areas.

In order to reduce these high overloads, two forms of longitudinal correction are applied to the teeth at each mesh in the gearbox. One is a fixed change of helix angle, "torsional correction," to take account of the fixed amount of misalignment occurring in the same direction for all loading cases. The second correction is crowning, which will take account of those misalignments that can cause tooth overloads at either end of the face width.

Once these longitudinal corrections have been established, they are combined with any end relief and used to calculate a set of coordinate values that are suitable for input to the computer-controlled form gear grinder. This ensures minimal deviation from the design intent to that manufactured and removes an element of possible influence on gear alignment. An example of the overlay chart and the accuracy required is shown in

Kevan Whittle

is chief engineer at Allen Gears, within the industrial business section of Rolls-Royce, managing the product design and development of low and high speed gearboxes for industrial and marine applications. Whittle is a chartered mechanical engineer trained with Eaton Ltd.'s transmission division. He later worked for Vickers Shipbuilding and Engineering as a design engineer on submarine main propulsion gearing, performing design studies and supporting gearboxes on test and in service with the Royal Navy.

Figure 1. The capability of the form grinding machine tool gives the greatest possible control to minimize manufacturing errors, with a process capable of machine-grinding to master gear quality and helix alignment of around 0.005 mm.

General industrial gears are manufactured to ISO 1328 Grade 4 with machine adjustments being made to keep profile and helix errors within specified tolerances. Allen gears are manufactured to ISO 1328 Grade 4 (AGMA Grade 12).

The alignment of the teeth is set by initially meshing the gear elements in the gear case while supported by accurately machined low-clearance bearings and then grinding the gear teeth to have conforming helix angles. This sets the static gear alignment and corrects for any errors resulting from the manufacture of the gears, case and bearings.

Static alignment is verified by rotating the gears slowly and visually observing the contact pattern between the meshing teeth. Proof of the mesh is obtained by witnessing the contact markings resulting from blue lacquer transferred from pinion to wheel, the thickness of lacquer being around 0.003 mm.

The gear tooth reliefs are subsequently applied and include end relief, helix-angle correction, and crowning. A static misalignment is consequently introduced in the gear mesh but ensures that, when operating at full power, the teeth will be aligned and within design limits.

Mounting System and FEA Model

The drive configuration and model is of a gearbox, gas turbine, gas turbine enclosure, support frame and resilient shock mounts. The model of the propulsion package included the connections between the support frame and the hull mounting points, which spanned some four separate lateral bulkhead structures manufactured from a sandwich composite material. A sandwich composite consists of two glass-reinforced plastic (GRP) plates with a compound "sandwiched" in between. Detailed finite element modeling of the system would prove the integrity of the propulsion system under severe load conditions expected at sea. A general elevation view of the machinery is shown in Figure 2.

A model can be created using, for example, version 5.3 of the ANSYS suite of software. Elastic straight pipe elements and 3-D beam elements are combined to create the gearcase, or the structure that holds the gears and bearings in place, as well as to develop shafts and system connections. To correctly model the total mass at a particular center of gravity, structural mass elements are used. The gearbox and gas turbine are bolted to the mounting frame while a flexible coupling connects the two assemblies. In the model, the gearbox and gas turbine are rigidly connected to a support frame by a system of constraint equations, the frame being a combination of four- and eight-node shell elements. It is considered that, to get the best indication of final gear alignment, the total system must be modeled and therefore includes the ship's hull, which was represented by beam elements with three translational and three rotational degrees of freedom, values of which were supplied by the naval architects. An isometric view of the model is shown in Figure 3.

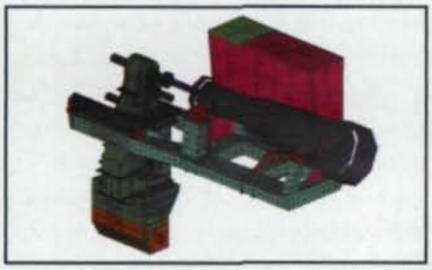

Figure 3—An isometric model of the total propulsion system using version 5.3 of the ANSYS software suite.

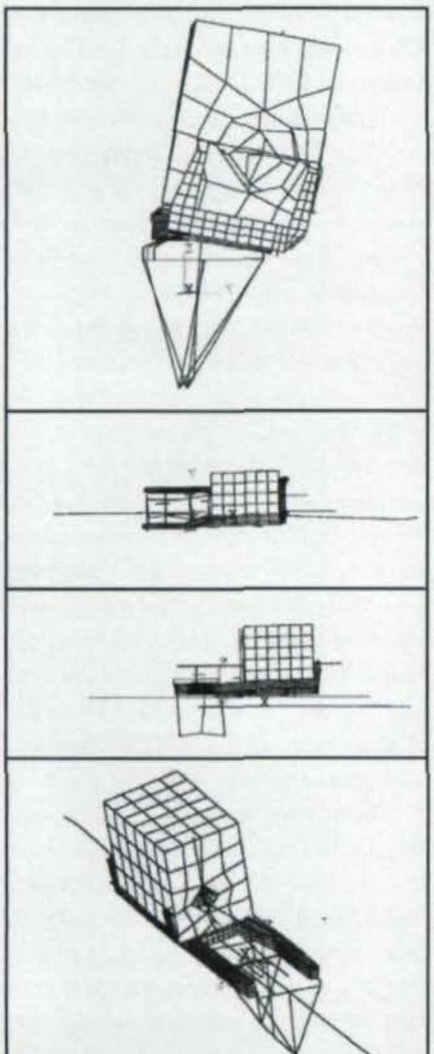

Figure 4—Model analysis of integrated gas turbine propulsion system.

Modeling work was conducted in parallel with the gearbox detail design activity and provided detailed requirements for methods of supporting the gearcase to achieve the necessary alignment of the gear to the gas turbine. The support of the propulsion unit was through 16 rubber mounts, which were modeled as unidirectional nonlinear spring elements to reproduce the force deflection curves specified. The resilient mount elements had a range of stiffness and dynamic magnifiers to allow the support frame to deflect in a controlled manner.

Analysis of the propulsion system model gave the following outputs:

a) dynamic behavior of the assembly to allow correct positioning of support structures, shown in Figure 4;

b) stresses and deflections of the gearcase at the gear-element bearing supports;

c) angular misalignment of shaft couplings;

d) natural frequencies of the propulsion package; and

e) maximum displacements of the assembly under shock loading.

This extensive analysis work allows the gear designer to gain detailed specifications on conditions of operation and provides data on the worst possible operating mode for the gear tooth alignment. This provides the means to optimize the gearcase design and stiffness and to minimize weight.

DNV Rating

Allen Gears has a long history of designing and rating gear teeth to a range of international standards, including AGMA 2001, American Petroleum Industries (API) 613, Lloyd's Rules, which involves the design of gearboxes for industrial machinery, and more recently DNV Classification Note 41.2 (Ref. 4). Most of the above calculations are performed using in-house computerized Fortran routines and are interactive programs that allow the user to evaluate various design options rather than simply rate a gear pair. The programs allow designers to have a baseline gear pair automatically selected based on a particular input power, speed, ratio, etc. Also, the programs give options to change any feature and have remaining features recalculated instantly. The programs calculate the gear service factors and data to define the salient points of the gearbox design that are suitable for supplying to a customer and to create manufacturing drawings.

DNV rating of gear teeth allows for finite and infinite life assessments of a gear pair for one or more load cases and gives service factors for contact, bending and scuffing resistance. As previously discussed, there are a number of points that are worthy of further discussion, including calculation of case depth "size factor" for

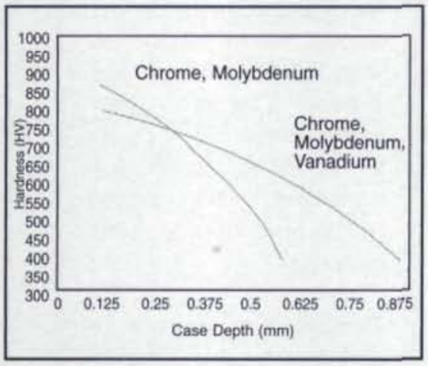

Figure 5—The difference between actual case depth profile for a 3% chromium-molybdenum nitriding steel and a 1.4% chrome-molybdenum-vanadium steel.

nitrided gears. Scuffing assessments for flash and bulk temperatures' tooth overload factors, which relates to the prediction of scuffing between two meshing gears also should be looked into further.

The size factor Zx is particularly onerous for the rating of nitrided gears. Nitrided gear tooth permissible stresses can be reduced by approximately 40%

when

$$Zx = \left(\frac{30t_{400}}{\rho_c} \right)^{0.4} \frac{1200}{\sigma H \lim}$$

Suitable materials exist that produce a case depth versus hardness curve that minimizes the effect of this penalty to around 30%. However, it is not Allen Gears' experience that nitrided gears have such a poor surface fatigue strength.

Comparisons of the actual case depth profile for a 3% chromium-molybdenum nitriding steel and a 1.4% chrome-molybdenum-vanadium steel is shown in Figure 5 to demonstrate the increased depth achieved at 0.7 mm at 400 HV. Under most circumstances, the designer would endeavor to choose carburizing steels where the case depths achieved easily produce a factor $Zx = 1$. This does, however, introduce limitations on the construction of large gear wheels and compromise the objective of lightweight construction.

Scuffing assessment to DNV consists of integral and flash temperature calculations. To achieve satisfactory safety margins, the specified oil load carrying capacity must be around FZG 12. To stop asperity contact, the oils must provide a film thickness between two gear flanks.

Oils with a low load capacity are likely to suffer from pitting and scuffing. With mineral oils having a viscosity grade of ISO VG 32 or 46, it becomes difficult to procure oils with the required FZG capacity. Allen Gears have specific rules on allowable flash temperatures and have had virtually no failures through scuffing. The requirements of DNV are onerous and make Allen Gears' current limits and practice unacceptable.

Alignment of Industrial Gearboxes

Alignment of gear teeth and the resulting overloads across the gear face widths in parallel shaft gears is controlled by the structural integrity. Loading on input and output shaft bearings is influenced by the stiffness and alignment capability of the connecting shafts and couplings, and the modeling work carried out on marine gears is vital to verify the integrity of the high speed couplings and also to prevent limits being exceeded on the connecting equipment's shaft bearings. The particular case study discussed in this paper used a membrane coupling with a length of 870 mm and had finite limits on axial, lateral and angular misalignment. The FEA model confirmed that these would not be exceeded in service.

The requirement to accurately align the gearbox to connecting equipment is key to satisfying gear tooth alignment and is more critical on some epicyclic gear products depending on coupling design features. The general configuration of a single-reduction star arrangement is shown in Figure 6. The gear cluster can be configured into a range of gearcases to provide free-standing,

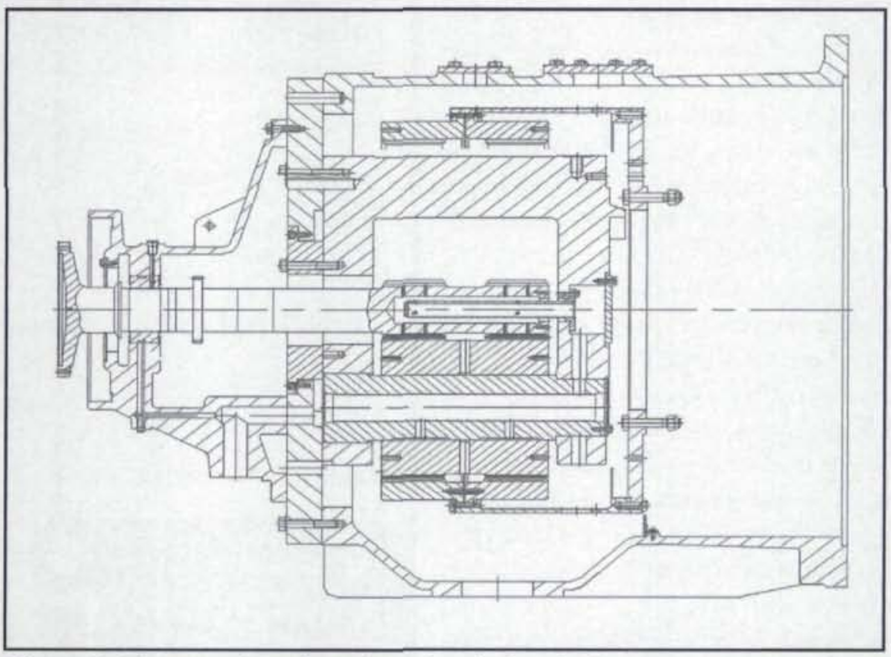

Figure 6—The general configuration of a single-reduction star arrangement in an epicyclic gear product.

engine-mounted or generator-mounted arrangements.

Figure 6 shows the generator-mounted gearbox where alignment of the gears is obtained by accurately locating the gear onto the driven (generator) shaft. This provides benefits in the package efficiency where there are no low-speed shaft bearings and benefits by reducing the gearbox length, cost and weight. Alignment of the input and output shafts is critical to control misalignment of the gear teeth and is achieved by accurate machining, assembly and measurement during the installation procedure. Any errors in alignment of the gearcase when attached to the generator will transfer through to the high-speed shaft via the gear mesh.

Accurate high-speed shaft alignment was traditionally achieved by manual methods using dial indicators, and the process could take approximately 24 hours. Indications would be that the equipment was in alignment, but the setting was carried out with a gearbox having no installed gears or bearings. The dial clock was fitted to the driven (output) equipment shaft and then checked to the gearcase driver (input) end and adjustments made. Consequently, when the gears were fitted, an alignment error was introduced due to the additional overhung mass.

Current techniques allow the equipment to be aligned in eight hours using electronic methods. Displacement probes can be fitted to the fully assembled gearbox. The gearbox is then driven, with outputs from the probes being processed by a personal computer to calculate misalignment and any necessary adjustments.

The customer support team within Allen Gears is regularly involved in the installation of gears and have a direct contribution to gearboxes in a variety of applications, from large water turbines to small high-gas turbines. In marine propulsion, the technology for aligning parallel-shaft gears and couplings has advanced in recent years with the use of laser alignment, which is accurate to 0.002 mm across distances of 5 m.

Concluding Remarks

Complex gearcases and shaft arrangements can now be designed and analyzed using FEA techniques to gain greater assurances on the safe performance of the system under load. Industrial gearboxes tend to be designed around a general type of configuration where structural integrity is known and is easily predicted. Also, the designs have evolved in recent years to allow for easier installation, and equipment/methods have also changed to improve the accuracy of the installation.

Calculations and predictions of alignment are critical to the validation of a gear design, and access to information on structural stiffness is key to determining the magnitude of the face overload factor and is easily extracted from an FEA model. Customers are demanding this level of analysis to lower the risk associated with main propulsion gearboxes.

The rating of gear teeth to DNV marine requirements can, in some circumstances, result in a larger gear pair than would be required to satisfy other standards. Current work within the British Gear Association involves researching the technology required to enhance material and lubrication technology to enable the optimum design to be obtained and to address this problem. O

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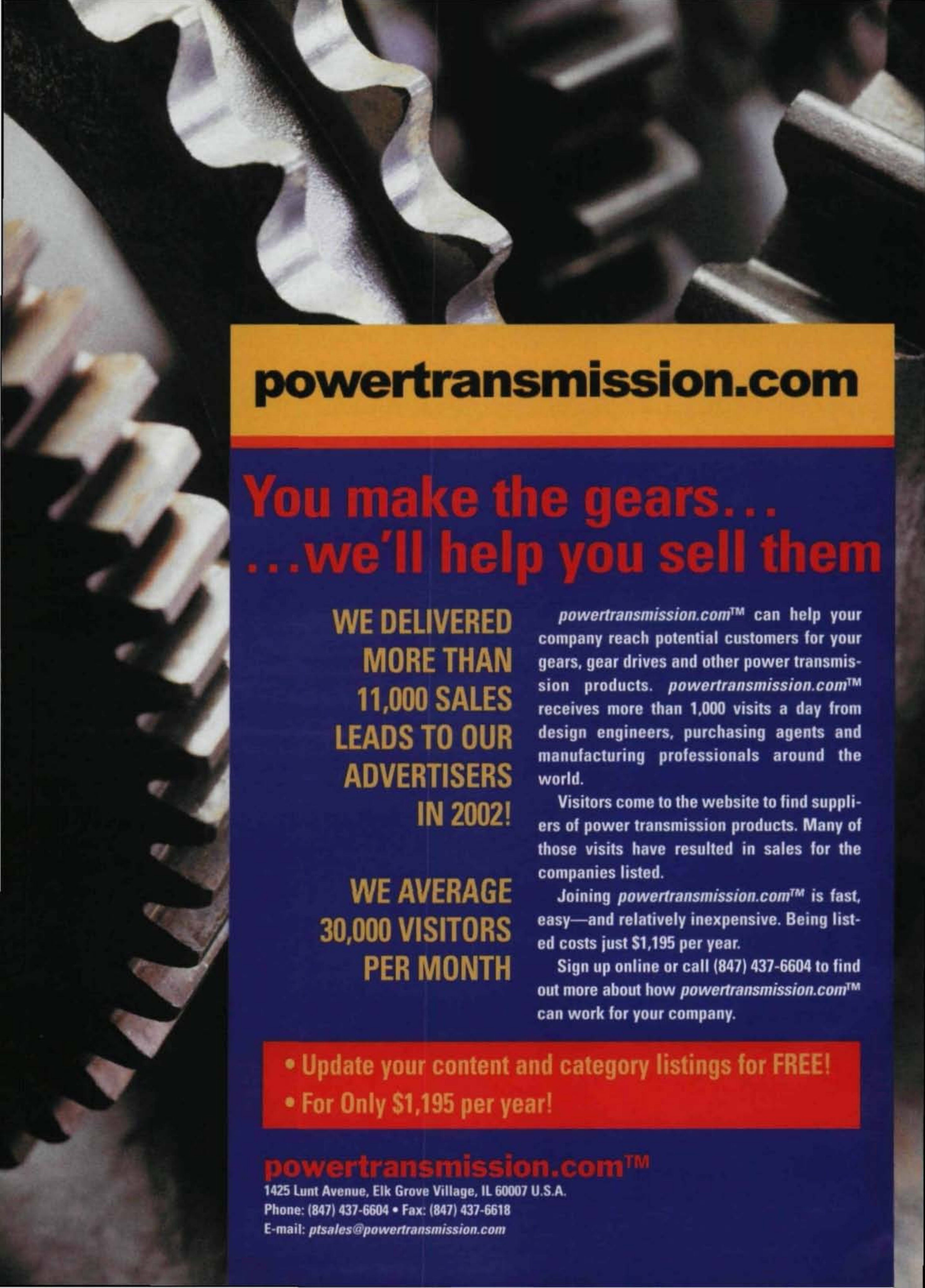
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Experimental Characterization of Bending Fatigue Strength in Gear Teeth

S.B. Rao and D.R. McPherson

Introduction

The effort described in this paper addresses a desire in the gear industry to increase power densities and reduce costs of geared transmissions. To achieve these objectives, new materials and manufacturing processes, utilized in the fabrication of gears, are being evaluated. In this effort, the first priority is to compare the performance of gears fabricated using these new alloys and processes with those fabricated using current materials and processes. However, once that priority is satisfied, it rapidly transforms to requiring accurate design data to utilize these novel materials and processes. This paper describes the effort to address one aspect of this design data requirement.

One of the modes of failure of a gear tooth results from breakage in the root fillet area. While sudden overloading (impact) can precipitate this type of failure, it usually occurs in practice due to bending fatigue. While consideration of sudden overloads is important in the design of gears, it is not the topic of this paper.

This article deals with bending fatigue failures in gear teeth. It describes the current method of experimentally characterizing bending strength and the deficiencies of this method. The paper also discusses an alternate approach being developed to address those deficiencies and to obtain and disseminate more accurate data characterizing bending fatigue strength.

Bending Fatigue

The cyclical nature of the loading of gear teeth in a transmission is the cause of bending fatigue. The origins of bending fatigue failures typically are imperfections in the surface of the root fillet (e.g., tooling "witness" marks) or nonmetallic inclusions near the surface. Cracks slowly propagate around the origin until the damaged area reaches the critical size for the case material at the prevailing stress level. For hardened, high-carbon material typically used for gears, this critical size is so small that cracks at this stage are very difficult to detect. When the crack reaches the critical size, it "pops" through the case (i.e., the entire case fractures in one or a few cycles). At this point, the rigidity of the tooth is reduced (compliance increases) and transmission error increases significantly. This produces a readily detectable increase in noise and vibration, and represents failure.

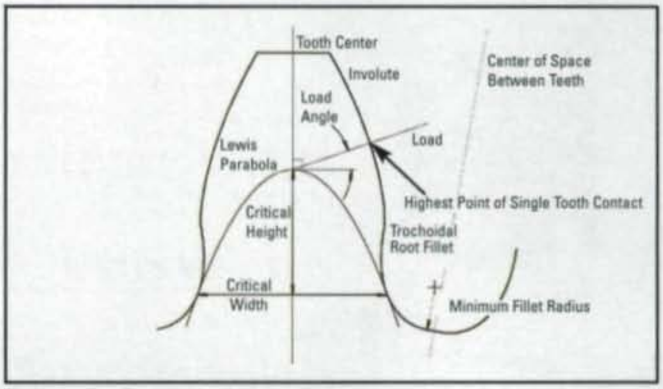

Figure 1—Bending stress calculation for spur gear.

While each situation can be different, in most instances, there is enough time between the onset of increased vibration and catastrophic failure for a vibration monitoring system to give sufficient warning to permit an orderly shutdown of the equipment. The mechanism that allows this is the reduced compliance of the cracked tooth, which transfers some of the load to adjacent teeth. The lower load, and the lower hardness of the core, results in slower crack propagation. This allows a brief interval between the occurrence of detectable cracking and fracture of the tooth. This feature notwithstanding, tooth fracture is the most catastrophic form of gear failure, and a substantial portion of gear test programs are dedicated to obtaining sufficient data to minimize its occurrence in service.

Dr. Suren B. Rao

is a senior scientist at the Applied Research Laboratory of The Pennsylvania State University in State College, PA, and managing director of the Gear Research Institute, which is located at the university. His primary research interests are in the area of manufacturing and materials related to mechanical power transmission components and systems.

Douglas R. McPherson

is a research engineer in the Drivetrain Technology Center at the Applied Research Laboratory of The Pennsylvania State University. His research interests include performance characterization of material systems for gearing, contact fatigue, surface topography characterization, bending fatigue, gear fatigue and gear manufacturing.

Bending Stress Computations for Root Fillets

Bending stresses are computed based on the assumption that the gear tooth is a cantilever beam with a stress concentration at its supported end. AGMA rating standards determine the form of the cantilever beam from the solution presented by Lewis, and use a corresponding stress concentration factor. ISO (and DIN) standards use different proportions for the beam and determine the stress concentration factor in a correspondingly different manner. Only the AGMA approach will be discussed here.

Figure 1 shows a spur gear tooth with a point load applied at the highest point of single tooth contact. This point of loading corresponds to the highest bending stress when there is effective load sharing between gear teeth. Specimen gears used in rig tests should have effective load sharing, so this is the appropriate point of loading for determining bending stress in rig tests. For gears tested in single-tooth bending fatigue, the actual point of loading established by the test fixture should be used in calculating bending stresses.

The Lewis parabola is drawn from the point the load line intersects the center of the gear tooth and is tangent to the root fillet. The methods used to lay out this parabola vary depending on how the root form is generated, and the full particulars are lengthy and presented in detail elsewhere (Ref. 1). The critical height and width are determined from the Lewis parabola as shown on Figure 1. The angle between the load line and a normal to the tooth center is termed the load angle (it differs from the pressure angle at the point of loading because of the thickness of the tooth). The bending stress is thus:

$$\text{Bending Stress} = \frac{\text{Load} \cdot \cos\,(\text{Load Angle})}{\text{Face Width}} \cdot \left[\frac{6 \cdot h}{s^2} - \frac{\tan\,(\text{Load Angle})}{s} \right] K_f$$

s = Critical Width from Lewis Parabola
h = Critical Height from Lewis Parabola
K_f = Stress Concentration Factor = $H + \left(\frac{s}{r}\right)^L \left(\frac{s}{h}\right)^M$
r = Minimum Fillet Radius
$H = 0.331 - 0.436 \cdot$ (Nominal Pressure Angle – Radians)
$L = 0.324 - 0.492 \cdot$ (Nominal Pressure Angle – Radians)
$M = 0.261 + 0.545 \cdot$ (Nominal Pressure Angle – Radians)

This equation for bending stress can be derived from first principles or from AGMA standards by taking the forms of relevant formulas pertinent to spur gears and setting all design factors at unity. A similar formula can be developed for helical gears.

Single-Tooth Fatigue (STF) Test

The single-tooth fatigue test is used to generate a statistically significant quantity of bending fatigue data at a comparatively low price. Teeth are tested one at a time with a fixed loading point. Consequently, failure will not occur via the other mechanisms (scuffing, pitting, wear, etc.) that affect running gears. This allows generation of bending fatigue data at comparatively high cycles without risk of losing tests to other modes of failure. Another cost-saving measure is that four or more tests can be conducted with each gear specimen.

Test Equipment. A gear is placed in a fixture so that one tooth at a time can be loaded while another tooth supports the reaction. The test is usually done in an electrohydraulic, servo-controlled universal test machine. The primary object of this test is to determine fatigue properties in bending. However, the same setup can be used to determine single overload properties (ultimate bending strength) as well. Frequently, enough teeth are tested to develop a stress-cycle diagram to define the bending fatigue characteristics of the material system.

Several arrangements for loading can be considered. One fixture arrangement is illustrated in Figure 2. This shows The Boeing Co. flexural design, which appears to have found favor with the aerospace sector. This fixture is designed for a 32-tooth, 5.333 DP, 3/8" face width spur gear with several teeth removed to provide access to test and reaction teeth. The gear is rigidly supported on a shaft. Load is applied through a carbide block contacting the test tooth at the highest point of single tooth contact. The loading block is held in the specified orientation to the gear by a flexural loading arm. This flexural design ensures accurate loading of the gear tooth with minimal migration of the point of loading. Reaction is carried through a block contacting the reaction tooth at the lowest point of single tooth contact. Load is cycled from the specified test load to a minimum load high enough to keep the slack in the system taken up (usually 10% of the test load). While most testing is conducted at 20 Hz, other frequencies are also possible. The fatigue test machine is instrumented to monitor instantaneous loads and tooth deflections. Changes in compliance can be utilized for monitoring crack initiation and propagation in the root fillet region. In addition, a crack wire can be incorporated to monitor catastrophic tooth failure. Typical fatigue load capacity of such types of equipment is in the range of 10,000–20,000 lbs., although higher loads, up to 110,000 lbs., can be used for single overload tests.

A second fixture arrangement is illustrated in Figure 3, and it appears to have found favor with many other industry segments. This fixture utilizes a 34-tooth, 6 DP, 1" face width spur gear and is derived

Figure 2—Boeing-type STF rig.

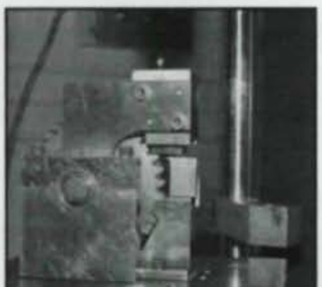

Figure 3—Gear Research Institute-type STF rig.

Table I—Test Data

Legend X – Failure; 0 – Run Out

Specimen Serial Numbers 9, 10, and 11. Specimens cut from bar stock, hobbed roots. R loading = 0.1, 25 Hertz.

Gear/Tooth	9/1	10/1	11/1	9/2	10/2	11/2	9/3	10/3	11/3	9/4	10/4	11/4	9/5	10/5	11/5	11/6	11/7	9/6	10/6	10/7	9/7	10/8				
Life, Cycles x 10^6	Run Out	0.126	Run Out	0.133	0.170	Run Out	0.196	0.241	Run Out	0.073	0.167	Run Out	Run Out	0.091	0.275	0.132	0.138	0.156	0.404	Run Out	Run Out	0.068				
Load	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	Failure Rate
10,500 lbs.																										
10,000 lbs.																										
9,500 lbs.		X		X										X		X	X	X								100%
9,000 lbs.			0		X		X				X		0		X											67%
8,500 lbs.	0					0		X		X		0										X				50%
8,000 lbs.									0										X		0					33%
7,500 lbs.																				0						0%
7,000 lbs.																										
6,500 lbs.																										
Load	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	Failure Rate

Searching Tests — Modified Staircase Test Sequence — Finite Life and Confirmation Tests

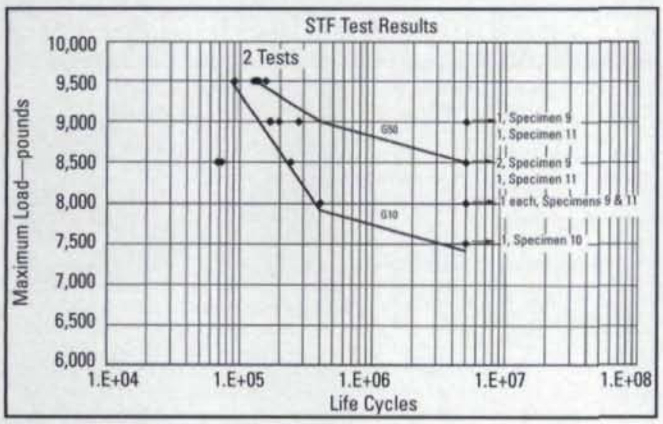

Figure 4—Load-cycle diagram from STF data.

directly from the SAE Division 33 STF fixture. Fatigue test loads up to about 15,000 lbs. are feasible with this fixture, and single overload tests up to about 50,000 lbs. can be accommodated. These STF fixtures are compact enough to be immersed in heated fluid; thus, fatigue testing can be conducted at elevated temperatures, up to 400°F.

Specimen Results. Table I summarizes results from a typical set of STF tests. Testing was conducted in three phases. Initial searching tests were conducted to establish loads that would result in failure in reasonable time. A "modified staircase sequence" of tests was conducted to develop data at a series of loads representing 0–100% failure. Further tests were conducted to fill in the stress-cycle relationship. Searching tests are started at a high load to ensure a failure and then stepped down until the tooth survives the specified number of cycles (here 5 million cycles has been selected as a run out limit). The modified staircase sequence is conducted by testing three specimen gears in sequence. If the tested tooth breaks before the specified limit, the next test is conducted one load step lower; if it doesn't break by the specified limit, the next test is conducted one load step higher. After the modified staircase sequence is completed, additional tests are conducted to ensure that all the specimen gears are tested at the lowest load. More tests are conducted to develop enough data for Weibull analysis at two loads resulting in 100% failure.

The load-cycle diagram shown in Figure 4 was developed from the data in Table I. Results at 9,000 lbs. and 9,500 lbs. were analyzed via Weibull statistical analyses to determine lives to 10%, 50% and 90% failure. The failure rates at 5 million cycles for loads from 7,500–9,500 lbs. were analyzed using normal probability concepts to determine 10%, 50% and 90% failure loads. The curves labeled G10 and G50 were then fit "by eye" using the results of these analyses as a guide. Results are reported in terms of load vs. cycles, and load can be converted to stress using the method discussed previously. Comparisons can be made between groups of gears with the same geometry on a load-cycle basis, or between gears with differing geometry on a stress-cycle basis.

Relationship of STF Data to Bending Strength of Running Gear Teeth

The item of interest to the design community is allowable bending stress for running gears rather than STF strength. The Gear Research Institute has developed a method to translate STF results to be comparable to stress results from running gears. When translating STF data to be comparable to stress for running gears, several considerations become significant. While STF data is based on breakage of several teeth on one gear, breakage of one tooth on a running gear constitutes failure. Consequently, consideration must be given to the statistical difference between four, eight, or more data points from a single STF specimen gear compared to one data point from a running gear specimen set. For example, considering a running gear with 18 teeth, 50% failure corresponds to one failure in 36 teeth tested, and 10% failure corresponds to one failure in 180 teeth tested. Further, in an STF test, loading is varied from 10–100% of the maximum load. In running gears, this cyclical loading varies from 0–100%, consequently STF data has to be adjusted for this difference.

One method to translate STF data to be comparable to bending stress data on running gears was proposed by the Gear Research Institute and is described in detail in Reference 2. It is briefly discussed here to explain its complexity. Though reasonable correlation between the proposed method and experimental data was obtained, the methodology has deficiencies, as will become apparent as it is presented.

Figure 5 shows the normal probability variant (NPV) plotted for STF data obtained at various maximum applied loads. The NPV is found in probability tables, such as those in Reference 3, which are based on the failure rate obtained at the specific load in the STF tests. Also plotted on Figure 5 are the "Mean" and "Conservative" fit lines for the load vs. failure rate, the proce-

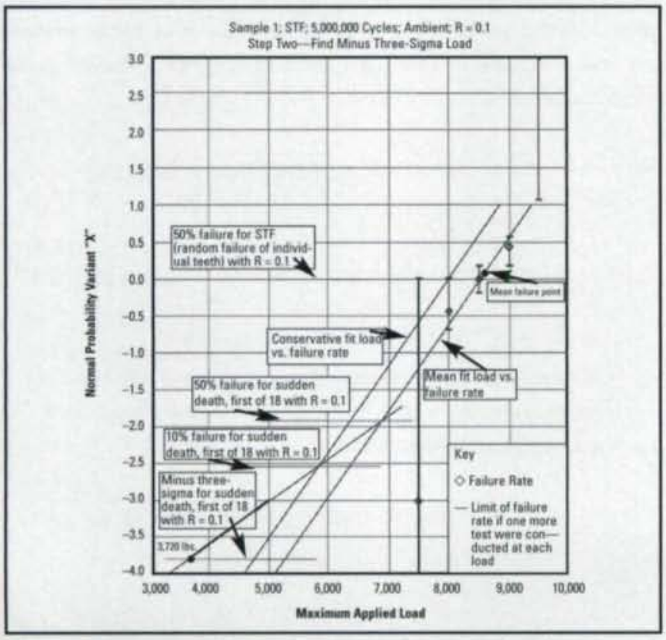

Figure 5—Maximum applied load vs. normal probability variant (NPV).

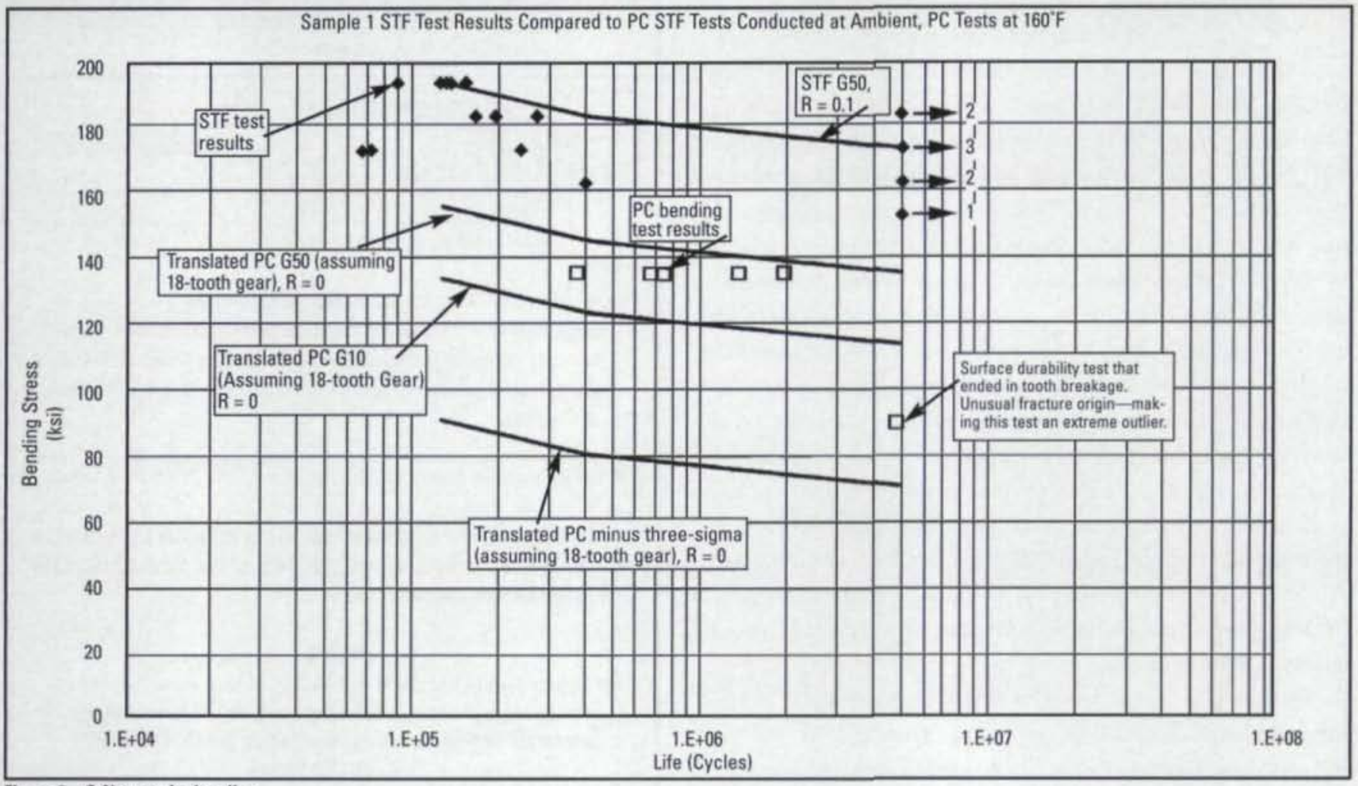

Figure 6—S-N curve for bending.

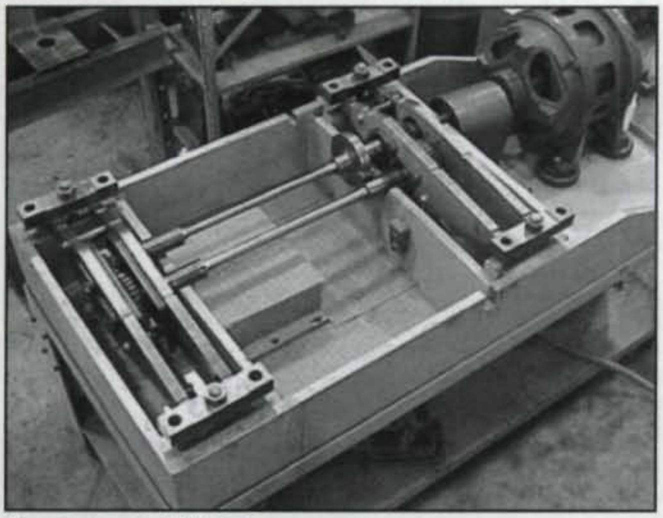

Figure 7—Low-speed PC test rig.

dure and logic for which are covered in Reference 2.

The power recirculating (PC) specimen gears that have been used to experimentally obtain bending stress data each had 18 teeth. Thus, 50% failure with PC specimen gears corresponds to one failure in 36 teeth tested, and 10% failure with PC specimen gears corresponds to one failure in 180 teeth tested. The normal probability variant for one failure in 36 pieces tested is –1.9145 (from probability tables such as those in Ref. 3), and that for one failure in 180 pieces tested is –2.5392. The load corresponding to 50% failure with PC gears is taken from the mean fit line at NPV = –1.9145, and that corresponding to 10% failures is taken from the conservative fit line at NPV = –2.5392. The aerospace community uses minus three-sigma (one failure in 840.84 parts) for the design bending strength curve. For 18-tooth gears, minus three-sigma corresponds to one failure in 13,333 teeth tested. The normal probability variant for this condition is –3.69. The load corresponding to minus three-sigma is found by drawing a line through the loads selected for 10% and 50% failure with PC gears described above, and picking off the value at NPV = –3.69.

In the STF test, the load is varied from 10% to 100% of the maximum load. These R = 0.1 stresses are converted to R = 0 stresses via allowable stress range (ASR) diagrams. The ASR diagrams are constructed to be representative of brittle materials following the method described in Reference 4. The pertinent equations are as follows:

$$\sigma_A = \text{Alternating Stress} = \frac{\text{Maximum Stress} - \text{Minimum Stress}}{2}$$

$$\sigma_M = \text{Mean Stress} = \frac{\text{Maximum Stress} + \text{Minimum Stress}}{2}$$

σ_U = Ultimate Stress (Ultimate stress is taken as the bending stress corresponding to the linear deviation point load from the fast bend single overload test.)

$$Y = \frac{1 - \dfrac{\sigma_M}{\sigma_U}}{1 + \dfrac{\sigma_M}{\sigma_U}}$$

$$\sigma_R = \text{Fully Reversed Stress} = \frac{\sigma_A}{Y}$$

These equations can be algebraically manipulated to yield the following expression for R = 0 stress:

$$\sigma_{R=0} = [(\sigma_U + \sigma_R)^2 + 4 \bullet \sigma_U \bullet \sigma_R]^{1/2} - \sigma_U - \sigma_R$$

Figure 6 shows the results of the application of the above analysis method to STF data. This figure illustrates the stress-cycle diagram showing STF results, PC test results and "curves" for STF G50, PC G50, PC G10, and PC minus three-sigma. The STF G50 line is laid in by eye. The other lines are constructed by moving the STF G50 line the distances determined in the foregoing analysis. The experimentally obtained PC bending results fall very close to the translated PC G50 curve. (This particular data set was selected because it comprises the longest-cycle PC bending failure data in the Gear Research Institute's archives, giving a better comparison to the portion of the stress-cycle relationship best defined by the STF test.)

In spite of the reasonable correlation between estimated PC bending stress data from STF data in Figure 6, the extent of numerical manipulation proposed is a drawback of this methodology. Consequently, a more direct approach to obtaining this data is proposed.

Power Circulating Bending Fatigue Tests

The power circulating bending test eliminates the need for most of the statistical adjustment described above. It consists of a test gear and mate gear running in mesh, under load, in a power re-circulating (PC or sometimes referred to as a 4-square) test rig. A low-speed rig (rotational speed less than 1,000 revolutions per minute), such as the one shown in Figure 7, is preferred so that time is available to stop the rig and avoid damage to it at the occurrence of a tooth failure.

Running gears can fail via a number of modes, many of which are shown generically in Figure 8. Consequently the challenge in conducting successful PC (bending) testing is to design tests where the other modes of gear failure do not occur.

The Gear Research Institute has conducted power circulating bending fatigue tests with 6-pitch specimen gears for a number of years, and data has been generated with carburized steel gears up to 50% failure life of about 500,000 cycles without undue influence from the other failure modes. More recently, data has been developed with induction hardened steel gears up to 50% failure life

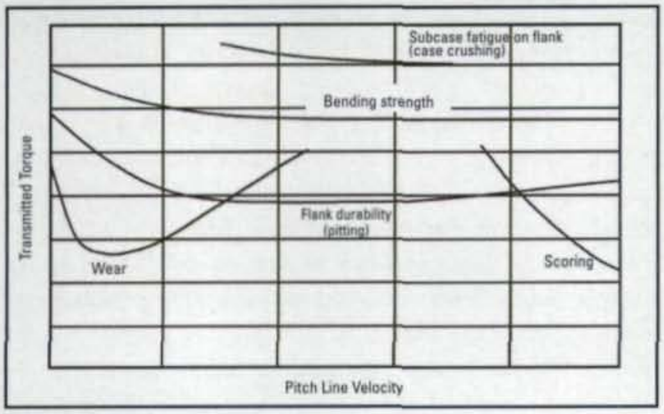

Figure 8—Gear failure map.

of 1 million cycles, again without undue influence from the other failure modes. To accomplish this, tests were conducted at high overload to promote bending failure before pitting would occur. Also, mate gears were finished with considerable (0.002") tip relief to avoid scoring. However, testing at high overloads at low speeds makes wear an endemic problem.

Further efforts to minimize wear and improve the accuracy of PC (bending) tests are ongoing. A brief break-in, starting with room temperature lubricant and reduced load, is conducted at the beginning of each test to reduce the ultimate wear rate. The test particulars in Table II show the current means of conducting PC (bending) tests with gear failures predominantly in the bending fatigue mode; however, it is still necessary to monitor wear.

It is desirable to develop PC bending data of up to 5 million cycles to 50% failure, to compare more directly to STF test results. This is planned with the use of finer pitch specimen gears to reduce the bending strength relative to surface durability, and finer surface finish and better break-in procedure to minimize wear. The case depth on the finer pitch specimens will be deeper than normal to avoid subcase fatigue below the contact surface. Such fatigue can be an issue with longer duration bending tests. In longer duration tests, subcase cracks will have time to propagate to the surface and potentially lead to untimely bending failure.

Conclusion

Single-tooth bending fatigue testing provides an inexpensive method to characterize bending performance of gears fabricated from new alloys using new manufacturing processes, but the needs of the design community for accurate design allowables have resulted in a critical examination of the method required to extract running gear bending performance predictions from single-tooth bending fatigue results. Based on the limited statistical reliability of the current method, a different approach, utilizing power circulating bending fatigue testing, is being evaluated. Initial efforts utilizing the power circulating bending fatigue method appear very encouraging, especially in the area of eliminating or minimizing the influence of other failure modes on the test. It is anticipated that, with further efforts, including the calibration of the rig under operating conditions, more accurate data characterizing the bending strength of gear teeth will be available to gear designers. O

Table II—Test Particulars, PC (Bending) Test.

Lubricant	MOBIL Synthetic Jet Oil II (MIL L-23699) (or as required for particular test program)		
Lubricant bulk temperature	140°F. The low-speed power circulating gear test rigs use splash lubrication. This bulk (sump) temperature is selected to emulate lubrication conditions in the 3.5" center distance high-speed test rigs where lubricant is sprayed onto the test gears at 115°F and reaches approximately 175°F before it is drained from the test gearbox.		
Lubricant change interval	2,000 hours (approximately)		
Lubricant filter	*10-micron ceramic filament*		
Specimen	18-tooth, 6-DP spur gear with 0.562" face width, as shown in Figure 4		
Mate	30-tooth, 6-DP spur gear with 0.87" face width, as shown in Figure 5		
Specimen Operating Speed	900 rpm (nominal)		
Velocities (on Specimen)		Rolling	Sliding
	Pitchline	48.4 in./sec.	0.0 in./sec.
	LPSTC	37.1 in./sec.	–18.0 in./sec.
	SAP	10.4 in./sec.	–60.7 in./sec.
Roll/Slide Ratio	LPSTC	–0.49	
	SAP	–5.81	
Run-in Procedure	Run one-half hour at one-half test load starting with room temperature lubricant.		
Test Loads	First test to be conducted at 6,500 lb.-in. (on specimen). This corresponds to approximately 150 ksi bending stress. Loads for subsequent tests will be determined based on the outcome of the first test, or according to project test plan.		
Run-Out	Tests will be suspended after 10 million cycles with no failure.		
Failure Criterion	Tooth Breakage or surface origin pits over 3/16" wide or 0.001" (approximately) profile change (see below) or progressive scoring or sub-case fatigue on flank that cracks through to a surface or severe vibration.		
Wear Monitoring	Due to the high loads and the rolling/sliding action of the gear mesh, material may be lost in the dedendum of the specimen. The condition of the gear contacting surfaces will be characterized visually at approximately 500,000 cycle intervals, and wear will be measured using a gage with a tooling ball sized to contact between gear teeth at the lowest point of single tooth contact. Tests will be stopped when wear reaches the point that load sharing between adjacent teeth is changed enough to skew bending fatigue results—this corresponds to approximately 0.001" loss of material in the dedendum of the specimen gear.		

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January 14–16—2003 Advanced Productivity Exposition. Orange County Convention Center, Orlando, FL. Sponsored by the Society of Manufacturing Engineers (SME), this event is Florida's largest machine tool and metalworking exposition. The exposition will also feature a series of half-day sessions focusing on lean manufacturing and Six Sigma. This exposition is free to anyone who registers on or before Dec. 21, $25 after Dec. 21. For more information, contact SME by telephone at (800) 733-4763 or on the Internet at *www.sme.org/orlando.*

February 11–12—Powder Metallurgy Value Seminars for Industrial OEMs. Hoeganaes Corp., Indianapolis, IN. Free seminars on the enhanced value of powder metal technology will be conducted at various locations by Hoeganaes Corp. The first day of the seminar will be devoted to the design of P/M parts and the second to case studies. For more information, contact Tim Hale via e-mail at *thale@hoegcorp.com.*

February 17–20—Falk Gear and Coupling Workshop. Falk Corp., New Berlin, WI. This workshop is a hands-on approach to working with Falk parallel, right-angle, concentric, Quadrive shaft-mounted and Omnibox worm gear drives. Successful graduates will qualify for workshop attendance certification. $1,495 includes tools, equipment and a daily luncheon. For more information, contact Falk Corp. by telephone at (262) 317-1428 or by e-mail at *renew@falkcorp.com.*

February 18–21—Basic Gear Fundamentals Course. Gleason Cutting Tools Corp., Loves Park, IL. This course is designed for individuals who are new to gear making and want a basic understanding of gear geometry, nomenclature, manufacturing and inspection. Additional content will focus on gear ratios, gear tooth systems and cutting methods. $895 includes a handbook, class materials, lunches and one group dinner. For more information, contact Gleason Cutting Tools by telephone at (815) 877-8900 or on the Internet at *www.gleason.com.*

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Spiral Bevel Gear Development: Eliminating Trial and Error with Computer Technology

Joseph L. Arvin and Thomas C. Mifflin, with contribution from James J. Cervinka

A New Era

Computer technology has touched all areas of our lives, impacting how we obtain airline tickets, purchase merchandise and receive medical advice. This transformation has had a vast influence on manufacturing as well, providing process improvements that lead to higher quality and lower costs. However, in the case of the gear industry, the critical process of tooth contact pattern development for spiral bevel gears remains relatively unchanged.

The procedures needed to develop spiral bevel gear sets for a new product can require months of trial-and-error work and thousands of dollars. In view of increasing global competition for lower-priced products, bevel gears are a prime target for the next generation of computerization. Answering this challenge, Arrow Gear Co. of Downers Grove, IL, has realized a new era through a shift in the way spiral bevel gear development is performed.

This article will provide some fundamental information pertaining to gear development and detail the procedures and techniques utilized by Arrow Gear to achieve maximum quality while substantially lowering development costs.

Understanding Contact Pattern and Gear Displacement

A critical attribute of a spiral bevel gear's design is its contact pattern. Simply stated, the contact pattern is the area in which the gear teeth come in contact as they engage and disengage during their rotation. This area of contact is checked by the following procedure.

The teeth are coated with a special marking compound and then run together in a tester. The area of contact can be seen in the disruption of the marking compound, and an experienced inspector is required to interpret the visual results. To document this contact, adhesive tape is then applied to the tooth surface and transferred to a piece of paper (see Fig. 1).

When a gear is installed in a gearbox and is powering the designated application, there are varying degrees of pressure, or load, on the gear teeth. These pressures include box deflections, bearing movement and temperature changes. When the gear teeth are subjected to these variables, the contact pattern will change.

Figure 2 shows the contact pattern from a gear with a very light load and a contact pattern from the same gear with a very heavy load. There is a general rule of thumb, which states that the heavier the load, the larger the contact pattern.

Now here is where the issue of contact pattern becomes so important. For a gear to perform properly under load, the contact pattern must be a certain shape and at a certain location. Typically, an ideal tooth contact pattern under load should encompass the bulk of the tooth surface while avoiding any contact with the edges of the tooth surface (see Fig. 3).

Another critical issue to consider, when assessing how the contact pattern will perform in an operating gearbox, is gear displacement.

In the operation of many gearboxes, the gears and their shafts do not remain in a fixed orientation. Thermal forces and stress from being under load can cause significant movement of the gearbox components from their original positions.

There are typically four different types of

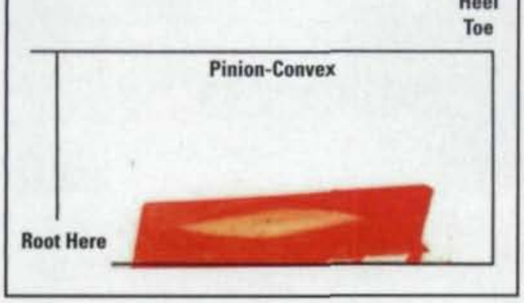


Figure 1—Typical contact pattern.

movement that can take place. These types are described as offset, pinion in and out of mesh, gear in and out of mesh, and shaft angle (see Fig. 4). It is this movement that is referred to as gear displacement, and it can occur in any combination of the four types.

In aerospace gearboxes, where keeping weight to a minimum is a high priority, the mass of the gearing used is usually smaller, and these displacements can be significant. On the other hand, in commercial applications where the gearbox components are typically more rigid, there is not the same degree of displacement.

Conventional Methods for Contact Pattern Development

The size and position of the contact pattern has always been a primary design consideration for gears. And for many years, achieving a good contact pattern was performed through the same methods that the vast majority of gear producers still use today.

The conventional method of achieving an ideal contact pattern is performed in the following way. First, an engineer will make an educated guess at the gear tooth geometry required to provide a correct contact pattern. Next, the part is fabricated and the gear teeth are machined to an undeveloped summary.

When the gear and its mating pinion are finished, they are run together in a tester. More often than not, the contact pattern will not be correct in this first attempt. This requires going back and changing the settings on the gear tooth grinder, then producing a new pinion. The parts are checked again. This trial-and-error process can continue through many cycles until the best educated guess for contact pattern location is achieved. But how will the gear perform under load in a gearbox, and what will the contact pattern look like then? Answering this question leads to more steps in the trial-and-error process.

First, the gears are mounted in the gearbox and run under light load to determine the contact pattern movement. Then, the gears are visually inspected to check the contact pattern, which is indicated by a light wear pattern on the mating tooth surfaces. If the pattern is not correct, which is commonly the case, the gear tooth grinder has to be set up again with new machine settings, and another pinion is ground. This cycle continues until a suitable contact pattern is developed when run under full load.

For a new gear design, this process can take several months to complete. And while this is a time-consuming and costly process, it was just the way it had to be done—or it was until new computer-based technologies for gear development became available.

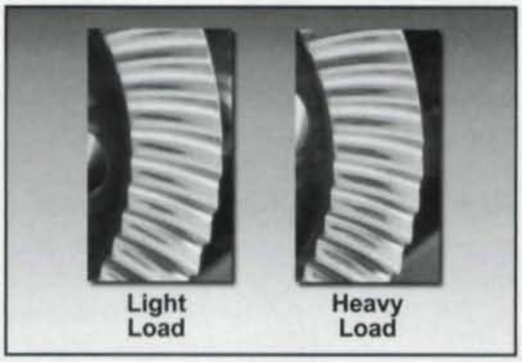


Figure 2—Same gear with light load and heavy load.

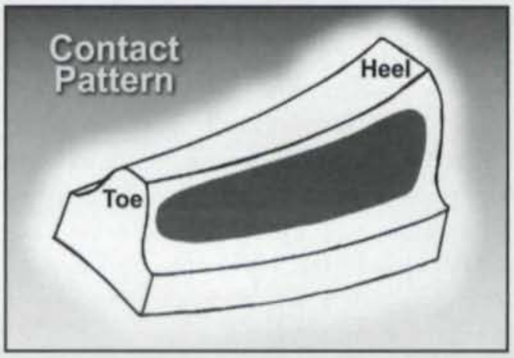


Figure 3—Ideal contact pattern under load.

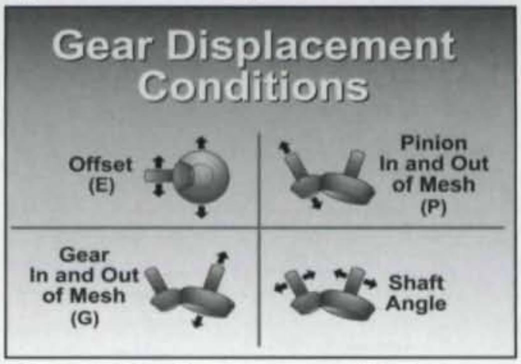


Figure 4—Gear displacement conditions.

A New Method for Contact Pattern Development

To address the traditional limitations of conventional methods, Arrow Gear implemented a highly advanced system for performing contact pattern development, a system that provides a dramatic reduction in the time and expense of the process when compared to conventional methods. This system uses a combination of state-of-the-art development software and machine tools. Among its key components are The Gleason Works' G-AGE, CAGE, MINIGAGE, loaded TCA and T-900 finite element analysis software packages. And for machine tools, the system utilizes Gleason Corp.'s Phoenix® CNC tooth cutters and Phoenix CNC tooth grinders, in conjunction with

Joseph L. Arvin
is president of Arrow Gear Co. of Downers Grove, IL. An industrial engineer, he started in the gear industry as a machinist at Indiana Gear of Plymouth, IN, in 1959. With Arrow Gear since 1972, Arvin helped introduce advanced manufacturing technologies and computer integration to the company.

Thomas C. Mifflin
is a gear engineer at Arrow Gear. Also with the company since 1972, he is responsible for gear manufacturing, perishable tool design and contact pattern design.

James J. Cervinka
is CEO and chairman of the board at Arrow Gear. He co-founded the company in 1947. Like Arvin, Cervinka helped introduce advanced manufacturing technologies and computer integration to Arrow Gear.

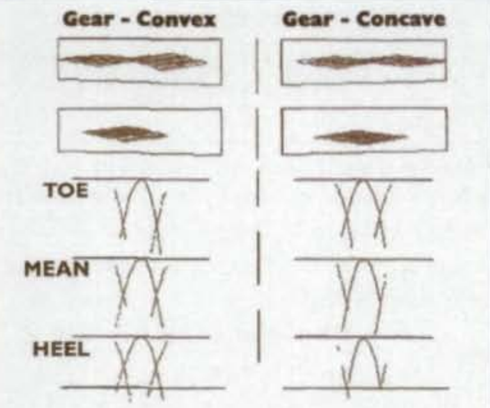
a Zeiss-Höfler CNC gear inspection system. More detailed information on the use of this system will follow, but here are a few highlights of its capabilities.

Using the development software, engineers can build virtual models to predict how the gear will perform in actual operation. This in turn generates the settings to be used by the machine tools. In addition, these settings for the machine adjustments are automatically downloaded to the machine tools, greatly reducing the time spent on setup. Perhaps the most dramatic aspect of this system is that ideal settings of the machine tools—which are required to produce the desired contact pattern—are typically achieved in the first or second attempt on the gear manufacturer's shop floor.

In essence, this system eliminates the trial-and-error process that was once required. And the bottom line is that development time is reduced and the gear producer is able to provide a significant cost savings to the customer.

Developing the Contact Pattern Through Computer Modeling: Overview of the Process

The process of developing a contact pattern with this system is very complex. However, to provide a clear understanding of how the system works, the conceptual highlights of a typical development will first be presented. A more detailed explanation of the steps involved will be presented later.



Arrow Gear Co.
Summary No. 3029
Ratio: 65 x 74 Date: 2/17/99

Generated Gear	Convex	Concave	
Pin Blade Angle	11.500"	16.300"	
Gear Blade Angle	16.000"	16.000"	
Point Diameter	9.812"	8.438"	
Contact Length Factor	0.310	0.310	
Delta Ax	0.600	−0.080	
Delta E	−0.100	0.040	
KP	5.000	6.000	
Delta EM	0.150	0.600	
Delta 60X	0.000	0.000	
Delta 2C	0.000	0.000	
Delta E-DR	−0.500	0.600	
Profile Separation Factor			
IN* 1,000	Mean	Toe	Heel
CX E(U)	−9	16	−28
P(H)	−3	−18	9
CU E(U)	13	−12	38
P(H)	−35	−29	−47

Figure 5—Printout of a TCA study.

The process begins by receiving the customer's design requirements. This would include drawings of the part detailing the critical geometry, such as ratio, diametral pitch and so on. In addition, it is helpful if the customer can supply specifications on operating torque and the gear displacements.

Engineers begin the process of contact pattern development by establishing a working file for the part based on its geometry. Using the CAGE software, a tooth contact analysis study, or TCA study, is performed. This indicates the location of the contact pattern without load.

Finally a loaded TCA is performed, taking into account all the displacement conditions. Once the TCA study is performed for all displacement conditions, the ideal contact pattern is identified. With this information, a finite element analysis is performed that predicts real stress on the tooth surface as well as the root fillet. This study allows the engineers to determine whether there is a potential for failure resulting from excessive or nonuniform pressures anywhere along the line of engagement of the gear tooth.

A more detailed explanation of how the TCA and finite element studies are actually performed is presented in the next section.

Developing the Contact Pattern Through Computer Modeling: Details of the Process

In this section, we will present the details involved in the process of designing the contact pattern through computer modeling and how the software integrates with the machine tools.

To begin, Figure 5 is a summary printout of a TCA study. This particular TCA is from an upper tower or PTO (power take-off) gear set, for use in an aircraft jet engine. For the purpose of illustration, we will be looking at the concave side of the gear and addressing the loaded TCA phase of the design work, when various displacements were taken into account. The different displacements came from thermal and external forces, in addition to the normal operating torque load.

Figure 6 shows the contact pattern design that was created to meet the load requirements and the different displacements that the gear set would encounter.

The different displacements that would result from varying thermal and external forces and from a load of 3,140 in.-lbs. of torque are shown in Figure 7. Some of those displacements are considerable, such as the pinion moving above the gear (E) by 0.013", the pinion going into mesh (P) by almost 0.029" and the gear going out of mesh (G) by 0.026".

The objective was to design a contact pattern that would have an acceptable shape and size, would never run off the ends, make contact in the root fillet, or run off the top lands—while taking into account the different displacements the gears would experience under normal operating conditions.

The contact pattern that was designed then met the requirements of the different displacements shown in Figure 7. Next to each requirement is the contact pattern from the loaded TCA study. If these contact patterns were overlaid or their contact areas combined, the result would—in essence—be a depiction of what the load zone will be for this gear set while it is in operation and encounters all of these different displacements at 3,140 in.-lbs. of torque.

In addition to each one of the different displacements on the contact study, the study will also look at the various pressures that are occurring along the path of engagement. In Figure 8, as the tooth comes into mesh, the path of engagement starts at point A. It then rolls through mesh and exits at point B.

Given a load of 3,140 in.-lbs. of torque, the table in Figure 9 shows what the surface pressure is at the start of engagement all the way through to the end of engagement. The pressures at the start of engagement are low, which are a result of tooth sharing—due to the high contact ratio. The pressures then start to climb and will reach a peak of 238,000 lbs./sq. in. in the center of the tooth. The pressures will then diminish, finally falling to nearly 84,000 lbs./sq. in., where this tooth has exited from mesh.

A key objective of this study in Figure 9 is to verify that there are no hard spots occurring in the pattern. Hard spots would show as spikes in these surface pressures. If a spike in these surface pressures is present, there is a strong indication that a failure mode may be present. As the teeth would come into mesh, the spike or ledge would create a nonuniform pressure, potentially causing pitting and subsequent failure. However, in this example, the pressures do not include any spikes. There is a gradual increase to the center of the tooth, followed by an equally gradual decrease. These gears will move in and out of mesh very smoothly.

This study will be performed for all of the dis-

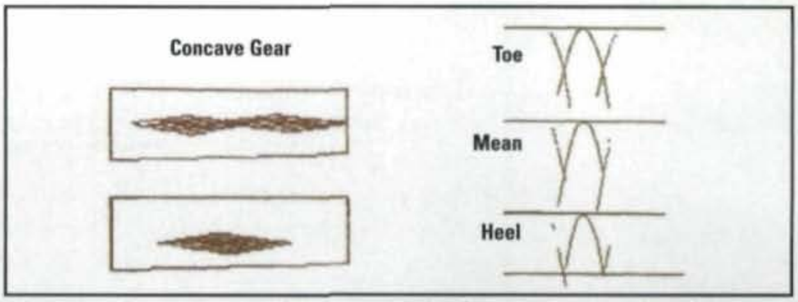


Figure 6—Contact pattern design to meet load requirements and displacements.

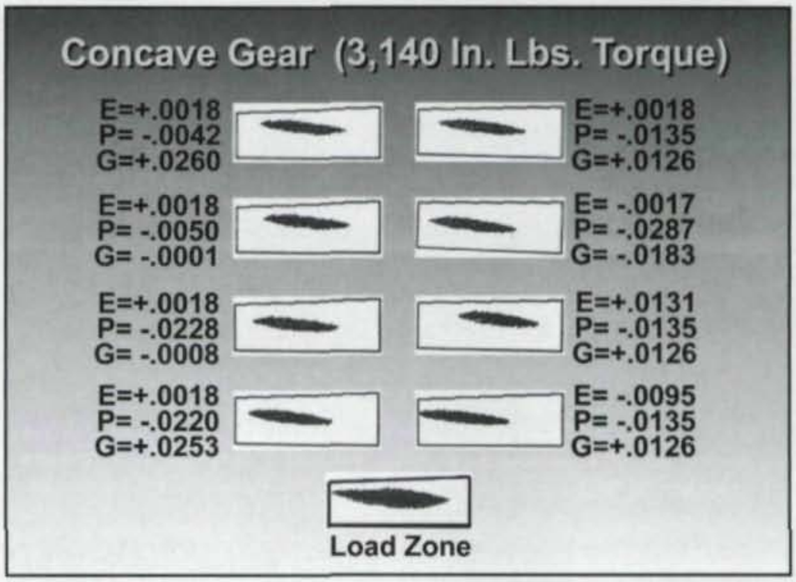


Figure 7—Displacements in aircraft jet engine project, corresponding contact pattern designs and combined load zone.

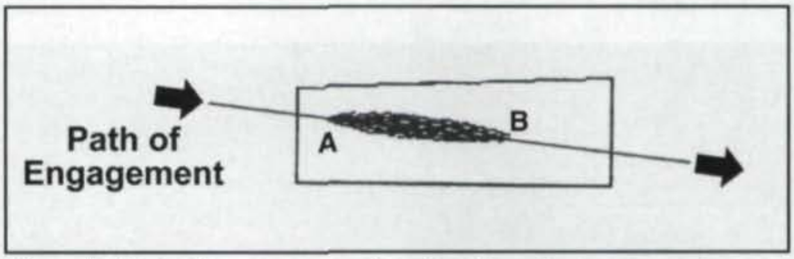


Figure 8—Study of pressures occurring along the path of engagement.

Summary No. 3029 Ratio: 65 x 74
Q 0.94983 THP 0.00000 QG 0.95819 THG 0.00000
DXGR 0.00000 U 0.00000 H 0.00000

Torque (in.-lbs.)	0	500	1,000	1,500	2,500	3,000	3,140
Max. Applied Pressure/1,000 (psi, Toe to Heel)							
	0	0	0	0	0	10	70
	0	0	0	0	114	139	145
	0	0	0	60	153	172	176
	0	0	0	116	171	189	193
	0	0	88	134	179	196	200
	0	0	123	152	190	205	209
	0	0	143	166	200	213	217
	0	0	158	177	208	220	224
	0	0	169	187	215	226	230
	0	0	179	195	221	232	235
	0	0	180	202	226	237	238
	0	0	178	205	226	233	235
	0	0	178	203	225	233	235
	0	0	177	192	215	225	227
	0	0	171	185	209	220	222
	0	0	157	174	199	210	213
	0	0	142	161	189	201	203
	0	0	124	147	178	190	194
	0	0	100	130	166	180	183
	0	0	53	109	153	168	172
	0	0	0	77	139	156	159
	0	0	0	0	112	129	133
	0	0	0	0	0	73	84

Figure 9—Table showing surface pressures.

placement conditions, and when complete, the design will be ready for finite element analysis.

The results of the loaded TCA are first downloaded to the T-900 finite element analysis software. The program then performs a real stress analysis of the tooth surface.

A report is generated (see Fig. 10). Through the use of different colors, the report shows the load distribution along the different areas of the tooth. The areas where there is the heaviest contact on the tooth are red. As the stresses decrease, the colors change and continue out until there is a base load, which is the lowest surface pressure or stress that will be seen on the gear tooth.

A similar study is then performed on the root fillet (see Figs. 11a and 11b). Again, the varying levels of stress are indicated by different colors.

A bar graph that specifies the corresponding pressure is generated on the reports for the tooth surface (see Fig. 10) and root fillet studies (see Fig. 11a). If the maximum value exceeds the rating of the material being used, there is a high potential that the gear will fail.

Another insight that is provided by the finite element analysis is the potential for ledges or edge contacts. As was mentioned before, a red area is an indication of the highest pressure. If the study indicates any red areas outside the center of the contact pattern, it would suggest that a failure might occur in these areas.

If the finite element study reveals any problems, the engineer can then go back to the CAGE software and modify the contact pattern as needed, then perform a second finite element analysis.

Once the TCA and finite element studies are performed, and the ideal tooth contact pattern size and location is achieved, the CAGE software creates the summary settings required by the Phoenix cutters and grinders to machine the parts. In addition, the G-AGE software is used to generate the inspection file for the Zeiss-Höfler CNC inspection system. Using the Zeiss-Höfler system, electronic digital topographical plotting of the tooth surface is performed and the G-AGE software automatically changes the machine settings to match the computerized tooth shape desired. Through a hard-wired network connection, both the summary settings and the inspection file are downloaded. After all these development procedures, the production process begins.

Customer Benefits: A Case Study of the PTO Gear Set

This advanced approach for design and contact pattern development provides numerous customer benefits. Foremost among these are dramatic savings of time and money.

An example of these two benefits to the customer was illustrated in Arrow's involvement with the previously mentioned aircraft jet engine project. The details of this project are presented in the following case study.

Arrow supplied gearing on two locations of the engine. The first bevel gear set was used in the upper tower shaft or power take-off. The second bevel gear set was used in the accessory gearbox.

As this was a new engine, Arrow was called upon to perform both the gear tooth design and the fabrication of these bevel gear sets.

As with all jet engine gears, this was a

Duplication of Operating Conditions with Universal Load Testers

As a supplement to the theoretical calculations that are performed to achieve the design of contact patterns, Arrow Gear utilizes universal load testers that are used to simulate the performance of the gears in a gearbox (see Fig. 1). These Gleason Corp. testers, which Arrow retrofitted, can test gears under loads of up to 700 in.-lbs. of torque. To accomplish this testing, an eddy brake was added to generate the load and a strain gage was integrated for monitoring the amount of torque applied. Control of revolutions per minute and load is performed by an onboard computer. The testers also allow for the adjustment of the gear and pinion position, *thus replicating the gearbox displacements.*

These testers first allow the engineers to view the contact pattern as they typically would in a gear tester during the manufacturing stage. Figure 2 shows the pattern that resulted from no more than a few inch-pounds of torque—and the contact pattern is a localized area halfway up the flank and toward the toe.

Figure 1—Universal load tester.

In Figure 3, the tester has applied 120 in.-lbs. of gear torque as well as the gear displacements. As can be seen, the contact pattern has moved from a toe location to a heel location and has grown in size and changed in shape.

The ability of these testers to monitor the movement of the contact pattern can be a valuable aid in manufacturing the gears, as well as for checking their actual performance without requiring them to be installed in a gearbox.

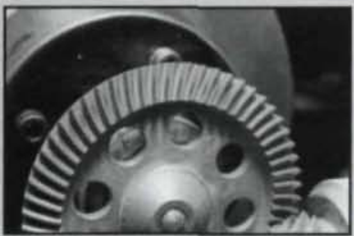


Figure 2—Contact pattern with no more than a few inch-pounds of torque.

There is another substantial benefit of these testers. Often, the customer is unable to determine what its gearbox deflections will be. If this is the case, the engineer can—in essence—back into the deflection values required for TCA and finite element analysis.

Figure 3—Contact pattern with 120 in-lbs. of torque.

Backing into those values is done in the following way. Using the operating torque and the wear pattern from a gear set that has been run in a gearbox, the engineer can duplicate the same wear pattern by adding displacement and load. This will generate all the numerical values required for further evaluation.

demanding application due to the high degree of gearbox deflections. Faced with the double-edged challenge of both a difficult job and a short lead time, Arrow began work on the project utilizing the design and manufacturing tools explained earlier.

The two different gear sets were then produced and shipped for installation in the engine. Here are the results.

First of all, a normal time frame for developing the desired contact pattern under full load can be up to six months. Arrow's initial development was performed in less than one week through computer modeling techniques—and this initial development required no further modifications when run under full load during engine tests. For the actual manufacture of a new gearing application, the typical time frame is 22 weeks. Arrow performed this work in 12 weeks.

After the gears were run in the engine for 75 hours, they were visually inspected. Shown in Figure 12 is one of the gears. Both contact patterns on the run side and start side were exactly as predicted. This approach saved a significant amount of expense and time for the company creating the engine.

Troubleshooting and Failure Analysis

This system for designing gears is used for the most part to design or improve designs on new or existing gear sets. However, there are additional capabilities of the system.

If the system is provided with the proper information, virtual models of the gear teeth can be created to predict the proper location of the contact pattern. This information compared with the actual contact pattern can provide valuable insight to the cause of a failure or other problems. In addition, this approach can improve beam strength of the tooth up to 30% and significantly increase gear life.

Conclusion

In today's competitive manufacturing environment, customer demands for fast delivery and lower costs are prevalent. In this climate, the computerized closed-loop approach to gear production is ideally suited. In addition, by reducing development time, this technique allows the product to be released to the market much sooner—substantially reducing costs to the OEM.

In view of the numerous benefits of this technology, the closed-loop methodology promises to become the standard development technique in the gear industry for years to come. ○

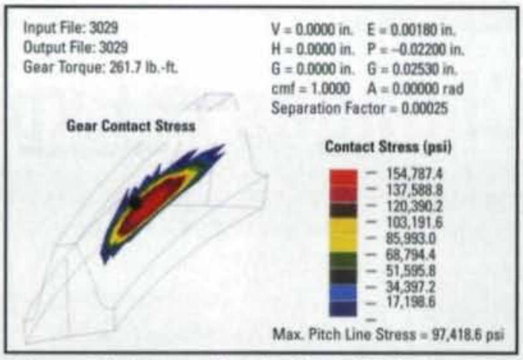


Figure 10—Report showing load distribution.

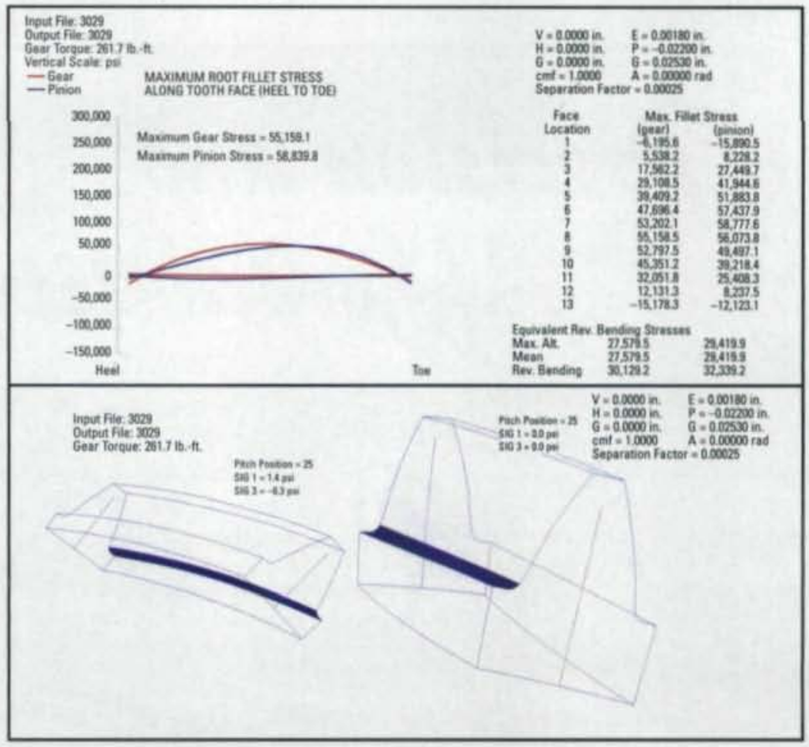


Figure 11—Load distribution study on root fillet.

Figure 12—A spiral bevel gear after running in the aircraft jet engine for 75 hours. Contact pattern is exactly as predicted.

Changes & Expansions

Remains of Ajax Magnethermic Combines With Tocco

The assets of Ajax Magnethermic are being combined with certain assets of Tocco to form Ajax Tocco. This was decided when the operating assets of Ajax Magnethermic Corp. of Warren, OH, were sold to a wholly-owned subsidiary of Park-Ohio, headquartered in Cleveland.

Tocco Inc., headquartered in Boaz, AL, is an induction heating equipment company.

The affiliation with Park-Ohio provides the stability to ensure that Ajax Tocco will continue to supply induction heating and melting equipment, according to Ajax Tocco's press release. Ronald J. Cozean will serve as president of Ajax Tocco.

Ajax Magnethermic closed in late June.

New Headquarters for Bodycote Metallurgical Coatings

Bodycote Metallurgical Coatings opened a new North American headquarters in Greensboro, NC. The new building houses a state-of-the-art coating facility.

This plant is equipped with a fully automated, ultrasonic cleaning line and several new physical-vapor-deposition coating lines. New acquisitions include a Hauzer 1200 series PVD coating machine and a Hauzer 1500 series PVD coating machine. Both series provide coating consistency between batches, according to the company's press release.

In addition to coating, Bodycote North America provides heat treating, brazing, hot isostatic pressing and materials testing. The company maintains nearly 70 U.S. facilities.

Gleason Expands Rockford Facility

Gleason Cutting Tools Corp. expanded its production capacity for high precision, CBN-plated wheels used for external and internal, straight and helical, parallel and bevel gear grinding.

According to the company's press

release, the facility in Loves Park, IL, is very advanced, with an expandable, ultramodern electrolytic and electroless nickel-plating capability currently producing up to 500 wheels per month.

Products available from Gleason Cutting Tools include nondressable grinding and dressing wheels manufactured with hardened steel bodies, precision ground profiles, and nickel-plated single layer CBN or diamond crystals.

A New Midwest Plant for Pioneer Broach

A 20,000 sq. ft. facility in Leroy, MI, was recently completed to house Pioneer Broach Co.'s Midwest division for manufacturing and broaching operations.

The company opened this operation to extend local service to Midwest shops for broach sharpening, reconditioning and complete broach design and manufacturing.

Ken Nemec, previously of Nachi Machining Technology Co. of Macomb, MI, will head the new division as plant manager.

Name Change for British Gear Association

Effective Jan. 1, 2003, the British Gear Association will be known as the British Mechanical Power Transmission Association.

Under the new name, the organization will be able to arrange seminars and projects on topics such as chains, belts, sprockets, bearings, seals and clutches.

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Products for the Gear Industry

New Threaded Wheel Grinder from Gleason

The 245TWG threaded wheel grinder from Gleason produces high quality finish grinding of automotive final drive ring gears, speed gears, planet pinions and planet gears up to 245 mm in diameter, with 3.25 module.

According to the company's press release, DIN 5–6 quality is achievable for most parameters, as is lead and profile accuracy to 2.5 microns.

Use of direct-drive spindles for the grinding wheel, worktable and dressing spindle has led to improvement in overall cycle times. The integration of on-machine dressing, coupled with a load/unload mechanism, enables the grinder to cut floor-to-floor time on a typical final drive ring gear to approximately one minute.

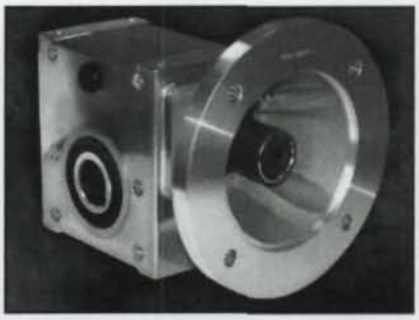
For more information, contact Gleason Corp. of Rochester, NY, by telephone at (585) 473-1000 or on the Internet at *www.gleason.com.*

New Worm Gear Unit from Renold Gears

According to Renold Gears, the JW Series worm gear unit is a high quality, low powered unit providing a drive solution to a range of applications, including printing and textiles, materials handling, conveyor drives and other industrial uses.

According to the company's press release, the gearbox is available in 10 sizes with power capacities up to 4kW and gear ratios of 5:1–100:1. Suitable motors include both standard IEC and NEMA motors, including the high efficiency EFF1 motor range. Double reduction units can be used to provide the option of a 4,000:1 ratio. Additionally, the product incorporates mounting surfaces on all sides of the gear case and the option of an output mounting flange on either side of the unit.

For more information, contact Renold Gears of Rochdale, U.K., by telephone at (44) 1706-751-000 or on the Internet at *www.renold.com.*

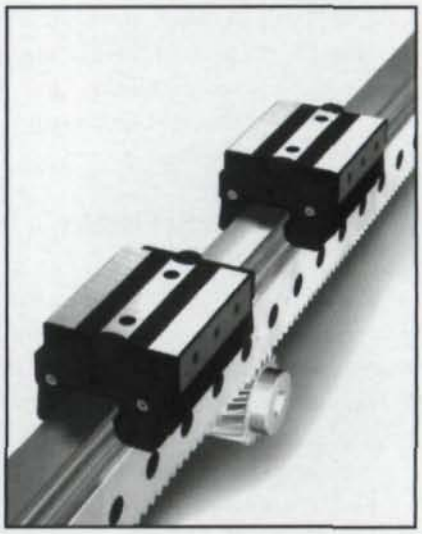


New Right-Angle Gearing Technology from Cone Drive

The Conex helicoidal gearing enables Cone Drive to offer increased torque in a smaller package.

Designed for use in conveyor handling systems in the paper and packaging industries and in small- to medium-duty manufacturing operations, Series B is offered in universal gear case sizes 02–11. According to Cone Drive's press release, this gear geometry has power capabilities up to 20 hp and maximum output torque capacity of 5,000 lb.-in., with a choice of 10 standard ratios from 5:1 to 60:1.

For more information, contact Textron Power Transmission of Traverse City, MI, by telephone at (888) 994-2663.



New Linear Guide with Integrated Toothed Gear Drive System from Schneeberger

According to Schneeberger Inc., the Monorail BZ 35 connects highly accurate toothed gear segments by creating a unified linear guide and rack that is ready for fast installation with a pinion drive.

The guide can handle forces large enough to be used in wood process machines as well as laser and water jet cutting machinery.

Single lengths up to 2,000 mm are available in the 400 mm module and single lengths up to 6,000 mm are feasible without a joint required. By lining up several BZ systems next to each other, large paths of travel can be achieved without loss of accuracy.

For more information, contact Schneeberger Inc. of Bedford, MA, by telephone at (781) 271-0140 or on the Internet at *www.schneeberger.com.*

New Grades of PBT from Ticona

The Celstran *PBT-GF30*, Celstran PBT-GF40 and Celstran PCT-GF50 materials from Ticona are long-fiber-reinforced polybutylene terephthalate (PBT) grades with 30%, 40%, and 50% long-glass fibers.

The new grades contain a PBT polymer specifically formulated to improve the mechanical properties of long-fiber composites, according to the company's press release.

Ticona added that the impact, tensile and flexural strengths of the new long-fiber-reinforced PBT grades are nearly identical to those of nylon.

For more information, contact Ticona of Summit, NJ, by telephone at (800) 235-2637 or on the Internet at *www.ticona.com.*

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For more information, contact HD Systems of Hauppauge, NY, by telephone at (631) 231-6630 or on the Internet at *www.HDSI.net.*

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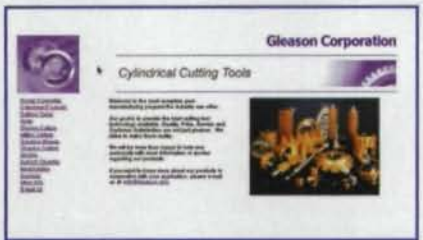

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A Man and His Mania for Antique Machines

Richard Spens has a hobby that leads him onto the Internet, through magazines, to auctions and into farmers' back yards.

It's a hobby that he succeeds at through obsessive-compulsive behavior—his joking description of his persistent interest. He says he looks everywhere and all the time for what he wants, to the limit of what his wallet—and his wife—can stand.

Richard Spens collects antique machinery. About six years ago, his hobby led him to a McDonald's parking lot near Midland, MI, to meet a woman taking her daughter to college in Michigan's upper peninsula. The woman's SUV was carrying Spens' then-latest acquisitions.

One of those acquisitions was a hand-operated gear-cutting machine that may be as many as 116 years old.

That age is based on the company name on the machine: Sloan, Chace and Co. That business was organized in 1886 as a partnership between Charles T. Sloan and George E.O. Chace. Sloan originally founded the business in the 1870s. The 1886 partnership later became Sloan and Chace Mfg. Co. All three versions of the business made small bench lathes, small bench milling machines and small gear-cutting machines.

Spens knows little else about the business, and that much he learned from one of its lathe catalogs and from *American Lathe Builders: 1810–1910*, a history by Kenneth L. Cope.

That day at McDonald's in 1996, Spens used 2 x 6s to slide his new acquisition from the woman's SUV to the back of his pickup truck, along with a second antique machine and some collets and attachments. Spens' total bill: $350 for two machines and other parts, $40 for the delivery service.

Now in his basement workshop, Spens' hand-operated gear-cutting machine can be used to make spur, face and straight bevel gears. The gears can be brass, cast-iron or steel, can have teeth as coarse as 24 DP, and can be as much as 4" in pitch diameter. Also, the teeth can be accurate to 0.002" of tooth-to-tooth error and 0.005" of total composite error on larger gears. According to Spens, the machine is more accurate when cutting smaller gears.

"For its time, that was pretty good," he says of the machine's accuracy, "especially on that larger size gear."

Spens himself has cut a brass spur gear with a 0.920" outside diameter and 24 DP to a quality level that he equated with AGMA Q7.

Spens explains that the machine cuts each type of gear based on the position of its arbor. The arbor can be moved anywhere along an arc radius just below and ahead of the gear-cutting tool. If the arbor is in a horizontal position, the machine cuts spur gears. If in a vertical position or at the arc's bottom, the machine cuts face gears. If at an angle, it cuts straight bevel gears in two or three passes.

The arbor and indexing adjustment can be finely adjusted downward to create gears of different diameters. The depth of cut can be adjusted by placing shim stock under the feed stop.

Also, the machine has a vice that can be placed anywhere along the arc. The vice has a feed adjustment that can be moved in thousandths of an inch.

Spens thinks the vice and feed adjustment were used to make racks, cutting one tooth at a time, then advancing the blank the proper distance and cutting the next tooth.

Spens has more than 40 antique machines of various types in his collection and wants more, including other gear-making machines. Currently, he's looking for what he terms the "elusive"

An antique, hand-operated gear-cutting machine manufactured by Sloan, Chace & Co.

1900–1920s Gleason bevel gear planer.

He explains that the planer's operation is very complex and quite interesting: The planer uses its single cutting tool like a shaper cutter, but it cuts gears by tracing an involute template or other tooth form template. He adds that the planer planes its tooth forms to any pressure angle on any size blank up to the machine's capacity.

Specifically, he's looking for the planer model that can cut blanks with outside diameters up to 24".

Given his interest, Spens' reaction to finding that model or another antique machine that interested him, can be easily predicted: "I'd buy it, if—you know—it was affordable; and I'd probably come out and get it."

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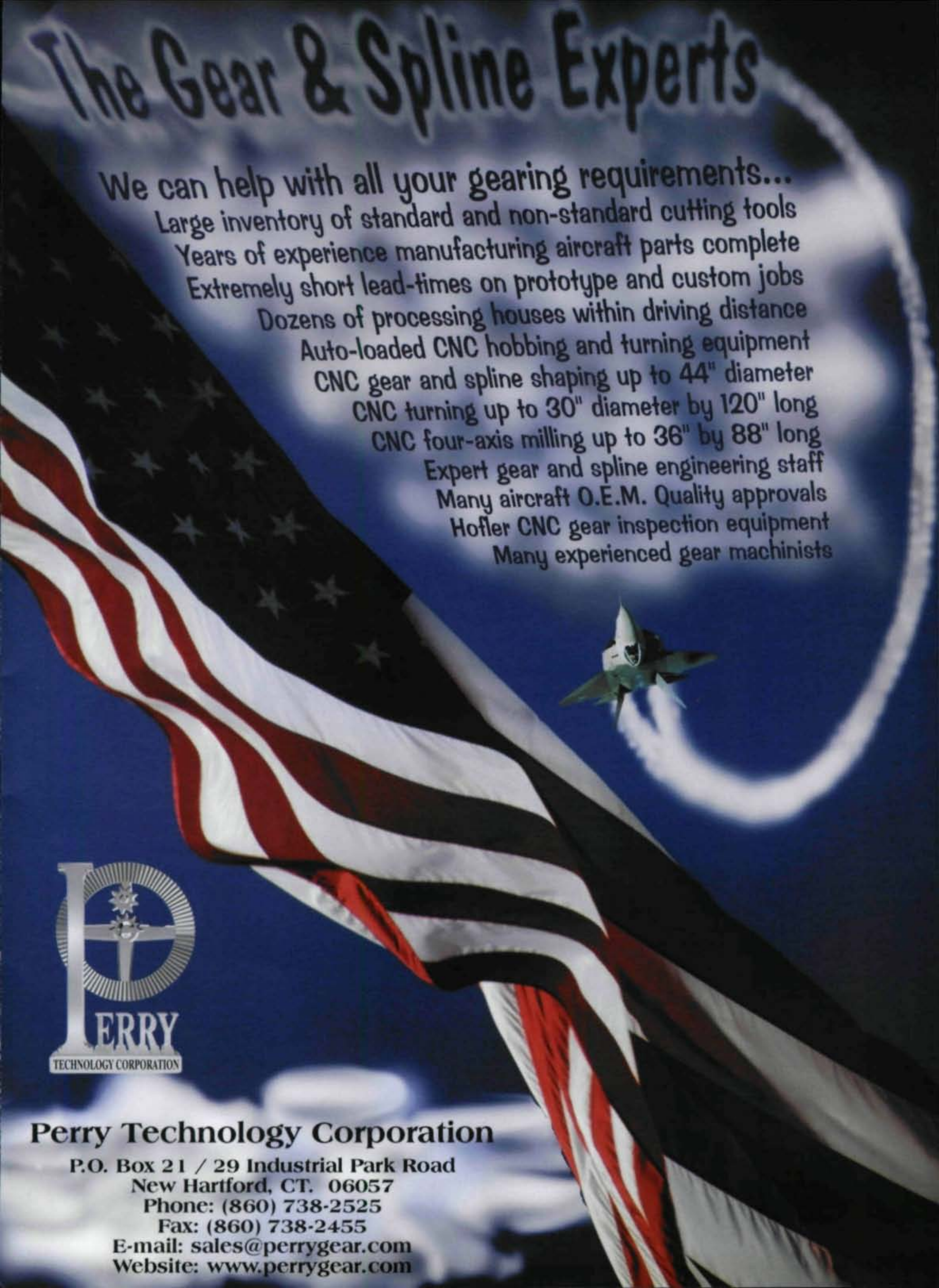
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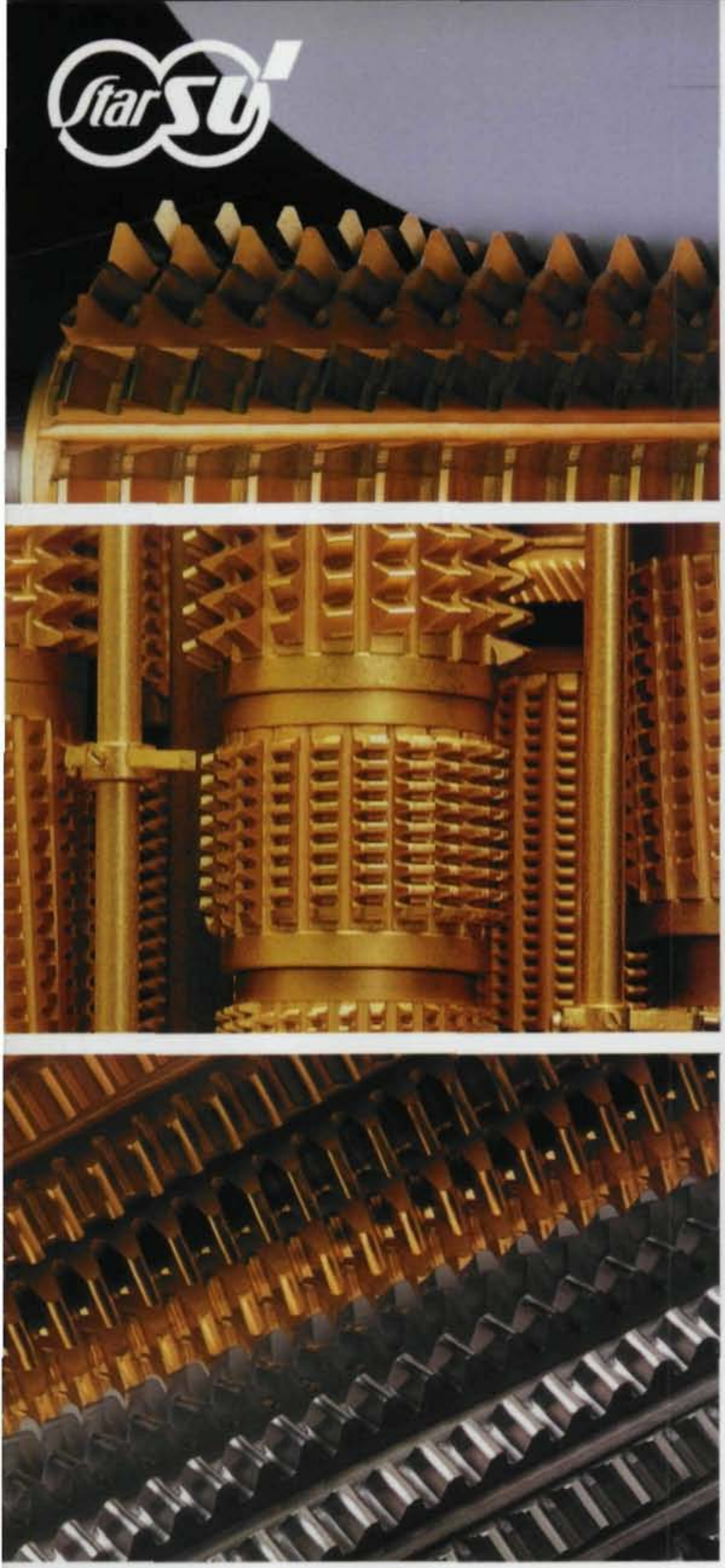
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