GEAR TECHNOLOGY

The Journal of Gear Manufacturing

GRINDING

May/June 1998

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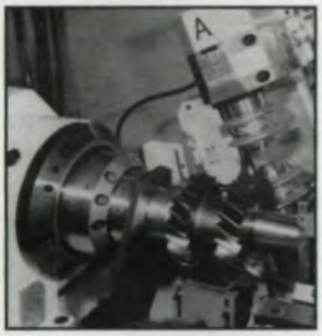
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MEETING THE CHALLENGE

Every once in a while something happens to fundamentally change the nature of your business. Despite the best of intentions and the most careful planning, there's no way we can anticipate every event. What do you do, for example, when your two biggest competitors merge, when the economy collapses in the region that imports your products or when key employees leave your company? Your reactions may make the difference between success and struggling to survive.

One of the keys is not to **over**-react. Overcoming challenges such as these can make your company stronger, more profitable or more efficient—but only if you approach the problem as an opportunity rather than a crisis. When things seem to be at their worst may be the best time to reexamine your organization. Find your strengths and ways to use and exploit them to turn negative into positive.

In today's economy, big companies swallow little ones, and competing interests form partnerships to gobble up more market share. When these marriages occur between your competitors, it's easy to feel like a bitter spinster. But it's important not to act like one. Don't be content to be left behind, and don't feel like it's your only course of action. Perhaps it's time to look into new areas of technology. Perhaps it's time to concentrate on what you do best. Perhaps it's time to carve out an unexplored market niche. The point is, it's probably time to do something. You just have to figure out what.

Chances are, if you've been minding your business all along, your company probably isn't in bad shape anyway. You probably already have many of the right people, skills, products or technologies to bring you through the struggle.

This topic hasn't come to me out of the blue. We've recently undergone some changes of our own at *Gear Technology*, and hopefully our reactions to them—and our approach to the problem—will make us stronger in the long run.

Many of you have come to know—if not in person, through her words—our departed senior editor, Nancy Bartels, who has recently accepted greater responsibility as managing editor at another publishing company. Over the past decade, Nancy has become the driving force behind the content of *Gear Technology*. If you read it in these pages, you can be assured that she read it first. Then she shaped it, polished it and made it into a finished product. Thank you, Nancy, for your years of hard work.

We've also recently said good-bye to Ed Mueller, who was our associate publisher. While many of you probably didn't get to know Ed, he was responsible for decision making, administration and overseeing of our magazine and two Web sites. Like Nancy, Ed left to pursue other opportunities.

In a relatively small operation such as ours, the loss of a key employee can be difficult. The loss of two can be crippling. We will certainly miss Nancy and Ed and the work they did here. But within every crisis lay the seeds of potential. Although the coincidence of these events seemed unfortunate, it forced us to act quickly. Instead of looking on our situation as a catastrophe, we looked on it as an opportunity to redesign and rebuild our organization.

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CIRCLE 136

PUBLISHER'S PAGE

You may already recognize the name of our new managing editor, William R. Stott. Some of you may even know him by "Randy," which is the name he uses when he's not in print. Randy has been our associate editor since 1994. He comes from a newspaper and magazine journalism background and will continue his efforts to make this publication the best possible information source for the gear industry. If any of you have ideas to make *Gear Technology* better, please give him a call.

Over the years Randy has also taken on the responsibilities of circulation manager for the magazine and Webmaster of our two Web sites. Because of his involvement in these other areas of our company, it seemed natural for him also to step into the administrative role that was formerly held by the associate publisher.

In addition to these changes, we've added strength from outside our organization. Our new associate editor, Charles M. Cooper, comes to us from the Illinois Pharmacists Association, where he was managing editor of their bimonthly journal and monthly newsletter. He will be doing a lot of writing and editing both for the magazine and our Web sites. He has already proven to be a valuable, productive member of our staff, and we hope you'll join us in welcoming him to our industry.

The challenge of an unusual coincidence of timing has forced us to make these changes to our staff. While this will undoubtedly change the personality of the magazine, we want to assure you that *Gear Technology* remains committed to bringing you the best technical information available on the gear industry. We will continue to bring you both the basics and the cutting edge technologies, as well as all the news and information that helps you do your job. It's our goal to help you compete more effectively, economically and successfully.

We met the challenge of our difficult situation by looking on it as a chance to move on. What seemed at first like a crisis has turned out to be an opportunity. We restructured and consolidated. We found talent within and brought more from without. There's no reason why your next crisis shouldn't be treated in the same way.

Michael Goldstein, Editor-in-Chief

P.S. Recently I had the honor of speaking before the AGMA on the topic of the Internet and how it can benefit companies in the gear industry. Many who attended have asked that we publish this information, so we will be including this material as an article in our July/August 1998 issue.

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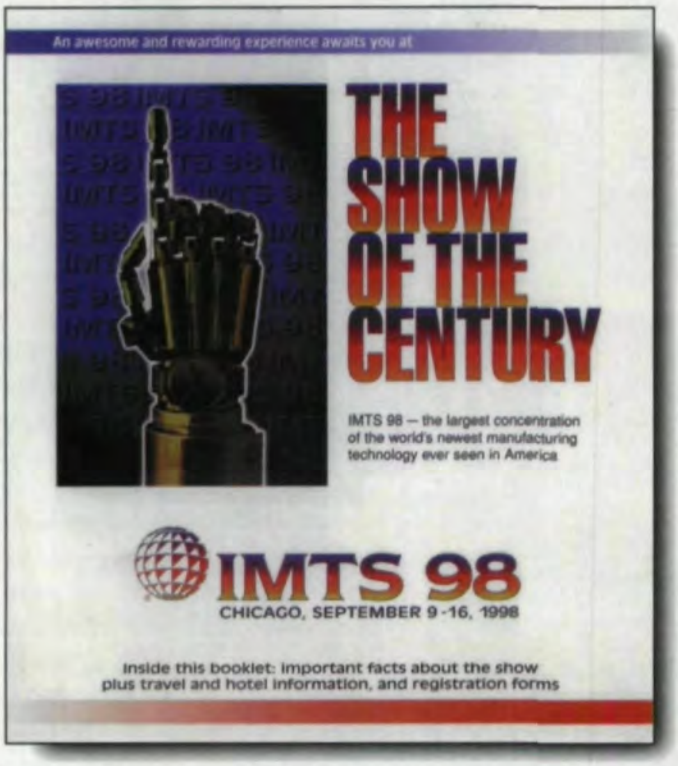

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CIRCLE 118

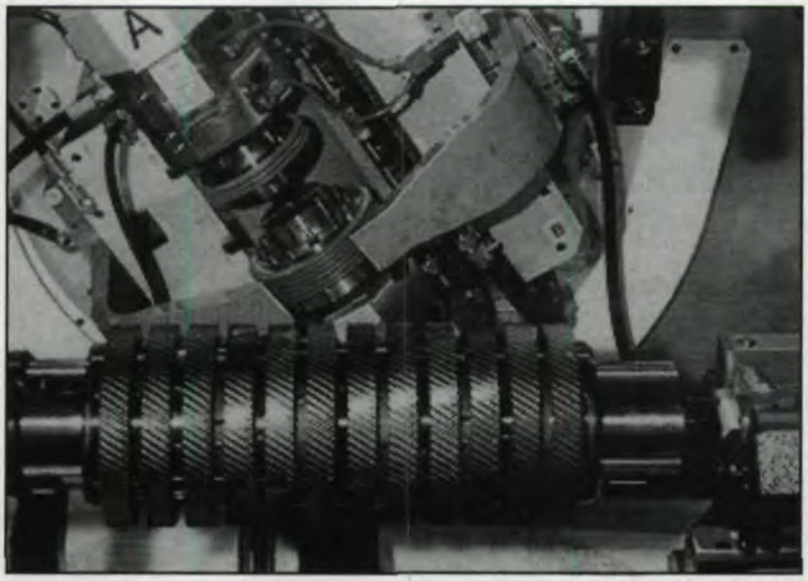
Hard Gear Finishing With CBN–Basic Considerations

Paul Brazda

For over 50 years, grinding has been an accepted method of choice for improving the quality of gears and other parts by correcting heat treat distortions. Gears with quality levels better than AGMA 10–11 or DIN 6–7 are hard finished, usually by grinding. Other applications for grinding include, but are not limited to, internal/external and spur/helical gear and spline forms, radius forms, threads and serrations, compressor rotors, gerotors, ball screw tracks, worms, linear ball tracks, rotary pistons, vane pump rotors, vane slots, and pump spindles.

Grinding as discussed in this article is the process of using abrasive grains for shaping workpieces into more precise form. A number of materials can be used for grinding. They include:

- Aluminum oxide (corundum) — for soft or hardened steel grinding;
- Silicone carbide — for cast iron, nonferrous metals and nonmetallic materials;
- Diamond — for cemented carbides, glass, ceramics and hardened tool steels;
- Cubic Boron Nitride (CBN) — a synthetic superabrasive used for grinding hardened steel and wear-resistant superalloys. The cubic shape of the CBN particles results in grains that have very pronounced cutting edges. The mechanical strength of CBN is more than double that of corundum. CBN also has the capacity to withstand thermal loads twice as high as those of diamond. These make CBN a good choice for a grinding abrasive.



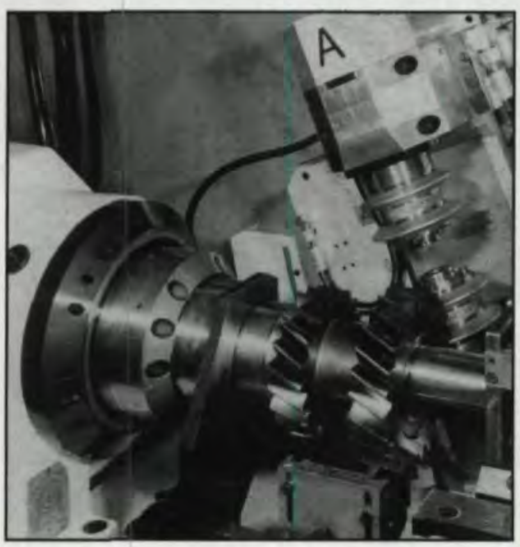


Fig. 1 — Multiple workpieces can be clamped in one setup for improved productivity.

Fig. 2 — Multiple spindle machines can grind dissimilar cluster gear members in one setup.

Several different types of bonds attach abrasive particles to the grinding wheel.

- Vitrified bond. Uses an inert, glass-like material. Not affected by grinding fluids or high temperatures generated during normal grinding. Vitrified wheels are impact-sensitive.
- Resinoid bond. Uses a highly flexible thermosetting plastic. Resinoid bonded wheels are often selected for their high operating speeds. These wheels have high impact resistance.
- Rubber bond. Gives more flexibility than resinoid bonded wheels.
- Metal bond. Used with diamond and CBN electroplated wheels.

The choice of which bonds to use is largely application-dependent and is a function of the varying degrees of softness in the bonds. Rubber-bonded wheels are the softest and metal-bonded the hardest.

Rubber-bonded wheels are useful in cut-off applications. When they make a cut, their flexibility will leave a very smooth, almost burr-free edge. A vitrified or metal bond wheel will roll a small piece of material over the cut end of the workpiece, leaving a sharp edge.

Resinoid-bonded wheels are harder than rubber, but softer and more flexible than vitrified wheels. If the workpiece is of an unknown type of metal or if the ground stock varies in thickness, resinoid-bonded wheels may be a good choice. Because they are flexible, they will give if they hit a high or rough spot on the material and will not shatter like a vitrified wheel. But there is a trade-off in terms of accuracy. Grinding with resinoid wheels cannot achieve the same degree of accuracy as grinding with vitrified wheels.

More than half of the grinding done today is done with vitrified wheels. They can produce a highly accurate profile and are used when a more accurate form is necessary than can be achieved with milling, lathing or other machining. The downside of using vitrified wheels is their fragility and the fact that the profiles must be dressed often to the desired form. They are impact sensitive. If they are suddenly bumped, or if they encounter an irregularity in the workpiece, they can shatter. This not only ruins the wheel, but also can damage the workpiece and, conceivably, injure the machine operator.

Metal-bonded CBN wheels are the hardest grinding wheels. They consist of a thin coat of CBN grit fixed to a hardened steel wheel, thereby eliminating the breakage problem found with vitrified wheels. They also provide the greatest degree of accuracy. This is why one of the biggest customers for CBN wheels is the aerospace industry, where the limits of profile tolerances are pushed every day. Only CBN wheels can provide the degree of accuracy required in aerospace quality gears.

However, there are other applications for CBN wheels as well. Because they provide a very stable profile, they are very useful in mass production situations. When several hundred gears of the same type need to be produced, CBN wheels can save significant amounts of setup and wheel dressing time. A third application for CBN wheels is for very large gears (say over 50") with many teeth. Again, because of their stable profiles, CBN wheels prevent first-to-last tooth errors and save setup and redressing time.

On the other hand, CBN wheels are more expensive than vitrified wheels. In prototype production, short runs or in applications where high accuracy is not required, they simply may not be economical.

Dressable or Nondressable Wheels

Both types are available and each has its uses. Dressable grinding wheels are used extensively in production grinding. They can be used in standard and specialized grinding machines for both form and generating grinding.

Most of our attention in this article will be devoted to nondressable CBN grinding wheels. While dressable CBN wheels are available, most CBN form grinding is characterized by the use of a non-dressable, single layer of CBN grains electroplated on a hardened and precision-ground steel body. Such grinding wheels must be manufactured and plated with the required profile form, since the machine operator may make little or no profile form adjustment.

Fig. 3 — In Coroning, cutting takes place when the crossed axes of the tool and the workpiece roll with each other.

This system is capable of simultaneously grinding both flanks, the root and, if desired, even a tip chamfer. This method improves load, noise and wear characteristics of the finished gear. It will also produce excellent accuracy and finish with no truing and dressing for the life of the wheel.

A non-continuous process of form grinding requires the part to be indexed after each grinding pass. Achievable metal removal rates depend on the grain size used, the required part geometry accuracy and specified surface integrity. Improved productivity can be realized by clamping multiple workpieces in one setup to reduce the indexing time required per part. Workpieces can be rough and finish ground in one setup (Fig. 1). Machines with multiple spindles are able to grind dissimilar gear members on cluster gears in one setup (Fig. 2).

The roughing wheel is generally designed using coarse CBN grains for aggressive grinding. The finishing wheel is designed to meet engineering specifications. Cycle times can be improved in certain applications by using a specially designed multiple wheel set which rough and finish grinds the workpiece at the same time. An additional advantage of using this wheel set is in the predetermined control of the stock removal condition between the roughing and finishing operations. The design and manufacture of high-quality CBN wheels capable of meeting or

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is Product Engineering Manager of Kapp Tech, Boulder, CO, manufacturer of grinding systems and wheels. He has a master's degree from CVUT, Prague, in the Czech Republic and 20 years' experience in power transmission systems, including gear form grinding wheel design and development.

Fig. 4 — A variety of CBN grinding wheels.

exceeding AGMA class 13 profile tolerance requires a high level of engineering and manufacturing expertise.

Coroning®*

Coroning is a newly developed gear finishing process that uses the same principle as traditional gear honing, but has some special features. The tool used in this operation is an internal gear with a hardened body, electroplated with a single layer of diamond. Coroning is a two-step process where roughing and finishing tools are used. Its cutting ability arises from the crossed axes of the tool and workpiece rolling with each other (Fig. 3). This process creates a superb surface finish structure with the grinding traces parallel to the rolling direction of the gear, which improves its performance. (Note that traditional form grinding creates grinding traces which are perpendicular to the rolling direction of the gear.) Another characteristic of this process is a relatively low cutting speed, which minimizes the thermal effect of generation of residual stresses.

Coroning is a highly productive, continuous grinding process which distributes the load of material grinding over a larger area of grains. This process has particular advantages for high volume transmission gears and for producing parts with large numbers of teeth. Noncontinuous form grinding cannot achieve the short cycle times of Coroning. Some gears which normally cannot be form ground due to limitations caused by adjacent shoulders or related design restrictions can be produced by the Coroning process.

Metallurgical Considerations

Certain measures must be taken in order to ensure optimum grinding results with respect to the structural condition of the workpiece. If the grinding process uses incorrect machining parameters, there is a danger that a thermally induced modification of the workpiece surface can take place locally. This is called "grinding burn." If the heating process exceeds the tempering temperature, a "tempered zone" is produced. If it exceeds the austenitizing temperature, a "rehardened zone" is produced.

Parameters that may lead to a thermally induced modification of a workpiece include:

1. Excessive feed rate during the grinding process;

2. Excessive infeed (or grinding depth) as a result of

- Errors in the preliminary machining of the workpiece, causing uneven distribution of the grinding allowance;
- Hardening distortions resulting from the heat treatment process;
- Positioning errors of the ground profile with respect to the workpiece clamping device;
- Alignment errors of the ground profile relative to the position of the grinding wheel.

The excellent heat conductivity of CBN produces a cool grinding action, thus reducing the danger of thermally induced part modifications. In spite of this, grinding burns can occur when the parameters of the grinding process are exaggerated or when the CBN grains are significantly worn out. To prevent this, some notion of just how aggressively one can grind in a particular situation needs to be arrived at.

Normalized Removal Rate Per Unit of Time—Q_w

This parameter is a function of the machine, the wheel, the grit size, and the feeds and speeds used. The grinding must take place at a reasonable depth and speed that at the same time will not produce thermal changes; i.e., grinding burns. Kapp uses the term Q_w to express this limit. This value refers to the volume of material removed (mm^3) per unit of time (seconds), based on the grinding wheel width in millimeters. The dimension is, therefore, expressed as mm^3/sec/mm.

Q_w can be obtained by calculating the product of feed rate and infeed. As the number of workpieces ground with an electroplated CBN grinding wheel increases, the abrasive grains (CBN crystals) become dull, whereupon the permissible values for Q_w decrease as well.

Other grinding wheel manufacturers may use other terms to describe this parameter. If this information is unavailable from the grinding wheel provider, the user has to arrive at some notion of what these grinding limits are on his own.

Grinding Fluids

The grinding fluid is probably the most overlooked factor in optimizing a grinding wheel's

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IF THE GRINDING PROCESS USES INCORRECT PARAMETERS, A THERMALLY INDUCED MODIFICATION OF THE SURFACE — GRINDING BURN — CAN TAKE PLACE.

performance. The type of fluid and its application are very important for the success of a grinding wheel's performance. Most production operations use some type of grinding fluid in order to increase the grinding wheel life and improve the quality of the ground part. These fluids contain a blend of ingredients which inhibit corrosion and extend fluid life despite very adverse conditions. Some of the grinding fluid functions are to

- Lubricate the grinding zone;
- Reduce the frictional heat;
- Cool the grinding wheel and workpiece to eliminate the danger of a grinding burn;
- Remove the chips from the grinding zone;
- Clean the grinding wheel surface to allow higher material removal rates.

CBN grinding wheels provide the most outstanding performance when used with waterless lubricating fluids. Such fluids reduce the chemical decomposition associated with CBN and high temperature steam. The formation of an oxide (B_2O_3) on the surface of a CBN particle creates a protective layer to deter further oxidation. However, this oxide layer is soluble in high temperature water and can allow further oxidation of the CBN particles. These reactions can cause accelerated breakdown of the CBN abrasive particles and shorten the grinding wheel life. The use of oil-based grinding fluids minimizes this effect.

Generation of Residual Stresses

While the grinding grains are applying pressure on the surface of the workpiece during its movement, tensile stresses are introduced under the surface. Chips are literally pulled away from the surface of the gear. When the tensile stress reaches the yielding point, plastic deformation will develop, reducing the level of tensile stress. After rapid unloading, plastic deformation turns into compressive stress in the direction of grinding and also perpendicular to it. The surface "snaps back." Simultaneously with this mechanical generation of stress, a thermal generation takes place that produces tensile stress. Whenever residual stresses are discussed, the term includes the sum of the two generating mechanisms and their impact on the ground surface. In order to improve the properties of the gear, these stresses should be of a compressive nature.

CBN as a cutting material produces higher residual stresses at the surface than aluminum oxide. The reason for this phenomenon is the thermal conductivity of CBN, which is approximately 40 times higher than that of aluminum oxide. This reduces the thermal load of the ground zone because a bigger portion of heat can be transferred out through the wheel body and particles of removed material. An additional effect arises from the greater protrusion of electroplated grain particles, which supports the free cutting ability of CBN and allows more coolant to be transported through the contact zone. The result of grinding wheel wear of both CBN and aluminum oxide is the loss of sharpness or cutting ability of the individual grain particles. Therefore, a greater amount of heat is generated in the process. However, the electroplated CBN wheels at the end of their useful lives still produce higher compressive residual stresses than a newly dressed aluminum oxide wheel. The CBN wheel can be economically replated many times, providing consistent and repeatable form.

Conclusion

Current trends and future demands for grinding technology are to produce power transmissions that have

- Improved energy efficiency,
- Higher power capacity,
- Lower weight and volume,
- Lower noise levels generated by gear mesh,
- Lower cost,
- Extended life.

Such goals can be achieved only through the higher overall system accuracy and improved quality of material. Research and development of grinding technology have to address these demands. CBN electroplated grinding tools have the best potential to reach these demands.

Gear designers are developing sophisticated and complex modifications of profile and lead in order to eliminate the vibrations and noise generated by the gear train. In addition to the geometrical requirements, it is necessary to develop the shortest possible grinding times for economic reasons. Close coorperation between gear designers and grinding process designers is therefore essential to the successful production of CBN-ground gears. ✪

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New Guidelines For Wind Turbine Gearboxes

Robert Errichello & Brian McNiff

Introduction

The wind turbine industry has been plagued with gearbox failures, which cause repair costs, legal expenses, lost energy production and environmental pollution.

A wind turbine gearbox has failed if it:

- Jams and stops rotating;
- Breaks and allows the rotor to turn without turning the generator;
- Exceeds allowable sound level or vibration limits;
- Has excessive oil leakage;
- Requires excessive maintenance.

Failed gears, shafts, keys, bearings and housings have caused jamming or breaking. Bending fatigue is usually the ultimate failure mode when gearboxes jam or break. However, bending fatigue is often a secondary failure mode initiated by other failure modes such as macropitting or scuffing. Some wind turbines have been destroyed when gearboxes broke and allowed rotors to overspeed.

Strict sound regulations in the United States and Europe require shutdown of noisy turbines because failed gears or bearings cause excessive sound or vibration.

Failed shafts, seals and housings cause excessive oil leakage and, in some cases, loss of lubricant and gearbox failure. Less severe leakage increases maintenance and costs. Soil contamination requires costly cleanup.

Three years ago, a group of individuals dedicated to wind power recognized that many gearbox failures were due to a lack of understanding of the severity of the wind turbine operating environment. They sought to define this environment and improve communication between gear and wind turbine manufacturers so reliable gearboxes could be obtained.

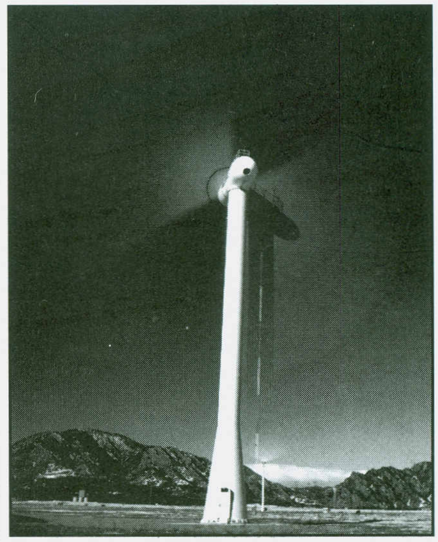
Therefore, the American Gear Manufacturers Association (AGMA) and the American Wind Energy Association (AWEA) formed a committee of wind turbine manufacturers and operators, researchers, consultants and gear and lubricant manufacturers.

The committee developed AGMA/AWEA 921-A97, "Recommended Practices for Design and Specification of Gearboxes for Wind Turbine Generator Systems." The document reflects the latest knowledge about wind loads and failures, plus the insights of gear and wind turbine manufacturers and operators. It describes wind turbine configurations, operating conditions and environmental factors that affect gearbox life. It offers guidelines for defining wind loads and specifying gears, bearings, operation and maintenance. All of these guidelines focus on one goal: ensuring reliable wind turbine gearboxes.

Wind Turbine Configuration and Operation

Two wind turbines with the same generator can experience extremely different gearbox loads. A good control system maximizes energy capture and minimizes structural loads. AGMA/AWEA 921 describes common wind turbines and the significance of their architecture and operation on gearbox loads.

Rotors can be horizontal, as in a HAWT (horizontal-axis wind turbine); or vertical, as in a vertical-axis wind turbine (VAWT). Some HAWTs are integrated systems where the gearbox supports the rotor and, in some cases, the generator and other components such as yaw drives. With this configuration, the gearbox housing must transmit rotor loads to the supporting tower without incurring excessive stresses or deflections. Integrated systems require detailed communication between the wind turbine designer and gear manufacturer to properly design interfaces.



Turbine at the National Wind Technology Center, Boulder, CO.

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is president of McNiff Light Industry of Blue Hill, ME, a company providing wind turbine testing and analysis services.

By contrast, a modular system consists of a rotor, rotor support shaft and bearings, gearbox and generator, each mounted to a common support plate. In this case, the gearbox doesn't need to support external loads.

Rotor torque is controlled by adjusting blade pitch (active control) or by aerodynamic stall (passive control). Active control increases gearbox complexity because, in many cases, the mechanical, electrical or hydraulic control passes through a hollow gearbox shaft to blade actuators. Passive control is simpler, but creates higher peak torques than active control.

Environmental Considerations

Wind turbines must withstand aggressive environments ranging from hot, dusty deserts to cold, wet marine environments.

Temperatures can vary widely, which affects lubrication. The oil sump temperature should be at least 5°C above the lubricant pour point during start-up and less than 95°C during operation. Otherwise, heaters or coolers may be necessary.

All sites require high-capacity, filtered breathers and positive shaft seals to control contamination. Desiccant breathers minimize water contamination. Protective coatings help prevent corrosion.

Defining Gearbox Loads

Many early wind turbine gearboxes failed because designers were uncertain of loads such as:

- Long periods of small oscillation when the brake stops the high-speed shaft, and the wind buffets the rotor;
- Long periods of low speed and light loads during light winds;
- Long periods of high speed and light loads when winds are below cut-in speed;
- High transient loads when the generator connects to the power grid;
- Rapid load fluctuations during normal operation;
- High transient and impact loads during braking.

Defining load spectra for these conditions is difficult due to the uncertainty of predicting loads. However, experience has made predictions more reliable. AGMA/AWEA 921 tells how to assemble load spectra that include both wind loads and transient loads, such as start-up synchronization, rapid blade pitch, actuation of blade control surfaces and braking.

Until recently, wind turbines were not classified according to their capabilities. However, the International Electrotechnical Commission's IEC 1400 initiative has established four classes based on wind severity. A Class 1 wind turbine, for example, can operate in the most severe conditions, with an average wind speed of 10 m/s, an extreme wind speed of 50 m/s and high turbulence intensity.

Specifying Gearbox Components

Experience has shown which gearbox components perform best in wind turbines. The following material summarizes recommendations for gears, bearings and lubrication systems.

Gears. Wind turbine gears are subjected to wind loads that vary from light to very heavy (wind gusts). Carburized, hardened and ground gears have proven reliable under widely varying loads.

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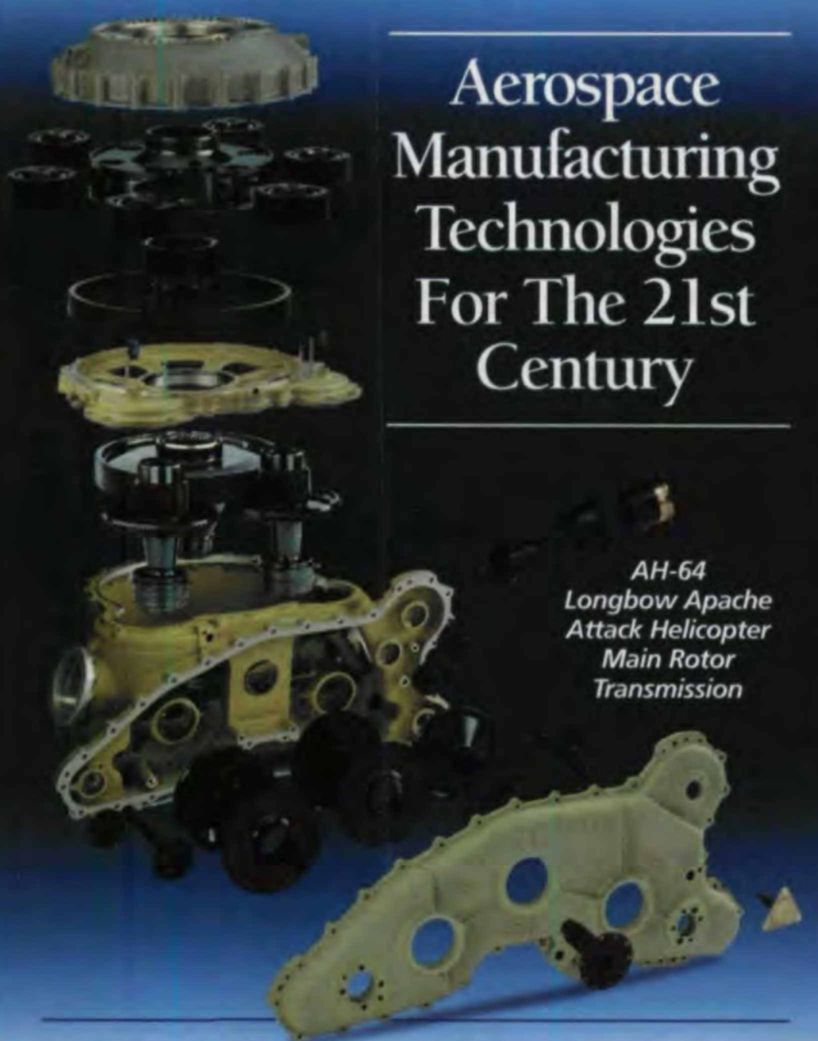

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Induction-hardened gears were tried, but lacked load capacity because of inadequate accuracy. Carburized and ground gears are the best choice because they have the necessary load capacity and accuracy.

Nitriding is not appropriate for external gears for wind turbines because it has poor resistance to overloads. It has been successful for annulus gears.

Through-hardened gears do not have sufficient load capacity because of their relatively low hardness. In addition, when run with carburized pinions, they promote polishing wear.

Metallurgical quality should meet requirements for Grade 2 material in accordance with ANSI/AGMA 2001-C95.

Gear teeth should be designed to maximize load capacity as explained in AGMA 901-A92. For example,

- Use at least 20 teeth on pinions to achieve good balance between pitting resistance, bending strength and scuffing resistance;
- *Because wind turbine gearboxes are* speed increasers, design profile shift for balanced specific sliding;
- To achieve uniform gear tooth load distribution, use a low aspect ratio (pinion face width to diameter ratio) and modify profiles and helices to compensate for deflections and manufacturing variations;
- To obtain quiet operation, use gears with high contact ratios and high accuracy (AGMA Quality No. 11 or higher).

Preferred gear types for wind turbines are spur, helical and double helical.

Helical gears are quieter than spur gears and have more load capacity. Double helical gears may balance thrust loads; however, they require high accuracy to control dynamic loads, and shaft couplings must be designed and applied to limit external thrust loads.

Spur gears are often used in epicyclic gearboxes because they avoid the tendency of some helical gears to misalign planet gears and interfere with floating sun gears. A combination of spur gears in the low-speed epicyclic stages and helical gears in the high-speed stage provides a compact, quiet gearbox.

As power increases, there are incentives to use epicyclic, split-power-path or hybrid gear arrangements to meet load capacity while limiting size and weight.

Bearings. Severe vibration in wind turbines precludes standard industrial practice for fitting bearing outer races and housing bores. Vibration may cause outer races to spin within their bores, causing wear that misaligns gears, generates wear debris, damages gears and bearings and contaminates lubricant. The worst damage occurs in steel housings because steel-on-steel sliding between outer races and the housing causes scuffing and severe wear. Damage may be less in cast iron housings because dissimilar materials are less likely to scuff. Nevertheless, for reliable wind turbine service, bearing fits must be tight, or races must be mechanically restrained from spinning.

Preferred bearing types for wind turbine gearboxes are spherical roller, tapered roller, double-row, cylindrical

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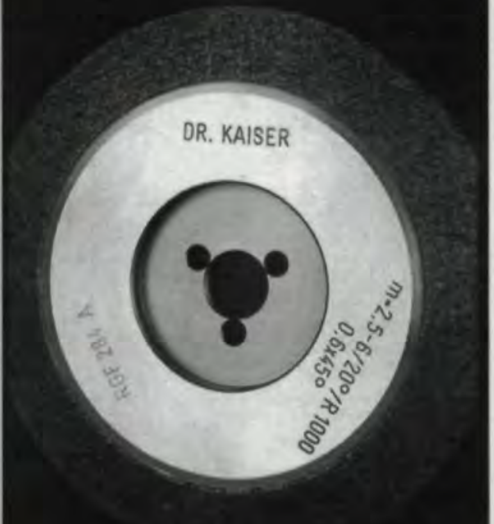

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roller with retainer, and ball.

Use single-row, tapered roller bearings only if end play is held within acceptable limits under all loads and operating conditions. Otherwise, excessive end play misaligns gears and may cause gear and bearing failures. Measure end play *in situ* so all factors affecting end play, including load deformations and thermal growths, are accounted for.

Full-complement, cylindrical roller bearings offer high load capacity. However, rollers have counterformal contact and opposed sliding that disrupt the lubricant film, allow steel-on-steel contact, create friction and make rollers prone to scuffing. Therefore, use these bearings only on low-speed shafts where sliding speeds are low enough to avoid scuffing.

Wind turbines frequently rotate at full speed under light loads. Under these conditions, bearing loads may be insufficient to maintain rolling contact between rolling elements and raceways. As a result, skidding, overheating and scuffing may cause bearings to fail. The highest risk occurs on high-speed shafts with four-point or angular contact ball bearings. Therefore, specify preload if necessary to prevent skidding.

Wind turbines with high-speed shaft brakes subject a gearbox to reversing loads during a brake stop. Therefore, minimize bearing end play to reduce impact loads on bearings caused by reversing thrust loads from helical gears.

Other tips:

- Equip bearings with bronze retainers. Plastic retainers are susceptible to contamination by hard particles that abrade rolling elements.
- For shaft-mounted gearboxes, minimize radial clearance of low-speed bearings to avoid misaligning the low-speed gear mesh. Modify the helix of the low-speed pinion or gear to compensate for misalignment.
- Except for epicyclic gearboxes, where the sun pinion has no bearings, mount all pinions between bearings. Don't use overhung pinions; they may cause gear misalignment and high bearing loads.

Lubrication

Micropitting, macropitting, adhesion, scuffing, abrasion and other lubrication-related failures are widespread in wind turbine gearboxes. To avoid such failures, the lubrication system should provide an adequate amount of cool, clean and dry lubricant to gears and bearings.

Pressure-fed systems should have a filter to clean the oil and can have a heat exchanger to cool the oil.

Maintenance and condition monitoring should be considered at the design stage.

Many factors need to be considered when selecting a lubricant, including viscosity, viscosity index, pour point, additives, micropitting resistance, scuffing resistance and cost. There are debates on the relative advantages of synthetic vs. mineral oils, especially regarding micropitting resistance. Operators have learned that experiments are required to find acceptable lubricants.

Wind turbine gears have relatively

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low pitch line velocity (typically 1–10 m/s) and high loads (contact stress over 1400 MPa). These conditions require synthetic or mineral gear oil with antiscuff additives and the highest practical viscosity (at least ISO VG 320). Too low a viscosity causes micropitting, macropitting, adhesive wear and scuffing.

Micropitting is prevalent in wind turbine gearboxes. Though it does not always lead to catastrophic failure, it generally reduces gear tooth accuracy, increases noise and can progress to full-scale macropitting and gear failure.

To prevent micropitting, maximize specific film thickness by using gears with smooth tooth surfaces and an adequate amount of cool, clean and dry lubricant of the highest viscosity permissible. You may need to experiment with different lubricants to find one with adequate micropitting resistance. FVA Project Number 54 (1) is a standardized test for micropitting resistance of lubricants. A ten-stage pass is minimum acceptable performance.

Running-in gears under controlled loads reduces tooth surface roughness and risks of micropitting and scuffing. However, manufacturers and purchasers disagree on who should perform this task. Gear manufacturers are reluctant to run-in gears because their test facilities are limited; and purchasers are reluctant because they can't always control loads. Therefore, run-in must be negotiated between the gear manufacturer and the purchaser.

Quality Assurance

The purchaser should write a procurement specification for wind turbine gearboxes that fully describes the application, load spectrum and minimum requirements for design, manufacturing, quality assurance, testing and gearbox performance.

The gear manufacturer should prepare plans for quality assurance, inspections and tests.

The purchaser should confirm that the gear manufacturer understands all requirements of the procurement specification before awarding the contract. Criteria for judging gear manufacturer competence are design review meetings, evaluations of manufacturing and QA plans and visits to manufacturing facilities.

To compare competing gearbox designs, AGMA/AWEA 921 recommends these standards for rating the components:

- Gearbox: ANSI/AGMA 6001, ANSI/AGMA 6010 or ANSI/AGMA 6023.
- Gears: ANSI/AGMA 2001.
- Bearings: ANSI/AFBMA 11.
- Shafts and keys: ANSI/AGMA 6001.

Plans should clearly define all acceptance criteria before manufacturing starts. After awarding the contract, the purchaser should audit manufacturing, inspection and testing for conformance to the procurement specification.

Prototype gearboxes should be qualified before full-scale production begins. Prototype gearboxes should not be manufactured until the purchaser approves all elements of first-article inspections and tests. After prototypes have met the requirements of the procurement specification, and manufacturing plans are

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sound, the transition to production should be straightforward.

Operation and Maintenance

The gear manufacturer, lubricant supplier and purchaser should establish plans for start-up, operation and maintenance before gearboxes are placed in service. Gearboxes should be run-in under controlled loads to reduce the risk of scuffing and micropitting and prolong gear and bearing lives.

Regular gearbox inspections should be done at run-in and at specified intervals. AGMA/AWEA 9021 gives guidelines for on-site monitoring and laboratory analyses of lubricants for viscosity, water content, acid number, solid contaminants and additive depletion. It also describes methods for monitoring vibration and temperature.

Safety is a major concern for wind turbine operators because of the risk of falling, being trapped in rotating components or being injured while maneuvering in restricted spaces. Gearboxes should have an adequate number of steps, handholds and lanyard hook points. Housings should have rounded corners, and all exposed rotating components should have safety guards. Because access is difficult and work space is limited, design the gearbox to ease maintenance. ○

References:

1. FVA-Information Sheet, "Micropitting," No. 54/7, July, 1993. Forschungsvereinigung Antriebstechnik e.V., Lyoner Strasse 18, D-60428, Frankfurt/Main, Germany.

Acknowledgement: *This article was presented at the 1997 AGMA Fall Technical Meeting in San Diego, CA. ©1997, AGMA. Used with permission.*

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AGMA and ISO Accuracy Standards

Robert E. Smith

Background

The American Gear Manufacturers Association (AGMA) is accredited by the American National Standards Institute (ANSI) to write all U.S. standards on gearing. However, in response to the growing interest in a global marketplace, AGMA became involved with the International Standards Organization (ISO) several years ago, first as an observer in the late 1970s and then as a participant, starting in the early 1980s. In 1993, AGMA became Secretariat (or administrator) for Technical Committee 60 of ISO, which administers ISO gear standards development.

ISO Organization

ISO has over 215 technical committees, with members from over 75 nations. Technical Committee 60 (TC 60) has several working groups (WGs) assigned to different standards, such as gear accuracy, gear ratings, nomenclature, etc. The gear accuracy standards were developed by WG 2. In addition to TC 60 and WG 2, AGMA convenes (chairs) about 20% of the working groups in TC 60.

ISO Accuracy Standards

The pre-existing ISO standard for gear accuracy is known as 1328-1975. This is a single document, covering all aspects of cylindrical gear accuracy.

The new standard has been split into several documents; tangential tolerances, radial tolerances, and several information sheets. The first two, 1328-1 and 1328-2, are standards, and the information sheets are known as Technical Reports (TRs). All have been approved by member nations, and most are currently in print.

The new documents are known as:

1. *ISO 1328-1. Cylindrical Gears–ISO System of Accuracy. Part 1: Definitions and Allowable Values of Deviations Relevant to Corresponding Flanks of Gear Teeth.* This standard deals with pitch variations (single and cumulative), profile (involute), helix (lead) and tangential composite variations (single flank).

2. *ISO 1328-2. Cylindrical Gears–ISO System of Accuracy. Part 2: Definitions and Allowable Values of Deviations Relevant to Radial Composite Deviations and Runout Information.* This standard deals with radial composite variations (double flank) and radial runout.

3. *ISO TR 10064-1. Cylindrical Gears–Code of Inspection Practice. Part 1: Inspection of Corresponding Flanks of Gear Teeth.* This Technical Report deals with methods of measurement and interpretation of data relative to 1328 Part 1.

4. *ISO TR 10064-2. Cylindrical Gears–Code of Inspection Practice. Part 2: Inspection Related to Radial Composite Deviations, Runout, Tooth Thickness and Backlash.* This Technical Report deals with the methods of measurement and interpretation of data relative to 1328-2. It also covers tooth thickness and backlash.

5. *ISO TR 10064-3. Cylindrical Gears–Code of Inspection Practice. Part 3: Recommendations Relative to Gear Blanks, Shaft Center Distance and Parallelism of Axes.*

6. *ISO TR 10064-4. Cylindrical Gears–Code of Inspection Practice. Part 4: Recommendations Relative to Surface Texture and Tooth Contact Pattern Checking.*

AGMA to ISO Differences

The current accuracy standard for cylindrical gears in AGMA is ANSI/AGMA 2000-A88. This was issued in 1988 and used the same tolerances as the previous AGMA 390.03 standard. Some of the notable differences between the AGMA and ISO standards are these:

1. Numbering System. The AGMA system of numbering for different classes of quality is from Q3 through Q15, in order of increasing precision. In other words, the higher the number, the higher the quality of accuracy (smaller tolerance). The ISO system is just the opposite. It consists of 13 accuracy grades of which 0 is the smallest tolerance and grade 12 is the lowest accuracy or largest tolerance. **A word of caution:** When specifying an ISO accuracy class, one must specify per ISO 1328 Part 1 or 2 with its date 1996 or 1997, respectively. The tolerances specified in these new replacement standards are different from the ones specified in the old ISO 1328-1975 for the same class numbers.

It is impossible to define a direct comparison between any AGMA and ISO accuracy grade. For example, one might find the same tolerance for one parameter, such as pitch variation, at a given module and diameter. Yet the tolerance will be different at other modules or diameters for the same class. A rule of thumb that can be *used with caution* is to subtract an

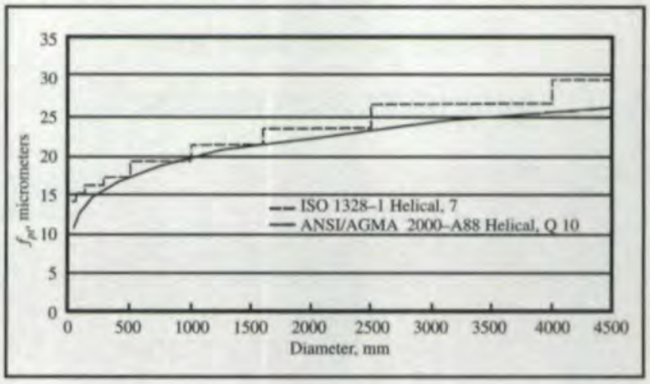

Fig. 1 – Tolerance comparison, single pitch variation, f_{pt}, AGMA Q 10 and ISO grade 7, module 8.

AGMA or ISO class number from 17 to find the corresponding class in the other system. This should only be used to get the mind thinking in a similar *range* of accuracy, not a specific class. For example, think of AGMA classes in different ranges of overall quality: 1) Low — Q3 to Q7, cast, molded, some cut; 2) Medium — Q8 to Q9, most cut; 3) High — Q10 to Q11, shaved and honed; 4) Precision — Q12 to Q15, ground. The above statement should be taken only in general terms. The "Rule of 17" should also be used in these general terms only.

2. Format of Standards. In an AGMA or ANSI document, the standard portion is a legally binding requirement. Anything that appears in an appendix or annex is for information only and is not legally binding. In ISO documents, a standard such as ISO 1328-1 has three categories of information. The main body of the standard is legally binding the same way as in AGMA. However, there are two types of annexes: Informative and normative. An informative annex is for information only and is not legally binding. A normative annex can contain requirements, such as tolerances, that can become binding if the methods or parameters within the annex are specified.

AGMA also issues information sheets. The Technical Reports associated with the 1328, Parts 1 and 2 standards are called Type 3 and might be considered more like AGMA information sheets.

3. Tables vs. Formulas. AGMA 2000 A-88 uses formulas for the basis of tolerances. These tolerances are then based on a continuous curve. Tables of tolerances are included only as a convenience for the shop users. The tolerances in ISO 1328-1 are based on the tables, rather than a formula. Formulas are included, but are based on ranges of modules (m), diameters (d) and face width (b), as shown in the tables. When applying the formulas, the given parameters m, d and b are to be introduced as the geometric mean values of the relevant range limits and not the actual values. If, for example, the actual module is 7, the range limits are $m = 6$ and $m = 10$, and the allowable deviations are calculated with $m = (6 \cdot 10)^{0.5} = 7.746$. This causes an unusual looking graph of tolerances with steps in it. (See Fig. 1). This also causes some unusual circumstances, where a minute change in diameter or module can cause a big change in tolerance if the change moves the example gear into the next step.

4. Runout and Accumulated (Total Cumulative) Pitch. AGMA 2000 A-88 uses pitch variation and radial runout for the control of the short and long term accuracy of the placement of teeth around a gear. Pitch variation is a tangential measurement, and runout is a radial measurement. ISO 1328-1 uses single pitch variation and total cumulative pitch variation. Both of these parameters are tangential in nature. Radial runout has been relegated to an informative annex in 1328-2. In the author's opinion, the ISO approach is better. Gears function tangentially, not radially. Also, after a gear has had a subsequent finishing operation, such as shaving, or with some methods of grinding, runout can be a very deceiving parameter of gear quality or accuracy. The finishing operation can reduce runout to a minimum, yet still leave a significant amount of accumulated pitch variation. (See AGMA Technical Paper 95-

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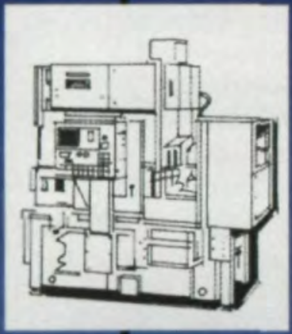

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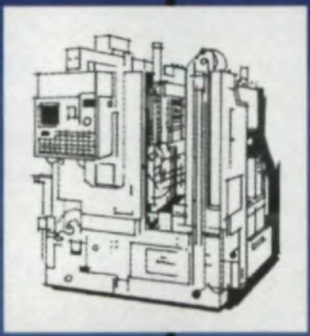

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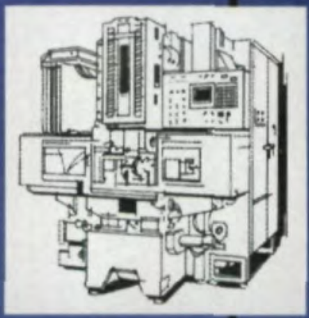

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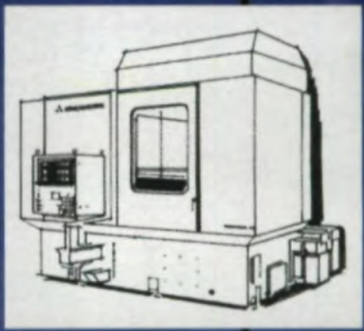

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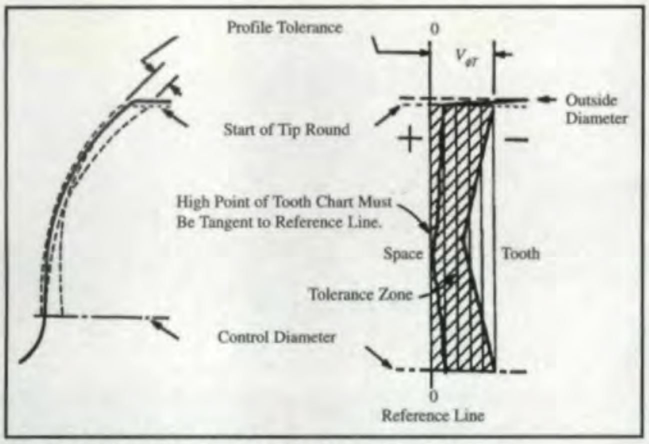

Fig. 2 – AGMA "K" Chart method of profile evaluation.

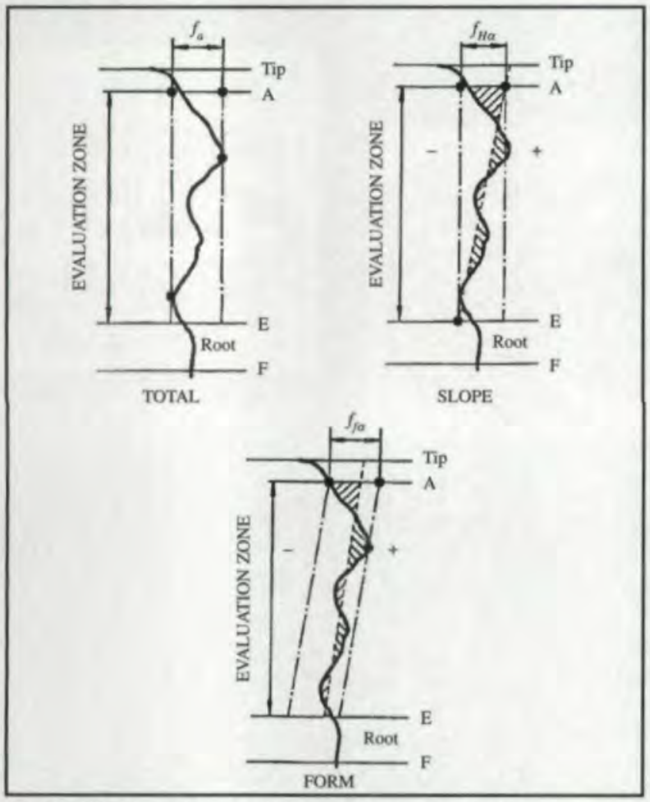

Fig. 3 – ISO Total, Form and Slope methods of evaluation.

FTM 1 by R. E. Smith, *et al.*, "Detection of Hidden Runout," for further details on this subject.)

5. "K" Charts vs. Slope, Form and Total. AGMA 2000 A-88 uses the familiar "K" charts for the evaluation of involute and tooth alignment (lead) data (see Fig. 2). ISO 1328-1 uses a rectangular tolerance zone in the main body of the standard for "total" deviation. However, an informative annex uses "slope" and "form." The problem with AGMA "K" charts and the rectangular tolerance zone of ISO is that they are like "go/no go" gages. They are not useful for today's statistical process control techniques. Also, the zones are wide enough to allow for the effects of runout on the profile or lead errors. For this reason, they are not very discriminating as far as determining the real involute or lead errors, independent of runout. The technique described in ISO 1328-1 allows the determination of slope and form deviations. These are useful for the determination of factors that have an influence on functional characteristics of the gears. For example, slope and mean slope are useful for the control of noise excited by gears.

The method of determining slope and form is described in ISO 1328-1, and the suggested tolerances are listed in an informative annex of the standard. For unmodified tooth forms, they are determined by using a least-squares-straight-line fit to the actual trace within a specified evaluation zone (see Fig. 3). The slope deviation ($f_{H\alpha}$) is the slope of the least squares line. The form deviation ($f_{f\alpha}$) is the bandwidth of the irregularities of the trace around the slope line. The total is F_α. If the tooth is a modified tooth form, other techniques for the best fit line are used. See ISO 1328-1 for a more detailed description.

6. Comparative Lead Tolerances. Tooth alignment (lead) is a very important parameter relative to life on highly loaded gears. For large diameter, higher accuracy gears, ISO is more restrictive on allowable lead tolerances. In addition, lead (or helix) tolerances are specified in the transverse plane in ISO 1328-1, while AGMA 2000 A-88 specifies them normal to the helix. Both are given in the base tangent plane.

Adoption of ISO Standards in the U.S.

Where do we go as far as adoption of ISO gear accuracy standards? The AGMA Inspection and Handbook Committee is now addressing that question. It is currently working on a bevel gear accuracy standard. This is AGMA 2009 AXX. Even though ISO has no bevel gear tolerance standards, the AGMA committee has endeavored to follow ISO practice for cylindrical gears as much as possible. So far, this practice has not met with complete agreement. The committee is now starting to rewrite AGMA 2000-A88. Again, the concept is to try to use ISO as much as possible.

Both ISO and AGMA are consensus standards. This means that they are arrived at by consensus of a large number of participants. This doesn't mean that everyone agrees with everything that results. For example, the AGMA committee feels that the ISO standard is too restrictive with some tolerances in the very small diameter or fine pitch region. The current feeling is that the tolerances are not relevant to current capabilities or practice. At this time, the outcome of this disagreement is not known. As far as the rewrite of AGMA 2000-A88, the work is just beginning. Whether AGMA decides to adopt ISO or a modified version of it remains to be seen.

Committee Participation

If anyone has strong feelings about standards issues, he or she should participate in their development. Everyone should feel free to get his interest on the table for consideration. It is also a great learning experience to participate on such committees. For further information, call AGMA Headquarters, 703-684-0211, attn. Bill Bradley. ○

Robert E. Smith *is the principal in R. E. Smith & Co., gear consultants of Rochester, NY, and one of* Gear Technology*'s technical editors. He has over 40 years of experience in gearing and is the author of numerous papers and articles. He is also very active in AGMA standards development.*

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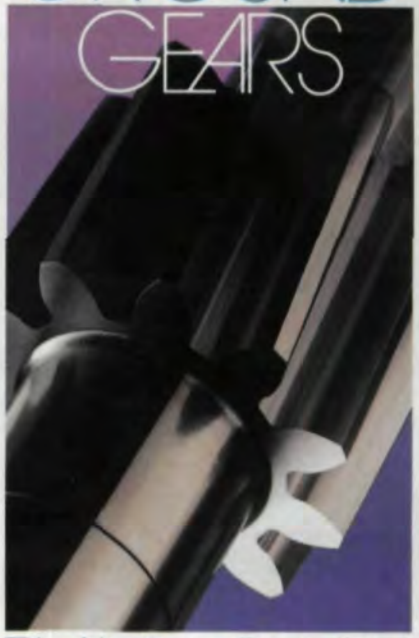

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CURDWORTH ACQUIRES SYKES IN U.K.

W. E. Sykes, specialist in the design and manufacture of gear generating machines, has been purchased by Curdworth Engineering (UK) Ltd. As of December, 1997, control of the manufacture, supply and sales of the Sykes line of machines, service and spare parts, passed to Curdworth's control. Curdworth will now offer a range of gear hobbing and shaping machines in addition to its other products.

MACSTEEL® ADDS TWO PLANTS

Macsteel has recently added Nitro-Steel in Pleasant Prairie, WI, and the Heat Treating Division in Huntington, IN, to its operations. The NitroSteel Division offers carbon and alloy steel bars and tubes surface hardened by the Nitrotec process, which gives the final product a rich, ebonized finish that is an alternative to hard chrome plating. The Heat Treating Division is a state-of-the-art induction heating facility specializing in bar and tube heat treating.

MITSUBISHI TO DISTRIBUTE INNOVATIVE HEAT TREAT TECHNOLOGY

Mitsubishi International has announ-ced that it has reached agreement with UK-based CoreFlux Systems to act as exclusive worldwide distributor of CoreFlux's Magnetic Field Technology. This patented process through-heats metal components up to 2000°F with ±2° accuracy and virtually no thermal gradient. Magnetic Field Technology generates heat by using hysteresis losses that result from stressing ferromagnetic and paramagnetic parts under a fluctuating magnetic field. Unlike conventional furnace and induction heating, which rely on thermal conductivity, hysteresis losses spread rapidly and evenly through the entire component, thereby avoiding skin effects and surface overheating.

VALENITE NORTH AMERICA ISO 9001 RECERTIFIED

Valenite Inc., a major supplier of cutting tools, has announced that all eight of its manufacturing sites and its six sales offices in North America have been ISO-9001 recertified. Valenite offers a complete line of standard and special indexable insert turning, boring, milling and drilling products. It also manufactures cutting fluids, carbide die and wear parts, high-speed steel cutting tools, special high-production tooling and electronic gaging systems.

FEINTOOL QS-9000 AND ISO 9002 CERTIFIED

Feintool New York, Inc., contract maker of fineblanked parts, has received QS-9000 and ISO 9002 certification. Feintool custom-engineered tooling and triple-action presses create complex precision parts in "one hit," enabling special shapes and exceptional tolerances at production economies. Automotive parts account for approximately 35% of the White Plains, New York plant's production. Other key users of fineblanked precision parts are the computer, office machinery and electronics industries.

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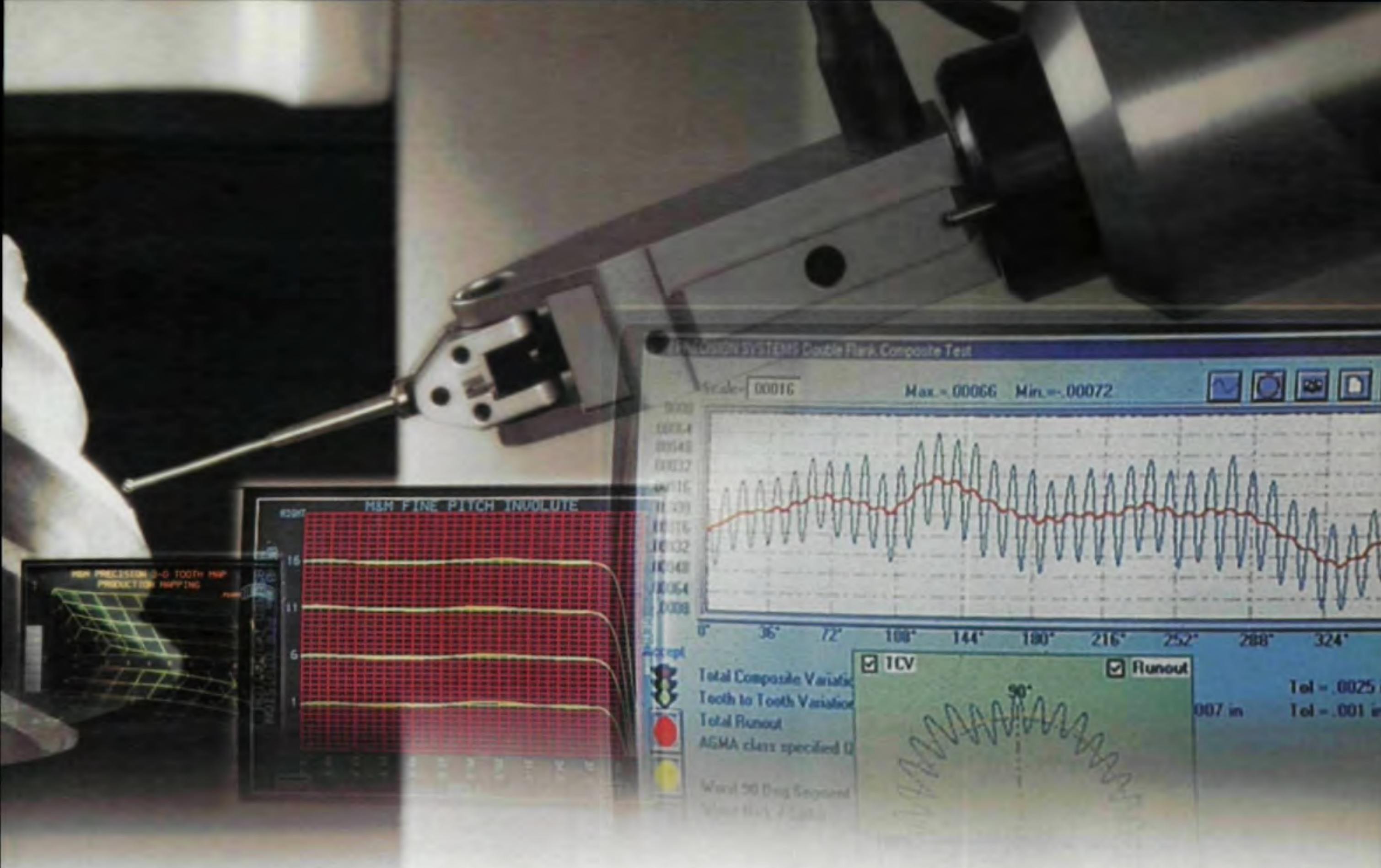

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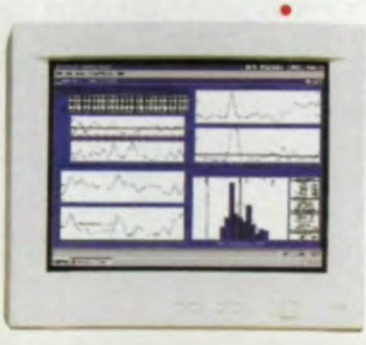

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CIRCLE 140

Relationship Between Wear and Pitting Phenomena in Worm Gears

Michel Octrue

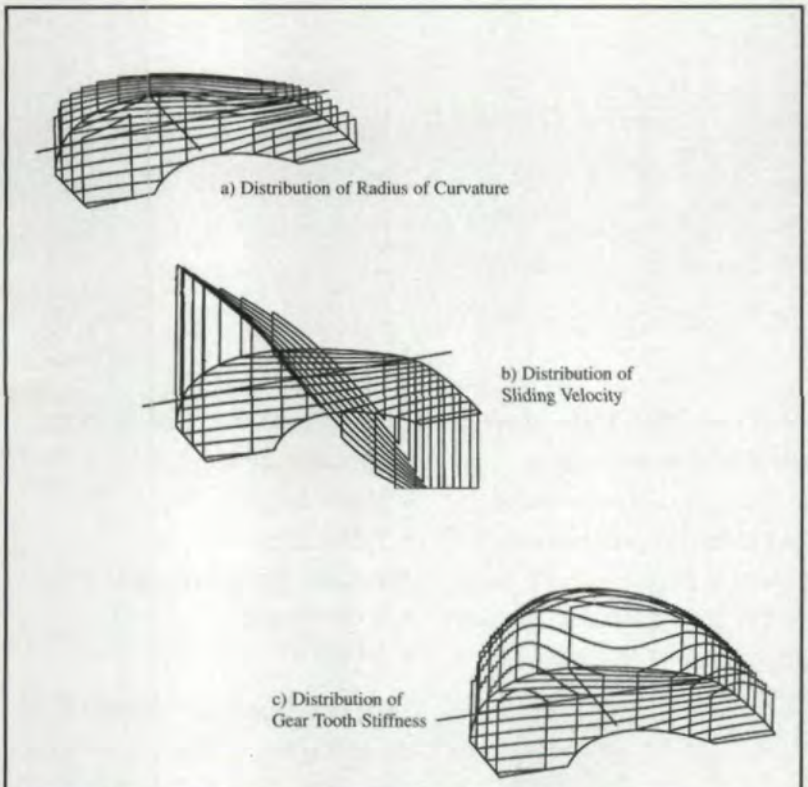

Fig. 1 — Physical parameters' distribution along lines of contact influencing wear phenomena in worm gears (Gear Ratio: 3/31).

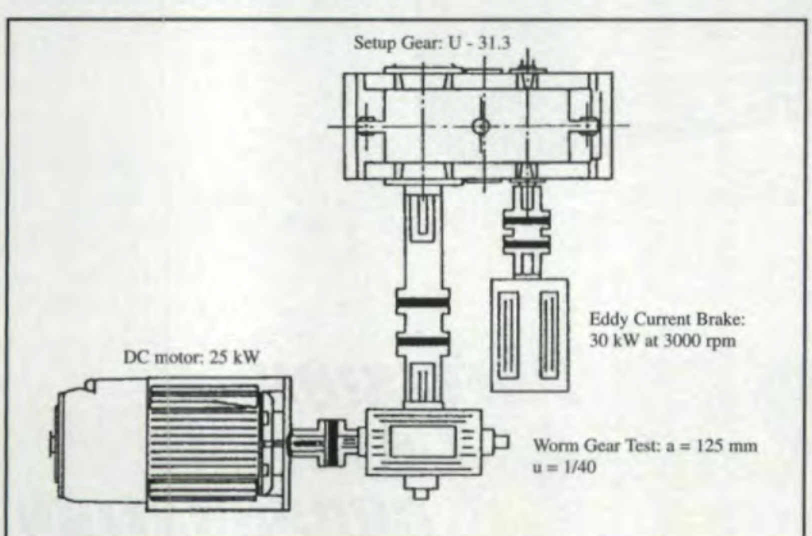

Fig. 2 — Worm gear test rig.

Introduction

Worm gears display unique behavior of surfaces because of the presence of wear phenomena in addition to contact pressure phenomena.

Physical Phenomena. In worm gears the contact between the two surfaces of the teeth, the worm threads and the worm wheel teeth, evolves along lines of contact with a sliding velocity induced by the translation to the worm threads and the rotation of the worm wheel and the rotation of the worm. This last element provides the major component of sliding phenomena.

Consequently, unlike the case with cylindrical gears, it is difficult to separate the usual effects caused by contact pressure, which stresses the subsurface of teeth, and the wear effects caused by this important sliding phenomenon which stresses only the surfaces of the teeth.

State of Calculation Methods. Calculation methods or standards (Refs. 2–3) used for worm gears define two main criteria for rating: contact pressure or wear and tooth bending. Comparison of these criteria gives, in general, a torque limitation caused by pressure or wear.

The complexity of the geometry of worm gears has led to the establishment of empirical rating methods.

More recently, the use of computers has allowed a more theoretical approach, taking into account physical phenomena such as contact pressure and elastohydrodynamic lubrication induced in worm gear meshes. These methods are classified as **analytical**.

Nevertheless, the wear phenomena inherent in worm gears are still too complex to be approached by theory. This results in the different conditions of sliding velocity, radius of curvature and local stiffness encountered along the lines of contact between a worm and a worm wheel (Fig. 1). This is why it is necessary to characterize wear phenomena quantitatively according to experimental results and observations.

Experimental Approach

For the last 12 years, CETIM has performed several endurance tests on worm gears. These tests were of long duration in order to reach a significant number of cycles of meshing on worm wheel teeth and to give pertinent data on the behavior of tooth surfaces.

To reach this goal, CETIM designed and developed three test rigs to test worm gear reducers.

Test Rigs' Description. Each test rig consists of a variable-speed, DC motor that drives the worm of the reducer to be tested with a torque meter. The worm wheel of the reducer is connected to an increaser that drives an eddy current brake used to apply the load. The brake is set up in balance in order to measure the applied torque (Fig. 2).

Each test rig is monitored, and the following parameters are measured continuously:

- Rotational speed of the worm,
- Torque on the input shaft of the worm,
- Applied torque on the output,
- Oil temperature in the sump,
- Oil temperature near the more heavily loaded worm bearing,
- The instantaneous wear of the worm gear.

The lubrication systems of worm gear reducers are composed of a pump, a heat exchanger cooled by water and a tank for oil. A water circulation bypass is used for temperature regulation of oil.

The configuration of the worm gear reducer is such that the worm is above the worm wheel. The oil inlet is above the worm near the more heavily loaded worm bearing and the outlet is at the bottom of the housing.

Worm Gear Data. Each test rig corresponds to a worm gear reducer.

Testing Process

Assembling: The housing is cleaned, and the contact pattern of the worm gear is tracked with a soft blue dye slightly on the leaving side of the worm wheel and adjusted accordingly.

Running in: The lubricant system is filled with new oil, and the worm gear reducer is run at its nominal speed with a load increasing as follows:

First hour: 10% of nominal torque

Two hours: 25% of nominal torque

Fifty hours: 50% of nominal torque

Then the oil is changed, the initial tooth backlash is measured in four positions of the worm wheel and the contact pattern is checked.

Testing: The torque is fixed at the test value, and each 300 hours the test rig is stopped. A sample of oil is taken for analysis just after stopping the test rig. The temperature of the worm gear reducer is decreased to ambient temperature. The level of oil is decreased so the worm wheel teeth are no longer in contact. The backlash is measured in four positions on the worm wheel, and the contact pattern is checked. The oil level is reestablished, and the test rig is started again with a progressive loading in order to stabilize the temperature.

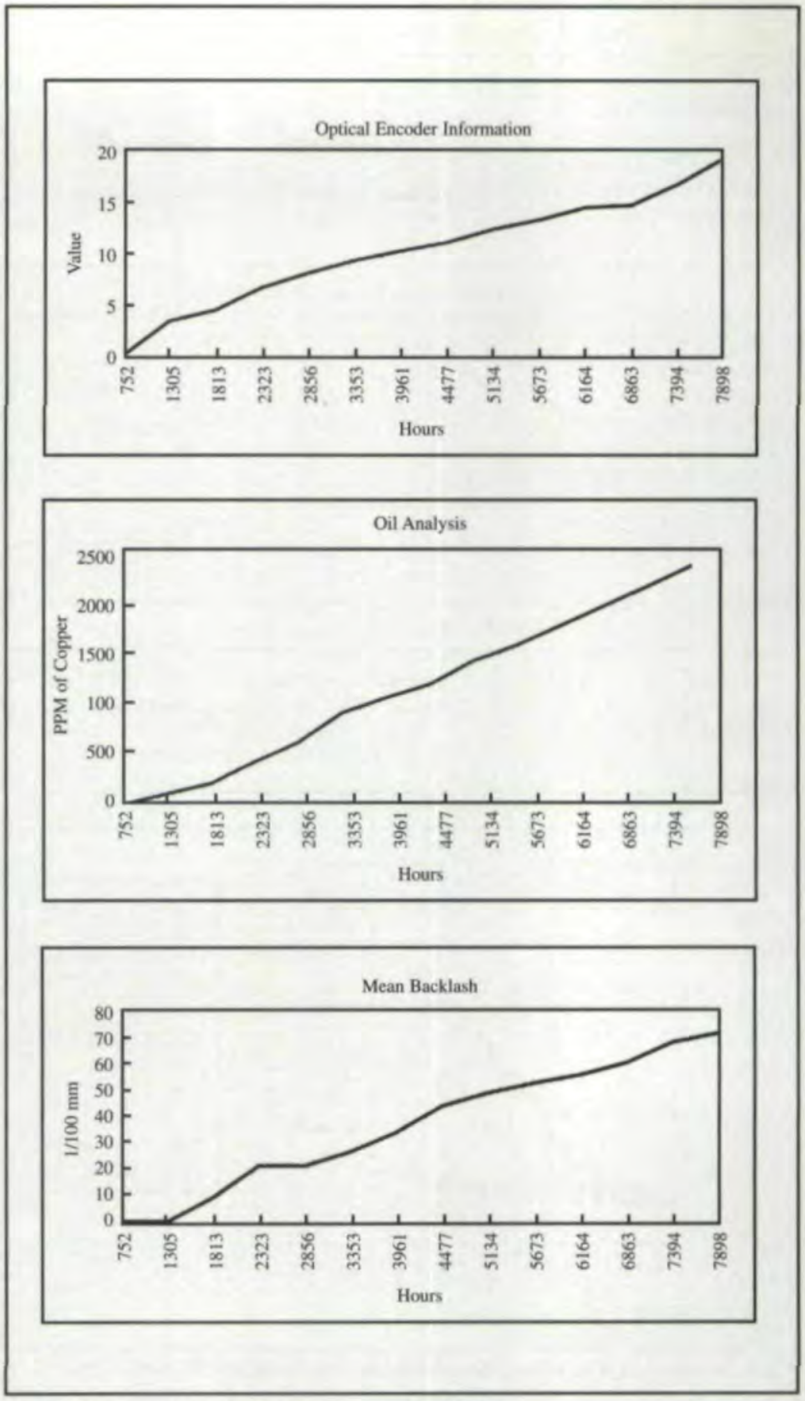

Fig. 3 — Comparison of wear measurement methods.

Evaluation of Wear on Worm Wheels

Wear is determined by two direct measurements:

- A continuous measurement made using two optical encoders linked on each shaft of the worm and worm wheel and connected to a specific board that calculates the variation of the relative

Dr. Michel Octrue

is manager of the gear department at the Centre Technique des Industries Mecaniques (CETIM) in Senlis, France. He is also the technical manager of the Institute de l'Engrenage et des Transmissions (IET) and president of the Technical Committee at EUROTRANS. He is the author of numerous books and papers on gearing subjects.

Table I — Characteristics of Worm Gear Test Rigs				
		A	B	C
Nominal Input Power	[kW]	5.5	18	31
Nominal of Input Velocity	[rpm]	1500	1500	1500
Maximum Input Power	[kW]	15	37	64
Min/Max Input Velocity	[rpm]	1600	1500	1500
Max Output Torque	[daN.m]	3000	2500	2000

Table II — Geometrical Data of Worm Gears				
		A	B	C
Center Distance	[mm]	125	125	125
Gear Ratio $u = z1/z2$		1/40	3/31	5/24
Axial Module	[mm]	4.95	6.45	8.36
Lead Angle [°]		5.4375	21.1369	40.2592
Normal Pressure Angle [°]		21.81	20.65	25
Worm Profile		A	A	A
Worm Wheel Face Width	[mm]	45	45	45
External Diameter of Worm Wheel	[mm]	220	220	220

Table III— Materials and Lubricant	
Worm Material	Case-Hardened Steel
Worm Wheel	Bronze with 12% of tin and x% of nickel
Lubricant	Synthetic Oil: Polyglycol Shell TIVELA A — 150 Cst at 40°C

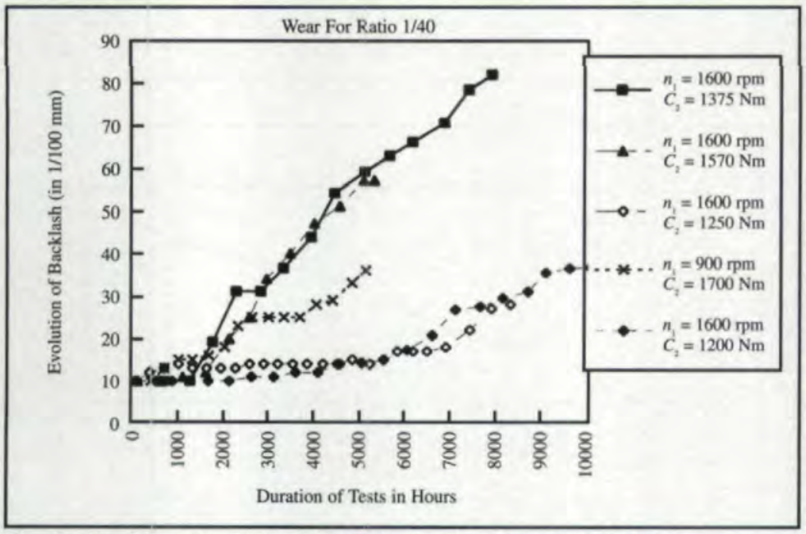

Fig. 4 — Evolution of wear for ratio 1/40 in function of number of hours.

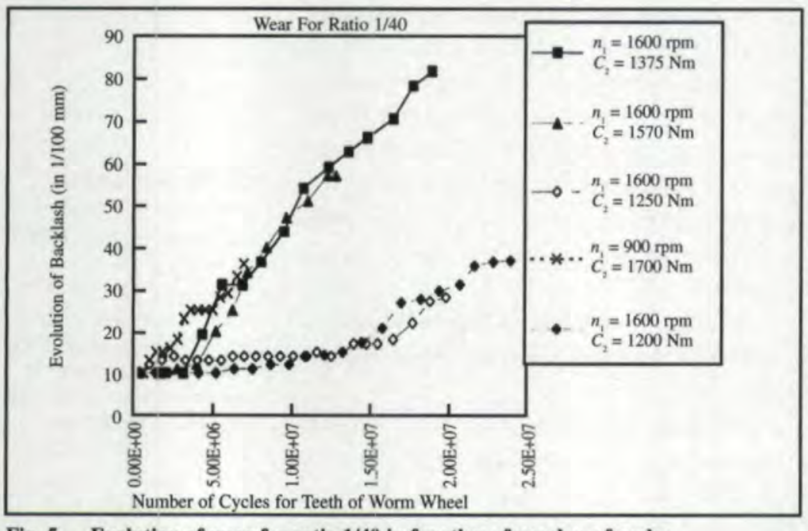

Fig. 5 — Evolution of wear for ratio 1/40 in function of number of cycles.

position of the worm for a defined space of the worm wheel teeth.

• A second measurement with the evaluation of backlash is made in four positions of the worm wheel at each stop of the test rig.

The last method takes into account the fact that the wear is not always a uniform phenomenon distributed on all worm wheel teeth. A mean value is deduced from the four measurements. (Note: In this article, the wear phenomena are discussed on the mean values.)

An indirect measurement of wear can also be obtained by the evaluation of the number of copper particles contained in the oil. This is obtained by the parts per million evaluated by oil analysis.

A comparison of these three methods is given in Fig. 3.

Results of Wear Measurement

Figs. 4–7 represent the evolution of wear interpreted as an increase of backlash. Only the variations of backlash are represented. Each curve corresponds to a specific working condition in terms of speed of the worm n_1 and torque applied on the worm wheel C_2.

For a 1/40 ratio, two graphs are plotted, one in functions of the duration of test; the second in functions of the number of cycles applied on the worm gear teeth (Figs. 4–5).

The representation in functions of the number of cycles gives a better coherence between the different working conditions. This is true for all ratios.

The observations of these curves show that for all loading conditions, a period of meshing exists during which no wear appears. It corresponds to the time it takes for the surfaces to adapt. Then the increase of backlash begins and evolves continuously.

This evolution is not really linear except for the 5/24 ratio; more often it evolves like stairs, and after the initiation of the phenomenon, there are periods of greater or lesser stability separated by periods during which wear evolves continuously.

Comparison With Predicted Results

The comparisons are done according to DIN 3996 standards. This standard proposes a calculation method to predict the evolution of backlash as a function of the duration of working.

These calculations have been done for each worm gear and for two working conditions (Figs. 8–10). (In these figures, the first number between brackets represents the rotational speed of the worm in rpm and the second, the applied torque on the worm wheel in mN.) The first observation is that the wear prediction in DIN 3996 is a linear function. The sensitivity of DIN 3996 is the same as for the experimental observations: If the torque increases, the wear increases. From the point of view of amplitude, theoretical predictions of wear and experimental results are not correlated: DIN 3996 predicts more wear for the ratio 1/40 than for the ratio 5/24. The experiment's observations show the contrary result.

Observations & Predictions

In order to explain these differences, qualitative observations of the evolution of flanks during working were made. The following pictures present the evolution of the same tooth flank during the testing process.

Figs. 11–21 represent the behavior of the surface of the worm wheel:

- Figs. 11–14 for 1/40 ratio during 7900 hours.
- Figs. 15–19 for 3/31 ratio during 8000 hours.
- Figs. 20–21 for 5/24 ratio during 1460 hours.

These figures show the progressive process of wear and pitting phenomena. These two physical types of flank deterioration are merged together.

The theoretical contact pressure distributions, determined with the analytical calculation method developed at CETIM (Ref. 5), are represented in Figs. 22–24.

If there is a uniform contact pressure distribution, as in the case of one-start worms, the propagation of cracks is low; otherwise the propagation is more localized near the high pressure of contact induced by low radius of curvature.

In all cases, it can be observed that the localization of the maximum of the theoretical prediction of the contact pressure distribution is well correlated with the zones where cracks appear. Consequently, in worm gears, the contact pressure distribution is not uniform along the lines of contact.

Interpretation of Phenomena

The first step of the process is linked to contact pressure and the induced cold working in the sublayer of surface where the maximum shear stress generally appears. Cracks are growing up under the surface near a depth close to 0.8 times the semi-width of contact. This depth is between 0.3 and 0.5 mm.

Cracks propagate more or less quickly toward the surface of the flanks as a function of the contact pressure level and the distribution of radius of curvature. When cracks have reached the surface, scales are removed, leaving small holes.

The transmitted load is redistributed on the part of the flank surface which is not scaled yet, and a new contact pressure distribution is established. This phenomenon evolves continuously up to the subsurface: Cracks have developed and reached the surface. An equivalent wear phenomenon on

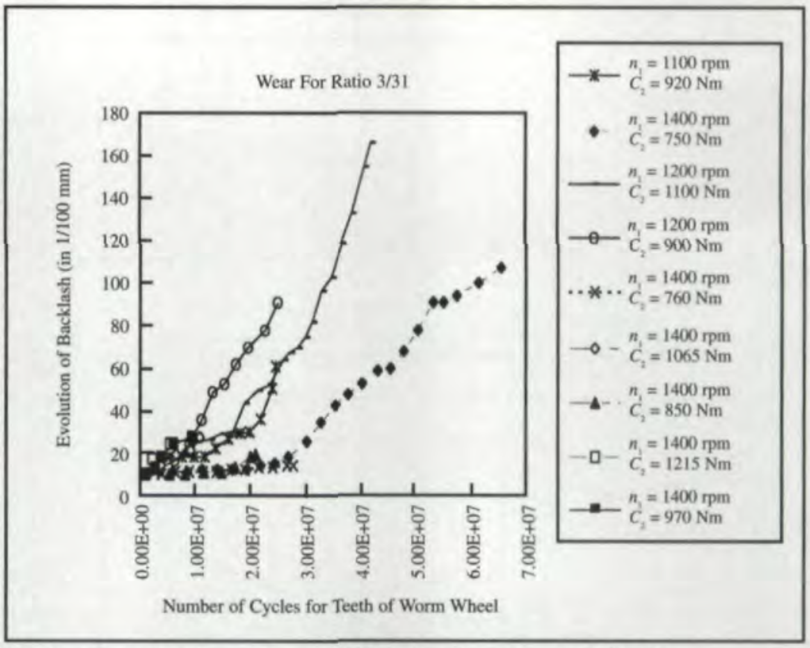

Fig. 6 — Evolution of wear for ratio 3/31 in function of number of cycles.

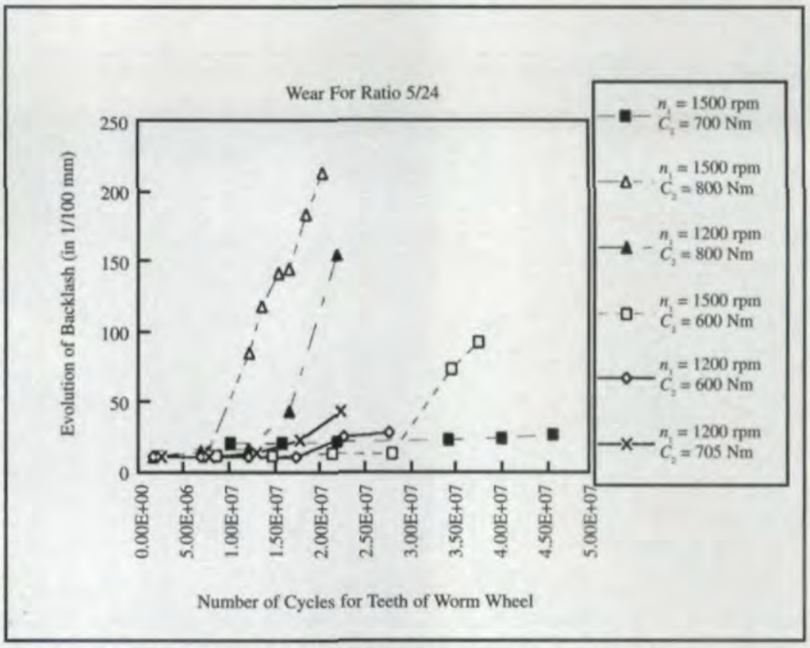

Fig. 7 — Evolution of wear for ratio 5/24 in function of number of cycles.

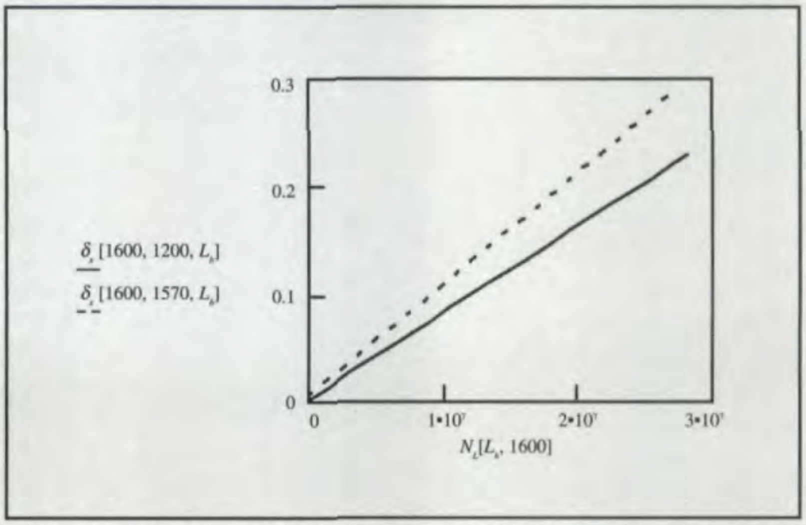

Fig. 8 — Prediction of wear by DIN 3996 for ratio 1/40 in function of number of cycles.

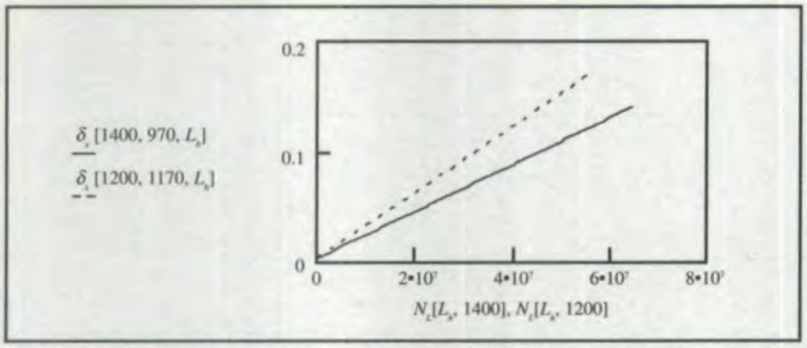

Fig. 9 — Prediction of wear by DIN 3996 for ratio 3/31 in function of number of cycles.

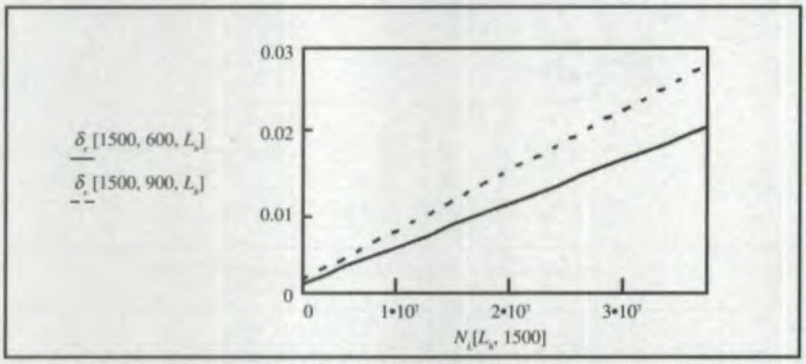

Fig. 10 — Prediction of wear by DIN 3996 for ratio 5/24 in function of number of cycles.

Fig. 11 — Worm gear 1/40: n_1 = 1600 rpm; C_2 = 1374 Nm after 2323 hours.

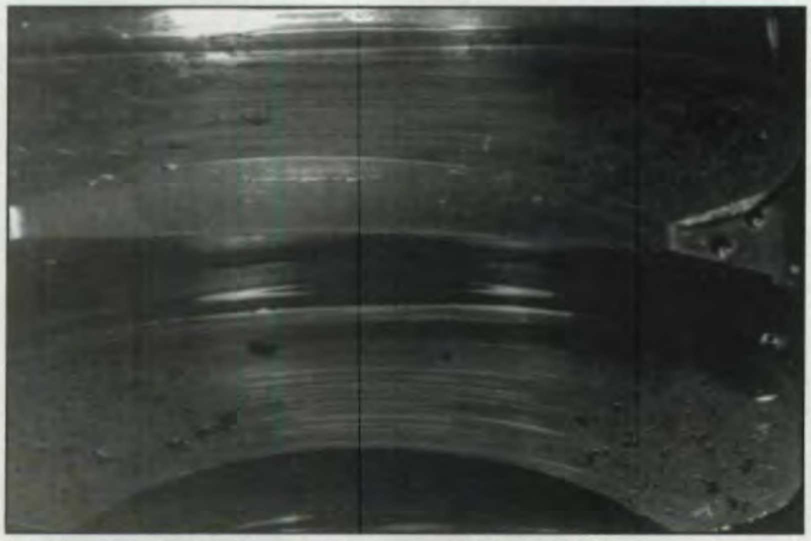

Fig. 12 — Worm gear 1/40: n_1 = 1600 rpm; C_2 = 1374 Nm after 4477 hours.

teeth is produced because material is removed, but the physical basis here is contact pressure.

The intensity of this phenomenon is directly linked to the radius curvature distribution of the worm gear. If this distribution is regular, as in one-start worm gears, the equivalent wear phenomenon will propagate slowly, and the resulting level of wear will be low.

When the pressure distribution is not regular, as in the case of 3- or 5-start worm gears, the equivalent wear phenomenon grows quickly, and the resulting wear level is high. This is observed in Figs. 5–7. In these cases, the heterogeneity of the worm wheel material can produce a non-homogeneity of tooth-to-tooth wear, thereby inducing pitch errors during working. These pitch errors can create internal dynamic effects caused by the bad load sharing between teeth, increasing the propagation of the equivalent wear phenomenon and, sometimes, the production of chocks during meshing.

A New Approach

In order to have a better prediction of worm gear life, it is necessary to have an approach based on the observed behavior of worm wheel flanks during working.

Our observations, described above, have shown that pressure contact phenomena are the basis of this kind of prediction. This implies the need to have a good knowledge of how the worm wheel is affected by contact pressure fatigue. To answer to this first question, CETIM has developed experimental investigations to determine S-N curves for several bronzes on roller-disk machines (Ref. 6).

In another way CETIM has developed an analytical method to determine contact pressure distribution, taking into account geometry (radius of curvature), kinematics (sliding velocity) and elasticity (gear tooth stiffness).

By coupling S-N curves with contact pressure distribution calculations, it is possible to predict the zones of tooth flanks on which scales will appear.

Then the calculation of removed surfaces can be made, and a new contact pressure distribution can be established. Taking into account the cumulative damages with S-N curves, the life of each point of flank surface can be predicted as the quantity of removed material and the equivalent wear phenomena.

This calculation has to be done iteratively: The contact pressure distribution is adjusted at each step, taking into account the leaving surface on worm wheel flanks. The surface strongly stressed is then removed, taking into account the contact

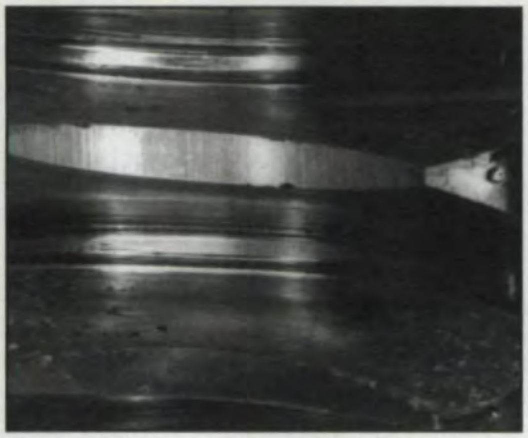

Fig. 13 — Worm gear 1/40: n_1 = 1600 rpm; C_2 = 1374 Nm after 6863 hours.

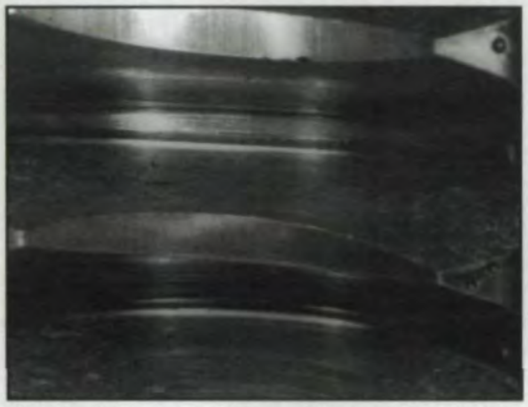

Fig. 14 — Worm gear 1/40: n_1 = 1600 rpm; C_2 = 1374 Nm after 7898 hours.

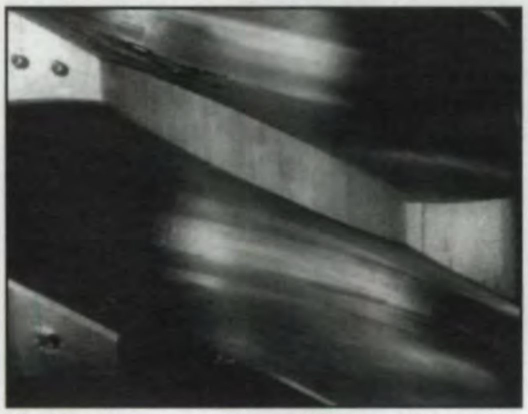

Fig. 15 — Worm gear 3/31: n_1 = 1400 rpm; C_2 = 750 Nm after 735 hours.

Fig. 16 — Worm gear 3/31: n_1 = 1400 rpm; C_2 = 750 Nm after 3302 hours.

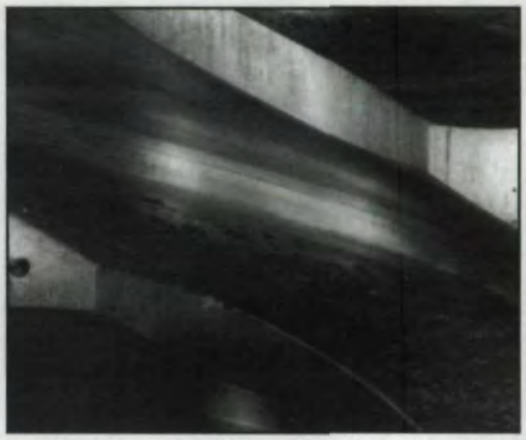

Fig. 17 — Worm gear 3/31: n_1 = 1400 rpm; C_2 = 750 Nm after 4032 hours.

Fig. 18 — Worm gear 3/31: n_1 = 1400 rpm; C_2 = 750 Nm after 5292 hours.

Fig. 19 — Worm gear 3/31: n_1 = 1400 rpm; C_2 = 750 Nm after 8033 hours.

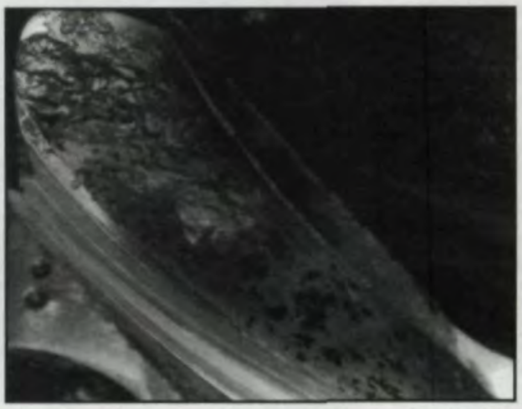

Fig. 20 — Worm gear 5/24: n_1 = 1200 rpm; C_2 = 800 Nm after 1113 hours.

FOR ALL LOADING CONDITIONS, A PERIOD OF MESHING, CORRESPONDING TO THE TIME IT TAKES SURFACES TO ADAPT, EXSISTS DURING WHICH NO WEAR APPEARS. THEN BACKLASH BEGINS AND EVOLVES CONTINUOUSLY.

Fig. 21 — Worm gear 5/24: n_1 = 1200 rpm; C_2 = 800 Nm after 1463 hours.

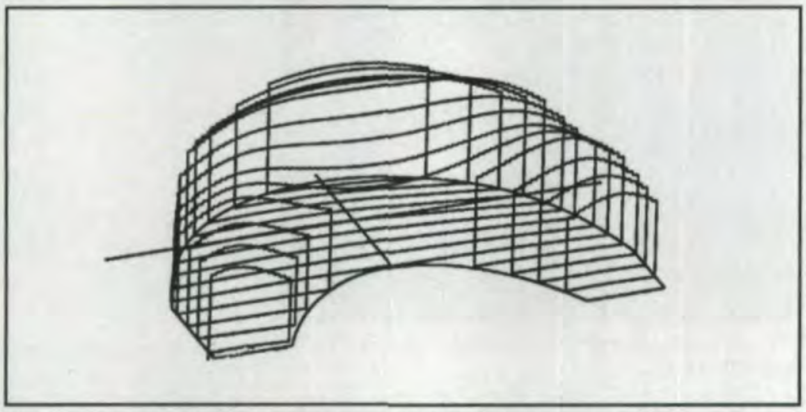

Fig. 22 — Pressure distribution on worm wheel flank for worm gear 1/40.

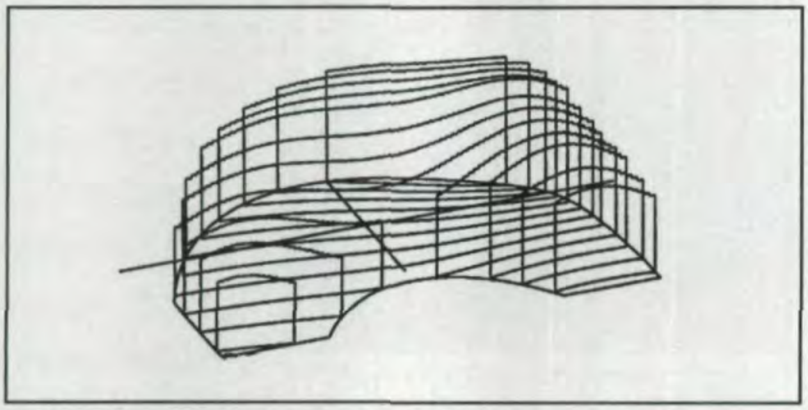

Fig. 23 — Pressure distribution on worm wheel flank for worm gear 3/31.

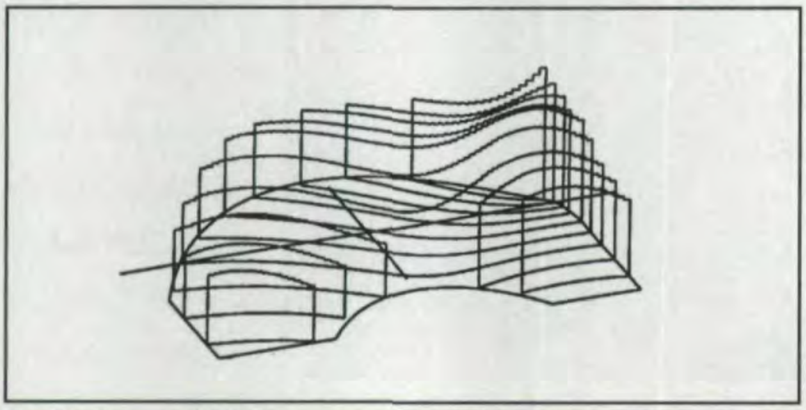

Fig. 24 — Pressure distribution on worm wheel flank for worm gear 5/24.

pressure S-N curve the material. A new leaving surface on the worm wheel flanks is determined for the next step of the calculation.

Conclusions

Experimental results presented in this paper, based on more than 50,000 hours of tests on real worm gear sets, provide a basis to evaluate wear phenomena.

It has been clearly established that analysis of the percentage of copper in the lubricating oil provides a good clue to the wear rate of worm gears. We also learned that:

- Wear phenomena are not a continuous effect starting as soon as the worm gear is running. A period of incubation is necessary during which contact pressure phenomena initiate cracks through the subsurfaces of the tooth flanks. When these cracks reach the surface, scaling appears, and an equivalent wear phenomenon appears which reflects a balance between weakness of material near the surface and redistribution of transmitted load on the surface.
- The second observation is that for worm gears, the contact pressure distribution is never constant on tooth flanks. The proposed analytical approach correlates well with observations.

The complexity of these fatigue surface phenomena have led us to propose a new approach in order to be able to predict surface durability of worm gears. It will be based on contact pressure.

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Dry Cutting of Bevel and Hypoid Gears

Dr. Hermann J. Stadtfeld

Introduction

High-speed machining using carbide has been used for some decades for milling and turning operations. The intermittent character of the gear cutting process has delayed the use of carbide tools in gear manufacturing. Carbide was found at first to be too brittle for interrupted cutting actions. In the meantime, however, a number of different carbide grades were developed. The first successful studies in carbide hobbing of cylindrical gears were completed during the mid-80s, but still did not lead to a breakthrough in the use of carbide cutting tools for gear production. Since the carbide was quite expensive and the tool life was too short, a TiN-coated, high-speed steel hob was more economical than an uncoated carbide hob.

It is now known that the coating is a key factor in the use of carbide tools. The coating lowers the friction between the chip and blade front face, and it protects the porous surface of the carbide. Heat and friction cause chemical reactions between the chip and the carbide binder material and "wash out" the carbide grains on the cutting front face. This causes a crater-like wear pattern. The combination of fine grain carbide with the well-known TiN coating is sufficient for skiving operations and leads to reasonable tool life (Ref. 1).

Improvements in carbide grades and sintering processes, in combination with new coating methods and the use of CNC machines, has led to a significant new trend in the way cylindrical gears are produced: High-speed hobbing using coated carbide tools without coolant. Provided that process parameters are set optimally, extremely short cutting times can be achieved with long tool life and high part accuracy. Process development in bevel gear cutting also benefits from the carbide and coating developments.

The Gleason results in carbide cutting with coolant are the product of substantial research work. Besides dramatically reduced cutting times of about 50%, improved surface finish and gear geometry quality are realized in comparison to conventional production with high-speed steel. This is largely due to the high stiffness and good dynamic behavior of the Phoenix® machines used for the cutting studies. The Gleason POWER-CUTTING™ process, as opposed to the carbide rough-cutting methods discussed by other machine tool manufacturers at the present time, is a finishing process which is suitable for later short-time lapping or grinding. AGMA Class 12 and 13 gear quality was achieved in all development studies.

Bevel Gear Dry Cutting

After successful investigations of the high-speed carbide cutting process with coolant (Ref. 2), the next logical step was to follow the general trend and proceed with the process development of a bevel gear dry cutting method. It was found that nearly all geometrical and technological parameters of the carbide wet cutting method also could be applied to bevel gear dry cutting. The surface cutting speed of the newly introduced method is 1000 ft/min.; that is, four times the value of conventional cutting. The cutting process combines conventional cutting in the continuous face hobbing method and plunge cutting in the single-index face milling case. Cutting feed rates of 70% of conventionally applied rates were found to be optimal as a result of the investigations.

Since the index speed of the single-index face milling cycle is the same for both POWER-DRY-CUTTING™ and conventional cutting methods, cycle time reduction

Fig. 1 — View into the work area of a Phoenix machine during continuous-index Power-Dry-Cutting (quarter of plunging depth).

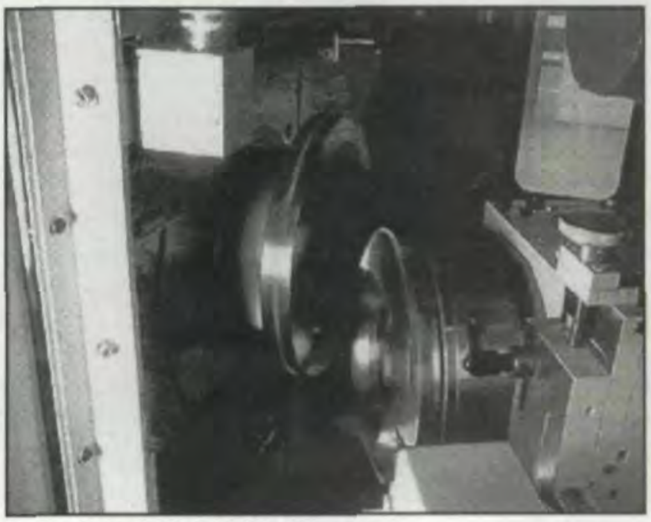

Fig. 2 — Power-Dry-Cutting (full depth).

Fig. 3 — Chips from dry cutting; plunging on the left, roughing in center, finishing to the right.

Fig. 4 — Phoenix 450HC-DRY with aerodynamic chip channel.

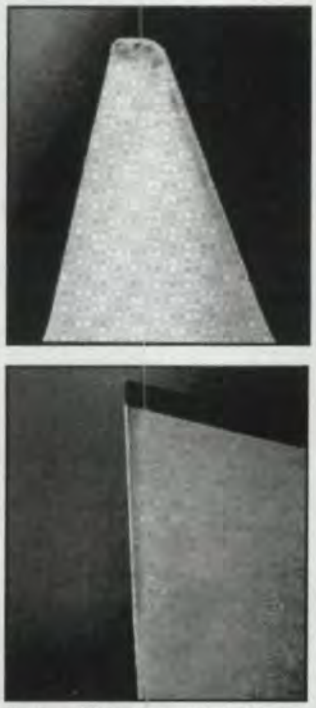

Fig. 5 — Blade wear pattern after cutting of 201 ring gears.

of 30 to 40% is possible in the case of face milling. In the continuous-index face hobbing case, the indexing speed is proportional to the cutter rpm, which reduces the cutting times by 75%. This and other advantages show the enormous economic potential of this method, especially if it is applied to the continuous indexing face hobbing process.

Fig. 1 gives an impression of the high surface speed and the high chip removal rate of the Gleason process. About a quarter of the plunging cycle has already been completed. The thin oil film for rust protection on the blank surface does not cause a noticeable smoke development at the beginning of the cycle.

Fig. 2 shows the end of the plunging cycle. The illustrated gear is a face-hobbed ring gear with 45 teeth and a module of 5.9 mm. The complete cutting time was 1.5 min. (as opposed to 6.5 min. for conventional cutting).

The carbide stick blades were TiCN coated. The projected tool life corresponds to three times that of high-speed steel tools. Even higher tool life times are expected in the future.

Fig. 3 shows chips from a ring gear, which was cut in the course of the parameter studies. The feed rate used to generate the chips was 0.005 in/blade. The highest temperatures of the chips in Fig. 3 were around the tempering temperature. Only parts of the chips turned blue. The chips to the left in Fig. 3 were created during the beginning of the plunging cycle; they are small and thick and have been cut by the tips of the blades. The chips in the center of Fig. 3 originate from the roughing portion of the cycle; they are wide and U-shaped and were generated simultaneously from the cutting edge, the blade tip and the clearance side of the blade. The chips to the right in the figure are wide and thin; they were cut primarily by the cutting edge of the blades near the end of the plunging cycle. At this time, the blades were completely engaged with the workpiece in the tooth profile direction.

The long and wide cut engagement between the blade and the work in case of bevel gear cutting prevents the generation of microchips. This is the reason why, in contrast to cylindrical gear hobbing, no chip welding in the root area occurs. The temperature of the workpieces amounted to 47°F above room temperature. The temperature of the cutter head stabilized at 90°F (12°F above room temperature).

A cost comparison to evaluate the economic viability of this method is based on the following factors:

- Cost of carbide blade sticks (four times price of HSS)
- Sharpening of carbide blades (last about three times longer than HSS)
- Coating of carbide blades (only once necessary for HSS)
- Building of blades in cutter head (1.5 times longer than HSS)
- Tool life of carbide (1.5 to 3 times that of HSS)
- Manufacturing time = machine occupation (20 to 30% that of HSS)

The cost of the carbide and the coating is more than compensated for by fewer resharpenings, requalifications and so forth. In this example, the total tool cost per manufactured gear is reduced by 28%.

Machines and Tools for POWER-DRY-CUTTING

To apply this method, it is necessary to use Phoenix bevel gear generators equipped with special high speed spindles. Cutter spindle speeds up to 1000 rpm cannot be realized with conventional machines that have the traditional cradle design. The complicated gear train in a conventional machine (with play between each gear set) cannot produce smooth coordinated motions and maintain stable temperatures at high cutting speeds. The six-axis, free-form Phoenix machine, however, has the cutting spindle motor mounted directly to the same vertical slide that houses the cutter spindle. Complicated couplings and mechanisms are then not required to transmit power to a tilted cutter head, as on cradle style machines. The new method can be applied to single-index face milling or continuous-index face hobbing and is always a completing operation.

To set up the cutting machine for the dry cutting process, a chip guiding channel was developed that takes advantage of the kinetic energy and the aerodynamic shape of the chips to remove them out of the work zone. In spite of cylindrical cutting, the chips fly tangentially to the cutter in the direction of the back guard, where they get removed into a chip container using a Venturi system. Except for two pressure air nozzles, no energy is applied

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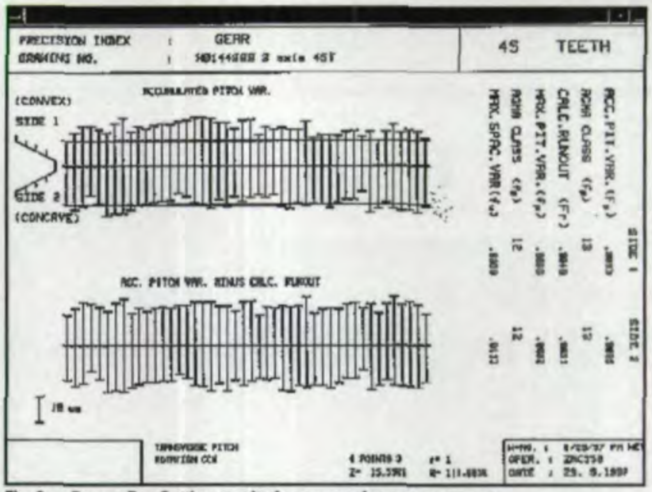

Fig. 6 — Power-Dry-Cutting results from a spacing measurement.

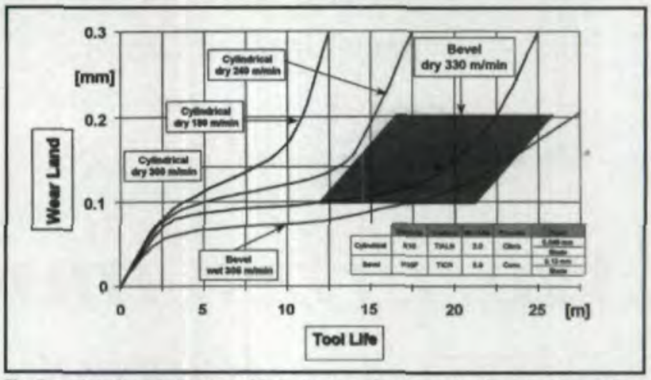

Fig. 7 — Relationship between blade wear, surface speed and cutting length (partially according to F. Klocke, Ref. 3).

to remove the chips and maintain a perfectly clean machine interior. This chip removal and the oil-free cutting process make for an environmentally and enerysaving overall concept. Up to 80% of the process heat is removed from the machine together with the chips without any heat transfer to the machine elements.

As a dry cutting option, the Phoenix machine has no oil tank and no chiller unit for cutting oil, as well as no chip conveyor. This results in a 30% reduced floor space and a very compact appearance of the machine tool. Fig. 4 shows a photograph of the Phoenix machine. The chip channel is visible in the center, surrounding the complete work area of cutter head and workpiece. The front part of the channel can easily swing aside and allows a convenient change of the cutter head.

The tools for dry machining are RSR® or TRI-AC® cutter heads. Solid carbide stick blades were used to conduct the cutting trials reported in this article. This simplifies the manufacturing of the blades and allows for the highest possible number of resharpenings. The carbide blades are resharpened on the Gleason 300 CG cutter/grinding machine with a diamond-coated grinding wheel. Fig. 5 shows photographs of "healthy" appearing blade wear after cutting 201 ring gears with dry cutting. The side relief wear land of 0.006" width leads to the conclusion that cutting of at least 50 more gears would have been possible.

The carbide grade with the best result in the parameter studies was Sandvik

H10F, with fine grain structure. The blades must be coated at least on the front face. In the reported investigations, a TiALN coating had been applied to the blades with a PVD method. The side relief angle was 6° (common value also used with high-speed steel). The side rake angle was ground to a positive value of 10°, which is about half the common value of a high-speed steel blade. The hook angle results from the tilted slots in the cutter head and therefore was a positive value of 12°.

The presented results of high-speed dry cutting were conducted in the continuous face hobbing and the single-index face milling method. The primary target of this new method is the manufacturing of both pinions and ring gears. Significant economic advantages exist for the high-speed cutting of Formate® ring gears. Due to the higher number of teeth, the cutting time savings is in general greater for the gear than for the pinion member. Besides, the elimination of the roll motion guarantees optimal and constant cutting conditions during the entire cutting cycle. The continuous change of the effective or dynamic blade angles always requires a compromise when generating a pinion. In spite of the smaller potential to reduce machining time and these additional complications, parameter studies for POWER-DRY-CUTTING of pinions are conducted parallel to the development of the ring gear cutting, since the gear manufacturing of the future shows a trend toward dry manufacturing, especially for small and medium-sized jobs.

Summary

The Gleason Works presents initial development results of a new dry cutting method for bevel and hypoid gears. An enormous potential will be available if the promising results of the laboratory investigations can be applied to daily manufacturing practice. Fig. 6 contains the graphical results of a tooth spacing measurement. Surface finish, tooth spacing and flank form correspond to a gear quality of AGMA Class 12.

An interesting comparison of the tool life potential can be seen in the diagram in Fig. 7. The tool life graphs for dry cutting of cylindrical gears, the tool life results of bevel wet cutting and expectations of the new process are compared in one diagram (Ref. 3). The definition of the tool life (cutting length) in metric units is defined as the length of material cut by one blade until a certain wear land occurs. The abscissa of the diagram in Fig. 7 shows the cutting length and the wear land width shown along the ordinate (Ref. 3).

The comparison of the bevel gear cutting graph in Fig. 6 with the cylindrical hobbing graph shows that today's results with Gleason wet cutting already have higher cutting lengths than the previous cutting results with cylindrical dry hobbing. 201 parts with 45 teeth and a face width of 45 mm results to 37 m, using an eleven-start cutter. With 1000 ft/min cutting surface speed and feed rate and 5/1000 in/blade, the projected tool life of POWER-DRY-CUTTING falls in the gray underlayed area.

The successful world premier of this method at the International Machine Tool Show (IMTS) in 1996 in Chicago was the result of development over more than two years, which subsequently led to the described method. The dry process version was introduced to the technical community at the 1997 EMO Machine Tool Show in Hannover, Germany. In addition to the dramatic reduction of cutting time, an advantage in surface finished gear quality is obtained in comparison to cutting with conventional high-speed steel tools. This was possible to a large extent because of the high stiffness and optimal dynamic behavior of the Phoenix machines. ○

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Hobs & Form Relieved Cutters: Common Sharpening Problems

Pfauter-Maag Cutting Tools, L.P.

Fig. 1 shows the effects of positive and negative rake on finished gear teeth. Incorrect positive rake (A) increase the depth and decreases the pressure angle on the hob tooth. The resulting gear tooth is thick at the top and thin at the bottom. Incorrect negative rake (B) decreases the depth and increases the pressure angle. This results in a cutting drag and makes the gear tooth thin at the top and thick at the bottom

In order to maintain correct tooth form, hobs and formed cutters must be resharpened with correct flute alignment (Fig. 2). Straight flutes should be sharpened parallel with the work axis and helical gashes should be sharpened with the correct lead. Gears cut by a hob with a flute lead error will not have the correct involute form. The teeth are unsymmetrical, each side of the teeth having a different pressure angle. The teeth are said to be "leaning" or have "cross bearing."

If the hob or cutter is not mounted true on the arbor, runout will result, and unequal amounts of stock will be ground from the faces of the teeth. Therefore, before sharpening and after the hob has been mounted on the machine, always check the hobs for runout. Runout produces errors and causes unsymmetrical profiles in the cut gear. Three forms of runout are shown in Fig. 3.

Runout can result from either a dirty and/or burred center, loose fit on the arbor or a sprung arbor. It could also be caused by machine misalignment, a worn index plate or pawls, a glazed wheel or improper finishing procedures.

Hobs sharpened with unequally spaced flutes have high and low teeth, which will produce unequal generating flats on the gear tooth as indicated in Fig. 4. High teeth (B) will produce low flats or hollows; low teeth (A) will cause high spots or bumps.

Fig. 5 shows gear tooth forms produced with correctly sharpened hobs and resulting teeth when errors are introduced into the sharpening of the same hobs.

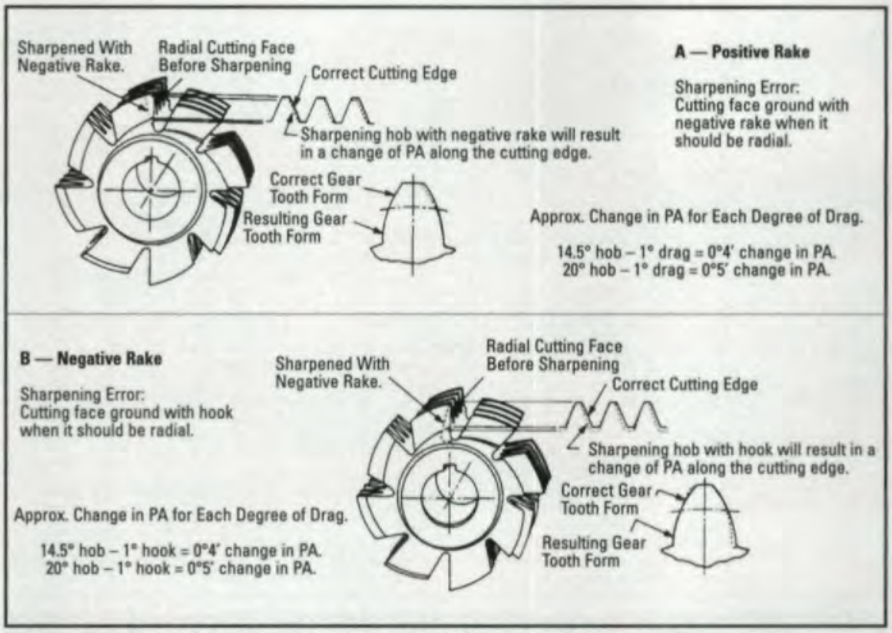

Fig. 1 — The effects of positive and negative rake resharpening error on profile.

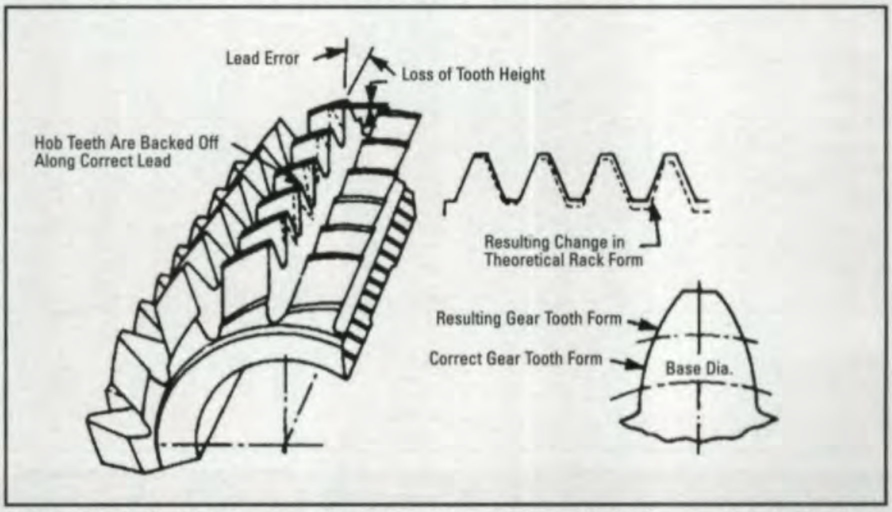

Fig. 2 — Effect of hob flute lead error. Since the hob is a reducing cylinder, incorrect flute lead resharpening destroys the integrity of the hob cylinder end to end, typically causing changes in workpiece size as the hob is shifted across its length.

Pfauter-Maag Cutting Tools, L.P., *Loves Park, IL, is a leading manufacturer of cutting tools for gear manufacturing.*

Table I — Common Sharpening Problems

Problem	Cause	Correction
Worn Index	Worn index plate or latch.	Replace worn item.
	Loose index plate.	Tighten retaining nut.
	Centers (work holding) misaligned.	Align centers.
Incorrect Radial Position and Rake Offset	Diamond positioned incorrectly.	Reposition diamond.
	Dull diamond.	Turn or replace diamond.
	Incorrect wheel position.	Reposition wheel.
	Excessive feed or wheel breakdown.	Reduce feed and dress wheel.
	Work eccentric or loose on arbor.	True between neutral centers on proper sized arbor. Do not use shims.
	Dirt on faces of collars and nut or in centers.	Clean faces and centers.
	Faces of collar and nut not parallel.	Grind faces parallel.
	Worn index plate or latch.	Replace worn item.
	Loose index plate.	Tighten retaining nut.
	Centers (work holding) misaligned.	Align centers.
Curved or Stepped Flute	Incomplete wheel dressing	Dress wheel and be sure diamond traverses entire cone face.
	Wheel not properly lowered into flute.	Lower wheel.
	Excessive feed or wheel breakdown.	Reduce feed and dress wheel.
	Wheel head swivelled incorrectly.	Set wheel head at proper helix angle.
Incorrect Lead	Tangent bar set incorrectly or not clamped.	Reset and clamp.
	Wheel "sings."	Dress with sharp diamond or faster traverse.
	Excessive table speed.	Reduce speed.
	Work eccentric or loose on arbor.	True between neutral centers on proper sized arbor. Do not use shims.
	Dirt on faces of collars and nut or in centers.	Clean faces and centers.
	Faces of collar and nut not parallel.	*Grind faces parallel.*
	Worn index plate or latch.	Replace worn item.
	Loose index plate.	Tighten retaining nut.
	Centers (work holding) misaligned.	Align centers.
Index Errors	Work arbor incorrectly tensioned between centers.	Tension correctly.
	Excessive feed or wheel breakdown.	Reduce feed and dress wheel.
	Wheel "sings."	Dress with sharp diamond, one pass only.
	Feeding at same flute each revolution.	Change feed cam setting.
	Work eccentric or loose on arbor.	True between neutral centers on proper sized arbor. Do not use shims.
	Dirt on faces of collars and nut or in centers.	Clean faces and centers.
	Face of collar and nut not parallel.	Grind faces parallel.
Burned Teeth	Improper coolant action.	Increase flow, redirect flow or change to different coolant.
	Glazed wheel.	Turn or replace diamond.
	Excessive feed.	Reduce feed.
	Insufficient table speed.	Increase speed.
	Sparking out too long.	Reduce number of revolutions of sparkout.
Rough Finish	Dressing too rapidly.	Dress wheel more slowly
	Wheel is too soft.	Use harder wheel.
	Excessive feed or wheel breakdown.	Reduce feed and dress wheel.
	Excessive table speed.	Reduce table speed.

Table II — Gear Hob Sharpening Tolerances—Total Indicator Reading

		Normal Diametral Pitch										
	Class	1–1.999	2–2.999	3–3.999	4–4.999	5–5.999	6–8.999	9–12.999	13–19.999	20–29.999	30–50.999	51 and finer
Spacing Between Adjacent Flutes	AA			.0020	.0015	.0010	.0008	.0008	.0006	.0006	.0006	.0006
	A	.0040	.0030	.0025	.0020	.0015	.0010	.0010	.0010	.0010	.0010	.0010
	B	.0050	.0045	.0040	.0030	.0020	.0015	.0015	.0010	.0010	.0010	.0010
	C	.0050	.0045	.0040	.0030	.0020	.0015	.0015	.0010	.0010	.0010	.0010
	D	.0060	.0060	.0050	.0050	.0030	.0025	.0025	.0020	.0017	.0017	.0015
Spacing Between Non-Adjacent Flutes	AA			.0040	.0035	.0025	.0015	.0015	.0015	.0015	.0015	.0015
	A	.0080	.0060	.0050	.0040	.0030	.0030	.0030	.0025	.0025	.0020	.0020
	B	.0010	.0090	.0080	.0060	.0050	.0050	.0050	.0040	.0035	.0030	.0025
	C	.0010	.0090	.0080	.0060	.0050	.0050	.0050	.0040	.0035	.0030	.0030
	D	.0120	.0120	.0100	.0100	.0080	.0080	.0070	.0060	.0050	.0040	.0035
Cutting Faces Radial to Cutting Depth	AA			.0010	.0008	.0006	.0005	.0005	.0003	.0003	.0003	.0003
	A	.0030	.0015	.0010	.0008	.0006	.0005	.0005	.0003	.0003	.0003	.0003
	B	.0050	.0025	.0015	.0010	.0008	.0007	.0007	.0005	.0005	.0005	.0005
	C	.0050	.0025	.0015	.0010	.0008	.0007	.0007	.0005	.0005	.0005	.0005
	D	.0100	.0075	.0050	.0040	.0020	.0020	.0020	.0015	.0015	.0010	.0010

		Face Width				
	Class	0–1"	1–2"	2–4"	4–7"	7" and up
Accuracy of Flutes Straight and Helical	AA	.0008	.0010	.0015	.0020	.0020
	A	.0010	.0015	.0025	.0030	.0050
	B	.0010	.0015	.0025	.0030	.0050
	C	.0010	.0015	.0025	.0030	.0050
	D	.0015	.0023	.0038	.0045	.0075

Table III — Effect of Sharpening Spacing Errors*

12° Cam Clear**		Lead Error Pressure Angle				
Sharpening Error	O.D. Runout	14 1/2°	20°	25°	30°	45°
.0005	.000106	.000027	.000039	.000049	.000061	.000106
.00075	.000159	.000041	.000058	.000074	.000092	.000159
.0010	.000213	.000055	.000078	.000099	.000123	.000213
.00125	.000268	.000069	.000098	.000125	.000155	.000268
.0015	.000319	.000082	.000116	.000149	.000184	.000319
.00175	.000372	.000096	.000135	.000173	.000215	.000372
.0020	.000425	.000110	.000155	.000198	.000245	.000425
.00225	.000478	.000124	.000174	.000223	.000276	.000478
.0025	.000531	.000137	.000193	.000248	.000307	.000531
.00275	.000585	.000151	.000213	.000273	.000338	.000585
.0030	.000638	.000165	.000232	.000298	.000368	.000638
.00325	.000691	.000179	.000252	.000322	.000399	.000691
.0035	.000744	.000192	.000271	.000347	.000430	.000744
.00375	.000797	.000206	.000290	.000372	.000460	.000797
.0040	.000850	.000220	.000309	.000396	.000490	.000850

*Pfauter-Maag Cutting Tools Standards

** 12° is a common clearance for unground hobs. Ground hobs are normally manufactured with 8° to 10° cam clearance; therefore, the amount of error for corresponding pressure angles will be less than shown above.

The above table can also be used to determine the pressure angle change due to radial face error at a given depth. PA Error = (Radial Face Error) x (Tan PA) x (Tan OD Cam Clearance)

Table IV — Form Milling Cutter Sharpening Tolerances*

		Cutter Diameter					
	Class	Up to 1" Incl.	1–2" Incl.	2–3" Incl.	3–4" Incl.	4–5" Incl.	Over 5"
Spacing Between Adjacent Flutes	Grd.	.0010	.0010	.0010	.0010	.0015	.0015
	Acc. Ung.	.0010	.0010	.0010	.0015	.0020	.0020
	Com. Ung.	.0010	.0010	.0015	.0020	.0020	.0020
Spacing Between Non-Adjacent Flutes	Grd.	.0020	.0020	.0025	.0030	.0030	.0040
	Acc. Ung.	.0030	.0030	.0030	.0040	.0040	.0050
	Com. Ung.	.0030	.0035	.0040	.0045	.0050	.0050

		Face Width of Cutter				
	Class	Up to 1" Incl.	1–2" Incl.	2–4" Incl.	4–7" Incl.	Over 7"
Accuracy of Flute Straight or Helical	Grd.	.0010	.0015	.0025	.0030	.0050
	Acc. Ung.	.0010	.0015	.0025	.0030	.0050
	Com. Ung.	.0010	.0015	.0025	.0030	.0050

		Depth of Form					
	Class	Up to 1/4"	1/4–1/2" Incl.	1/2–3/4" Incl.	3/4–1 1/4" Incl.	1 1/4–2" Incl.	Over 2"
Cutting Faces Radial To Cutting Depth	Grd.	.0003	.0005	.0008	.0010	.0020	.0030
	Acc. Ung.	.0005	.0007	.0012	.0015	.0028	.0040
	Com. Ung.	.0010	.0012	.0020	.0025	.0035	.0050

*Pfauter-Maag Cutting Tools Standards

Table V — Multiple Thread Milling Cutter Sharpening Tolerances — Total Indicator Reading*

		Cutter Diameter					
	Class	0–1"	1.001–2"	2.001–3"	3.001–4"	4.001–5"	5.001–6"
Spacing Between Adjacent Flutes	AT	.0010	.0010	.0010	.0010	.0015	.0015
	BT	.0010	.0010	.0010	.0015	.0020	.0020
	CT	.0010	.0010	.0010	.0015	.0020	.0020
	DT	.0010	.0010	.0010	.0015	.0020	.0020
Spacing Between Non-Adjacent Flutes	AT	.0020	.0020	.0025	.0030	.0030	.0040
	BT	.0020	.0030	.0030	.0030	.0035	.0045
	BT	.0030	.0030	.0030	.0040	.0040	.0050
	CT	.0030	.0035	.0040	.0045	.0050	.0050

		Face Width of Cutter				
	Class	0–1"	1.001–2"	2.001–4"	4.001–7"	7.001" & Over
Accuracy of Flute Straight or Helical	AT	.0010	.0015	.0025	.0030	.0050
	BT	.0010	.0015	.0025	.0030	.0050
	CT	.0010	.0015	.0025	.0030	.0050
	DT	.0010	.0015	.0025	.0030	.0050

		Depth of Form			
	Class	.034" or Less	.034–.096"	.097–.170"	.171" & Over
Cutting Faces Radial To Cutting Depth	AT	.0003	.0003	.0003	.0003
	BT	.0003	.0004	.0004	.0004
	CT	.0003	.0005	.0010	.0010
	DT	.0005	.0010	.0015	.0015

*Metal Cutting Tool Institute Standards

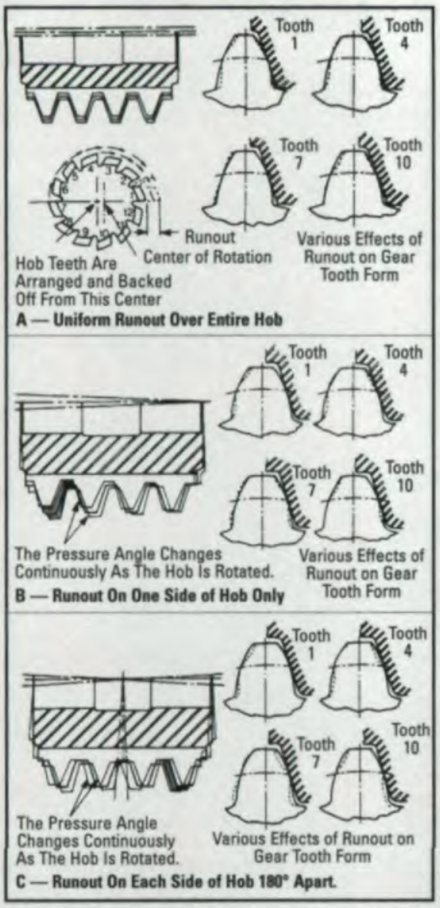

Fig. 3 — The effect of three types of runout on the hobbed profile.

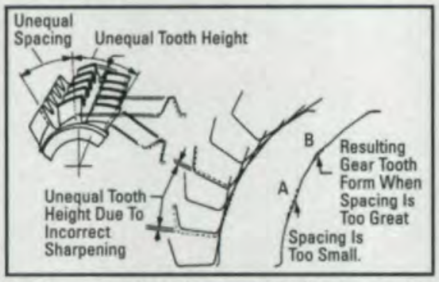

Fig. 4 — The effect of accumulated flute spacing error on profile.

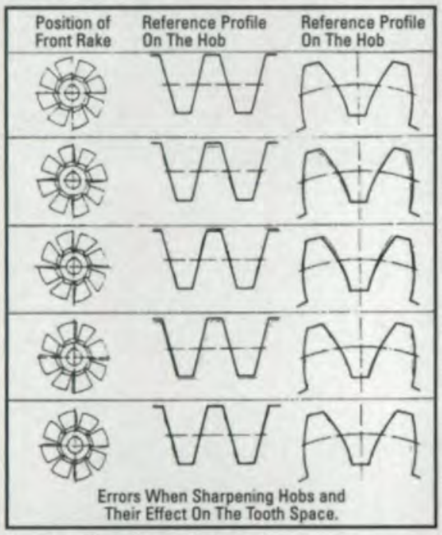

Fig. 5 — The effect of hob resharpening errors on the hobbed tooth profile relative to the basic rack profile of the hob.

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Acknowledgement: *Thanks to Keith Liston of Pfauter-Maag Cutting Tools for help with the technical editing of this article.*

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Welcome to our Product News page. Here we feature new products of interest to the gear and gear products markets. To get more information on these items, please circle the Reader Service Number shown.

Ono Sokki Linear Gage Sensor

Ono Sokki has introduced the Model GS251W linear gage sensor, which is designed to measure dimensions, thickness, curvature, eccentricity, displacement, height, depth, flatness, variation, runout, roundness, distortion, deflection and position. It uses linear glass scale technology, allowing the maintenance of highly consistent accuracy throughout the entire range of measurement. The GS 251W produces accurate measurement to 0.0004" throughout its 1" measuring range.

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Kolene Corp. has developed a low-temperature, environmentally friendly cleaning process for removing tough soils from parts made of steel, aluminum and heat sensitive alloys. Applications range from removing carbon, oils and gasket material from engine components to cleaning aluminum bearing housings. The Kolene® No. 6 Process consists of a compact immersion tank and proprietary cleaning salts. Cleaning takes place at approximately 600°F. The company says that in most applications, soils can be removed without heat distortion in less than 15 minutes.

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ISO 9000 Micrometer Calibration Sets

The L.S. Starrett Company has introduced the only complete 1" and 25 mm micrometer checking sets available in the industry that meet ISO 9000 requirements. The company says that the new Starrett/Webber Gage Block Sets are the fastest and most convenient method developed to meet the ISO 9000 recommendations. The 1" English Micrometer Checking Kits are available for U.S. Federal Accuracy Grades 1 and 2. The chromium carbide gage blocks in the sets are accurate to several millionths of an inch.

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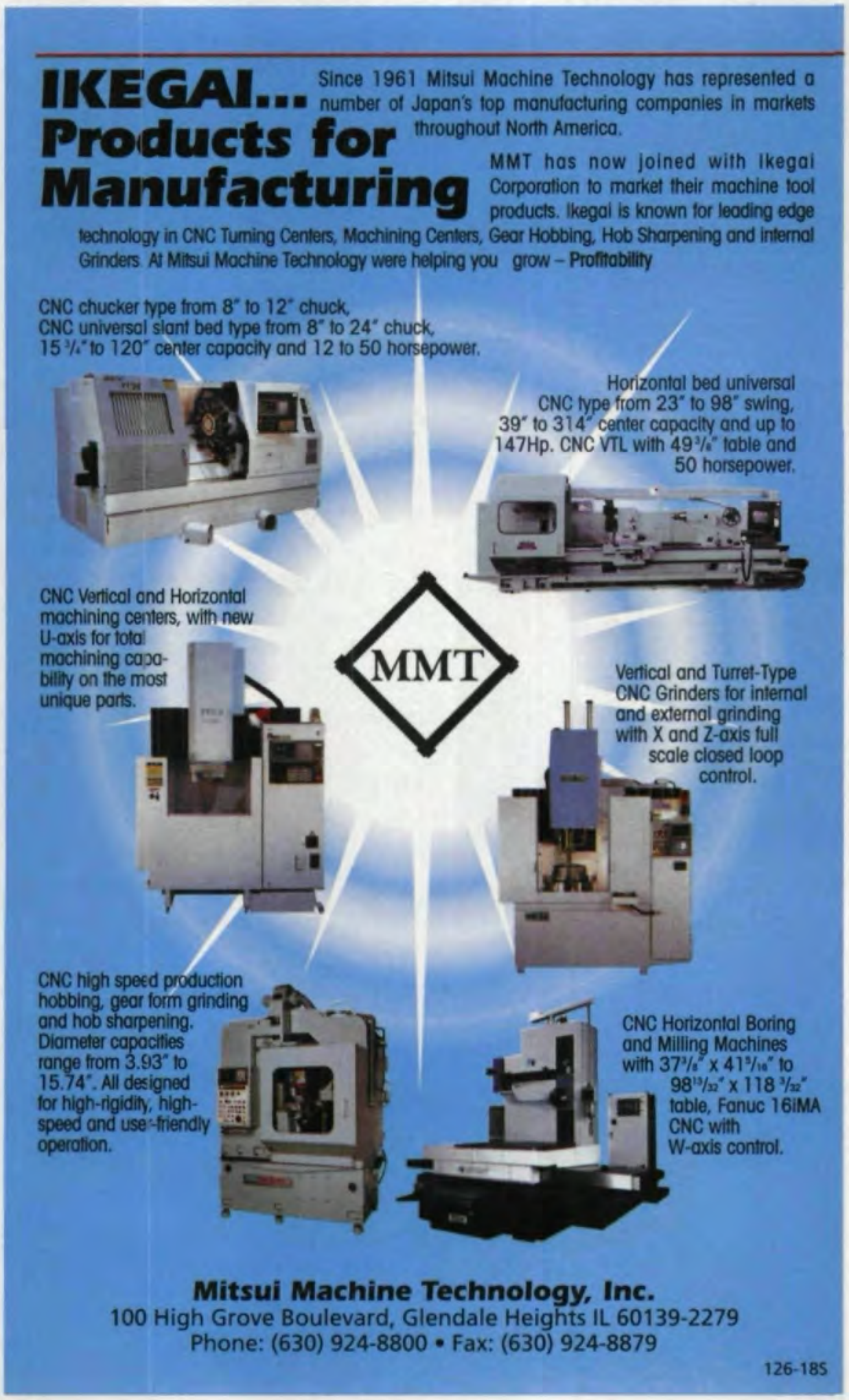

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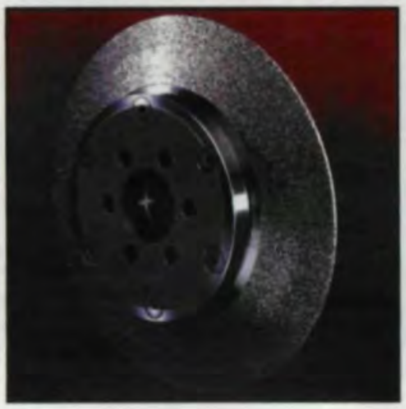

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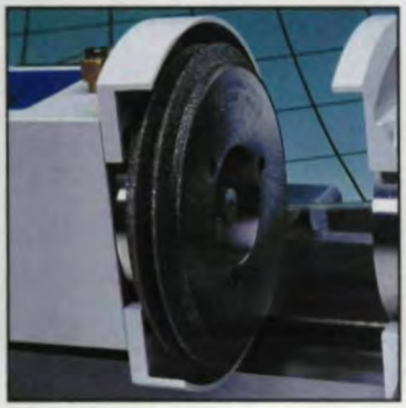

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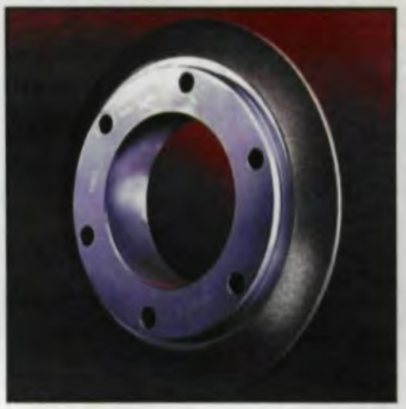

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CIRCLE 129

THE GUY DOWN THE HALL WANTS HIS COPY OF *GEAR TECHNOLOGY* BACK!

FILL OUT THE CARD BETWEEN PAGES 24 & 25 AND

GET YOUR OWN!!

Thoughts For Gear Technologists

Gear Technology's bimonthly aberration — gear trivia, humor, weirdness and oddments for the edification and amusement of our readers. Contributions are welcome.

Every now and then, it strikes us as wise to keep our thoughts to ourselves and let our betters speak for us. Therefore, we present to you a collection of observations on work, science and other items of interest to gear engineers.

The opinions expressed are those of the people cited, not the Addendum team, although a number of them appear on various bulletin boards around the office.

On Work

It is impossible to enjoy idling thoroughly unless one has plenty of work to do. — *Jerome K. Jerome*

If you have great talents, industry will improve them: If you have but moderate abilities, industry will supply their deficiency. — *Joshua Reynolds*

I have long been of the opinion that if work were such a splendid thing, the rich would have kept more of it for themselves. — *Bruce Grocott*

That state is a state of slavery in which a man does what he likes to do in his spare time and in his working time that which is required of him. — *Eric Gill*

The test of a vocation is the love of the drudgery it involves. — *Logan Pearsall Smith*

Philosophy is written in that great book which ever lies before our eyes—I mean the universe . . . this book is written in mathematical language, and its characters are triangles, circles and other geometrical figures, without whose help . . . one wanders in vain through a dark labyrinth. — Galileo

A man who has no office to go to—I don't care who he is—is a trial of which you can have no conception. — *George Bernard Shaw*

It is wonderful, when a calculation is made, how little the mind is actually employed in the discharge of any profession. — *Samuel Johnson*

Work is love made visible. If you cannot work with love, but only with distaste, it is better that you should leave your work and sit at the gate of the temple and beg alms of those who work with joy. — *Kahlil Gibran*

On Engineering & Science

Let no one enter who does not know geometry. — *Inscription on Plato's door, probably at the Academy of Athens.*

Praise without end the go-ahead zeal
Of whoever it was invented the wheel;
But never a word for the poor soul's
Sake that thought ahead, and invented the brake. — *Howard Nemerov*

God made the integers. All else is the work of man. — *Leopold Kronecker*

Someone told me that each equation I included in the book would halve the sales. — *Stephen Hawking on* A Brief History of Time.

There are no such things as applied sciences, only applications of science. — *Louis Pasteur*

On Working With Others

Never explain—your friends do not need it, and your enemies will not believe you anyway. — *Elbert Hubbard*

The world is divided into people who do things and people who get the credit. Try, if you can, to belong to the first class. There's far less competition. — *Dwight Morrow*

People who could not tell a lathe from a lawn mover and have never carried the responsibilities of management never tire of telling management off for its alleged inefficiency. — *Keith Joseph*

And finally, an Addendum All-Purpose Solution of Last Resort for awkward situations . . .

. . . there are few situations in life that cannot be honorably settled, and without loss of time, either by suicide, a bag of gold, or by thrusting a despised antagonist over the edge of a precipice upon a dark night. — *Ernest Bramah*

The Addendometer: If you've read this far on the page and enjoyed it, please circle 225.

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