
GEAR TECHNOLOGY
The Journal of Gear Manufacturing
IMTS 98 SHOW ISSUE
September/October 1998
WHO'S WHO AT IMTS
THE KEYS TO ISO 6336-1
THE BASICS OF GEAR METROLOGY
DESIGNING RELIABILITY INTO INDUSTRIAL GEAR DRIVES
THE GEAR INDUSTRY'S INFORMATION SOURCE

INTRODUCING THE REVOL

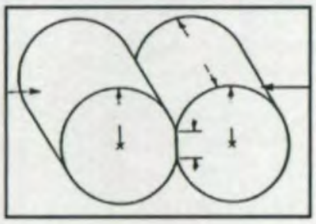
GEAR TECHNOLOGY

SEPTEMBER/OCTOBER 1998

The Journal of Gear Manufacturing

SPECIAL SECTION

FEATURES



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DEPARTMENTS

64

Cover art courtesy of M&M Precision Systems, West Carrollton, OH

GEAR TECHNOLOGY

The Journal of Gear Manufacturing

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LOOKING AROUND THE CORNER

Listen carefully these days and you'll hear a faint rumbling among the economic masses. It's probably nothing to worry about. It'll most likely go away. It's only the naysayers and skeptics who predict that the end is near. They've been doing so for almost all eight years of our current economic boom, and they've been wrong so far.

As a matter of fact, most of today's statistics and news clippings on the U.S. economy seem to point toward more good times ahead. "We enjoy impressive growth, low inflation and unemployment, and unprecedented wealth creation," says *U.S. News and World Report*. "The U.S. continues to manufacture almost twice as much wealth as any other country," says *Industry Week*. The good news crosses my desk every day.

But an uncanny feeling persists that there's a monster hiding around the corner, that good times can't last forever. It's just a feeling, but it hints that Alan Greenspan's perpetual motion machine is doomed to run out of momentum just like every contraption ever invented by a backyard mechanic chasing a dream.

While factories are still cranking out goods, and consumers are still buying them, economic headlines lately have been raising some eyebrows. Economic turmoil in the Far East. Hmm. Atomic weapons being tested. Hmm. Russia raising interest rates to 150%. Hmm. Stock market "corrections" of 100-200 points in a day. Hmm. Big companies announcing layoffs. Hmm.

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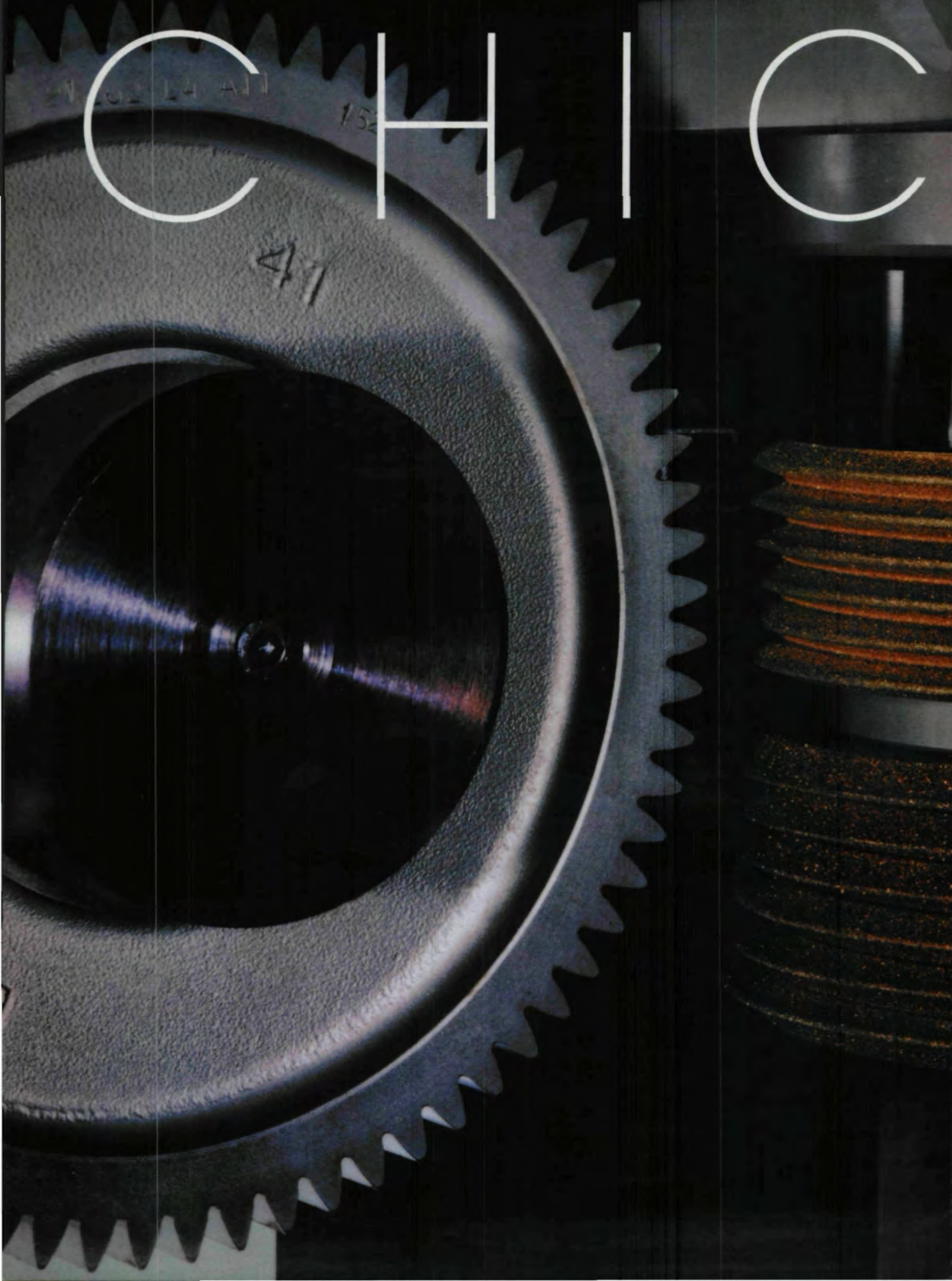
Other than the fact that these are tough times for prognosticators, what does all of this mean? I don't claim to have all the answers. Each business owner must make his own decisions. But it would be prudent, at the very least, to have a contingency plan in place. Weighing your options now may mean the difference between weathering the storm or becoming tomorrow's driftwood.

We usually look to IMTS for ideas about how we can make our manufacturing operations more efficient, more productive and more profitable. Certainly now—when business is still good—is a good time to consider these things. This year, show organizers are promising even more exhibits taking up even more floor space than ever before.

But this year, I'll be looking at more than the technology. I'll be looking at IMTS itself as a bellwether of industry confidence in the future of manufacturing demand. I'll be looking at attendance levels and reports of machine tool orders that come out after the show.

These reports should give us an idea whether the monster is really around the corner or whether he is merely the result of an overactive imagination.

Michael Goldstein,
Publisher & Editor-in-Chief


CHIC

Pfauter-Maag Cutting Tools now brings you the best in bevel gear tools

Pfauter-Maag Cutting Tools now offers more than just cylindrical gear-cutting tools. We are now responsible for the North American sales and service of the fast-growing line of Gleason bevel gear-cutting blades and heads. Even better, many of these products will be manufactured at Pfauter-Maag's ultra-modern production facility in Loves Park, IL, helping raise quality and responsiveness to unprecedented new levels.

Bevel gear-cutting blades and heads are the latest addition to the Pfauter-Maag line of hobs, shaper cutters, shaving cutters, form cutters, CBN grinding wheels, thin film coatings, heat treat services, and resharpening services.

Pfauter-Maag Cutting Tools
CORPORATION

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E-Mail: Sales@pmct.com

IMTS 98
CHICAGO, SEPTEMBER 9-16, 1998

BOOTH NO. B1-7150

CIRCLE 168

Proudly Announcing . . .

ISO 9001 Registered

Sponsors Predict Biggest IMTS Ever

In 1927 the first precursor of IMTS was held in Cleveland. Back then, lasers, robots and computer controls were just science fiction. At IMTS 98 they will fill nearly every last corner of the recently expanded McCormick Place.

Sponsors expect IMTS 98 to break records. Already it's the biggest IMTS in terms of exhibit space (1.4 million square feet) and exhibitors (more than 1,400). And if registration continues at its current pace, we could also see new highs for attendance and sales made at the show.

The following are just some of the companies that might be of interest to gear manufacturers.

IMTS 98 BASICS

The International Manufacturing Technology Show

September 9–16, 1998

McCormick Place, Chicago, IL

Registratration: (800) 322-IMTS

Additional information:

www.imts.org

GUIDE TO THE BOOTHS

A.G. Davis/AA Gage (Booth E1-2332) will have on display their line of Hydra-grip hydraulic expansion arbors and chucks for gear grinding, hobbing, shaping, honing and other manufacturing operations. They will also show their line of face gear couplings.

Bourn & Koch Machine Tool Co. (Booth C2-5473), will exhibit the model 400H 5-axis gear hobber. The unit at the show will be an extended bed version with an optional sliding operator guard. The machine is capable of producing AGMA quality 10 on parts with diameter up to 400 mm. The machine features a hob slide that can accommodate up to 6" X 7" hobs and is CNC controlled to allow programming cycles for crowning, tapered root spline hobbing and the shifting between different hobs or cutters on one arbor. Bourn & Koch is a manufacturer of OEM gear hobbers, gear shapers, specialty gear machines, dial machining centers, extrusion milling machines, specialty machines, turning centers, machining centers, grinders, boring machines and more.

Emuge Corporation (Booth E1-2352) is a premier manufacturer of quality workholding and electronic control systems for gear manufacture. Emuge will feature a precision clamping product line that includes arbors, chucks, diaphragm chucks, spindles, draw bars and machine operation measuring systems configured to accomplish a broad range of turning, grinding, hobbing, milling, drilling, lapping, balancing, inspection and assembly operations. Emuge's mechanical, hydraulic and mechanical/hydraulic systems achieve an excellent runout accuracy—typically .00008 to .0002—and are renowned for reliability, functionality, clamping element interchangeability and ease of maintenance. Custom application services are offered. Emuge's Web site at *www.emugecorp.com* outlines details.

Euro-Tech Corporation (Booth E1-2530) will show the Frenco line of gages and workholding devices, Mytec hydraulic expansion arbors and chucks and the Euro-Tech Power Block III toolholder. Frenco's product range includes extremely long-wear go/no go gages, bevel gear testing gages, completely automatic spline and cluster shaft inspection systems and a full range of clamping arbors, chucks and nests for inspection and machining applications. Mytec rupture-proof arbors and chucks are ideally suited to precision applications. Mytec arbors and chucks provide normal runouts under .00012" for grinding, hobbing, shaving, shaping and inspection of gears. The Euro-Tech Power Block III mounts directly to your bench or work table to make tool changes easy and risk-free. Vertical and horizontal axes allow access to the bottom of the toolholder. Either axis may be outfitted with like or different receptacles to accommodate any standard or custom toolholder tapers, including HSK.

Gleason Pfauter Hurth (Booth B1-7150) will exhibit several new products for gear manufacturing. The new Power Dry Cutting process for bevel and hypoid gears will be demonstrated on the new Gleason 175HC Power Dry Cutting machine. The 175HC will demonstrate face milling and face hobbing on bevel gears. Gleason will also premier the new 600HTL Hypoid Turbo Lapping machine for hard finishing bevel and hypoid gears. The turbo lapping process (ultra high speed) is possible due to Gleason advances in machine dynamics and compound application technology. Gleason-Pfauter will exhibit for the first time the new (16") P400G profile grinder with integrated gear measuring and integrated CNC wheel dressing. Gleason-Hurth will demonstrate the capabilities of the ZH125 CNC spheric honing machine for hard finishing cylindrical gears using an internal abrasive honing tool. The machine will be shown with automation. Gleason and Pfauter will also exhibit, for the first time, a new joint engineering and manufacturing cooperation in machine design with a new gear manufacturing machine to be unveiled at IMTS.

Gold Star Coatings (Booth E1-2701), a subsidiary of Star Cutter Co., will feature thin film coatings used to either improve tool life on cutting tools and dies or improve wear life on parts.

H.B. Carbide Co. (Booth E1-2700) a subsidiary of Star Cutter Co., will feature its line of carbide pre-forms used to manufacture finished tools and wear parts.

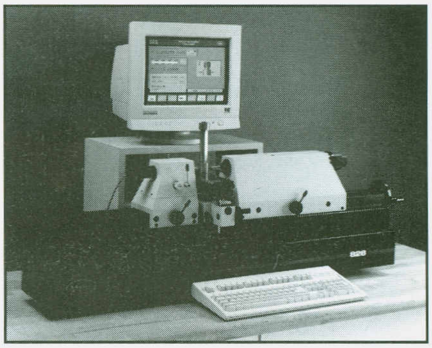
Holroyd (Booth B2-6516) will show, for the first time anywhere in the world, the TG150E thread grinder. This is a 'sister' version of the TG350E launched two years ago as the first in a planned new generation of machines. The TG150E has been specifically designed for grinding smaller components, with profiles up to 70 mm wide and less than 300 mm in diameter (the TG350E has a maximum 350 mm capacity and 110 mm profile width). An important feature of the new machine is integrated 3D component profile measurement with automatic machine compensation to all axes. Also on display will be a range of worm gears, screw compressor rotors and other helical products manufactured by Holroyd's subcontract facility.

Kapp Sales & Service L.P. (Booth B1-6981) represents the Kapp and Niles product lines of internal and external grinding machines featuring stock dividing, on-board measuring, dressable wheels and CBN wheels for the manufacture of internal and external spur and helical gears, ball screw tracks, worms, compressor rotors, rotary pistons (root type), pump spindles (IMO type), pump rotors (gerotor type), vane pump rotors, constant velocity tracks (CVT and the like) and more.

Koepfer America, L.L.C. (Booth B1-6961) will display a new high-speed, eight axis gear hobbing machine with computer controlled flexible automation. The Koepfer Model 160 is designed for shaft, pinion and gear work up to 60 mm and 2.36" diameter. The machine includes eight axes of CNC control, including tailstock and gantry loading arm positions to reduce setup time and improve overall efficiency. A unique slant bed design provides ideal chip flow and allows the user the choice of wet or dry hobbing. High spindle speeds (cutter up to 5,000 rpm and work spindle up to 1,000) allow optimum use of coated carbide cutting tools even when hobbing parts of small diameters or low numbers of teeth. The automation allows loading of a part in two seconds.

M&M Precision Systems Corporation (Booth B1-7149) will feature metrology systems for parallel axis and spiral bevel gears, gear cutting tools, thread gages and turbine blades. 3515 and 200 Series CNC inspection systems for gear manufacturing process control, running a full complement of inspection software, will be networked to a remote workstation with M&M's GearNet™ software. M&M will highlight multiple probe technology with demonstrations of ID, 3D and laser probe scanning. New LMS laser measuring systems will showcase non-contact scanning for thread gages and turbine blades. The GRS-2 double flank gear roller system and durable ODM-8 dimension over pins gage will also be shown.

Mahr Corporation (Booth D2-4437) will present metrology products for measuring gears as well as surface texture, form and length on a variety of parts. The Extramess 2000 is a new analog/digital inductive comparator accurate to .000012". The M1 and M2 are the first of a new series of surface texture measurement instruments designed for the shop floor. The Optimar 100 is a new bench-top calibration system for dial indicators, dial comparators, dial test indicators, incremental probes and LVDTs. A staff of applications engineers and technical specialists will be on hand to discuss the new products.



The 828 PC calibration center from Mahr Corp.

From the desk of Fred Young, President, Forest City Gear

Noise, Misalignment, Banding, Transmission Error, Harmonic Resonance, Binding, Roughness

These are a few frequently occurring problems with gears. Are you losing business to competitors because of noise and poor efficiency? Forest City Gear has resolved these problems for many diverse companies with our high accuracy cut gears.

CROWN HOBBING

We have pioneered ***fine pitch crown hobbing*** (putting a radius on the flank of the tooth along the longitudinal axis) which has dramatically reduced noise especially in helical applications and compensated for various misalignment scenarios. Almost single-handedly we introduced this concept to the American power tool industry and many small motor manufacturers.

CARBIDE REHOBBING

We reduced gear error by ***carbide rehobbing*** gears after they have been heat treated to 60Rc. Only a few gear producers utilize this advanced technology. Precise index accuracy for x-ray positioning equipment, telescopes, printing presses, copiers, plotters & printers— these are a few applications assisted by Forest City Gear.

ANALYTICAL GEAR INSPECTION

Few people have the ***inspection capability*** to investigate and monitor profile error as does FCG. Our analytical gear checkers verify lead and profile error to ten millionths of an inch. You may already be one of about eighty **gear manufacturers** assisted by FCG to produce quieter, more accurate gears. Accuracy of profile is determined by the quality of the hobs, ***hob re-sharpening*** (another of our services) and mounting on the gear machine. Problems here effect transmission error and noise. Lead error and crossed bearing cause premature failure and contribute to excess backlash as well as noise variation.

STATE-OF-THE-ART MACHINERY

Gear cutting equipment at Forest City Gear is offered for resale after three to five years, to be replaced by more efficient ***state-of-the-art new machinery***. Index error, another bugaboo contributing to roughness, is common to gear producers using obsolete gear equipment with very worn master index gear and transmission components. Accumulated spacing error can occur during heat treatment or from multistart hobs as well as from badly worn gear equipment. This non-adjacent index error excites transmission errors, banding and harmonics. Finally all of these problems are additive to plain old runout or eccentricity which can be caused by bad gear blanks, inadequate and inaccurate work fixturing as well as worn master drive worm gears and cutter spindles in the gear machines. These conditions, unfortunately, are all too common in the American Gear industry and overseas.

LET US HELP

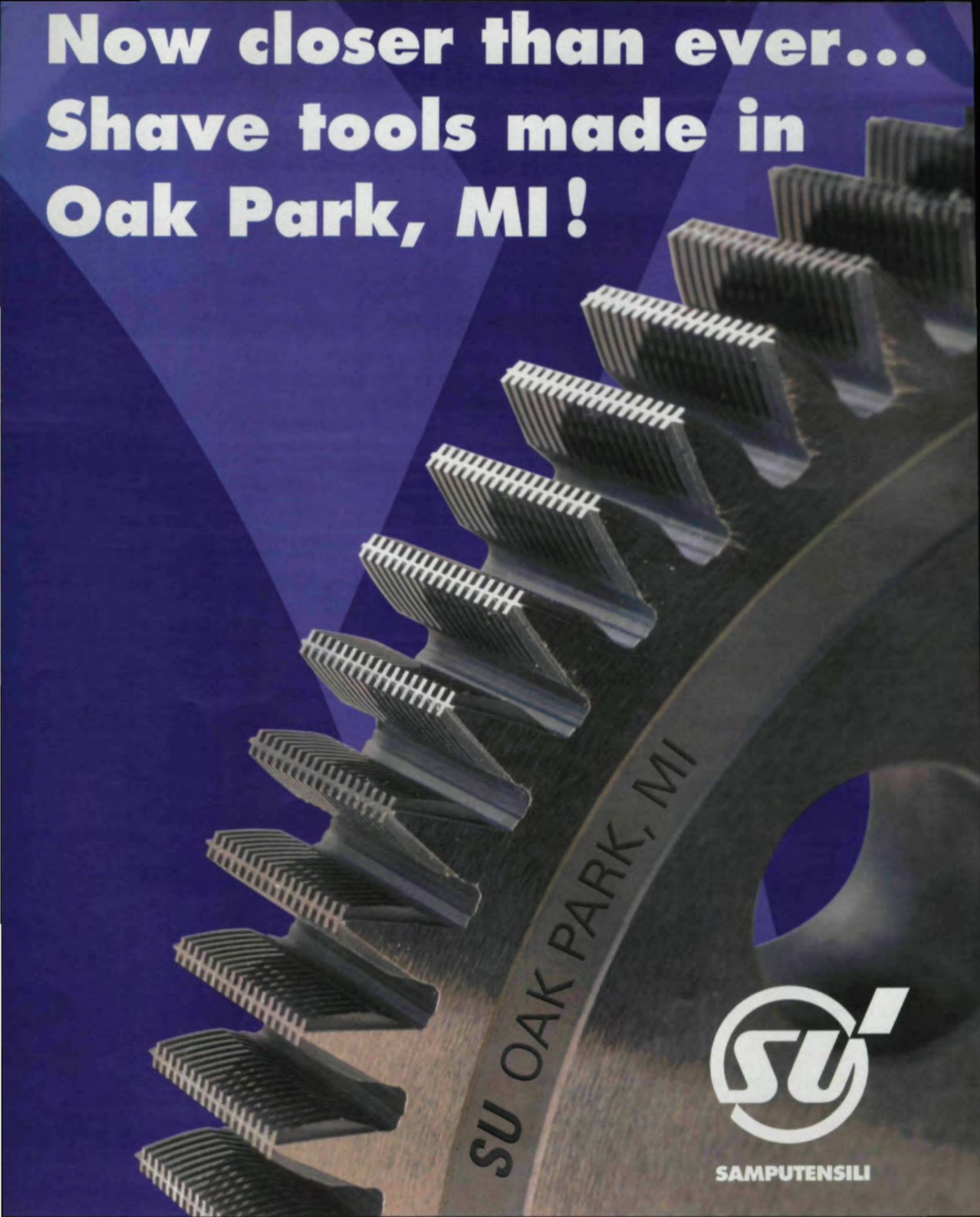
Our ability to measure these attributes and determine their causes has allowed us to help countless people in the power transmission and motion industry through the years. We would like an opportunity to confirm the accuracy (or inaccuracy) of your gears, which we offer as a service, and provide you opportunity to improve your products. We look forward to hearing from you.

Sincerely,

Frederic M. Young

Fred Young

FOREST CITY GEAR


Now closer than ever...
Shave tools made in
Oak Park, MI!
SU OAK PARK, MI
SAMPUTENSILI

Mitsubishi Machine Tools (Booth A1-8242) will demonstrate its line of gear hobbers, shapers, shavers and grinders. Mitsubishi's gear shapers and hobbers have sophisticated cutting mechanisms for high productivity shaping and hobbing of gears up to one meter in diameter. Their gear shavers can handle gears up to 450 mm diameter.

Mitsui Machine Technology Inc. (Booth A1-8733) represents Ikegai Corporation with their full line of CNC machining centers, including gear hobbing, boring, milling, turning, grinding and combination machines. MMT also will show the O-M Ltd. line of vertical CNC lathes and the Howa Machinery Co. line of vertical and horizontal CNC machining centers.

National Broach (Booth B1-7490) introduces three new gear machines at IMTS 98. The NBV 5-8 is a low-cost, small part broaching machine with a low table height and transfer height, eliminating the need for pits and platforms and improving user accessibility. The Red Ring Shavemaster 400 combines advanced software, rigid design and kinematics to enable higher quality, greater efficiency and more complex parts than ever before. The CLP-35 CNC gear checker can inspect profile, lead and pitch of gears, hobs, shave cutters and worm gears. All assemblies on the CLP-35 are hand scraped for absolute static accuracy. In addition to the new machines, visitors will see demonstrations of broaches, shave cutters, hobs, hones, roll form racks and master gears.

Pfauter-Maag Cutting Tools (Booth B1-7150) is a world leader in the manufacturing of hobs, shaper cutters, shaving cutters, form cutters, CBN grinding wheels, thin film coating, bevel gear tools and heat treat service. Their products are sold throughout the United States, Mexico, Canada, Asia, Europe and South America. Pfauter-Maag features tools made of premium high speed steel as well as carbide.

Radyne (Booth C2-5364) will exhibit its Power Integrated Solid State Induction Heating systems—the Power Integrated ScanMaster, incorporating a 250 kW/10kHz IGBT solid state induction power source and Windows-based PC control. The scanner drive uses an AC brushless servo motor, providing some of the fastest scan speeds with the highest degree of positioning accuracy in the industry. Also featured will be the Dual Position Power Integrated Pop-Up Fixture with its integral 160 kW/30kHz IGBT solid state induction power source. This design incorporates three positions, allowing one position to load, a second to heat, and a third to quench. In addition, Radyne will provide a hands-on demonstration of its Apex QA Quality Assurance Monitoring System. Finally, a pre-show press release suggests that "a few other surprises may be awaiting you at the Radyne booth."

Reishauer Corporation (Booth B1-7164) is demonstrating the RZ820, the biggest machine in the Reishauer product line. It is capable of efficiently grinding large, heavy duty gears to a very high quality level. This machine replaces all previous ZB, RZ701 and RZ801 machines. The RZ820 has improved software for increased productivity and improved positioning accuracy of the shift axis, which greatly reduces idle times. Hydraulic tailstock and on-machine fine balancing of the grinding wheel are two new features that enhance throughput.

Richardon (Booth B1-7164) will display the R200 CNC high production hobbers. The Richardon machine comes with a modern 6-axis control coupled with the mechanical stiffness of all cast iron and all vee way construction. It has been configured to handle high helical jobs (-45° to +60°) that are difficult to cut on competitive machines. As standard the R200 CNC can be used to mill or hob gears, and its compact design is intended to please equipment engineers.

S.L. Munson & Company (Booth B2-6555) will be displaying products from Dr. Kaiser Precision Diamond Products Company and DWH Super Abrasives. They will display a complete line of rotary diamond dressers for all gear dressing applications; a new single- and double-side dresser design including an integral rotary root relieving tool for Reishauer SPA and Fässler DSA system dressing units; wear parts produced with polycrystalline diamond surfaces lapped to extremely close tolerances; CNC pro-

THE PAVILIONS

IMTS 98 features three new pavilions this year, bringing the total to 10 shows within the show. Below is a list of the pavilions and their locations in the McCormick Place complex.

Abrasive Machining/ Sawing/Finishing	B1 & B2	North
EDM	C1 & C2	North
Environmental Safety & Plant Management	C1 & C2	North
Factory Automation	D1 & D2	East
Gear Generation	B1 & B2	North
Lasers & Laser Systems	B1 & B2	North
Metal Cutting	A1, D2 & E1	S & E
Metal Forming &Fabricating	B1 & B2	North
Quality Assurance	D1 & D2	East
Tooling & Workholding Systems	E1	East

IMTS FUN FACTS

33 football fields of exhibit space

50 million pounds of equipment

11.8 miles of carpeting

$550 million worth of metalworking equipment on display filing dressers; and rotary diamond dressers for plunge form applications. DWH products include examples of vitrified CBN and vitrified diamond wheels for precision grinding applications.

Schunk Inc. (Booth E1-2471) manufactures and sells a wide range of tooling products, including hydraulic chucks, hydraulic arbors, lathe chucks, chuck jaws, FUNDO—a new hydraulic dowel pin, and TRIBOS—a revolutionary new tool-holding system. Schunk also manufactures grippers for factory automation.

Sigma Pool (Booth B1-7170) will present three new machines, including the Liebherr-Emag hobbing machine, a Klingelnberg-Hoefler crank-shaft/camshaft inspection machine and an Oerlikon bevel gear tester. Also at the booth will be the Sigma Pool's full line of bevel gear generators, plunge shaving machines, shaping machines and the latest in dry cutting technology.

Star Cutter Co. (Booth B1-7182) will display Elk Rapids Engineering CNC sharpening machines as well as its full line of cutting tools, including hobs, milling cutters, pressure coolant & non-pressure coolant drills & reamers, solid carbide tooling and PCD tooling.

SU America (Booth B2-6657) will feature a CNC gear grinding machine, which the company says is accurate, flexible and affordable. It is a form grinder that can use ceramic and/or conventional grinding wheels. It can grind internal gears as well as externals and splines, and it has the latest-generation numerical controls. SU's entire line of gear cutting tools, including carbide hobs, will also be on display.

Modular Vertical Honing System (MVH) from Sunnen Products Company.

Sunnen Products Company (Booth B1-7303) will introduce the new MVH modular vertical honing system incorporating a menu driven industrial PC control. The machine comes in two modules to take advantage of either of Sunnen's honing tool systems. The single stroke honing module incorporates rigid, single-pass plated diamond tooling. The Krossgrinding™ module uses plated diamond in-process expansion tooling for extremely precise roundness and straightness and control of bore size and finish. The Krossgrinding™ system also produces an oil-bearing crosshatch pattern. The two basic modules can also be combined for rough and precision honing in one unit.

Tell Us What You Think . . .
If you found this article of interest and/or useful, please circle **200**.

SuperDry

Cut & Dry.

Mitsubishi presents the world's first dry hob cutting system – the *SuperDry* GN Series.

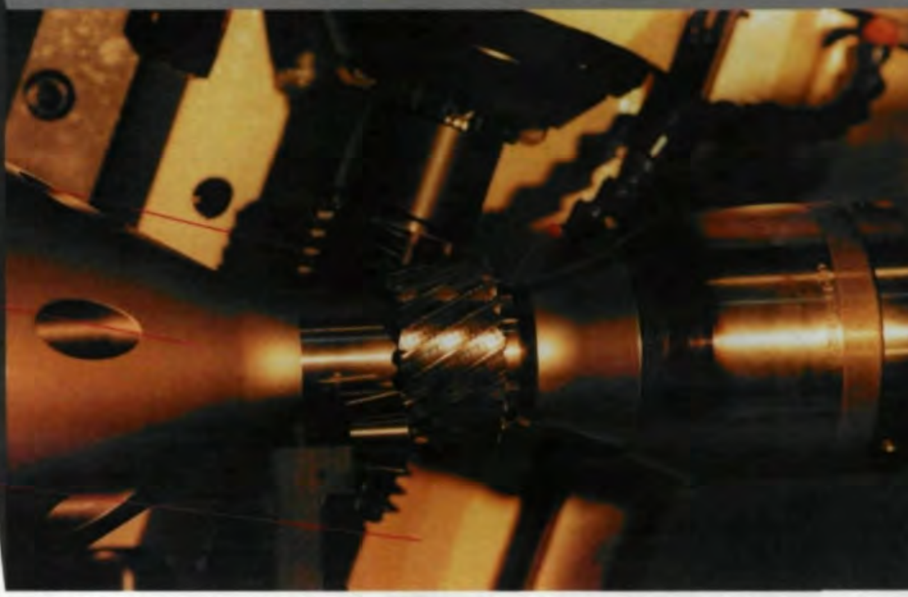
The Mitsubishi team of engineers has successfully developed a totally revolutionary hobbing process that eliminates the need for cutting fluids and the hassle of disposing them. Here are the facts:

1) The *SuperDry* GN Series provides a cutting speed *twice* that of conventional hobbers.

2) Faster cycle times mean less capital investment and reduced labor hours ultimately leading to at least a 40 percent decrease in production costs.

3) Our *SuperDry* hob is made of MACH 7 HSS material with a proprietary coating that dissipates heat and reduces tool wear resulting in an extended tool life of *five times* that over traditional wet cutting.

In short, Mitsubishi's dry hobbing process is safe, clean, and clearly, cut and dry.



Mitsubishi revolutionary "SuperDry" HSS hob cutter design.

A hardened ground helical gear drive table permits highly accurate indexing.

With an innovative horizontal machining method, chips simply drop beneath the workpiece into the conveyor and out of the machine – without the flushing action, or mess, of coolant.

For more information on the new *SuperDry* GN Series dry hobbers and the entire family of our gear machines, contact Mitsubishi today.

See us at IMTS '98 Booth A1-8242

MITSUBISHI
MACHINE TOOLS

MHI Machine Tool U.S.A., Inc.
907 W. Irving Park Road • Itasca, IL 60143-2023
Phone: (630) 860-4222 • Fax: (630) 860-4233
http://www.mhi-mmt.com

CIRCLE 109

Mitsubishi introduces revolutionary dry hobber that reduces gear cutting costs by up to 40%

Kenji Ueno

Itasca, IL – MHI Machine Tool U.S.A., Inc. has announced the introduction of the *SuperDry* – a revolutionary dry hobber system for gear cutting. The new system was featured in the company's booth #A1-8242 at IMTS '98.

The highly efficient Mitsubishi GN10A, GN20A and GN25A dry hobbers eliminate the cost of purchasing cutting fluids and the hassle of disposing of them – protecting the environment and enhancing safety – while reducing overall gear production costs by more than forty percent.

As a key part of its new system, a proprietary dry hob has been developed by Mitsubishi engineers. The *SuperDry* hob is a distinctive cutting tool made of MACH 7 HSS material with a proprietary coating that dissipates heat and reduces tool wear. It allows a hobbing speed of two hundred meters per minute which is two times faster than conventional machining, with tool life up to five times that of a conventional hob. The *SuperDry* machine was redesigned so that the hob operates horizontally to accommodate chip removal by allowing the chips to drop beneath the workpiece directly into the conveyor and out of the machine.

Longer tool life means users will need a smaller tool inventory, with less frequent tool changes or re-sharpening, and less overall machine downtime. Faster cycle times mean fewer machines are required – thereby reducing capital investment, labor hours, and floor space needed. The process is ideal for gears that range from automotive final drive gears and larger truck gears, to automotive pinion and sun gears.

The *SuperDry* machine also offers excellent unmanned operation capabilities, high productivity, and full FMS compatibility. The standard spindle speed is 300–3,000 rpm for the GN10A and 200–2,000 rpm for the GN20A and GN25A *SuperDry* hobbers.

Advantages of Mitsubishi *SuperDry* HSS Hobs

- Doubled productivity with high cutting speeds of (150-200m/min) – 2-times faster than standard tin-coated HSS
- Tool life up to 5-times longer than wet cutting with HSS
- Clean and safe

The Curse of Great Potential

Over recent years, much has been said about the potential environmental and cost benefits of dry gear hobbing, but results have often been inconsistent, costly and generally disappointing. This led to a lull in the application of the technology while the industry reassessed the results of these early efforts.

Mitsubishi is convinced that environmentally friendly processes will eventually be mandated, but decided not to offer dry gear hobbing machines until a truly workable and cost-effective solution was found. With continued R&D in tooling and equipment, the company is now confident that the solution has arrived. Following is a brief recap of the developmental process.

The Mitsubishi *SuperDry* Hob

Initially, Mitsubishi was among the builders who experimented with carbide hobs. After extensive testing Mitsubishi engineers determined that the solution to dry cutting might come from a more user-friendly cutting tool. Unlike carbide, which is expensive and requires great care in sharpening and handling, a High Speed Steel (HSS) solution would be ideal if the base material and coatings could be improved enough to handle the high speeds.

Mitsubishi has now found the solution, with the combination of a new, patent-pending MACH 7 HSS base material and a new, also patent-pending *SuperDry* coating. The new *SuperDry* hob cutter operates at speeds approaching those of carbide cutters, without the high cost or chipping problems associated with carbide hobs.

Dr. Eng. Kenji Ueno
is Vice President,
Gear Machines and Grinders,
at Mitsubishi Machine Tool U.S.A., Inc.,
of Itasca, IL.

The *SuperDry* Dry Hobbing Systems

Mitsubishi engineers also redesigned the machine tools to ensure an optimum fit of the hob and the cutting equipment. (The redesigned machines are fully capable of using carbide hobs, but only the HSS hob provides the optimal performance advantage.) As stated earlier, in the new machines, the hob operates horizontally to accommodate chip removal by allowing the chips to drop beneath the workpiece directly into the conveyor and out of the machine. Compared to conventional gear hobbers, chip-to-chip dead time has been greatly reduced, rapid traverse rate has doubled, and maximum hob length and hob shift length are now the longest of any gear hobber in its class.

A hardened and ground helical gear drive table permits highly accurate indexing. When used in an FMS situation, the *SuperDry* is easy to maintain with chip conveyor and simple hydraulic maintenance schedules. Various monitoring functions simplify FMS operation and ensure that hobbing is performed smoothly.

For more information on the GN10A, GN20A and GN25A *SuperDry* dry hobbers and the complete line of Mitsubishi gear production machinery, contact MHI Machine Tool U.S.A., Inc., Marketing Division, 907 W. Irving Park Rd., Itasca, IL 60143-2023 (630) 860-4222; fax (630) 860-4233.

CIRCLE 175

Designing Reliability Into Industrial Gear Drives

Peter A. Mayo

Introduction

The primary objective in designing reliable gear drives is to avoid failure. Avoiding failure is just as important for the manufacturer and designer as it is for the end user. Many aspects should be considered in order to maximize the potential reliability and performance of installed gearing.

This article is intended to provide some insight into the important elements applied to the design and production of industrial gearing and how the reliability of the gear and drive train is influenced from such measures. Fortunately these days, the gear designer and gear manufacturer have some very sophisticated tools at their disposal to achieve these goals.

There are many gear design codes in use worldwide, including AGMA & DIN standards. The long-awaited ISO standard for gears has recently been approved after more than 20 years in the making. While the gear design codes provide formulas for the determination of various parameters, these equations do not yield a unique or definitive solution.

The actual design process proceeds by the intuitive selection of parameters by the experienced gear designer, who then applies the design code to establish compliance with certain criteria. Regardless of the standard employed, the gear design codes share the common objective of assessing the ability of the teeth to resist surface pitting and cracking when subjected to cyclic loads. The standards also provide guidelines to avoid surface damage to the active tooth flanks by scoring due to inadequate lubrication.

Tooth distress due to pitting is a manifestation of excessive contact (Hertzian) stress. Even more significant is the development of cracks in the critical tooth root fillet region when the tooth bending stress exceeds the endurance strength.

Gears may fail by other means such as wear, plastic flow, case crushing, quench cracks and corrosion, but these modes are not so readily derminant or predictable. Also, more than just the tooth design affects the reliability of the gearing. Other factors influencing reliability include lubrication, construction, the characteristics of the prime mover, bearings, application, assembly and maintenance.

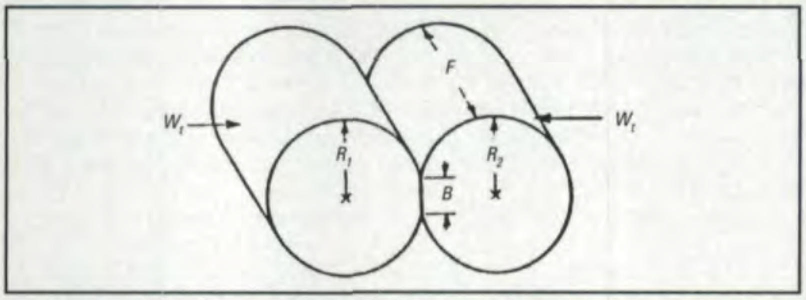


Fig. 1 — Parallel cylinders in contact and heavily loaded. Courtesy of Technomic Publishing (Ref. 1).

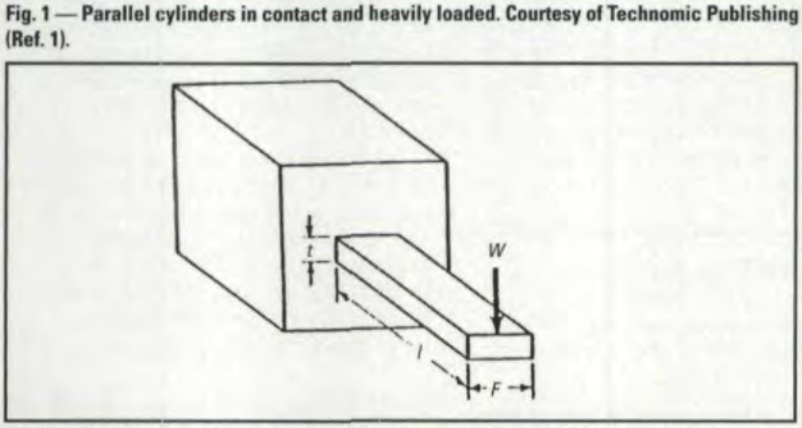


Fig. 2 — Gear tooth as simple cantilever. Courtesy of Technomic Publishing (Ref. 1).

Theory

The power capacity of gears is most often referred to as the gear rating. In order to understand the fundamental design criteria, a brief explanation as to the origins of the rating equations is appropriate. The following derivation should serve as an adequate introduction to the subject, but the reader should refer to Ref. 1 for more information.

For spur and helical gears, the basic equation for assessing the pitting resistance of two engaging teeth is based on the simple analogy of two cylinders of length F pressed together under load W_t, as shown in Fig. 1.

The Hertzian stress for the band of contact is given by

$$B = \sqrt{\frac{16\ W_t(K_1 + K_2)R_1R_2}{F\ (R_1 + R_2)}}$$

where

$$K_1 = \frac{1 - v_1^2}{\pi E_1}$$

$$K_2 = \frac{1 - v_1^2}{\pi E_2}$$

The maximum compressive stress is

$$S_c = \frac{4\ W_t}{F\pi B}$$

These two equations may be combined with Poisson's ratio as $\upsilon = 0.3$ to give

$$S_c = \sqrt{0.35 \frac{W_t}{F} \frac{(1/R_1 + 1/R_2)}{(1/E_1 + 1/E_2)}}$$

The radius of curvature ρ at the pitch circle d is

$$\rho = \frac{d \sin \phi}{2}$$

Radius of curvature for the pinion is $R_1 = \dfrac{d \sin \phi}{2}$ and for the gear

$$R_2 = m_G R_1$$

The compressive stress at the pitch line of a pair of spur gears is therefore:

$$S_c = \sqrt{\frac{0.70}{(1/E_1 + 1/E_2) \cos\phi \sin\phi}} \quad \sqrt{\frac{W_t}{Fd}\left(\frac{m_G + 1}{m_G}\right)}$$

Which is the same as the fundamental rating formula for pitting from AGMA Standard 2001-1995 except for the addition of some derating factors to take account of the uncertainties that prevail in the real world situation over that of the theoretical.

$$S_c = C_p \sqrt{W_t K_o K_v K_s \frac{K_m}{d\ F} \frac{C_f}{I}}$$

From AGMA Standard 2001-1995, the pitting resistance power rating is given by:

$$P_{ac} = \frac{n_p F}{126\,000} \frac{I}{K_o K_v K_s K_m C_f} \left(\frac{d\, S_{ac}}{C_p S_H} \frac{Z_N C_H}{K_T K_R}\right)^2$$

where

P_{ac} is the allowable transmitted power for pitting
n_p is the pinion rotational speed (rpm)
F is the net face width of the narrowest member
I is the geometry factor for tooth pitting resistance
K_o is the overload factor
K_v is the dynamic factor
K_s is the size factor
K_m is the load distribution factor
K_R is the reliability factor
K_T is the temperature factor
C_f is the surface condition factor
C_H is the hardness ratio factor
C_p is the elastic coefficient factor
d is the operating pitch circle diameter of the pinion
Z_N is the stress cycle factor for pitting resistance
S_{ac} is the allowable contact stress number
S_H is the safety factor

Similarly, the tooth bending strength considers the teeth as essentially short cantilever beams. The strength of gear teeth was conceived by Wilfred Lewis in 1893 by inscribing a parabola inside the outline of a gear tooth as shown in Fig. 3. By doing this, the stress along the surface of the parabola is constant.

The location (a) on the tooth where the largest inscribed parabola is tangent to the root fillet region determines the position of maximum stress.

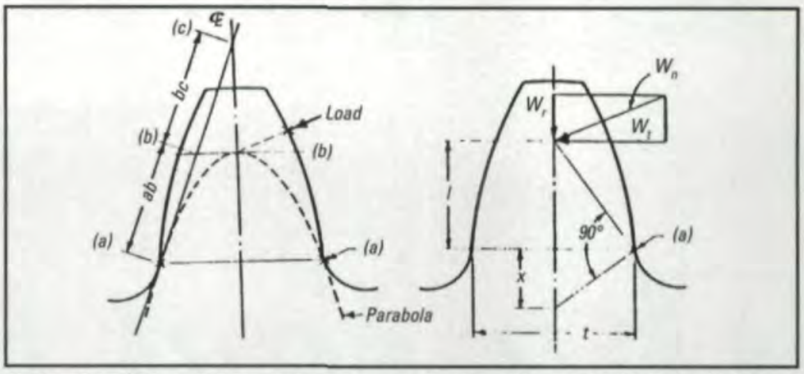


Fig. 3 — Inscribed parabola on gear tooth section. Courtesy of Technomic Publishing (Ref. 1).

The tensile stress at the base of a cantilever is:

$$S_t = \frac{6Wl}{Ft^2}$$

From Figure 3:

$$x = \frac{t^2}{4l}$$

By substitution:

$$S_t = \frac{W}{F(2x/3)}$$

The circular pitch may be introduced into the equations to give:

$$S_t = \frac{Wp}{F(2x/3)p}$$

The term $2x/3p$ was called y by Lewis. This was a paramater that could be determined from a layout of the tooth. With the addition of a stress concentration factor and other aspects, the term is now referred to as the geometry factor J. The latter can be determined with very little effort nowadays using various gear geometry software programs.

The Lewis Formula is written as

$$s_t = \frac{W}{Fpy}$$

Replacing y with Y/π and then diametral pitch P_d for π/p, the Lewis equation becomes

$$s_t = \frac{WPd}{Fy}$$

which is the fundamental rating formula for bending strength from AGMA 2001 (with the addition of some derating factors).

The fundamental rating formula for bending strength from AGMA 2001 is as follows:

$$s_t = W_t K_o K_v K_s \frac{P_d}{F} \frac{K_m K_B}{J}$$

Accordingly, the bending strength power rating from AGMA 2001 is given by:

$$P_{at} = \frac{n_p\, d}{126000\, K_o K_v} \frac{F}{P_d} \frac{J}{K_s K_m K_B} \frac{s_{at} Y_N}{S_F K_T K_R}$$

where

P_{at} is the allowable transmitted power for bending strength
n_p is the pinion rotational speed (rpm)
d is the operating pitch circle diameter of the pinion
F is the net face width of the narrowest member

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J is the geometry factor for tooth bending strength
K_o is the overload factor
K_v is the dynamic factor
K_S is the size factor
K_m is the load distribution factor
K_R is the reliability factor
K_T is the temperature factor
S_{at} is the allowable bending stress number
Y_N is the stress cycle factor for pitting resistance
S_F is the safety factor for tooth bending strength

From these two fundamental rating formulas, the important elements can be considered in terms of dynamic effects, material properties, loading characteristics and tooth geometry. There are also some basic differences between straight spur gears and helical gears to consider.

The way these attributes are chosen, applied and controlled during design and manufacture can significantly influence the reliability of the gears. Further elaboration of these aspects is therefore most relevant to the subject at hand.

Gear Selection

Single helical gearing provides significant advantages over spur gears. It is the meshing of the helical teeth along multiple contact lines inclined at an acute angle to the pitch line that contributes most to the ability of helical gears to transmit more load than straight spur gears.

Moreover, smooth transfer of load occurs gradually and uniformly by a combined sliding and rolling action as successive teeth come into contact along the engaging helicoidal surfaces. This situation is usually referred to as helical overlap. The helical overlap ratio is equal to the ratio of the face width to the axial pitch. Similarly, the ratio of the length of action (length of engagement) of the meshing gear teeth to the transverse base pitch is the transverse contact ratio.

The choice of helix angle for low speed gears is usually a balance between minimizing the axial thrust and maintaining a helical overlap ratio of at least 1.1. This means that the face width of the gear is at least 10% wider than the axial pitch and ensures that before a tooth begins to leave the mesh, the next tooth has already begun to take some share of the load.

Spur gears, on the other hand, rely totally on the conjugacy and contact ratio of the meshing involute tooth forms for the smooth transmission of the load. The average number of teeth in mesh (transverse contact ratio) is usually about 1.2 to 1.7 for both spur and helical gears. Helical gears have typically twice this average amount of teeth in mesh since the overall contact ratio of helical gears consists of the helical overlap ratio plus the transverse contact ratio.

Herringbone or double helical gears typically have helix angles from 20° to 30°. This results in a larger number of teeth in mesh in any given instant and hence high helical overlap ratio. The gears then operate smoothly and are much more tolerant of tooth variations.

Double helicals can, however, be sensitive to variations in accumulative pitch between the two helices that are not synchronized. The floating member otherwise tends to shuttle back and forth, or due to its inertia, the dynamic tooth loads become amplified to the detriment of the gear drive.

The critical tooth bending load for spur gears can occur either at the tip or close to it at the highest point of single tooth contact, but for normal helical gears, the load is designed to be distributed evenly over the oblique helical contact line (or in reality, finite width contact band due to elastic material properties) which extends from the bottom of the tooth to the tip as shown in Fig. 4.

By virtue of their load sharing ability, helical gears can have approximately 50% greater load carrying capacity than the equivalent spur gear of the same physical size when rated to AGMA standards.

Materials

The type and choice of materials obviously plays a vital role in the design and performance of gears. Steel of one type or another tends to be favored for gear materials because it has a high strength to cost relationship. Since gear teeth are subjected to cyclic loads, the fatigue strength rather than the normal mechanical strength determines the allowable design stresses.

The allowable stresses (S_{ac} and S_{at}) are influenced by many factors including hardness, chemistry, cleanliness, residual stress, microstructure, quality, heat treatment, processing practices and number of stress cycles.

There has been a trend over the years towards ever increasing gear tooth hardness. The reason for this is simple. Generally, the harder the material and tooth surface employed, the greater the resis-

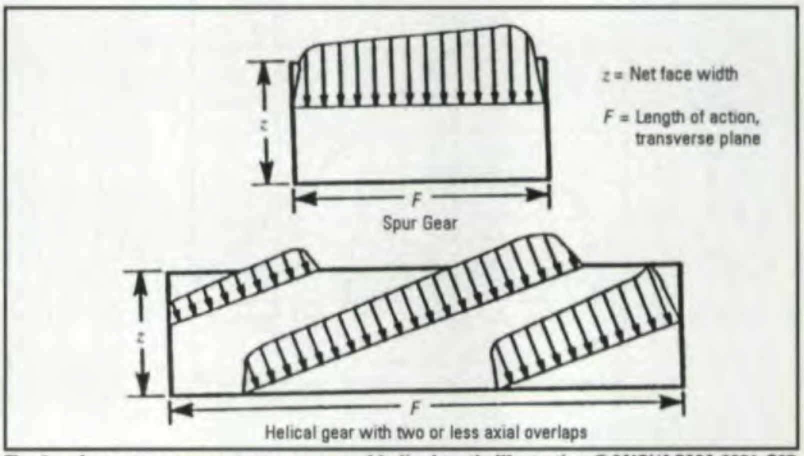

Fig. 4 — Instantaneous contact on spur and helical teeth. Illustration ©ANSI/AGMA 2001-C95.

tance to pitting and tooth bending fatigue. This allows gears to be made smaller in size for a given torque. A reduction in diameter reduces pitchline speeds, and reductions in face width increase the prospects for good gear tooth alignment.

Due to the nonlinearity of the allowable stresses as a function of hardness, the pitting resistance of through hardened gearing varies exponentially with hardness to the power of 1.6. On the other hand, the tooth bending strength varies to the power of only 0.6. The latter explains why increased tooth bending strength is usually achieved by a larger tooth size or greater face width.

Gear steels typically have carbon contents in the range of 0.3% to 0.5%, and they are alloyed to enable the desired hardness on the flanks and roots of the teeth. Below 0.3% carbon the gears have poor wear resistance. The 0.3% carbon alloy steels produce the greatest toughness compared with higher carbon steels, but 0.4% carbon has higher hardness potential. Above 0.5% the toughness of these steels tends to be quite low.

The AGMA gear rating codes assign allowable design stresses based on the verification undertaken to confirm the cleanliness, quality, homogeneity and integrity of the materials employed. It should be noted that for a given hardness, the selection of allowable stress must be made from a design range.

The lower values for allowable stress might be appropriate for castings which are relatively free of harmful defects, but which may contain some innocuous defects such as discrete gas holes and porosity, but not cracks and shrinkage.

Intermediate values for allowable stress may be applied to commercial grade forgings and maximum values for extra high quality steel forgings, such as those offered by the Electro Slag Refined steel making process. Vacuum degassing has also been a steel making process of significant benefit to improving gear steel quality.

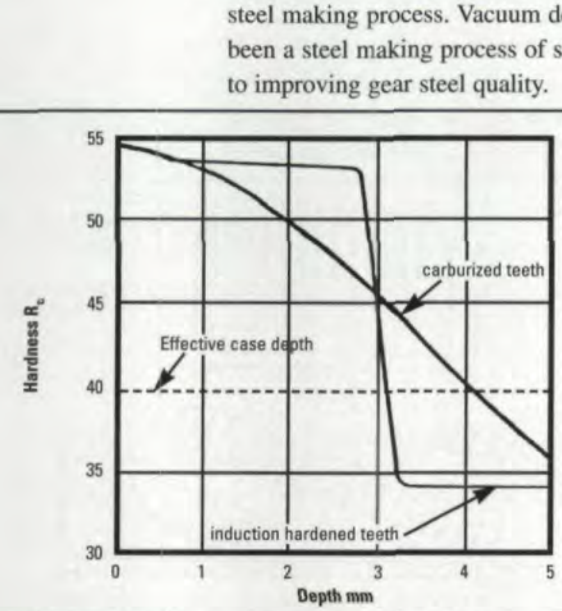

Fig. 5 — Induction hardened & carburized case core relationship.

The most important part of a gear contains the teeth, and so it often pays in the long run to invest in the best material available. For cyclically loaded components such as bearings, it has been well established that cleaner steels improve the life expectancy. The ESR steels boast extremely clean microstructures, which are inherently very resistant to the initiation of fatigue.

Lower alloy steels can be employed where low hardness will suffice or when the teeth can be gashed prior to heat treatment. The structure of the gear should also maximize the potential of the heat treatment. In order to encourage quicker quenching rates, the sides of large solid gears or pinions can also be dished to reduce the mass effect.

Even with high hardenability alloy steels, the teeth of coarse pitch, through hardened gears benefit significantly when practical from pregashing the teeth prior to heat treatment. This ensures uniform and effective hardening to flank and root when the hardness required is greater than 300 HB.

These days, foundry facilities and expertise have been developed to the point where even large gear castings can now be water quenched to produce a very tough and much improved fatigue resistant martensitic microstructure not previously possible by air hardening (normalizing), which produces a bainitic microstructure.

Surface Hardened Gearing

The pursuit of ever harder gears has seen the emergence and now dominance of case carburized gears in gearboxes. Typical hardnesses for such gears would be 58 to 62 Rc, so the teeth are invariably precision ground after hardening. The gears therefore boast high accuracy, high surface load carrying capacity and smoother operation.

As a consequence of surface hardening gear teeth, beneficial residual compressive stresses are induced into the surface layer. This effectively reduces the tensile bending stresses in the critical tooth root fillet region. It is most important therefore not to grind the roots of the teeth as the grinding can remove the residual compressive stress. Shot peening, however, can be employed to restore the residual stress if grinding becomes necessary to remove excess distortion.

Where the gearwheel itself might be too large to be case hardened, a significant benefit can still be gained by the use of a case hardened pinion meshing with a through hardened gearwheel. This has evolved even though pinions typically need only 20% to 40% hardness differential over and above that of the gearwheel to account for the greater number of stress cycles experienced by the pinion. The tooth hardness of the gearwheel might typically range from 300 HB up to 400 HB.

The benefit of this combination may be explained in terms of the ability of alloy steel gear teeth to strain harden under the influence of cold work by the much harder surface of the engaging pinion teeth. AGMA Standard 2001 recognizes this phenomenon and applies the hardness factor C_H to achieve a rating gain corresponding to the subsequent increased hardness. Actual measurements would suggest that alloy steels can increase the active tooth surface hardness by as much as 15% in this action.

Even though the carburized teeth can develop beneficial residual compressive stresses in the surface layer, the drawback experienced with producing carburized gear teeth is the volumetric expansion that occurs when the tooth surface is enriched with carbon and subsequently transformed to martensite when quenched. The associated distortions require further finishing in order to correct these distortions.

These movements can be quite uneven. Single helical teeth tend to unwind slightly, and long pinions can develop an hourglass shape. Therefore, subsequent finishing removes more material than expected, and thin localized case depths can occur.

Testing of the surface hardened layer is an important issue. Some tests, such as hardness, can correlate the success of the hardening process on the actual gear teeth with the test piece. Most of the testing, though, must be done on test samples independent of the actual work piece. It is therefore essential that the test samples be totally representative of the gear.

Induction hardening offers an alternative means to provide high surface hardness and increased strength in the root of gear teeth. Hardnesses typically 50 to 56 Rc can be readily achieved depending on the carbon equivalent, the type of steel and the quenching rates employed. The advantage of induction hardening is considerably less distortion than with carburizing.

However, induction hardening requires a heavier case thickness. This is because the induction hardened case has a more abrupt transition from case to core in contrast to the more gradual transition of carburized surface layers. This is illustrated in Fig. 5. Induction hardened case depth is also measured differently.

In specifying the case depth, it is important to ensure that the maximum subsurface shear stresses do not occur at this transition zone.

Nitriding also provides a very useful surface hardening technique where very high hardnesses are needed. It is limited, though, where only a thin surface layer can be used such as that found in small and high speed gears.

Using a very hard pinion is contrary to the practice of using a sacrificial pinion with particularly soft teeth. The disadvantage of allowing the pinion teeth to become worn and misshapen is uneven wear on the gearwheel teeth.

Very hard pinion teeth do not yield or suffer so easily from the usual perils of through hardened surfaces such as pitting, spalling and scoring (assuming adequate lubrication). As a result, the pinion can then be expected to always maintain its true involuted shape and in turn help to maintain the tooth profile of the gearwheel. The gearwheel teeth might otherwise suffer if the pinion tooth form became distorted for any one of the forgoing reasons.

Dynamic Considerations

The dynamic factor K_v takes into account the internally generated gear tooth loads induced by non-conjugate action (non-uniform motion) of the meshing teeth. Dynamic forces arise from the relative accelerations between the gears as they vibrate in response to excitation referred to as "transmission error."

Ideally, gears should have uniform transfer of motion from input to output gear. It is impossible to produce perfectly true gears, and it is the departures from the ideal geometry that contribute to non-uniform motion and transfer of load. These deviations can be caused by many factors including residual stresses in materials, variations in material metallurgy and hardness, poor machine tool condition, inaccurate tooling, poor machine setting practice and many more. Stiffness of the teeth and mesh also contribute to non-uniform motion.

The common source of transmission error occurs from variations in the elemental tooth parameters. These individual elemental parameters include the tooth spacing (pitch), involute profile, tooth alignment (helix) and runout (eccentricity).

Gear metrology is used to identify and quantify the various elemental parameters of the teeth and gear body. The somewhat peculiar geometry of gear teeth requires some unique measuring methods and facilities. In addition to verifying compliance with allowable tolerances, gear metrology can be used as a diagnostic tool to identify the source and cause of the deviations.

The objective is therefore to minimize the deviations by taking appropriate action during design and manufacture. In some situations, the deviations can be removed by subsequent processes such as grinding. In many cases, though, the deviations remain in the installed gearing. By improving gear accuracy, the dynamic load induced by non-conjugate meshing of the gear teeth can be reduced.

Fig. 6 provides some indication of the accuracy required as a function of pitchline speed.

The measurement precision needs to be better than that of the component tolerance by a factor of approximately 10. Gear tolerances vary from 50 to 100 μm right down to only several microns. Therefore, gear measuring equipment needs to have an accuracy of just 1 to 2 microns when measuring a component having a tolerance of 0.01 mm.

High accuracy, particularly in gears, can incur considerable manufacturing cost. The desired gear accuracy should therefore by chosen carefully so as to obtain maximum benefit without excessive cost.

Load Conditions

The load distribution factor K_M accounts for less than uniform load across the width of the teeth and from one tooth to the next. Ideally, the load should be uniform over the full width of the teeth and the full working depth.

Many factors can affect this condition, which in turn greatly affects the reliability and performance of the gears. Poor load distribution can occur due to misaligned shafts, excessive clearance in bearings, deflections in teeth, shafts and gear structures. Obviously, misalignment of shafts must be minimized or avoided. The deflections, on the other hand, are somewhat unavoidable since they invariably result from the applied loads.

Traditionally, the supporting structures of gears (webs, stiffeners, diaphragms and tubes) have relied almost totally on empirical or intuitive techniques. As a consequence of the ability of modern gearing to transmit high loads from the use of very hard tooth surfaces, higher achieved accuracy and higher pressure angles, the imposed loads increase the stress and strain on the supporting structures.

Empirical data becomes somewhat scarce for such situations, and so to improve the potential reliability of these high specific load (load per mm of face width) gear wheels failing to perform satisfactorily from lack of strength or rigidity, the method of finite element analysis (FEA) for stress distribution (and strain) provides a valuable tool for the evaluation of the loads in the structure of gearwheels.

Fig. 6 — Gear quality as function of pitchline speed.

Accurate stress analysis also permits the elimination of redundant material, which contributes to reductions in unnecessary cost, weight and inertia effects. However, it has been found from these studies that it is in fact the need for stiffness rather than the level of stress that most often dictates the selection of structural dimensions. The reason is that uniform load distribution and load sharing between adjacent teeth is significantly influenced by the deflections of teeth, rim and supporting structure.

The dynamic gear alignment can be very much determined by the accuracy of manufacture. During gear cutting, significant movement can occur in the blank due to the release of residual stress. The circularity of a gear can be affected by as much as 10 to 12 mm, which would be considerably outside the permissible runout tolerance.

Residual stress in the form of tensile hoop stresses induced by improper welding practices or simply by the excessive interference with shrink fitted gear rims onto a hub can create a parasitic stress condition that combines with the normal (tensile) tooth bending fatigue stresses, which in turn often leads to broken teeth.

The dynamic load distribution can also be affected by elastic tooth deflections. Since the deflected tooth is slightly behind where it should be, the approaching tooth engages with an impact. Tip relief and sometimes root relief is applied to account for this interference. It is imperative that the tip relief, especially with spur gears, does not reduce the contact ratio below 1.0, though. A minimum contact ratio of 1.2 ensures conjugate action (uniform motion) is maintained.

The correct amount of tip relief is a function of the tooth stiffness and applied load. Tip relief can be imparted to the teeth by modifications incorporated into the gear cutting tool. The actual tip relief produced in the teeth is a function of the diameter and addendum modification.

For standardized tooling, gearwheels tend to receive a generous amount of tip relief, but pinions tend to acquire little or none. Fortunately, in speed reducing drives, the tips of the gearwheel teeth engage first. The interference of the deflected tooth (Ref. 3) can be seen in Fig. 7.

Uniform load distribution can likewise be affected by the elastic deflections of long and slender pinions. Such pinions can suffer from excessive bending and torsional deflections. It is especially important with case hardened gearing


HEAT TREATING

to apply a small amount of end relief to avoid loading the ends of the teeth.

Rim Proportions

The rim factor K_B considers the reduction in ratings when the rim thickness is less than 1.2 times the tooth depth. Excessive deflections of the rim can seriously impair the reliability of the gear to perform properly (Ref. 4).

While AGMA suggests a factor of 1.2 times tooth depth as a safe minimum, the gears with higher specific load and those gearwheels with very wide face widths require proportionally greater rim thickness or other means of support to counter these deflections.

Tooth Geometry

The tooth shape or geometry also plays a major role in the overall performance of gearing. The constant angle between the line of action of involute gear teeth and the common tangent to the kinematic pitch circles at the pitch point is called the pressure angle.

Standard pressure angles evolved to rationalize the tooling required to produce gears. The Australian standard AS 2938-1995 for gearing promotes the use of ISO 53 tools with 20° pressure angle. Various other standard pressure angles are used, though.

The operating pitch line speed of the gears tends to govern the choice of pressure angle for a given application. At one extreme, pressure angles of 14.5°, 15°, 16° or 18.5° are employed to minimize noise and vibration excitation at high pitch line speed by virtue of a greater number of teeth in mesh (higher contact ratio) at any particular instant with such pressure angles.

With higher loads and slower speeds, it becomes more important to maximize tooth bending strength and pitting resistance. Pressure angles of 20°, 22.5°, 23° or 25° can be used on gears with high specific tooth loads and low pitch line speeds.

The disadvantage of high pressure angles is the reduction in transverse contact ratio (number of teeth in mesh) and a narrower top land thickness. Despite this, the use of a 25° pressure angle for low speed, high torque drives has achieved wide acceptance in many industries following its success in the gearing of draglines used in open-cut coal mines.

The geometry factors I for pitting and J for tooth bending strength both benefit from pressure angles larger than the traditional 20°. This is because for a given diameter, as the pressure angle increases, a lower radius of curvature on the teeth can be obtained, and so in turn, the Hertzian stress is reduced.

Similarly, a gain in tooth bending strength results from the increased tooth thickness at the base of the tooth. The 25° tooth form will carry about 20% more torque than the 20° nominal tooth form, all other factors being equal.

Additionally, the tooth geometry and effective pressure angle can also be influenced by addendum modification. Positive addendum modification is considered crucial for low numbers of teeth (typically < 17) where the base circle, from which the involute originates, intrudes on the active portion of the tooth profile.

Positive addendum modification alleviates the undercut in the root fillet region that ensues from this intrusion. Undercut can seriously affect the tooth bending strength by reason of the narrower (undercut) tooth thickness at the base. The contact ratio can also be adversely affected from the reduced length of active tooth profile.

The gearset likewise benefits from an increase in the relative radius of curvature on the pinion tooth, thus improving the pitting resistance. The maximum sliding velocities occurring at the extreme points of engagement can also be optimized with judicial selection of addendum modification.

The gears will not perform very well or at all without proper lubrication. Besides reducing friction and preventing wear, the lubricant is also relied upon to remove heat from the tooth surfaces.

In the case of the total loss spray systems used on the open gears of metalliferous grinding mills, cement mills, kilns and sugar mills, the air blast used to purge the lubrication system should not be so great that it displaces the grease from where it has just been deposited on the pitchline of the teeth. This can occur particularly with the low base oil viscosity grades or the latest nonchlorinated solvent group of lubricants.

A reasonable surface finish is necessary to ensure the contact between meshing teeth. While a smooth surface finish is desirable, it is sometimes the bane of manufacturers to achieve a satisfactory surface finish with the materials employed. As the purity of gear steel increases, the machinability of such materials tends to decrease. The cleaner steels improve fatigue resis-

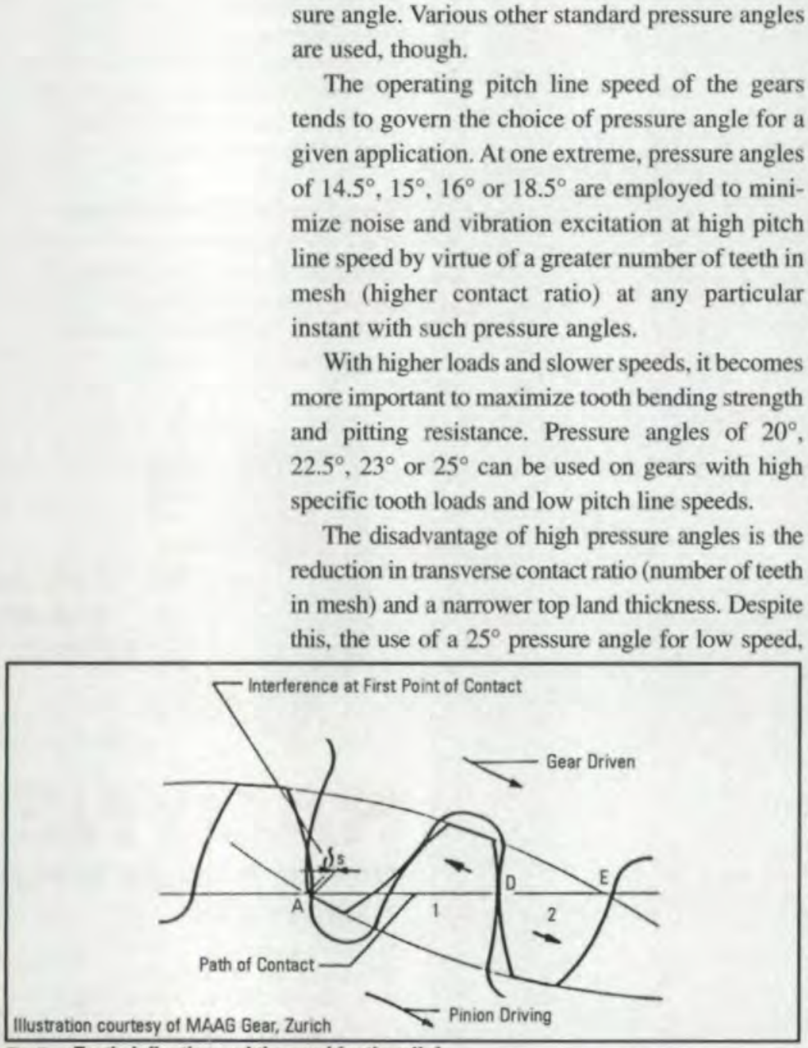

Fig. 7 — Tooth deflection and the need for tip relief.

tance but some trade-off sometimes occurs with surface finish. To date, none of the international gearing standards specify limits on surface finish other than the benefit of smooth pinion surfaces to encourage work hardening.

In addition to maintaining an acceptable surface finish on the teeth, the root fillet region deserves special attention. In this region, removal of lineated tool marks aligned with the axis of the tooth by finishing or shot peening can eliminate undesirable stress raisers.

Installation and Commissioning

Upon completion of the individual gears, the two components can be meshed together in the workshop to show how well the gears might ultimately perform. A simple mesh test with the gear and pinion set at the correct center distance and planar relationship can verify satisfactory compliance of runouts, profile, backlash and load distribution rather than taking for granted that these will be correct upon installation.

Dynamic testing methods that simulate the operating conditions can be employed to provide further information. A no-load test can be used to verify satisfactory gear design, manufacture and assembly in terms of potentially acceptable noise and vibration levels, satisfactory gear tooth alignment and bearings and lubrication operation.

A load test using either a brake or back-to-back test can simulate the actual service conditions for a much better validation. The back-to-back test can only be done with a pair of mirror-image gearboxes and not without incurring some additional cost.

The assembly of gearboxes has been found to directly affect their reliability when put into service. Since the gear case will have been machined to suit the bearing race size, final assembly must maintain the proper fit. The overzealous use of sealing compound on the gear case halves or simply a gap between top and bottom will allow the outer bearing races to spin, causing premature failure of the bearings and damage to the gear case.

In the case of rotating shafts, the fit of bearings to the shaft is also critical. The current design of some roller bearings uses reduced race thickness. The consequence can be increased hoop stress culminating in breakage of the race, particularly if the shaft size is at the upper tolerance or greater.

The installation can obviously have a direct bearing on the reliability of the gearset. The gear alignment and uniformity of load distribution across the width of large gears can be determined from non-contact temperature measurement using an infrared pyrometer (thermometer). The foundations and gear support structure must also be adequate and stable to maintain these alignments.

In bolting down the gearbox, an uneven foundation will distort the gear case. Even just a small amount of twist in the gearcase can significantly affect the gear alignments. This aspect can sometimes be used to advantage to achieve proper gear alignment when the gearbox bores are not machined just right.

Couplings play a vital function in the success of a gear drive. A case in point is herringbone or double helical gearing, which has the special need where the couplings must not restrict the axial movement of the floating member. The choice of couplings may also need to consider the necessity to isolate the gear elements from sources of resonant vibrations (or the coupled components from the gear mesh excitation).

Proper care and maintenance practices form an essential ingredient in order that the potential or intended reliability will be realized in service. The benefit of a regular inspection program should not be underestimated. Condition monitoring methods such as vibration levels, temperature measurements, oil analysis, visual inspections and nondestructive testing provide valuable information to assure the long-term performance and reliability of any critical or important gear elements.

In conclusion, the reliability of any gear set is influenced by many factors. The production of gears involves some very sophisticated machine tools and specialized processes and procedures, so gears tend to be rather expensive items. The designer plays a vital role in the success of these gearsets, but so too do those involved with the manufacture, installation and maintenance.

The whole of life cost should be considered since mediocre gear quality gearing rarely proves worthwhile, since the downtime can also be very costly. A considerable effort is expended in producing both good and not-so-good gearing, but hopefully the preceding information provides some insight into the many aspects that contribute to achieving the best and most reliable gearing. ○

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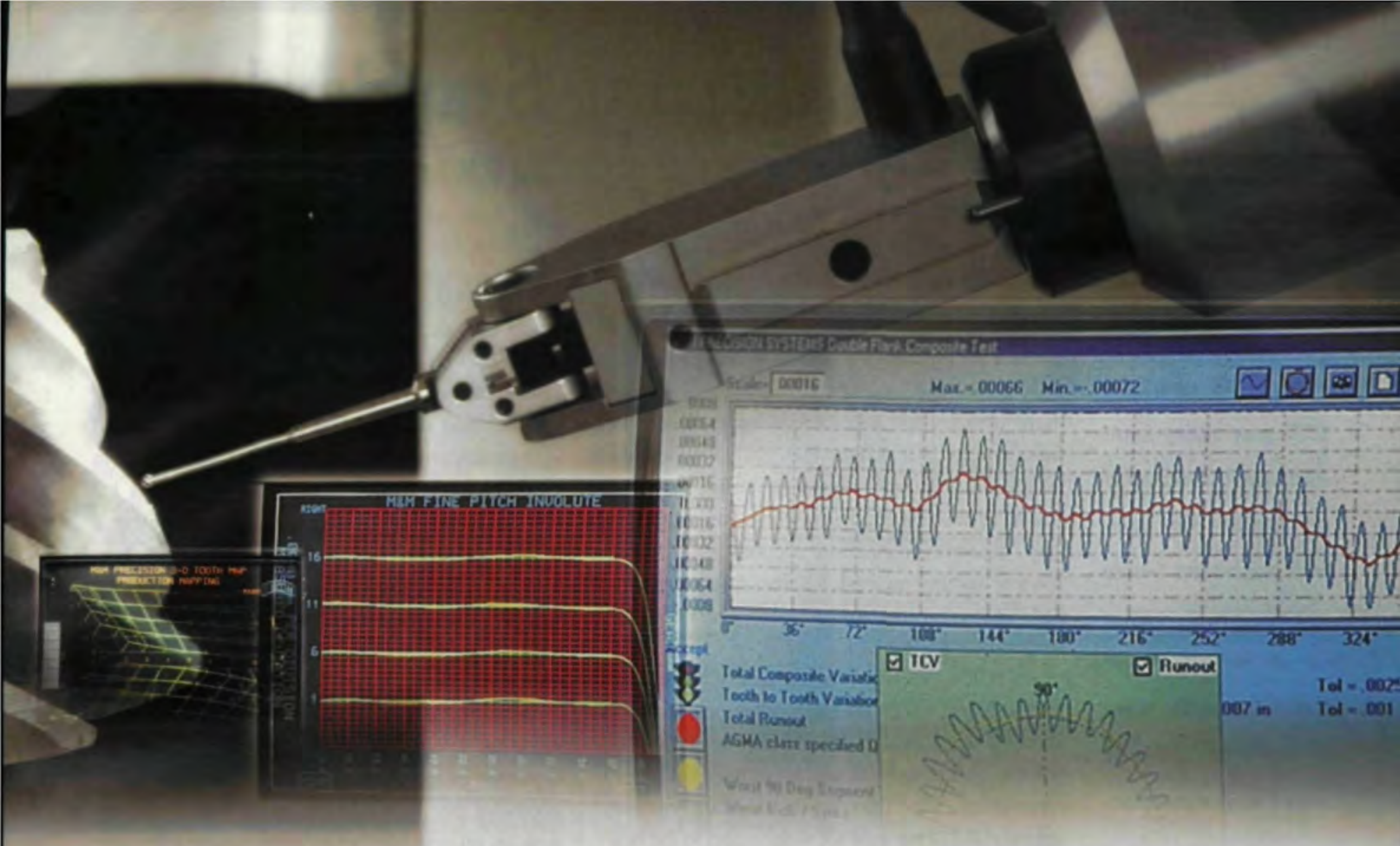
This article first appeared in the proceedings of the "Gear and Shaft Technology Seminar" held by the Institute of Materials Engineering Australasia Ltd (formerly Institute of Metals and Materials Australasia Ltd) at Wollongong, NSW, July 29, 1997.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Comparing Standards

The keys to understanding ISO 6336-1 gear rating

Don McVittie

This is the second of a series of articles introducing the new ISO 6336 gear rating standard and its methods of calculation. The opinions expressed herein are those of the author as an individual and not necessarily those of any organization of which he is a member.

One of the best ways to learn the ISO 6336 gear rating system is to recalculate the capacity of a few existing designs and to compare the ISO 6336 calculated capacity to your experience with those designs and to other rating methods. For these articles, I'll assume that you have a copy of ISO 6336, you have chosen a design for which you have manufacturing drawings and an existing gear capacity calculation according to AGMA 2001 or another method. I'll also assume that you have converted dimensions, loads, etc. into the SI system of measurement.

ISO 6336 looks into more details of your design, so it requires about 80 input values compared to about 60 for AGMA 2001. The additional information required includes details about things like blank construction, root fillet finish, lubricant viscosity, arrangement of the gear set with respect to the bearings and location of light load contact pattern with respect to the bearings. We'll cover these differences as they apply, section by section.

Input Data

Most of the input data is similar to other rating standards, like AGMA 2001. The basic gear geometry values of numbers of teeth, center distance, outside diameters and face widths are the same. ISO 6336 requires the input of minimum safety factors for surface contact (pitting) and root bending stresses. The safety factors are used in the calculation of allowable working stresses in parts 2 and 3. While safety factors as low as 1.0 are permitted, most users will want to select a higher value for root bending, such as 1.2 or more. If the required safety factor is not achieved with the input values of materials, geometry and load, one or more of the inputs should be changed until the required safety factor is reached.

Module, normal pressure angle and helix angle are determined at the reference (generating) diameter, based on a carefully defined basic rack geometry per ISO 53. The basic rack is a gear with an infinite number of teeth whose reference plane is located where the tooth thickness and the space width are equal. The basic rack is not the cutting tool. It is conjugate to a hypothetical rack shaped tool which defines the tooth shape.

Tooth thickness is defined in terms of the profile shift factor X of a "zero-backlash" gear pair derived from the basic rack geometry. (See the article in July/August 1993 and a correction to one of the formulas in January/February 1994 for more information on basic racks and profile shift.) The X factor and the tool geometry—tool addendum, tip radius, protuberance and finish stock allowance—are critical to the calculated result. Figs. 1 & 2, from the AGMA ISO 6336 program user's manual, illustrate the principle. It is important to note that the definition of tool addendum used in ISO 6336 may not be the same as you are using in your present gear rating program, so that value may have to be adjusted for input to the ISO 6336 calculation. Fig. 3 illustrates the ISO definition.

The input power or torque must be specified, since the ISO dynamic and load distribution factors are load dependent. Although the ISO standard lists numerical application factor values which are identical to AGMA's, those values should be used with caution. The ISO definition of application factor is broader than AGMA's, including many "dynamic" effects, such as the effect of tooth spacing deviations which are part of the AGMA dynamic factor. The differences are most noticeable in the calculat-

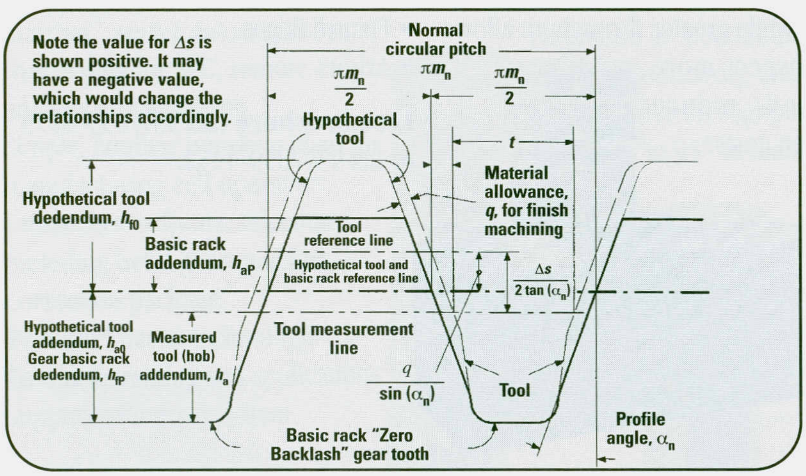

Fig. 1 — Basic rack and hypothetical tool represented in rack form (©AGMA)

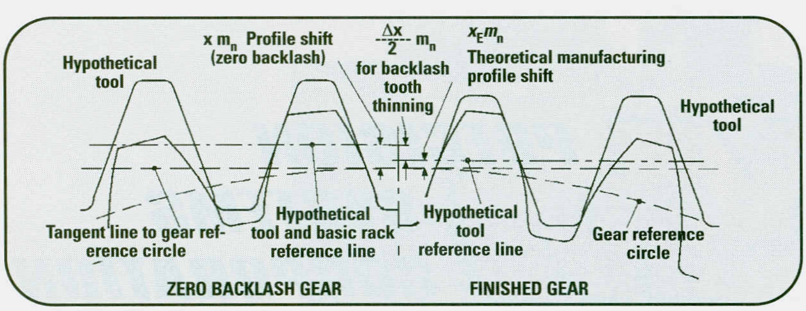

Fig. 2 — Hypothetical tool with zero backlash and finished gear (©AGMA)

ed capacities of large, low speed, relatively less accurate gears, such as those found in bridges, mill drives, train positioners and antennas. The ISO dynamic factor estimates gear pair resonance as an equivalent single mass system with the mesh stiffness as a spring. The ISO dynamic factor is strongly influenced by the ratio between the mesh pass frequency and the calculated resonant frequency of the single mass model. High speed, lightly constructed, lightly loaded gear sets have high ISO dynamic factors and low speed, heavily constructed, heavily loaded gear sets have very low ISO dynamic factors. Many of the "dynamic" effects which AGMA includes in its dynamic factor are shifted to the application factor by ISO. You will need to be careful in your evaluation of the differences and in the application factor you choose to evaluate your gear sets. The product of $K_A \bullet K_v$ should be nearly the same in both systems if rated gear capacities are to be comparable.

Calculating the Dynamic Factor

The ISO dynamic factor calculation estimates the rotational moments of inertia of the pinion and wheel from the inside diameters of their rims and their calculated root diameters with an adjustment for webs, if any. The gear blank geometry must be provided for this calculation. The mesh stiffness and root diameters are calculated from the X factors and tool geometry, so accuracy of these input values affects the dynamic factor. A smaller portion of the dynamic factor is a function of tooth accuracy, expressed as an ISO 1328 quality number. There is no consistent way to estimate ISO 1328 quality from AGMA 2000 quality numbers, so the actual tolerances must be known or calculated, then ISO 1328 quality numbers recalculated from the tolerances. The fundamental equation in ISO 6336–1 for dynamic factor is:

$$K_v = (N \bullet K) + 1$$

Where:

N is the ratio of mesh pass frequency to resonant frequency.

K represents the effect of gear tooth accuracy.

Typical values of K_v for industrial enclosed drive gears are 1.05 or less.

The basic ISO dynamic factor is Method B of ISO 6336-1. Methods C and D are simplified versions of method B with applicability restricted by their underlying simplifying assumptions.

The AGMA dynamic factor is included in ISO 6336-1 as alternative method E for those who prefer it. If method E is used, it might be appropriate to use a lower (AGMA) value for application factor.

Evaluating Load Distribution

Much of ISO 6336-1 is devoted to various methods of evaluating load distribution across and between the teeth. Although different influence factors are defined for load distribution in root bending stress and contact stress in the face and transverse directions, the principal influence on the load capacity of the gear set is $K_{H\beta}$, the face load distribution factor for contact stress. $K_{F\beta}$, the face load distribution factor for root bending stress has a similar value and the transverse load distribution factors $K_{H\alpha}$ and $K_{F\alpha}$ are usually nearly 1.0.

The correct evaluation of the load distribution factor is critical to obtaining satisfactory results with ISO 6336, just as it is with other gear rating systems, such as AGMA 2001. I suggest that you try the various optional methods in ISO 6336 while making comparative capacity calculations with your own gear designs, to develop experience with them. Since the mathematical definition of face load distribution factor is the same in ISO 6336 as in AGMA 2001, you should expect similar values of this factor from either method. If the values are not within 10% of each other, you should investigate further and resolve the difference. It may be necessary to make an experimental investigation or a detailed calculation of deflections under load to get the "right" answer. It is important to consider the effects of manufacturing variations as well as the average meshing conditions in this analysis. Method A is commonly used to evaluate $K_{H\beta}$—assuming that a mathematical model gives satisfactory results. Recent investigations by an ad hoc group sponsored by ISO/TC60/SC2/WG6 in which the same gear sets are calculated by different "correct" mathematical models have shown that the value of $K_{H\beta}$ can vary more than 10% depending on small variations in the calculation method and the underlying assumptions. Future editions of ISO 6336-1 are

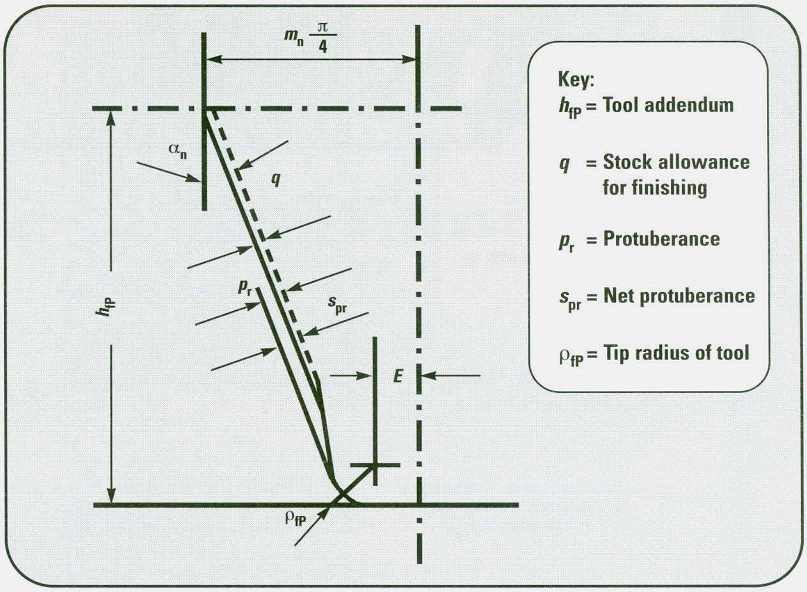

Fig. 3 — Dimensions and basic rack profile of the teeth (finished profile with undercut).

Don McVittie
is one of Gear Technology*'s technical editors. He is president of Gear Engineers, Inc., Seattle, WA and a former president of AGMA. McVittie is a licensed professional engineer in the state of Washington and has been involved with gear standards development for more than 25 years.*

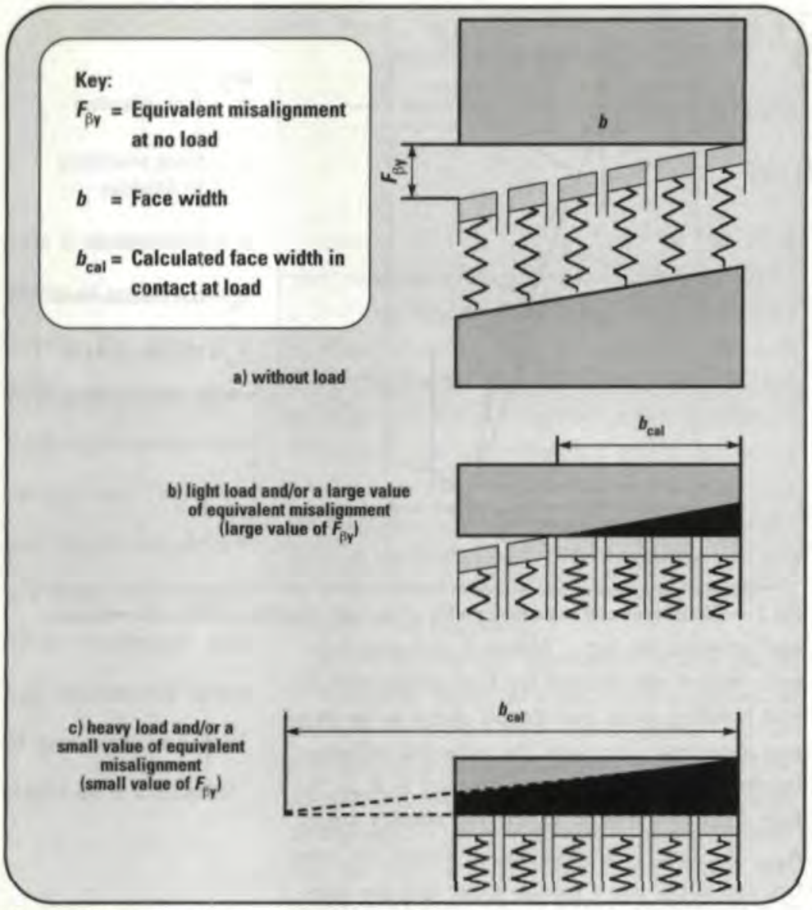

Fig. 4 — Distribution of the load along face width with linear equivalent misalignment.

expected to specify a more detailed calculation method as method B and to restrict method A to experimentally verified values of $K_{H\beta}$. In general, values of C_m calculated by AGMA 2001's empirical method are somewhat more conservative than the values of $K_{H\beta}$ commonly used as method A by European manufacturers of industrial enclosed drives.

Methods B, C and D are based on a simple mathematical model of mesh deflection under load. The mesh stiffness constant is the same as the spring constant used in the method B dynamic factor. That constant is about 1/3 higher (more stiff) than the factor which was used in the AGMA standards, so load distribution factors calculated by methods B, C and D tend to be higher than AGMA values.

We will be back on more familiar ground in the next article, which looks at the details of calculating surface contact and root bending stresses in ISO 6336-2 and -3.

Method B assumes a linear load distribution model with constant mesh stiffness, similar to the analytical method which was used in pre-1995 editions of AGMA 2001 (See Fig. 4). The fundamental input is the total loaded mesh misalignment including the effects of manufacturing variations, and elastic deflections due to load. The deflection of the wheel shaft can optionally be included in the total mesh misalignment. Method B doesn't implicitly recognize the benefits of tooth crowning, but it does recognize the benefits of running in and helix modification to compensate for torsional and bending deflections of the pinion. The evaluation of total misalignment is very important, since it has a large influence on $K_{H\beta}$ and the calculated capacity of the gear set. If you elect to use method B, be sure to read all of the detailed instructions and limitations which are included in section 7 of ISO 6336-1.

The AGMA computer program for ISO 6336 allows the input of a $K_{H\beta}$ value (method A), the input of a mesh misalignment (method B) or the input of detailed information about the mesh to calculate $K_{H\beta}$ according to method C. Method C is limited to the case where the center of the pinion is within 30% of the center of the bearing span. Gear sets which do not meet this limit must be calculated by methods A or B. A subset of method C (C1) is a further simplification for symmetrically located pinions, but the same inputs are required. Method C considers the gear set arrangement, pinion shaft deflection, type of crowning or lead correction and the location of the light load contact pattern, so inputs for all of those factors are required. Method C is a bit complicated to use in hand calculations but, once programmed, allows the user to see the effects of changing pinion proportions and crown or lead corrections. Method C can give results which are similar to the empirical method of AGMA 2001 for heavily loaded narrow face width pinions. Lightly loaded wider faced pinions can be heavily penalized. Method D is a further simplification of method C1 for symmetrically located pinions.

This concludes our exploration of ISO 6336-1. It contains the most important influence factors, which are also the most difficult to evaluate and which have the most potential to be different from AGMA or other calculation methods.

The calculation methods for surface contact (pitting) stress and root bending stress in ISO 6336 are very similar to AGMA 2001. The nominal stresses, which depend only on load and geometry, have similar values. The combined effects of the general influence factors from ISO 6336-1 modify the nominal stresses to "calculated" values which have essentially the same definition as AGMA's "stress numbers." Most of the differences between gear ratings by ISO 6336 and AGMA 2001 are explained by the general influence factors of ISO 6336-1. These general influence factors offer the greatest opportunity for further research and improvement of the standard. Users of the standard will need to exercise good judgment in picking the calculation methods to be used and the values of input variables in order to be satisfied with the calculated results.

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Industry In Motion

SCHAFER GEAR ACHIEVES QS 9000/ISO 9002 STATUS

Schafer Gear Works, a manufacturer of gears and related components for industrial, automotive, agricultural, aircraft avionics and consumer products companies, has met the requirements of the International Standards Organization for its South Bend, IN plant and has been awarded QS 9000/ISO 9002 registration, according to Bipin N. Doshi, Schafer president.

The qualification process, according to Schafer quality assurance manager, Robert Worrell, was undertaken on the company's own initiative. Although not a Tier 1 supplier providing products directly to the end-product manufacturer, but rather through other sub-suppliers, Schafer management initiated the process to improve its own internal performance and to offer its customers the benefits of Tier 1-level quality and service.

IMTS 98 YOUTH SUMMIT PLANNED

It is common knowledge that the machine tool industry faces a shortage of skilled workers—the workforce is getting older and there are not enough young people entering manufacturing to replace them. To meet the need of introducing students to careers in manufacturing technology, a diverse group of industry leaders has planned the IMTS 98 Youth Summit.

The primary aim is to utilize IMTS as a means of attracting students to careers in manufacturing technology. At the show, students will learn that the number of job openings within the industry is high and that begining precision machinists and toolmakers command impressive salaries. They will also see that grimy, dimly-lit, rust-belt workplaces are history, replaced by high-tech, computer-driven "clean room" environments and that manufacturing desperately needs an infusion of skilled and trainable workers.

Members of the group putting on the IMTS 98 Youth Summit include the American Machine Tool Distributors Association, The Association for Manufacturing Technology, Focus:HOPE, Gardner Publications, Inc., National Coalition for Advanced Manufacturing, National Institute for Metalworking Skills, National Tooling and Machining Association, Precision Metalforming Association, Society of Manufacturing Engineers, Tooling and Manufacturing Association and Vocational Industries of America.

IMTS 98, September 9–16, 1998, at Chicago's McCormick Place, will be the largest International Manufacturing Technology Show in the 71-year history of the event. Over 1,400 exhibitors will showcase 50 million pounds of advanced technology over 1.3 million square feet of exhibit space.

CARBIDE MANUFACTURING CELL HELPS PFAUTER-MAAG CUTTING TOOLS MEET INCREASED DEMAND FOR SOLID CARBIDE HOBS

Faced with unprecedented industry demand for its solid carbide hobs, Pfauter-Maag Cutting Tools has established the industry's first carbide manufacturing cell at its ultra-modern manufacturing facility in Rockford, Illinois.

This dedicated resource, consisting of the most advanced CNC grinding machines available, allows Pfauter-Maag to meet both increased demand and address the special handling and quality issues carbide tool production involves.

TIMKEN STEEL WINS CHRYSLER SUPPLIER AWARD TWO YEARS RUNNING

For the second year in a row, the Timken Company Steel Business has been awarded the Chrysler Corporation's Gold Pentastar Award for exceptional quality, delivery and warranty performance.

"The Timken Company has a reputation for manufacturing quality products," said Dan A. DeMare, senior buyer, Chrysler Corp. "But this award also recognizes its exceptional service and delivery to Chrysler. Once again, Timken's performance—as a supplier and a partner—is deserving of our company's special recognition."

Timken supplies Chrysler with seamless steel tubing used to manufacture gears and races for automatic transmissions.

BALZERS TOOL COATING APPOINTS PETER BJORKMAN PRESIDENT

Balzers Tool Coating, Inc., a supplier of thin film wear protection coatings to manufacturers and users in the metalworking industry, has appointed Peter Bjorkman to the position of president, replacing Roger D. Bollier who left to become president of N.C. Industries. Previously, Bjorkman was the president of Balzers Sandvik, a joint venture company in Stockholm, Sweden, serving the wear protection coating market in Scandinavia.

Peter Bjorkman

AJAX MAGNETHERMIC APPOINTS NEW DIRECTOR OF MARKETING

Frank C. Wilson has been appointed Director of Marketing for Ajax Magnethermic Corporation. Wilson has been involved in the induction heating business for over thirty years. His experience includes all areas of sales and marketing, as well as technical application and machine design concept expertise.

In his new post, Wilson will be responsible for the four U.S. service centers that provide coil repairs and rebuilds, replacement coils and other maintenance and repair parts and services. Wilson will also provide other marketing services in support of Ajax's sales efforts.

LINDBERG EXPANDS OPERATIONS—RECEIVES QUALITY RECOGNITION

Lindberg Corp. (Nasdaq: LIND), the largest commercial heat treater in the United States, has announced the establishment of its first heat treating facility in Monterrey, Mexico. The site is expected to be fully operational during the second quarter of 1998. "We will process materials for a variety of metal-working customers that were identified in a recent market study," said Leo G. Thompson, president and CEO. "This will establish a base of operations that could expand over time to meet other market needs. We are excited about the opportunities we have discovered in Mexico, particularly in the growing metal-working markets of northern Mexico."

Lindberg also announced the acquisition of Houston Heat Treating Company, which primarily serves the oil and gas industries. Privately held Houston Heat Treating reported sales of about $6 million in 1997. Terms of the cash deal were not disclosed. "This will be our second division in the Houston market," said Thompson. "Our other recent acquisitions have focused on the aerospace industry. In keeping with our expansion strategy, we are continuing to search for profitable heat-treat businesses serving a variety of industries."

In an unrelated matter, Lindberg Corporation's Industrial Steel Treating Division, located in Huntington Park, CA, has been awarded a letter of compliance to the AS-9000 aerospace quality standard. The division had previously received and continues to maintain ISO-9000 and NADCAP (North American Defense Contractors Accreditation Program) accreditation. Lindberg's Vac-Hyd Division, located in Rancho Dominguez, CA, was the first commercial heat treating operation in North America to receive an AS-9000 letter of compliance in late 1997. The Industrial Steel Treating Division provides heat treating and brazing services for customers primarily in the aerospace industry.

NADCA ASSUMES MANAGEMENT OF THE DIECASTING DEVELOPMENT COUNCIL

The North American Die Casting Association will assume management of the Diecasting Development Council (DDC), the marketing affiliate of NADCA. The new Executive Director of the DDC will be Leo J. Baran and, under the new management, the DDC will relocate to 9701 W. Higgins Rd., Suite 855, Rosemont, IL 60018-4721. The new phone number will be 847-292-3625 and the fax will be 847-292-3613.

The DDC's mission stresses the need to expand the North American marketplace for die cast components of all major alloys. It provides design, specification and sourcing assistance to OEMs as well as literature, manuals, video production, regional and on-site seminars and marketing and advertising to help generate qualified sales leads for DDC member companies.

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The Basics of Gear Metrology and Terminology Part I

Edward Lawson,
Senior Applications Engineer, Mahr Corporation

It is very common for those working in the gear manufacturing industry to have only a limited understanding of the fundamental principals of involute helicoid gear metrology, the tendency being to leave the topic to specialists in the gear lab. It is well known that quiet, reliable gears can only be made using the information gleaned from proper gear metrology.

Part I: Gear Inspection

Gears are one of the most common devices within the world of engineering, offering an elegant solution to the problem of effective power transmission. Modern gear drive designs must provide quiet, reliable service at high power densities, which can only be achieved by using gears which accurately embody a geometry like the involute helicoid system. Gear metrology may be divided into two subtopics, functional gaging and analytical testing. These two categories of gear inspection provide fundamentally different types of information, each with its advantages and disadvantages. They can each be further divided into single flank and double flank testing procedures. It is important to understand the capabilities and limitations of these categories because misconceptions about the proper meaning and usage of the information they provide are very common.

The functional gaging type of gear inspection can be characterized as an "attribute inspection," meaning that it determines if a given production piece will function as intended in the product. It does not determine whether the various elemental specifications affecting functional performance are in tolerance or control since such elements often combine in either a cumulative or compensatory fashion. Functional gaging is, therefore, more qualitative than quantitative. More sophisticated versions of gear functional gaging instruments can provide an assortment of numerical test data. However, since most of this information is based upon a fundamentally composite observation, it is usually best applied to process performance rating exercises rather than to control of process variables as these relate more directly to elemental test parameter data.

Functional gaging observations can be based upon either single flank or double flank meshing configurations of the master and production gears. The single flank version provides a direct observation of transmission errors, while the double flank version provides observation of variation of center distance.

The analytical testing type of gear inspection would be characterized as a "variable inspection," meaning that it provides numerical information pertaining to given elemental parameter specifications of a production piece. This type of test data often serves as the basis for accept/reject decisions. However, since analytical testing is unavoidably based on sample type data (the number of teeth tested, number of test traces per tooth), it could fail to detect anomalous errors such as nicks or hard spots. Composite action testing, which includes observation of all surfaces of all teeth, would be a more reliable method for detecting such errors which, though not systematic, could adversely affect product performance.

Analytical testing is generally quantitative rather than qualitative and is usually the most valuable source of process control information since process variables usually relate more directly to elemental parameters. Like functional testing, analytical testing observations can be based upon either single flank or double flank inspection practices. AGMA tolerances are provided for involute profile, tooth alignment (formerly called lead), pitch and pitchline runout parameters. All are single flank parameters except pitchline runout.

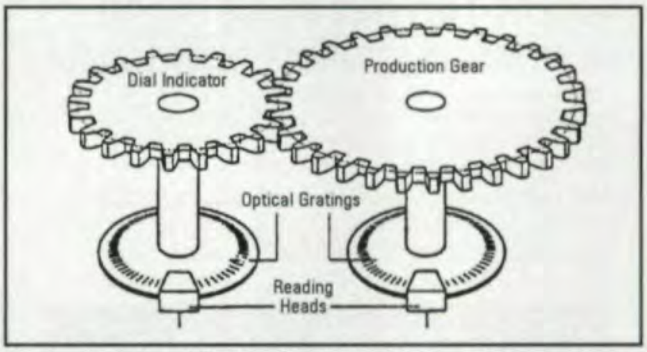

Fig. 1 — Functional Gaging: Single Flank Composite. © ANSI/AGMA 2000-A88.

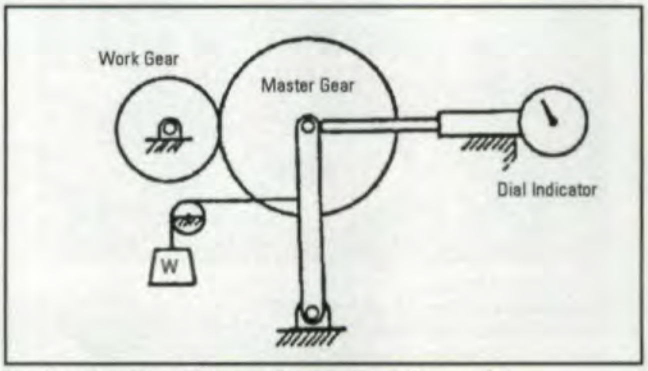

Fig. 2 — Functional Gaging: Double Flank Composite. © ANSI/AGMA 2000-A88.

Single Flank vs. Double Flank

Single flank testing provides observations (analytical or functional) of gear geometric quality involving only one flank at a time. The data provided is tangential rather than radial in direction, thereby offering information about the way the gear operates—an advantage over double flank testing operations. A single flank composite testing instrument (see Fig. 1) provides two spindles, to carry the master and production gears, mounted in fixed locations on the instrument to simulate the mounting of the gears at their proper center distance with backlash. Each spindle is fitted with a high-precision angular encoder as well as a means to apply a braking load to one of the gears as they are rotated through mesh, thereby maintaining contact on the loaded flank. The gears are placed on the spindles, brought into single flank contact with backlash and rotated through at least one revolution of the production gear. During that rotation, variation in the relative rotational velocities of the gears is observed. This procedure is based upon the assumption that two perfect gears would produce zero variation in rotational velocity, or no transmission error.

Double flank testing provides observations (analytical or functional) of gear geometric quality involving both flanks simultaneously. It provides radial rather than tangential data, information related only indirectly to the way the gear operates. The double flank composite testing instrument (Fig. 2) provides two spindles to carry the master and production gears. One of the spindles is mounted in a fixed location on the instrument and the other is mounted on a linear slide which is arranged to permit the center distance between the two gears to vary. A means is also provided to apply a load to the slide mechanism which will serve to maintain zero backlash between the gears. In operation, the gears are mounted on the spindles, brought into zero backlash mesh and rotated through at least one revolution of the production gear. During that rotation, variation in center distance between the gears is observed. This procedure assumes that two perfect gears thus tested would produce zero variation in center distance.

Double flank composite action test data can reveal radial eccentricity or out-of-round errors that can produce gear transmission error. It cannot, however, reveal angular tooth position errors which also produce transmission errors. Certain manufacturing processes (i.e. shaving) often produce gears with significant angular errors that cannot be detected by double flank testing. It is also not possible with this testing method to directly relate large tooth-to-tooth errors to gear function, including noise problems. It can, however, find non-systematic errors such as nicks, burrs or hard spots and it does offer an ideal means for evaluating functional tooth thickness based upon observations of the average center distance during testing with a calibrated master gear. Occasionally, one of the spindles is fitted with a gimble mounting to permit tilting in response to line of contact errors in the production gear. For spur gears this observation relates very well with tooth alignment errors. However, for helical gears the observation is equally and inseparably affected by both lead and profile errors.

Errors Detected by Composite Action Testing

If the error observed during either a single flank composite action test, or a double flank composite action test is plotted, the resulting trace will typically consist of long term and short term error components.

The long term component is composed of two categories of error, the most common occuring in a sinusoidal pattern once per revolution of the production gear and relating to the eccentricity of its pitchline. The second category relates to errors of the gear's shape or roundness. For example, a thin-walled ring gear which has been held in a three-jaw chuck with excessive force could display a long term error of three cycles per revolution.

The short term component is normally observed at a frequency of one cycle per mesh cycle. In a single flank test, this type of error relates to errors of tooth geometry and is directly related to noise problems, which may only be inferred from double flank tests. Frequency spectrum analysis of single flank test data usually correlates very well with the noise patterns generated by problem gears. Also, because short term errors occuring in regions of substantial slope on the long term component are affected accordingly, some standards permit the removal of the long term component from the test data before observations of the short term component proceed.

Occasionally, another pattern of short term error is observed in the single flank composite action test which does not correlate with the meshing cycle frequency. Commonly referred to as ghost harmonics, these patterns typically relate to kinematic errors in the associated machining operations.

Testing Machines

The classic method of testing an involute is to employ a base circle disk made to the same diameter as the base circle of the gear to be tested. That disk is mounted on a spindle which can also carry the gear. The device must also provide a linear slide arranged so as to operate in a direction tangential to the base circle disk. The slide carries a straight edge which is held in firm contact with the disk. A sensitive measurement probe is also carried by the slide. The probe is placed so that it will contact one of the gear teeth within the plane of action.

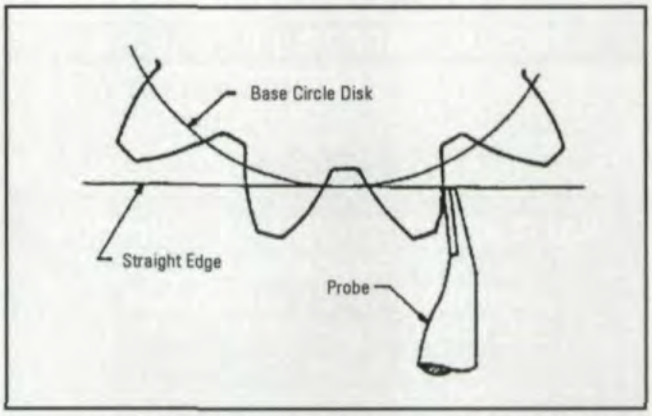

Fig. 3 — Involute Profile Testing Probe. © ANSI/AGMA 2000-A88.

The device is moved through a course of motion which will cause the probe to traverse the gear profile from root to tip following an involute path (Fig. 3). Because of the arrangement of the inspection device, rotating the gear with the disk and carrying the probe with the straight edge, this is automatic. As the disk and straight edge roll past one another, the straight edge and probe travel a linear distance equal to the circumferential distance upon the base circle disk associated with the angle through which the disk and gear have been rotated. During this motion, the probe is carried along within the plane of action by the straight edge. If the cam (the involute gear tooth) is an accurate involute, the sensitive probe will measure no error during the motion.

There are several different kinds of test machines. The involute test instrument uses the method described above. The straight edge mounted on the slide rolls tangentially with the disk mounted on the spindle. The probe is carried within the plane of action while contacting the gear tooth that is carried along with the disk. A related device uses a master involute cam on the spindle instead of the base circle disk. This cam drives a follower on the slide which carries the probe. The gear tooth's involute profile can also be inspected using a coordinate measurement machine (CMM). This method considers the involute helicoid surface in rectilinear coordinates, a considerably more complex procedure than the classic generative methods described above and not very common.

The most common category of involute test instrument today is the CNC tester. These devices employ a rotary axis and linear slides that are not kinematically connected to one another. Instead, each axis is fitted with a high resolution scale so that its movements can be controlled by a CNC module. Typically, an axis radial to the rotary axis positions the measurement probe to contact the involute tooth flank within the plane of action. The rotary axis and tangential linear slide are then commanded to move at constant velocities such that the linear distance travelled by the slide is equal to the circumferential distance upon the theoretical base circle of the gear associated with the angle through which the rotary axis travels.

Another type of CNC tester uses the computer controlled axes in a fundamentally different way to inspect the involute. Such instruments move the measurement probe in a radial direction only, while the rotary axis is moved in a nonlinear relationship according to the given involute. This practice lowers the cost of the instruments due to the lack of the tangential measurement axis normally used to generate the involute according to the constant rise cam principal.

Part II: The Involute Profile

An ideal gear would provide both the smooth running properties of friction disks and the positive power transmitting qualities of teeth. This can be accomplished by using teeth with a geometry which conforms to the law of gearing: "In order for two gears to transmit uniform rotary motion, the common normal of the mating profiles must pass through the same point on their line of centers at every point of contact." Such gears exhibit conjugate action, which is to


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say they exhibit progressive contact between mating teeth while their pitch circles rotate together at uniform, proportionate velocities.

There are limitless geometries which will satisfy the law of gearing, but no geometry accomplishes this task with the elegant simplicity of the involute helicoid system. Since it meshes with a straight sided theoretical rack, it can be accurately manufactured with relatively simple, straight-sided cutting tools. The involute helicoid geometry also has the property of operating on varied center distances without affecting its ability to provide uniform transmission of rotational motion at the same ratio. All other geometries are conjugate only at their design center distance.

Involute Geometry

An involute can be defined as the locus of a point on a line rolling on its base circle or, in three dimensions, a base cylinder. It can be imagined like this: envision a tin can as the base cylinder. Fasten a string to some point on the can and wrap it part way around while holding it taught. The string now represents a line or plane tangent to the base cylinder. This is the line (or plane) of action. Focus on a point where the string lies wrapped next to the can and begin to slowly raise the string away, keeping it taught all the time (Fig. 4). As it rises, the path followed by the point on the string as it moves up and away from the can will be an involute. At first, the point will move nearly straight up from the surface of the can. It will then quickly begin to follow a curved path similar to an Archimedes spiral. That path is the involute curve.

Two things are important to note here. The first is that this only occurs within the plane of rotation, perpendicular to the axis of the base cylinder. It is also important to note that, at any point you wish to consider along the involute curve, the string (line/plane of action) is perpendicular to the involute. Further, the curvature radius of the involute is always equal to the length of the string (line/plane of action) from the involute to the point of tangency of the string with the can (base circle/cylinder).

Consider now the same base circle (Fig. 5) with a single line of action tangent to it. In this case, the base circle can rotate about its axis when the line of action is pulled to the left. Two equally spaced points along the line of action have generated two parallel involutes as the line of action was pulled left. The distance between the involutes along the line of action is equal

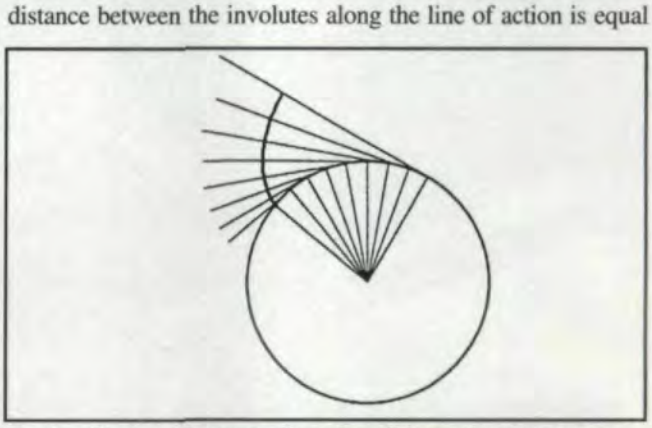

Fig. 4 — Circle with tangent lines describing an involute curve. Courtesy of AGMA.

to the distance along the circumference of the base circle. This is true because the same points on the line of action (string) that have generated these two involutes were once resting upon the surface of the base circle (can) before the involutes were generated by pulling away the string.

The Involute Cam

This observation gives rise to the most important property of the involute, that it will serve as a constant rise cam to a follower that is constrained to move within the plane tangent to the base cylinder of that involute. Imagine that such a follower has been positioned within the plane of action and in contact with the involute curve to the right. Now consider what will occur if the base circle is rotated counterclockwise through the angle required to move the involute at the right to the position of the involute at the left. The follower will be driven to a position within the plane of action contacting the left involute. It has been driven a distance along this line *equal to the length of the base circle circumference swept* by the counterclockwise rotation of the base circle. Any such rotation of the base circle and its involute "cams" will produce such a constant displacement of such a follower.

Figure 6 shows two base circles with a single line of action tangent to both. Involute profiles from both base circles interact with the string as we have seen before but now, with the two circles arranged to share this single line of contact, we can also see the two associated sets of involutes interacting. It can be seen that these involute profiles only contact one another within the plane of action. This condition also exists for all involute helicoid gear sets, where the mating tooth flanks only contact one another within the plane of action which is tangent to the base cylinders of both gears.

Observe the interaction of these mating constant rise involute cams. Begin with the point of contact labeled 2 near the base circle of the upper gear (driver) and at the OD of the lower gear (driven) in Figure 6. Now, rotate the driver counterclockwise through an angle necessary to move its involute to location 3. As this move proceeds from location 2 to 3, the point of contact with the mating involute is driven along the line of action, carrying the driven profile also to location 3. The distance travelled along the line of contact is equal to the circumferential distances along both base circles swept by their rotations which will be in proportion to their diameters.

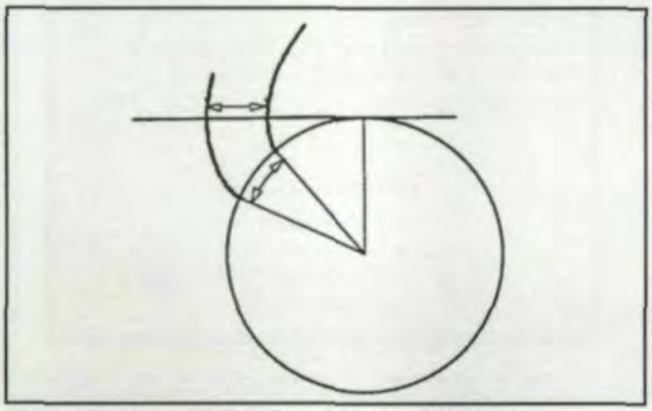

Fig. 5 — The Constant Rise Cam. Courtesy of AGMA.

Therefore, any rotation of the driver will cause an exactly proportionate rotation of the driven gear according to their diameter ratios.

Profile errors — Symptoms

Involute profile errors can result in gear noise, strength problems associated with dynamic loads promoting fatigue and durability problems associated with localized contact stress. Gear noise invariably relates to transmission error or inconsistent rotational velocities caused by geometry errors. Profile errors have a particularly troublesome effect upon transmission error because they tend to be consistent in the teeth of the gear. This causes transmission errors at mesh frequency which is usually in the range of human audio acuity. Also, since the error is typically consistent for all teeth, the associated mesh frequency transmission error is usually of consistent amplitude throughout the rotation of the gear. This is perceived by the ear as a pure tone which is much more objectionable than a noise source of equal average amplitude that exhibits a modulated frequency of amplitude.

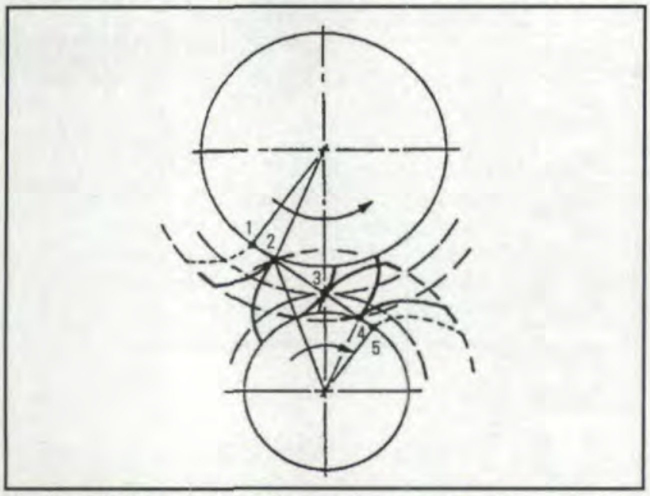

Fig. 7 — The Constant Rise Cam produces constant transmission of rotational motion. Courtesy of AGMA.

Involute profile errors can also adversely affect the strength and durability of a gear. Tooth strength ratings are calculated assuming that the torque load will be applied consistently and that loads applied at the critical region near the tooth tip will be shared by adjacent teeth entering mesh. Profile errors can increase dynamic loading and tip loading conditions thereby promoting fatigue and premature failure. Localized tooth contact stresses are also increased by profile errors that accelerate pitting and similar durability failures.

Profile errors—Causes

It is clear the primary contributors to involute profile errors are cutting tool accuracy and mounting errors. Cutting tool geometry errors typically transfer consistently to the gear profile on a one-to-one basis. Since other influences can also affect the gear profile, it is important that the cutting tools are significantly more accurate than the gears they are expected

to produce. Mounting errors can be even more problematic. A perfect cutting tool mounted inaccurately will perform no better than a low grade, inaccurate cutting tool. The use of bent or dirty tool arbors and failure to check truing diameters are possibly the most common and costly sins occurring in gear cutting operations. Errors in cutting tool accuracy or mounting tend to produce involute errors that are consistent when one observes teeth located at various positions around the gear. Errors in gear blank accuracy or mounting tend to produce involute errors that vary when one observes teeth located at various positions around the gear.

Gear blank accuracy and mounting errors are composed of two categories, eccentricity and out-of-round. Eccentricity conditions produce a sinusoidal pattern of variation in the slope trend of the involute test traces taken at various positions around the gear. Out-of-round conditions produce deformation of involute traces according to the given roundness error pattern. Radial runout of a gear caused by either an eccentric blank or an eccentric mounting of a good blank will cause a characteristic error pattern in which the profiles will display a slope error that varies in a sinusoidal pattern around the gear). This category of apparent profile error will not adversely affect the strength or durability of a gear or contribute to the generation of noise. Runout can be the source of several types of problems that affect gear performance. However, the sinusoidal pattern of profile slope errors it creates is not one of those problems.

It is possible to produce a gear with a substantial out-of-round condition that would exhibit proper conjugate action with a mate. Such gears are sometimes produced when a cyclic acceleration/deceleration is desired in a mechanism. However, when a gear that is intended to be round is deformed into an out-of-round condition during manufacturing, it cannot be expected to operate without an associated detrimental effect, as would be the case with simple eccentricity.

Recognizing the absence of detrimental effects from eccentricity-based apparent profile error, procedures have been employed that tolerance only averaged profile errors. This practice is inadequate because averaging may also remove the detrimental effects of out-of-round conditions. Rather, it is only correct to adjust test results according to the gear's simple eccentricity condition which has first been determined by analysis of its radial runout or by a more complex geometry-based analysis of the profile traces. Watch for the conclusion in the next issue of *Gear Technology*.

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Aug. 10–14. Heat Treatment of Steel. Materials Park, OH. A comprehensive course covering practical heat treating of steels, atmosphere control, quenching, temperature control and equipment types. For more information, e-mail: *mem-serv@po.asm-intl.org* or call 440-338-5151.

Sept. 9–16. IMTS 98. McCormick Place, Chicago, IL. Pavilions for Abrasive Machining/Sawing/Finishing, EDM, Environmental Safety & Plant Management, Factory Automation, Gear Generation, Lasers, Metal Cutting, Metal Forming & Fabrication, Quality Assurance and Tooling and Workholding Systems. The Society of Manufacturing Engineers will also hold its Manufacturing 98 conference. For IMTS information or registration, log on to *www.imts.org*. For SME conference information, log on to *www.sme.org*.

Sept. 21–25. AGMA Advanced Inspection Course. Richard J. Daley College, Chicago, IL. Advanced gear inspection and troubleshooting workshop. For more information, log on to *www.agma.org* or call 703-684-0211.

Oct. 12–15. ASM International Materials Solutions Conference and Exposition 1998. Rosemont Convention Center, Rosemont, IL. Where research and development meets industry for information exchange and networking. For more information contact ASM on-line at *www.asm-int.org/event98* or by phone at (440) 338-4634.

Oct. 20–24. Euro-BLECH 98. Hanover, Germany. The 15th year for this outstanding international industrial trade show with over 1000 exhibitors from across the world. For more information visit the Euro-BLECH Web site at *www.Euro-BLECH.de*.

Oct. 25–27. AGMA Fall Technical Meeting. Cincinnati, OH. The Fall Technical Meeting features seminars and papers on the design, analysis, manufacturing and application of gears, gear drives and related products, processes and procedures. For more information, call 703-684-0211, fax 703-684-0242 or log on to *www.agma.org*.

Oct. 26–29. IMET. McCormick Place, Chicago, IL. IMET is an umbrella event comprising M/TECH (Computer Technology for Design and Manufacturing), Motions Systems Technology Week '98 (PT Design Show, Hydraulics & Pneumatics Show, Motion Controls & Sensors Show), Industry Week's Best Practices from America's Best Plants Conference, Computer Aided Engineering's Getting the Most from 3D and Parametric Design, Industrial Equipment & Maintenance Expo '98, Electronic CAD Conference and Metal Components Expo. For more information, log on to *www.imetcongress.com*.

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LITERATURE MART

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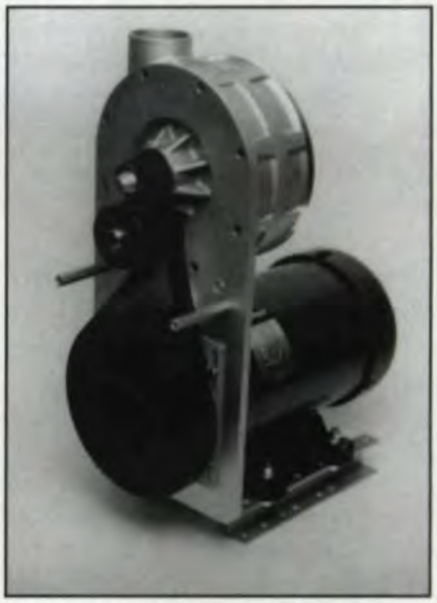

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Gear Technology's bimonthly aberration — gear trivia, humor, weirdness and oddments for the edification and amusement of our readers. Contributions are welcome.

Come with us now to those thrilling days of yesteryear...OK, this is not the Cisco Kid, but we do have a little game for you. Guess the year the following advertisements and excerpt were printed—they all appeared in a single issue of *Machinery* Magazine. Fax us at (847) 437-6618 or e-mail *people@geartechnology.com* with your answer and if you are the first to guess the year we'll print your name in the next Addendum and declare you the winner.

On Advertising: Seeing the Market Whole

"We have been looking over the mechanical world with the sales manager, noting that (1) the modern market for shop equipment is universal, and is rapidly increasing in volume and extent; (2) that customers are found in industries of almost every description; (3) that the buyers are not office men, but practical shop men who originate the orders and specify makes and brands as well as sizes wanted.

"Fortunate the sales manager who has the gift to visualize this boundless field, its unlimited possibilities, its manifold, increasing opportunities, its definite objectives. If he sees it all, his advertising plans will show it, his copy will demonstrate it. If he finds inspiration in so vast a market, there will be intelligence, method, conviction, force, and possibly power, in his advertising. His copy is read not only in America, but in Europe, Asia, Japan, China, India, Australia, New Zealand, Africa, South America. It is read wherever the wheels of industry revolve. It is read for business.

"American metal-working tools admittedly lead the world, and descriptions of them are eagerly read and studied by engineers everywhere. How anxious the sales manager should be to see that these intelligent, interested, responsible men are fully and accurately informed about the tools which it is his mission to market.

"The sales manager who sees with his mind's eye the engineering world of readers will endeavor earnestly to make every advertisement convey some definite data, idea or suggestion. To do less than this is to trifle with serious business and real opportunities. Ask any foreign dealer representing American tools for his opinion of the value of good copy in reaching the foreign engineer, interesting him, arousing his curiosity.

"The modern advertising method developed by Machinery, which shows the machine demonstrating high efficiency under actual shop conditions, makes a lasting impression on an engineer no matter where he is. He sees the tool doing the work and reads the authentic figures of production. He, too, wants results , and his competitors want them. Once he knows the best he is not likely to be satisfied with anything less. Advertising copy carefully planned to definitely and specifically inform the engineering reader, and persistently carried on month after month and year after year, builds a granite foundation under a business. In time this is called Good Will, and sometimes it is the most valuable asset that a business retains."

Thanks to Roland Ramberg of The Gear Works-Seattle, Inc. for letting us raid his library.

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

Did you guess? Did these machines and that bit of sage advertising wisdom appear in 1910, 1925, or somewhere in between? If you think you know, fax or e-mail us by September 30, 1998 and maybe your answer will appear in the next Addendum. Also, if you or someone you know owns a working machine from this period, let us know. If so, we may work up a story on it for The Gear Industry Home Page.

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