IMTS 92 Pre-Show Issue

GEAR TECHNOLOGY

THE JOURNAL OF GEAR MANUFACTURING

JULY/AUGUST 1992

GRINDING SPUR AND HELICAL GEARS

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CALENDAR

AGMA Technical Meetings
July 7-8 Flexible Couplings, Pittsburgh, PA; **July 9-10** Fine Pitch Gears, Alexandria Bay, NY; **July 22-23** Marine Gears, Milwaukee, WI; **Aug. 27-28** Helical Rating, San Francisco, CA Call (703) 684-0211, AGMA Headquarters, for more details.

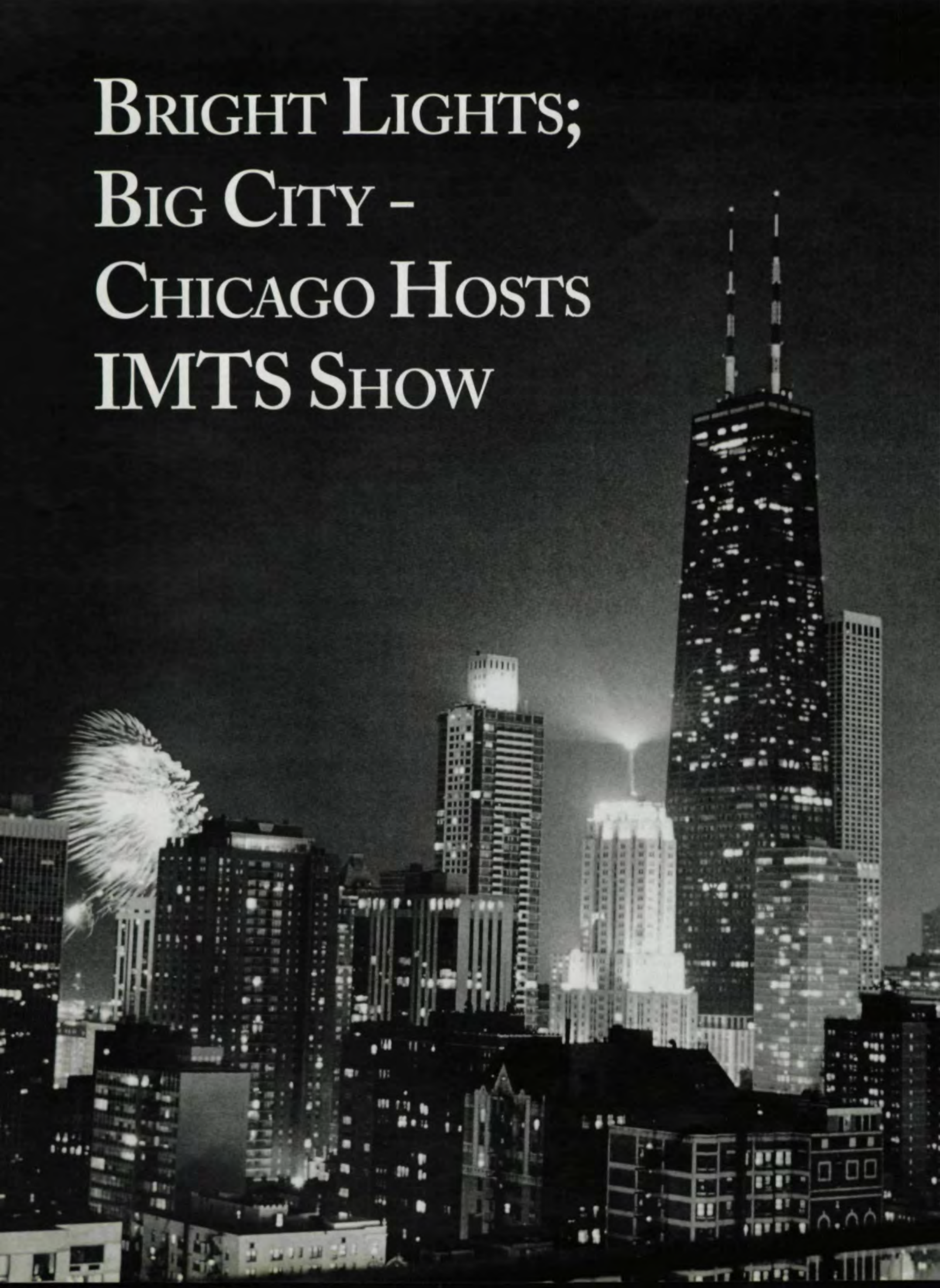
JULY 22-23
CIATEQ 1st Seminar for Gear Users and Manufacturers. Holiday Inn, Queretaro, Mexico. Technical meetings, plant tours, opportunities to contact Mexican gear manufacturers. Seminars run by Dudley Technical Group covering gear failure, rating practices, heat treating, lubrication, noise, etc. For more information, contact Rodrigo Lopez (52) (42) 16 38 08 at CIATEQ A.C.

SEPTEMBER 9-11
Gear Noise Symposium. Ohio State University, Columbus, OH. Contact Carol J. Bird, Conference Coordinator, (614) 292-3204 or Dr. Donald R. Houser at (614) 292-5860 for more information.

OCTOBER 20-22
SME Advanced Gear Processing & Manufacturing Clinic. Ritz Carlton Hotel, Dearborn, MI. Seminars on gearing subjects. Tabletop exhibits. Contact Mike Traicoff at SME for details: (313) 271-1500, ext. 596.

OCTOBER 26-28
AGMA Fall Technical Meeting. Baltimore, MD. For more information call (703) 684-0211.

Bright Lights; Big City - Chicago Hosts IMTS Show



New Product Area Pavilions, SME Conferences Highlight IMTS 92

By Donna-Marie Weir

IMTS 92 - The International Manufacturing Technology Show - opens in Chicago September 9 and runs through September 17 at Chicago's McCormick Place. IMTS is the Western Hemisphere's largest trade show. Over 800 companies from all over the globe will be showing products in exhibits covering some 931,000 sq. ft. of space.

The show, sponsored by The Association for Manufacturing Technology (AMT), turns McCormick Place into an international marketplace. AMT president Albert W. Moore describes the gathering as an opportunity for potential buyers to investigate, analyze, and compare $315 million worth of the newest machine tool technology from around the world. IMTS 90 broke an all-time attendance record set in 1980 with over 116,000 registrants.

One new pavilion and two new focus areas will be introduced this year at IMTS 92. The EDM Pavilion will feature CNC wire EDMs, RAM type die sinker EDMs, CNC wire EDM die sinkers, and manual die sinkers. Back by popular demand is the Forming & Fabricating Pavilion, which will offer demonstrations of high-speed presses, bending, drawing, and spinning machines, and press tooling, along with a showcase of other demonstrations.

Two focus areas will also be part of IMTS this year. The Environmental Safety & Plant Engineering Focus Area will feature equipment and products in the fields of safety, coolant purification, air quality controls, and allied interests. The Tools & Tooling Focus Area will feature cutting tools and tool holders, jigs and fixture tooling, and a host of other tools and tooling products.

This organizational system has been developed for added attendee convenience, allowing visitors to save time and energy by seeing all related equipment in one place. This will also give attendees more time to see the rest of the exhibits at IMTS 92.

IMTS 92 will be complemented by the International Manufacturing Technology Conference (IMTC). Cosponsored and organized by the Society of Manufacturing Engineers (SME), the conference features presentations by experts on future developments and applications in manufacturing technology. This year's conference will feature specialized technology tracks in six areas, including machining, forming and fabricating, automation for flexible manufacturing and CIM systems, robots/vision, environmental/waste management, and quality assurance.

Show hours for the West building will be 9-5 daily including Saturday, and 9-4 Sunday, Sept. 13, and Thursday, Sept. 17. East and North building hours will be 10-6 daily including Saturday, and 10-4 Sunday, Sept. 13, and Thursday, Sept. 17.

McCormick Place has contracted with a food and beverage service for the '92 show. Levy Restaurants, operators and owners of over 20 restaurants in the Chicago area, will be taking over all food operations at the facility. A host of casual restaurants plus one upscale dining facility will also be added.

Located on Chicago's lake front, McCormick Place is easily accessible by both automobile and public transportation. Direct public transportation and major highway access to the exhibition halls are also available from O'Hare International Airport. McCormick Place is also near major hotels, restaurants, shopping, and entertainment centers.

For more tourist information, contact the Chicago Tourist Bureau at 312-567-8500. For more information on IMTS '92 call 1-800-322-IMTS. ■

SEE OUR COMPLETE IMTS EXHIBITORS INDEX ON PAGE 50.

A FLOOD OF LESSONS

About the time we were in the midst of planning the editorial content for this issue of *Gear Technology,* we, like everyone else in the metro area, found ourselves diverted by the Great Chicago Flood. For a week, it seemed to be all we thought about. Then the tunnels dried out, the stores reopened, and we all went back to work.

But some of the lessons from this crash course in urban engineering have remained. And, as I've looked over the line-up for this issue, it occurs to me that some of the lessons learned from this soggy disaster can be applied a lot of places - including the American gear industry. And, coincidentally, some of our coverage for this issue reflects these issues.

"Infrastructure" was a word we all learned during Flood Week. Suddenly, live at five, was a vivid demonstration of the importance of all the non-glamorous basics that make a great city run. The lesson was clear: Neglect the basics, and sooner or later, you will have to pay the price.

Some in the American gear industry have been neglecting their infrastructure the same way cities all over the country have been, and it's starting to show. In David Goodfellow's "Viewpoint" in this issue, he points out that the U.S. gear industry has the oldest inventory of machine tools in the industrial world.

It's IMTS time in Chicago again, and some of the most up-to-date, efficient machinery in the world will be on display. This is a great opportunity to go looking at new equipment. And if you haven't thought about upgrading, it's certainly at least time to do that much. The exhibitors at IMTS are anxious for your business; they need the sales today, and they need healthy customers tomorrow. Competitive machinery is essential to our success, and we can't "wait a while" to become competitive.

Some would say that the American gear industry is already a leaking tunnel; and we've all seen what happens when the necessary repairs are put off too long.

Another lesson learned from the Great Chicago Flood was about the importance of planning. One of the phenomena of that strange week in this city's history that didn't attract a lot of attention around the country was the nearly flawless evacuation of close to a million people from the Loop area in a matter of six hours.

How did that brilliant bit of logistics happen in the midst of what seemed to be a comedy of errors? Because some unnamed souls in the appropriate departments planned ahead. They asked themselves, what would we do if...? and then drew up a plan accordingly. They thought about the details and were prepared.

PUBLISHER'S PAGE

Our "Management Matters" column in this issue and the next deal with planning a successful trade show, which, at least in terms of the details involved, is not unlike planning the evacuation of a major city. There's a lot to do, a lot that can go wrong, a lot to think of, but if you succeed, you're a hero.

Sure planning is hard work. Sometimes it's a bit boring. It involves the little details that can make you crazy. But it's the kind of careful, plodding, non-glamorous work that makes any project or any business succeed.

The gear industry is no different from any other: to succeed we need to think about the future, plan for it, invest in it. Otherwise we'll find ourselves up to our collective basements in far more than dirty water.

Michael Goldstein
Michael Goldstein
Publisher/Editor-in-Chief

Operation Trade Show

Planning ahead and keeping an eye on the details are keys to a successful trade show exhibit.

Nancy Bartels

Organizing a successful trade show exhibit is not unlike running Operation Desert Storm. The logistics can be a nightmare; the expense, horrendous; the details, mind-boggling. About the only thing you won't have to cope with is having someone fire SCUD missiles at you.

On the other hand, the sales leads you can generate at such an event can make it well worth the effort. According to the National Association of Exposition Managers, when you've added up all the numbers, the cost per face-to-face contact at a trade show is nearly 50% <u>less</u> than the cost of the average industrial sales call.

So where to begin? How do you make best use of the trade show option in your marketing plan? *Gear Technology* spoke with two trade show veterans, John Lawrence of Design Origins, Inc. of Madison Heights, MI, and Pam Felgenhauer of Exhibit Installation Specialists, Inc. with corporate offices in Philadelphia. They have shared with us some trade show planning basics.

Planning, Planning, Planning

The first key to success is *advanced* planning. Going to a trade show is not a decision to be made on a whim, nor is it a plan that can be implemented in two weeks or a month. Successful trade show exhibitors often begin planning six months to a year prior to the show dates. At many shows, the first opportunity to reserve a booth comes at the close of the prior year's show (in the case of shows like IMTS or the AGMA Gear Expo, that's <u>two</u> years in advance), and many of the best booth locations are snapped up early.

Furthermore, for maximum effectiveness, your trade show effort should be a part of your overall marketing plans. It should not be viewed as something-we-can-do-if-we-get-around-to-it. What you want to show at your booth and how you show it should be part of an overall plan and should supplement your other advertising and marketing efforts.

Once you've decided that trade shows should be part of your strategy, the next question to ask is which trade shows you should attend. Research the shows at which you think you belong. John Lawrence suggests, "Get a prospectus from the show. It should have a count of the attendees and total audience and usually a breakdown of the industries and job functions. Evaluate if you belong there...Look at past exhibitors. If your competitors are there, it seems logical that you belong too." The next important decision to make is what you are going to show and what you want to tell the audience about it. Decide early what story you want to tell and how you are going to tell it.

The question to ask is, what will draw people to my booth? There's no point in being at a show if no one stops to see you. Hardware attracts visitors to booths at engineering shows. If you have a new machine, bring it. If your budget or the story you want to tell doesn't lend itself to that,

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Nancy Bartels
is Gear Technology's Senior Editor. She is also a freelance writer and literature and writing instructor.

show instead a smaller part - perhaps just a cut-away example of your newest feature. Or bring in samples of manufactured parts to show your booth visitors. People like to have things they can see and touch and manipulate.

How Big A Booth?

The decision about what to bring has to be made early because it affects the next important question - the size of your booth. Obviously, if you're going to show a large machine, you're going to need more than a tabletop to do it.

But other factors should also go into your size decision. "You get better exposure with a 10' x 20' booth than you do with a 10' x 10," says Lawrence. "It's just like having a bigger ad in a magazine."

Show regulations have to be considered as well. When you register for a show, you will get an exhibitor's manual. READ IT CAREFULLY. COVER TO COVER. SEVERAL TIMES. It contains a wealth of information, including any restrictions that the show management or the location will place on your booth construction.

"The best thing you can do is read those rules and regulations and make sure when you design your booth, you have followed them," says Pam Felgenhauer. "Show management has restrictions about height and different kinds of booths and not blocking other booths, etc. I've seen people come to a show with a fifteen-foot tower and discover that the ceiling is only twelve feet. They've had to take a saw and cut off the top before the show started."

Costs will also have to be factored in. On custom construction of booths you should expect to pay around $1,000 per foot for the back wall and standard graphics. Product illustrations would be extra. A folding "suitcase" display will run between $2,500 and $3,500 for a 10' section. These costs are in addition to space rental, shipping, installation, and employee salaries.

If these numbers cause you to reach for your heart medicine, remember that you can do effective things with small booths or tabletops. The key to success with them is the old adage, "You get what you pay for." Lawrence puts is this way: "[With a table top...] good photography is important. A lot of people will say, 'I'll take my 35mm and go out in the shop and shoot the pictures,' and then they're disappointed with the outcome. Use good photography and quality sign work. Remember that you can use things like this for more than one show - and they do make a difference in your presentation."

Rolling Your Own - or Not

This brings us to the knotty question of when to bring in professional help. It is, of course, theoretically possible to design, build, ship, and run your booth all on your own, using only in-house talent. And it's tempting to think you'll save a lot doing it that way. In most cases it's also probably not a good idea - especially for the first-time exhibitor.

Booth design and graphics is as specialized a field as any of the engineering disciplines, and, in the end, you will probably be more satisfied with the results if you use experienced professionals. These are the people who can give you the most "bang" for your buck.

Remember, too, that appearances are more than three-fourths of the battle at a trade show. You want a booth that looks good, one that leaves the best impression of your company and its products. Sloppy, amateurish design and construction isn't going to do that.

Also keep in mind that booth design and construction are time-consuming. If someone on your staff is going to do it, that someone will have to take time away from his or her regular duties.

It may help to look at booth design and construction costs as a long-term investment. An experienced professional can design one for you that is flexible enough to meet your needs at a variety of shows and can be used for a number of years.

Moving Right Along

Once your booth is designed, there's still the question of getting it - and its contents - shipped to the show and set up on time - and then of taking it down and getting it back again.

If this seems like a monumental task, you can turn it over to an installation and dismantling (I&D) house. Says Jan Felgenhauer, "We have some customers that just let us handle the whole thing. They call us and say, 'Here's where the show is. Here's when we need to be there,' and they let us take it from there."

The advantages of letting the I&D house handle all or part of your project come down to two factors; efficiency and cost-effectiveness. Because the I&D house will be working with several exhibitors, it will usually have access to the pick of the labor crop and the pick of the schedule. Also, on the basis of experience alone, your booth will be assembled more quickly. Remember that all the construction help you need will be paid hourly, and you can reap great savings by avoiding situations where people are standing around waiting for material to arrive or taking extra time to assemble your booth.

But many companies do choose to handle their own shipping and setup. Here again, careful planning is absolutely essential. Choose a trucking company that has experience with trade shows. Companies that are familiar with trade show practices can save a lot of headaches.

The Union Label

One area of concern to

many first-time exhibitors is the question of dealing with union help in setting up your booth. Horror stories abound, and usually they are much worse than the reality.

Says Lawrence, "Attitude is so much different than it was 20 to 25 years ago. Unions are much more agreeable. They would like to do your work because they need employment. They won't fight you like they used to...In almost all cities now, especially with small booths, they are getting very lenient."

Still, there are areas of possible conflict. The first place to check to avoid it is that exhibitor's manual. It will list which unions must be involved in your setup and what their respective jurisdictions are.

Attitude is important here. Felgenhauer reports, "Some people come in and say, 'Well, you're union, but so what? You're going to do what I say.' That attitude is not going to cut it. The best advice is, if you're having problems with a particular union person, don't argue with them. Go back to the service desk or that person's supervisor and talk to them. Let them handle it. That's what they're paid for."

Working within the system and playing by the rules are the keys to smooth operation here.

Details, Details, Details

Whether you handle all your own setup or have an I&D house do it, you will need to assign one person from your company to be responsible for the trade show booth. Pick the most obsessive person on your staff. "You need someone who will dot every i and cross every t and then check five times after that," says Felgenhauer.

Your guide (or your designated person's) to the i's that need dotting is the show manual mentioned earlier. This book will become your "bible" for the duration.

The show manual is the source of much important information, such as setup schedules, building and zoning rules that will apply, regulations about deliveries, storage, pickup, and the vital telephone numbers of the people to call if a question comes up that the book does not answer.

Paying close attention to these dates, times, and rules can save you big money. "Missing a deadline on an installation can cost you big money," says Felgenhauer. "For example, if you get your electrical outlets in one day late, the cost will go up $100 per line for a 110 outlet at most shows. It could be $300 or $400 for a 480 outlet."

But your show manager should also be flexible. "I've never been to a show where, no matter how carefully you've planned, something doesn't go wrong at the last minute. But when you've studied the manual and know the options, you know how to respond," says Felgenhauer.

It's Show Time

Planning for trade show success doesn't stop with the minute the booth is set up. The wise exhibitor remembers to bring a few "survival supplies" to the booth as well. Some of these are common sense items, but they are easily forgotten in the rush of opening day.

Bring a small first-aid kit. Aspirin, vitamins, band-aids, and cough drops are the kinds of things that seem unimportant until you need one. And don't assume you won't need them.

The same is true for small office supplies. Be sure to pack extra pens, a stapler, ruler, scissors, paper clips, pads of paper, etc.

Paper towels, all-purpose cleaner, a small whisk broom and dust pan, and a waste basket aren't bad ideas either. These can prevent your booth from getting "tired" before the end of the show. A basic set of tools - a hammer, screwdriver, some duct tape, extra extension cords - can also stave off minor disaster in the middle of the show.

Also think about your own comfort. Comfortable shoes are a must. You're going to be on your feet for hours on end. (If the show is running more than a day or two, consider carpet padding. The cost is minor compared to the difference it will make to your comfort.) Bring along some mints, hard candy, or chewing gum to help tide you over until you can break for a meal.

Bring enough people. The number is a trade-off between overworking your people and the space in your booth. You probably should not have more than one or two people in a 10' x 10' booth at one time, according to John Lawrence. On the other hand, remember that these people need a break. "If you've got two or three people at a show, let two walk through the show and have one work the booth for an hour. Then trade off," he suggests.

What you want to avoid is having five or six people standing around in your booth doing nothing. That looks bad and may actually drive people away.

Follow Up

Once the show is over, your work is not done. Along with all the repacking and shipping of your booth and its contents, you should be taking notes on what went right and what went wrong at the show. Write down the things you want to do differently next time. Don't depend on your memory. You also should have a system in place for following up on all the contacts you made during the show. They're the reason you went to all this trouble in the first place. Failure to make good use of these contacts is to waste your entire trade show effort.

It's true. Creating and running a successful trade show exhibit is a little like organizing the Normandy Invasion. But when the contacts you make turn into good sales later, you will find that the effort has been worthwhile. ■

Grinding of Spur and Helical Gears

Dr. Suren B. Rao
National Science Foundation
Washington, D.C.

Grinding is a technique of finish-machining, utilizing an abrasive wheel. The rotating abrasive wheel, which is generally of special shape or form, when made to bear against a cylindrical-shaped workpiece, under a set of specific geometrical relationships, will produce a precision spur or helical gear. In most instances the workpiece will already have gear teeth cut on it by a primary process, such as hobbing or shaping. There are essentially two techniques for grinding gears: form and generation. The basic principles of these techniques, with their advantages and disadvantages, are presented in this section.

A general introduction to the basic principles of the grinding process, however, precedes the discussion of gear grinding techniques. This is based on the belief that the discussion of the grinding process, in combination with the description of the gear grinding techniques, would constitute a more complete treatise on gear grinding.

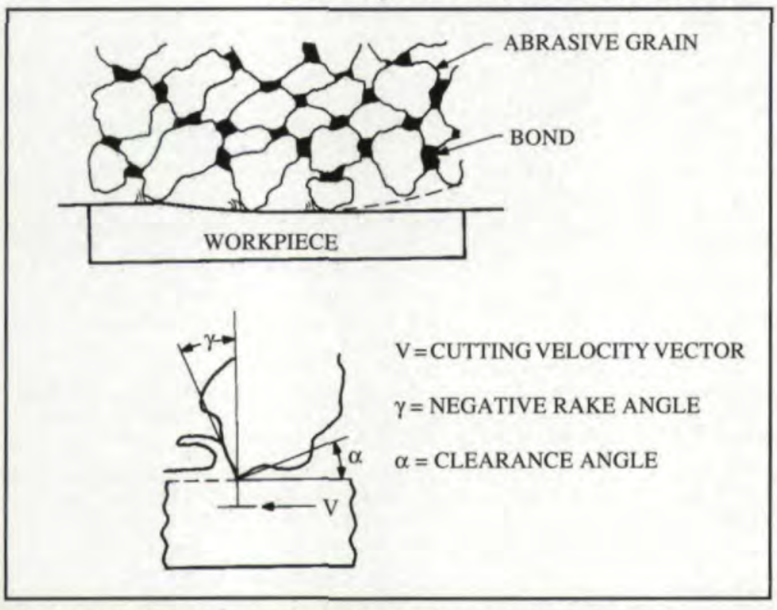

Fig. 1 - Grinding process schematic showing negative rake angle.

Reasons For Grinding

There are two primary reasons for grinding gears. In spite of several attempts to the contrary, it still remains one of the most viable techniques of machining gears once they are in a hardened state (50 Rc and above). Also, the process, in combination with highly accurate machines, is capable of gear manufacturing accuracy unmatched by other manufacturing techniques. AGMA gear quality 12 and 13 are common, and AGMA gear quality 14 and 15 are not unusual. With the advent of cubic boron nitride (CBN), grinding has been tried, with some success, as a primary operation on hardened material instead of hobbing or shaping before heat treatment (sometimes referred to as "direct grinding"). Obviously this is of greatest consequence only if a gear is being made from through-hardened material, a process that is not very common. It is much more common for gears to be made of case-hardened material where the economics of grinding from the solid are not as beneficial.

Gear grinding is an expensive operation and has to be justified on the basis of required gear quality in the hardened condition. The basic principles of grinding are now presented.

Grinding Process Mechanics and Process Parameters

Grinding is a metal cutting process not unlike single- or multi-point machining, such as turning, milling, hobbing, etc., but with some major dissimilarities. Grinding is characterized by the fact that the cutting tool, in this case the grinding wheel, consists of a very large number of randomly oriented cutting edges machining small amounts of material, thus resulting in extremely fine chip thicknesses. While chip thicknesses of 20 μm (0.0008") or more are common in operations like turning, and chip thicknesses of 8 μm (0.0003") are common in operations like milling, chip thicknesses of less than 1 μm (0.00004") are the norm in grinding.

Though the abrasive particles in the grinding wheel are randomly oriented, by virtue of their shape, they generally present a large negative rake angle to the cutting velocity vector as seen in Fig. 1. Negative rake angles always result in higher cutting forces than do positive rake angles. Also, small chip thicknesses result in higher specific cutting forces, where *specific cutting force* is defined as the force required to cut a unit area of chip cross section (kgf/mm^2 or lbf./in^2). The combination of negative rake and low chip thickness gives rise to high specific power requirements in grinding. Specific power is defined as power required to machine unit quantity of material in unit time (hp/mm^3/min or hp/in.^{3}/min). This is not only indicative of the low efficiency of the grinding process, but also its high susceptibility to burning damage, as all the power consumed by the operation is converted into heat. The small chip thickness, however, also enables the generation of a high-quality surface and tight dimensional tolerances that make this process critical to the manufacture of high-precision gears and components.

The combination of large negative rake angle on the abrasive grain and small chip thickness also results in cutting process stiffnesses in grinding that are almost several times the cutting process stiffnesses in other machining processes, such as turning and milling. *Cutting process stiffness* here is defined as the force per unit chip thickness and is generally expressed in kgf/mm^2 (lbf/in.^{2}) of chip thickness. Since the rate of reproducibility of error due to a machining process is given by the formula:

$$\delta = \frac{\mu}{1+\mu}$$

where δ is the rate of reproducibility and μ is expressed by the formula:

$$\mu = \frac{R}{K}$$

where K is the stiffness of the machine tool and R is the cutting process stiffness.

In a typical turning operation K is several times R, and δ typically computes to less than 0.25. This signifies that only 25% of the initial work piece error will remain after the first turning pass. In grinding, however, since R is extremely large compared with K, values of δ = 0.95 are not uncommon. This signifies that in grinding almost 95% of the initial work piece error may remain after the first grinding pass. It is obvious from this discussion that careful execution of all previous processes to ensure a good preground gear is essential to an economic and successful grinding operation.

Wear of the grinding wheel is an essential part of the process. As the sharp cutting edges of the abrasives wear out, cutting forces on that particular abrasive increase until either the grain fractures reveal new sharp cutting edges or the abrasive is pulled out of the bond and a new abrasive grain is exposed. In essence, if the process were in perfect harmony, the grinding wheel would be self-sharpening. But even though wheel wear is an accepted phenomenon, the rate at which it occurs is critical. For wheel wear results, not only in wheel replacement, but also in other nonproductive wheel preparatory operations, such as trueing, dressing, and profiling. Therefore, the ratio of work material removed to volume of wheel lost, also called the G ratio, is a measure of grinding efficiency. G ratios can range from less than one to several hundred, depending on these variables.

One other distinguishing feature of the grinding process is the fractional amount of time the abrasive grain is actually creating a chip, in comparison with the total time this grain is in contact with the work piece material. Three distinct phenomena have been recognized as occurring as the abrasive grain comes in contact with and leaves contact with the work surface. These three regions have been defined as rubbing, plowing, and cutting, and the actual cutting or chip formation may be occurring for only about 30% of the time the abrasive and the workpiece are in contact. The force at which the transition occurs from rubbing and plowing to cutting is called the *threshold force*. When the mechanisms holding the grinding wheel and the work exert a force in excess of this threshold force, grinding and metal removal will occur.

Finally, here is a word about spark-out. As a grinding process proceeds with the rotating grinding wheel being fed into the work material, cutting forces are generated. These cutting forces cause the electromechanical structure that holds

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the grinding wheel and the workpiece to deflect away from each other. At a certain instant in the process, the infeed of the wheel and/or the workpiece is stopped, resulting in a reduction of the cutting forces. This causes the strain energy stored in the structure to overcome the deflection and return the system to a state of equilibrium. As this happens, the wheel and workpiece move into each other and continue to grind as the forces decay to the threshold force level, after which no more grinding occurs. This part of the grinding process, where no infeed occurs but grinding continues, is called *spark-out*. The time taken to complete spark-out is a measure of the stiffness of the structure of the machine tool-tool-workpiece system. In general, some amount of spark-out in a grind cycle will improve work piece quality.

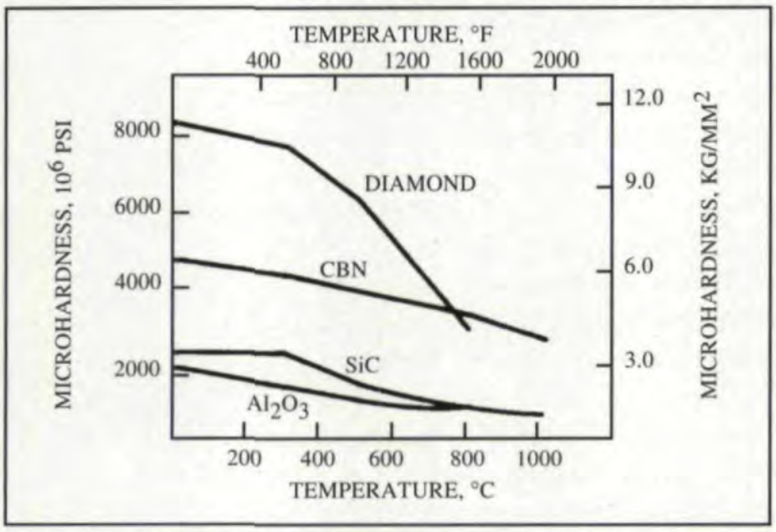

Fig. 2 - Comparative hardness of abrasives at elevated temperatures.

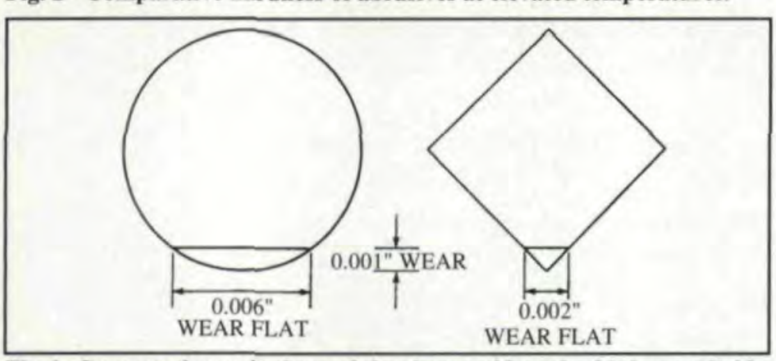

Fig. 3 - Comparative grain shape of aluminum oxide and cubic boron nitride.

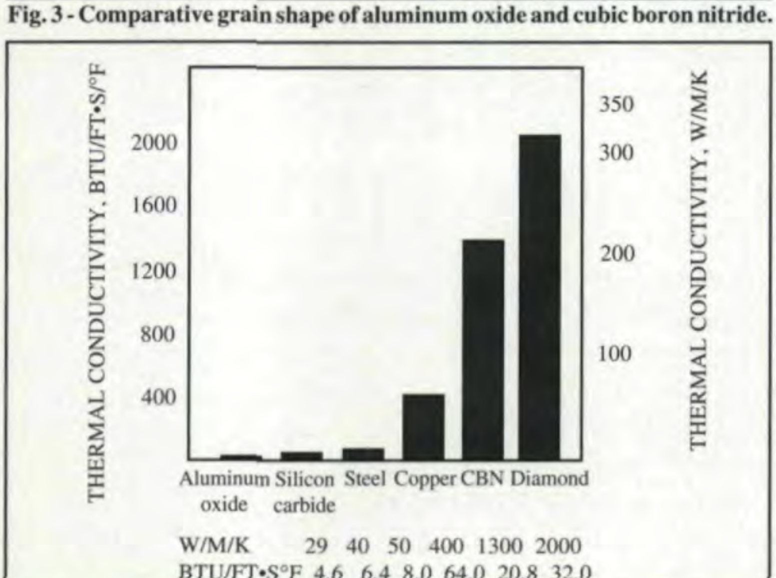

Fig. 4 - Comparative thermal conductivity of abrasives and other materials.

Abrasives

Though aluminum oxide (Al_2O_3), silicon carbide (SiC), diamond (C), and cubic boron nitride (CBN) are generally considered in the category of abrasives where grinding is concerned; only aluminum oxide and cubic boron nitride are discussed further. This is because in gear grinding we are usually dealing with ferrous alloys, and diamond and silicon carbide tend to perform poorly when grinding steel. High wear rates of the diamond or silicon carbide abrasive when grinding may be due to interatomic diffusion of the carbon atoms present in these two abrasives, since steel is characterized as "carbon hungry" at the elevated temperatures that are encountered during grinding.

The characteristics of aluminum oxide and cubic boron nitride that impact their performance as abrasives are now presented in a comparative manner. It is obvious from the following that cubic boron nitride is a considerably superior abrasive, though more expensive than aluminum oxide.

Hardness. Fig. 2 shows a comparative plot of diamond, cubic boron nitride, silicon carbide, and aluminum oxide hardness at elevated temperatures. It is obvious that cubic boron nitride is several times harder that aluminum oxide and even harder than diamond at temperatures higher than 1472°F (800°C). The chemical inertness of cubic boron nitride is also of significance, since any chemical affinity to iron would result in increased wear rates.

Grain Shape. When comparing grain shapes of aluminum oxide and cubic boron nitride, the former is known to have a more pronounced spherical form, while the later has a block form. For a given amount of crystal wear, a spherical form exhibits a larger wear area than does a block form (Fig. 3). Tendency to burning has been related to wear flat area, indicating that higher degrees of burning and surface damage are possible with aluminum oxide than with cubic boron nitride.

Thermal Conductivity. Fig. 4 shows a comparison of thermal conductivity of the various abrasives and some common metals. Though diamond has the highest thermal conductivity, cubic boron nitride is not far behind and considerably higher than aluminum oxide. The high thermal conductivity of cubic boron nitride allows more of the heat generated at the abrasive-work material interface to flow into the abrasive and into the wheel than into the work piece, resulting in reduced tendency for surface damage. It must be remembered, however, that it is possible to produce thermal damage with cubic boron nitride. The combination of high thermal conductivity and lower wear flat area owing to grain shape allows for much higher metal removal rates to be achieved and higher spindle powers to be utilized before thermal damage can occur. The impact of a grinding abrasive on the work piece will generally induce a compressive stress on the work surface. However, the localized heating and subsequent cooling that is more predominant when grinding with aluminum oxide overcomes the compressive stress due to mechanical impact, and the residual stresses in the uppermost layers of the work piece are highly tensile. The absence of this heating when grinding with cubic boron nitride results in residual stress that is a compressive on the work surface. Fig. 5 shows typical residual stress profiles produced by plunge grinding with the two abrasives. Since tensile stresses are accompanied by lowered fatigue life, clearly grinding with CBN offers distinctive advantages. The only drawbacks to the application of cubic boron nitride are its costs and the need for stiffer, higher-powered machine tools to fully utilize the advantages that cubic boron nitride has to offer.

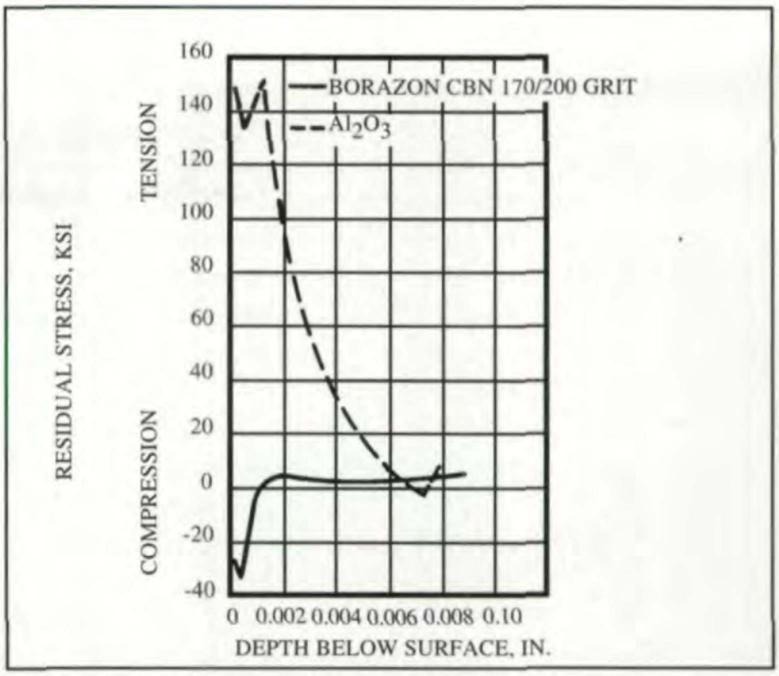

Fig. 5 - Comparative values of residual stress distribution.

Grinding Wheels

Grinding wheels and their properties are only briefly discussed here, as a considerable amount of literature is in existence, especially from wheel manufacturers, that covers this in detail. All grinding wheels, except electroplated wheels, consist of an abrasive held in a bond. The physical size of the abrasive is a major determining factor in abrasive grain concentration and in the number of cutting edges engaged in the process of grinding at any given instant in time; that is, the larger the abrasive size, the fewer the number of abrasives and cutting edges, and vice versa. This in turn impacts the chip thickness as grinding proceeds and, consequently, the surface finish that is obtainable. In general, coarse abrasive grain sizes, also called grit sizes, result in rougher ground surfaces and finer grit sizes in lower surface roughness values. Surface roughness values of 0.4 μm (16 μin.) are generally possible with 60 grit size abrasives, and values better than 0.1 μm (4 μin.) are possible with abrasives of 200 grit size or finer.

Wheel hardness is another important characteristic of the wheel and is related to the amount of bond used in the manufacture of the wheel. A hard wheel has more bond, resulting in a greater abrasive retention ability. The abrasive will have to become considerably dull, in regard to its cutting edges, before sufficient forces are generated to tear it away. On the other hand, a soft wheel has less bond and consequently will lose its abrasive grains more readily. In general, soft wheels are used with hardened materials because the abrasive grain is known to dull rapidly when machining the hard materials, and fresh, sharp abrasives will be required to continue grinding without the occurrence of high temperatures and surface damage. On the other hand, hard wheels are generally used with soft materials, since the abrasive is expected to last longer, and abrasive grain retention is a property that is desired from an economic point of view.

Wheel structure is another important characteristic. An open structure allows greater chip clearance and is preferable in roughing operations where large quantities of material may be removed. Lack of sufficient space for chips would result in the loading of the wheel with subsequent burning of the work surface. However, open-structure wheels are also softer, because of the reduced amount of bond material.

Abrasive grain size is specified by the wire mesh size that will allow the abrasive to pass through. The smaller the number, the larger is the grain size. It must be remembered that the mesh size specified only indicates that grains larger than the specified value do not exist in the wheel, but smaller abrasive grain sizes do. In general, the grain size distribution can be assumed to follow a normal distribution. Wheel hardness or wheel grade is specified with a letter, with A being the softest and Z the hardest. Wheel structure is generally specified with a number, with 1 representing a close structure and 10 representing a very open structure.

It must be remembered that there are no absolute relationships between work piece and wheel characteristics. The aforementioned facts are only guidelines, and the exact choice of a wheel for a particular work material-grinding operation combination has to be arrived at on the basis of trial and experience.

Wheel Preparation. In most grinding operations where a dressable wheel is used, four distinct operations may be present, singly or in combination: 1) wheel trueing, 2) wheel dressing, 3) wheel profiling, and 4) wheel crushing. The purpose and procedure for these four operations are now discussed. The mechanism for accomplishing these operations is described later.

Wheel Trueing. In this operation wheel material is removed to eliminate wheel nonuniformities of shape and geometry due to wheel manufacture and mounting. The grinding wheel is mounted on its wheel holder, balanced, and then mounted on the machine spindle. Trueing is then carried out by the motion of a diamond tool in a direction along the axis of wheel rotation as the wheel is spinning at speeds close to or at grinding speeds. After all the nonuniformity is eliminated, the grinding wheel will need to be balanced again. Wheel trueing is generally necessary only when the wheel is mounted for the first time unless nonuniform wheel wear has occurred during the grinding operation. Trueing will reduce forced vibration problems due to nonuniform wheel shape and geometry, resulting in improved surface finish.

Wheel Dressing. This is required to eliminate the uppermost layer of dulled abrasive grains and expose the sharp, next layer of abrasive grains in the wheel to obtain efficient cutting. On a new wheel it becomes necessary to do this when the wheel is very hard and the bond material completely encloses the abrasive grain. For softer wheels the trueing operation is generally able to expose abrasive grains, and the first wheel contact with the work piece is sufficient to break down any bond material that may still be covering the abrasive. On harder wheels the bond material may need to be pushed back with a stick of silicon carbide or naturally occurring abrasives, such as corundum, etc. Too much bond removal is, however, detrimental, as abrasive grains would be unsupported and consequently lost easily, leading to loss of the wheel.

Dressing also clears the chip-loaded surface of the wheel, which may cause burning of this work piece. A loaded wheel, in combination with dull abrasive grains, will have a smooth, glazed surface. After dressing, the wheel surface will be rougher to the touch.

Wheel Profiling. In this operation the wheel is shaped to a specific profile in order to generate the required geometry on the work piece. This is of special significance in gear grinding, as the wheel is either representing a rack in some generating-grinding operations or the normal space between two adjacent teeth in form-grinding operations. Wheel trueing, dressing, and profiling can, however, be combined into one operation on a machine, especially if a medium or soft wheel is used. For hard wheels trueing and profiling can be combined, while initial dressing to push back the bond material is carried out as a separate operation.

Wheel Crushing. This is a technique used for rapidly removing wheel material to profile a wheel. A crushing roll, generally made of high-speed steel, with the required profile machined on it, is brought into contact under pressure with the grinding wheel, with no relative tangential velocity. Wheel speeds are generally reduced during this process to about one-fifth to one-tenth the actual grinding speeds.

Except for crushing and dressing of very hard wheels, all other wheel preparation operations are combined on most grinding machines. Profiling, trueing, and dressing can be done with a single-point diamond traversing the wheel surface in a specific relationship to generate the required profile. Where profile accuracy is influenced by the wear of the diamond point, a rotating diamond disk that represents many diamond points will improve results, since the diamond wear is distributed over many points. However, the disk should run true in axial and radial directions in order to maintain profiling accuracy. For much faster profiling, in combination with dressing and trueing, formed diamond rolls can be used. These rolls are, however, expensive, and sufficient part volume may be necessary to justify the investment. With a formed diamond roll, intermittent dressing when the grinding wheel is out of the cut or continuous dressing during the grinding operation are possible. Continuous dressing is especially effective for high-speed, creep-feed grinding, which is discussed later.

Grinding Processes

There are two distinct grinding processes used in gear grinding, as in most grinding operations. They are as follows:

Conventional Grinding. Fig. 6 illustrates the basic properties of this process, which is characterized by a wheel rotating at surface speeds of about 30 m/s (6,000 ft/min), infeeds of 0.01 to 0.050 mm (0.0004 to 0.0002 in.) and work velocities of 1.25m/min to 10m/min (50 to 400 in./min). The chips generated are short, due to the small arc of contact between an abrasive grain and workpiece in this process, and are easily disposed of. Wheel wear rates are generally high in this process, resulting in low to medium G ratios. This is because of repeated impacts between the edge of the work piece and the wheel due to the to-and-fro oscillations of the workpiece, for which reason this process is also sometimes referred to as pendulum grinding. Researchers have also found that in this mode of grinding, the average force per abrasive grain is high, further contributing to rapid wheel breakdown. Coolant is generally used, though dry grinding can be done if the amount of infeed is in the 0.0005-mm (0.0002-in") range when grinding hardened steel. This is due to the fact that metal removal rates are very small, with a small fraction of the wheel surface cutting at any given instant, with low power consumption and consequently low amounts of heat generation.

Creep-feed Grinding. Fig. 7 illustrates the basic properties of this process that are characterized by large infeeds into the work in excess of 0.5 mm (0.0125") and up to 10 mm (0.4"), depending on machine power and stiffness; but accompanied by much lower work velocities, which could be as low as 50 mm/min (2 in./min) and seldom exceeding 500 mm/min (20 in./min). Work velocity is inversely proportional to the infeed. Work velocities exceeding 250 mm/min (10 in./min) are generally accompanied by special dressing processes, such as continuous dressing, which enable the maintenance of a sharp, unclogged grinding wheel.

Since the infeeds are large, arc of contact between wheel and work is extremely large in comparison to conventional grinding. This results in each abrasive grain cutting a long chip. The wheel consequently has to have a very open structure to accommodate long chips.

The large arc of contact, which results in a large number of grains in simultaneous cutting action, requires high spindle power, which in turn results in large cutting forces and the generation of greater quanitites of heat than with conventional grinding. The machine tool has to have the necessary power and stiffness to withstand the larger forces, and a copious supply of

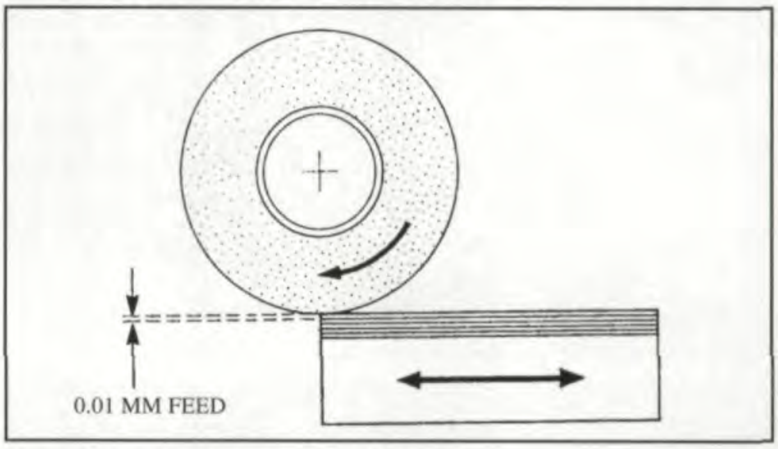

Fig. 6 - Conventional or pendulum grinding.

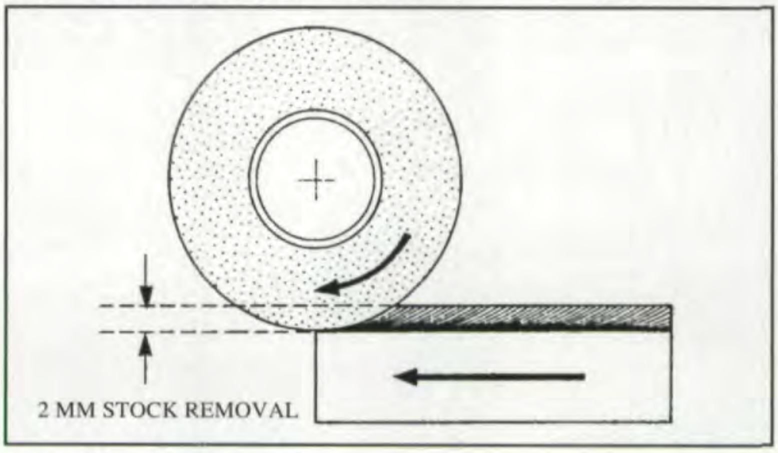

Fig. 7 - Creep-feed grinding.

well-directed coolant to carry away the heat generated in the process.

In spite of the larger power requirement, creep-feed grinding generally enjoys a higher G ratio than does conventional grinding when grinding similar materials. The lowered wheel wear is attributed to lower forces per abrasive in creep-feed grinding and also to the fact that in conventional grinding many wheel-work piece impacts are present as the work piece oscillates from side to side about the wheel.

Any means of eliminating the long chips produced in creep-feed grinding from loading the wheel will only improve the efficiency of the process. The use of high-pressure coolants to flush the wheel has been one techniques allowing higher work velocities. Another technique has been continuous dressing. Here the dressing roll, which may have the required form, is continuously fed into the wheel during the grinding process, with the grinding wheel being continuously fed into the work to compensate for reduction in wheel size. This continuous dressing keeps the wheel clean and sharp, allowing higher work velocities during creep-feed grinding. Experimental work where work velocities were in the 1m/min (40 in./min) range and higher have been reported.

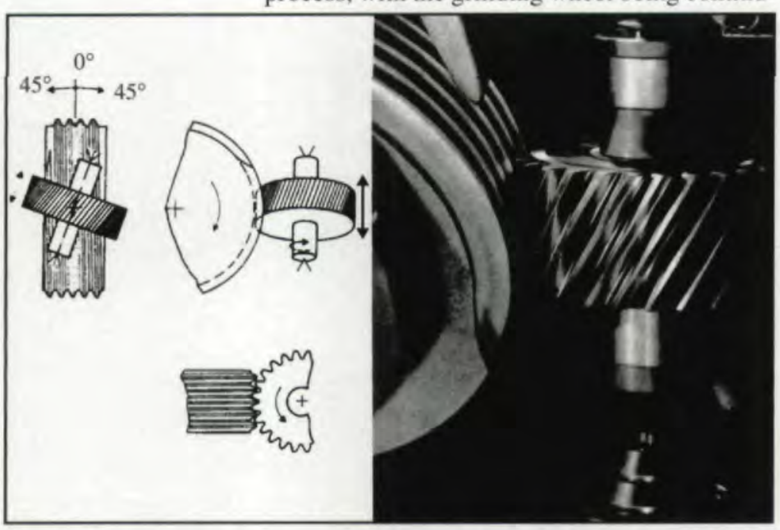

Fig. 8 - Kinematic representation of threaded wheel method.

Fig. 9 - Electronic threaded wheel-type gear grinder.

With this introduction to the various aspects of the grinding process, it is now possible to discuss gear grinding as practiced by the industry. The two most common techniques are form grinding and generating grinding. The techniques are now discussed in detail.

Generating Grinding

There are several basic techniques of generating grinding; each technique is associated with a specific machine-tool manufacturer. These distinct techniques are now presented.

Threaded Wheel Method. The basic machine motions that generate the gear in this method are kinematically illustrated in Fig. 8. The similarities of the mechanics of this technique to gear hobbing are very obvious, with the threaded grinding wheel replacing the hob. The ratio of work speed and wheel speed when grinding spur gears is a simple ratio of number of teeth on the gear and number of starts on the wheel. For helical gears this has compensated (differential indexed) for the traverse of the grinding wheel along the face width of the gear.

In order to be able to carry out the grinding process, the threaded grinding wheel, unlike the hob, has to achieve surface speed in excess of 25 m/s (5000 ft/min). The indexing mechanism has to be considerably more accurate in order to achieve the gear quality required in grinding. In the past, complex gear arrangements were normally used to obtain the simple and differential indexing requirements between the grinding wheel and work piece. However, *electronic gear boxes* (EGBs) are now commercially available to maintain the kinematic relationship. Fig. 9 shows a typical machine with an EGB for generating grinding.

The quality of the gear being ground is also significantly affected by the rack-type profile of the grinding wheel. It must first be introduced on a cylindrical wheel and then maintained through the grinding process as the wheel breaks down due to wear.

Introduction of the rack-type profile on a cylindrical wheel is done in two steps. A rough rack-type profile is crushed into the wheel using a steel crushing roll. This can then be finished by a variety of techniques using diamond tools, such as a single-point dresser or coated dressing disk. Profile modifications are introduced in this second step as required. With single-point dressing tools, profile modidifications are made using special cams. If coated disks are being used, the modifications are lapped into the disk by the machine manufacturer.

Saucer Wheels Method. This is another generating technique where two saucer-shaped wheels are used as shown in Fig. 10. The grinding surfaces of the two wheels represent the rack, and the involute profile is generated by the gear rolling relative to and in contact with the two grinding wheels. The wheels may be set parallel to each other or at an angle up to 10°. The work piece is reciprocated in the axial direction to provide the feed motion as two flanks of two different teeth are ground in one pass. At the end of the pass the entire gear is indexed using mechanical index heads so that two flanks of two or more teeth are then ground by the wheel. The depth of cut is determined by the infeed of the two grinding wheels toward each other.

For spur gears, the axes of the grinding wheels are perpendicular to the axis of the gear and only simple motion, to simulate the rolling of the gear on the rack represented by the grinding wheel, is needed to generate the involute. This generating motion is produced by steel tapes fixed to a stationary tap stand, the other end of which is wound over a rolling block that is generally the same diameter as the base-circle diameter of the profile being ground. This is illustrated in Fig. 11. When grinding a helical gear, the rolling motion that is necessary to generate the involute has to be compensated for the helix angle as the grinding wheels move along the face width of the gear. This is accomplished by a helix guide mechanism attached to the tape stand that is used to generate the rolling motion. The helix guide is set to the base helix angle, and, as the gear moves along its axis, additional motion is imparted to it to produce the helix along with the involute.

On older machines of this type, changeover from one gear to another required the change of the rolling block for each change in base-circle diameter. On modern machines, mechanisms have been developed that allow a range of base-

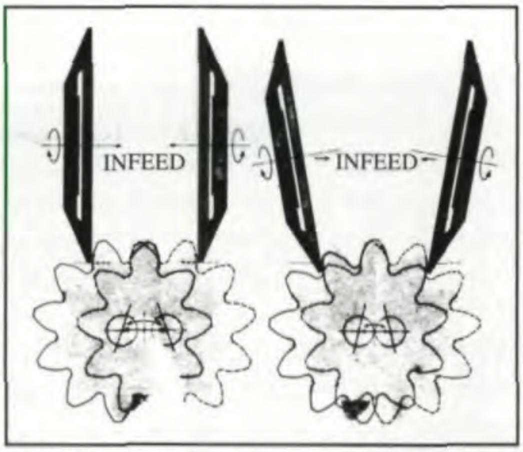

Fig. 10 - Saucer wheel method.

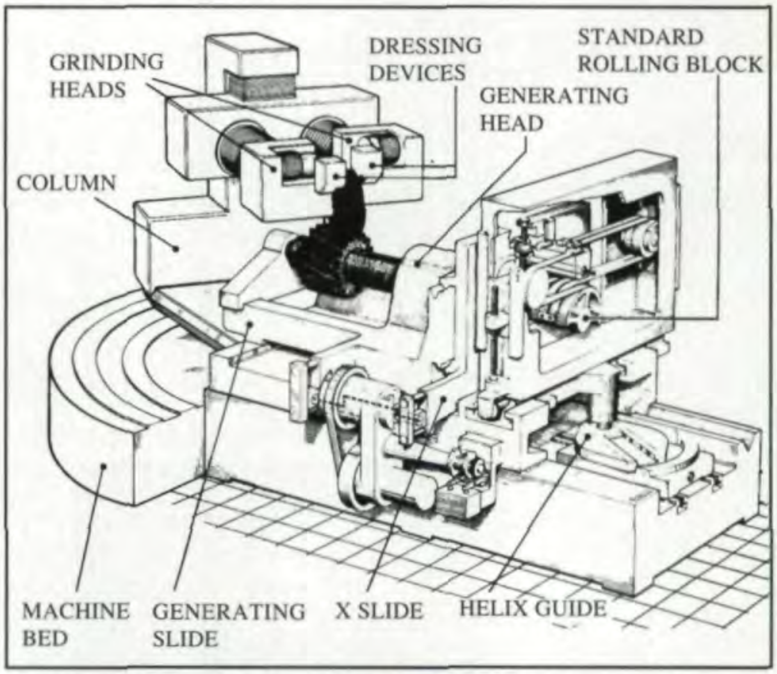

Fig. 11 - Typical saucer wheel grinder showing basic components.

circle diameters that can be ground with the same rolling block.

The contact between the saucer-shaped wheel and the tooth flank is generally restricted to a very small area at any given time. This generally makes this technique of gear grinding time consuming and slow. However, it also enables point-by-point profile and lead modification along the flank of the tooth technique, termed *topological modification*. Consequently, the profile of the gear tooth can be different along the entire face width of the gear, a feature that none of the other gear grinding techniques, form or generation, can duplicate. Use of computers to control this topological grinding feature allows an infinite variety of tooth forms to be ground. It must be remembered, however, that this feature is at the price of slower cycle time, and tradeoffs have to be examined before a decision is made to use this technique. Also, at the present time, apart from a few gear tool applications, no other applications of topological grinding have been applied.

Vitrified aluminum oxide wheels are most commonly used in this method of gear grinding, and the grinding process is generally done "dry." Dressing is carried out with wheel compensation, using single-point diamonds. The purpose of dressing is to ensure that the rotating surface of the wheel represents a straight tooth of a generating rack. Single-layer, cubic boron nitride-plated wheels have also been tried, as they eliminate the need for dressing, and wheel life is high for reasons already explained in the text.

Conical Wheel Grinder. This is another version of generating grinding using a grinding wheel that represents a single tooth of the rack, as shown in Fig. 12. The sides of the wheel correspond to the pressure angle of the gear being ground. The work gear rotates and translates linearly to generate the rolling action required to generate the involute profile.

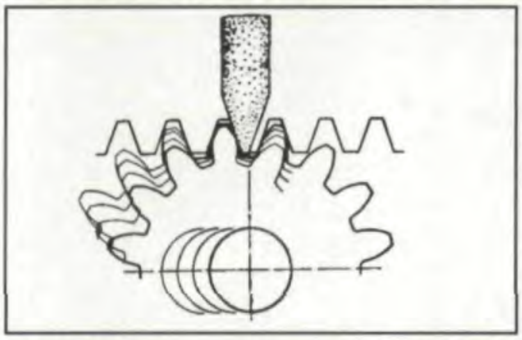

Fig. 12 - Basic concept of a conical wheel grinder.

Fig. 13 - Conical wheel grinder finishing a double-helical gear in one setup.

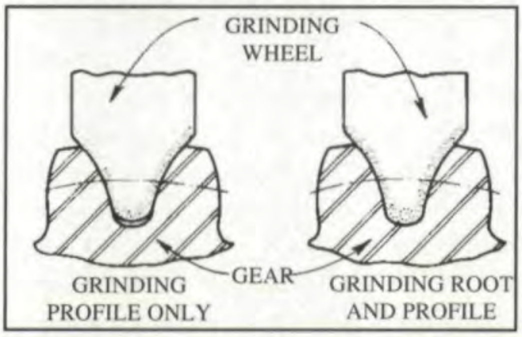

Fig. 14 - Basic concept of form gear grinding.

The simultaneous rolling and linear motion is generally obtained by having a master gear with the same number of teeth mounted on the work spindle rolling on a stationary master rack. The master gear and rack must correspond to the gear being ground in terms of number of teeth, pressure angle, diametral pitch, etc. Electronic means of varying the base roll diameter to correspond to the gear being processed are, however, now available. In this process, as with the previously discussed saucer wheels method, two flanks of two different teeth are finished before the gear is indexed to grind two more flanks of two more teeth.

Helical gears can also be generated by this technique, though helical master gears and racks are required. If the wheel needs to be dressed, diamond points operating at the specific pressure angle are required. Since simple straight line forms need to be dressed, the dressing mechanism is relatively simple. Tooth profile modifications are produced by the modified grinding wheel. For lead modifications, the tool slide with the grinding wheel is radially advanced in synchronism with the stroking motion of the grinding slide, controlled through a tracer roll following the slope of the template. A moderate-sized conical wheel grinder, grinding a double helical gear is shown in Fig. 13.

Form Grinding

In this technique, the abrasive grinding wheel is profiled to represent the space between two adjacent teeth on a gear. The wheel is then passed through the space while grinding occurs on the two adjacent teeth flanks and the root, if required, as shown in Fig. 14. This is one of the primary advantages of form grinding in that various simple and compound root forms can be produced. Form grinding also enables the grinding of internal gears and external gears positioned against a shoulder.

When a spur gear is being ground, the wheel is simply moved along the axis of the gear. When a helical gear is required, the axial motion of the wheel is combined with a motion of the gear about its axis in order to produce the lead. The various axes of motion required to manufacture a gear on a horizontal axis grinder are illustrated in Fig. 15. The A axis provides the tooth-to-tooth index and, when interpolated with the X axis, generates the lead. The Y axis provides size control and, in combination with the X axis, provides lead modifications. The B axis allows the wheel to be set to the helix angle of the part. On a horizontal axis machine, two more axes are generally required to dress the profile on the grinding wheel. These are marked as the V and W axes.

An analysis of current gear grinding equipment indicates that form grinders are ahead of the generating machines in the application of computer control to gear grinding. In most generating machines it was found that only some aspects of the process were under computer control, while other aspects used mechanical control devices such as index plates, sine bars, or cams. However, contemporary form grinders appear to have completely abandoned mechanical devices in favor of computer control and appear to be doing as well or better than the older form-grinding machines. Also, computer control enables these form grinders to be more flexible and require less setup time than their generating counterparts.

Since the wheel profile is constant, modifying the lead by Y axis motion, which results in a change in center distance between the grinding wheel and the gear, will result in slight distortions to the profile. Lead modifications through change in the interpolation relationships between the A and X axes are also possible.

It is also important to note that, when grinding a helical gear, the normal tooth space that is represented by the grinding wheel has to be modified to account for the interference that occurs between the wheel and the helical groove, commonly termed *heel-toe action*. This heel-toe action is a function of the wheel diameter. Consequently, this has to be compensated for wheel diameter reduction during dressing to avoid errors in profiles.

In the past, mechanical devices such as sine bars, index plates, and cams were used to generate the helix, index, and profile, respectively. On modern computer-controlled machines, such as the one shown in Fig. 16, software generates and controls the relationships. Consequently, compensating for the involute as the wheel diameter

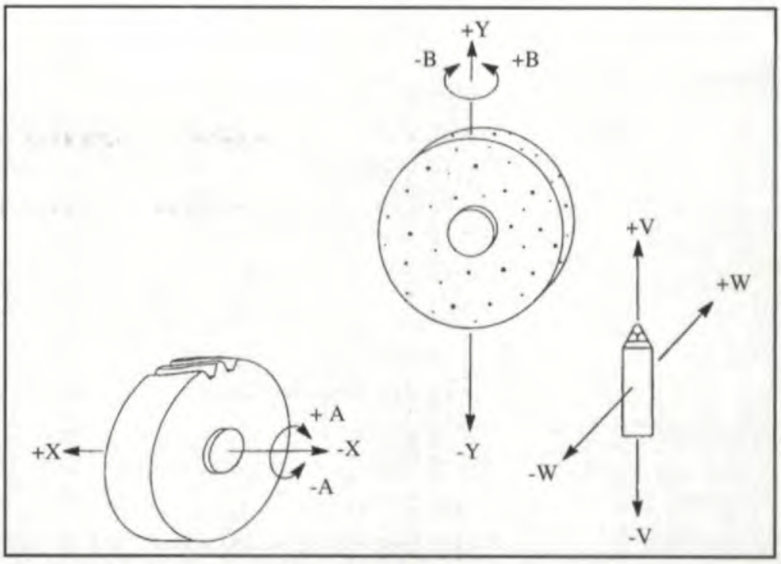

Fig. 15 - Axes of motion for form grinding.


Fig. 16 - CNC form gear grinder.

changes due to dressing can be done just as easily as speeding up the spindle is carried out for changes in wheel diameter in order to maintain constant wheel surface speeds. The only necessity is that the machine constitute a set of necessary accurate linear and rotary axes.

Wheeling trueing, profiling, and dressing are accomplished by the dressing mechanism. A diamond disk or single-point diamonds can be used. If production volumes can justify it, a diamond preformed dressing roll can be used to reduce dressing times and increase productivity. Alternatively, electroplated preformed cubic boron nitride wheels can be used, and dressing times can be completely eliminated (keeping in mind that preformed wheels may cost up to 100 times the cost of a dressable aluminum oxide wheel). Vitrified, dressable cubic boron nitride wheels can also be used. These need to be dressed, but not as often as aluminum oxide wheels, and are generally cheaper or about the same cost as a plated wheel.

The coordinates describing the profile that is used to control the dressing device are also generated by software. Two basic approaches are evident. One is a more fundamental approach based on solid geometry, where the grinding wheel and work piece are considered as two cylinders intersecting each other at a present distance and angle between the two axes. The shape of the intersecting surface on one of the cylinders, that is, the workpiece, is defined by the specified profile. Consequently the shape of the wheel surface can be computed. The profile may be an involute with modifications or a noninvolute if required. The other basic approach is heuristic or data-based in which profile coordinates corresponding to different pressure angles, modules (diametral pitch), base-circle diameters, and helix angles at P. D. are stored. Interpolated values of coordinates for other profiles can then be obtained. This approach is more limited in scope and may need a few trials to arrive at the right profile.

The ability of form grinding to produce noninvolute forms cannot be overstressed. Generating grinding is limited in this area as the gear profile is due to the rolling action of the work against the wheel. Form grinding is, on the other hand, limited only by the type of forms that can be generated on the wheel.

Since the accuracy of profile obtained in form grinding is directly impacted by wheel wear, any technique that could reduce wheel wear is obviously of benefit to the economics of the operation. Plated cubic boron nitride wheels, where wear of the wheel is almost nonexistent, represent one approach, providing it can be cost-justified. Creep-feed grinding, with its accompanying reductions in wheel wear, is another. In order to accommodate the creep-feed grinding process, current machines have been designed and built with high spindle power and high static stiffness to utilize the power, low table speeds, and large coolant flows. All these features have enabled the application of advanced processes to the technique of form gear grinding.

Cycle Time Estimates

These are essential in job shops for quoting purposes before a job of grinding a gear can be started. Keeping in mind the variety of gear grinding techniques available and the variety, of grinding processes that could be utilized, development of specific formulae to suit each process and technique was considered futile. Instead, a more general approach is now presented: an approach that can be modified to suit each technique or process as necessary.

The total time required to grind a gear is given by the expression:

Total time = grind cycle time + work handling time + setup time per gear.

Since the work handling time and setup time per gear are functions of sophistication and type of work-handling equipment and machine tool and the skill of the operator, further discussion is restrained to the grind cycle time only.

The *grind cycle time* is given by the generalized expression:

Grind cycle time = grind time + index time + wheel dress time + reset time.

Further, *grind time* is given by the expression:

$$\text{Grind time} = \frac{\text{gear face width} + \text{overtravel}}{\text{work traverse velocity}} \times \text{number of traverses} \times \text{number of gear teeth.}$$

Gear face width and number of teeth can be obtained from a part print; the amount of overtravel is a value necessary to clear the part for the purposes of indexing; and the work traverse velocity is a parameter that is dependent on a variety of factors, including the process and the type of machine tool being utilized. The number of traverses required for grinding is given by the formula

$$\text{Number of traverses} = \frac{\text{rough stock}}{\text{rough infeed}} + \frac{\text{semifinish stock}}{\text{semifinish infeed}} + \frac{\text{finish stock}}{\text{finish infeed}} + \text{number of spark-out traverses.}$$

All these aforementioned parameters are part- and process-dependent variables. The number of spark-out traverses is also dependent on the incoming quality of the gear, the required outgoing quality, and the stiffness of the machine tool being utilized. If the work traverse velocity in the expression for grind time changes during the rough, semifinish, finish, and spark-out, different grind times have to be calculated for each part of the process and summed to get the total grind time.

The *index time*, which does not exist in the case of generating grinding with a threaded wheel, is given by the expression:

The *index time* = time per index × number of gear teeth × number of index traverses.

The time per index, usually a few seconds, is dependent on the type of machine tool and the number of teeth and is a part-dependent parameter. The number of index traverses is based on the processes and can be computed from the expression:

$$\text{Number of index traverses} = \frac{\text{number of rough traverses}}{\text{number of traverses/index}} + \frac{\text{number of semifinish traverses}}{\text{number of traverses/index}} + \ldots$$

This combination is due to the possibility of a number of traverses with grinding infeed on the same tooth before indexing, a technique that is used occasionally while roughing on a form grinder or generating grinder using saucer-shaped or conical grinding wheels. If the grinding process being used requires indexing after every traverse, then the number of index traverses is the same as the number of traverses.

The *dress time* is given by the expression:

Dress time = time per dress + number of dresses per gear.

In some types of generating grinding a number of gears may be finished between dresses and so a fractional value will have to be used for the parameter "number of dresses per gear." Also, the time per dress during roughing may be different from the time per dress during semifinishing, finishing, or spark-out, in which case the formula may have to be expanded to account for all these variables. Since a certain amount of time is usually required to bring the dressing mechanism into action at the start of each dressing cycle, this time should be added for a more accurate estimate of dress time.

The gear being ground, in almost all instances, is held between two elements on the machine during the grinding process, for example, a headstock and a tailstock for an external gear. Of the two, one is general the stiffer member, and consequently, it is preferable to grind against this member in what is generally characterized as unidirection grinding. (This is not to say that bidirectional grinding cannot be done, though this is restricted to roughing passes only.) The *reset time* is the idle time lost to reset the machine to do unidirectional grinding and given by the formula:

$$\text{Reset time} = \frac{\text{face width + overtravel}}{\text{wheel return speed}} \times \text{number of traverses} \times \text{number of gear teeth.}$$

The wheel return speed is generally the rapid traverse rate on the machine, though depending on the amount of travel, the table of the machine may never reach that speed. A lower rate should generally be used to account for acceleration and deceleration. If a combination of bidirectional and unidirectional grinding is used, the formulae have to be modified to suit the requirements.

As stated earlier, only the general approach to estimation of cycle time is presented here. These have to be modified to suit the grinding technique and precess selected. Above all, process parameters, such as feed rates and infeeds, have to be valid because the quality of the cycle time estimate is vitally dependent on them. ■

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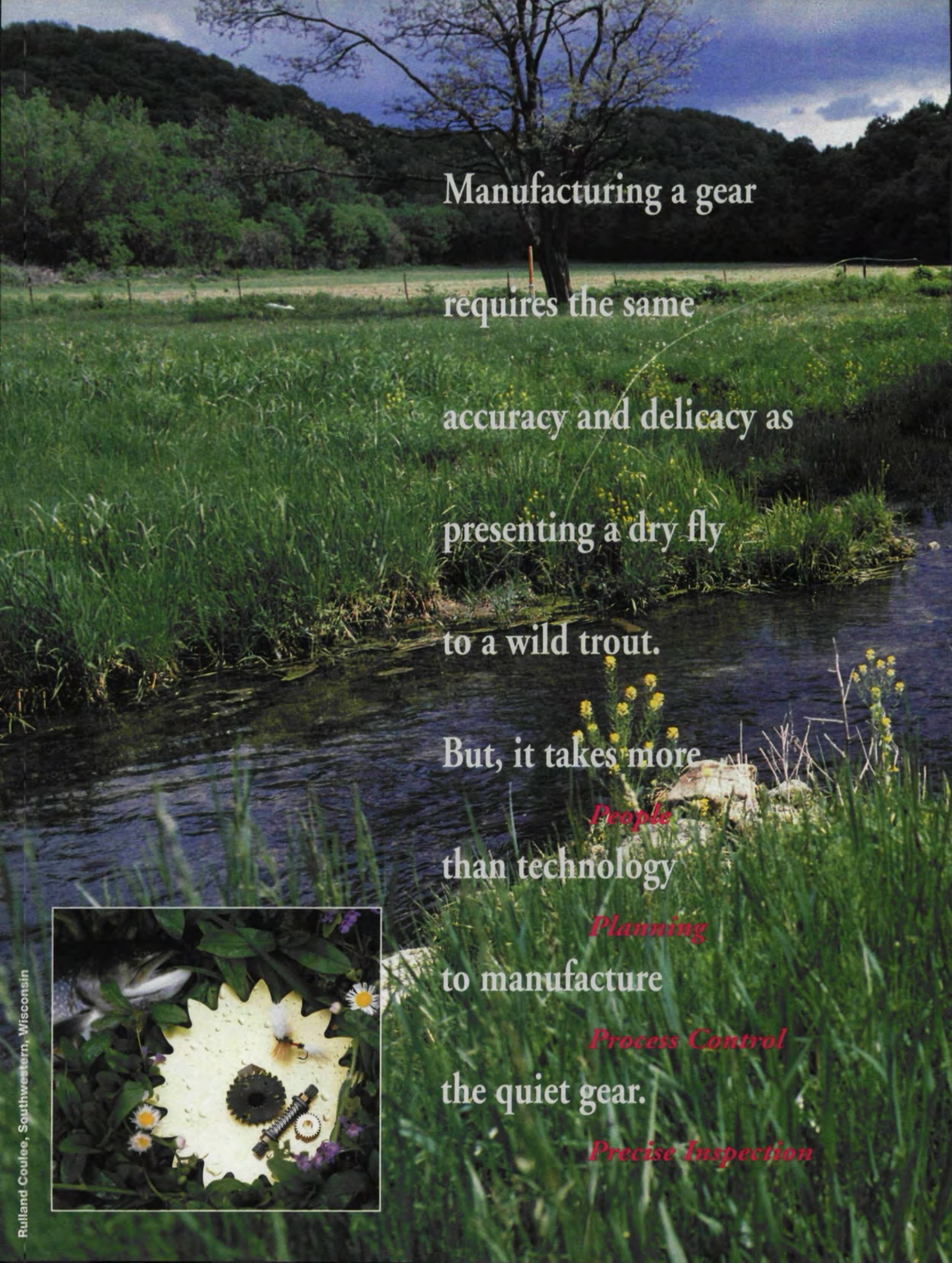

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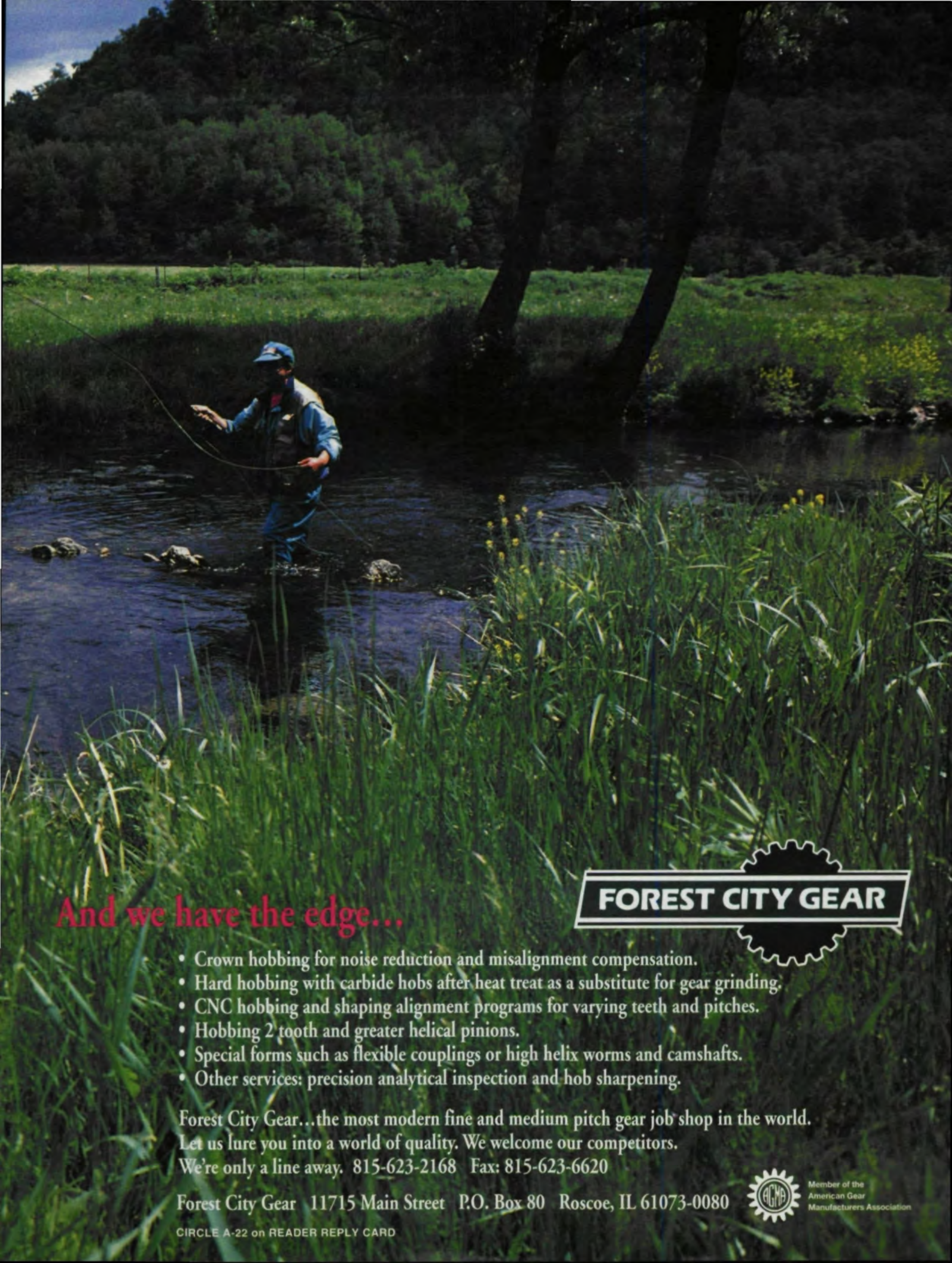
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Comparing Surface Failure Modes in Bearings and Gears: Appearances vs. Mechanisms

Charles A. Moyer
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Table 1 - Damage Classification For Bearings

I. FATIGUE

Contact Fatigue - Subsurface Origin

- Inclusion - with macroshear classical shear stress zone below contact surface
- Inclusion - near surface zone of microshear greatly influenced by surface roughness (asperities)
- Subcase fatigue - origin near case-core interface if yield strength is exceeded by applied stress

Contact Fatigue - Surface Origin

- Origin at the end of contact aggravated by edge geometry
- Multiple origins of micropitting (peeling or frosting)
- Point surface origin - at localized stress risers (dents, grooves, surface inclusions)

II. PLASTIC FLOW

- Brinelling or debris denting
- Load excursions above the material yield point
- Yielding aggravated by high temperature excursions

III. WEAR

A. Adhesive
  - Normal - mild or "controlled" - usually identified as "run-in"
  - Severe - irreversible scuffing, scoring, smearing, or seizure

B. Abrasive
  - Normal - usually 3 body system, medium to fine particles that are also associated with "run-in"
  - Severe - grooving, gouging, denting with ridges that cause serious surface stress risers

C. Corrosive
  - Water or acidic constituents from lubricant breakdown or temperature chemically aggressive additives in lubricant

D. Fretting
  - Microscale adhesive and abrasive wear
  - Corrosion involvement depending on environment and contacts

Introduction

In the 1960's and early 1970's, considerable work was done to identify the various modes of damage that ended the lives of rolling element bearings.[1-5] A simple summary of all the damage modes that could lead to failure is given in Table 1. In bearing applications that have insufficient or improper lubricant, or have contaminants (water, solid particles) or poor sealing, failures, such as excessive wear or vibration or corrosion, may occur, rather than contact fatigue. Usually other components in the overall system besides bearings also suffer. Over the years, builders of transmissions, axles, and gear boxes that comprise such systems have understood the need to improve the operating environment within such units, so that some system life improvements have taken place.

Those of us who manufacture bearings realized that identifying the damage modes was not enough, but that an understanding of the causes and underlying mechanisms at play was important to improve the ability to predict bearing life in this variety of systems. However, whenever adverse operating conditions prevailed, and actual life was below expected life, the understanding of modes of failure within bearings could be used to determine what improvements were needed to allow bearings to have extended life even under adverse conditions.

Considerable work has been done to accomplish this, and more remains to be done, but through understanding expected failure modes based on known operating conditions and applying this knowledge to the design cycle, significant improvements in bearing performance have been achieved. For line contact bearings, the prime factors are (1) significantly cleaner steel, (2) much improved surface finishes, which include roughness, waviness, and other factors beyond R_a (the arithmetic center-line average of roughness), and (3) geometry that optimizes internal stress profiles, so that even under high-load or nonaligned conditions much more uniform stresses can be maintained along the contact line.

The same development path seems to have been occurring for gear contacts. Thus, we can review the failure modes that occur in gears and, based on the understanding of what mechanisms underlie the various identified failure modes, see what design changes may be appropriate that would also extend gear performance life beyond present limits.

One way to do this is to compare the modes of damage identified in line contact bearings and gears to determine what is happening within their respective contacts that cause the final failures. Since both bearings and gears function primarily with fluid lubrication, a tribological model of line contact, as shown in Fig. 1, can be the starting point to discuss the mechanisms that contribute to the various failure modes reported.

The Basic Line Contact Model

Whether gears or roller bearings, the contacting surfaces can be represented as two cylinders. It is possible to include transverse profiles or radii on these cylinders and determine the appropriate elliptical or truncated elliptical contact formed; but for this discussion two simple cylinders of the same or different radii are sufficient. Under running conditions, each cylinder has surface velocities either close to the same or considerably different that result in tangential or frictional forces within the contact that can range from less than 1% to well over 20% of the normal force that forms the rectangular contact area between the two cylinders.

The two cylinders under load determine the Hertzian contact area on the X-Y plane shown on Fig. 1. The cylinder length along the Y axis and 2b, the width of contact along the X axis, form the Hertzian contact shown by the straight dashed lines. The Hertzian contact pressure reaches a

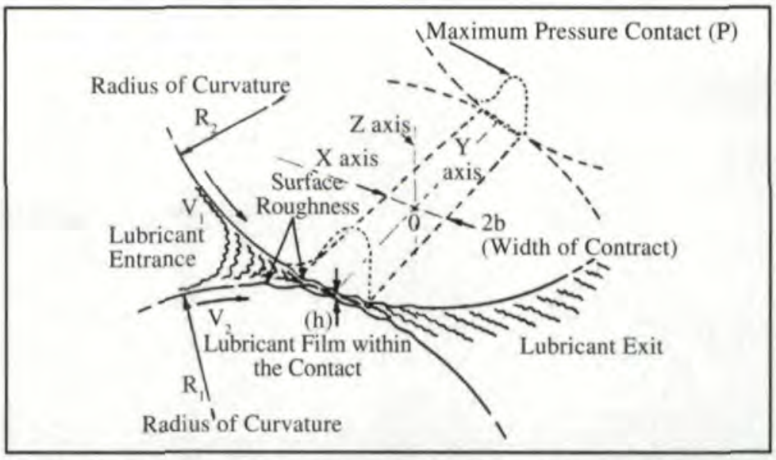

Fig. 1 - The tribology model of line contact. Two cylinders in contact (R_1 and R_2) under load with surfaces moving (V_1 and V_2). X axis (in the direction of motion) and Y axis (at right angles to motion) define the contact plane. The Z axis at right angles to the contact plane refers to the distances into the contacting cylinders. The contact length on the Y axis and the contact width 2b define the Hertzian stress contact area.

maximum P as indicated by the dotted lines above the contact rectangle. P directly relates to the subsurface principal and shear stresses to be discussed later.

If there is sufficient lubricant in the entering meniscus to the contact, an elastohydrodynamic lubricant (EHL) film (h) will be formed. Recognizing that the cylindrical surfaces have some amount of roughness, the lubricant film (h) and roughness (σ) can be compared to give an indication of the quality of the lubricant operating regime within the contact; that is, the EHL film (h) divided by the composite roughness of the two surfaces (σ) provide h/σ or lambda (λ).

Testing on ball and roller contacts, as for example in a tapered roller bearing,[6] demonstrates that the contact itself (2b) actually operates as a functional filter, and only the surface roughness wavelengths that are approximately one-fourth 2b up to twice 2b are active within the contact. Considering gear contacts, Wellauer and Holloway understood this in their paper on the application of EHL lubricant films to gears presented in 1975.[7] A simple approach to address such a concept was presented as a modified lambda ratio in 1989.[8] The concept was included in a paper at the 1989 AGMA Fall Technical meeting[9] and will be in the next revision of the ANSI/AGMA Standard 2001-B88, Appendix A.

The modified lambda can be expressed as:

$$\lambda m = \underline{h}\ \_\ \frac{L}{2b}^{1/2} \qquad (1)$$

where λm = the modified lambda

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L = large end wavelength cutoff used to measure surface roughness [usually L = 0.8mm (.030 inch)]

2b = width of contact in the direction of motion

Any further discussion of lambda, especially as used later in Table 3, is primarily based on λm rather than λ not modified.

Under load the two cylinders, besides producing the contact stress profile shown in Fig. 1, also produce stresses below the surfaces. Of most interest are some specific shear stresses formed from the principal stresses σ_x, σ_y, σ_z and the shear stresses T_{xz}, T_{yz}, and T_{xy} that occur within the stressed volume below the Hertzian contact area. The first of these is the maximum shear stress (T_{45}) that is in the center of contact:

$$T_{45} = 0.5\,(\sigma_z - \sigma_x) \qquad (2)$$

where T_{45} = shear stress on 45° plane from the contact surface in the rolling direction and at x (or b) = 0.

σ_x = principal stress in the x direction (Fig. 1).

σ_z= principal stress in z direction (Fig.1).

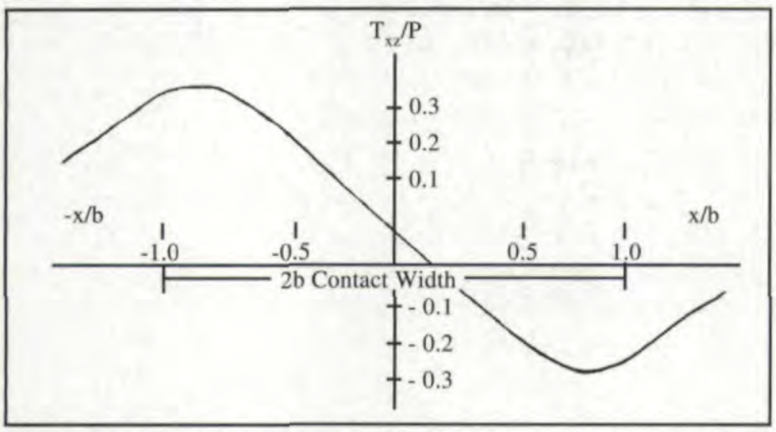

Fig. 2 - The range of orthogonal shear stress T_{xz} as a fraction of the Hertzian contact pressure P. The values go from a maximum of -0.25P through zero to +0.25P at 0.5b below the contact surface for pure rolling.[10]

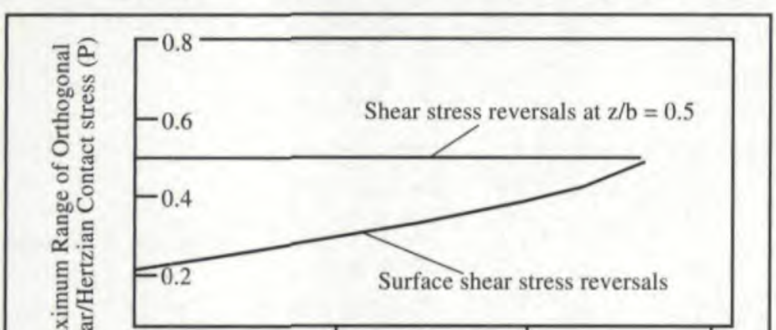

Fig. 3 - Influence of the coefficient of friction on the maximum range of orthogonal shear stress in terms of the ratio $|2T_{xz}|/P$ where P is the maximum Hertzian contact stress for line contact.[10]

The maximum of T_{45} occurs on the center line of the rectangular contact (b = o) and below the surface at 0.78b, where b is one-half the width of contact. Under rolling conditions, T_{45} is equal to 0.3P (P equals maximum Hertzian pressure in Fig. 1).

While T_{45} seems to relate to the yield limit of stressed material, the orthogonal shear stress, usually T_{xz} (in the direction of motion) seems to be related to rolling or rolling/sliding contact fatigue. Of interest here is the range of orthogonal shear that occurs under a contact area as shown in Fig. 2. This figure is based on the results published by Kannel and Tevaarwerk [10] and shows for pure rolling that the maximum T_{xz} occurs about 0.5b below the contact surface and has opposite values across the contact of ±0.25P; that is, the range of shear stress is 0.5P, and this magnitude along with the depth below surface is strongly related to fatigue. There is not space here to develop the equations used to calculate T_{xz}, but the reader is directed to Refs. 10 and 11 for further details.

Another equivalent stress considered with contact fatigue is the von Mises yield parameter (σ Mises) that is represented by:

$$\sigma\ \text{Mises} = [T_{xy}{}^2 + T_{yz}{}^2 + T_{xz}{}^2 + 1/6\ \{(\sigma_x - \sigma_y)^2 + (\sigma_x - \sigma_z)^2 + (\sigma_y = \sigma_z)^2\}]^{1/2} \qquad (3)$$

where all stresses are the principal or shear stresses on or below the contact surface in either the direction of motion (x-axis) or in the mutually perpendicular directions (y-axis and z-axis).

These three stresses, along with a few others, have been used to evaluate what stress best relates to contact fatigue and the onset of plastic deformation.

When the surface velocity difference between the velocities of the two cylinders is low (low-sliding), and the surfaces are separated by a lubricant film ($\lambda m > 1.0$), the shear stress will be below the surface and contribute only to material related fatigue. As the velocity difference increases (larger slide/roll ratio), and/or with considerable surface contact ($\lambda m < 1.0$), the shear stresses are larger (contribution from higher frictional forces) and, as an example, the orthogonal shear stress can extend up to the surface. This is shown in Fig. 3, taken from Kannel and Tevaarwerk,[10] and indicates how the maximum shear stress reversals in the surface increase with friction force increase. This means that from the increased friction a larger volume of material below the surface sees significant shear stresses.

Surface roughness, besides contributing to the λm calculation, determines the actual contact area that occurs in comparison to the contact calculated from Hertzian theory. Surface roughness is very often represented in two dimensions as a field of asperities, and under load only a finite number are in contact. As the load increases more contacts occur, and some existing contacts grow larger, but the actual contact area never reaches the full theoretical contact. The run-in that occurs for bearings and gears includes the portion of these asperities in contact that are plastically deformed and worn away, depending on the amount of tangential force (sliding) that accompanies the normal load in the contact. It is clear that the size of these asperities, their side slopes, hardness, and the lubricant film that tries to separate the two rough surfaces, all contribute to run-in and to the mode of damage the surfaces may eventually see.

A real surface can be represented by a string of asperities in a line. These multiple contacts form micro-contact stresses and subsurface shear stresses in miniature below each loaded asperity so that, depending on λm, shear stresses may be formed below the surface. Figs. 4 & 5 provide representations of real surfaces that are stressed under two different λm values. Besides the variation of contact stresses that deviate from the Hertzian stress calculated assuming ideal surfaces (zero roughness), these figures show the subsurface shear stresses and the shear stresses in miniature below the surface as modified by the roughness, and illustrate the changes that can occur with a change in λm.

These figures come from the work of Zhou and Cheng[12] based on the actual surface finish on specimens in a two-disk machine described in Ref. 12. They illustrate the subsurface and near surface shear stresses that must be included in any analysis of the basic line contact model. They have used the von Mises stress, and this stress illustrates the influence of the surface asperities as they are able to penetrate the lubricant film. For λm greater than 3.0, the maximum von Mises stress is 620 MPA (90 KSI) and located 338 μm (0.013") below the contact surface. For λm equal to 0.33, the maximum von Mises stress is 1450 MPA (210 KSI) and only 40 μm (0.0016") below

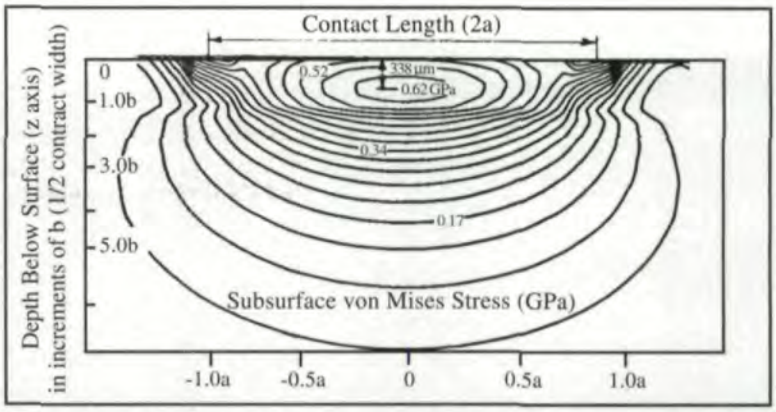

Fig. 4 - Contours of von Mises stress below the contact. Hertzian contact pressure 2.3GPa, lambda ratio above 3.0. Maximum stress is 0.62GPa, occurs 338 μm below the surface or 0.78b (width of contact) below the surface showing no influence from surface asperities.[12]

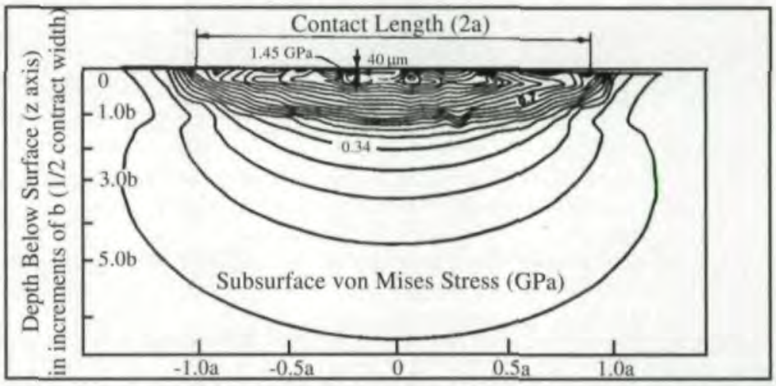

Fig. 5 - Contours of von Mises stress below the contact. Hertzian contact pressure 2.3GPa, lambda ratio 0.33. Maximum stress is 1.45GPa, only 40 μm below the surface or 0.09b below the surface showing considerable asperity interaction.[12]

the surface. All these aspects of the line contact model must be kept in mind as we discuss the fundamentals of wear mechanisms.

The Basic Wear Mechanisms

Within the context of the line contact model, wear can be considered in its broadest terms. As expressed in an excellent review paper on "Wear and Wear Mechanisms" by Vingsbo,[13] wear can be defined as the removal of material from surfaces in a tribology system. And basic to this definition is that the fundamental phenomenon in material removal is fracture. Vingsbo considers four mechanisms.

The first is abrasive wear, termed ploughing or microcutting, that separates a fragment or chip from the main body. Also associated with this mechanism are various amounts of plastic deformation. Abrasive wear also assumes that one surface has harder asperities than the surface it abrades, or that hard particles (three body system) are moving across the surface. This mechanism implies the presence of slip or considerable sliding between the surfaces. In steels, with which we are primarily concerned here, a large portion of energy expended is used in the plastic deformation and primarily forms ridges along the furrow sides.

The second mechanism is adhesive wear or shearing of junctions. This considers asperity-asperity contact or asperity-surface contact, depending on the relative surface finishes. Again considerable plastic strain and frictional work from the localized slip (hot spots) or sliding takes place. Obviously, the energy into the points or areas of contact dissipates as heat that helps remove oxide layers or contaminants on the surfaces. This then allows the asperities to form intimate contact or "weld." If the binding energy of the join (weld) is stronger than material away from the surface contact, fracture will take place within the adjacent material. This causes material transfer, and with continued plastic flow from sliding, the transfer areas may break loose and form third body fragments.

Table 2 - Damage Classification for Gears[18]

I. (4.) SURFACE FATIGUE

(4.1) Pitting - Initial (0.4-0.5mm size pits) to destructive (considerably larger) usually takes place in the dedendum (negative sliding).

(4.2) Spalling - Larger irregular shaped spalls, but quite shallow compared to destructive pitting.

(4.3) Case Crushing - Subsurface fatigue failure just below hardened case with longitudinal cracks ending on the surface.

II. (5.) PLASTIC FLOW

(5.1) Cold Flow - Extreme surface deformation from heavily loaded contacts - appearance of extrusion or peening.

(5.2) Rippling - Periodic, wave-like deformation with fish scale appearance usually seen on hardened gear surfaces.

(5.3) Ridging - Deep ridges from plastic deformation from high stresses.
(-) Indentation - Deep plastic flow from rolling, brusing, peening, or brinelling.[26]

III. (3.) WEAR

(3.1) Polishing - Fairly uniform mild wear - associated with run-in process

(3.2) Moderate Wear - Visible contact pattern - more material loss than in polishing.

(3.3) Excessive Wear - Pronounced wear with sufficient material loss to degrade gear design life.

(3.4) Abrasive Wear - Lapped, radial scratch or grooving on tooth contact surface from hard or metallic debris particles.

(3.5) Corrosive Wear - Surface deterioration from chemical action from liquid contaminants, lubricant breakdown, acid products, or moisture.

(3.6) Frosting - Etched appearance, micropitting with field of micropits 2.5m deep (0.0001 in.).

(3.7) Scoring - From moderate to destructive or localized, advanced occurrence of adhesive wear, alternate welding and tearing, ad vanced frosting, or significant material loss.

The third mechanism is fatigue. It is normally found in high-cycle stress applications, usually with stresses below the yield point of the material. Thus, fatigue seems to lack the extended plastic strains that occur during the operation of the first two mechanisms. However, there is ample evidence that within bearings, especially those run at higher loading, there is microstructural alteration within the stressed volume due to cyclic stressing.[14] At the stress levels that cause fatigue, evidence of plastic deformation occurs in the highest stressed zone and around localized stress risers, such as non-metallic inclusions. Fatigue can occur without sliding (without significant frictional force) and cracks can nucleate below the contact surface at either the subsurface shear stress regions or below the asperities at miniature shear stress areas as noted in Figs. 4 and 5, and in contribution with subsurface stress risers for either shear stress locations. These stress risers and those on the surface all contribute to the statistical nature of contact fatigue; it tends to follow a different life distribution than wear.

The fourth mechanism Vingsbo considers in his paper is impact, which may have some relevance here; however, most impact fractures tend to occur in brittle material, so it will not be considered further.

The underlying phenomenon that ties these wear mechanisms together is material fracture. The factors that influence the wear rate and wear severity in different degrees for each of these mechanisms are the forces involved, quality of lubricant, ranging from full separating film to boundary lubrication, surface roughness, surface velocity in terms of rolling speed and sliding, and the surrounding environment that inhibits or promotes surface absorption, oxide formation, and other chemical reactions.

Besides material loss (wear) and debris or particle generation, these factors also influence the friction levels, heat, and temperature, which can alter the surface and subsurface areas. The additional changes that take place can directly influence the failure modes, especially for steels.

Microstructural changes of the material can occur just below the contacting surface, and even compositional changes can occur as heat input enhances diffusion into the near surface material. Probably the greatest changes occur from the plastic deformation that accompanies almost all of these mechanisms and also modifies the near surface material structure leading to the microcracks that can be the starting point of surface/subsurface degradation.

Observations of Structural/ Mechanisms Interaction

In recent years, considerable work has been done to understand the changes occurring within the material from repeated stressing from surface contacts. Some of the results indicated that with significant tangential forces present, the mechanism leading to wear particle formation was less adhesion-related, but primarily due to plastic deformation and fracture under a delamination mechanism.[14-15]

This concept considers that sliding surfaces go from asperity-asperity contact with adhesion, ploughing, and deformation in the contacts to form larger actual contact areas that continue the subsurface deformation until cracks nucleate somewhat parallel to the surface. As the cracks propagate, some migrate to the surface (or possibly start from the surface), then flat, thin, plate-like wear particles are formed. Although there is not universal agreement to the delamination theory,[16] there is certainly evidence that very close to the surface a deformed, "fragmental" layer forms, and below a layer of deformation microstructure forms that is "cellular," smaller grained, and relatively free of dislocations.[16] Some excellent work has also been done to relate the significance of plastic deformation of the surface layers to actual gear pitting.[17]

Cummins and Doyle[17] ran annealed and case-hardened gears under conditions to produce surface distress and concluded that some of the debris formed during running-in consisted of submicron wear particles that by themselves would produce only mild wear and contribute to the mutual polishing of the gear surfaces. In addition, they found larger platelets and wedge-shaped particles that were formed after the run-in when larger areas of gear surfaces were in contact. In careful examination of the wear surfaces and the debris particles, as observed in both scanning electron and transmission electron micrographs, it was clear that the fine debris came from surface layers that had been significantly deformed and consisted of very fine grain material. The large particles were primarily thin platelets or larger flat pieces with wedge-shaped undersides. Based on the pits on the gear surface, the thin platelets come from the shallow pits lacking fracture-type bottoms, and the wedge pieces seemed to come from the deeper, more severe pits with some fractographic topography on the pit bottom.

Table 3 - Relating Contact Fatigue Damage Mode to Modified Lambda

Modified Lambda	Contact Fatigue Mode (Initiation)	Material Influence	Surface Roughness Influence	Geometry Influence
Ratio > 3.0	Subsurface Fatigue Inclusion[1]	Important	Minor	Important
Ratio 3-1.0	Subsurface/Near Surface Mostly Inclusion	Important	"Sharp"/High Asperities Important	Important
Ratio 1-0.3	Some Inclusion, Some Surface Related	Somewhat Important	Important for Surface & Near-Surface Fatigue	Somewhat Important
Ratio < 0.3	Surface Related[2]	Minor	Important	Less Important
Any Ratio	Localized Stress Risers	Mixed	Mixed	Mixed
	(PSO[3] - dents, grooves, and surface inclusions)	Minor	Somewhat Important	Minor
	(GSC[4] - edge fatigue misalignment)	Somewhat Important	Minor	Important

(1) Fatigue originates at non-metallic inclusion in the maximum shear zone below the surface for both bearings and gears.

(2) Called peeling or micorpitting for bearings[4]; spalling for gears.[18]

(3) PSO - point surface origin - fan-shaped spall propagation starting on the surface.

(4) GSC - geometric stress concentration starting at end of line contact.

A cross section of the gear surface showed developing cracks at about the depth of the large pit bottoms and crack branching that was parallel to the surface. This would indicate some repeat load occurrences were required to propagate the cracks and separate the particles from their original surface. Severe deformation, along with significant localized tangential forces, preceded the near-surface cracking, and that multiple contacts were required before the particle fractured enough to break loose from the surface. Considering individual asperity, random, repeat contacts, the developing of the surface micropits, and the larger pitting were probably related to a fatigue mechanism.

This review of the basic wear mechanisms that can occur within the tribo line contact model underlines the similarity of the operating conditions that promote damaging wear, plastic flow, or fatigue. The lack of a good lubricant film, rougher contacting surfaces, higher operating temperature, the severity of the loading, and operating speeds either too low or too high promote gear or bearing degradation. Table 2 is taken from the ANSI/AGMA 110.04-1980 National Standard, "Nomenclature of Gear Tooth Failure Modes".[18] Included are the three failure modes, Surface Fatigue, Plastic Flow, and Wear, placed in the same order as in Table 1. As to be expected, there is considerable similarity.

The greatest loading difference is the amount of tangential force that accompanies the normal force for gears in comparison to bearings. The difference this makes is that the near-surface zone with asperity interaction is probably more critical for gears, and this confounds any clear differences between polishing and moderate wear, micropitting, and initial pitting and spalling and possibly the propagation of these into more destructive failure modes. However, with the possible overlap of some of these modes in mind, next consider Table 3, which was based on our recent bearing experience and suggests what consequences this approximate guide might have for gears.

Relating Contact Fatigue Damage Model To Modified Lambda

The concept of h/σ has been popular for over 20 years as a means to assess the contact lubricant condition within either bearings and gears.[8] The λm values are most helpful within the lubricant conditions that are primarily described as mixed EHL. This occurs when asperity contact and some lubricant separation of the surfaces exist together, but in varying degrees. Appropriate lambda ratio values for this cover the range of λm from less than 0.3 to greater than 1.0. A simplified summary of this is shown in Table 3. The lubricant regime range represented by λm is a continuum, and the specific λm values picked to represent portions of the continuum are somewhat arbitrary. However, Table 3 indicates a λm range that goes from essentially complete surface separation to boundary lubrication. Also included in the table are the three prime factors for altering bearing performance, material (primarily cleanness), surface roughness, and for line contact, internal geometry.

When λm is greater than 3.0, almost complete separation of the contact surfaces occur. Under this lubricant condition fatigue damage initiates subsurface to the contact, primarily at inclusions within the material. Thus, under this condition material cleanness is important, and fatigue life can vary greatly, especially in the direction of increased life. Geometry is also important, since proper surface profiles can minimize contact stress and permit the full potential of today's clean steel to be achieved. The influence of surface roughness is minor for this condition. For line contact bearings, the fatigue mode is observed as an elliptical spall with fairly slow propagation. In gears, this fatigue mode is called surface fatigue - destructive pitting in Table 2. This pitting originates subsurface at discontinuities (i.e. non-metallic inclusion or other stress-points) near the maximum subsurface shear stress region.

Next, considering λm in the range from 3.0 to 1.0, contact fatigue mode is still predominantly material related, but depending on the traction occurring on the surface, initiation can move closer to the surface. Surface asperities and asperity slope start to play a role when λm approaches 1.0

When λm is in the middle range 1.0-0.3, fatigue damage may be mixed, with surface roughness being important. Depending on the sharpness (asperity slopes) and relative heights of the surface asperities, fatigue can be surface-related or near-surface at the higher microstress regions just below the higher asperities. Fatigue life within this near surface regime is also influenced by material cleanness and very careful damage analysis is required to assess the actual crack initiation sites in this mixed EHL regime. Thus, the recent understanding of this near surface stress region helps explain the role of clean steel in improving fatigue performance under these essentially poorer operating conditions as illustrated by Fig. 5.

For gears, because the increased sliding in the surface contact area permits higher coefficients of friction, the shear stress region occurs over a wider range and closer to the surface (as in Fig. 3) than in bearing contacts that have nearly pure rolling.[10-12] Because of this, the fatigue mode may change and "spalling" occurs as given in Table 2. According to Drago[19] and Ref. 18, spalling is a different fatigue mode for gears. "In its early stages...cracks occur and spread from the origin, in a fan-shaped manner, in the direction of sliding until a piece of material is removed from the surface."[19]

It would seem that, depending on the λm value, the fatigue modes for gears in the light of the discussion under basic wear mechanisms, destructive pitting or spalling could compete, so that damage analysis takes careful evaluation. Such care in doing this is reported by Clark et al.[20] on geared roller test rollers. They detected subsurface pitting and surface spalling for tests with λm of about 0.1 to 0.4 and with 35% sliding between the rollers.

Thus, the fatigue mode and material/surface influences given in Table 3 for the λm range of 1.0-0.3 are fairly judgmental, but have validity based on Ref. 20 damage analyses.

Finally, when λm drops below approximately 0.3, surface related micropitting tends to predominate so that surface roughness is the prime influence on fatigue. Under these low modified lambda conditions, increased surface traction and localized frictional heating can take place with increased lubricant and additive interaction with the surfaces. For gears with increased sliding the traction forces, combined with the normally rougher surfaces, seem to require EP additives in the lubricant as protection for the contacting surfaces.

There is some evidence in Ref. 21 that EP additives improve fatigue life under λ values less than 1.0 and reduce life for λ values above 1.0, in disk-on-disk tests run with 10% sliding. The tests were with and without 5 wt. percent sulphur-phosphorus EP added to the same base oil. However, in the presence of debris, increased wear has been shown with S-P/EP in lubricants both for gears[22-23] and tapered roller bearings.[24]

For line contact bearings with much lower slip between the contacting surfaces, the role of additives is unclear. However, in bearings operating with low lambda values that promote boundary lubrication, chemical reactions and higher tractions can lower fatigue life to less than is predicted by using λm value assessments alone.

In bearings, the terms peeling and micropitting are used to describe the surface-related fatigue as Table 3 indicates. For gears, the term spalling is used for surface fatigue as given above. In Ref. 18 λ seems primarily applied to scoring and wear. From Table 2, frosting is considered surface distress that resembles bearing micropitting. There are similarities between the approaches to lessening peeling-micropitting in bearings and those to alleviating the occurrence of micropitting in gears. Both forms of surface distress are reduced by a reduction in surface roughness or increasing the oil film. That micropitting could be related to fatigue was demonstrated by Macpherson and Cameron[25] in some specific sliding disk tests they conducted in the early 1970's. They called the "new form" of gear failure "fatigue scoring." The work of Cummins and Doyle[17] reinforces this conclusion that micropitting is fatigue.

There are also localized stress risers that can occur either from handling damage before running, large debris particles going through the contact, or serious misalignment that causes end of contact edge damage. Within the bearing contact, failures from these stress risers are termed point surface origin (PSO); i.e., spalling from a localized point that propagates much as the "spalling" Drago describes for gears. At the end of contact, the failure is called geometric stress concentration (GSC). These stress risers are usually generated well beyond the elastic limit for the contact material so, based on their size relative to the lubricant film, they may influence fatigue life almost independently of λm. In bearings, the defects from debris are observed as isolated, pronounced dents, while in gears the debris develop long scratches or a series of grooves and ridges from the considerable sliding present in the contacts.

A fair volume of material around these "localized" stress risers may also be highly stressed and may be influenced by material cleanness. Of course, GSC and edge stress problems can be overcome with adequate design and proper profile geometry for the application. For line contact, as in gears or tapered roller bearings, from a fatigue standpoint only, the largest debris particles are capable of life-limiting dents in bearings or deep scratches in gears.

Summary

It is possible with the tribology line contact model to consider all the factors that influence and interact within the loaded concentrated contact. Two of the important factors are the film thickness (h) and combined surface roughness (σ). Together in the modified lambda, they are an indicator of the lubricant regime operating in a gear contact or roller bearings. A corollary to this model is the stress field developed below the concentrated contact that consists of subsurface shear stress and near surface shear stresses related to the surface asperities. These shear stresses are important in the understanding of near-surface plastic deformation accumulation and near-surface and subsurface fatigue.

The three basic mechanisms that lead to metal loss in steel gears or bearings are abrasive wear, adhesive wear, and fatigue in the presence of various levels of plastic deformation. Very often in real operating contacts, all three of these mechanisms exist together, and all contribute to the nucleation and propagation of the fractures that lead to the loss of material. An aid to separating out the primary mechanism is a careful examination of the contact surface and the debris formed. Optical microscopes, scanning electron microscopes, particle size analysis, and ferrography may be required to clearly identify the actual mechanisms present.

Table 3 presents the approximate relationship of contact fatigue mode for bearings and the modified lambda ratio. Although it is based on fatigue, an understanding of the other mechanisms at work that degrade gear and bearing performance and that as relate to λm may allow Table 3 to be used in a broader sense. Our review of the various surface and near surface damage modes in gears and the development of cracks and fractures that precede the formation of debris, when compared to the development of fatigue cracks for bearings, makes the relationships in Table 3 a reasonable starting point for defining the lubricant regime in gears. ■

Acknowledgement:
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GEAR FUNDAMENTALS

Gear Inspection and Measurement

Robert Moderow,
ITW, Illinois Tools, Lincolnwood, IL

The purpose of gear inspection is to:

• Assure required accuracy and quality,

• Lower overall cost of manufacture by controlling rejects and scrap,

• Control machines and machining practices and maintain produced accuracy as machines and tools wear,

• Determine heat treat distortions to make necessary corrections.

Basic Gear Inspection Terms

Runout (Radial) - The total variation of the radial distance of a gear's teeth from its center (or bore). Runout errors contribute to gear noise, binding, and additional stress in mating gears.

Runout (Sectional) - A condition of radial runount where much of the error occurs in a few teeth and is not evenly distributed over the entire gear. This is a more severe error than evenly distributed runout.

Runout (Axial) - Also called wobble or face runout. The total variation of a gear's teeth along its axis, measured from a reference plane perpendicular to its axis. While in itself not detrimental, it is almost always accompanied by lead variation, which causes excessive stress, noise, and possible binding in mating gears.

Lead - The axial advance of a screw thread (or gear tooth) in one turn (revolution, 360°).

Lead Error - The difference between the theoretical lead trace and the actual lead trace. Usually measured from one end of the tooth to the other, normal to the theoretical lead trace. Contributes to end-of-tooth contact with the mating gear, causing possible noise, surface crushing, binding, and early failure.

Lead Average - The average of the lead error of the gears teeth. Usually based on four lead traces taken around the gear at 90° intervals.

Lead Variation - The condition in a gear where some teeth vary in lead, plus and minus, from the average lead.

Total Lead Variation - The total amount of lead variation, plus to minus. Usually based on four lead traces taken around the gear at 90° intervals.

Plus Lead, Minus Lead - Plus lead is minus helix angle. Minus lead is plus helix angle. (See Fig. 3.)

Involute (Profile) - The curved shape of the gear teeth, usually from the S.A.P. or T.I.F. to the end of the tooth.

Involute (Profile) Error - The difference between the theoretical profile and the measured profile. Can cause noise, stresses, and early failure in mating gears.

Involute (Profile) Average - The average of the profile error the gear's teeth. Usually of four profile traces taken around the gear at 90° intervals.

Involute (Profile) Variation - The total amount of profile variation, plus to minus. Usually of four profile traces taken around the gear at 90° intervals.

Total Involute (Profile) Variation - The total amount of profile variation, plus to minus. Usually of four profile traces taken around the gear at 90° intervals.

Plus Involute, Minus Involute - Plus involute is plus material at tip of gear. Minus involute is minus material at tip of gear. (See Fig. 4.)

Spacing - The measured distance between corresponding points on adjacent gear teeth.

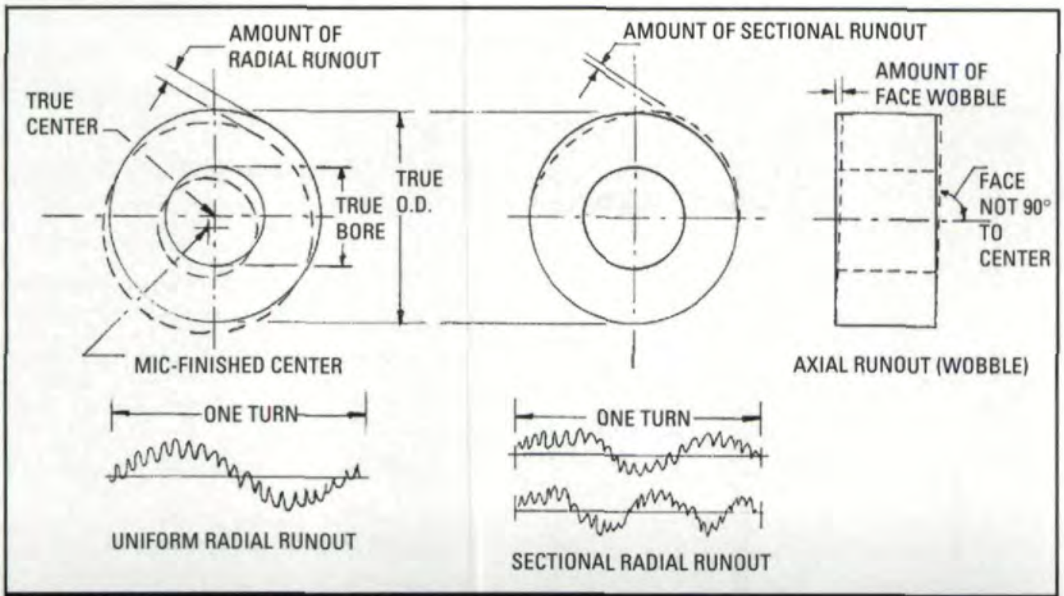

Fig. 1 - Uniform, radial, and axial runout.

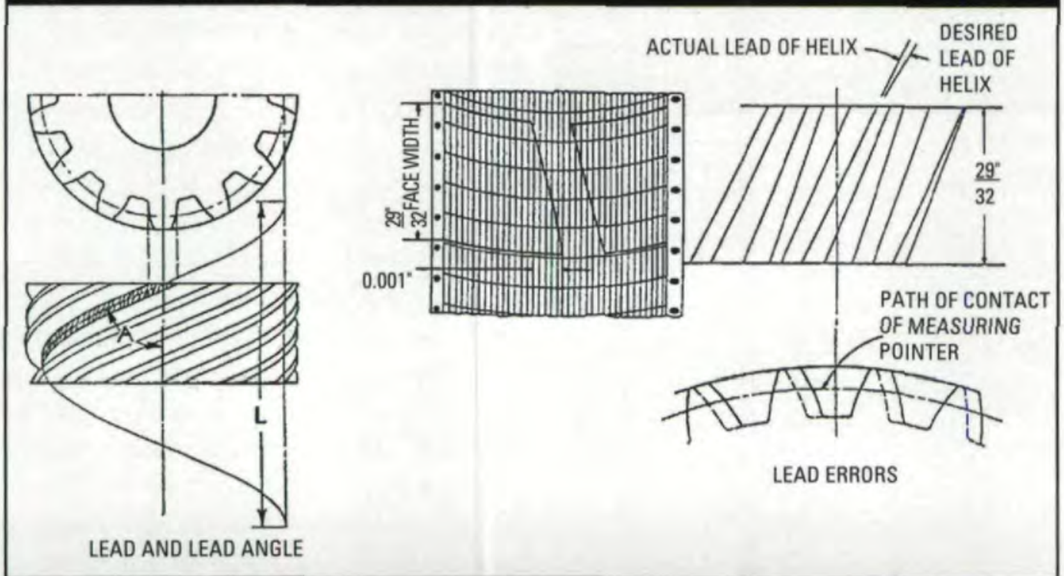

Fig. 2 - Types of lead errors.

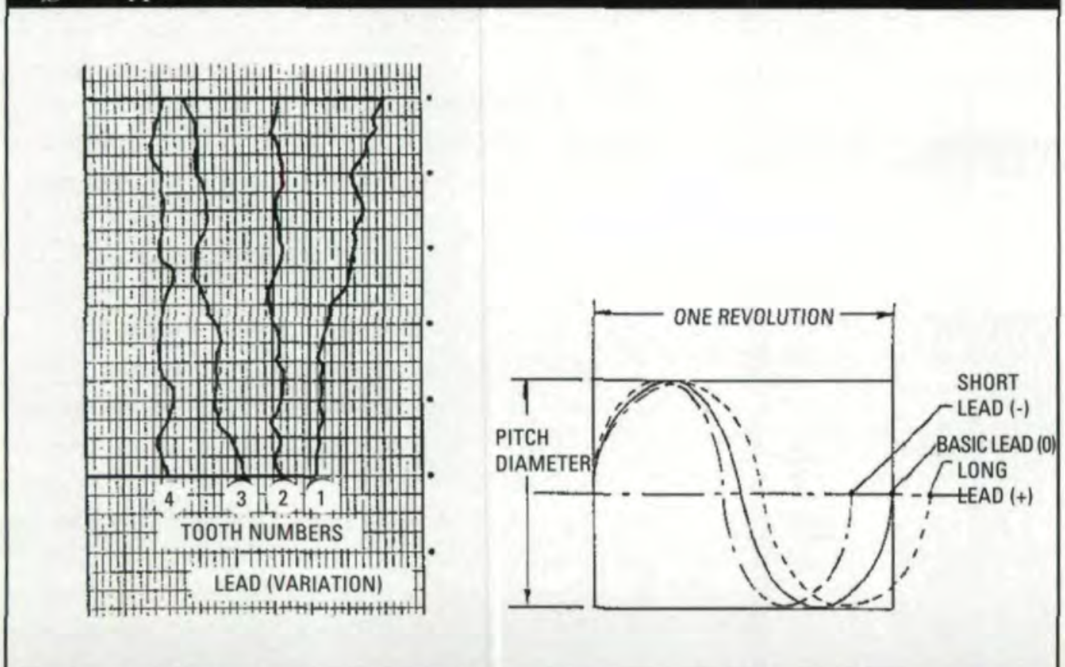

Fig. 3 - Charting lead variation caused by face runout - wobble.

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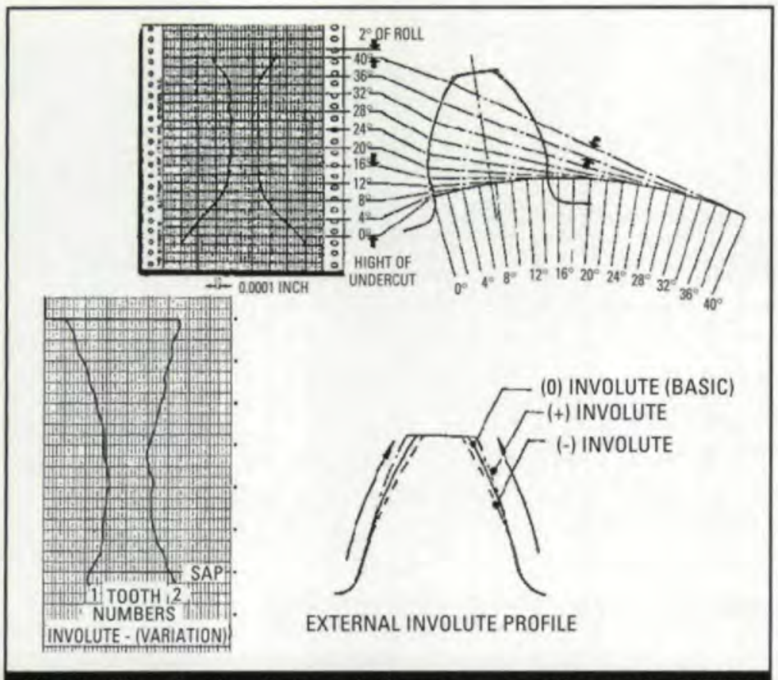

Fig. 4 - Charting involute variation on consecutive teeth; 2-THD hob cutting even number of teeth.

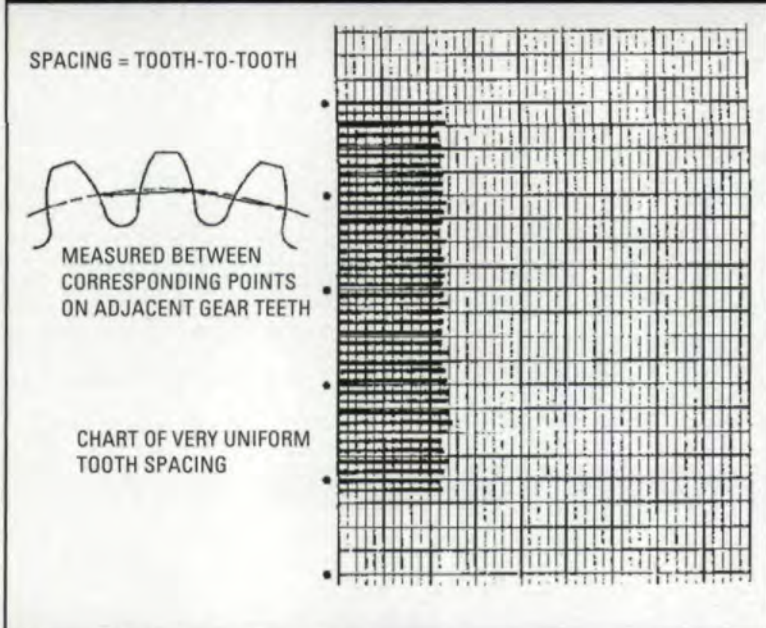

Fig. 5 - Tooth-to-tooth spacing.

Fig. 6 - Angular position of teeth.

Spacing Variation (Tooth to Tooth) - The difference between any two adjacent measurements of spacing. Can contribute to noise, stress, and early failure.

Pitch - The theoretical distance between corresponding points on adjacent teeth.

Pitch Variation (Error) - The difference between the theoretical pitch and the measured spacing.

Index - The theoretical angular position of teeth about an axis.

Index Variation (Error) - The displacement of any tooth from its theoretical position relative to a datum tooth.

Angular Velocity Error (Variable Velocity) - A tooth positioning error in a gear which was cut with runout. Subsequent operations (shaving, rolling) are unable to remove this runout, as teeth are not diametrically opposite, through the correct center. These subsequent operations tend to mask this runout, making this error difficult to find. It can be detected with a specially designed "high P. A." master gear if the gear is helical. It can also be detected with equipment that can check index error or with a "single flank" type gear roller. This type of error causes the driven gear of a gear set to speed up and slow down in one revolution, causing noise. In a planetary set, this error may cause a binding condition or will alternately carry load. This shifting load deflects the sun gear, causing noise and early failure. In an indexing gear set, this error will cause the indexed gear (driven) to vary from true position.

Nick - Actually a "rolling interference," which causes noise with the mating gear. May be a bright, shiny "nick." May be a burr left by a machining operation or a slight "plus material" condition left from a handling nick or burr, which has been "peened over" and barely noticeable.

Gear Inspection Methods

In *functional gear inspection* the gear is inspected by meshing with a known quality (master) gear. It is an attempt to inspect a gear as it is used. (See Fig. 8.)

In the *dual flank (tight mesh)* method the work gear is mounted on a fixture that allows it to rotate only. The master gear is mounted on a fixture that is free to vary the center distance with the work gear. The master gear's teeth are forced into the work gear's teeth (no backlash -

tight mesh) by pressure. An indicator senses the variations in center distance as the two gears are rolled together by hand or under power. A recorded chart can be obtained. (See Fig. 9.) *Total composite error* is the total variation of the center distance in one turn (360°) of the work gear. *Tooth-to-tooth composite error* is the variation of the center distance when the work gear is rotated through an angle of 360/N degrees, where N is the number of teeth in the work gear. (See Fig. 9.)

Terms Associated With the Dual Flank Method

Hand Roller - A dual flank gear roller where the gears are rotated by hand (no power). Usually just an indicator, rather than recording equipment is used.

Gear Charter - Similar to a hand roller, but having charting equipment and the power to drive one gear.

Red Liner - Same as a gear charter, but made by Fellows Corp.

Red Line Chart - The chart that comes from a "red liner." Also any chart showing the composite or rolling action of two gears.

Runout, nicks, and burrs can all be evaluated by this method. Functional tooth thickness can be determined by measuring the center distance between the work and the master. Involute, lead, spacing, fillet interperence, and other errors affect the tooth-to-tooth composite reading, but generally cannot be determined directly.

In the *single flank (fixed center distance) method*, the master gear and work gear are mounted on fixed center distance with backlash. (See Fig. 10.) The master gear drives the work gear. The relative angular rotation of the master and the work are compared and charted. This method can find the same errors as the dual flank method and can also detect angular velocity errors.

Analytical Gear Inspection

Analytical gear inspection is used for checking the individual elements of a gear - runout, lead, involute, tooth spacing, tooth thickness, etc.

Runout can be checked with a master gear (See previous paragraphs on functional gear inspection). It can also be checked by indicating over a ball or pin in a tooth space, progressing to the next tooth space. (See Fig. 11.) Some instruments, a tooth space comparator, for example, will inspect runout by plunging a ball into the

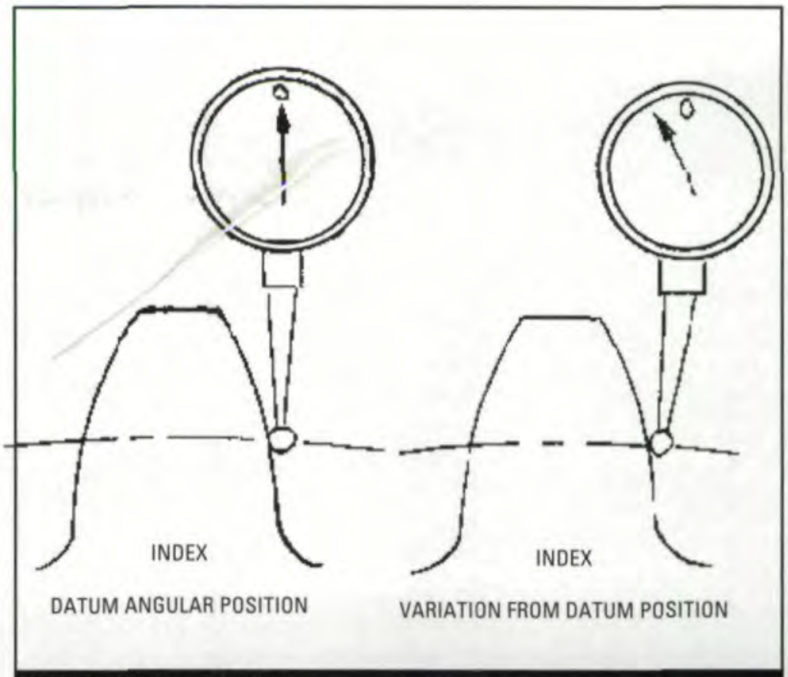

Fig. 7 - Pitch variation.

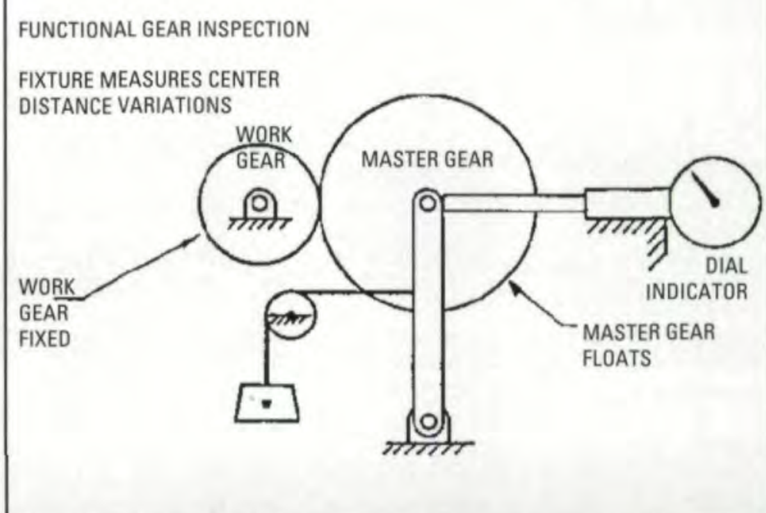

Fig. 8 - Schematic of dual flank rolling fixture.

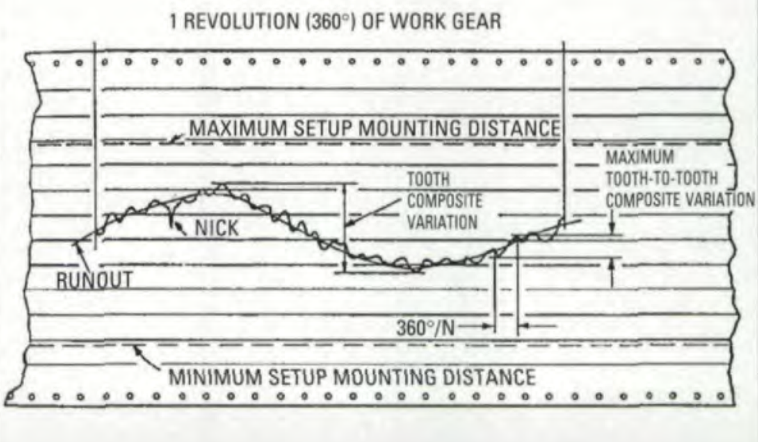

Fig. 9 - Chart of typical gear tooth errors - work gear rolled with master gear in dual flank fixture.

tooth spaces and also provide a chart.

Pitch (tooth spacing) can be check using a fixed finger and a movable (indicating) finger, measuring the distance from one tooth to another, progressing to the next pair of teeth. A tooth space comparator can do this automatically and can provide a chart.

Index can be checked by a coordinate measuring instrument with a single probe which is programmed to rotate the gear in increments of 360°/N (Number of teeth) and take readings on each tooth. This measures the true position of the gear teeth relative to the theoretical position.

Involute (profile) measuring instruments generate an involute curve. This is accomplished mechanically by rotating the work gear and moving the probe (which contacts the tooth surface) in a straight line path. It is also accomplished by coordinate measuring instruments programmed to move the probe and rotate the work gear in timed relationship. The instrument then detects any deviations from a true involute curve. A true involute will be charted as a straight line. (See Fig. 12.)

To determine a point (P) on an involute curve (tooth), we must convert a given diameter to involute measuring instrument movements. Since the instrument rotates the gear, we use degrees of rotations, (degrees of roll) to determine a given point (P). Also, the probe or contact finger moves in a straight line path so some instruments use this motion to determine a given point (P). This is call the *length of line of action* to point (P).

The chart paper on which the gear tooth involute trace is recorded is fed out of the recorder relative to either of the above two motions. When a gear's involute tolerance calls for zero to minus at the tip and zero to minus at the S.A.P., the resulting tolerance band resembes a "K"; hence, the term "K chart." Plastic overlays are often used to determine if a given involute chart is in or out of tolerance.

Lead checking instruments generate a lead path. This is accomplished mechanically by rotating the work gear and moving it along its axis in a timed relationship past a stationary indicating probe. This is also accomplished by coordinate measuring instruments programmed to move the probe and rotate the work in a timed relationship. The instrument then detects any deviations from this true lead path. (See Fig. 13.)

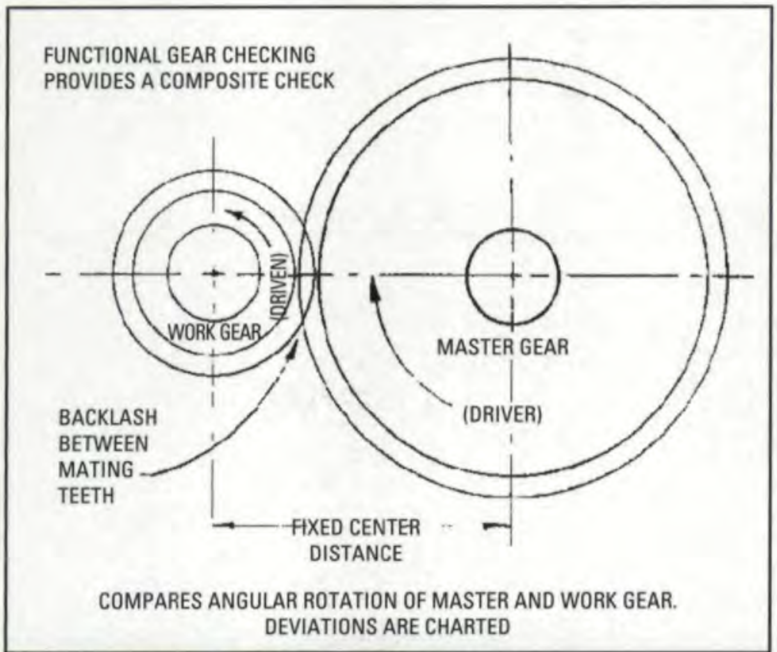

Fig. 10 - Schematic of single flank gear roller.

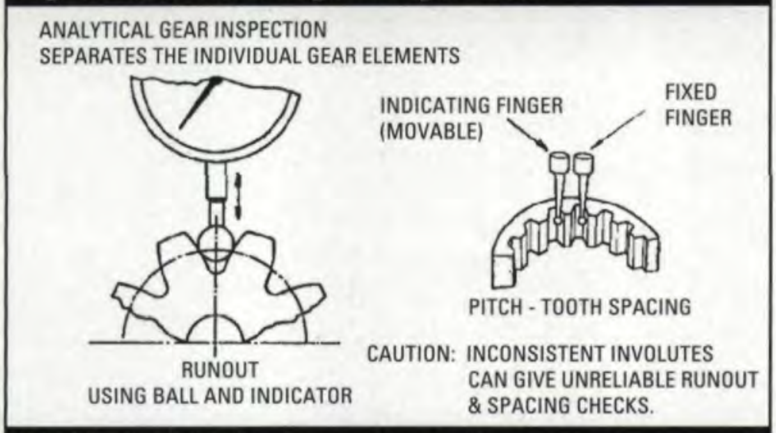

Fig. 11 - Analytical gear inspection systems.

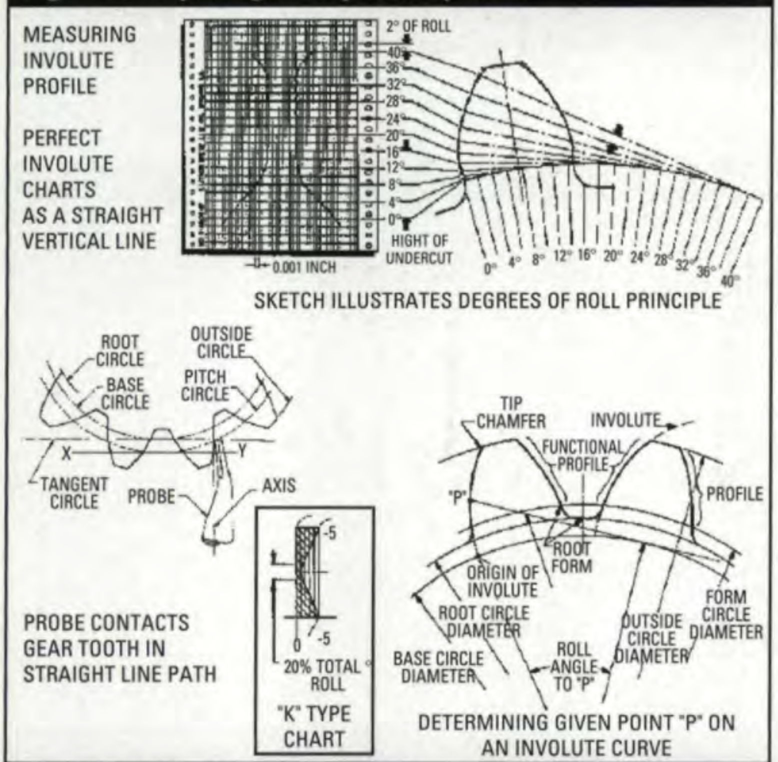

Fig. 12 - Measuring involute profile.

The chart paper on which the gear tooth lead trace is recorded is fed out of the recorder relative to the face width of the gear. To determine if a given lead chart is within a given tolerance band, plastic overlays are often used. They are especially helpful when the gear has crown, taper, etc.

A gear's tooth thickness can be checked in many ways. The most common method is to measure over balls or pins positioned in diametrically opposite tooth spaces on *external gears*. Internal *gears* are measure between balls or pins. Balls are usually used for helical gears. (See Fig. 14.)

Tooth thickness can also be measured by determining the center distance when in tight mesh (no backlash) with a master gear. Tooth thickness determined by a master gear is usually called "functional tooth thickness," as it will include other variables, such as lead, involute, spacing, and runout errors.

All of the analytical and functional inspection measurement (lead, involute, spacing, tooth thickness, runout, composite error, etc.) can now be analyzed by computers. It is no longer necessary for an inspector to judge if the part is in tolerance or not, as the computer will judge. Data storage is also accomplished so that the history of a given part can be stored for years and recalled at moment's notice.■

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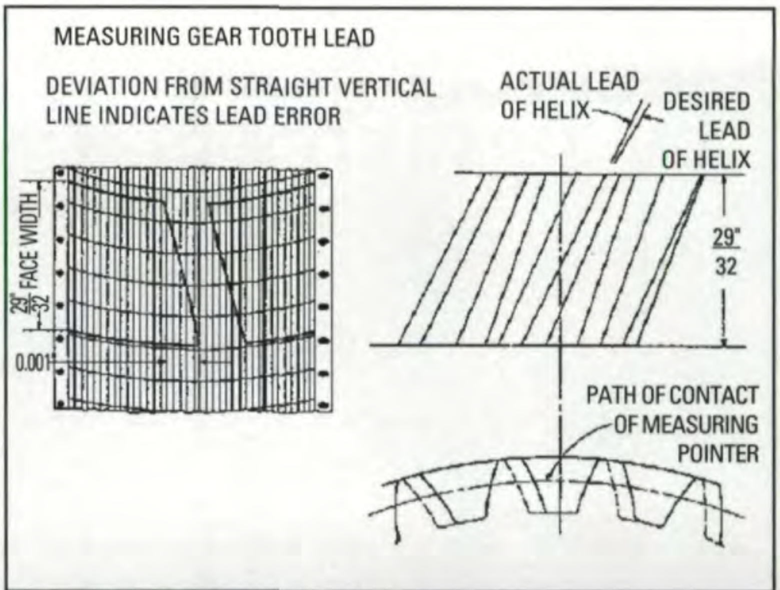

Fig. 13 - Measuring gear tooth lead.

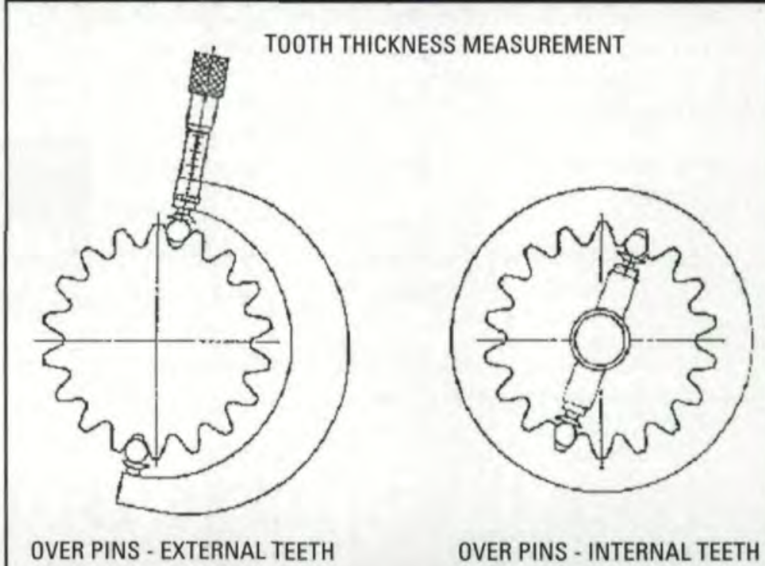

Fig. 14 - Tooth thickness measurement.

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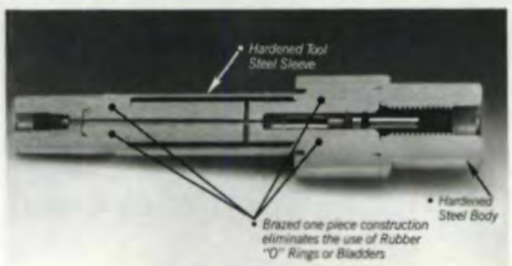

CIRCLE A-19 on READER REPLY CARD

Cutting Low-Pitch-Angle Bevel Gears; Worm Gears & The Oil Entry Gap

Robert E. Smith
William L. Janninck

Quuestion: Do machines exist that are capable of cutting bevel gear teeth on a gear of the following specifications: 14 teeth, 1" circular pitch, 14.5° pressure angle, 4° pitch cone angle, 27.5" cone distance, and a 2.5" face width?

Bob Smith replies: Machines made by The Gleason Works in years past could cut such gears, except for the very low pitch angle. These machines were known as "planers" and "planing generators." Some of these machines are still being used today by gear jobbers.

The planers used an involute former and a single point tool to generate the conjugate tooth profiles. The planing generators also used a single point tool, but had a relative motion between cradle and work spindle that generated the tooth shape. The most common planers were the 37, 54, 77, 120, 144, and 192. These numbers represented the approximate workpiece diameter capacity of the machines. The most common planing generators were the 60, 70, and 90. Again, these numbers represented the approximate workpiece diameter capacity.

Investigation shows that these machines were typically able to get down to a pitch cone angle of approximately 7°. But the questioner requires an angle 3° smaller.

He has resorted to milling the teeth and has asked for guidance on how to select cutter sizes and machine settings in order to achieve a quality result.

For many years, *Machinery's Handbook,* available from Industrial Press, has had a section on the use of Brown & Sharpe form cutters for the milling of bevel gear teeth. It also has a lengthy discourse on how to calculate machine settings. However, this method is used primarily for replacement gearing when no other proper method is available. The method creates a compromise tooth form that is not very conjugate. The result may very well be noise and vibration that is intolerable.

A solution to the questioner's problem might be the use of a gear type that has been known for years, but not widely used. H. E. Merrit, in the third edition of his book *Gears,* published by Pitman & Sons, Ltd., and others referred to these as conical involute gears. A more recent AGMA technical paper by L. Smith (AGMA paper 89-FTM-10. *Gear Technology,* Vol. 7, No. 6, Nov/Dec, 1990.) refers to them as "taper gears." The taper gears have an involute helicoid tooth form that is generated from a cylindrical surface, the base cylinder. All sections normal to the axis have a common base circle diameter and, therefore, the same involutes. However, the tooth thickness at

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Address your gearing questions to our panel of experts. Write to them care of Shop Floor, Gear Technology, P. O. Box 1426, Elk Grove Village, IL 60009, or call our editorial staff at (708) 437-6604.

Robert E. Smith
is the principal in R. E. Smith & Co., Inc., gear consultants in Rochester, NY. He has over 40 years' experience in gearing methods, manufacture, metrology, and research.

William L. Janninck
does gear and tool design and consulting. He has been inolved with gears and gear manufacturing for 45 years, 40 of them with Illinois Tools-ITW, Inc. He is the author of numerous articles on gear-related topics.

any diameter increases linearly from the front face to the back face. (See Figs. 1 and 2.) The gear has the appearance of a bevel gear, but each transverse section represents a spur gear. (See A. Beam, "Beveloid Gearing," *Machine Design*, Dec., 1954.) This tooth form will produce conjugate gears and can be made with very little transmission error. This type of gear has the additional benefit of being insensitive to mounting errors, as are spur gears.

Calculations for this tooth form can by found in *Involutometry and Trigonometry* by W. F. Vogel and in several technical papers by Mitome, ASME, and JSME.

The taper gear can be produced by any rack type tool generator or hobbing machine which has a means of tilting the work axis and/or coordinating simultaneous transverse and infeed motions. (See Smith.) In the past, mechanical machines of this type have been rare. This is the main reason that they have not come into common use. However, any CNC hobbing machine in common use today should be able to produce taper gears. CNC abrasive hobbing machines can also be used to grind them.

To address questions to Mr. Smith, circle Reader Service No. 78.

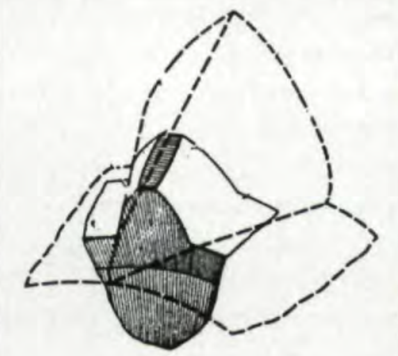

Fig. 1 - Spur taper gear tooth. Complex helicoid

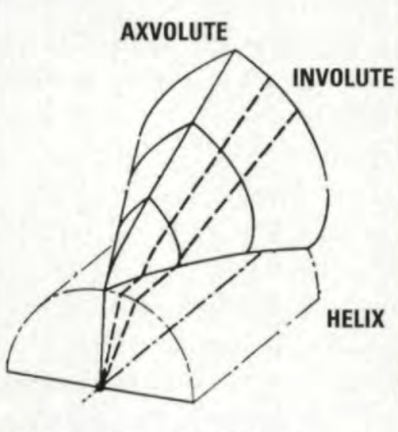

Fig. 2 - Taper gear tooth

Question: What is meant by the term "oil entry gap," when referring to the mesh condition of a worm and worm wheel?

Bill Janninck replies: One of the requirements frequently specified, among the other geometric and dimensional values and tolerances for a worm gear set, is the one defining the area of contact. The driven flank of the gear is examined for the location and percentage of area of contact shown by the transfer of a color-

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Fig. 3 - Ideal contact.

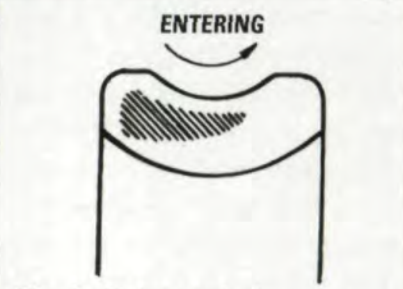

Fig. 4 - Entry contact.

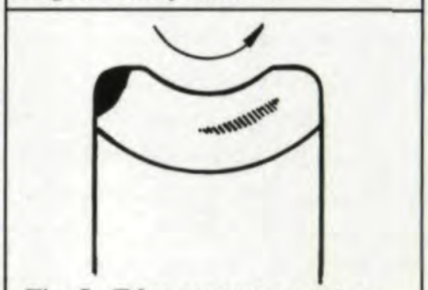

Fig. 5 - Edge contact on entry.

ing medium from worm to gear while in rotation. The contact pattern can generally be described as high or low, leaving or entering, or edge, this last pattern being sure cause for concern or rejection. For some sets the contact patterns are checked at both low load and in the anticipated operating range to allow for possible deflections during use.

Since worm gear sets are classified as high sliding gearing, because the worm flank constantly slides across the gear teeth, lubrication is extremely important. The contact pattern is usually specified to be biased toward the leaving or outgoing side of the gear teeth. This is the ideal contact pattern and is shown in Fig. 3. That area of the gear flank which does not show contact is obviously in clearance, and as one proceeds across the gear face from entry edge toward the contact area, clearance diminishes. This tapering space forms a wedge or oil entry gap. It allows the oil to migrate or sweep into the mesh zone.

If the area of contact occurs near to the entry side, as is shown in Fig. 4, the oil film can be reduced or thinned out and may even be broken, possibly causing overheating and failure of the set. If a hard edge contact occurs, as is shown in Fig. 5, the oil can be substantially diverted from the mesh, and the oil film broken. Edge contact is immediate cause for rejection.

Question: How is this gap produced on the gear? Can it be produced with a standard hob? How can it be produced by the flycutting method?

The clearance gap is produced on the worm gear flank by the amount of oversize designed into the hob. The more the oversize, the more the clearance and the smaller the contact pattern. Standard worm gear hobs generally have ample oversize built in to form some gap clearance, but if a

special specification is given on the contact pattern, the hob design might have to be reviewed. Actually hob oversize serves two purposes. It also permits some resharpening life when the hob becomes dulled. Every time a hob gets sharpened the diameter is reduced along with the oversize of the hob, and as the oversize is reduced, the contact pattern will change, slightly increasing in width. On a single start hob almost the entire hob can be consumed, resharpening gradually until the hob diameter is reduced to standard and no oversize remains. If the hob goes undersize, it is considered unusable. When the hob is new and has the largest oversize, the contact area is the smallest, and at the end of life, when the hob is a near match to the worm, the largest area of contact is seen. If there are strict limits on the contact pattern, then the hob oversize as well as sharpening life

may have to be restricted to suit.

During the worm gear cutting procedure, the contact pattern can be moved across the face of the gear by altering the hobbing machine, setting angle and observing the results, moving the contact into the ideal area.

The use of flycutting methods works in much the same manner. However, if the fly cutter is made of a radially adjustable blade inserted in a body, the oversize can be established by moving the blade out to the desired position. Once the pattern is accepted, the blade can be kept at the same oversize by resetting it after each sharpening. This gives consistent control on the contact pattern. If the flycutter is solid, then the same problems arise as experienced with conventional hobs. In any case the target oversize is usually selected by the hob designer or user based on experience and mathematical modelling. ■

To address questions to Mr. Janninck, circle Reader Service No. 79.

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Investing in Ourselves is the Key to Revitalizing American Manufacturing.

David Goodfellow

VIEWPOINT

Popular wisdom has it that manufacturing in the United States is no longer a viable entity. We are told that quality is poor, skilled labor is difficult to obtain, if not impossible, demand is low, and the government is helping to discourage business. So what should we do, give up?

That, of course, is the worst possible thing to do. We have several manufacturing models today which should encourage us to rebuild our manufacturing base; models such as Motorola, Inc., a Chicago-based company building electronic communication devices. Through a combination of aggressive marketing, commitment to quality manufacturing, employee education, and savvy investment, Motorola has been successful in becoming a world leader in its field, even in Japan. None of Motorola's techniques for success are patented. They could be implemented by other companies as well.

We also have the teachings of Dr. Deming, who is travelling tirelessly around America, trying to convince American businessmen to reorganize and restructure their companies to increase the quality of their products, the skills of their people, and to advance their competitiveness in the world marketplace today.

He - and others - have taught that TQM (Total Quality Management) is a key to success in improving the quality of our products, as well as in the reduction of production costs of those products, in order to be more price competitive in the marketplace. Quality does not cost more; it costs less.

Dr. Deming has been something of a folk hero in other countries for years. One wonders why he has had such a hard sell here at home.

Marketing strategies and quality control questions are matters under individual company control. The question of the government's role, if any, in the success of the economy is a more complicated one, and it is necessary to remember that frequently government action does more harm than good. There are, however, several areas in which government can assist business in America to become more competitive.

First, we need a revamping of the workers' compensation laws. Recent legislation and court rulings have favored the employee to the detriment of the employer. It's very easy for the employee to claim he or she is "hurt"; this translates into added costs for the employer. Second, and perhaps most important, is the revitalization of the education process in America. That will involve not only retraining existing workers, but also better educating and training those young people who will first enter the work force.

Approximately 75-80% of the jobs in industry do not require a baccalaureate degree. What they do require is a high level of competency in basic skills and adequate training in technical areas. We need to rebuild the apprenticeship programs, vocational schools, and technical training institutions to help educate those young students who are not necessarily college bound.

We also have to convince the high school students today and their parents that there are good jobs for those

David Goodfellow
is the president and CEO of American Pfauter, Limited Partnership and of Pfauter-Maag Cutting Tools, Limited Partnership, of Loves Park, IL.

who follow the path of apprenticeship or other technical programs. Industry can be a good place to work, and we need to get that message across.

Third, we need a complete restructuring of the product liability laws. These laws are costing American business millions of dollars per year for insurance rates, legal representation, and frivolous law suits - money that could be much better spent elsewhere.

Finally, American businessman must have the courage and the dedication to invest in new equipment and facilities to improve the quality and the productivity of the products they produce. America has the oldest inventory of machine tools in the industrial world today. No wonder we have trouble keeping up with our competitors from overseas.

But many seem reluctant to make the investments necessary. We hear many businesspeople say, "If only we had the investment tax credit back, we would invest." But I wonder. A few weeks ago I was in a meeting with approximately 100 industrial leaders of our community, and I asked the question, "If we now had an investment tax credit of 10%, how many of you would go out and buy new machine tools because of that investment tax credit?

Not one person raised his or her hand.

In the end, reviving America's manufacturing base is a matter of individual responsibility. If we don't do things for ourselves, no one else is going to do them for us. Businesses must have the confidence in themselves and their products and the staying power to invest in new equipment, technology, techniques, and education to stay competitive in the marketplace today.

We can rebuild our industrial base. The problems are tough, but not unsolvable. But we - both government and business - must get moving. At present we are handicapped by several circumstances that give great advantages to our competitors. We must change these circumstances and change them quickly if we are to be competitive in the world economy. ■

VIEWPOINT

Government can help by revamping workers' comp laws, by upgrading education, and by restructing product liability laws.

WHERE TO FIND THE "TOP TEN GEAR BOOKS"

Bob Errichello's "Shop Floor" column in our last issue, "The Top Ten Books For Gear Engineers," evoked a great deal of reader response. As a service to all our readers who called wanting to know where to find these books, we provide the following information:

Dudley's Gear Handbook, 2nd ed., Dennis P. Townsend, ed. and *Machinery Vibration - Measurement and Analysis* by V. Wouk are both published by McGraw-Hill, NY. The *Handbook* is priced at $75.00 and *Machinery Vibration* is $49.50. They can be ordered from the publisher by calling 800-2-MCGRAW.

Drago's *Fundamentals of Gear Design* is published by Butterworth-Heinemann of Stoneham, MA. The price is $85.00, and it can be ordered from the publisher at (800) 366-2665.

Lynwander's *Gear Drive Systems* is priced at $89.75 and can be ordered from the publisher, Marcel Dekker, NY, by calling the customer service number, (800) 228-1160.

The Maag Gear Book can be ordered from the Publication Department at AGMA Headquarters. Call (703) 684-0211. The price is $95.00.

John Colbourne's *The Geometry of Involute Gears* is $73.00 and can be purchased from the Order Dept. at Springer-Verlag, NY, by calling (212) 460-1500.

A limited number of copies of *Steel Selection - A Guide for Improving Performance and Profits* by Kern and Seuss can be purchased from Kreiger Publishing, Inc., in Melbourne, FL, by calling (407) 724-9542. The price is $54.50.

The following three books are out of print and no longer available: *Gears for Small Mechanisms* by W. O. Davis (N.A.G. Press, Ltd., London); *The Exact Over-Wire Measurement of Screws, Gears, Splines, and Worms* by W. F. Vogel (Wayne State U. Press, Detroit, MI); and *The Influence of Microstructure on the Properties of Case-Carburized Components*. (American Society for Metals, Materials Park, OH.) The best chance of locating copies of them would be through a used book dealer or book search service. ■

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For more information on the TOCCO TPH profile hardening process, contact George D. Pfaffmann, Vice President of Technology and Service Operations, TOCCO, Inc., a subsidiary of Park-Ohio Industries, Inc., 30100 Stephenson Hwy., Madison Heights, MI 48071. Or call our toll free number: 1-800-468-4932.

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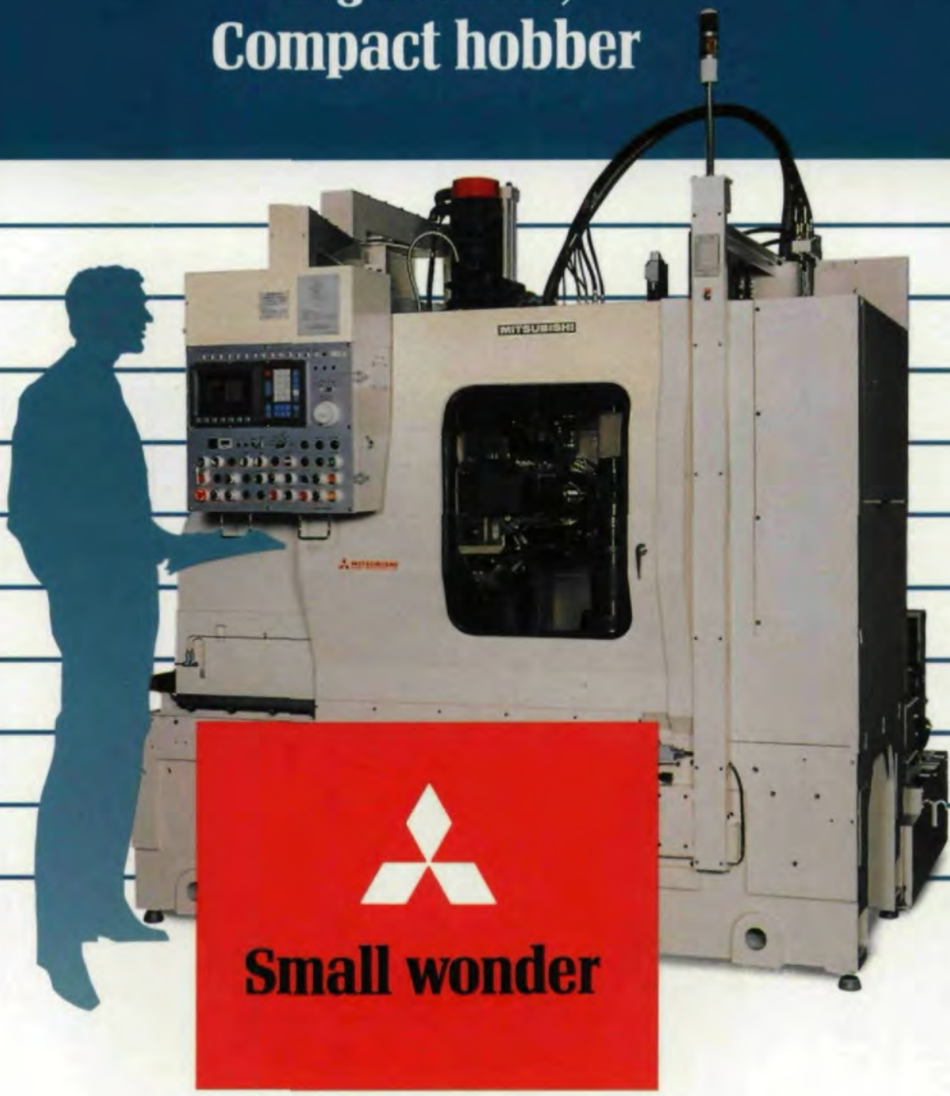

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