

IMTS 92 Show Issue

GEAR TECHNOLOGY

THE JOURNAL OF GEAR MANUFACTURING

SEPTEMBER/OCTOBER 1992



CNC TECHNOLOGY & THE MANUFACTURE OF SPIRAL BEVEL GEARS

INVESTIGATION INTO GEAR RATTLE PHENOMENA

GEAR FUNDAMENTALS - BASE PITCH TABLES

IMTS COVERAGE

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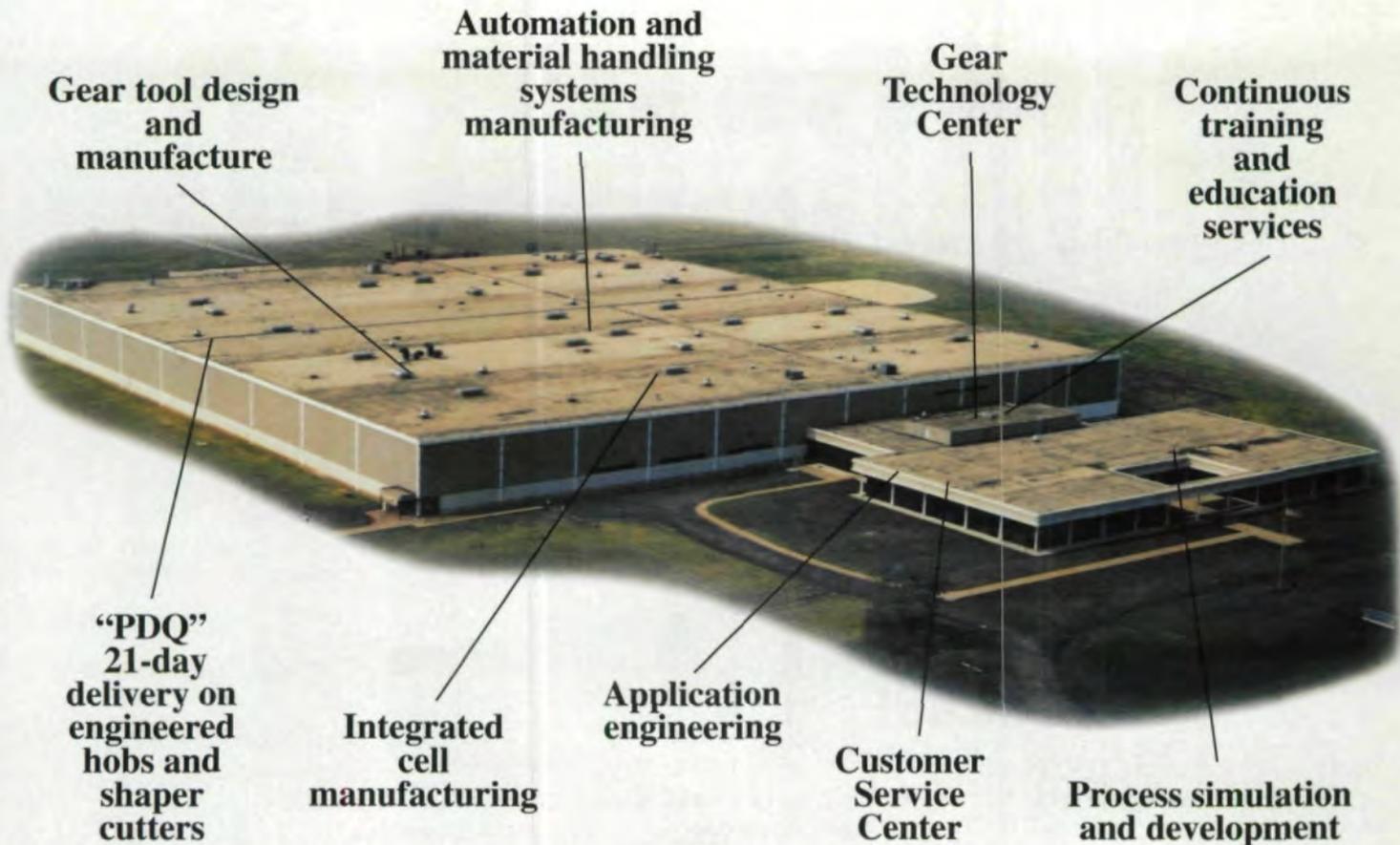
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ON THE CUTTING EDGE OF GEAR TECHNOLOGY SINCE 1896

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Do they have CNC automatic loading gear hobbers with multiple cut capability?

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Do they crown hob or single index cut?

Do they hard cut gears from the solid to the fifty rockwell range?

Does your vendor support the gear equipment with new CNC milling machines, new grinders and new cross grinding honing equipment?

Does your gear manufacturer have \$7,000

worth of extensive gear analysis software?

Does your gear house talk to you about gear blank quality?

Does your gear producer welcome his competition to see the latest state-of-the-art equipment?

Will your supplier do one piece or a million, with the last one just as the first one?

Does your gear house supply a whole lot of other gear producers?

Does your gear source periodically send their people to gear training schools and clinics?

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We believe there are many excellent gear companies, but with any supplier other than Forest City Gear, we doubt you could answer all of the above questions affirmatively.

In fact, for the strength of the American Gear Industry, we hope that many of our competitors will follow our footsteps. Of course, we don't intend to stop, so they will have to take aim at an advancing target.

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The Seeds of Great Enterprises

“Opportunity is the start of great enterprises.” said the Greek statesman Demosthenes, and what was true 2300 years ago is no less true now. Plenty of opportunities which can grow into great - and successful - enterprises are waiting for us right now if we only have the foresight to take advantage of them.

IMTS 92 is in Chicago this month. Hundreds of companies from around the world will be there, and they are eager to do business with you. This could be the opportunity for you to upgrade equipment or make important contacts that will be the source of some great future enterprise for your business.

If IMTS is not in your future, plenty of other opportunities are coming up. They will require that you plan ahead to take the greatest advantage of them, but they too are out there waiting, ready to be turned into great enterprises.

In the fall of 1993, AGMA will again sponsor its GearExpo, a trade show devoted exclusively to our industry. Its smaller, more intimate setting may be the perfect opportunity for you to buy needed equipment, learn about advances in the industry, or show your products or services to the people most eager to buy.

Consider also the educational opportunities out there for the taking. AGMA, SME, and other organizations offer regular conferences and seminars on gearing subjects. So do a number of machine tool manufacturers. The AGMA Small Business Committee is also working in conjunction with INFAC in Chicago to set up a number of training courses, specifically with the small gear shop in mind.

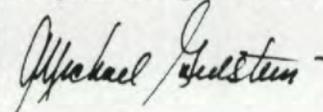
Education is the sort of opportunity that Demosthenes had in mind. In it are certainly the seeds of all great enterprises. Ask yourself whether everyone at your plant is so well trained that none of these courses would be useful to them.

In addition to the valuable information they offer, these kinds of conferences and seminars

provide you with the chance to share your expertise and build your company credibility. Every one of the technical organizations regularly issues “Calls for Papers.” They need knowledgeable gear people to present papers on a variety of gearing subjects. *Gear Technology* is also always looking for interesting, well written articles on gearing subjects. You could be the author or presenter we’re all wanting to hear.

Contacts made and opportunities opened up by becoming an “expert” on a particular gearing subject are less quantifiable than those gained from a trade show appearance, but they are just as real and also have the potential to grow into great enterprises.

Exhibiting at trade shows, taking or teaching courses, or writing papers are not easy. They all require a lot of effort on your part. But Demosthenes never said that turning opportunity into a great enterprise would be easy; he simply implies that not taking the opportunity guarantees that you won’t be part of the great enterprise at the other end.



Michael Goldstein,
Publisher/Editor-in-Chief

PUBLISHER'S PAGE



When Horsburgh & Scott discovered CPM® REX® 20...



An operator installs a CPM REX 20 spline cutter used to form internal splines in a worm gear. A tool made from CPM REX 20 was also used to cut this gear's teeth.

...gear cutting shifted into high



Tools made of CPM REX 20 are used for a variety of gear cutting operations at H&S.

Horsburgh & Scott, Cleveland, Ohio, knows what it takes to be a world class gear manufacturer...like using Crucible's CPM REX 20 steel for gear cutting tools that run longer and better between sharpenings.

Crucible CPM REX 20 not only lasts about 50% longer than M-2 or M-42, but the tools are faster to make and sharpen. Steve Lyncha, H&S tool room supervisor, said, "When you find something that grinds this easily, you think it'll never hold up, but CPM REX 20 has handled the stop-and-go action of gear cutting better than we could ever imagine."

According to Lyncha, the long-term savings far outweigh the greater initial investment. "As long as CPM REX 20 is available, you could offer me M-2 at a \$1.00 a pound, and I still wouldn't buy it," he added.

CPM REX 20 is only one of nine CPM high speed steels specified for gear cutting tooling in major automotive and other heavy industrial applications. To find out more, contact your nearest Crucible Service Center or call toll free: **1-800-PAR-XCEL** (1-800-727-9235)



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Where's the class?

Monthly courses are held at ITW Components and Tools. Regularly we take the course "on-the-road" for client location sessions. Call for information.

Who says it's worth it?

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How do I sign up or get more info?

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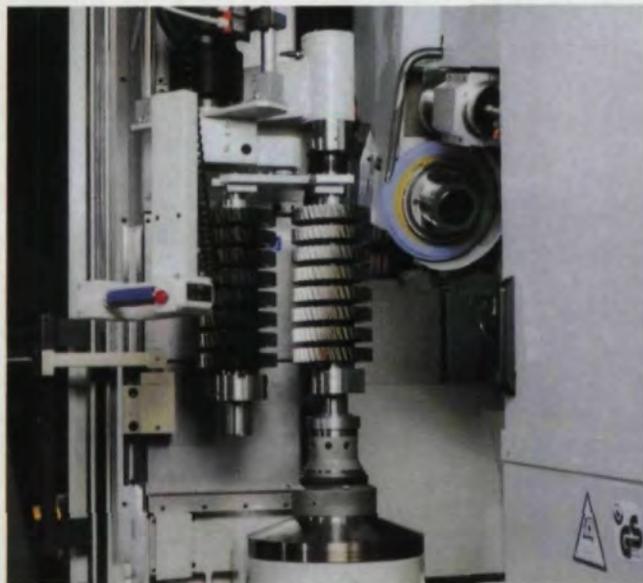
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CIRCLE A-30 on READER REPLY CARD

The Bottom Line on Trade Shows

Turning Trade Show Leads Into Sales Is The Name of The Game.

Nancy Bartels

The whole point of a trade show is to get leads that will turn into sales. No matter how attractive your booth was, no matter how smoothly the setup and the show ran, no matter how many visitors you had at your booth, if your presence at a show didn't net you any sales, then your considerable investment of time, money, and effort has been wasted.

But turning trade show leads into sales does not happen by magic. Like the other parts of a successful trade show, this transformation is the result of planning and hard work.

Getting Them In

Step 1, of course, is getting people to your booth, where you can qualify them and determine if they are valid prospects. Every trade show is full of noise, color, and activity, and just getting a visitor to notice your booth can be a challenge.

The key words here are comfortable and inviting. People have to want to stop

at your booth. It shouldn't be too crowded or cluttered. Avoid graphics, furniture, and colors that will cause visual overload. One new machine, attractively displayed, will be more effective than several similar models all jammed together.

You want to be different enough to stand out from the crowd, but beware of novelty for its own sake. You may attract a lot of lookers with a magician or a girl in a bathing suit, but will any of them stop to discuss your product after the dog and pony show is over?

Your booth should also be neat and clean, of course, and your people should stand out. Consider having them wear distinctive hats, sports jackets, or ribbons that will identify them as the people with the information about your products. Remember you want to make it easy for the visitor to ask questions and get the information he or she wants.



MANAGEMENT MATTERS

At the same time, avoid a booth that is too perfect or too comfortable. John T. Lawrence, of Design Origins, Inc., says, "I've seen booths that are so neat, pristine, and pretty that no one wants to walk into them. They're beautiful. People walk by and say, 'God, that's gorgeous,' but no one stops. They're afraid to mess up the white rug or whatever."

Lawrence also warns against having a lot of sofas or comfortable chairs. "You don't want people to use your booth as a rest area." A table and upright chairs that say, "Let's talk business," are more appropriate.

Are You Qualified?

Once you have people

Managing a business today is hard work. Let "Management Matters" lend a hand. Tell us what management matters interest you. Write to us at P.O. Box 1426, Elk Grove, IL, 60009, or call our staff at (708) 437-6604.

Nancy Bartels

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

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CIRCLE A-13 on READER REPLY CARD

in your booth, your most important task is to separate the serious prospects from the tire kickers. A stack of business cards from booth visitors by itself will be next to useless in terms of generating sales. These names have to be evaluated - or qualified - for their sales potential.

Qualifying leads is a process that should begin before the show even opens. Your salespeople need to have the proper qualifying forms, and they should be trained before the show to fill them out properly.

Your qualifying form should answer the following questions: Is the prospect using your product or a similar one? Is he or she considering purchasing one? Can he or she influence the decision to buy? When is the purchase planned? What is the budget for the purchase? You may also wish to add questions that apply uniquely to your product, but these five questions should be the basis for your form.

After you have developed your qualifying form, you will need to establish some criteria for evaluating the information you receive. What will constitute an "A" lead - a really hot prospect? What will make a "D" lead - probably not really interested? A chief decision-maker with a half-million dollar budget already approved and looking to buy in the next six months is obviously a better prospect than a someone with no

influence on purchasing at a company with no budget for acquisitions in the coming year. A clear set of criteria for evaluating leads understood by everyone working your booth and back in the home office is necessary to make the most of your trade show effort.

But your planning shouldn't stop with the qualifying form. *Absolutely crucial to your success in turning prospects into sales is what happens to these leads - both at the show and afterwards - once they have been qualified.*

At the show, your sales staff should be instructed to fill out a form every time. Too much goes on at a trade show to trust anything to memory. A system should be developed so that properly qualified leads get into the right hands as soon as possible. You don't want good leads to end up languishing in a coat pocket or briefcase. Don't wait to bring all the leads back from the show with you. Get them to the home office or to your literature fulfillment house right away.

Literature & Freebies

The question of what to give away and how much to give are open to discussion. The issue should always be looked at in the context of the basic purpose of your goal at a trade show - to qualify potential customers. If freebies help, then they're worth the trouble and expense; if not, trouble and expense are all they are.

John Lawrence puts the freebie question this way: "The only thing more important than getting your name out is getting the prospect's name, address, and business card. If the booth visitor isn't really a prospect, then you don't care so much about that, but, in that case, it's sure nice if they took a notepad or ruler with your name on it back to the office."

Lawrence recommends personalizing literature. Write something - a quote, a phone number. He says, "Rather than just handing out literature, you should be discussing the brochure with the prospect. Make a note of

ris RF Communications Group in Rochester, NY, has a different view of the literature question: "I don't recommend handing out literature in the booth at all. It's like flushing it down the toiletProspects leave it behind or throw it out because they don't want to carry it."

Instead Cicotta recommends setting up a system to mail literature to prospects during the show. He suggests either having a local fulfillment house do the mailing or have the leads sent to your home office each day. Literature packets for each of your products should be prepared

MANAGEMENT MATTERS

An industrial sales call costs \$250, whether it is made on a hot prospect or someone with no interest. A trade show contact costs only half that.

some kind and give it to them. That makes it more valuable than just a printed piece. People will say, 'I'm going to throw this away - oops! - I gotta keep this one.'"

John Cicotta, an experienced exhibit manager, formerly with Gleason Works and now with Har-

Richard Erschik, of ahead of time and mailed to the prospect the day after he or she visits your booth. "Think how impressed the prospect will be when he returns from the show and finds the literature he requested waiting on his desk," says Cicotta.

Richard Erschik, of

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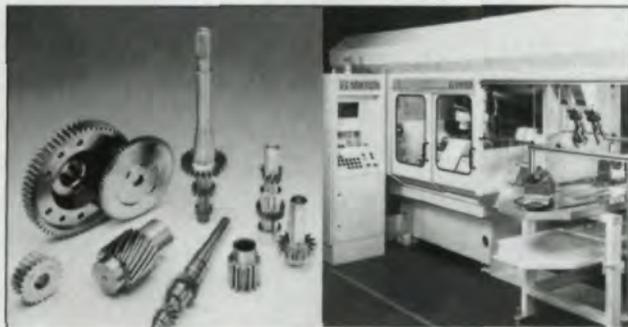
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CIRCLE A-17 on READER REPLY CARD

Leads to Sales, a marketing consulting firm in Carol Stream, IL, goes one step farther. He recommends: "Send your literature only to qualified people. As a courtesy, you should send something to everyone who stopped at the booth, but save your big, glitzy, expensive catalog or product kit for qualified prospects."

After the Ball is Over

Once the show is over, your work with your prospects is only beginning. Follow-up is the key to turning leads into sales, and it is here that many companies negate all the hard work and effort they have put into making a successful trade show appearance. According to a survey run in *New Equipment Digest*, less than 20% of the leads brought home from trade shows are ever followed up!

Why? Not because sales forces are lazy or inefficient, but because following up on leads from trade shows tends to be the lowest priority for the sales team.

Here's why. Consider that today it takes approximately 12 tries to get through to a person on the phone. You have to deal with the voice mail, leave messages, get busy signals, miss a couple of returned calls, and then, maybe, if you're lucky, you will actually make contact with the person to whom you wanted to speak - only to get a reply like, "XYZ Gadgets? I left a

card at your booth? Which show? When? No, I'm not interested in your new line of computerized portable gadgets. There's no money in the budget."

Not even your most aggressive and persistent salesperson will be willing to spend a great deal of time in conversations like that, especially when he or she can be having much more pleasant - and productive - ones servicing current customers.

The solution? Requalify all those names you got at the show. It sounds like a lot of work, and it is, but in the end, while you will have fewer leads to turn over to your sales department, they will be better leads, much more likely to turn into sales. And, after all, it's results, not total number leads, that are important.

The requalification should be done by someone from your marketing department. The phone call to requalify should be a conversational one whose aim is not to sell products, but to discover the level of interest in buying your product. It should answer the same five basic questions you used to qualify prospects at the show: Present use of product; consideration of use; decision-maker status; date of purchase; budget. Only the prospects that survive this second cut are the ones that should be sent to your sales department.

This is a system it takes

time to implement. Begin by requalifying only the people who were "A"s and "B"s in your show qualification. Others can be sent literature or put on a mailing list. Expect that the process will take two or three months. Also expect that it will take your sales force a while to get used to the fact that the leads now coming from trade shows are truly valuable potential customers.

Realize that this system will work much better if you put it on a computer data base. This is valuable information you can use not just to generate sales, but also in your other advertising and marketing

turn into solid numbers on your sales chart.

The trade show route to sales is expensive and a lot of work. If you find yourself wondering if it's really worthwhile, if the money and time might not be better spent some other way, consider one set of numbers. The cost of the average industrial sales call in the U.S. is \$250.00 - whether the call is made on a hot prospect ready to sign a contract or on someone who "might be interested some day." According to the National Association of Exposition Managers, the cost of a contact generated at a trade show is nearly 50%

MANAGEMENT MATTERS

Less than 20% of leads brought home from trade shows are ever followed up. One reason is because the leads have never been properly qualified.

plans. You're going through a lot of effort to get it. Be sure to make the most of it.

Once you get this system in place, either using your own staff or an outside fulfillment service, you will find that the prospects gleaned from a successful trade show do

less than that.

So in spite of the complexity of planning and implementing a successful trade show strategy, it is still one of the most cost-effective means available to get your message to interested prospects. And that, of course, is the point. ■

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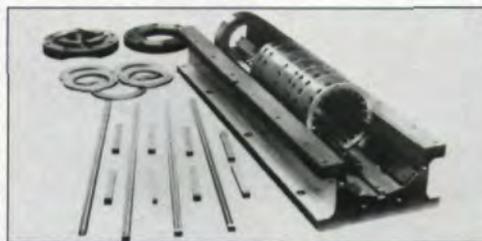
In sprocket tooth broaching, two types of tools are primarily used: a stick-type (surface broach) and a ring-type tool. The main pot is a two-piece split holder that has four ground locating bars for supporting a series of precision ground, broach-holding bushings.

These bushings have internal slots that hold precision, high-speed steel broach sticks. The sticks can have from as little as one, to as many as three splines each, to rough the part sprocket teeth by generating the form. The broach sticks are assembled in a staggered formation with many teeth engaged at once for a smooth, quiet cutting action. These sticks can be easily removed for simple sharpening operations.

A series of individual precision-manufactured, high-speed steel annular rings or wafers are assembled behind the sticks and holder bushings in the main pot. These rings shave the entire form, i.e. the involute flanks, the full fillet root radius tangent to these flanks, and the root diameter of the parts.

Finish pot broaching of sprockets is a significant improvement over the present accepted process. It takes ten (10) machines to duplicate the output of one (1) pot broach machine. This translates into direct cost savings due to reductions in floor space, material handling, spare parts, and manpower. The overall result is less variation within the

process and a net gain in quality with considerable cost savings.



Red Ring stick- and wafer-type pot broaching tools disassembled to show construction.

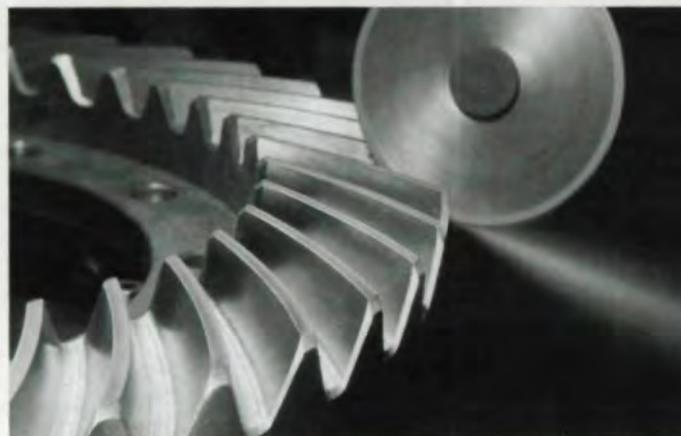
Presently, a Canadian manufacturer is producing external sprocket-toothed parts using a RED RING 20-ton, 56-inch stroke, push-up pot broach machine and tools. The parts have 39 teeth, a 7.2 diametral pitch, 31.5 degree pressure angle, and a 5.4 inch pitch diameter. The machine is set up to produce 150 parts per hour.

Broaching, especially gear and spline tooth broaching, has been a large part of National Broach & Machine Co.'s business since its founding 63 years ago. National Broach first successfully pot broached an external gear in 1958. Since then, RED RING pot broaching machines and tools have been continually improved to a point where today, a variety of precision external configurations and tooth forms are produced daily using this high production, economical method.

For more information, contact your RED RING sales representative at:

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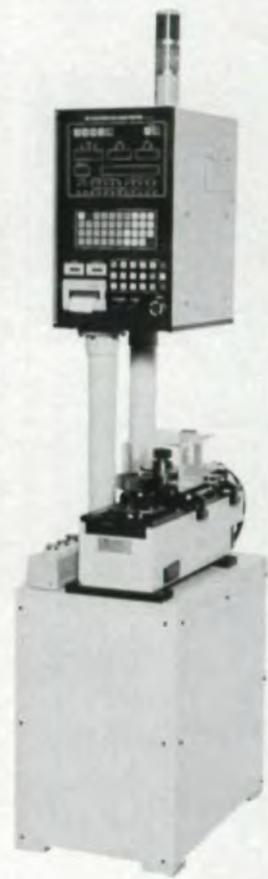
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2	189.640	031	010
3	189.635"	032	009
4	189.633"	033	010
5	189.638	033	010
6	189.633"	032	009
7	189.635"	036	010
8	189.636"	030	009
9	189.635"	031	009
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8 *Gear Technology* advertisers will have booths at IMTS 92, the largest trade show in the Western Hemisphere. The show opens in Chicago on Sept. 9 and runs through the 17th. More than 800 companies from around the world will cover some 931,000 sq. ft. of exhibit space to show the latest manufacturing technology - everything from forming and fabrication products to environmental and plant safety equipment.

Of particular interest to *Gear Technology* readers will be the gear cutting machinery and cutting tools.

The *Gear Technology* advertisers will be showing a wide variety of latest equipment and processes at their booths.

American Pfauter Limited Partnership of Loves Park, IL (Booth 6337) will be demonstrating a host of new products, including new CNC gear shaping and honing machines, CNC profile grinders, and new hobs, shaper cutters, and other tooling.

Balzers Tool Coating, Inc., of N. Tonawanda, NY, (Booth 8612) will have TiN and TiCN-coated hobs and shaper cutters on display, along with a variety of PVD-coated cutting, forming, and molding tools. An engineer will be on duty throughout the show to answer coating questions and review the latest application results on coated tools.

Koepfer America, Inc. of South Elgin, IL, (Booth 6681) a supplier of hobbing machines and cutting tools, will show a Koepfer 153B high-speed production hobber with automatic loading and an automatic skiving system for recutting hardened work pieces. The booth will also feature a Koepfer 200 CNC high-speed hobber with quick-change capability. Demonstrations will show complete part-to-part changeover in less than five minutes. Also on display will be a Haas Model HB 1035 CNC hob and tool sharpener with CNC setup and flexible sharpening for straight and helical gash hobs and tools.

Kanzaki (Booth 7335) produces complete CNC gear shaving machines that offer con-

ventional, diagonal, plunge, and underpass shaving capabilities. Kanzaki offers full, 5-axis CNC hard gear finishing systems, incorporating tool wear compensation.

Liebherr Machine Tool, a Division of Liebherr-America, Inc., Saline, MI (Booth 6466) has responsibility for sales and service of Liebherr CNC gear cutting machines and material handling equipment and Lorenz CNC gear shaping machines and gear cutting tools. One of the company's LC 252 CNC hobbing machines and a Lorenz LS 154 CNC shaping machine equipped with a flexible automatic load/unload system will be on display at the booth, along with a full range of Lorenz cutting tools.

Mikron Corporation of Elgin, IL, (Booth 4300) offers a wide range of CNC gear machines for the hobbing and shaving of gears within the range of 1/2" to 21". Mikron's "A" series machines (up to 8" dia. and 27" long) do high precision finish hobbing with a horizontal work axis; the "C" series does vertical workpiece hobbing through 21" diameter; and the "S" series machines do shaving with the conventional, diagonal, underpass, and plunge methods. Complete tooling is available as well.

MHI Machine Tool U.S.A., Inc., of Itasca, IL, (Booth 4197), in a departure from displaying small capacity machines, will demonstrate a 16" capacity hobber and an 18" capacity shaving machine. The GC40 is a 5-axis, 400 mm (15.7"), 8 module (3.2 DP) hobbing machine with fully conversational programming. The FA45 is a 5-axis, 450 mm (17.7"), 10 module (2.5 DP) state-of-the-art shaving machine that is also fully conversational in its programming.

Pfauter-Maag Cutting Tools, L.P., Loves Park, IL, (Booth 6337) offers a full range of gear cutting and finishing tools. The original Wafer Shaper Cutter and High Speed Wafer Hob will be demonstrated on new model machines by Pfauter. ■

The photograph for this article was shot by Richard Goldstein.

IMTS EXHIBITOR INDEX

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CNC Technology and the System-Independent Manufacture of Spiral Bevel Gears

Dieter Wiener
Klingelnberg Söhne,
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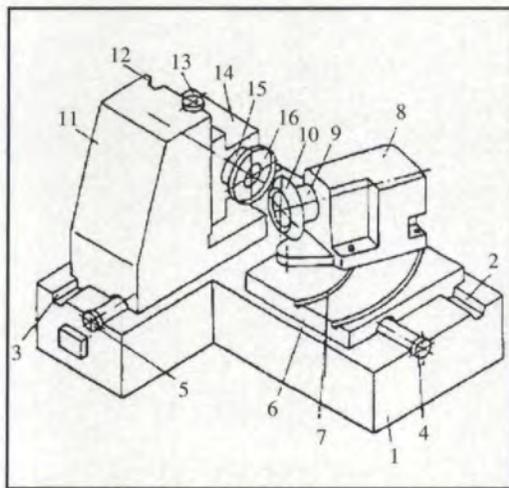


Fig. 1 - Russian machine with X-Y cradle.

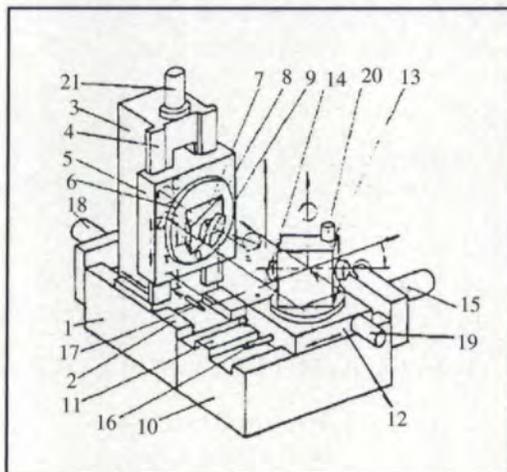


Fig. 2 - Modul machine with CNC-controlled X-Y cradle.

Introduction

CNC technology offers new opportunities for the manufacture of bevel gears. While traditionally the purchase of a specific machine at the same time determined a particular production system, CNC technology permits the processing of bevel gears using a wide variety of methods. The ideological dispute between "tapered tooth or parallel depth tooth" and "single indexing or continuous indexing" no longer leads to an irreversible fundamental decision. The systems have instead become penetrable, and with existing CNC machines, it is possible to select this or that system according to factual considerations at a later date.

This article, by giving a brief overview of the machines and systems available on the market or currently at the development stage, discusses the possibility of using different types of gear cutting on the same machine. It also contrasts the characteristics of different gear cutting systems, and, finally, discusses the possibilities of gear cutting optimization offered by current calculation and manufacturing technology in connection with CNC measurement technology.

CNC Machine Systems

Former Soviet Union system - The first machine known to us with a cross slide working at the workpiece side is shown in Fig. 1. It has

the Russian Patent Application No.SU 724487 A, dated March 30, 1980. On the figure's left side, the cross slide with its ways "3" and "12" describes the circular motion of the tool. The actuation of the cross slide is a mechanical-hydraulic system.

Modul system - A CNC-controlled bevel gear cutting machine with cross slide is mentioned in Patent Application No. 255 296 A 1 of the company Modul in the former German Democratic Republic. The principle of this machine is shown in Fig. 2. The generation motion of the tool is achieved by overlaying slides "F" and "E." This machine is also equipped with a computer controlled tilt axis and swivel axis. These are the axes "D" and "C."

If the workpiece is cut with "tilt," the C-axis has been moved in a certain relation to the axes "F" and "E" (continuous path controlled), while the tilt-axis D is only a fixed setting axis.

Japanese system - In the Japanese Application for Patent No. 3643967 (Fig. 3) a machine is introduced in which all the motions are determined by the CNC control as well. As with the previously mentioned machines, the generating motion is achieved by a cross slide. The X-axis and the Y-axis are used to achieve the generating motion of the tool. As in the case of the Russian machine, no tilt and no swivel is provided.

Grinding Machine - Fig. 4 shows a full CNC-controlled bevel gear grinding machine, WNC 80, made by Klingelnberg. The generation motion of the tool, in this case the grinding wheels, is performed by the Y-axis (cradle motion). The advantage of this system is the fact that only three axes are in motion during the generating process: the Z-axis (workpiece rotation), the Y-axis (workhead), and the X-axis (helical motion). By intelligent superimposition of these three axes, the tilt setting is unnecessary. The double spindle wheelhead allows the grinding of pinions, which were cut by the traditional five-cut method at the concave and the convex flanks separately in one setting.

Phoenix - On the Phoenix machine (Fig. 5) made by Gleason, the generating motion of the tool is done, as with the first three machines, by the cross slide, visible in the left side of the picture. The ways 16 and 20 are the machine ways for the cross slide. Therefore, the cross slide motion is performed by the tool in the direction opposite to the previously described

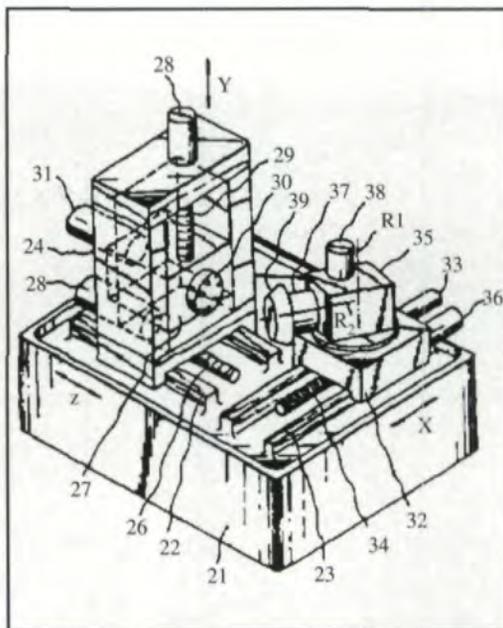


Fig. 3 - Japanese CNC machine.

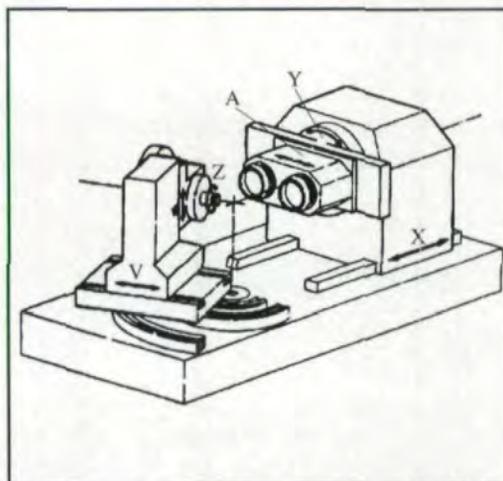


Fig. 4 - CNC bevel gear grinding machine, Klingelnberg WNC 80.

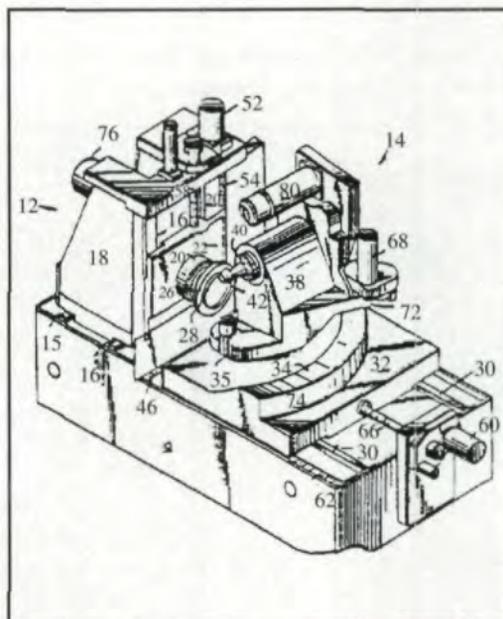


Fig. 5 - CNC bevel gear processing machine, Gleason Phoenix.

Dr. Dieter Wiener

is the executive vice president of the Remschied Gearing Division of Klingelnberg Söhne. He has also worked for Hofler Company and his own gearing company. He received his education at the Technical University of Aachen.

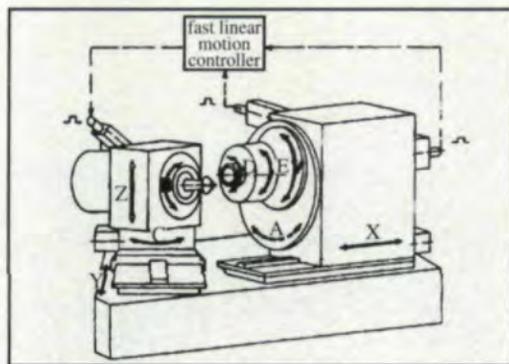


Fig. 6 - CNC bevel gear generating machine, Klingelberg KNC 40/60.

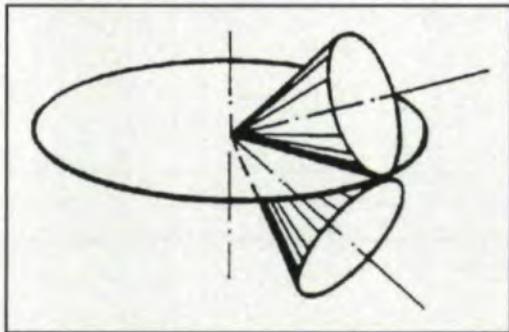


Fig. 7 - Manufacture on precise pitch cones.

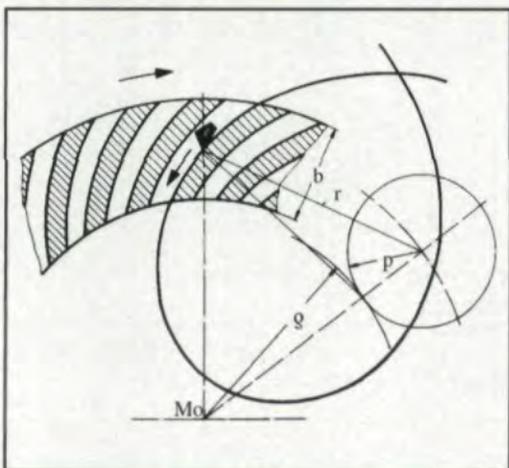


Fig. 8 - Tool radii in a continuous process.

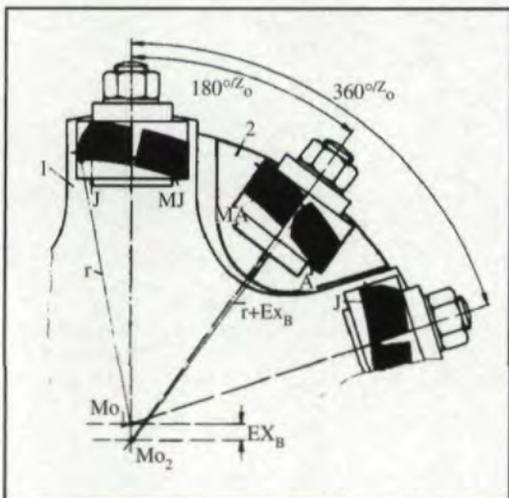


Fig. 9 - Split blade head for individual and small series manufacture.

machines. This machine has the characteristic of continuously moving the machine root angle during the generating process. This motion eliminates the tilt setting and the swivel setting.

KNC - Fig. 6 shows a full CNC bevel gear cutting machine, the KNC 40/60 series by Klingelberg. On this machine all the axes are CNC controlled. The E-axis is only used for single piece and small lot size production. This axis sets the crowning eccentricity automatically when cutting bevel gears with the divided cutter head. This axis is not used for high-volume production, regardless of whether the face milling or face hobbing process is applied. The advantage of this machine is its great stiffness, since in the single indexing process, only three axes are in motion at the same time: the workpiece rotation axis B, the workhead rotation axis A, and the infeed axis X.

Gear Cutting Systems

The two decisive differentiating features of gear cutting systems are the continuous indexing or single indexing processes and the parallel tooth depth or the tapered tooth system.

In principle, all varieties can be combined with one another, but in practice, however, the following principle combinations are common: Continuous with parallel tooth depth and epicycloid, and single indexing with tapered tooth and arc.

In the case of parallel tooth depth, manufacture is performed on precise pitch cones (Fig. 7). Unless corrections are made, no crowning will occur. The gear generating process is precise and does not result in profile bearing, and as a result of the identical tool radii of the concave and convex flanks in a continuous process (Fig. 8), without a desired correction, no longitudinal convexity is produced. The advantage is that it is simple to conduct all calculations on the imaginary crown gear in question; the complicated three-dimensional bevel gear problem becomes a simple two-dimensional crown gear problem. Profile bearing is produced precisely and specifically via the tool, and the longitudinal crown via the difference in radius of the tool flanks for convex and concave flanks. The difference in radius is achieved via a separated blade head (Fig. 9) in individual and small series production; and in large series production (Fig. 10), via the tool inclination or the gear generating process.

The continuous process is always a complet-

ing process: both flanks of the gear and pinion are completed simultaneously in each case.

The tapered tooth depth is normally not arrived at according to a theoretically precise process and, therefore, necessitates a precise pre-calculation. Depending on the branch, highly different methods are used: They are the five-cut method, in which the ring gear is prepared in two cuts (roughed down) and completed (finished), the pinion is roughed down, and each flank individually finished; and the completing method, in which the gear and pinion are completed in a single operation. As in circular arc gear cutting, the blades for the inner and outer cutters have different radii, and the completing method requires a special design, e.g., using the duplex-helical method. For circular arc gear cutting, highly efficient processing methods have been developed which can be used to particular advantage if the ring gear can be produced in a plunging process on the basis of the transmission ratio.

Very often the question arises whether there is a difference of load carrying capacity between parallel tooth depth and tapered tooth depth. (See, for example, Krenzer, AGMA Paper 91-FTM-1.) Apparently, other design criteria are more important than the gear cutting system, provided the main drive data are similar.

One definite advantage of circular arc gear cutting is that it is possible to grind completing designs very economically with cup wheels. It is also possible to grind a continuously produced gear cutting using a cup wheel with an epicycloid as the longitudinal tooth line. The difference between the arc form of the grinding wheel and the epicycloid form of the tooth in lengthwise direction is within the stock allowance. However, this form of gear cutting requires separate processing of the right and left flank. The completing design of the circular arc gear cutting, on the other hand, enables completion of both flanks via grinding in a single step.

Calculation

For all processes, calculation methods have been developed which simulate the manufacturing process and the running conditions in the drive unit. The important factor is that the calculations function independently of the system. Only in this way is it possible to achieve a comparison which is independent of the process. In addition to the pure design calculation on the



Fig. 10 - Single-part blade head for series manufacture.

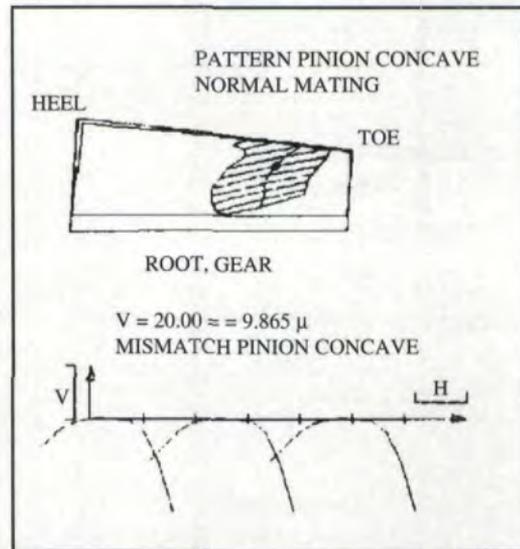


Fig. 11 - Contact area and single-flank gear generating error of a gear set (original design).

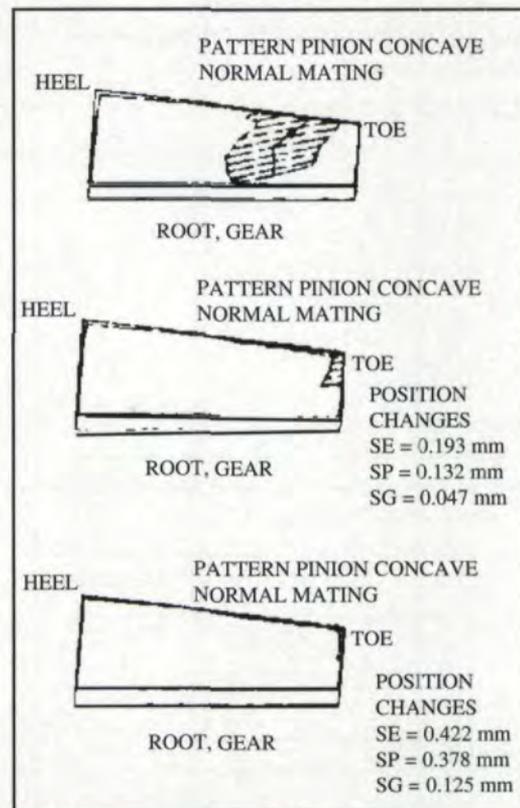


Fig. 12 - Contact areas with zero-load and two-load classes (original design).

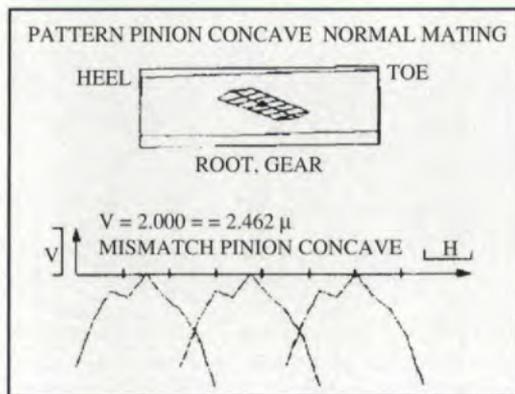


Fig. 13 - Contact area and single-flank gear generating errors optimized.

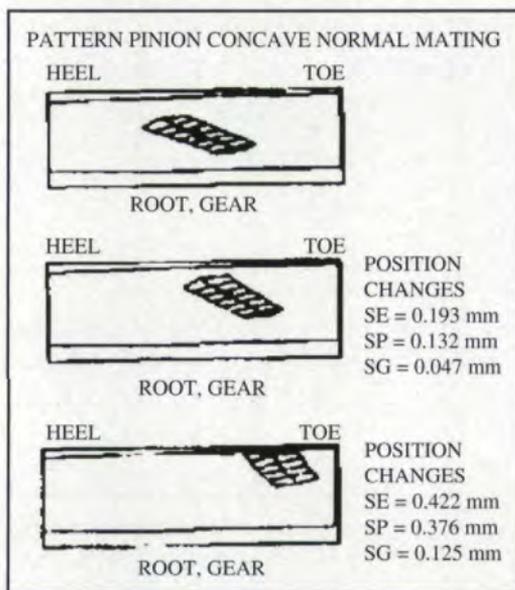


Fig. 14 - Contact area at zero-load and two-load classes optimized.

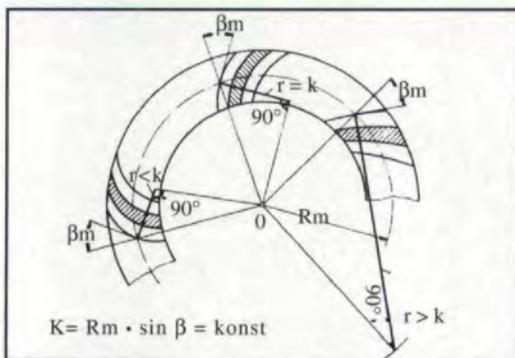


Fig. 15 - Arrangement of bevel gear and cutter in a "right-angled case."

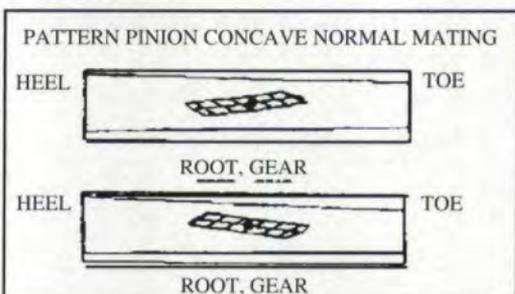


Fig. 16 - Contact area, example 2.

basis of the load capacity calculation according to AGMA or DIN standards, contact area analysis provides a reliable indication of the running behavior of a bevel gear in the drive unit. Optimization of a truck bevel gear set is given as an example.

Optimization With Regard to Noise Behavior - As the gear set is noise-critical in the original design, importance was attached to a low single-flank gear generating error.

Fig. 11 shows the contact area and single-flank gear generating error (motion curve) of the original design. The design has low convexity values due to the high noise requirements. For the drive, dislocation values below Load Classes I and II are known. These load classes correspond to 30% and 100% nominal load. Fig. 12 shows load-free contact areas for the zero position and for the dislocations which correspond to Load Classes I and II.

The large contact area indicates small crownings. Even at 30% nominal load, sharp edge wear occurs, which worsens at full load.

With knowledge of dislocation values, a gear cutting method was designed which was far less sensitive to dislocations, although the single-flank gear generating error was further improved (Fig. 13). The relevant contact areas are illustrated in Fig. 14. The load-free area appears smaller, but nevertheless, the single-flank gear generating error is improved by a factor of 2.5. No edge wear occurs at partial and full load (Fig. 14). The drive is far quieter in the entire load area than in the original design.

The following measures were implemented:

- As the contact area tends to wander in the direction of tooth length, the contact area sensitivity was reduced by switching from a large to a small cutter design. Sensitivity to contact area dislocations in the direction of tooth length is least in the so-called right-angled case (Fig. 15).

- By means of the pinpointed use of helical contacts (bias in) the overlap ratio was improved for all load areas without making the gearing more sensitive to dislocations.

The bevel gear set ground in this way is quieter than a conventionally manufactured gear set. Lapping would worsen the noise.

Optimization with Regard to Load Capacity - Another example is intended to show how the load capacity of a gear set can be optimized using calculation and CNC-controlled manu-

fracture. The gear in question is an offset gear set for a rail drive. The gear set suffers pitting, which occurs initially as micro-pitting on the tooth head of the pinion, in case of overloading. Obviously the Hertzian stress is too high.

Standard tooth contact area analysis reveals a good contact pattern in the lengthwise direction (Fig. 16.) Only the tooth contact analysis (Fig. 17) under load demonstrates that the tension level at the tooth flanks gets too high under overload. The pitting stress can be reduced by increasing the profile crowning, which is simply done by altering the input data for the grinding process. One then receives a contact pattern which is more distinctive in the direction of the tooth height (Fig. 18). The tooth contact analysis indicates a decrease of the maximum Hertzian stress from 2,600 N/mm to 2,120 N/mm, but tension peaks appear in the root relief of the pinion at the top of the gear. This can be prevented by means of tip relief at the gear root relief at the pinion. In this case, the Hertzian stress is not increased, but the tension peaks at the top of the gear teeth are avoided (Fig. 19). The extended lines indicate that these tension peaks are reduced from 2,180 N/mm to 2,135 N/mm. By means of special software, the tip relief can also be conducted in a circular motion or elliptically (Fig. 20).

Closed Loop of Correction

With today's methods of calculation, manufacturing, and inspection, it is possible to achieve a closed loop from the calculation (including contact area analysis with ease-off and single-flank gear generating error) to manufacture and inspection (Fig. 21). The figure shows an example calculation on a PC. The machine setting data resulting from this are transferred to the cutting machine and the grinding machine either via a data storage medium (diskette) or via a direct line. At the same time, performance data for the coordinate measuring machine are produced during the calculation. If deviations from the performance data are revealed during inspection, then the machine setting is automatically adjusted so that the desired geometry is produced. This "development" is particularly necessary if existing master gears need to be duplicated. ■

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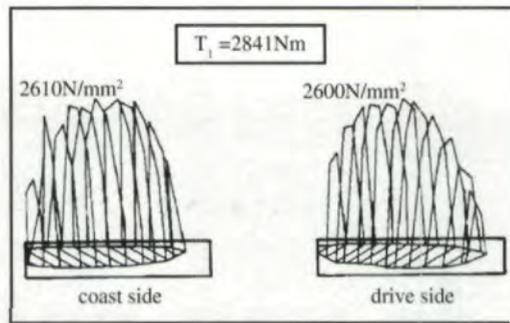


Fig. 17 - Load distribution.

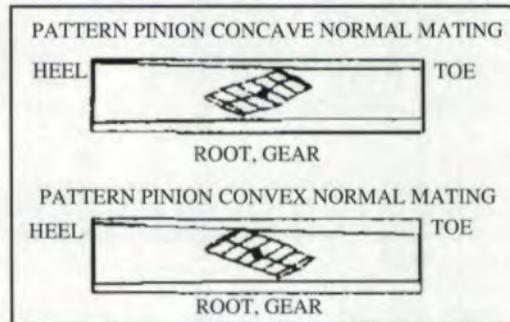


Fig. 18 - Contact pattern with smaller profile crowning.

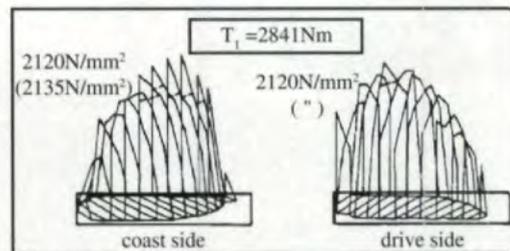


Fig. 19 - Load distribution with greater depth convexity.

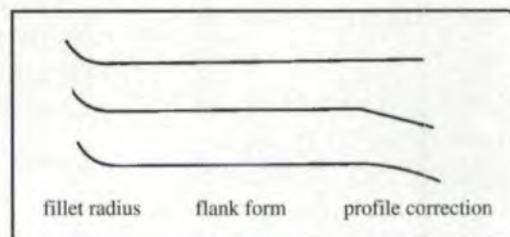


Fig. 20 - Circular tip relief.

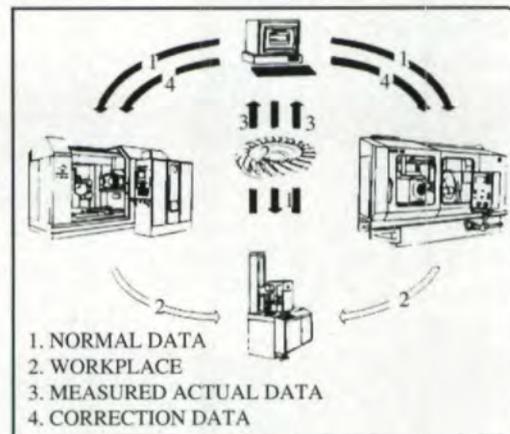


Fig. 21 - Closed loop of calculation, manufacture, measurement, and correction.

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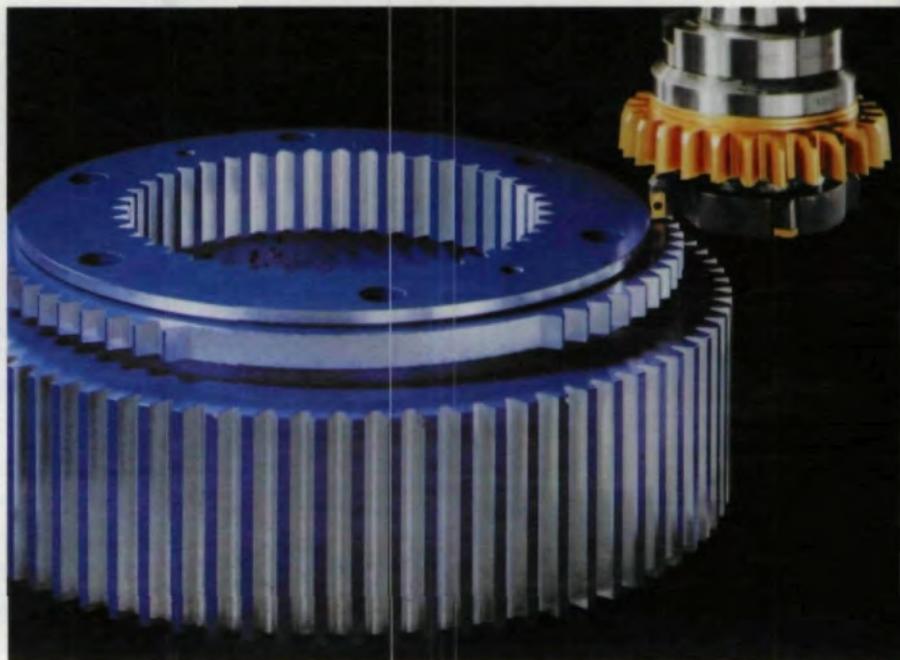
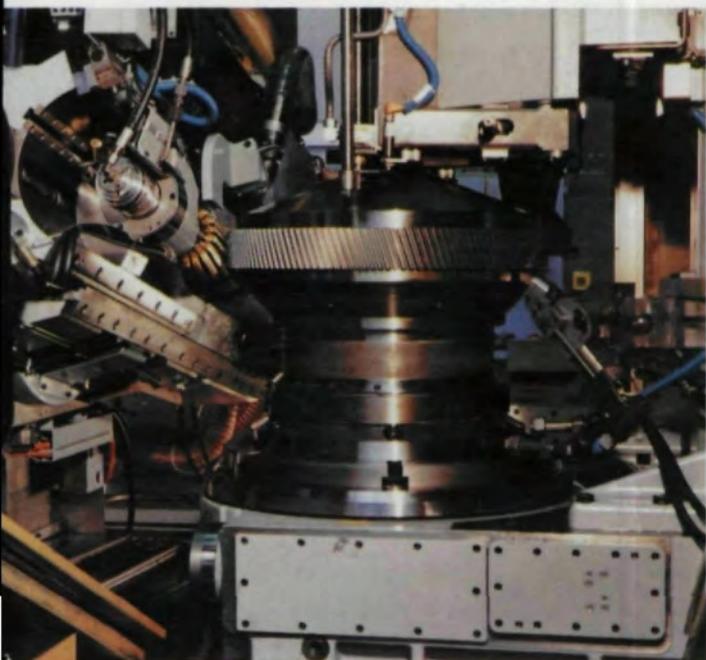
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Investigation of Gear Rattle Phenomena

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AVL List GmbH, Graz, Austria

The acceptance by discerning customers of passenger cars is dependent upon both the actual noise level and the subjective noise character.

The subjective noise character itself can contain, among other features, undesirable noise phenomena which become apparent at certain points in the vehicle operating range. One such critical phenomenon is gear rattle, which is mainly present under low speed, high load conditions. Due to changes in the angular velocity of the crankshaft, gear rattle under driving conditions occurs at the unloaded gears and splines. It is influenced by a combination of features, such as the inertia of the flywheel, clutch, and all gears, the transmission drag, and the resonant characteristics of the driveline. The effect on airborne noise is also influenced by the characteristics of the vibration transfer paths from the gear teeth meshes via shafts and bearings to the transmission housing. To analyze the interactions of these parameters, it is necessary to conduct both vehicle tests, either on the road or with a chassis dynamometer, and to investigate the bare engine-transmission unit in an anechoic test cell. The first part of the investigations reported in this article deals with the definition by conventional methods of the noise phenomena in the passenger compartment and a parallel application of a subjective

noise character evaluation system. This is followed by a correlation analysis of the noise phenomenon recorded in the passenger compartment with the noise at the source.

Finally, a comprehensive investigation of the noise phenomenon on the bare power unit and the potential of key parameters for reduction at the source will be described.

Introduction

Generally, the noise level of today's passenger cars is of an acceptable standard, in terms of both the objective noise level and the subjective noise character. Legislative regulations limit the objective noise level. The subjective noise character, which consists of more than simply the objective noise level,¹ has to be of high quality to meet the demands of discerning customers. Under these circumstances, individual noise phenomena can become annoying, although the objective noise levels are hardly affected. Such a noise phenomenon can arise from the gear rattle of a manual transmission, which occurs either in neutral or under drive conditions.² Neutral rattle is generated at idle with the transmission in neutral and the clutch engaged. Drive rattle occurs at the unloaded gear meshing points under driving conditions, usually at higher loads and lower speeds. The annoying character of rattle noise is additionally intensified by the fact that it occurs in the speed range where transmission noise is clearly audible, as it is hardly masked by the engine noise due to the different speed dependencies.

At very low speeds, transmission noise can be similar to or even higher than engine noise, particularly in the case of a petrol engine installation. However, the rate of noise increase with speed is higher for engine noise:³⁻⁶

$$I_{\text{transm}} \propto n^2 \rightarrow 6 \text{ dB/octave} \quad (1)$$

$$I_{\text{engine}} \propto n^3 \dots n^5 \rightarrow 9 \dots 15 \text{ dB/octave} \quad (2)$$

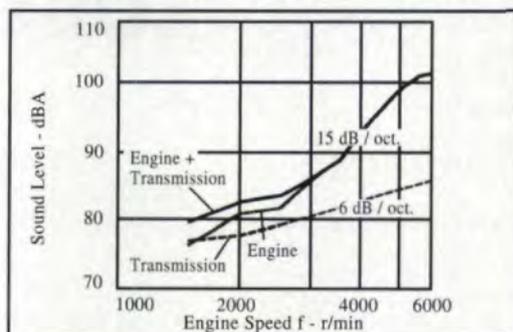


Fig. 1 - Noise of a petrol engine and transmission (full load, fifth gear).

I = sound intensity
n = speed

As shown in Fig. 1, for a power unit with a 1 ltr. petrol engine, the transmission noise is completely masked by engine noise at speeds above 2500 r/min. The results in Fig. 1 were obtained from noise analysis of the complete power unit and of the engine only (with the transmission wrapped in lead.) The transmission noise was calculated as the difference between these results. However, at speeds higher than 2500 r/min, the difference became too small for an accurate determination of the transmission noise level. For this reason, the curve for the transmission noise was extrapolated (as fine line) from the level at 2500 r/min using a slope of 6 dB/octave. This article will consider only drive rattle phenomena. The analysis of a typical gear rattle problem will be described. This occurred in a front wheel drive passenger car equipped with a 1.1 ltr. petrol engine and five speed manual transmission, and appeared as a disturbing component in interior noise.

Identification

The basic investigations were performed in an acoustic test cell with the bare engine-transmission unit. To ensure that the actual noise problem was treated on the bare power unit, the rattle noise was identified in the vehicle during preliminary road tests. To achieve this, a microphone was located close to the gearbox surface in the engine compartment. An accelerometer was fixed to the gearbox housing and an artificial head measuring system⁷ positioned at the co-driver's seat. A digital audio tape recorder was used as a high quality signal storage device. By comparative listening, the actual interior noise, the airborne sound from the gearbox, and the structural vibration of the gearbox housing via headphones - the gear rattle phenomena as well as their typical range of engine operating conditions - could be identified. Fig. 2 shows the frequency spectra of airborne noise measured first near the gearbox, and at the co-driver's seat position during road tests; then in the test cell, again close to the gearbox, and at 1 m from the power unit. The 25% load condition indicates the beginning of gear rattle. Obviously, the gear rattle is a major contributor to high-frequency noise, which can be seen in the good correlation between each diagram; thus, the engine operating range in which gear rattle

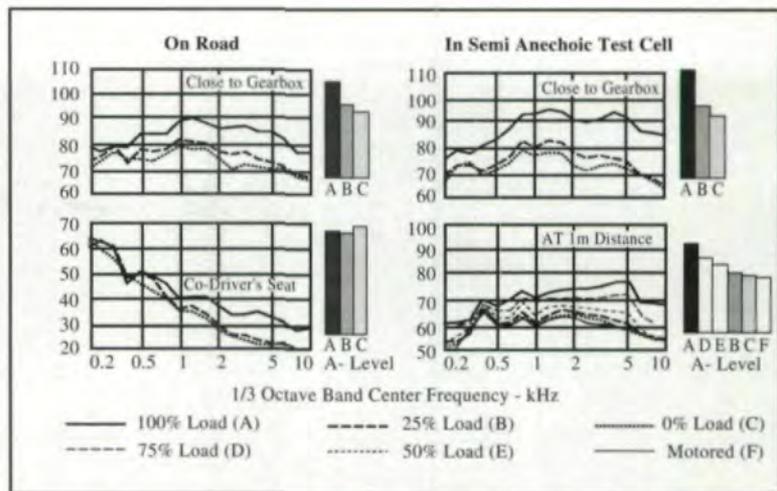


Fig. 2 - Comparison between road test and test cell results (1500 r/min, third gear).

occurred could be limited to the speed range from low idle to 2000 r/min and to the load range from about 25% to full load.

Origin Of Drive Rattle

As described in Ref. 2, drive rattle originates from teeth impacts at the meshes of unloaded gears after passing the backlash point. The impacts occur whenever the angular acceleration is high enough to make the inertia torque at the driven wheel greater than its drag torque:

$$J \cdot \ddot{\phi} > T_{\text{drag}} \quad (3)$$

J = moment of inertia of driven wheel

$\ddot{\phi}$ = angular acceleration of driven wheel

T_{drag} = drag torque acting on driven wheel

Using this relationship, the threshold of gear rattle can be defined in terms of the critical angular acceleration, f_{crit} , at which drive rattle begins to occur:

$$\ddot{\phi}_{\text{crit}} = T_{\text{drag}}/J \quad (4)$$

Therefore, rattle noise is generated when the angular acceleration reaches high values. This fact is demonstrated in Fig. 3a, representing the time trace of vibration acceleration recorded simultaneously with the speed fluctuation measured at the primary shaft, secondary shaft, and flywheel. Since the angular acceleration is the first derivative of the speed fluctuation (= angular velocity), and since the shape of the speed fluctuation is similar to a sine wave, the maximum angular acceleration occurs during the zero-crossing of the speed fluctuation. Within these time periods, high peaks of structural vibration, which are caused by the rattle impacts, can be observed. The propagation time of typically 80 ms. for the vibration transfer from the gearbox housing is negligible. At an engine speed of 1500 r/min, a time of 80 ms. corresponds to a

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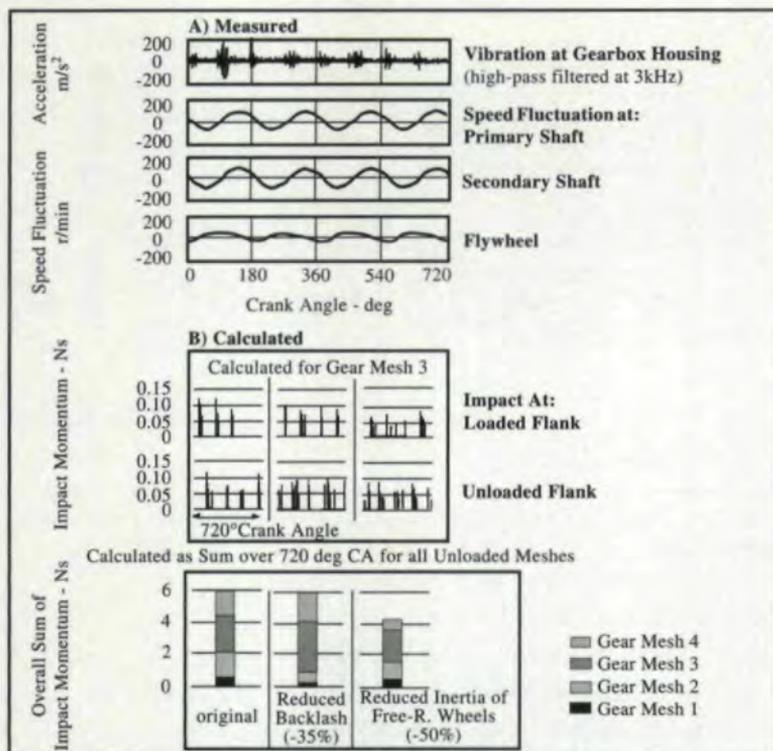


Fig. 3 - Measured and calculated time history of drive rattle impacts (1500 r/min, full load, fifth gear).

crank angle interval of less than one degree.

Some further interesting aspects can be seen in Fig. 3a. At first, the speed fluctuation is dominated by the second order frequency of the engine speed due to the rotary force characteristics of the four-cylinder internal combustion engine. For this reason, impacts occur eight times per engine operating cycle on the fully rattling transmission. Secondly, the speed fluctuations of the primary and secondary shaft are greater than those of the flywheel, and they are additionally shifted in phase. As will be illustrated later, this fact is caused by a torsional resonance in the power train with the clutch as the dominant torsional spring. Thirdly, at one gear meshing point, more than one impact can occur at a given maximum angular acceleration. This is evident from Fig. 3a, where the results were obtained from the transmission equipped - apart from the gear engaged - with only one free-running wheel. In this case all the impacts could only originate from the one unloaded gear mesh. A computer simulation of gear rattle in the fully equipped transmission system, carried out in parallel to the experimental investigation, confirmed this fact and indicated typically several impacts per reversal of tooth loading, and characterized the impacts as elastic collisions. Fig. 3b shows the result of some of these calculations (including the effect

of some parameter variations).

Key Parameters

The "primary" parameters which have an influence directly on the generation of rattle noise, can be found in Eq. 4. These are the moments of inertia of the free-running wheel, the angular acceleration, and drag torque acting on the free-running wheel. A fourth parameter not appearing in Eq. 4, but representing a fundamental condition for the generation of rattle noise, is the tooth backlash, since zero backlash precludes any gear rattle. Further possibilities for treating gear rattle are provided by "secondary" parameters acting on the propagation of rattle noise to the outside of the transmission. Here the question arises whether the direct airborne noise contribution or the structural vibration component (coming from the meshes via transfer paths) is more significant in the radiation of airborne noise from the surface of the transmission housing. To find an answer, the insertion loss of the gearbox housing was determined experimentally using a loudspeaker excitation inside the gearbox. A very high insertion loss was measured so that the contribution of the direct airborne sound can be neglected. For example, at 1500 r/min engine speed, full load, and in fourth gear, more than 99% of the total sound power being radiated from the transmission surface originated from the structural vibration component. For this reason, it is useful to consider only those secondary parameters which have an influence either on the vibration transfer from the gear meshes to the outer surface of the transmission or on the airborne sound radiation from the transmission surface.

Backlash. In accordance with experience,² the computational analysis indicated only a small effect from backlash on gear rattle (Fig. 3b). Within the limits of current tolerances, the effect of backlash is insignificant. However, zero backlash prevents gear rattle, since the tooth flanks can never lose contact and impact other flanks. Unfortunately, zero backlash is unrealistic for other reasons, except where it can be provided by an additional device such as an anti-rattle plate.²

Drag Torque. For lower rattle noise, the drag torque of the free-running wheel has to be increased as shown in Eq. 4. Two major components contribute to the drag torque; the friction due to the lubricant (viscosity, oil depth) and the

drag torque in the bearing of the free-running wheel. Within the experimental work, only the bearing drag torque was investigated. For this purpose, the radial clearance between free-running wheel and shaft (via the intermediate needle bearing) was reduced by 40% on all gears. No effect on rattle noise could be found. From the computational analysis it was concluded that to be effective, the drag torque must be increased to a certain extent so that condition (3) is no longer satisfied. If the drag torque increase remains below this limit, then the impact energy at the one tooth flank decreases, but increases at the opposite flank. Clearly the 40% reduction of the radial clearances did not increase the drag torque beyond the limit.

Angular Acceleration. Benefits can be obtained from reducing the angular acceleration acting on the free-running wheels. There are some possibilities for lowering the angular acceleration. First the output speed fluctuation of the engine depends on, among other things, the moment of inertia of the engine flywheel. The properties of the clutch disk also strongly influence the input of speed fluctuation into the transmission due to torsional resonances.⁸ Finally, the transmission arrangement; i.e., its geometry and gear ratios, determines the "local" angular acceleration of each free-running wheel. The torsional resonance of the driveline system was found to be the key feature affecting drive rattle. Fig. 4a shows the driveline resonance at full load conditions. The speed fluctuation at the rattle threshold is plotted to illustrate the range of drive rattle (shaded area). Here the speed fluctuation is defined as a percentage ratio of the difference between the maximum and minimum instantaneous speed to the mean speed. Fig. 4a clearly illustrates the procedure for the elimination of drive rattle. The task is either to raise the rattle threshold above the maximum speed fluctuations or to lower the maximum speed fluctuations below the rattle threshold or to move both limits simultaneously. In Fig. 4a the rattle threshold is near the speed fluctuation of the flywheel, and the rattle condition disappears at higher speed due to the "vibration isolation" effect between flywheel and primary shaft. This occurs theoretically - for a simple mass-spring system - at frequencies greater than $\sqrt{2}$ x resonant frequency. In Fig. 4b the effect of a lower speed fluctuation output from the engine is demon-

strated by a 30% increase of the flywheel inertia. The result is a clear reduction of the speed fluctuation over the whole speed range, but particularly at the resonant speed. The possibility of moving the resonant speed out of the lower operating speed range was investigated by means of a soft clutch (with 27% less stiffness) as well as by a progressively stiff and highly damped clutch. Both clutch disks were tested in combination with the heavy flywheel giving the results in Fig. 4b. It transpired that it is impossible, within practical limits, to shift the resonant speed below the lowest operating range; i.e., to obtain an overcritical condition which would have been the best solution. On the other hand, the stiff clutch drastically reduces the speed fluctuation input to the transmission approaching the rattle threshold. However, the rattle condition now continues up to higher speeds, and the overall noise character becomes rougher in the higher speed range due to the high damping of the clutch, which weakens the vibration isolation between crankshaft and primary shaft in the mid-frequency range. The effect of these changes on the airborne noise radiated from the whole power of 1500 r/min. Since the frequency range above 1 kHz is most significant for gear rattle noise, the range of drive rattle is shaded only above 1 kHz in this figure. It indicates the great benefit to noise reduction which is obtained when the torsional resonance is suppressed. By chance, at 1500 r/min engine speed, the speed fluctuation is the same for both the soft and the stiff clutch versions, so both noise spectra nearly

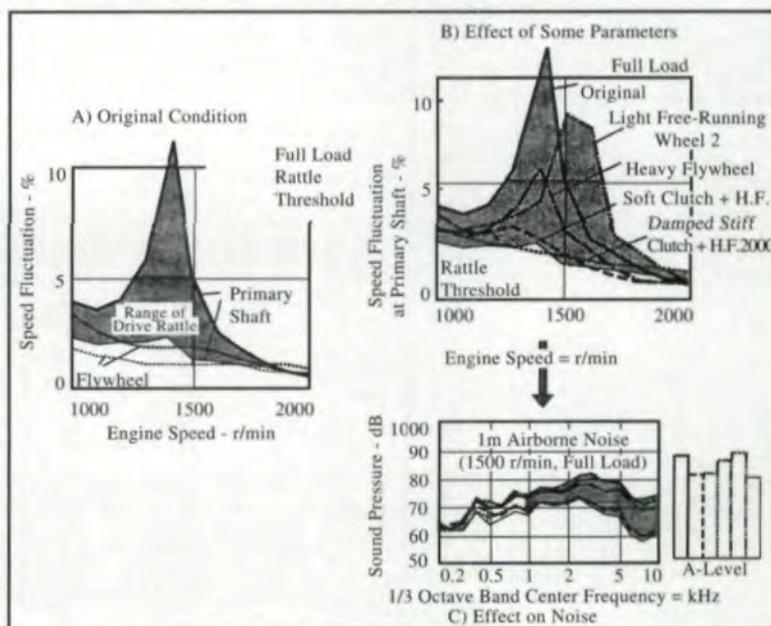


Fig. 4 - Characteristics of driveline torsional resonance (fourth gear).

coincide in Fig. 4c. The gear arrangement in the transmission was found to be disadvantageous with respect to gear rattle. As shown in Fig. 5, the original design (with the free-running wheels of third and fourth gear placed on the primary shaft) results in a very non-uniform distribution of the local angular acceleration. It favors gear rattle particularly at the free-running wheels Nos. 3 and 4, because of the high angular acceleration occurring at these wheels. The situation does not change even when the moments of inertia of the free-running wheels are taken into account (Fig. 5, right side). A re-design of the gear arrangement - placement of

the free-running wheels Nos. 3 and 4 on the secondary shaft - would give a far better starting position for low gear rattle. A re-designed transmission was not available as hardware, and it could not be tested, but the benefits for low gear rattle noise in terms of a high rattle threshold are evident in Fig. 5 due to the more even distribution and much lower maximum values of the local angular acceleration.

Inertia Of Free-Running Gear Wheel. There are three effects on gear rattle arising from the moment of inertia of the free-running wheels. The first and most important one is its influence on the rattle threshold. The rattle threshold is high if the inertia torque and, consequently, the inertia of each free-running wheel is low according to Eq. 4. The second effect is its influence on the characteristics of the tooth impact, and the third one is its effect on the driveline torsional resonance. To test these effects the free-running wheel of the second gear was selected, although the free-running wheels Nos. 3 and 4 were found to be the most prone to gear rattle (cf. Fig. 5). However, the wheels 3 and 4 could not be prepared as low inertia wheels because of their small size. Therefore, the larger wheel No. 2 was used for this test. Its original inertia was about three times greater than that of wheel Nos. 3 or 4 (cf. Fig. 5). Two samples of a low-inertia wheel No. 2 were prepared for experimental purposes. The first was extremely light with an inertia reduction of 38%. The second version, used for the final vehicle tests, was reduced in inertia by 15%. The effect of the extremely low-inertia wheel on the torsional driveline resonance can be seen in Fig. 4b, which shows a clear up-shift of the resonant speed. This speed shift, however, is not only caused by extremely light free-running wheel No. 2, but also to an equal extent by the removal of the sliding sleeve, which could not be installed together with the extremely light wheels for technical reasons. The combined effect (i.e., in combination with the heavy flywheel and damped stiff clutch) on noise is illustrated in Fig. 4c as the lowest spectrum defining the threshold of rattle noise. The computational simulation of a 50% reduction of the inertia of the free-running wheels confirmed the noise reducing effect as shown in Fig. 3b. Moreover it can be seen that the impacts decrease in magnitude and increase in number with a gain in its overall effect.

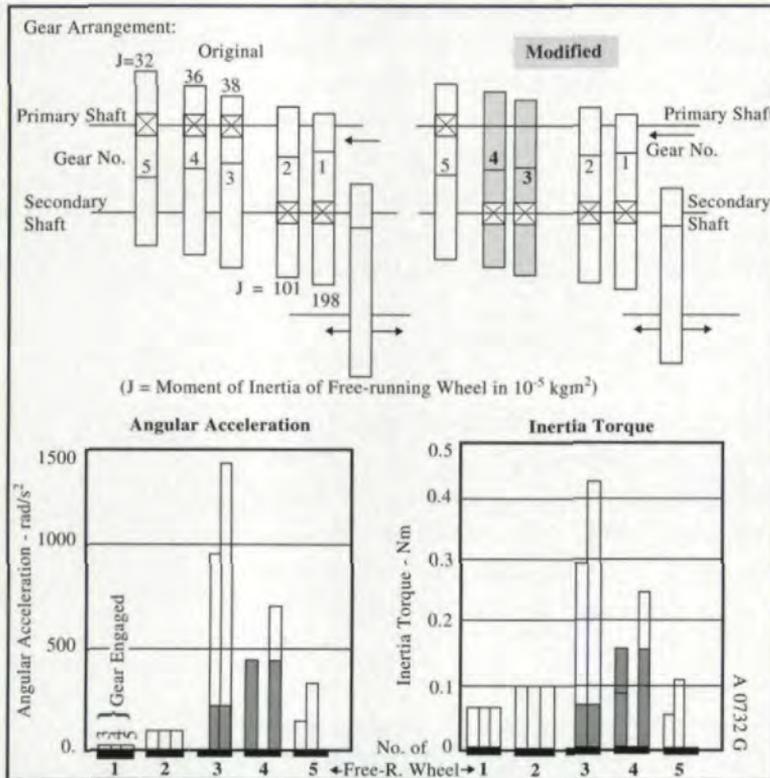


Fig 5 - Effect of gear arrangement on local rattle condition (assuming a speed fluctuation input of 0.5% at 1500 r/min).

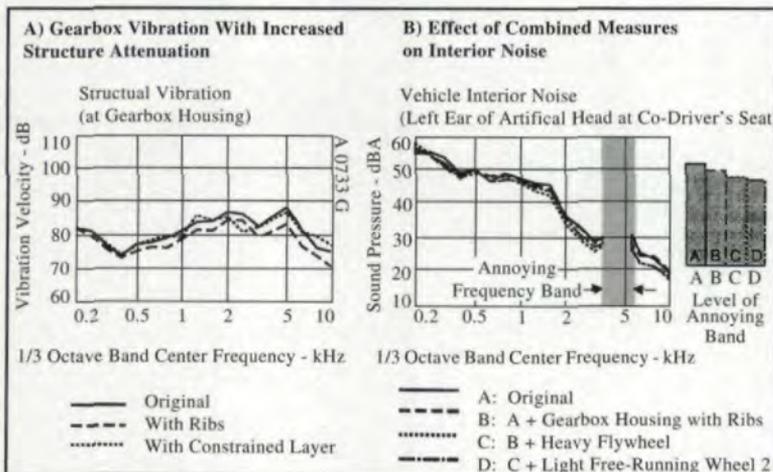


Fig. 6 - Effect of various improvements (road test, 1500 r/min, full load, fourth gear).

Vibration Transfer. Significant elements in the vibration transfer path from the mesh to the outer gearbox surface are the shaft bearings and the structure attenuation of the gearbox housing. In Ref. 3 the effect of the bearing design is described with a gain in low noise of up to 4 dB. Within the present work, the influence of the structure attenuation was investigated. From vibration measurements on the running transmission using a laser vibrometer, it was found that at frequencies above 2kHz the shaft vibration was scarcely higher than the vibration at the gearbox housing indicating a very low intermediate structure attenuation.

To determine the potential for noise reduction in terms of the structure attenuation of the gearbox housing two tests were carried out. First the walls of the gearbox housing were damped by means of a constrained layer. Secondly the walls were stiffened by ribs. Fig. 6a shows that the ribbing of the gearbox housing proved to be very effective. The reduction in vibration velocity reached up to 5 dB. The application of the constrained layer also improved the situation, but to a smaller extent. Finally the combined effect of several measures (by successive addition of single measures) was tested in the vehicle during road tests. As can be seen from the noise levels (in the frequency band significant for the rattle audibility in the interior nose) in Fig. 6b, the combination of improved gearbox housing (by ribs), increased flywheel inertia, and reduced inertia of free-running wheel No. 2 (second version) gave the best results, so that the gear rattle in the vehicle cabin was no longer audible.

Conclusions

In automotive transmissions drive rattle is excited by the angular acceleration arising from low frequency speed fluctuation caused by the fluctuating torque output of the combustion engine due to its gas and inertia forces. Work to eliminate gear rattle threshold - expressed by the critical speed fluctuation - above the actual maximum speed fluctuation. The rattle threshold depends upon the drag torque and the inertia of the free-running gear wheels, as well as on the local angular acceleration. Therefore, the following potential for improvement exists. The gear arrangement provide the opportunity to raise the rattle threshold. The proper lay-out of the gear arrangement at the design stage is a powerful

tool to minimize gear rattle or at least to provide a good condition for later improvements. In this case, the major item to be taken into account is the wheelshaft arrangement. The great influence of the driveline resonance requires a careful treatment of the clutch and the distribution of inertias. Since the resonance can hardly be avoided, the solution will always be a compromise. Therefore much effort should be concentrated on accompanying measures, such as a low speed fluctuation output from the engine and a high rattle threshold of the transmission. Since the vibration transfer from the gear meshes to the outer surface has a great effect on the rattle noise emitted by a transmission, its structure attenuation has to be of a high level. There can be some potential for improvement by stiffening or damping the housing walls and thereby increasing the "audible" rattle threshold. ■

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Base Pitch Tables

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There is one dimension common to both members of a pair of properly mating spur gears - the base pitch (BP). This base pitch is equal to the circular pitch of the gear on the base circle (see Fig. 1). For a helical gear, the base pitch can be described in either the transverse or normal plane, and is called the transverse base pitch (TBP) or normal base pitch (NBP), respectively. For parallel axis helical gears, both the TBP and NBP must be the same on both mating gears. For skew axis helical gears, only the NBP must be common.

Gears are traditionally designed on the basis of diametral pitch and pressure angle rather than solely on base pitch. This is for purposes of standardization, interchangeability, minimal tooling use, and simplicity of specifications.

Two pieces of information, diametral pitch

and pressure angle, are required to arrive at the base pitch.

For spurs: $BP = CP * \text{Cos}(PA)$

$CP = PI/DP$

where CP = Circular pitch

DP = Diametral pitch

PA = Pressure angle

For helicals:

$NBP = NCP * \text{Cos}(NPA)$

$NCP = PI/NDP$

where NCP = Normal circular pitch

NDP = Normal diametral pitch

NPA = Normal pressure angle

There are three different base pitch tables included here, one based on a metric module array (Table 1); another based on an inch circular pitch series (Table 2); and the last based on a diametral pitch sequence (Table 3). The base pitches in all cases are listed in inches.

While the tables can be used in span or block readings for gear tooth thickness measurements, the main use of these tables is usually to define the diametral pitch, circular pitch or module, and the pressure angle of an undefined or unknown gear.

Frequently a broken or worn gear is removed from a piece of machinery, and a replacement is wanted. If the original gear specifications are not available, the gear engineer may be handed this piece of broken gear or, perhaps, the mating gear and asked to define the original specifications so a replacement can be made. The base pitch tables offer a starting point to define an

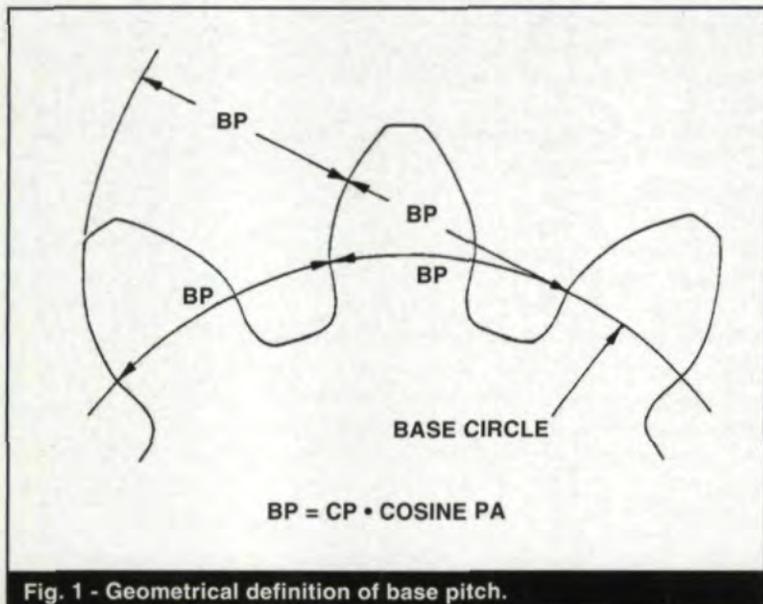


Fig. 1 - Geometrical definition of base pitch.

Table 1 - Metric Module Array

MOD	14.5	15	17.5	20	22.5	25	28	30
26.00	3.11337	3.10623	3.06697	3.02187	2.97102	2.91451	2.83939	2.78497
25.00	2.99363	2.98676	2.94901	2.90564	2.85675	2.80241	2.73018	2.67785
24.00	2.87388	2.86729	2.83105	2.78942	2.74248	2.69032	2.62097	2.57074
23.00	2.75414	2.74782	2.71309	2.67319	2.62821	2.57822	2.51176	2.46363
22.00	2.63439	2.62835	2.59513	2.55696	2.51394	2.46612	2.40256	2.35651
21.00	2.51465	2.50888	2.47717	2.44074	2.39967	2.35403	2.29335	2.24940
20.00	2.39490	2.38941	2.35921	2.32451	2.28540	2.24193	2.18414	2.14228
19.00	2.27516	2.26994	2.24124	2.20829	2.17113	2.12983	2.07494	2.03517
18.00	2.15541	2.15047	2.12328	2.09206	2.05686	2.01774	1.96573	1.92805
17.00	2.03567	2.03100	2.00532	1.97584	1.94259	1.90564	1.85652	1.82094
16.00	1.91592	1.91152	1.88735	1.85961	1.82832	1.79354	1.74731	1.71383
15.00	1.79618	1.79205	1.76940	1.74338	1.71405	1.68145	1.63811	1.60671
14.00	1.67643	1.67258	1.65144	1.62716	1.59978	1.56935	1.52890	1.49960
13.00	1.55669	1.55311	1.53348	1.51903	1.48551	1.45725	1.41969	1.39248
12.00	1.43694	1.43364	1.41552	1.39471	1.37124	1.34516	1.31049	1.28537
11.00	1.31720	1.31417	1.29756	1.27848	1.25697	1.23306	1.20128	1.17826
10.00	1.19745	1.19470	1.17960	1.16226	1.14270	1.12096	1.09207	1.07114
9.00	1.07771	1.07523	1.06164	1.04603	1.02843	1.00887	0.98286	0.96403
8.00	0.95796	0.95576	0.94368	0.92981	0.91416	0.89677	0.87366	0.85691
7.00	0.83822	0.83629	0.82572	0.81358	0.79989	0.78468	0.76445	0.74980
6.00	0.71847	0.71682	0.70776	0.69735	0.68526	0.67258	0.65524	0.64268
5.00	0.59873	0.59735	0.58980	0.58113	0.57135	0.56048	0.54604	0.53557
4.00	0.47898	0.47788	0.47184	0.46490	0.45708	0.44839	0.43683	0.42846
3.50	0.41911	0.41815	0.41286	0.40679	0.39994	0.39234	0.38223	0.37490
3.00	0.35924	0.35841	0.35388	0.34868	0.34281	0.33629	0.32762	0.32134
2.75	0.32930	0.32854	0.32439	0.31962	0.31424	0.30826	0.30032	0.29456
2.50	0.29935	0.29868	0.29490	0.29056	0.28567	0.28024	0.27302	0.26779
2.25	0.26943	0.26881	0.26541	0.26151	0.25711	0.25222	0.24572	0.24101
2.00	0.23949	0.23894	0.23592	0.23245	0.22854	0.22419	0.21841	0.21423
1.75	0.20955	0.20907	0.20643	0.20339	0.19997	0.19617	0.19111	0.18745
1.50	0.17962	0.17921	0.17694	0.17434	0.17140	0.16814	0.16381	0.16067
1.25	0.14968	0.14934	0.14745	0.14528	0.14284	0.14012	0.13651	0.13389
1.00	0.11975	0.11947	0.11796	0.11623	0.11427	0.11210	0.10921	0.10711
0.90	0.10777	0.10752	0.10616	0.10460	0.10284	0.10089	0.09829	0.09640
0.80	0.09580	0.09558	0.09436	0.09298	0.09142	0.08968	0.08737	0.08569
0.75	0.08981	0.08960	0.08847	0.08717	0.08570	0.08407	0.08191	0.08034
0.60	0.07185	0.07168	0.07078	0.06974	0.06856	0.06726	0.06552	0.06427
0.50	0.05987	0.05974	0.05898	0.05811	0.05713	0.05605	0.05460	0.05356
0.40	0.04790	0.04779	0.04718	0.04649	0.04571	0.04484	0.04368	0.04285
0.30	0.03592	0.03584	0.03539	0.03487	0.03428	0.03363	0.03276	0.03213

unknown gear.

Fig. 2 shows how a flange micrometer can be used to measure over a span or block of teeth on a gear, in this case over three teeth. In Fig. 3 a digital caliper is shown for a similar measurement over four teeth. Fig. 4 shows the requirement that the anvil or face of the measuring tool contacts tangent to the involute profile, and not on the fillet or across the tooth corners.

The difference between any two adjacent span readings, for example, the difference between the reading over three

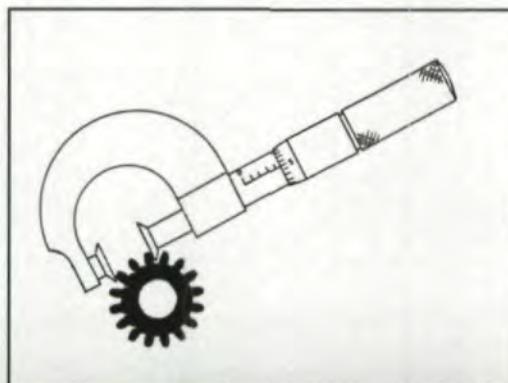


Fig. 2 - Flange micrometer measuring the span over 3 teeth on a 16-tooth spur gear.

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Table 2 - Inch Circular Pitch Series

CP	14.5	15	17.5	20	22.5	25	28	30
4.0000	3.87259	3.86370	3.81487	3.75877	3.69552	3.62523	3.53179	3.46410
3.7500	3.63055	3.62222	3.57644	3.52385	3.46455	3.39865	3.31105	3.24760
3.5000	3.38852	3.38074	3.33801	3.28892	3.23358	3.17208	3.09032	3.03109
3.2500	3.14648	3.13926	3.09958	3.05400	3.00261	2.94550	2.86958	2.81458
3.0000	2.90444	2.89778	2.86115	2.81908	2.77164	2.71892	2.64884	2.59808
2.7500	2.66241	2.65630	2.62272	2.58415	2.54067	2.49235	2.42811	2.38157
2.5000	2.42037	2.41481	2.38429	2.34923	2.30970	2.26577	2.20737	2.16506
2.2500	2.17833	2.17333	2.14586	2.11431	2.07873	2.03919	1.98663	1.94856
2.0000	1.93630	1.93185	1.90743	1.87939	1.84776	1.81262	1.76590	1.73205
1.7500	1.69426	1.69037	1.66900	1.64446	1.61679	1.58604	1.54516	1.51554
1.5000	1.45222	1.44889	1.43058	1.40954	1.38582	1.35946	1.32442	1.29904
1.2500	1.21018	1.20741	1.19215	1.17462	1.15485	1.13288	1.10368	1.08253
1.0000	0.96815	0.96593	0.95372	0.93969	0.92388	0.90631	0.88295	0.86603
0.8750	0.84713	0.84519	0.83450	0.82223	0.80839	0.79302	0.77258	0.75777
0.7500	0.72611	0.72444	0.71529	0.70477	0.69291	0.67973	0.66221	0.64952
0.6250	0.60509	0.60370	0.59607	0.58731	0.57742	0.56644	0.55184	0.54127
0.5000	0.48407	0.48296	0.47686	0.46985	0.46194	0.45315	0.44147	0.43301
0.3750	0.36306	0.36222	0.35764	0.35238	0.34645	0.33987	0.33111	0.32476
0.2500	0.24204	0.24148	0.23843	0.23492	0.23097	0.22658	0.22074	0.21651
0.1875	0.18153	0.18111	0.17882	0.17619	0.17323	0.16993	0.16555	0.16238
0.1250	0.12102	0.12074	0.11921	0.11746	0.11548	0.11329	0.11037	0.10825
0.0625	0.06051	0.06037	0.05961	0.05873	0.05774	0.05664	0.05518	0.05413

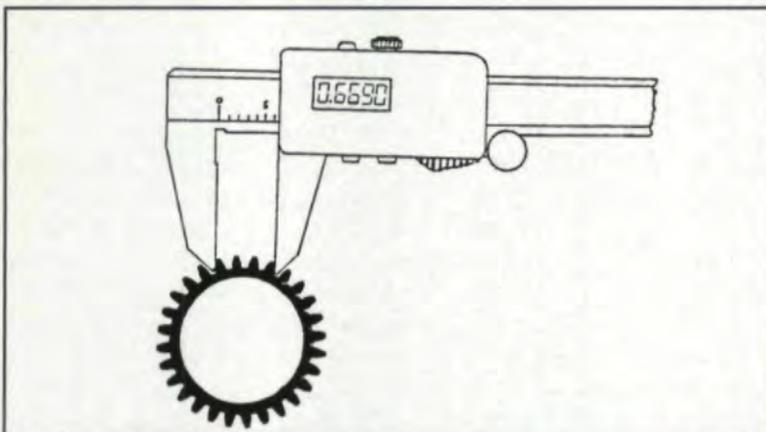


Fig. 3 - A digital caliper measuring the span over 4 teeth on a 28-tooth gear. Reads to nearest .0005 inches.

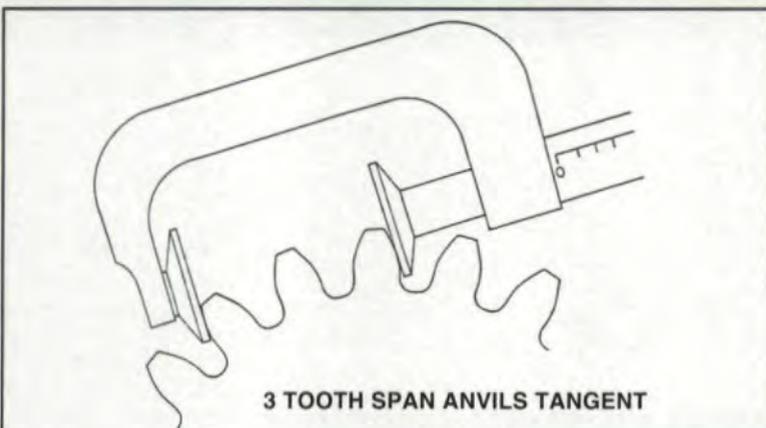


Fig. 4 - Contact on anvils must be tangent for a valid measurement.

teeth minus the reading over two teeth, yields the base pitch. This base pitch value can be compared to the tables to see if a match can be found.

EXAMPLE:

Using a flange micrometer.

Spur gear
20 teeth
Outer diameter 1.908
Root Diameter 1.528

4 tooth span = .9104

3 tooth span = .6642

Difference .2462 = Base pitch

The nearest match in the tables is a base pitch of .24601 for 12DP, 20°PA. A further check shows the pitch diameter falling on the tooth form, so there is a good probability that the gear is identified.

A dial or digital caliper can also be used, but does not have the accuracy of a micrometer. Good results can be had by averaging five or ten sets of readings.

These hand measuring tools are not very expensive compared to an involute checking instrument, although if such equipment is

Table 3 - Diametral Pitch Sequence

DP	14.5	15	17.5	20	22.5	25	28	30
1.00	3.04153	3.03455	2.99619	2.95213	2.90245	2.84725	2.77386	2.72070
1.25	2.43322	2.42764	2.39695	2.36171	2.32196	2.27780	2.21909	2.17656
1.50	2.02768	2.02303	1.99746	1.96809	1.93497	1.89817	1.84924	1.81380
1.75	1.73801	1.73403	1.71211	1.68693	1.65854	1.62700	1.58506	1.55469
2.00	1.52076	1.51727	1.49810	1.47607	1.45123	1.42363	1.38693	1.36035
2.25	1.35179	1.34869	1.33164	1.31206	1.28998	1.26544	1.23283	1.20920
2.50	1.21661	1.21382	1.19848	1.18085	1.16098	1.13890	1.10954	1.08828
2.75	1.10601	1.10347	1.08952	1.07350	1.05544	1.03536	1.00868	0.98935
3.00	1.01384	1.01152	0.99873	0.98404	0.96748	0.94908	0.92462	0.90690
3.50	0.86901	0.86701	0.85605	0.84347	0.82927	0.81350	0.79253	0.77734
4.00	0.76038	0.75864	0.74905	0.73803	0.72561	0.71181	0.69347	0.68017
4.50	0.67589	0.67434	0.66582	0.65603	0.64499	0.63272	0.61641	0.60460
5.00	0.60831	0.60691	0.59924	0.59043	0.58049	0.56945	0.55477	0.54414
5.50	0.55300	0.55174	0.54476	0.53675	0.52772	0.51768	0.50434	0.49467
6.00	0.50692	0.50576	0.49937	0.49202	0.48374	0.47454	0.46231	0.45345
7.00	0.43450	0.43351	0.42803	0.42173	0.41464	0.40675	0.39627	0.38867
8.00	0.38019	0.37932	0.37452	0.36902	0.36281	0.35591	0.34673	0.34009
9.00	0.33795	0.33717	0.33291	0.32801	0.32249	0.31636	0.30821	0.30230
10.00	0.30415	0.30345	0.29962	0.29521	0.29025	0.28473	0.27739	0.27207
11.00	0.27650	0.27587	0.27238	0.26838	0.26386	0.25884	0.25217	0.24734
12.00	0.25346	0.25288	0.24968	0.24601	0.24187	0.23727	0.23116	0.22672
13.00	0.23396	0.23343	0.23048	0.22709	0.22327	0.21902	0.21337	0.20928
14.00	0.21725	0.21675	0.21402	0.21087	0.20732	0.20338	0.19813	0.19434
16.00	0.19010	0.18966	0.18726	0.18451	0.18140	0.17795	0.17337	0.17004
18.00	0.16897	0.16859	0.16646	0.16401	0.16125	0.15818	0.15410	0.15115
20.00	0.15208	0.15173	0.14981	0.14761	0.14512	0.14236	0.13869	0.13603
22.00	0.13825	0.13793	0.13619	0.13419	0.13193	0.12942	0.12608	0.12367
24.00	0.12673	0.12644	0.12484	0.12301	0.12094	0.11864	0.11558	0.11336
28.00	0.10863	0.10838	0.10701	0.10543	0.10366	0.10169	0.09907	0.09717
32.00	0.09505	0.09483	0.09363	0.09225	0.09070	0.08898	0.08668	0.08502
36.00	0.08449	0.08429	0.08323	0.08200	0.08062	0.07909	0.07705	0.07557
38.00	0.08004	0.07986	0.07885	0.07769	0.07638	0.07493	0.07300	0.07160
40.00	0.07604	0.07586	0.07490	0.07380	0.07256	0.07118	0.06935	0.06802
44.00	0.06913	0.06897	0.06810	0.06709	0.06596	0.06471	0.06304	0.06183
48.00	0.06337	0.06322	0.06242	0.06150	0.06047	0.05932	0.05779	0.05668
52.00	0.05849	0.05836	0.05762	0.05677	0.05582	0.05475	0.05334	0.05232
56.00	0.05431	0.05419	0.05350	0.05272	0.05183	0.05084	0.04953	0.04858
60.00	0.05069	0.05058	0.04994	0.04920	0.04837	0.04745	0.04623	0.04534
64.00	0.04752	0.04741	0.04682	0.04613	0.04535	0.04449	0.04334	0.04251
72.00	0.04224	0.04215	0.04161	0.04100	0.04031	0.03955	0.03853	0.03779

available, using it would be the next step for accurate profile definition.

On a helical gear, the result of these measurements yields the NBP and, from the tables, the normal diametral pitch and normal pressure angle. This is the usual plane, for basic tooling information and many times, the design basis. Some further determination of the gear lead, such as might be accomplished on a helical lead inspection machine, is necessary to fully define the basic gear geometry.

The table for metric module gears is in-

cluded because metric gear applications are frequently found in machine tools, printing equipment, textile machinery, packaging machinery, and transmissions. Metric gears are quite widely used.

Consideration also should be given to the quality and condition of the gear being inspected as this will influence the accuracy of the values. ■

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Kanzaki-Quieting a Noisy World

As consumer demand for quieter vehicles increases, there is a greater need for exceptionally quiet transmissions. Noise reduction measures, up to now, have been difficult and expensive, due to the equipment available.

Industry experts believed that "ground" gears would provide the level of "quietness" desired, however, this assumption has generally been proven wrong. Methods of providing smoother surface finishes were necessary to eliminate and/or greatly reduce gear noise. As a result, the hard gear finishing machine was developed.

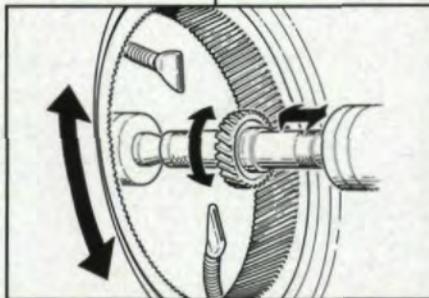
Innovative envelopment

An internal abrasive gear honing wheel offers far greater contact than a conventional, parallel axis gear honing wheel. This enveloping action creates the additional contact necessary for effective surface finishing, resulting in dramatic noise reduction.

Throughout Europe and Asia, hard gear finishing is a supplemental process for improving the surface finish of heat treated gears which have been hobbled and hardened, hobbled, shaved and hardened, or ground. The process has proven beneficial for improving surface finishes, and achieving corrections in both profile and lead. In addition, removal of burrs and nicks as well as the elimination of shavings are added benefits.

Surface finishes where $R_a = 5$, $R_t = 4$, and $R_z = 3.2$ are not uncommon. Measured at the above values, surface finishes have been achieved on gears which have been ground, shaved and hardened, or only hobbled and hardened, to greatly reduce noise produced by the actual gear teeth.

The internal finishing wheel features a master dressing device. This process is adaptable to spur, helical and internal gears. While hard gear finishing (HGF) can be used to eliminate or supplement various manufacturing methods, the amount of



improvement is dependent on the quality of the gear to be honed. Typically, improvements of one or two AGMA quality levels can be achieved.

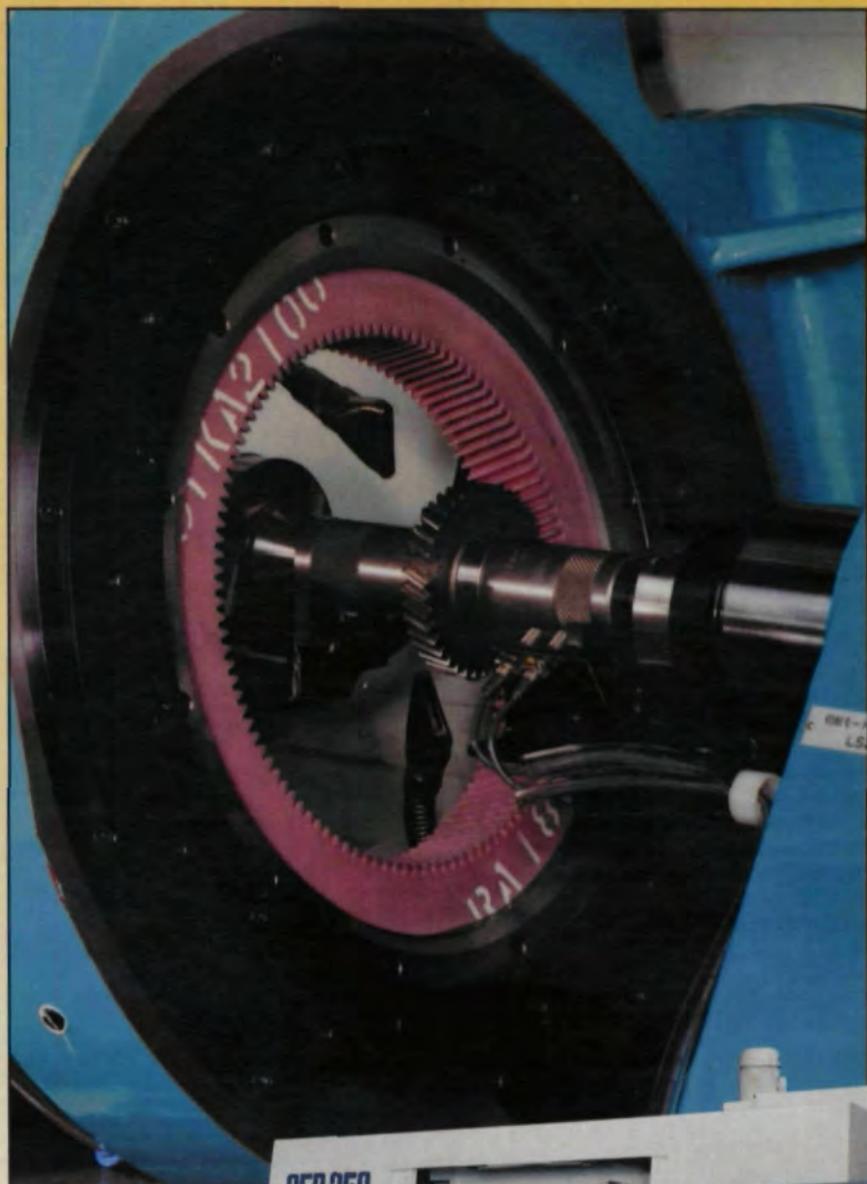
Kanzaki manufactures equipment utilizing Fanuc 5 axis controls. The operator-friendly system incorporates both a tool sensing system, which determines when the gear honing tool requires dressing, and an automatic tool dressing system. The result is assured quality in improved geometric gear accuracy and surface finish, after heat treat, to substantially reduce noise levels.

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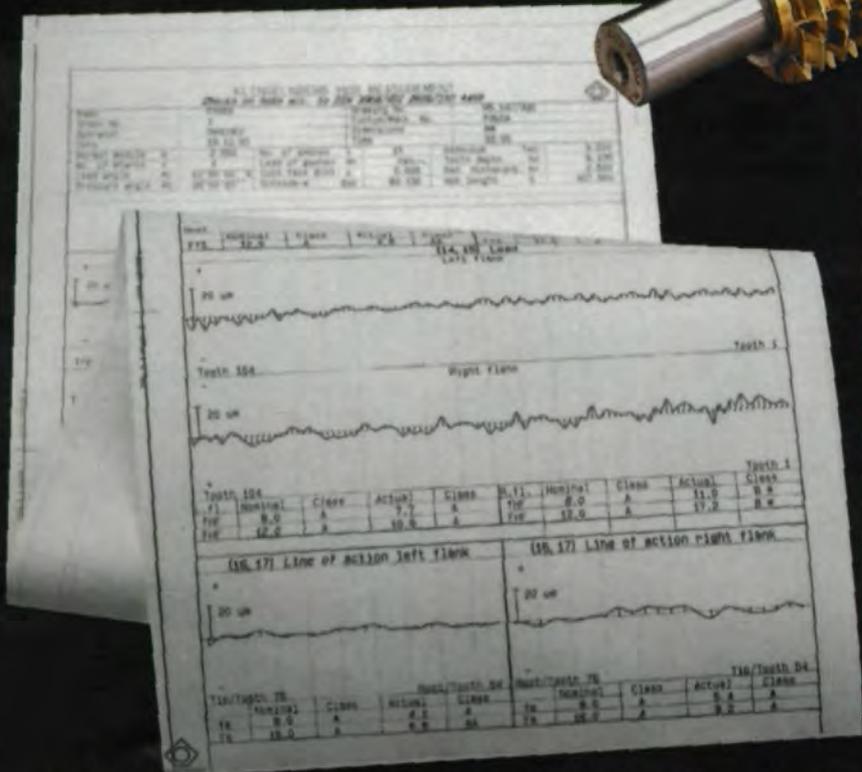
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Reader Dialogue: Functional Measurement of Gears; More Good Gear Books

Robert E. Smith
Robert Errichello

From time to time, the editors of "Shop Floor" receive correspondence from readers relating to particular articles they have written for past issues. As one of the purposes of this column is to provide a forum for the exchange of ideas, we reproduce here two of these letters and their replies. The subject of the first is the functional measurement of gears. (See *Gear Technology*, Sept/Oct, 1991, p. 17.) Robert E. Smith writes the reply.

From: Dr. Awny Y. Attia, Professor of Machine Design, Ain Shams University, Cairo, Egypt.

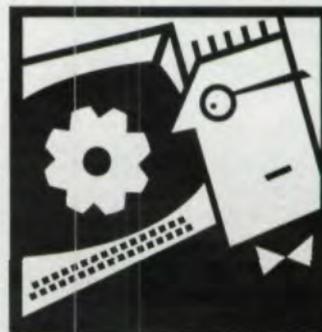
...I believe that the aim of functional measurement is to determine the deviations from uniformity of motion between two gears in action as they run and function. From the design point of view, besides running accuracy, functional errors are used to determine the maximum momentary tooth load when running at speed. They are the instantaneous deviation from correct relative angular displacement and are usually specified as a percent of the gear ratio. The value of the functional error is that it changes according to the changes in many factors, such as resultant manufacturing errors, errors in assembly, and eccentricities in bearings. Under load, more functional errors, caused by teeth deflections and torsional deflections of shafts and gear bodies, are added. When running at speed, more functional er-

rors are added, generated from the dynamic increase in tooth load, inertia of the gears, and attached masses and vibrations of the geared system.

In my opinion, there is no fixed value for the functional error in a pair of gears in mesh, as this value is changing according to this multitude of factors. This has been proven to me in many investigations where the functional errors were measured by photocells. Elemental and roll measurements do not give any indication of errors in functioning, but are very useful as a fast check on dimensional tolerances and the fulfillment of the product according to certain standard specifications.

I remember that in a gear firm, I picked some gears that had been rejected for having pitch errors exceeding the limits of tolerances, and I assembled them in pairs, those having positive errors with those having negative errors. Functional measurements of these pairs sometimes gave more accurate running than that of unrejected gear pairs. I am trying to design a set up for functional measurement of gears based on assembled metrology, hoping to give values of functional errors of any pair of gears in mesh in geared systems.

Bob Smith replies: Professor Attia is just taking what I said a step further to the measurement of assembled gears. In the Sept/Oct column, I was mainly



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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.

Robert Errichello

is the principal in GEARTECH, a gear consulting firm in Albany, CA. He is a member of AGMA, ASME, and a Registered Professional Engineer in the State of California.



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talking about the functional measurement of unassembled gears at the time of manufacture. The AGMA and ISO quality standards are written for unassembled gears.

When gear pairs are run in their actual assemblies, other factors do add to the functional characteristics. As Professor Attia points out, it is possible to put two gears together, each of which is out of elemental tolerance, and have them function perfectly well. For example: two gears can have excessive "positive tip" profile errors, but they will run together quietly. I have seen this happen many times. The profile errors complement each other in such a way that the gear "pair" is conjugate. They just happen to end up with a different pressure angle than specified. If the gear "pair" was measured by single flank methods, it would show good functional characteristics. If the gears were assembled in a gear box with relatively light loads, as many applications are, they would run just fine. If the application has loads such that the teeth deflect considerably, they would not run as well.

However, single flank transmission error measuring instruments can be applied to loaded gear boxes also. As Professor J.D. Smith of Cambridge University commented in the Nov/Dec issue, (See *Gear Technology*, Nov/Dec, 1991, "Viewpoint," p. 9) tests can and have been run at operating loads and speeds. This takes into account gear and housing errors and deflections, as well as dynamic conditions (vibration). Single Flank transmission error measuring instruments already exist that can do this.

The second letter, regarding Top Ten Books for Gear Engineers," comes from Dr. George W. Michalec, consulting engineer of Pleasantville, NY.

Dr. Michalec writes: I read with interest your article, "The Top Ten Books for Gear Engineers," in the current issue (May/June, 1992) of Gear

Technology. The coverage is both wide and in good detail. It should be helpful for many engineers.

A disappointment is that my book, *Precision Gearing, Theory and Practice*, published by Wiley in 1966, did not make the top ten. I assume you are familiar with it. Although it is now "old" I still encounter engineers who asked for it. Unfortunately, after three printings, Wiley has ceased publication, and it has been out of print since 1989. About ten years ago it was printed in China, both in English and Chinese. Thirty thousand of the latter were printed and sold out. Pleasingly, I was presented with a copy.

One of the main thrusts of this gear book is identification and handling of gear errors using statistical probabilistic techniques. An intent of the book was to teach engineers statistical mathematics applied to gearing. Some of the

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statistics present in Chapter 5, "Gear Train Performance and Error Analysis," are original. To my knowledge, the treatment of gear errors and their sources is unique, and I have not seen the equivalent in any other gear book.

I write this information with the hope that it is of interest. Also, that should you in the future choose an eleventh book, you would consider it.

Bob Errichello replies: Of course I am familiar with Dr. Michalec's excellent book. After arbitrarily deciding to limit the number of books to ten, I was faced with the unhappy task of eliminating certain useful texts.

Although the book is a treatise on statistical analysis of gear errors, I felt that the *Gear Handbook* and *Gears for Small Mechanisms* gave adequate treatment to the subject, while covering many other issues of concern to readers. With these considerations in mind and to remain within the scope of the article, I reluctantly dropped Dr. Michalec's book from the list. ■

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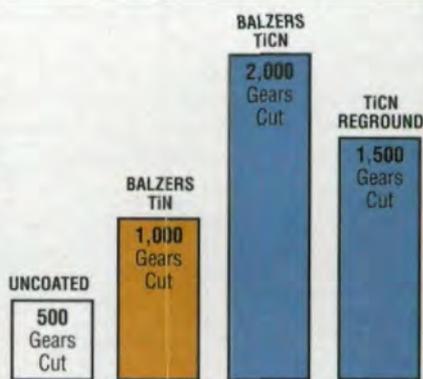
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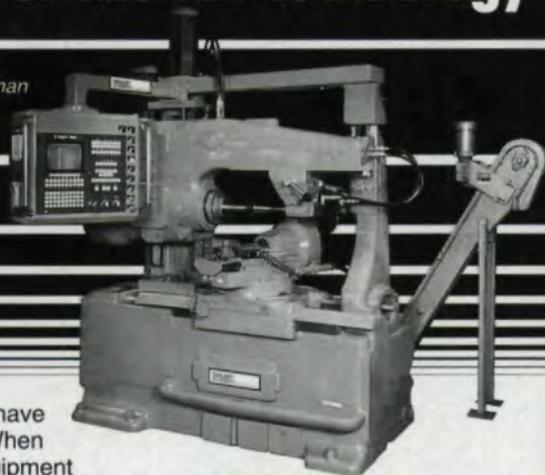
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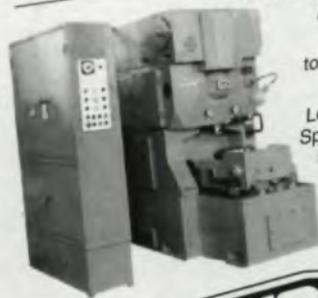
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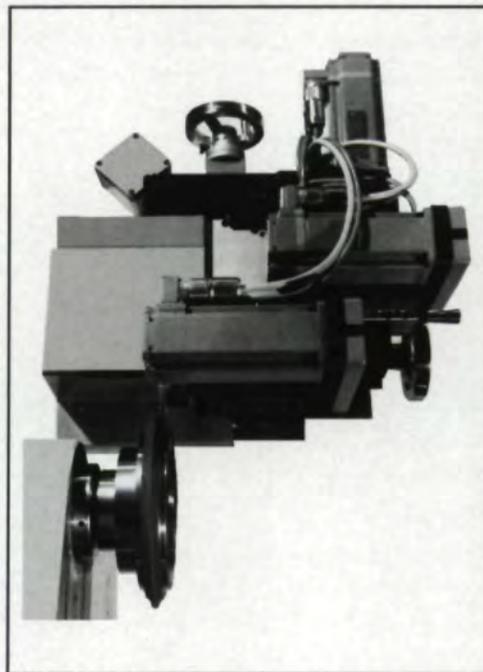
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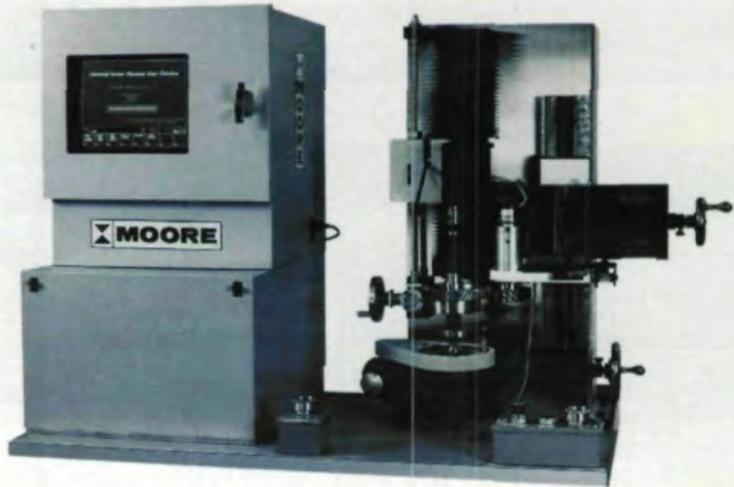
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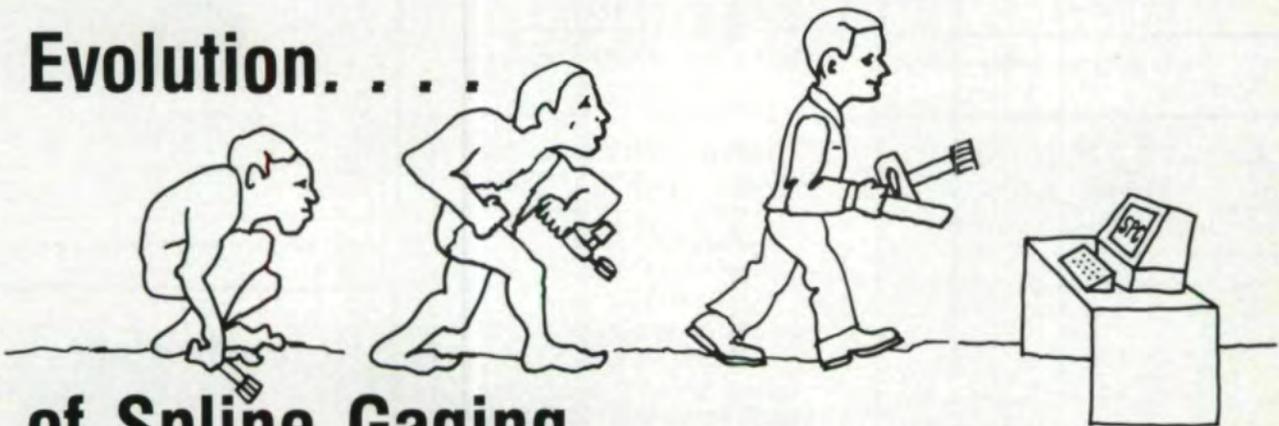


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A leading worldwide drivetrain supplier of off-highway equipment located in a desirable southeast U.S. location is searching for an experienced parallel axis gear specialist. (Spiral bevel gear knowledge is a plus.) Hands-on experience in hobbing, shaping, and shaving is a must, along with a general understanding of gear grinding. The candidate must possess good project management skills and have experience in the justification and installation of new capital equipment. The successful candidate will be an aggressive, highly motivated individual with a minimum of 7 years experience and have a BSME or equivalent. For consideration, please submit your resume and salary requirements in strictest confidence to:

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Contact: Ann Hunsucker, Excel Associates, P.O. Box 520, Cordova, TN 38018, or call (901) 757-9600; Fax: (901) 754-2896.

Gear Tool Agent/Distributor needs sales engineering person with gear tooling and manufacturing experience. Will train for sales. Established territories: Carolinas & Virginia. Salary commensurate with experience. Send resume to Box TT Gear Technology, P. O. Box 1426, Elk Grove Village, IL 60009.

Koepfer America, Inc., world leader in fine-to-medium pitch gear hobbing systems, seeks sales engineer. Responsibilities include sales and engineering support of Koepfer mechanical and CNC gear hobbing equipment, Haas hob and tool sharpening machines, cutting tools, and related equipment. Please submit resume and salary requirements in confidence to

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Fax 708-931-4192.

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A leading worldwide gear cutting tool and machinery manufacturer is searching for representatives for its full line in the following areas: •Wisconsin •Pennsylvania •Texas •Iowa •New Jersey •Colorado •Minnesota •New England •Washington State •Illinois •Kentucky •Ohio •Arkansas •Missouri. Send resume to Gear Technology, Box ST, 1425 Lunt Avenue, Elk Grove Village, IL 60007.

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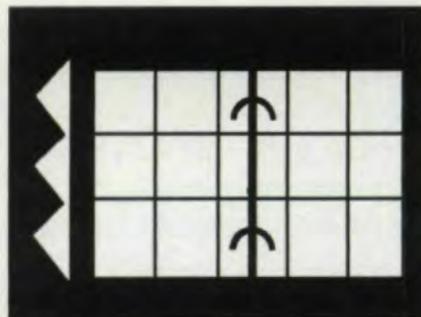
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CALENDAR

SEPTEMBER 9-11

Gear Noise Symposium. Ohio State University, Columbus, OH. Three-day course covering general noise measurements and analysis, causes of gear noise, gear noise reduction, dynamic modeling, signal analysis and problem diagnosis, housing dynamics, and housing noise radiation. For more information, contact Carol J. Bird, Conference Coordinator, (614) 292-3204 or Dr. Donald R. Houser at (614) 292-5860.

SEPTEMBER 9-17

IMTS 92, McCormick Place Exposition Center, Chicago, IL. Largest machine tool exposition in the Western Hemisphere. Affiliated technical conferences run by SME. See p. 20 for more details.

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. (313) 271-1500, ext. 596.

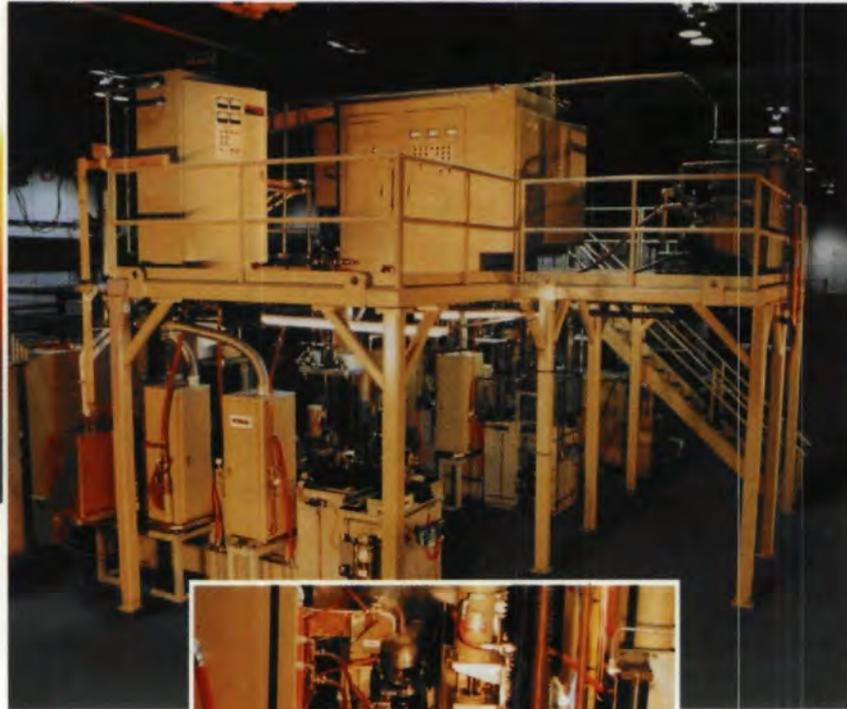
OCTOBER 26-28

AGMA Fall Technical Meeting. Marriott Inner Harbor, Baltimore, MD. Presentations and papers on Plastic Gearing, Worm Gearing, Noise & Vibration, Transmission Error, Gear Materials, CBN Grinding, Planetary Gears, Non-Involute Spur Gears, and more. Call AGMA at (703) 684-0211 for more information.

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