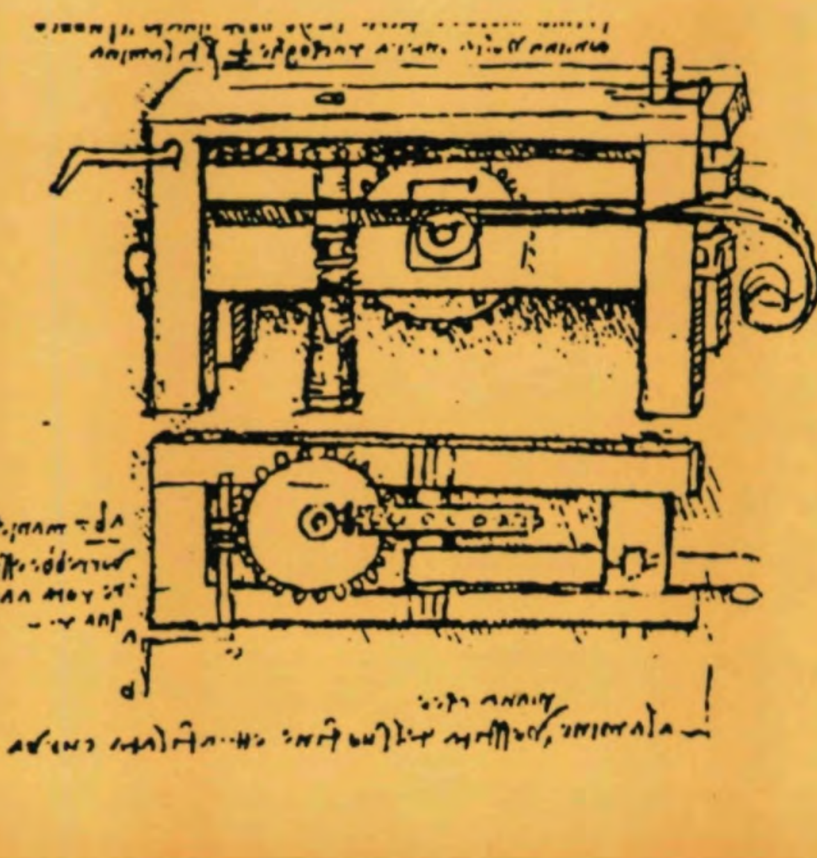
GEAR TECHNOLOGY

The Journal of Gear Manufacturing

MARCH/APRIL 1988

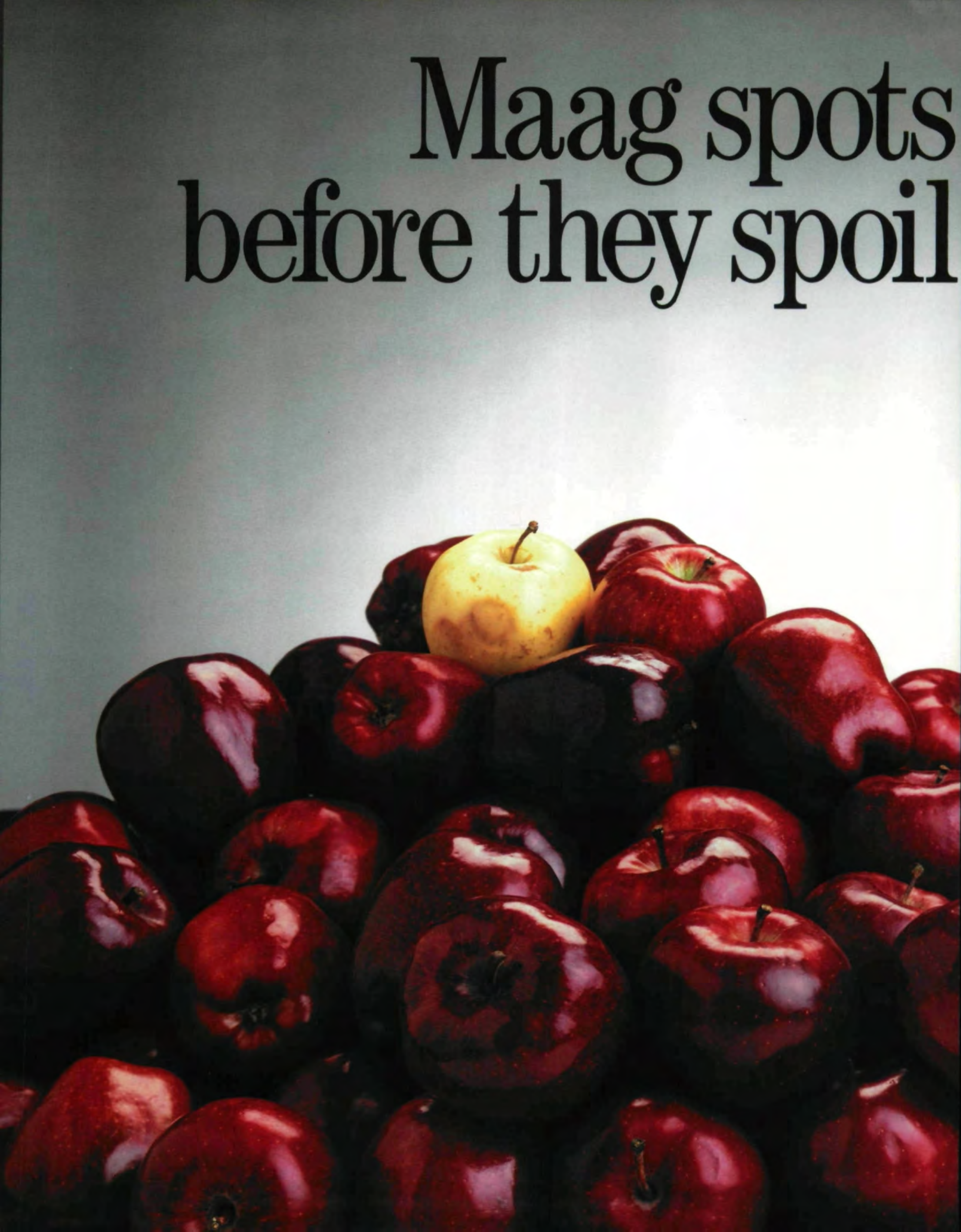


Hard Gear Finishing

Acoustic Intensity Measurements And the Effect of Tooth Profile on Noise

Contact Surface Topology of Worm Gear Teeth

Internal Helical Gear Design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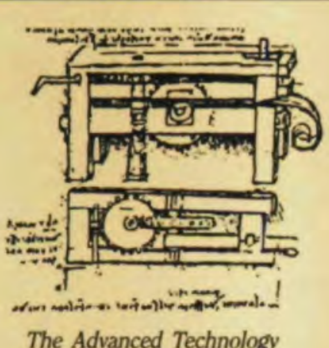

The Advanced Technology of Leonardo Da Vinci 1452-1519

COVER

Our cover shows both an elevation and a plan for a small rolling mill designed by Leonardo. The mill stretched and rolled copper strips into thin, even pieces that could be polished and used as mirrors — an expensive luxury in Renaissance Italy.

The machine is powered by a winch, which turns a large horizontal gear through a worm. A second worm on the wheel's shaft turns another gear wheel, moving the shaft around which the copper strip is wound. The strip is stretched and rolled by passing it through dies. Pressure is placed on the dies by a wedge on the top of the mill at the point where the copper enters the machine.

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Postmaster: Send address changes to GEAR TECHNOLOGY, The Journal of Gear Manufacturing, 1425 Lunt Avenue, P. O. Box 1426, Elk Grove Village, IL 60007.

MANUSCRIPTS: We are requesting technical papers with an educational emphasis for anyone having anything to do with the design, manufacture, testing or processing of gears. Subjects sought are solutions to specific problems, explanations of new technology, techniques, designs, processes, and alternative manufacturing methods. These can range from the "How to . . ." of gear cutting (BACK TO BASICS) to the most advanced technology. All manuscripts submitted will be carefully considered. However, the Publisher assumes no responsibility for the safety or return of manuscripts. Manuscripts must be accompanied by a self-addressed, self-stamped envelope, and be sent to GEAR TECHNOLOGY, The Journal of Gear Manufacturing, P.O. Box 1426, Elk Grove, IL 60007, (312) 437-6604.

EDITORIAL

HIGH TECH MANUFACTURING—CHALLENGES FOR THE 1990s

This issue's editorial is a reprint of the keynote address given by Michael Goldstein at the Computer Aided Gear Design Seminar held at the University of Northern Iowa, Cedar Falls, IA, on November 9, 1987.

When I was asked to give this speech, I received a set of specifications not unlike the ones engineers frequently receive from their bosses or customers. It was suggested that the speech be "general, inspirational, appropriate, meaningful, and comprehensive and no longer than twenty minutes." And so, with that in mind, I would like to relate a little story that I think typifies some of the things I want to talk about today.

On the sixth day of creation, God called all the angels together for a production meeting. He said, "I have some good news and some bad news. The good news is that the Creation of the World Project is coming along just great—much better even than We had imagined. The day/night idea is just the ticket, and the Market Research people tell us that oceans and especially the male/female business are going to be very popular. Production is well ahead of schedule. In fact, things are going so well, I'm going to give all of you the day off tomorrow.

"The bad news is that as soon as we're done, we have to file an environmental impact statement."

This little story summarizes one of the ironic truths about technological advances in general and the advent of the computerized workplace in particular. Overall, the results are spectacular—better even than any of us could have dreamed or imagined. A computerized workplace opens up markets and opportunities and ideas that were unheard of just two decades ago. But, as with every new invention, there are environmental impacts. Things change because of innovation, sometimes in ways we never expected. It's not that innovations bring problems with them; it's more a question of presenting us with new challenges to be met.

This is certainly the case with the introduction of the computer to the machine tool industry. CAD/CAM, CNC controls and related products that go with them have changed industry drastically. For the most part, these changes have brought nothing but good things. We can build tools and parts faster and cheaper. We can design and cut gears more accurately. We can manufacture ones that run better, longer, more silently and efficiently in smaller lots with faster changeover times. Design problems that a few years ago would take days, weeks or even months of work to solve can now be worked out in hours or even minutes, sometimes by means of computer simulations instead of with expensive prototypes. The moderately priced computer has become as much a part of the engineer's working equipment as his reference manuals, his calculator and, before that, his slip stick.

At the same time, the computerized workplace has brought challenges that we in industry will have to meet in the next decade.

New machines capable of state-of-the-art gear manufacture require educated workers. Getting and keeping such a workforce will be one of the basic challenges of the 1990s. Providing society and industry with people who have the high level of training and education necessary to meet this challenge will require some serious reordering of our priorities toward education in this country.

A good basic education is absolutely critical to the success, not only of the gear industry, but also of every other industry in the United States. In fact, it is critical to the well-being of the United States itself. Furthermore, "good" basic education is not going to be good enough. In our present highly competitive global business environment, "excellent" basic education is the least we can afford to settle for. And the painful reality is that at present, the public education system is not even coming close to providing us with this level of education. Our toughest competitors in global markets have emulated us in developing educational systems to provide them with the skilled workers they need, but now other nations are surpassing us in literacy and general skill levels. We see the results of this neglect of basics in our prospective employees and in our national educational statistics. It's unconscionable that the richest nation in the world can tolerate sending functional illiterates into the job market after twelve years of schooling—kids who cannot speak or write effectively, much less be trained to operate and maintain complex computer aided equipment.

Today 27 million adult Americans—one in five—meet this definition of functional illiteracy. It's unacceptable that in a world business environment where our strongest competitors routinely graduate some of the best prepared engineers in the world, we have allowed our government to cut educational spending by 14% in the last 9 years.

These facts indicate to me that our priorities have become terribly twisted. This kind of short-sightedness harms not only our near-term competitive ability, but also leaves a grim legacy to our children and to their children as well. John F. Kennedy said, "A child miseducated is a child lost." Anton Campanella, president of New Jersey Bell, echoes this same sentiment. He says, "We have to rely on the public schools to produce the people who will lead our business and our society. There are no 'spare' people. Society needs us all." In a world that is going to continue to become more complex and competitive, the United States cannot afford to lose a single child through simple indifference and neglect of public education.

But this is not the place to address everything that is wrong with public education. It is enough to remind you that no brilliant technological breakthroughs or progress we make in education and research on other levels will be worth anything if we have not addressed the question of a solid, basic education for all our citizens. Training our society in fundamentals simply cannot be neglected. As businesspeople, engineers, taxpayers and parents we must demand that our educational system do better, and we must force the issue with our legislators and educators until it is done.

If we are to take full advantage of the abilities of computers, CAD/CAM and CNC equipment, providing a labor pool thoroughly trained in fundamentals is a minimum requirement. The advent of this kind of technology has increased the rate of change and advance in every area, including gear manufacturing. We have to provide employees with a way to keep up with the technological changes. Continuing education is essential for everyone at our companies, from the machine operators to senior engineers and corporate officers.

As publisher of GEAR TECHNOLOGY, I have always supported and encouraged continuing education. One of the main goals we set for ourselves at GEAR TECHNOLOGY was to be an on-going gear clinic that would help disseminate the best writing of an educational and teaching nature from all over the world. We have been an ardent supporter of the technical societies and various technical exhibits and conferences.

But obviously, meeting the challenge of staying current in a changing technical environment must go beyond subscribing to a magazine. There is no such thing as a free lunch, either in life or engineering, so we should not be surprised to discover that there is no quick cure for supporting and encouraging continuing education.

Companies committed to maintaining a top-notch work force should be supporting the following strategies:

- Encourage attendance at seminars, roundtables and shows—events like this one.
- Encourage not only membership, but also *active participation* in technical societies like SME, AGMA, ASME, and ASME/GRI. These societies provide a wealth of information and resources as well as intellectual stimulation and friendships.
- Provide funding for continuing education and encouragement for employees to complete degrees.
- Provide time for reading, writing and research in gearing and other important technical fields.

Knowledge and experience should be looked upon as a capital resource, a critical component in the ultimate success of our individual companies and our country as well.

In addition to these corporate approaches, we all must nurture our own commitment to our education and professional growth. Government and the company can only do so much. If we are not concerned with our own professional development, we cannot expect anyone else to be concerned either. Greater emphasis and higher priority must be given to investing in ourselves.

John Gardner, critic and observer of American life says, "A nation is never finished. You can't build it and then leave it standing as the pharoahs did the pyramids. It has to be *re-created* for each generation by believing, caring men and women. *It is now our turn.* If we don't care, nothing can save the nation. If we believe and care, nothing can stop us."

This emphasis on continuing education and personal growth, benefits not only our companies and our society, but it also is one of the best ways to prepare ourselves to meet the challenges of a changing future. These challenges make career planning a much more difficult process than it has been in the past. A rapidly changing economic environment and equally rapid advances in technology make the old days of going to work for one company for your whole career a thing of the past. One study estimates that in the next twenty years, professionally trained people may change careers—not just jobs or companies, but careers—as many as four times. In a volatile work environment, no one can afford to neglect his or her educational development.

But this presents another dilemma for the manufacturer. Supporting continuing education for a mobile work force is a gamble. The engineer whose education you have encouraged or paid for may be working for your closest competitor next year. This is yet another challenge to management—to create a work environment that encourages the forming of long-term working relationships.

The brightest and the best of our society are the ones who tend to take their own continuing education seriously. One inducement to stay at a company is a *tangible* commitment on the part of management to the idea of continuing education. This commitment can be demonstrated through sponsorship of attendance at technical conferences and workshops, such as this one, tuition rebates, flex time policies to allow employees to attend classes and the encouragement and funding of writing and research projects.

Improving education for everyone is not the only challenge presented to us in the high-tech work place. More corporate research and development is necessary to fully utilize the potential of the computer and its related products. New products and processes ultimately create new markets. But the average gear company, like other companies, is caught in a bind that makes financing basic gear research difficult. The gear manufacturer can devote only a small portion of his profits—assuming he has any—to research. He has capital investment to provide for, wages and incentives to pay, marketing and sales forces to support and stockholders to consider. In short, he is contrained by all the requirements of running a successful business.

Government cost cutting has hit hard in this area as well. In the 8 years between 1979 and 1986, federal funding for non-military research and development fell by 25%. Of course, we cannot expect the government to do all our

(continued on page 48)

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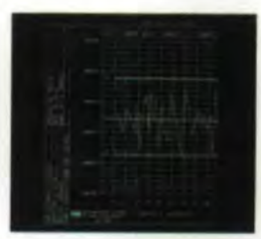

SPC Run Chart

Topological Map

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CIRCLE A-6 ON READER REPLY CARD

Hard Gear Finishing

A. Donald Moncrieff
Consultant
Zigmund Grutza,
Di-Coat Corporation

Abstract:

Hard Gear Finishing (HGF), a relatively new technology, represents an advance in gear process engineering. The use of Computer Numerical Controlled (CNC) equipment ensures a high precision synchronous relationship between the tool spindle and the work spindle as well as other motions, thereby eliminating the need for gear trains. A hard gear finishing machine eliminates problems encountered in two conventional methods – gear shaving, which cannot completely correct gear errors in gear teeth, and gear rolling, which lacks the ability to remove stock and also drives the workpiece without a geared relationship to the master rolling gear. Such a machine provides greater accuracy, reducing the need for conventional gear crowning, which results in gears of greater face width than necessary.

Hard gear finishing offers many potential benefits, including elimination of heat-treat distortion, elimination of nicks and abrasions due to handling, greater load carrying ability through the use of highly accurate gears, the opportunity to design smaller, lighter gear boxes and reduction of gear noise caused by inaccuracies in gear teeth.

These benefits involve a minimal addition to direct labor costs because hobbing or shaping are done at higher production rates, while gear honing, gear shaving and de-nicking are eliminated. Furthermore, capital costs for gear cutting and material handling are reduced, less floor space is required and the overall operating costs of machines is lowered.

Background

Millions of gears in use today have teeth that are not finished after hardening, resulting in diminished accuracy of the gear. The reason for this is the difficulty and expense of finishing hardened gears. Methods of finishing in the soft also present certain problems to the manufacturer aiming for the most accurate gear mesh.

Gear shaving has been widely used for many years to finish gears before hardening. The shaving cutter works in tight mesh at crossed axes between the workpiece and the cutter. However, shaving fails to correct some errors between the cutter spindle and the work spindle.

In gear rolling, a method evolved from gear burnishing, the gear is rolled in tight mesh with a master gear or a master rack. Gear rolling lacks the ability to remove stock. It compresses or cold forms metal instead of shaving metal from the tooth flanks. Another drawback to gear rolling, as in gear shaving, is the inability to correct some errors in gear teeth because of the lack of synchronization between the cutter spindle and the work spindle.

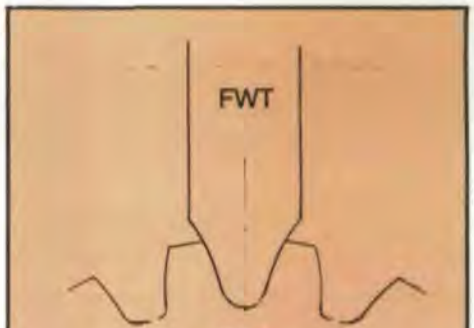

Fig. 1

Heat treating gears to reduce wear causes another set of problems. The helix angle of helical gears tends to unwind during heat-treat, and their profiles become distorted. To compensate for distortion, these gears are finished before hardening with a modified profile and a larger helix angle than the design specifies. This compensation is only partially successful because distortion caused by heat-treat is not always predictable; hence, the helix angle is crowned, and the profile is made full at the pitch line to overcome the effect of heat-treat distortion. A crowned gear reduces helical overlap, and a modified profile reduces involute overlap, both of which are necessary for quiet, efficient transmission of power.

Consequently, automobile transmission gears are made with a greater face width than would be required if they were made more accurately, for example, with the accuracy of aircraft gears that have their teeth ground after hardening. However, grinding gear teeth is a slow and expensive process and is not used for gears made in large quantities, such as those for automobiles, tractors and trucks.

Types of Hard Gear Finishing Machines

There are three types of hard gear finishing machines offered on the market today. In the following discussion these types are treated as CNC machines. These types, based on the cutters they use, are the Formed Wheel Type (FWT); the Gear Wheel Type (GWT); and the Worm Wheel Type (WWT).

The formed wheel type (Fig. 1) uses a machine that indexes the workpiece so as to grind one tooth at a time. A gear hob-

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bing machine with a suitable grinding head and a single index attachment is used for grinding large gears, while a hob sharpening machine has been developed to grind small gears.

The machines developed for this type of grinding use CBN coated or vitrified bond aluminum oxide wheels, depending on the size of machine used. However, it is generally accepted that when properly coated, a CBN wheel will give much greater tool life than an aluminum oxide one. One of the advantages of using a formed wheel is that it is easier to coat accurately with CBN than are either of the other two types.

Formed wheel type machines are used to grind either spur or helical gears and non-standard tooth forms. These wheels are used to grind the fillets as well as the root diameters of gears as required for pump gears or for aircraft gears. Each FWT hard gear finishing tool is designed for a gear of a given pitch, pressure angle, helix angle and number of teeth. It may not be used for other gears.

Continuous indexing on a FWT machine does not grind gears as rapidly as a worm wheel type machine. Index (spacing) accuracy of gears ground on a FWT are also a concern, but with the use of a CBN coated cutter, index accuracy is not a problem.

CBN coated wheels work best at higher surface speeds than are commonly used by hard gear finishing machines. The FWT machine, when using a 10" wheel, may run between 5,236 fpm and 16,439 fpm, while the developer of CBN recommends 20,000 fpm. At present, wheel speeds are limited by several factors, including spindle design and effective application of coolant to the workpiece. The formed wheel hard gear finishing machine uses creep feed to remove about 0.004" of stock from each tooth flank of the workpiece in one pass. The creep feed is approximately 23.6 fpm.

CBN coated wheels give 3,000 to 4,000 more times resistance to wear than aluminum oxide wheels. For formed wheel type grinding, tool life is estimated at 10,000 times the grinding wheel diameter for each pass across the face of a one inch gear (as advertised by the manufacturer). When estimating the tool cost for a given workpiece, it is probably best to include in the estimate the original cost of the tool plus the cost of three recoatings. For a formed wheel type hard gear finishing tool the cost of the wheel depends on its size. For instance, a 10" diameter wheel would probably cost $3,000.00, and recoating would cost $600.00.

If a 10" diameter wheel were used to grind a 30 tooth gear with a one inch face width, the cost per piece would be 36¢ as shown in Example 1.

Example 1.

A	= DP	10
B	= PA	20°
C	= HA	0°
D	= No. of teeth	30
E	= PD	3.0"
F	= Face width	1.0"
G	= Circular pitch	0.314"
H	= Creep feed	23.6 fpm
I	= Cost of HGFT	$3000.00
J	= Cost of recoating	$600.00
K	= Dia. of HGFT	10"
N	= No. of recoats	3

Tool cost =

$(I+3J) / [\{(1+N) \times (10{,}000) \times K\}/D \times F]$

Tool cost = 36¢

Production estimate =

$\{[(F/\cos C)+(G/2 \times \tan C)]/H\} \times D$

Production estimate = 88 seconds

The Gear Wheel Type HGF Machine I

From burnishing to shaving to rolling to hard gear finishing with a gear wheel type tool has been the empirical development history of the gear wheel type hard gear finishing machine. For many years, gears were burnished, shaved or rolled in an effort to machine the most accurate gears possible. In each of these processes, the cutter drives the workpiece at crossed axes without the two spindles being driven in a timed relationship.

The phrase, "You can make a good gear better, but you can't make a bad gear good," was coined to explain the limitations of these processes. Special steel treatment and the use of accurate gear blanks, protuberance hobs or shaper cutters and accurate gear cutting machines with reduced feeds and speeds before the shaving operation made an acceptable gear, but all these processes had to be performed before the gear was hardened.

The first gear wheel type hard gear finishing machine used techniques developed for plunge type shaving, but added a method of controlling the hard gear finishing tool spindle in timed relationship to the workpiece spindle with this basic change to the machine: Improvements were made to workpiece accuracy that could not be made by gear shaving. Errors in accumulative tooth spacing could be removed which would improve transmission error, as shown by single flank rolling tests. Moreover, this important improvement could be made after hardening the gear.

The gear wheel type tool emerges as a separate element,

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ZIGMUND GRUTZA *is vice president of Di-Coat Corporation. He has worked for the company for seventeen years in various capacities, including head of the research in coating processes department. He holds a Bachelor of Science Degree in Chemistry from Wayne State University, Detroit, MI. He is also a member of the Industrial Diamond Association, the Diamond Wheel Manufacturer's Institute, American Electroplaters and National Metal Finishers.*

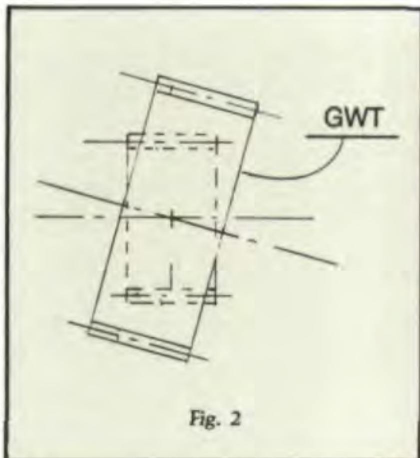


Fig. 2

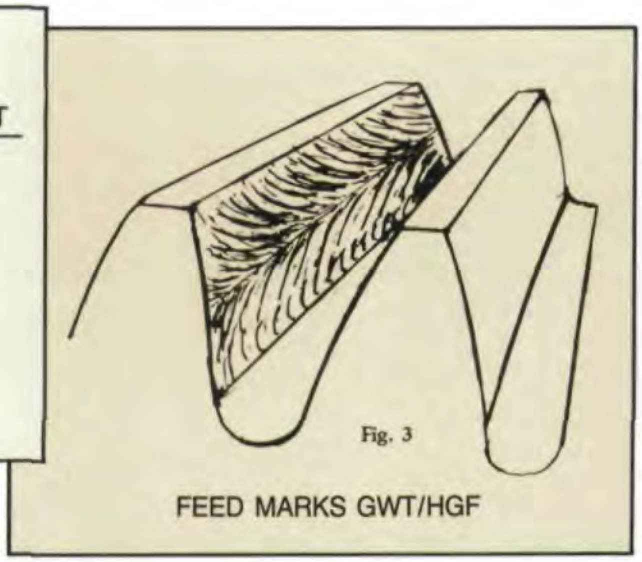


Fig. 3

but one of equal importance with the machine. The shape of the teeth on the gear wheel type tool controls the accuracy of the tooth profile, as well as the accuracy of the lead trace, including the crown in the face width of the workpiece. The success of this operation is dependent on how well the cutter is made. (See Fig. 2.)

The gear wheel type hard gear finishing machine grinds one flank of the teeth on the workpiece while being fed to depth by closing the center distance between the cutter and the workpiece. The other flanks are finished by changing the direction of rotation of the cutter and its position so as to contact the opposite flanks. The gear wheel type tool is developed to the profile and amount of crown required. A resin bonded wheel is ground by the manufacturer, and test pieces are run by the user. This routine is repeated until the cutter has been developed to produce the required profile and crown. Then, the manufacturer makes a single layer CBN wheel. In this way the workpiece teeth are designed and redesigned until the part runs satisfactorily.

Tool life varies with the size of the cutter and of the workpiece, the surface speed used and the length of sliding action between the gear wheel type tool and the workpiece. The grinding action used by a gear wheel type hard gear finishing machine is a combination of specific sliding obtained from involute action, and crossed action sliding, which happens at the same time. This results in diagonal motion of the CBN crystals across the flanks of the workpiece teeth, which motion changes direction at the rolling diameter (pitch diameter) of the workpiece. (Fig. 3)

Tool life is established for a gear wheel type tool by comparing it to the tool life for a worm wheel type tool. A test while using a worm type cutter while grinding 40,000 16-tooth helical pinions developed satisfactory tool life/cost. While 3,092 pinions were ground, 1.45 sq. in. of crystals were worn 0.002". Full information is given for this test under the description of the worm wheel type hard gear finishing machine. Since no empirical data on tool life/cost is available for gear wheel type hard gear finishing machines, this data is used to estimate tool life/cost. The number of pieces that can be ground by the gear wheel type machine is found using the formula in Example 2. This data is then divided into the cost of the gear wheel type plus three recoatings to give tool cost per piece.

Gear Wheel Type Hard Gear Finishing Machine II

A second gear wheel type hard gear finishing machine is being introduced with a development background similar to the one described above. Hard gear finishing machines developed from gear shaving are based on one or more of the gear shaving methods. These are conventional shaving, which moves the crossed axes contact point across the face width of the workpiece by feeding the workpiece along its axis; diagonal shaving, which moves the crossed axes contact across the face of the cutter by feeding the workpiece at an angle of 30° to 60° to its axis; and plunge shaving, which uses a cutter that envelopes the teeth of the workpiece and which eliminates all motion between the workpiece and the cutter, except infeed and rotation.

This second gear wheel type finishing machine is made to use all these methods, which are adapted to hard gear finishing. The machine may be used for gear shaving as well as hard gear finishing. The machine adds synchronization between the workpiece spindle and the cutter spindle, and controls this and the other required motions with CNC.

While it is possible to use the "conventional" type gear shaving principle for hard gear finishing, its use is unlikely. (See Fig. 4.) With the generating action concentrated on the crossed axes point of the gear wheel type wheel, the area of CBN coating that does the grinding on the workpiece is greatly reduced. This results in a much greater tool cost, which makes a gear wheel type wheel expensive to use for

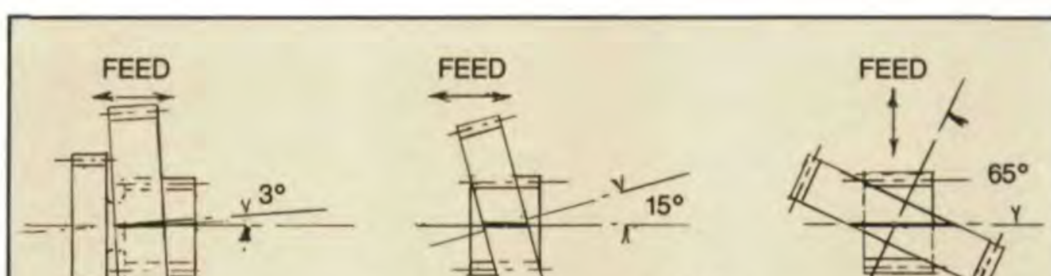


Fig. 4

Example 2 – Plunge Type GWT/HGFT.
Estimated tool life for other HGFTs compared to WWT/HGFT:

3092 (17.22/length of workpiece teeth) × (area of GWT/HGFT/1.45)

	Workpiece Data:		GWT Center		WWT Center*	
A	NDP	10	a	10	a	0.202"C.P.
B	NPA	20°		20°	b	20°
C	HA	20°	c	Spur	c	35' 45"
D	No. teeth	50	d	90	d	1 thd.
E	PD rad.	2.66"		4.5"	e	3.084"
F	Base rad.	2.4996"				
G	O.D. rad.	2.76"	g	4.635"	g	3.1496"
H	Add.	0.1"	h	0.135"	h	0.065"
I	Base pitch	.3141				
J	Face width	1.0"	j	1.064"		
K	R.P.M.	3721	k	2067	k	3300 RPM
L	Crossed Axes			20°		
M	Grinding time			80 sec.		
N	Contact area of by ctr.		n	21.15 sq. in.	n	1.45 sq. in.
P	No. pieces for one position for WWT ctr.					3092 pcs.
R	Total pcs. for original and 3 coatings WWT.					403,184
S	Cutter cost			$3100.00		$5250.00
T	Cost of recoating ctr.			$1000.00		$1800.00

*(Figures given for the WWT are used for 16 tooth pinion.)

Length of workpiece teeth = (J×D)/cos.C

Contact area for GTW/HGFT = d × (H + h) × J

Estimated tool life for GTW/HGFT compared to WWT/HGFT; 3092 (17.22/length workpiece teeth) × (area GTW/HGFT/1.45) Estimated tool life = 15,518 pcs.

Tool Cost = (S + 3T)/4 × Est. tool life
Tool cost for the gear data in the example equals 10¢ ea.

Example 3 – Conventional Type Gear Wheel Type Hard Gear Finishing Tool

Contact area of conventional gear wheel type = (d) × cross axes contact

Contact area = 90 × .08 sq. in. = 7.2 sq. in.

Estimated tool life = 3092 × (17.22/50) × (7.2/1.45) = 5,281 pcs.

Estimated tool cost = $4800/21,145 = 23¢ ea.

conventional type grinding. As an example, when figures for the use of this wheel are entered in the formula used for plunge grinding, the cost per piece is shown. (See Example 3.)

The plunge type gear wheel type tool described for hard gear finishing machines may also be used by gear wheel type II. However, the wheel that will be used to the best advantage will have fewer teeth and have a larger crossed axes angle to provide more grinding action similar to a worm wheel type tool. For example a gear wheel type II/gear finishing tool with a 65° helix angle, grinding a spur gear would give the results shown in Example 4.

Internal Gear Wheel Type Hard Gear Finishing Machine

There has been a need for a hard gear finishing machine to grind internal gears. The configurations of internal gears used in automatic transmissions are sometimes designed so that machining causes stresses in the material that distorts the gear teeth. This condition is further aggravated by heat-treat distortion. Because of these conditions an unusually high scrap rate for some internal gear parts exists. There is also a need for hard gear finishing of internal gears to the same accuracy now available for external gears.

To meet this requirement two machines have been developed, both from gear shapers. These are vertical type

Example 4 – 65° Helix Angle GWTII/HGFT

		Workpiece		GWTII/HGFT		
A	=	Dia. pitch	10	Norm. DP		10
B	=	Pressure angle	20°	Norm. PA		20°
C	=	Helix angle	0		c	65 RH
D	=	No. teeth	50		d	37
E	=	Pitch dia.	5.0"			8.754"
F	=	Base dia.	4.698"			
G	=	O.D.	5.200"			
H	=	Add.	0.100"		h	0.135"
I	=	Base pitch	0.295"			
J	=	Face width	1.000"		j	2.366"
K	=	RPM	3,268			4,300
L	=	Crossed axes angle				65°
M	=	Cost of HGFT				$2,100.00
N	=	Cost of recoating				$ 650.00

Contact area for GWT/HGFT = d × (.08) × (j/cos c) = 16.57

Estimated tool life = 3,092[(17.22/(j×D)]×(area GWT/HGFT/1.45)

Estimated tool life = 12,158 pcs.

Estimated tool cost = (M + 3N)/4 × Est. tool life
Estimated tool cost = $0.083 ea.

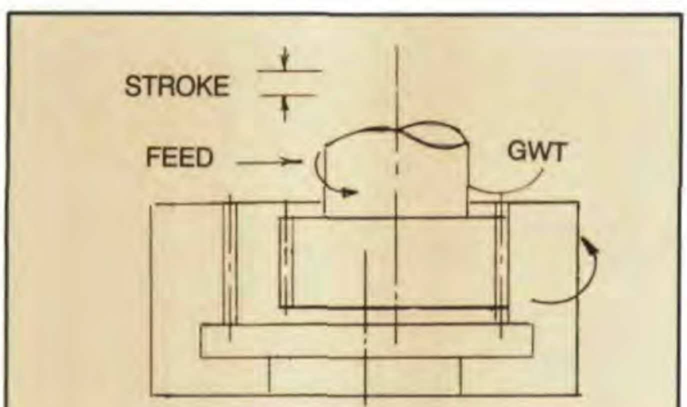


Fig. 5

machines which use state-of-the-art developments to reciprocate a gear wheel type hard gear finishing tool at 1,500 to 2,000 strokes per minute. The other motions required are controlled by CNC. (See Fig. 5.)

These machines produce very accurate gears at reasonable production rates. They are available not only for producing internal gears, but also for grinding close shoulder gears, such as those found in cluster gears used in transmissions.

When figuring tool life/cost, the efficiency of the gear wheel type tool used for grinding internal gears must be compared to that of a worm wheel type hard gear finishing tool. The gear wheel type tool used by this machine reciprocates at a speed of 100 surface fpm, while the worm wheel type rotates at a speed of at least 5,191 fpm. The expected 3,092 pieces produced by the worm wheel type have to be reduced by the efficiency of the slower moving gear wheel type. Arbitrarily, we have chosen a conservative 40% as an efficiency factor for the gear wheel type hard gear finishing tool used to grind internal gears. (See Example 5.)

Example 5 – GWT/HGFT for an internal gear.

		Workpiece		GWT/HGFT	
A	=	NDP	15.58	a	15.58
B	=	NPA	20°	b	20°
C	=	HA	18°LH	c	18°RH
D	=	No. teeth	70	d	51
E	=	Add.	0.064"	e	0.073"
F	=	Face width	1.22"	f	0.864"
G	=	Cost HGFT		g	$1,500.00
H	=	Cost recoat		h	$ 300.00
J	=	Strokes per tooth			5
K	=	No. infeeds			5
L	=	Strokes per minute			1200
M	=	Load/unload			30 sec.

Active area of internal HGFT = d × (E + e) × f = 6.037

Tool life:
3,092 × {[17.22/(D×F/cosC)]×(Active area/1.45)} × 0.4

Tool life = 1,092 pcs.

Estimated tool cost = (g+3h)/(4×tool life) = 55¢ ea

Estimated production = [(D × J × K)/(L)] + M = 118 sec.

The Worm Wheel Type Hard Gear Finishing Machine

Two manufacturers have developed the hobbing process as well as the threaded wheel grinding process into successful hard gear finishing machines. One of the makers has developed the process using as a basis a hobbing machine which was altered by replacing the hob head with a grinding head suitable for the speed and ridgity required for hard gear finishing, and using a six inch diameter hardened worm coated with CBN as a grinding wheel. The other manufacturer altered a gear grinding machine by using a 20" aluminum oxide wheel dressed to the required shape for plunge grinding.

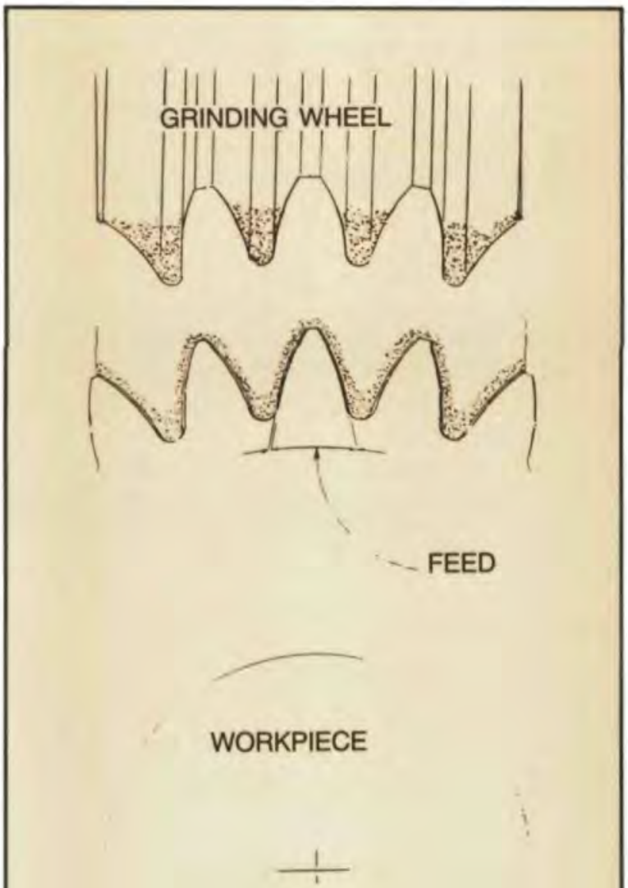


Fig. 6

Plunge Grinding With an Aluminum Oxide Wheel

Grinding in this manner makes it necessary to dress the wheel so that it will double-envelope the workpiece. This is done with a diamond coated gear which is equal to a perfect workpiece, with all the involute modifications as well as a lead trace crown, if required. The diamond coated dresser gear is plunge fed into the grinding wheel to full depth; then the synchronization between the grinding wheel and the dresser is changed to make the tooth space wider in the grinding wheel. This allows room for stock on the flanks of the workpiece and for the workpiece to be fed rapidly to the correct center distance before grinding.

First one side of each tooth is ground by a series of feeds moving the workpiece relative to the wheel in a side trimming motion. The other sides of the teeth are ground by reversing the direction of feeds. These motions are in sequence, resulting in a grinding rate of three to four seconds per tooth. (See Fig. 6.)

One of the advantages of this method of hard gear finishing is the short grinding and dressing times once every 30 to 40 pieces or as required. The grinding wheel may be dressed on the machine by feeding the dress gear through the automatic loader in place of one of the workpieces. The dress gear is moved into the wheel to dress each side of the teeth by the same sequence of motions used to grind a workpiece. Some accuracy may be sacrificed dressing in this way. However, to overcome any loss in accuracy the dress gear may be mounted behind the workpiece on the same spindle. When this is done, the machine is programmed to move into dress position and dress the wheel automatically after a preset number of pieces have been ground.

Using three to four seconds per tooth as a basis for production time and a 10 NDP, 20° NPA, 20° HA, 50 tooth, 1" face width gear, it takes approximately 200 seconds to grind this gear.

For tool life, both the vitrified grinding wheel and the diamond coated dress gear are considered. If the vitrified wheel is redressed for every thirtieth gear, its life is estimated at 7,000 pieces when a 20" diameter wheel is used. The estimated useful life of the diamond coated dress gear is estimated at 300 dressing cycles. The results are shown in Example 6.

Hard Gear Finishing With a CBN Coated Worm

Generating involute teeth with a modified worm as a tool is one of the oldest methods in use today. The accuracy of the machine and the tools used for this purpose have been improved since their invention 90 years ago; however, the basic machine and tool remain the same. It was, therefore, no surprise that this principle of gear generation was used as a basis for a worm wheel type hard gear finishing machine.

The worm wheel type hard gear finishing tool is a hardened precision ground master worm made of tool steel with the

Example 6

The tool cost is estimated to be

A	=	Cost of the vitrified grinding wheel	$ 300.00
B	=	Cost of the diamond coated dress gear	$3,000.00
C	=	Cost for recoating the dress gear	$1,600.00
D	=	No. of pieces per dressing cycle	30
E	=	No. of dressing cycles	300
F	=	No. of pieces for the grinding wheel	7,000

Tool cost = (A/F) + [(B+3C)/(D×E×4)] = 26¢ each required workpiece modifications ground into the tooth shape of the worm. (See Fig. 7.) A section through the pitch line of the cutter and the pinion shows the setting angle Φ, which is equal to the helix angle plus the thread angle. Also, the direction and length of feed of the workpiece across the hard gear finishing tool is shown. (See Fig. 8.) The tool is finished by coating its tooth flanks with CBN. Then it is carefully balanced to avoid chatter marks when the workpiece is ground. The size of the tool is 6.3" dia. by 3.54" long. Tools may have multiple threads, however, single thread tools are used most frequently. The tool rotates at 3,300 to 6,000 RPM. A feed parallel to the axis of the workpiece is equal to its face width plus a short overtravel. It is quite common to use feeds of 0.1" to 0.140" per revolution of the workpiece for roughing passes and 0.04" per revolution for a finishing pass. The tool feeds into the workpiece upon completion of each pass. Most gear teeth are cleaned up by removing 0.004" to 0.006" from the tooth thickness, which is done with three roughing passes and one finishing pass. Workpiece and tool data are shown in Example 7. Calculations are given in Example 8.

More than 45,000 pieces were ground to establish tool life. During these tests 6,000 workpieces were run in each of several positions along the hard gear finishing tool. Involute and lead tests were made on every hundredth part. (See Fig. 10.) These inspection charts show a slight improvement in surface finish between the first workpiece and the 6,000th workpiece ground.

Example 7 – WWT/HGFT.

Workpiece Data			
A	=	NDP	15.58
B	=	NPA	20°
C	=	HA (LH)	18°
D	=	No. teeth	16
E	=	PD (rad)	0.5399"
F	=	Base (rad)	0.507"
G	=	OD (rad)	0.62"
H	=	Add.	0.08"
I	=	Base pitch	0.1991"
J	=	Face width.	1.024"
K	=	Norm. circ. pitch	0.2017"
L	=	Full depth.	0.154"
WWT/HGFT Data			
a	=	NDP	15.58
b	=	NPA	20°
c	=	Thd. angle	35'45"
d	=	No. thds.	1
e	=	PD (rad)	3.0845"
f	=	Infeed (per pass)	0.0004"
g	=	OD (rad)	3.1495"
h	=	Add.	0.065"
i	=	Full depth	0.1756"
j	=	Length of HGFT	3.150"
k	=	RPM	3300
m	=	Circular pitch	0.20165"
s	=	HGFT cost	$5250.00
t	=	Recoating cost	$1800.00
u	=	Roughing feed	0.120"
v	=	Finishing feed	0.040"
w	=	Total no. of passes	4
x	=	HGFT set. angle	18.595°
y	=	Feed overtravel	0.07"

Fig. 7

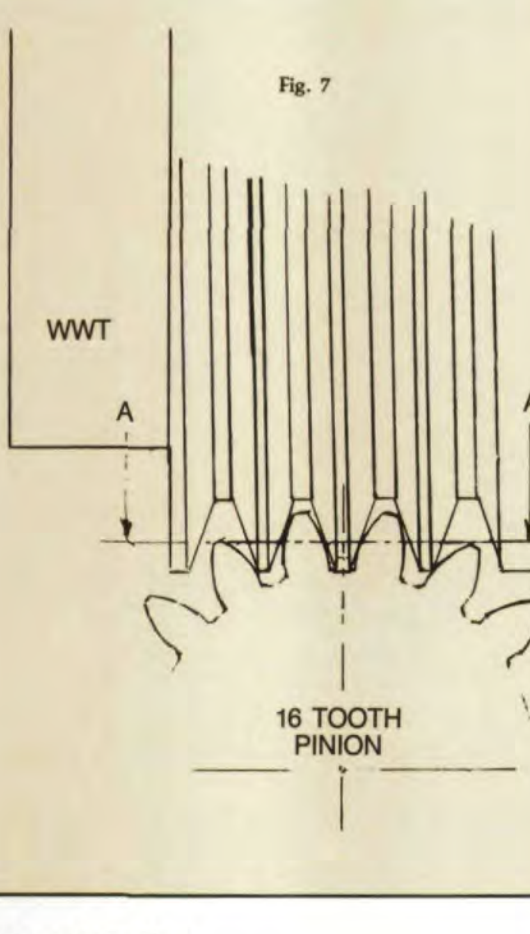
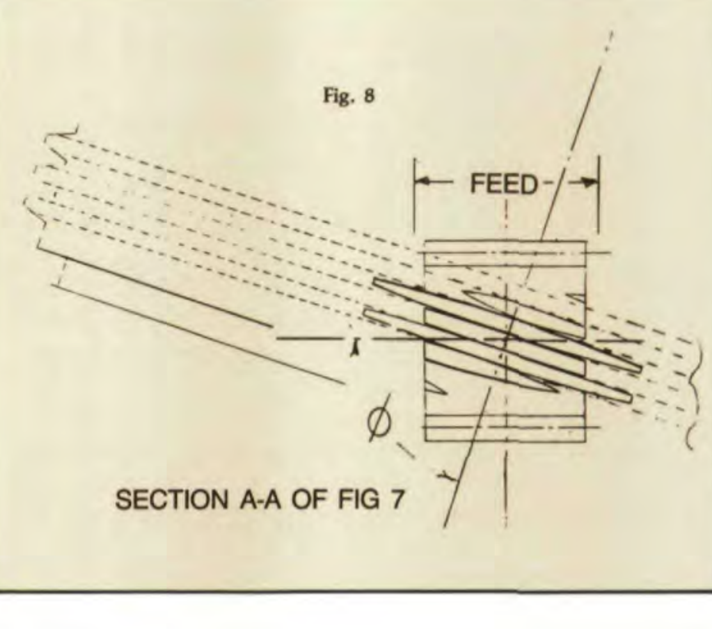


Fig. 8



Example 8 – Worm Wheel Type Tool
(See Fig. 9)

Active profile of WWT/HGFT:

$$\left[\left\{\sqrt{\frac{(G+F)\,(G-F)}{I}}\right\}/I\right] \times \left[\,(2e) \times \Phi\,\right] = 34.7363$$

Width of active profile:

$$2 \times \sqrt{E^2 - [E-(f\times w)\times \sin B]^2} = 0.049$$

Contact area of WWT/HGFT =
active profile × width × 1.702

$$Le = J - [\,2(h/\tan.B) + (m/2)\,] = 2.5092$$

$$N = L/\text{width of active profile} = 4.181$$

T = Active profile/Circum. at PL of WWT/HGFT
= 1.792

$$Pa = T \times L = 0.276$$

No. usable positions for WWT/HGFT:
$Le/(pa/N) = 39$

Tool life at one position
$53{,}236/\{\,(D \times J)/\cos C\,\} = 3{,}092$ pcs

Total tool life:
Tool life at one position × No. of usable positions
= 120,588 pcs.

Tool cost per workpiece
$(s + 3t)/4 \times$ total tool life = 2¢ ea. pc.

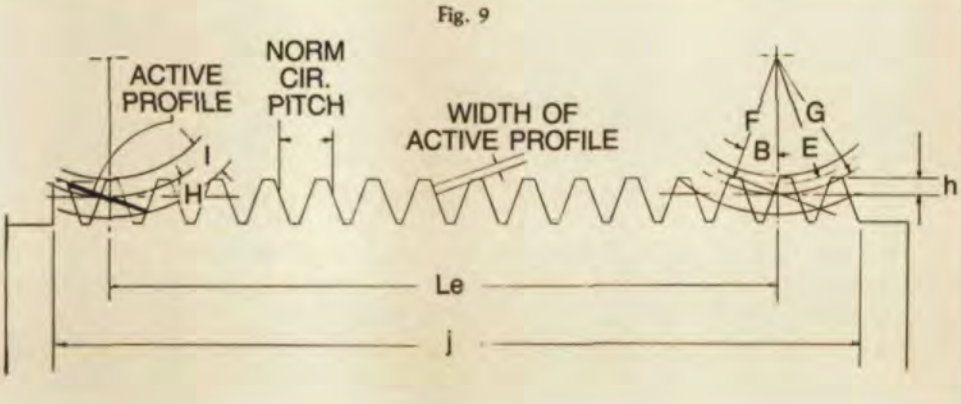
Estimated production time (allows 10 sec. load/unload)
$(D\times J\times \text{No.passes}/k\times d\times u) + (D\times J/k\times d\times v) + (w+9)/30 = 41$ sec.

Feed overtravel (include in J above)
$H\times \cos C\times \tan x/\tan B = 0.07''$

Along with the inspection for accuracy, two other factors were considered. First, a torque meter was monitored to measure the power required to drive the hard gear finishing tool spindle to prevent reaching a predetermined torque, which would disturb the synchronization between the hard gear finishing tool spindle and the workpiece spindle. Second, every fifth workpiece was Nital etched to determine surface damage. This was necessary because there was no surface damage that could be seen with the naked eye.

The Nital etch process consists of dipping the workpiece into a solution of 5 parts nitric acid and 95 parts of ethyl alcohol for 5 minutes or until the workpiece turns black, followed by a second dip into 10% HCl mixed with water. This test does not determine whether or not the surface damage affects the life of the workpiece. For this reason, additional testing using X-ray diffraction and a goniometer were conducted in an effort to measure the extent of surface damage. However, several gears were given a "back to back" test which shows exceptionally good results. For the reasons given above, 3,092 workpieces were used for tool life when grinding a 16 tooth pinion instead of 6,000

Fig. 9



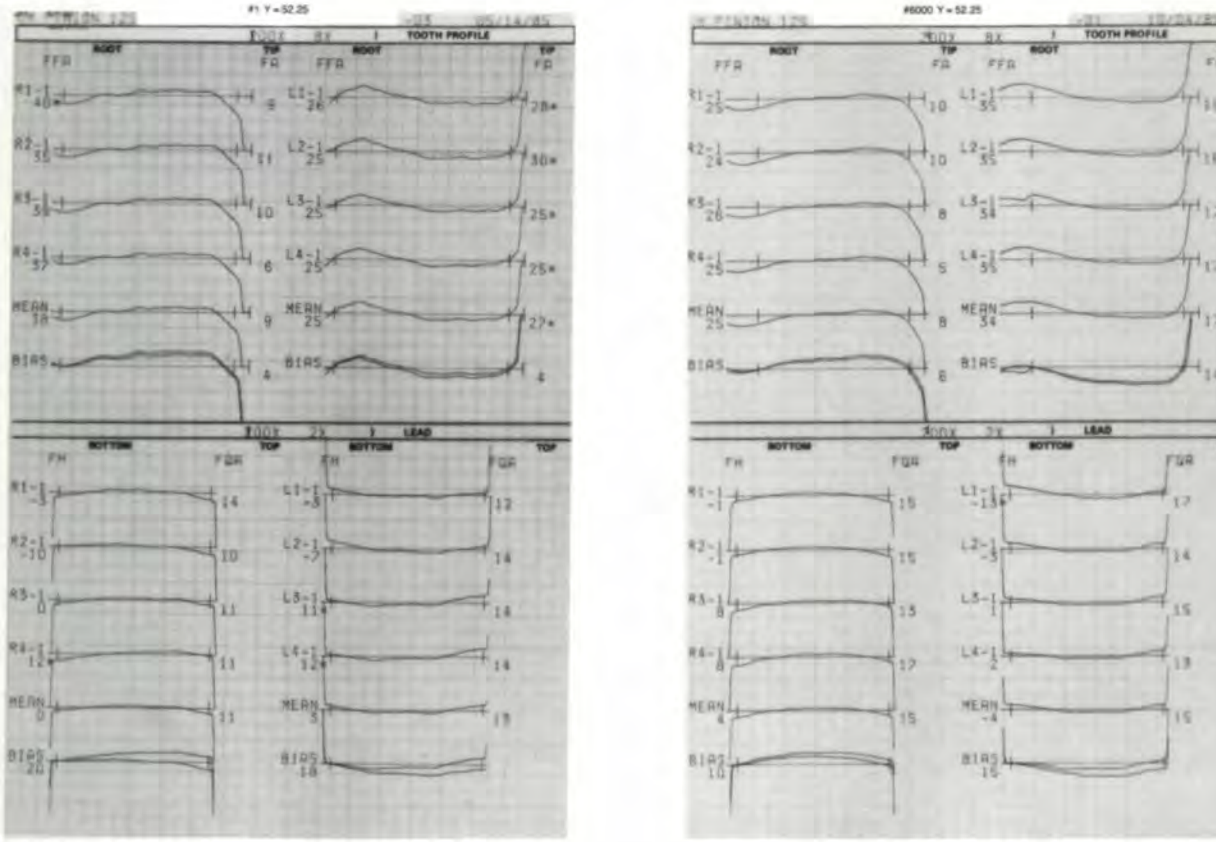


Fig. 10

pieces, which was established during accuracy tests.

This test was completed with the hard gear finishing tool rotating at 3,300 RPM. The drive motor used for the hard gear finishing tool spindle was a 3.75 HP DC servo motor, which was considered not to have enough horse power, but was the largest motor available. A later machine uses a system which provides for the use of a 5 HP DC servo motor that will drive the hard gear finishing tool up to 6,000 RPM.

Accuracy of Gears Ground by Hard Gear Finishing

It is generally accepted that all types of hard gear finishing machines are able to grind gears to AGMA Quality 13. To accomplish grinding gears to this accuracy, different types of hard gear finishing machines have various basics that must be held to close tolerances. The most difficult wheel to make is the gear wheel type used for plunge grinding. This is because the profile, the helix angle with a reverse crown, the tooth spacing and the concentricity must all meet master gear tolerances. In addition, the tool requires a wide face to envelop the workpiece.

The gear wheel type hard gear finishing tool made for conventional grinding or the wheel made to work at a large crossed axes angle do not envelop the workpiece. The workpiece lead trace is made tapered, crowned or true lead by the machine's CNC control. The gear wheel type hard gear finishing tool made to grind internal gears is made to master gear tolerances. Changes required from a true lead angle are made by tipping the machine column.

The worm wheel type finishing tool presents different problems. The teeth in cross section are modified rack teeth. The wheel must run true, and the thread angle must be held to close tolerance because runout or thread angle weave will cause errors in the workpiece teeth profiles. Electroplated CBN wheels used for grinding profiles are not easily dressed, and it is difficult to improve the wheel topography before use. Some worm wheel type finishing tools have been dressed using a diamond coated wheel or with a diamond coated duplicate of the workpiece. It is difficult to cut down the protruding crystals without disturbing the more deeply imbedded crystals using these methods. Every effort is made to coat the wheel with CBN crystals whose mesh size is uniform and to coat with uniform distribution.

The three involute/lead charts (Figs. 10 & 11) show external pinions ground with a worm wheel type finishing tool and an internal gear ground with a gear wheel type. None of these wheels were ground or dressed after they were CBN coated. The fact that it was not necessary to dress the crystals to obtain accuracy gave ultimate tool life. With sharp crystals, which are not partially blunted by dressing and which project from the bond material undamaged, the hard gear finishing tool is free cutting and permits grinding without surface damage due to heat. With sharp cutting edges, the involute chart of the first gear shows slightly rougher surface

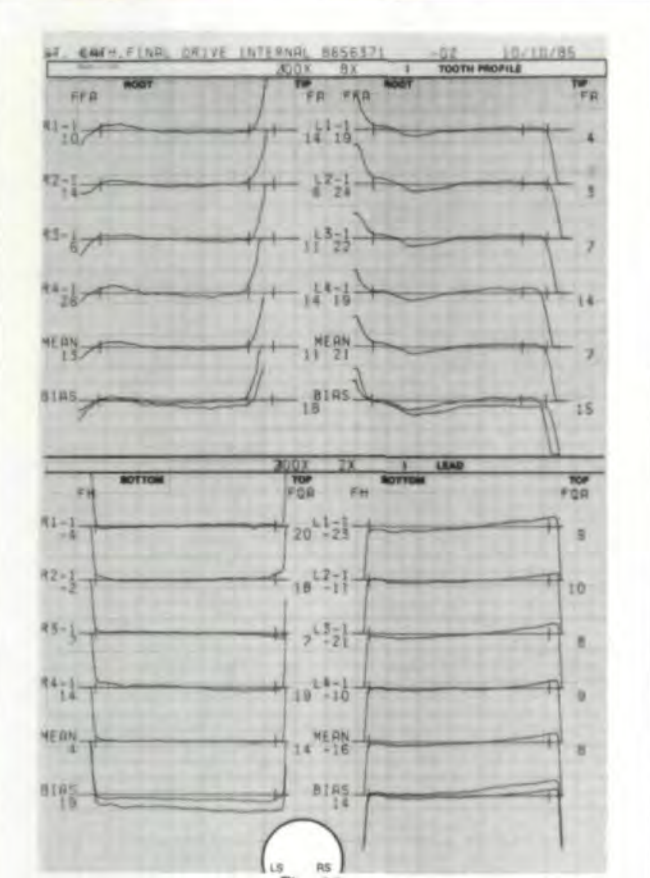


Fig. 11

finish. Now, as the crystals wear, the more deeply set crystals come into play increasing the number of cutting edges, which give a slightly smoother involute. These conditions are shown by the charts of the first and the 6,000th pinions. The difference in the progression of these conditions gradually increases the power required and the heat generated until it becomes necessary to change the hard gear finishing tool.

Hard gear finishing when using a worm type finishing tool is capable of finishes of two microns, peak to valley. The finish is determined by the grit size of the CBN crystals as well as the feed used for the finishing pass.

Surface Roughness vs. Durability

The surface durability of a pair of gears is affected by many factors, such as, roughness of the teeth, sliding speed on the tooth surfaces, hardness, material specification, accuracy of the gear teeth, loads applied, etc. One of the positive factors is that gears ground with CBN are left with compressive stresses. While there are different theories predicting the location of origin leading to tooth failures, experiments have shown that surface fatigue cracks are almost completely eliminated when metallic contact between meshing teeth is eliminated by reducing the roughness of the teeth. Table 1 lists the results of experiments which show reduced pitting of gear teeth when surface roughness of the teeth is reduced. However, surface durability cannot be increased by an improvement in surface finish alone. The accuracy of tooth profile and lead variation must be increased at the same time.

In these experiments, the theoretical oil film thickness (h min.) between meshing teeth was calculated using Dowson's equation. The D value was about 0.03 between a hobbed and a skived gear. When the driver and follower were ground, the value of D was 2, which suggests full separation of meshing teeth. D value = h min./(R max. driver + R max. follower).

Effect of Workpiece Hardness

From tests done on a 16 tooth pinion made of SAE 5140 steel drawn back to 35 Rc, it is apparent that wheel wear becomes greater while grinding low hardness steel gears. It has been noted in a report by S. Tanaka, et. al. that wear of the CBN coated wheel used for their tests increased appreaciably when they ground low hardness steel gears.[2]

This was attributed to higher plastic deformability which caused a difficulty in the production of chips. However, these tests were done while using wheel speeds of 3,700 to 7,400 fpm., and while using a very slow generating motion of 0.784 to 2.236" per min.

Worm wheel type hard gear finishing tool tests used a wheel speed of 5,190 fpm and a generating motion between the wheel and work of 21.2 fpm. The tests have shown the same surface texture and accuracy between gears of 35 Rc and 60 Rc. Fourteen hundred pieces were ground without shifting the hard gear finishing tool. This results in 44,240 pieces for one coating of the tool. (See Fig. 12.)

Coolant and Filtration

Wheel wear is closely related to the temperature of the

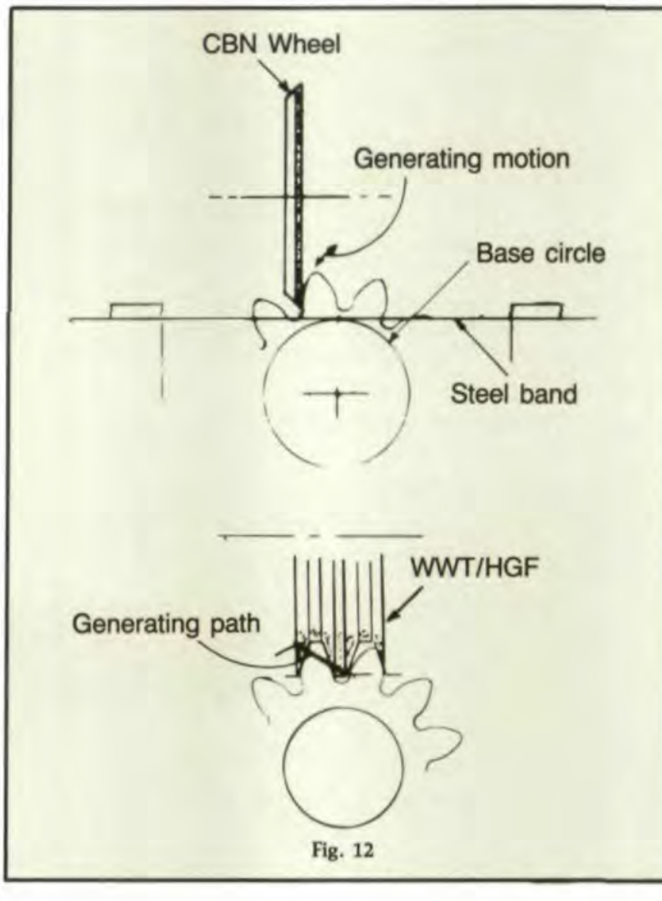


Fig. 12

TABLE 1

	Driver*			Follower**					
	Kind of Material	Hardness BHN Rc.	Surface Roughness R_{max}um	Kind of Material	Hardness BHN Rc.	Surface Roughness R_{max}um	Hertzian Pressure MPa***	Revolutions of Follower	Pitting Area Ratio (%)
HOBBED 9 AGMA 9	AISI 5135	300 BHN	10	AISI 5135	300 BHN	10	780	6.9×10^6	2.69
SKIVED AGMA 8	AISI 5135	64 Rc	2	AISI 5135	41 Rc	3	1180	10×10^6	3.3
GROUND AGMA 12	AISI 5135	64 Rc	0.1	AISI 5135	185 BHN	0.1	1180	31×10^6	0.58

*Driver - 8.466 Pitch, 20 P.A., 25 teeth, 590 face **Follower - 27 teeth ***780 = 53.5 Kg/mm^2, 1180 = 120 Kg/mm^2

TABLE 2

TYPICAL PHYSICAL PROPERTIES OF OIL USED AS COOLANT

Appearance	Color	ASTM L2.5
Specific Gravity @	60°F	0.87
Flash Point	C.O.C.	388°F
Viscosity @	100°F	118 SUS
Copper Corrosion @	210°F x 1 hr.	1 (a)
Total Acid Number	Milligrams K.O.H./gm	0.86

wheel matrix. As the temperature increases, the wear of the wheel increases. The best results are obtained by grinding with oil as a coolant. The properties of such a coolant oil are shown in Table 2.

While grinding with a worm wheel type finishing tool, two coolant nozzles are used, one pointing toward the arc of cut from each side of the axis of the wheel. Each nozzle is supplied with coolant by its own pump. The pump used is driven by a one-half HP 1,200 RPM motor and delivers 16 GPM.

Coolant Filtration

The level of coolant filtration has an important effect on the power required for grinding and the quality of surface finish, as well as on wheel wear. The system used should remove particles larger than five microns.

Discussion

Some information has been given above on currently available machines that use CBN coated wheels with brief mention of a machine which uses an aluminum oxide wheel. Research and development is a slow, painstaking process. The machines developed will serve as a corner stone for a process that is exciting and useful. Other machines will be developed that will improve on the basic process.

Further development is needed in the following areas:

- More even coating of wheels and distribution of CBN
- More exact control of grit size for CBN distribution
- Closed circuit TV monitors to check CBN coatings
- CNC controls providing for use of bigger, faster motors
- Multi-thread worm wheel type finishing machines to increase production
- Means of eliminating surface damage
- A program to determine surface damage by test rather than by Nital etching.

There are other means of manufacturing hard finished gears; i.e., the use of a carbide tool by the skiving hobbing principle and by the skive shaving principle. Skiving hobbing is slow and not competitive with the use of CBN wheels. Skive shaving shows few demonstrable production results. Therefore, these methods of hard gear finishing are not described above.

A major U.S. corporation has hosted an informal meeting to exchange views on the development of hard gear finishing using CBN coated tools. This meeting was attended by representatives from automotive and tractor companies, as well as many others interested in hard gear finishing. The meeting disclosed that Japan's consumption of CBN is well ahead of the USA's and Europe's. Further, because higher accuracy of gears was required and conventional grinding of gears was too expensive, a Japanese car manufacturer designed and built ten hard gear finishing machines. These machines are reported to use 12" diameter electroplated CBN wheels that run at 15,700 sfm. The configuration of these machines was based on existing "threaded wheel" gear grinding machines. Because of the close correlation between surface speed, production rates and tool life for hard gear finishing, it will be of interest to learn more about these machines.

Conclusions

(a) From data obtained through tests and production runs on existing machines, it is clear that higher surface speeds should be used for hard gear finishing tools. This would increase tool life as well as production rates.

(b) Higher surface speed and more powerful stepping motors must wait for new developments in CNC control equipment. Meanwhile, it is possible and desirable to design a machine with a single drive motor to drive the hard gear finishing tool spindle and the workpiece spindle through change gears.

(c) Grinding the roots and fillets of teeth by hard gear finishing will strengthen gears by a smooth blend of the fillet with the profile, which will leave tensile stresses in compression.

(d) Hard gear finishing should be used to finish internal gears. With the use of machines now available on the market this is possible.

(e) Accuracy after hardening is equal to AGMA 13. This means accuracies of AGMA 9 or 10 are easily available with the use of these production methods.

(f) Oil as a coolant supplied with medium pressure at high volume will increase wheel life and improve surface finish.

(g) Chip separation is important. A filtration system to remove particles larger than 5 microns will improve tool life and lower the power required to drive the wheel.

(h) Up time is important because it affects both direct labor cost and production rates. Different types of hard gear finishing tools should be considered with this factor in mind.

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Automated Acoustic Intensity Measurements and the Effect of Gear Tooth Profile on Noise

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Abstract:

Acoustic intensity measurements were made at NASA Lewis Research Center on a spur gear test apparatus. The measurements were obtained with the Robotic Acoustic Intensity Measurement System developed by Cleveland State University. This system provided dense spatial positioning and was calibrated against a high quality acoustic intensity system. The measured gear noise compared gearsets having two different tooth profiles. The tests evaluated the sound field of the different gears for two speeds and three loads. The experimental results showed that gear tooth profile had a major effect on measured noise. Load and speed were found to have an effect on noise also.

Introduction

The NASA Lewis Research Center investigated the effect of tooth profile on the acoustic behavior of spur gears through experimental techniques. The tests were conducted by Cleveland State University (CSU) in NASA Lewis' spur gear testing apparatus. Acoustic intensity (AI) measurements of the apparatus were obtained using a Robotic Acoustic Intensity Measurement System (RAIMS). This system was developed by CSU for NASA to evaluate the usefulness of a highly automated acoustic intensity measurement tool in the reverberant environment of gear transmission test cells.

The purpose of this article is to report on the results of noise tests of two different spur gear profile configurations which included a total of 12 different speed and load conditions. Also, the useful features of an automated acoustic intensity measurement system are demonstrated through the presentation of the test results.

RAIMS

RAIMS consists of a two-channel spectrum analyzer (FFT), a desktop computer, an instrumentation robot arm, a digital control unit for the robot and an acoustic intensity probe as shown in Fig. 1. The computer, analyzer and digital control unit module are connected via an IEEE-488 interface bus to provide computer coordination of the robot and data acquisition system. A description of the components and an evaluation of this automated system have been reported by Flanagan and Atherton.[1,2]

RAIMS measures acoustic intensity by the two microphone techniques, utilizing the imaginary part of the cross-power spectrum. Other researchers[3,4] have used this technique for

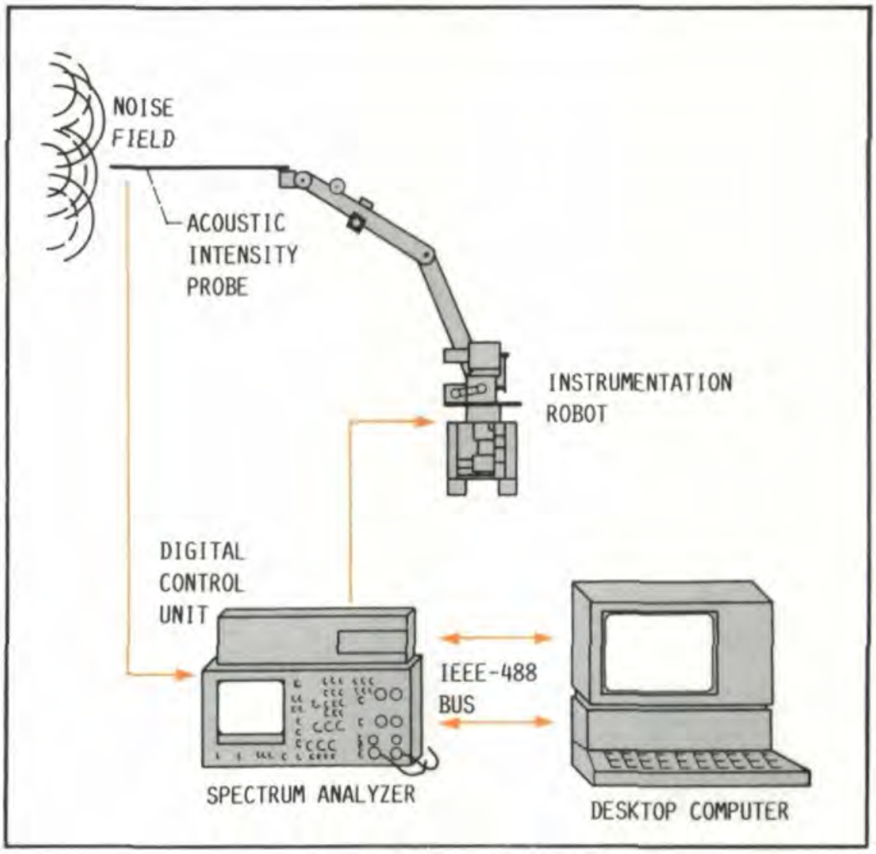

Fig. 1—Schematic of the RAIMS system

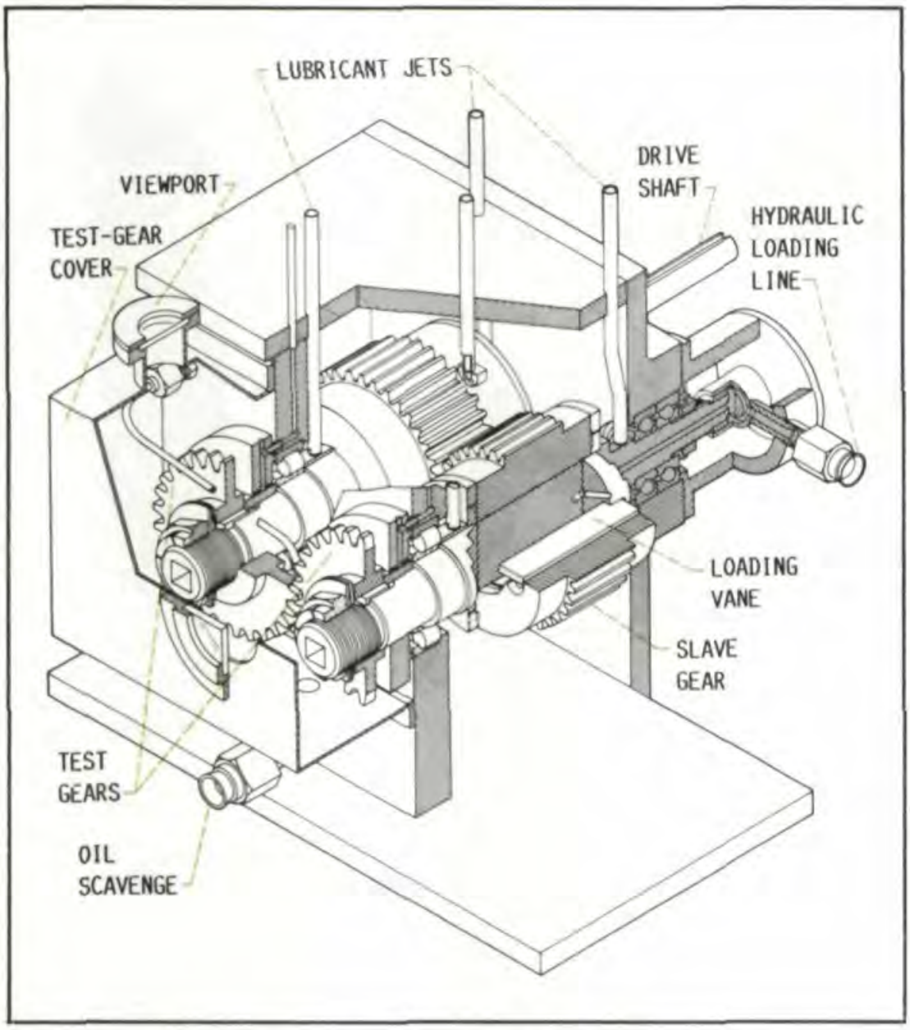

Fig. 2—NASA Lewis Research Center's gear fatigue test apparatus.

acoustic measurements in reverberant environments and have investigated the source and effect of various measurement errors in this method.[5,6] The microphones and the instrumentation consist of high quality, commercially available equipment. Because the acoustic intensity algorithm was programmed in the desktop computer, the system was calibrated against a commercially available, precision system which computes acoustic intensity directly. This process involved the comparison of the acoustic intensity from a noise source by the two systems and provided verification and calibration of RAIMS in the frequency domain.

Acoustic intensity is the net flow of sound power per unit area as measured at a point in space. It contains both magnitude and direction information and is generally presented in the frequency domain to display the frequency content of the intensity vector. Measurements at points which form an inclusive envelope around a noise source can provide information on the source location and total emitted sound power. The total sound power is a useful quantity because it is a characteristic of the noise source and is unaffected by the environment.

The AI emitted from the spur gear test apparatus depends on the nature of the excitation and the manifestation of the surface vibration into the acoustic farfield. Surface mounted accelerometers are frequently used to identify vibration amplitudes, but they cannot characterize the noise field phenomena. Also, accelerometers are limited to measuring the vibration at the attachment point to the structure. At the beginning of testing, it is difficult to determine the accelerometer placement to pick up the most dynamically active points of the structure. Accelerometers on thin, flexible housings can mass load the structure and corrupt the dynamic response as well as the resulting acoustic field. Sound pressure measurements from single microphones can also be biased by the reverberation and acoustical absorption characteristics of the surrounding environment. Because acoustic intensity is a vector quantity and does not measure standing waves, it has been shown to be a viable technique to characterize radiated sound power and identify acoustic sources in reverberant environments.

Apparatus and Test Hardware

The experimental noise tests were performed in the NASA Lewis Research Center gear fatigue test apparatus shown in Fig. 2. The test apparatus is of the four-square power loop type with the torque preload supplied by a rotary hydraulic actuator built into one of the shafts. An electric motor attached at the extension of the second shaft provides the power to drive the system. Changes in load conditions are made by adjusting the hydraulic pressure on the actuator loading vanes. Speed changes are made by exchanging sheaves. The slave gears are simply supported by the bearings and the test gears are overhung cantilever fashion at the front of the apparatus. The two bearings are supported by vertically rigid mounting plates. A metal cover with a transparent viewport in the center encloses the test gears.

Two sets of gears were tested. The first set consisted of the NASA standard fatigue tester gears whose dimensions are shown in Table 1. This set of gears has tip relief starting at about 27° roll angle, which is just before the start of single tooth contact during mesh. As the roll angle increases, the tooth profile has a linear deviation from the true involute. The teeth of the second set of gears had tooth profile modifications consisting of slightly more tip relief and the addition of root relief. The gear tooth profiles were measured on an involute checking machine and their traces are shown in Fig. 3.

Test Program

The acoustic intensity measurement program was carried out at the operating conditons indicated in Table 2. Implementation of the 12 tests was accomplished by adjusting the pressure to the hydraulic actuator of the test rig and ex-

(continued on page 26)

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AUTOMATED ACOUSTIC INTENSITY . . .
(continued from page 23)

changing the sheave diameters of the input shaft. For the two operating speeds of the test, the meshing frequencies of the 28 teeth test gear and 35 teeth slave gear are indicated in Table 3.

To carry out the acoustic tests, RAIMS was placed in front of the test rig facing the test gear cover. The robot was then programmed to measure AI in the four planes to the left, front, right, and top of the cover in square patterns of 2.54 cm extent as indicated in Fig. 4. The tip of the acoustic intensity probe was held between 5 to 10 cm from the surface of the test rig. A total number of 163 scan positions were used.

Taking the average of 32 measurements, calculating the acoustic intensity, storing the data, and positioning of the robot required about one min. This automated sequence was repeated for all of the 163 spatial points of the total scan. The 163 points did not represent a complete enclosure scan. Consequently, the spatial integration of the acoustic intensity represents only a partial measurement of the total sound power. A complete closure scan was not possible due to piping obstructions and limited access space. The partial sound power is still a useful measurement for the comparison test of the two gear pairs.

Test Results

The automated test program produced a great deal of data, portions of which are presented in the following graphs. Fig. 5 shows the AI spectrum at one of the 163 points for the standard test gears operating at 10 160 rpm and 1615 N tangential load.

The spectrum of Fig. 5 shows three regions of high amplitudes which are present (to a greater or lesser degree) in all of the AI spectrums from each of the 163 measurement points. The first region extends from 500 to 1500 Hz, and is characterized by several peaks that are separated by the operating speed of 170 Hz. The high amplitudes in this region are attributable to the excitation from the bearing passing frequencies which may be amplified by the test gear cover.

The second region extends from 2800 to 3500 Hz. The amplitudes in this region are caused by the coincidence of the torsional natural frequency predicted by Mark[7] to be at 3500 Hz. Region three extends from 4300 to 6000 Hz, and it represents the contribution from the fundamental meshing frequencies of the test and slave gears. The sidebands are caused by the errors of the gear tooth profiles.

The fundamental and first harmonic meshing frequencies

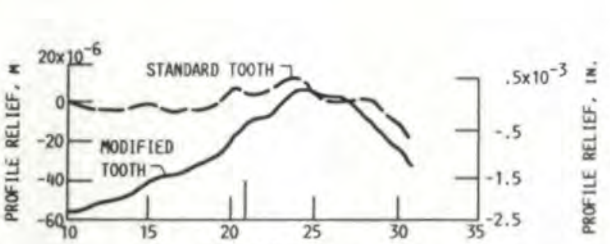

Fig. 3 – Tooth profiles of test gears (zero profile relief corresponds to a perfect involute profile).

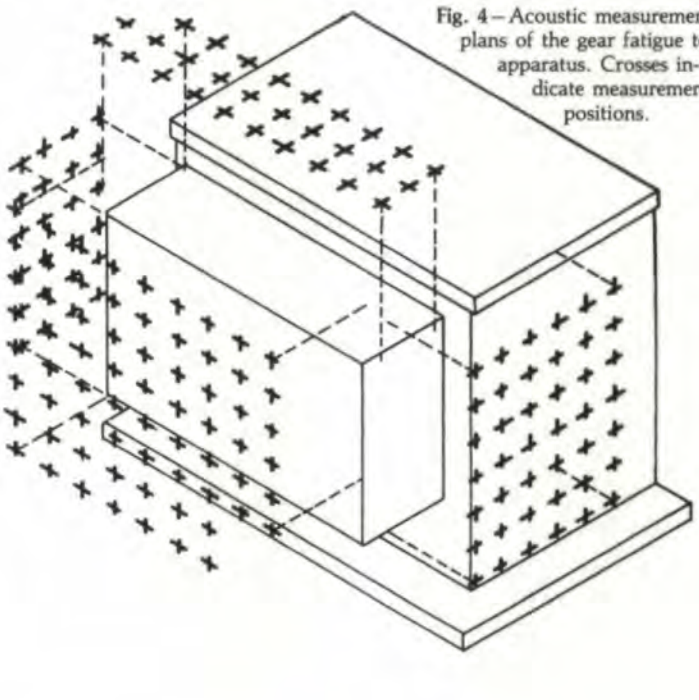

Fig. 4 – Acoustic measurement plans of the gear fatigue test apparatus. Crosses indicate measurement positions.

Table 1 — Spur Gear Data

Test gear — standard:	
Number of teeth	28
Diametral pitch	8
Whole depth, cm (in.)	0.762 (0.300)
Addendum, cm (in.)	0.318 (0.125)
Pressure angle, deg	20
Pitch diameter, cm (in.)	8.890 (3.500)
Tooth width, cm (in.)	0.635 (0.250)
Outside diameter, cm (in.)	9.525 (3.750)
Root fillet, cm (in.)	0.102 to 0.152 (0.04 to 0.06)
Measurement over pins, cm (in.)	9.603 to 9.630 (3.7807 to 3.7915)
Pin diameter, cm (in.)	0.549 (0.216)
Backlash, cm (in.)	0.0254 (0.010)
Tip relief, cm (in.)	0.001 to 0.0015 (0.0004 to 0.0006)
Test gear — Modified: see Fig. 3 for modifications	
Slave gear	
Number of teeth	35
Diametral pitch	10
Tooth width, cm (in.)	3.81 (1.5)

Table 2 — Load and Speed Conditions

Test number	Test gear	Tangential load, N (lb)	Speed, rpm
1	Standard	1615 (363)	10 160
2		1615 (363)	7 470
3		1292 (290)	10 160
4		1292 (290)	7 470
5		969 (218)	10 160
6		969 (218)	7 470
7	Modified	1615 (363)	10 160
8		1615 (363)	7 470
9		1292 (290)	10 160
10		1292 (290)	7 470
11		969 (218)	10 160
12		969 (218)	7 470

Table 3 — Meshing Frequencies

Operating speed, rpm	Test gear, Hz	Slave gear, Hz
10 160	4741	5927
7 470	3486	4358

of the test gear are at 4741 and 9482 Hz. The fundamental mesh frequency of the slave gear is at 5927 Hz. Note the strong presense of the slave gears. This is not unexpected, since the test apparatus is used for fatigue testing and the slave gears are lightly loaded.

An indication of the housing dynamic behavior can be obtained by plotting lines of constant intensity at a given frequency for a complete measuring plane. Fig. 6 shows such an iso-intensity plot for the right side of the spur gear testing apparatus using the same operating condition as in Fig. 5 at 5927 Hz. This planar representation shows concentration of high and low amplitudes across the plane which appear to derive their origin from the structural dynamics of the housing.

Figs. 7 to 10 are the results when, at a given load/speed condition, the acoustic sound power of the scanned area is

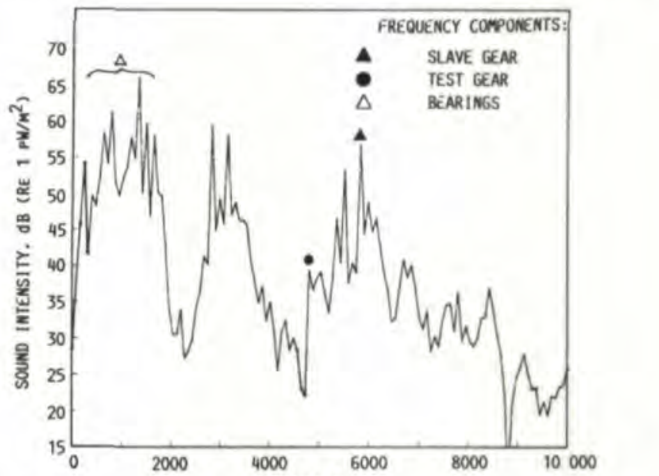

Fig. 5 — Frequency spectrum of standard gear in middle of right side — at 10 160 RPM and 1615 N load.

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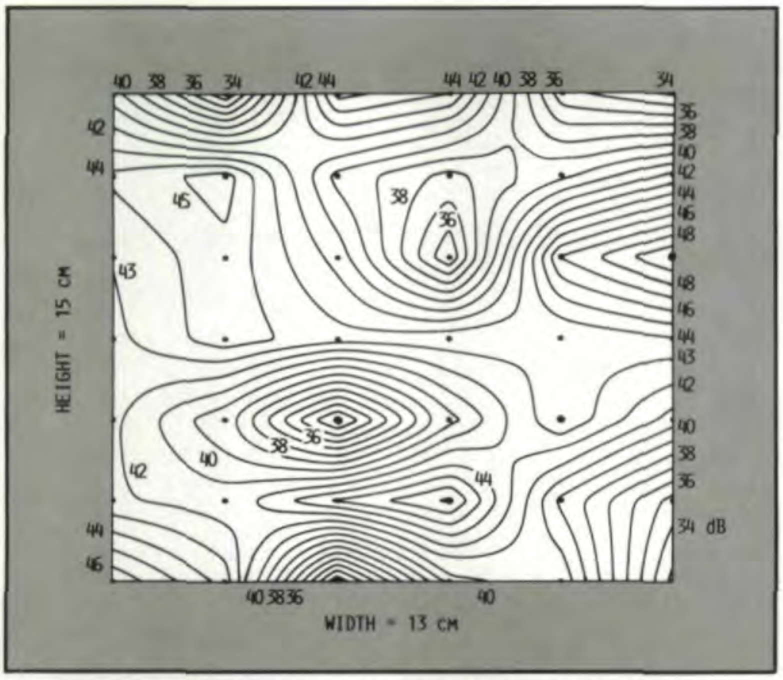

Fig. 6 – Iso-intensity plot of the right scan at 5927 Hz for the standard gears, 10 160 RPM, and 1615 N tangential load.

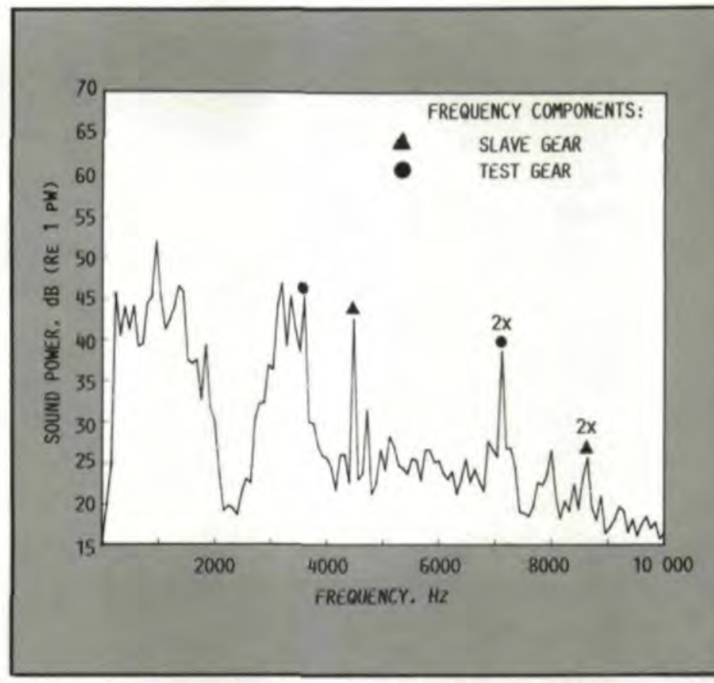

Fig. 8 – Sound power for the four planes investigated (conditions: standard gears, 7470 RPM, 1615 N).

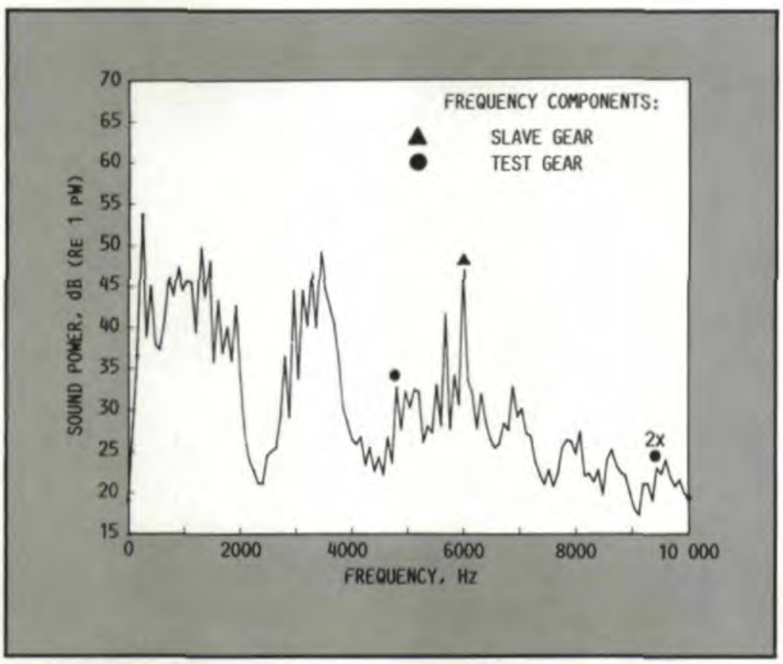

Fig. 7 – Sound power for four planes investigated (conditions: standard gear, 10 160 RPM, 1615 N).

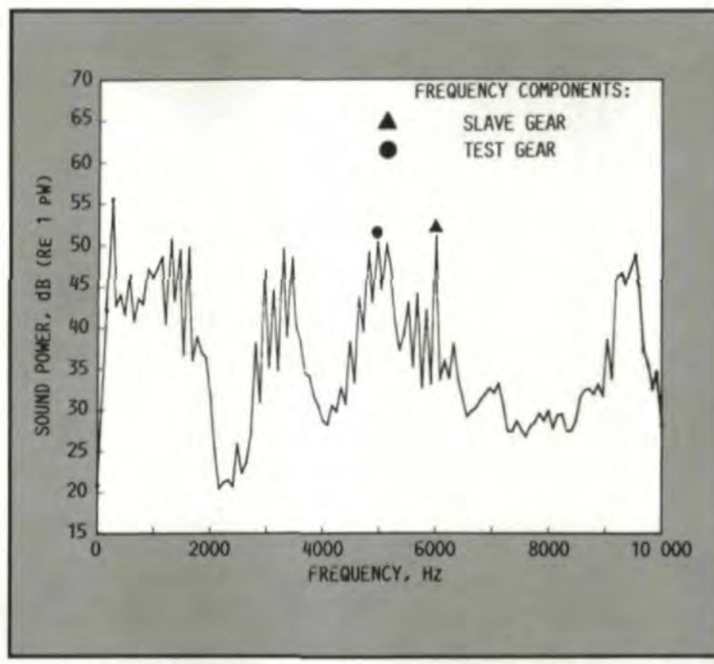

Fig. 9 – Sound power for the four planes investigated (conditions: modified gears, 10 160 RPM, 1615 N).

determined. The plots show the sound power from standard and modified gears at the two speed conditions and 1615 N load.

Notice that the gear mesh and sideband frequencies and bearing passing frequencies are still present. Inspection of the four plots shows the strong signal from the slave gears and bearings. For the standard test gear cases, the amplitude increased by 5 db from the low to the high speed tests at the slave gear fundamental frequency (Figs. 7 and 8). For the modified test gear cases, the increase was nearly 10 db at the slave gear fundamental frequency (Figs. 9 and 10). The cause for this higher increase could be the influence of the modified test gears, which have a relatively high amplitude, on the slave gears (cross-coupling).

The standard test gears show a 10 db increase in amplitude from the high to the low speed test (Figs. 7 and 8). The explanation is that the test gears operated near the predicted torsional natural frequency range of 3500 Hz at the low speed tests. Finally, this effect is noticeable even in the modified gear test data. The difference between the amplitudes of the standard and modified test gears at the high speed test is nearly 16 db.

Fig. 11 is a comparison of the six tests performed for each gearset. The measurements indicate a significant difference

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  - e) Speeds — Feeds
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  - a) The Hobbing Machine
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  - g) Machines
- B. Rolling

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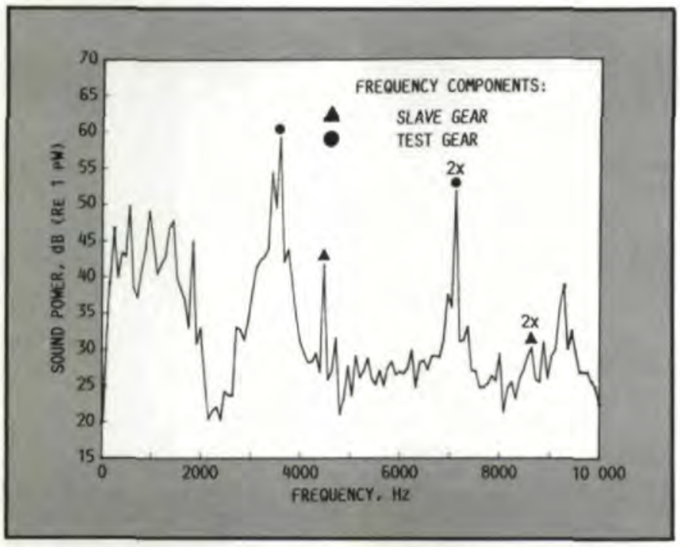

Fig. 10 – Sound power for the four planes investigated (conditions: modified gears, 7470 RPM, 1615 N).

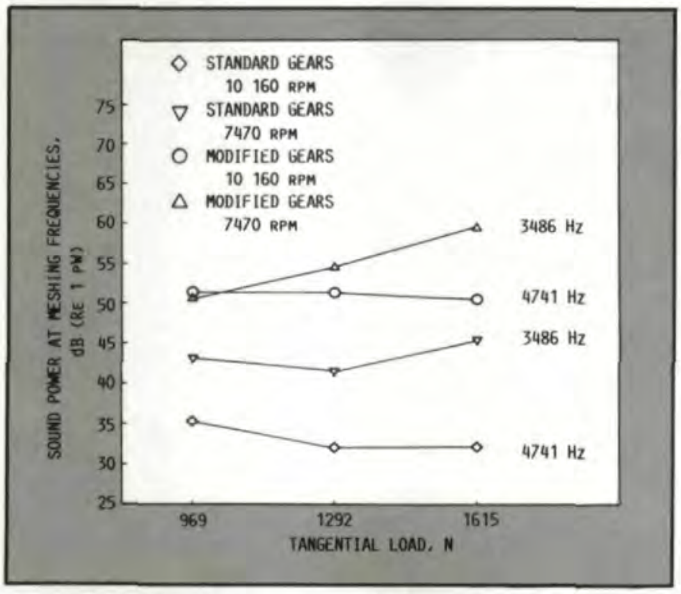

Fig. 11 – Sound power versus mesh frequencies and loading.

in the performance of the standard and modified gearsets and slight variations due to load and speed.

Conclusions

Review of the experiments and the theories for acoustic intensity and spur gears leads to the following conclusions:

1. Acoustic intensity identified dominant frequencies.

2. Robotic acoustic intensity measurements allow determination of total sound power from a noise source not withstanding the difficulty in getting around obstructions.

3. The acoustic intensity method can locate "hot spots" (surface sources and leaks). However, the measured acoustic intensity is related only to the surface phenomena while the item of interest is the source excitation. Identification of the excitation from the surface phenomena is dependent upon the dynamic behavior of the structure.

4. The test data indicates that the modified test gears are noiser than standard test gears. This shows the marked sensitivity of gear noise to the influence of tooth profile.

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Contact Surface Topology of Worm Gear Teeth

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Abstract:

In a mating worm and worm gear set, the inspection of the worm member is accomplished by available analytical inspection procedures. The mating enveloping worm gear with its warped tooth surfaces is generally accepted by the contact pattern developed while running the gear with a qualified worm. These patterns will only show that area with a minimal separation between the worm and worm gear tooth surfaces and the actual separation beyond the contact area are unknown.

A mathematical modelling procedure has been developed to predict the initial contact pattern, as well as the surface separation topology over the entire worm gear tooth surface. Equations and procedures are presented to permit an analysis for any gear set.

Gear and tool design parameters can be studied in relation to the computed results in advance of actual cutting of the components.

Introduction

Among the various types of gearing systems available to the gear application engineer is the versatile and unique worm and worm gear set. In the simpler form of a cylindrical worm meshing at 90° axis angle with an enveloping worm gear, it is widely used and has become a traditional form of gearing. (See Fig. 1.) This is evidenced by the large number of gear shops specializing in or supplying such gear sets in unassembled form or as complete gear boxes. Special designs as well as standardized ratio sets covering wide ratio ranges and center distances are available with many as stock catalog products.

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Fig. 1 – A typical industrial speed reducer gear set using a ground form worm and a bronze gear.

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or more. Reduction ratios in the range of 4 to 1 and 400 to 1 are possible. While a large percentage of the worms are single thread or single start, it is not uncommon to see ten or more threads in the worm. Applications range from very precise drives, such as dividing heads and indexing tables, through power drives and the less precise situations requiring motion direction changes or adjusting purposes.

Examples of the applications of this type of gearing are: Speed reducers, indexing tables, positioning tables, screw jacks, hoists, passenger and freight elevators, machine tools, capstans, conveyors, tensioners, actuators, stoker drives, printing machines, antenna drives, electric clocks, floor polishers, food processors, irrigation drives, speedometer drives, washing machines, waste water processors, valve operators and mining machines.

The enveloping worm gear tooth surface that is required in the set creates a rather unusual manufacturing problem. The warped tooth surface must comform very closely in conjugate action with the worm so that good load support, long wear life and smooth transmission of motion are maintained. In fact the one advantage of the worm gear set over a simple crossed axis set of helical gears, is load carrying ability. A throated worm gear set can carry some 15 to 20 times the load of a comparable set of crossed axis helicals. This is attributable solely to the larger area of contact available with enveloping worm gears.

The Worm Member

There are five popular ways to make a worm.

Thread chasing. A straight sided tool is positioned in a lathe and tranversed axially through the turning worm blank. If the tool cutting edge lies in the worm normal plane that is normal to the worm lead angle, a "chased helicoid" will be formed. If the cutting edges of the tool lie in the axial plane of the worm, an Archimedean or common screw thread is formed. This method is not too popular if a ground worm surface is needed, since the grinding wheel shape becomes geometrically complex and difficult to dress.

Thread milling. This method utilizes a special lathe which has a milling cutter drive head set in place of the tool post. As the worm is turned the milling head is passed axially through the worm, developing a lead. The double conical vee form of the cutter produces a worm called the "thread milled helicoid." This method is far more productive than thread chasing, but still lacks somewhat in accuracy or finish on the worm.

Thread grinding. In place of the relatively small cutter used in thread milling, a larger double conical vee shaped grinding wheel is used. The machine motions are the same as above, and a "thread milled helicoid" is produced. With the need in many cases for a good smooth finish on the worm member, this is a popular way to produce worms. Besides good finish, accuracy is also available with the worm surface in a fully hardened state.

It is possible to use a special dressed form on the grinding wheel other than the straight vee form, so that an involute helicoid will be developed on the worm. Or alternately, a single flank grind could be used with the flat side of a grinding wheel generating the involute helicoid. The involute helicoid is chosen sometimes as the design basis of the worm, and in some countries it is the standard form.

Hobbing. If the worm has two or more starts, it becomes possible to consider regular hobbing practices for manufacture. A gear hobbing machine and a hob are employed to cut the worm, and the result is a generated involute helicoid worm. Considerations for the resulting hobbed finish and accuracy must be made. This is a productive way of making multiple start worms.

Roll forming. With the capability of very high productivity the use of roll forming of worms has become popular. Because of the large relative size of the rolling dies compared to the

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worm, the resulting worm is usually a very close fit to an involute helicoid form. However, since roll forming relies on the plastic deformation of the worm material, some rolling die development work might have to be done to produce the desired results. While the surface finish is usually quite acceptable, the chance of distortion after heat treat does exist.

The Worm Gear Member

The worm gear member is almost exclusively produced by one of the gear hobbing processes:

Radial feed. This is also known as the infeed method and uses a multiconvolution full face width hob and a gear hobber with a power infeed or radial feed cycle. This is the most common way of producing the worm gear and is the fastest method if it can be used. Almost all single thread applications are made this way. The hob is plunged to depth into the blank and is dwelled until full final cutting is completed.

Tangential feed. Whenever it is not possible to use the radial feed method, either because of the worm thread versus gear teeth relationship or because of the need for a smoother finish, the tangential feed method is used. It requires a gear hobbing machine with a tangential feed capability which travels the tool tangentially past and through the worm gear throat. The long path of tool travel requires longer cutting times than radial feeding. There are four different designs of tooling that can be used in this process.

- Multiple convolution cylindrical hob. This hob uses the machine cycle of a radial feed followed by at least one axial pitch of tangential travel.
- Multiple convolution tapered end hob. This hob is fed only in the tangential direction leading into the cut with the tapered roughing section and following with the finishing section. This combined roughing and finishing hob is favored for use with the coarser pitch gears starting at about .600 axial pitch or bigger. This hob requires more time to cut the gear than the radial-axial feed.
- Pancake hob. Pancake hobs are tangential feed tools which permit lesser tool costs, having a narrow face and only one or two finishing teeth lying in each thread. Because of fewer cutting edges tool wear is higher and gear cutting time is increased.
- Fly cutters. Having a hobbing machine capable of tangential feed permits the cutting of worm gears with very minimal tool costs as only a single finishing hob tooth or tool point is required. Of course, it also requires the maximum machine time, and the wear demands on the single point tool are high. For development work, short delivery demands or limited production of parts the method is ideal. Contact pattern is not locked in and is more controllable.

Inspection of the Worm

The accuracy of the worm thread lead and the thread spacing on multiple thread worms is sensed by a lead measuring instrument. The worm profile can be checked by using a worm and hob profile checker with a co-ordinate system, or if the worm is an involute helicoid, by a straight line check along the generatrix. The worm can be readily checked for accuracy and compared to specified values and tolerances.

Fig. 2—Basic gear rolling inspection unit with fixed 90° axis angle.

Fig. 3—Floor model instrument with a rotatable swivel feature allowing deviation from 90° axis angle.

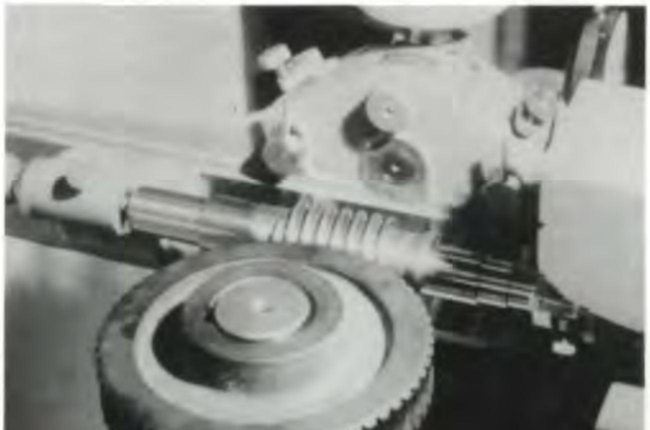


Fig. 4—Close up view of unit in Fig. 3.

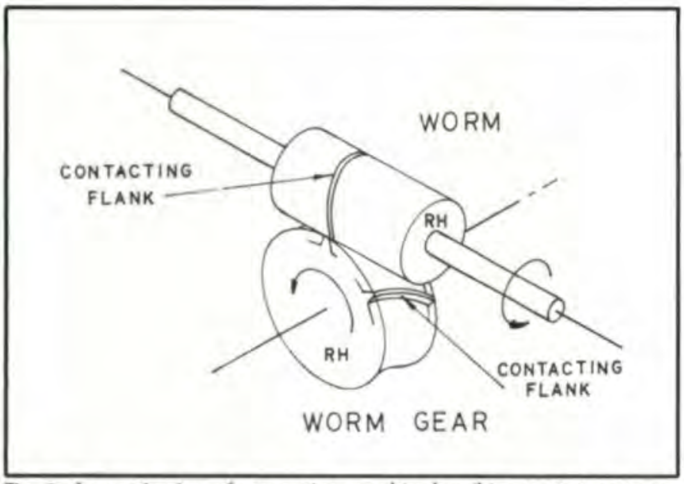


Fig. 5—Isometric view of conventions used in describing contact patterns.

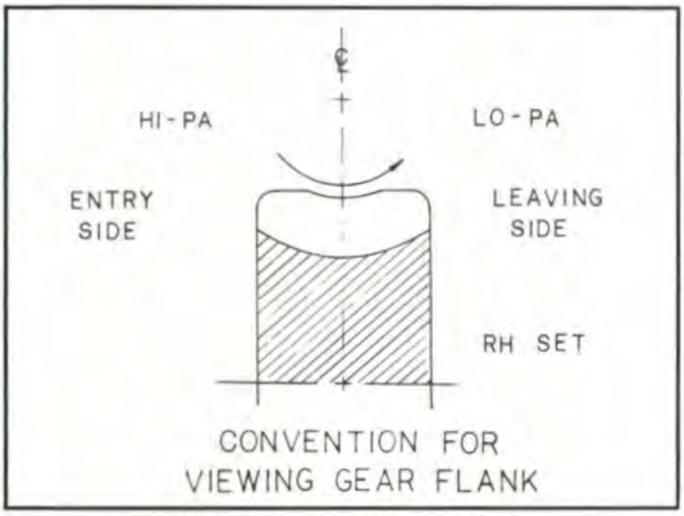


Fig. 6—Plan view of gear tooth flank.

Inspection of the Worm Gear

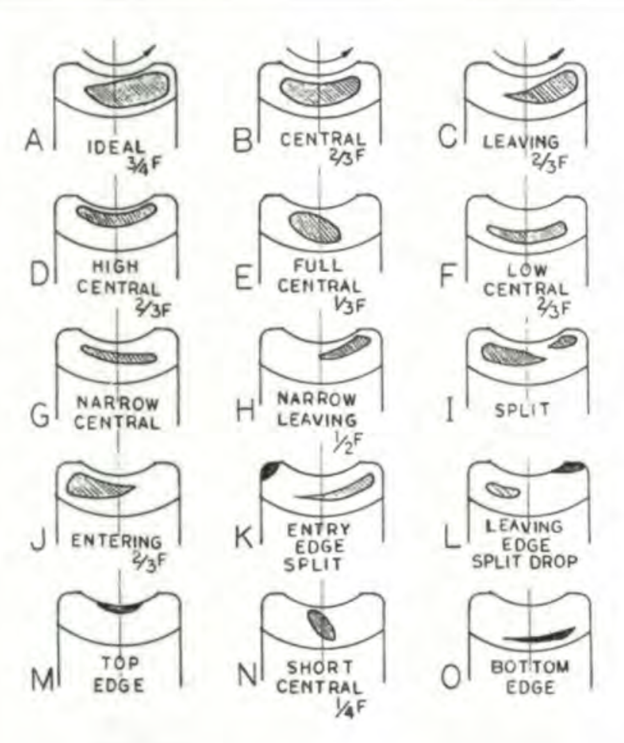
Although tooth spacing and pitch diameter runout can be readily checked analytically, there are no simple analytical methods for checking the worm gear tooth surface. Functional methods are commonly used for two reasons: to make a composite check, rolling the worm gear in mesh with a qualified production part or a master worm, measuring the total composite action and tooth to tooth action, and to develop a contact pattern. The latter check is done at a fixed center distance, with backlash, under light load, rolling in both directions. The contact pattern is enhanced by using a marking medium such as a colored pigment, coating the worm and rolling, thus, transferring the pigment to the gear to develop the contact pattern. Usually this check is performed with the gear axes at right angles, but some rolling instruments can swivel the worm support for a measure of the amount of readjustment for the hob swivel setting. For those gearsets that are sensitive to the swivel setting, this feature can reduce the trials and recuts necessary to close in on an acceptable contact pattern.

Fig. 2 shows a basic gear rolling unit with a fixed 90° axis angle. Fig. 3 shows a more complex gear rolling unit that includes a rotatable and measurable swivel, which permits departures from the 90° axis angle setting. Fig. 4 is a close up of the same instrument and shows a motor drive attachment for turning the worm.

Contact Pattern Analysis

The convention for viewing the contact pattern on a gear set is shown in the isometric view on Fig. 5 and the plan view in Fig. 6. The entering and leaving sides are identified.

Fig. 7 displays some of the contact patterns that may be encountered.



CODE: F – Face width
A – Acceptable
MA – Marginally acceptable
NA – Not acceptable

A. The ideal accepted contact pattern with clearance on the entering side which allows for the lubricant film to be carried into mesh. The contact is about 2/3 of the gear face, is mostly located on the leaving side of center and has no edge contact.
B. Center contact .... A
C. Leaving side contact .... A
D. High central contact .... A
E. Full central contact .... A
F. Low central contact .... A
G. Narrow central contact .... MA
H. Narrow leaving contact .... MA
I. Split contact .... MA
J. Entering side contact .... NA
K. Entry edge - split .... NA
L. Leaving edge - split/drop .... NA
M. Top edge .... NA
N. Short central .... NA
O. Bottom edge .... NA

Fig. 7—Description of various contact patterns.

Edge contacts are usually considered unacceptable because of possible lubricant diversion or blockage on the entering side, or because of probable broken contact or a poor rolling action on the leaving side. Top and bottom edge contacts are unacceptable and also may be accompanied by a poor rolling action.

Narrow areas of contact can cause heating and, depending on loading, may be destructive. Welding of material between gear and worm may occur. Worm gear sets involve a significant degree of sliding in the contact zone and lubrication is highly important.

For high precision drives not only is good rolling action desired, but also a substantial area of contact to assure long consistent accuracy.

The Worm Gear Cutting Procedure

The initial set up of the worm gear hobbing machine is always tentative until the first piece can be inspected and passed. If necessary, adjustments are made in the hobbing machine settings. For inspection, the gear is transferred from the gear cutting machine to a gear rolling tester where it is run against a test worm. Observing the contact pattern on both flanks can help decide if a centering or hob swivel change is necessary. Traffic continues between the cutting and checking machines until an acceptable pattern is established. The process starts over if the cutting tool is sharpened or changed.

A New Approach to Contact Pattern Planning

We have now seen the framework in which most worm gear sets are manufactured. If a problem arises with the contact pattern, a good amount of time can be spent in deciding on a course of corrective action. The information seen in the paint markings may leave many unanswered questions as to the future prospects for the gear set. An analysis solely by the contact pattern can be a very frustrating matter on certain gear set configurations.

If a part of the surface does not contact, one cannot tell just how much separation exists. Thin coatings of marking materials may only sense separations of .0003 to .0005". Use of shims or feelers has proven unreliable because contact separation is a dynamic thing. Correct placement of the shim is difficult to estimate. Accelerated wear testing is expensive. Full load run in tests usually leaves questions too.

A mathematical modelling procedure is presented here to help in describing the clearances that exist between the worm and gear teeth as they rotate through mesh. Naturally this involves the cutting tool, as this is the part that establishes the worm gear tooth contacting surface. If the tool is a hob, it involves the exact design, whether based on normal or axial pitch, and includes profile modifications. The results of this procedure are called the contact surface topology of the worm gear tooth. The calculated data can be mapped along equal separation lines so contact areas may be predicted and observations of future contact prospects may be made. As with all mathematical modelling, the relationships are rather exact, and in reality this is seldom true. Experience has shown a good correlation between this model and real life experiences, and it can be used as a basis for some logical and practical decisions.

The Mathematical Approach

As mentioned earlier there are several different families of worms and the first step is to identify that family. We have done work on these three:

1. Archimedian or screw helicoid
2. Thread milled helicoid
3. Involute helicoid

Because of the individual geometry characteristics of each different worm family, separate programs are used for each, *but the procedure is the same. The equations will only be* given for the screw helicoid, but examples will be shown for screw and involute helicoid. Figs. 8-14 illustrate various aspects of this gear geometry. Following is a list of the nomenclature used in the calculations.

Nomenclature

AX	– Axial pitch of worm
TH	– Number of worm threads
L	– Lead of worm
D2	– Worm pitch diameter
LA	– Worm lead angle
R2	– Worm pitch radius
PN	– Normal circular pitch of worm
OS	– Hob design oversize on diameter
D7	– Hob pitch diameter
NH	– Number of hob threads
HL	– Hob lead angle
LH	– Hob lead
CD	– Actual center distance of gear set
CH	– Hobbing center distance
NW	– Number of gear teeth
R5	– Theoretical gear pitch radius
CC	– Theoretical center distance of gear set
D	– Offset plane distance
RX	– Radius to a point in offset plane
TA	– Turning angle of worm
AP	– Axial pressure angle of worm
XX & YX	– Co-ordinates of point in offset plane
PP	– Axial pressure angle at point XX, YY in offset plane
Q	– Interim value
RT	– Interim value - radians
RW	– Interim value - radians
RZ & RQ	– Polar co-ordinates of point on worm gear flank
RG	– Interim value - radians
C	– Parametric value - radians
SA	– Swivel angle of hob
AL	– Interim value - radians
DH	– Interim value
RB	– Interim value - radians
BB & X2	– Co-ordinates of trace in hob axial section
XC & YC	– Co-ordinates of trace rotated to hob axial PA

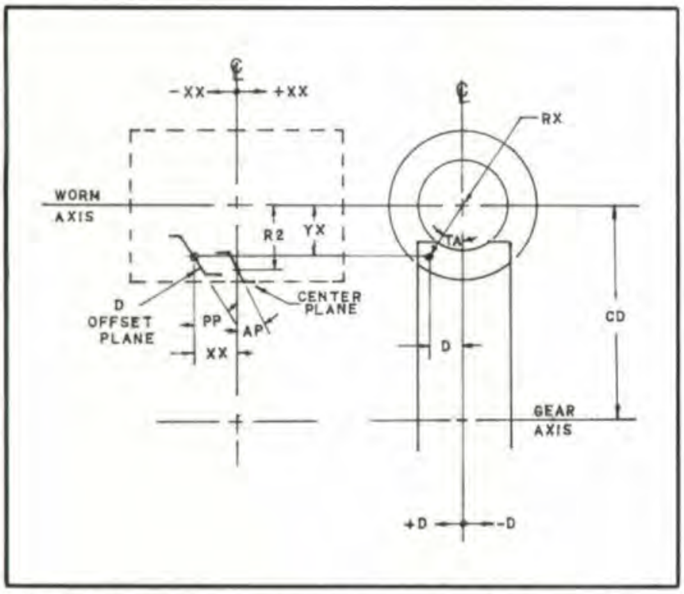


Fig. 8 – Diagram of geometry and dimensions on the worm and gear.

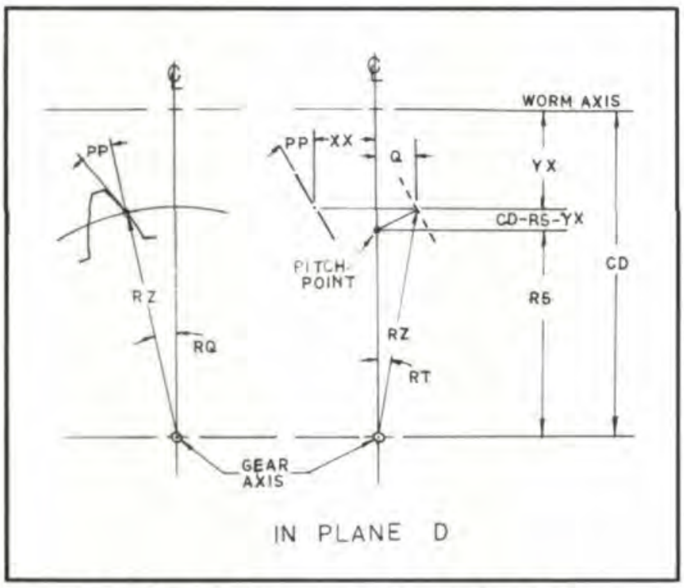


Fig. 9 – Diagram of geometry on the set in the "D" plane.

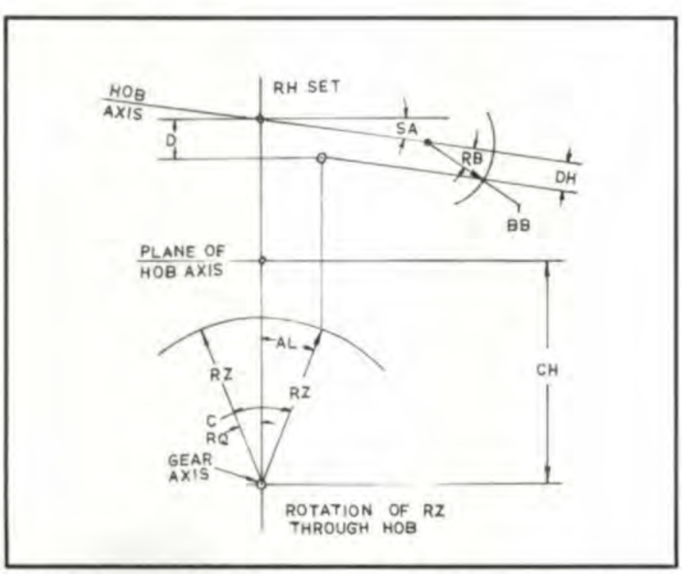


Fig. 10 – Diagram of the relationship of a point on the worm gear and the hob.

These equations are for a right hand worm and right hand gear at 90° axis angle and utilizing the normal pitch hob design method.

$$L=AX*TH \tag{1}$$
$$\text{Tan } LA=L/(D2*PI) \tag{2}$$
$$R2=D2/2 \tag{3}$$
$$PN=AX*\text{Cos } LA \tag{4}$$
$$D7=D2+OS \tag{5}$$
$$\text{Sin } HL=NH*PN/(D7*PI) \tag{6}$$
$$LH=NH*PN/\text{Cos } HL \tag{7}$$
$$CH=CD+OS/2 \tag{8}$$
$$R5=NW*AX/(2*PI) \tag{9}$$
$$CC=R5+R2 \tag{10}$$
$$YX=\text{sqr}(RX^2-D^2) \tag{11}$$
$$\text{Tan } TA=D/YX \tag{12}$$
$$XX=(RX-R2)*\text{Tan } AP-TA*L/(2*PI) \tag{13}$$
$$\text{Tan } PP=(2*PI*YX*\text{Tan } AP*RX+L*D)/(2*PI*RX^2) \tag{14}$$
$$Q=(CD-R5-YX)/\text{Tan } PP \tag{15}$$
$$\text{Tan } RT=Q/(CD-YX) \tag{16}$$
$$RZ=\text{sqr}(Q^2+(CD-YX)^2) \tag{17}$$
$$RW=(XX-Q)*2*PI/L \tag{18}$$
$$RG=RW*TH/NW \tag{19}$$
$$RQ=RG+RT \tag{20}$$
$$AL=RQ+C \tag{21}$$
$$DH=D*\text{Cos } SA-RZ*\text{Sin } AL*\text{Sin } SA \tag{22}$$
$$\text{Tan } RB=DH/(CH-RZ*\text{Cos } AL) \tag{23}$$
$$BB=\text{sqr}(DH^2+(CH-RZ*\text{Cos } AL)^2) \tag{24}$$
$$X2=LH*(RB-NW*C/NH)/(2*PI)+RZ*\text{Sin } AL*\text{Cos } SA+D*\text{Sin } SA \tag{25}$$
$$YC=(BB-D7/2)*\text{Cos } WAP+X2*\text{Sin } WAP \tag{26}$$
$$XC=(BB-D7/2)*\text{Sin } WAP-X2*\text{Cos } WAP \tag{27}$$

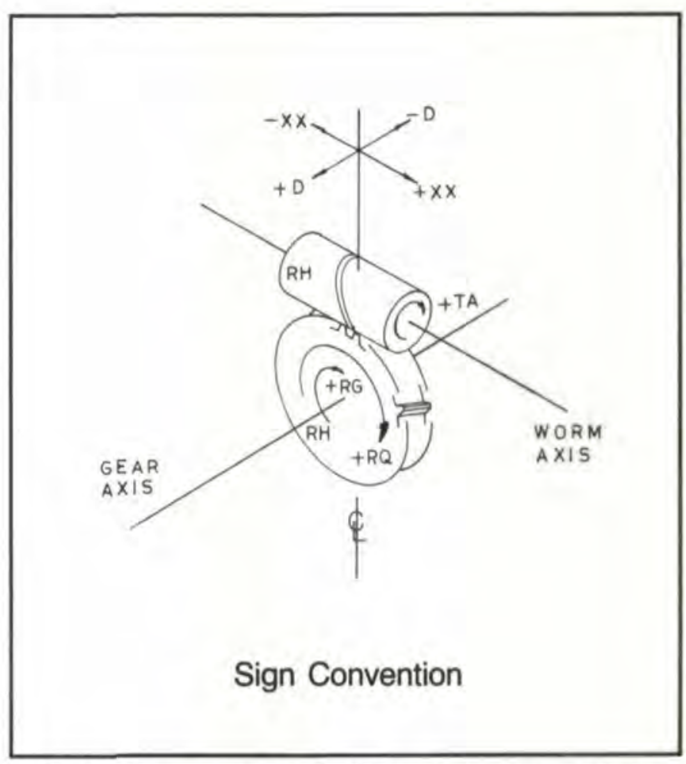


Fig. 11 – Sign convention used in the equations for worm and gear.

Analysis Procedure

To make a contact analysis all the pertinent worm, worm gear and hob data necessary to describe the set is listed. The face of the gear is sectioned by a plane offset a distance D from the centerline. To cover the entire gear face some planes are taken both to the right and left of center; that is, plus and minus D values. For a particular value of D, various values of RX, an arbitrary radius on the worm, are selected beginning with the worm outside radius. Radius RX is gradually reduced observing the value of RZ, which is on the worm gear, seeing that it remains within the limits of the gear outer radius. This point is then rotated back into the gear hob by varying the value C, tracing the point path until it passes near or through the hob helicoid surface. If a point penetrates the hob surface, material will be removed at that radius point, RZ. If the trace passes outside the hob surface, excess material will be left. Thus for each value of D & RX, a value RZ is calculated and an associated separation value is determined. Fig. 15 shows the trace of several points as they are tracked back into the hob surface. On a magnified basis the separations are easily measured, and the profile separation based on actual hob profile is determined as is shown in Fig. 16. Data is plotted in an array format as D & RZ on a gear face layout according to the convention in Fig. 17. It yields a field which can be contoured into a topological map.

The sample data (Table 1) was used in this presentation

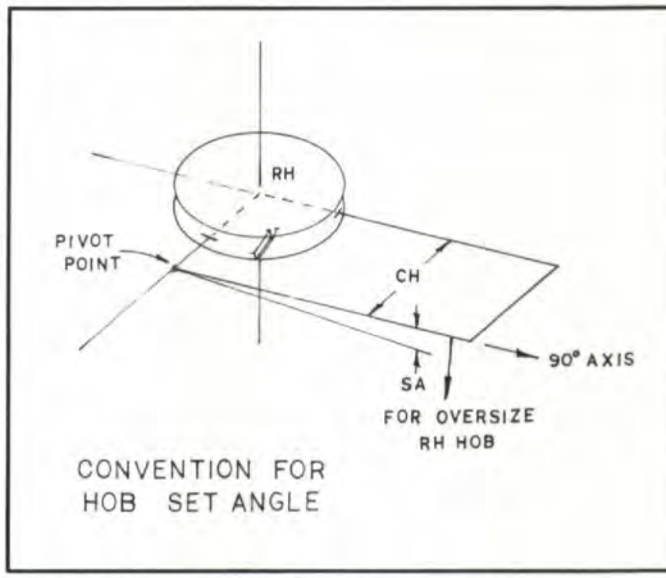


Fig. 12 – Diagram of the hob swivel or setting angle convention.

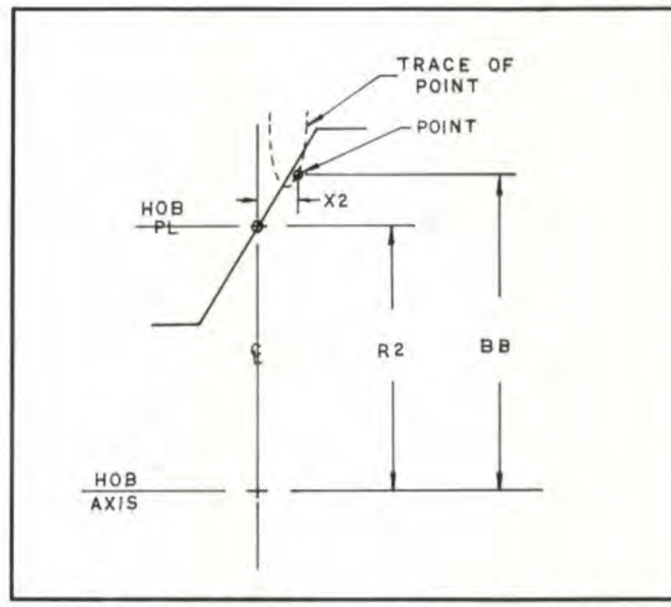


Fig. 13 – Co-ordinates of the point trace path in the axial hob section.

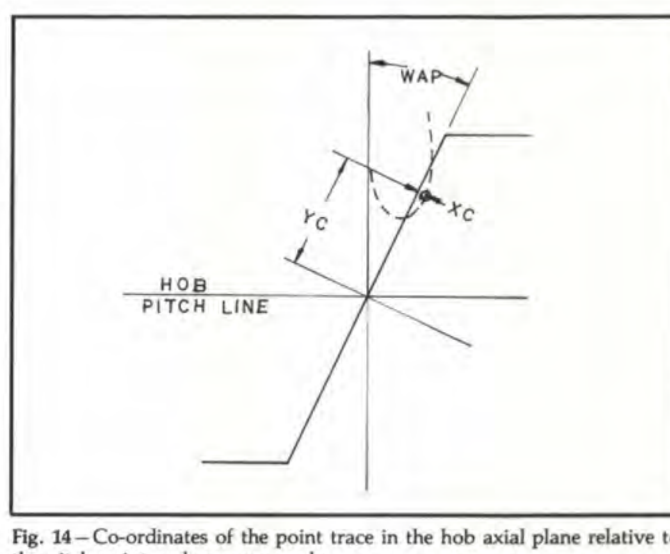


Fig. 14 – Co-ordinates of the point trace in the hob axial plane relative to the pitch point and pressure angle.

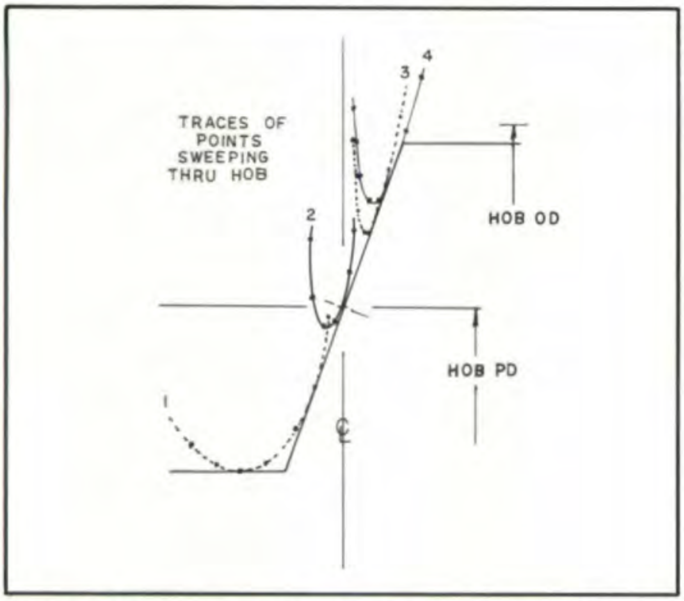


Fig. 15 – Trace of several different points sweeping through the hob.

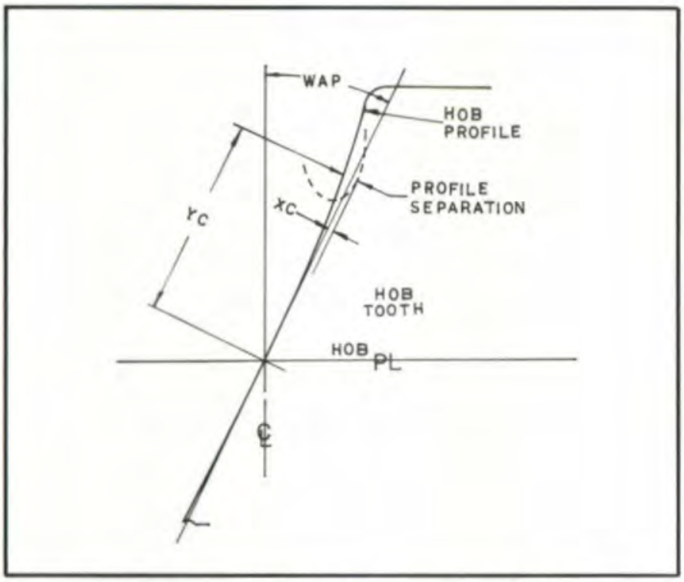


Fig. 16 – Determination of the profile separation value for a point.

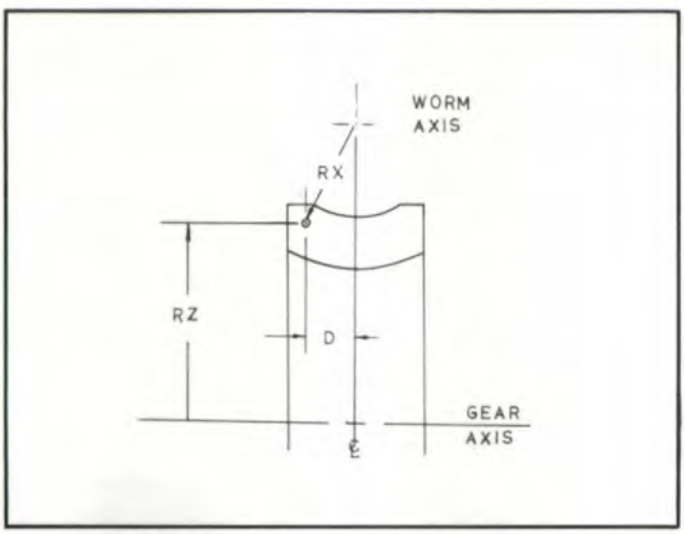


Fig. 17 – The location of the separation values on the gear flank.

Table 1 – Sample Data for Contact Analysis

Worm Data:

Thread form – screw helicoid
Axial C.P. – 1.0000
Normal C.P. – .96907
Threads – 2
Lead – 2.0000
Worm O.D. – 3.136
Worm P.D. – 2.5000
Axial P.A. – 20.0000°
Lead angle – 14.2866°

Gear Data:

Teeth – 25
Pitch radius – 3.9789
Face – 2.000
Outside radius – 4.430
Throat radius – 4.297
Actual center distance – 5.2289
Theo. center distance – 5.2289

Hob Data:

Normal C.P. – .96907
Axial C.P. – .99645
Hob oversize – .150
Hob pitch diameter – 2.65
Hob lead – 1.9929
Axial pressure angle – 19.9346°
Hobbing center distance – 5.3039
Hob lead angle – 13.46223°
Hob set angle – 0.82°
Theo. hob set angle – 0.82438°
Profile mod. – .0015 hollow

and uses a gear hob specified according to the normal circular pitch design method. The plot of separations resulting from a sequence of calculations using this data is shown in Fig. 18, displayed as an array. In Fig. 19, equal levels of separation measured from the high point are connected by lines to create the map.

Some observations concerning the expected contact area and its location can be made as well as noting the degree of separation across the entire gear face. With light test load and color transfer a .0005 level is used. With some "running-in" the .001 level can be considered. The progressive potential for future contact patterns can be assessed, either by plastic deformation or, later, by normal wear. The opportunity to see the amount of entry side easement in consideration of lubrication can be particularly helpful.

An additional cross sectional plot is useful graphically and represents the separation plot along the worm mid form trace. This is plotted just below the topological map.

The first contour map shown in Fig. 19 is for a hob with .150 oversize. Fig. 20 and Fig. 21 show a series of maps with

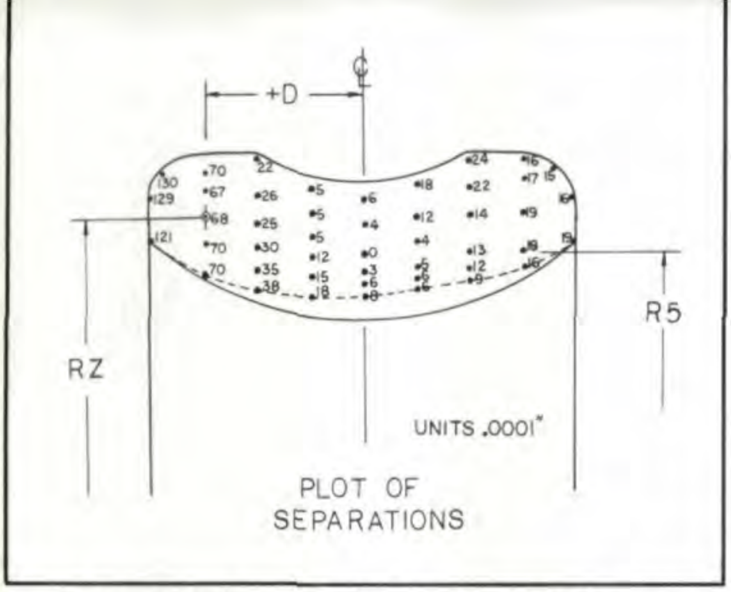


Fig. 18 – Array of separation values plotted for the sample case.

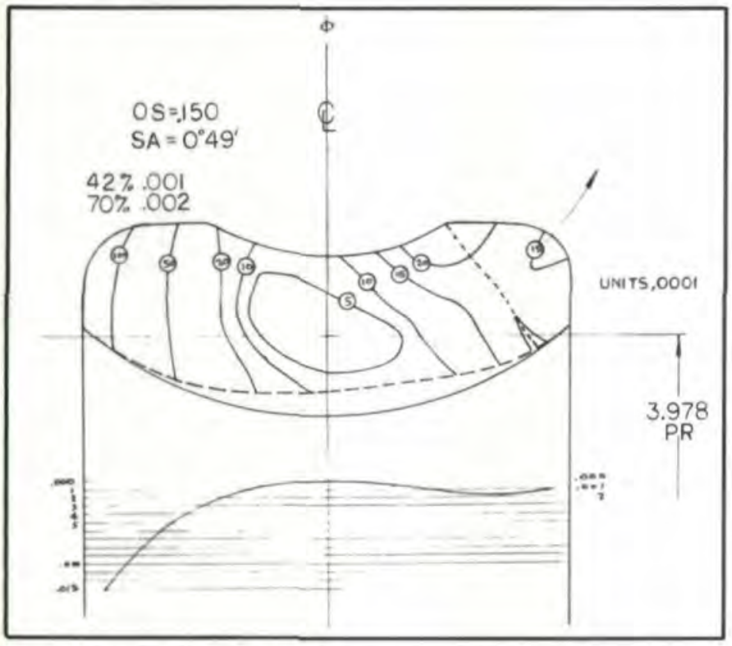


Fig. 19 – Point of equal separation are delineated, creating the separation map for .150 oversize.

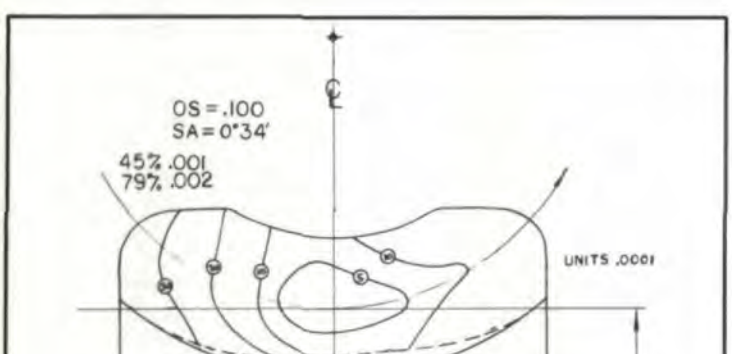


Fig. 20 – Map for same case data with hob oversize .100.

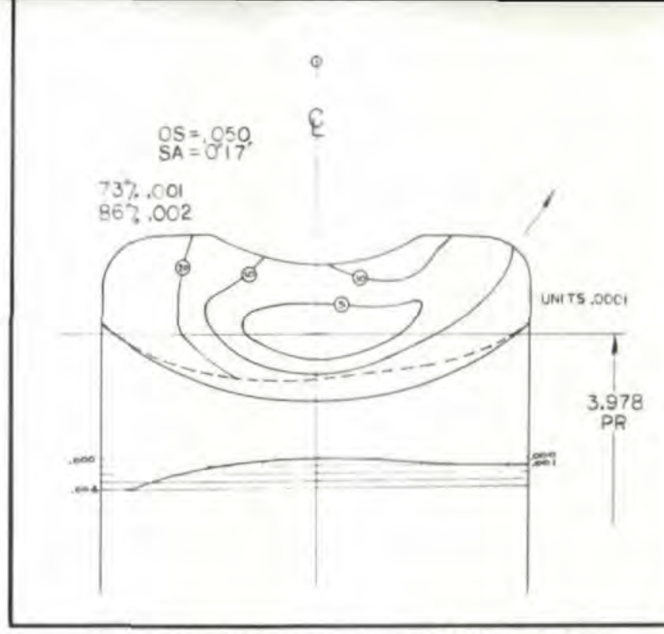


Fig. 21 – Map for same case data with hob oversize .050.

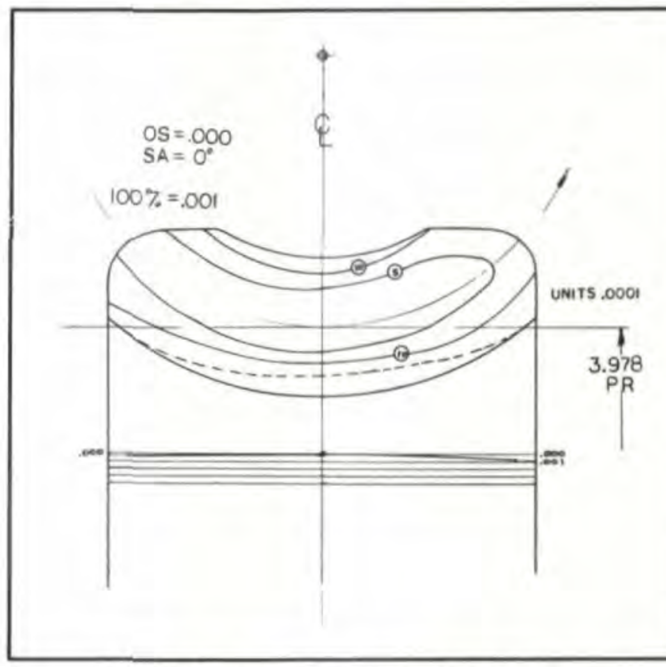


Fig. 22 – Map for same case data with .000 oversize.

.100 and .050 oversize respectively. Fig. 22 shows the results from a hob with no oversize.

General Comments

- As the oversize is reduced, the longitudinal easement reduces.
- The curvature geometry favors the entry side with increased separations.
- The leaving side frequently exhibits a cusp or valley phenomenon, which inhibits the attempt to get the contact pattern toward the leaving side of center. Small swivel adjustments may get a split or leaving edge contact.
- Low oversize hobs make for critical swivel settings, and subsequently, gear alignment in assembly will be sensitive too.

Fig. 23 is an additional case in the series in which the hob is undersize .015 on diameter, and the contact pattern develops into the well-known split.

Using an Extra Thread in the Hob

Occasionally a worm gear hob will be made with one more thread than is in the worm, as when a seven thread hob is used to cut a worm gear that mates with a worm of six threads. The hob is labelled as one thread oversize. The actual amount of oversize in this case is one-sixth of the worm pitch diameter. By usual standards this amount of oversize would be considered rather large and a significant amount of crowning or ease-off will be developed on the gear tooth. Such a case was investigated and the results are shown in the contour separation plot in Fig. 24. In this case, the hob oversize is .500 and, while the area of contact at the .002 level appears good, the entry side easement is high at .030. It will take substantial wear to develop a broad contact.

Another case of a three-thread hob being used to cut a gear mating with a two-thread worm was also investigated. The oversize is one-half of the worm pitch diameter and is quite excessive and is unusable in most applications. The contour separation plot is shown in Fig. 25. This hob has the equivalence of 1.250 oversize; the contact is a narrow band, and it will not widen very quickly with progressive wear. If loaded comparably with other throated worm gear sets, a failure is likely.

At times it would be advantageous to use the concept of the extra hob thread, particularly when the worm threads and gear teeth are of an even ratio, and where tangential feed might be required. For example, in the case of a six-thread worm mating with a 30 tooth gear, using a seven-thread hob would permit using the faster infeed hobbing method, providing the reduced area of contact is acceptable.

The extra thread design is unique in that both the axial and normal pitch and the axial and normal pressure angles are matched between the worm and hob. The hob swivel angle is normally set at zero.

Involute Worms

One of the other thread forms used is the involute helicoid. The calculations are more complex due to the curvature of the worm form and there is a greater sensitivity on the worm gear profile on the leaving side. In Fig. 26 a map of the contact is shown for a hob set at the theoretical swivel setting angle, and then to the right the same gear is mapped for a slightly smaller swivel setting. A significantly different contact pattern is produced for only an eleven minute angular change. With the first setting the gear is unacceptable, and after the adjustment, it is acceptable. The results shown here are typical of the higher lead angle worms of both involute and "thread milled" helicoids.

APPENDIX

Worm Gear Hob Oversize

With the concept that a hob should duplicate the worm in the set exactly, a problem arises as soon as the hob is sharpened and goes undersize. Besides the impractical or uneconomical aspect of a very limited hob life, the result of

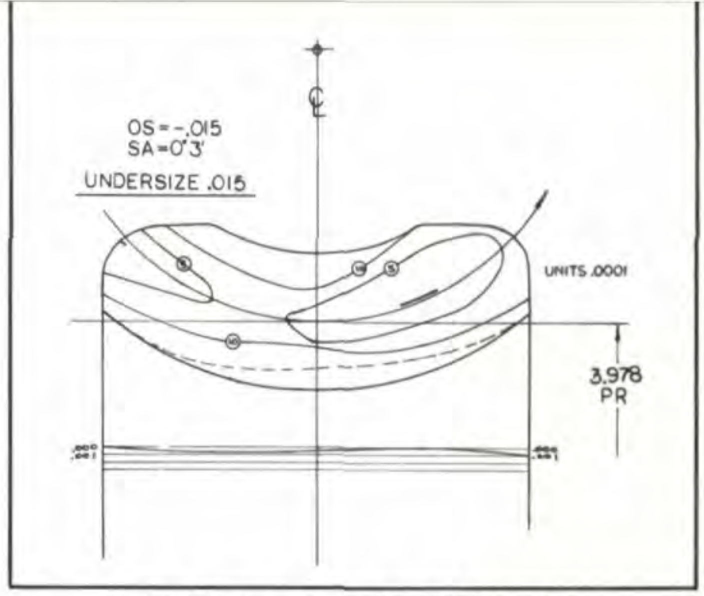


Fig. 23 – Map for same case data with .015 undersize.

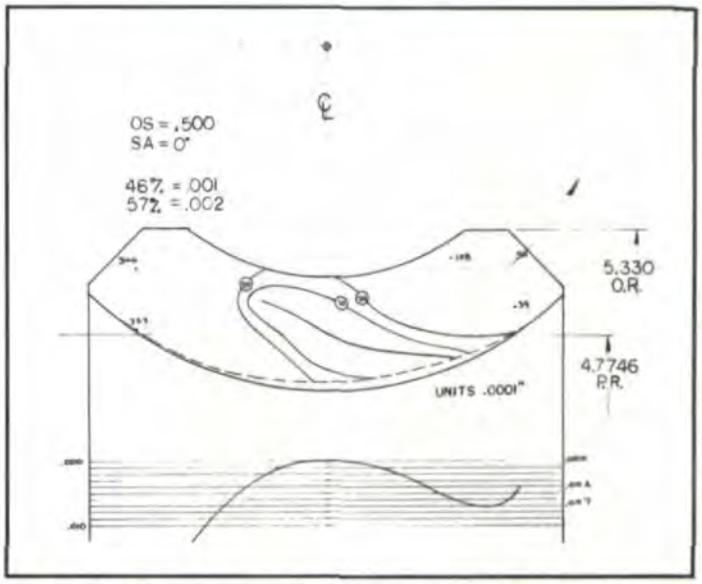


Fig. 24 – Map of one thread oversize hob. Seven thread hob, six thread worm.

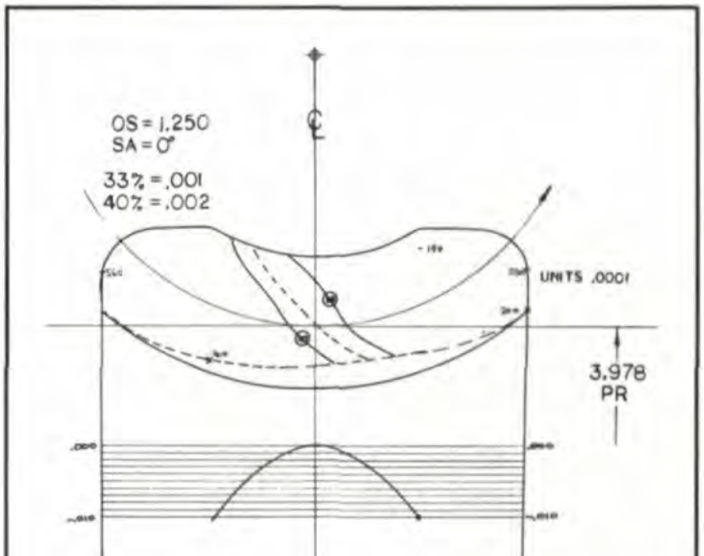


Fig. 25 – Map of one thread oversize hob. Three thread hob, two thread worm.

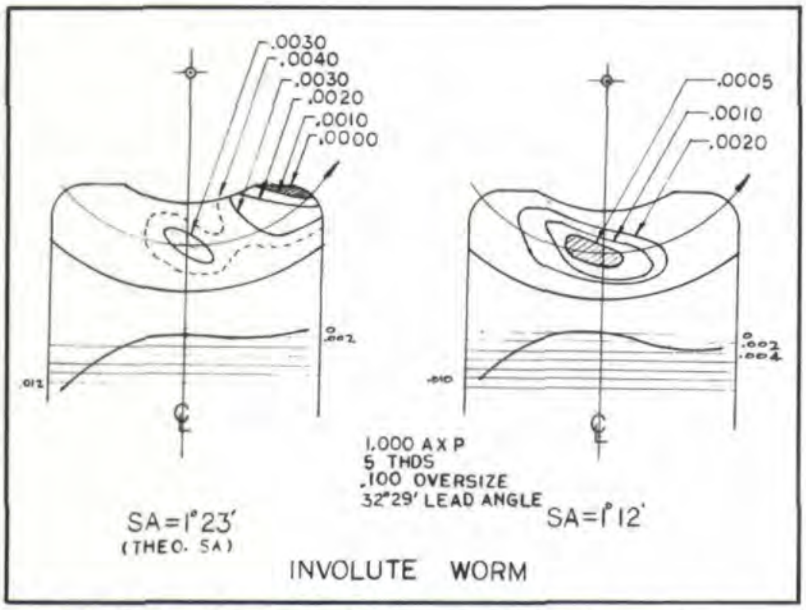


Fig. 26—Map of the same involute form worm flanks with two different swivel angles.

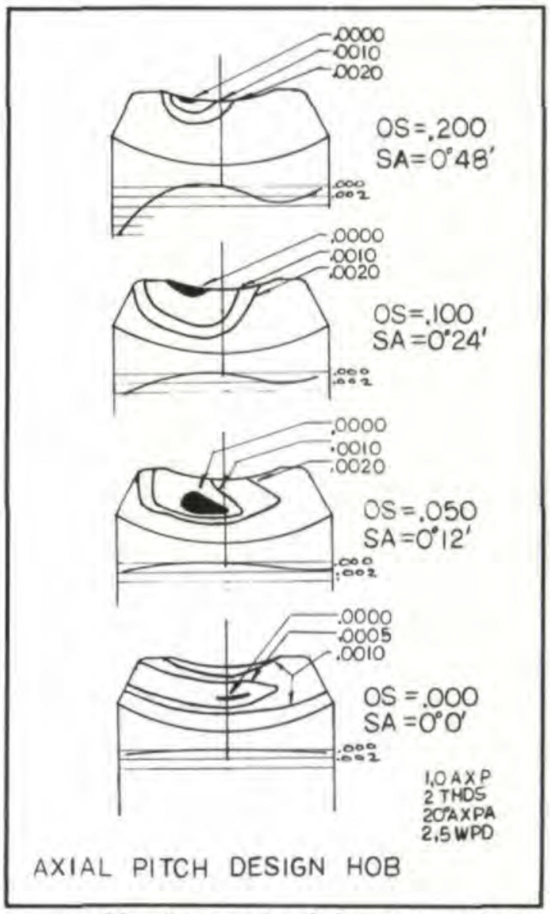


Fig. 27—Maps for an axial pitch design hob with four different amounts of oversize and the results.

using an undersize hob is the development of one of the undesirable split contact patterns.

Although a duplicate design type hob can yield the full maximum available area of contact, there is little room for error and some edge contact will very likely occur. It is obvious that some hob oversize is needed. The demands for oversize selection are in opposite directions, one asking for maximum oversize to fully utilize the hob life, and the other demanding a significant area of contact.

The Axial Pitch Design Concept

One approach to designing a worm gear hob is to match the axial pitch, axial pressure angle and number of threads with that of the worm. In essence this has the hob matching the worm exactly only when the oversize is zero. Since some oversize is always used for a new hob, it is mismatched and is long on the normal circular pitch.

The oversize is an arbitrary choice guided by recorded experience. The oversize used is made as large as possible while still maintaining a passable contact pattern on the gear. Then as the hob is gradually sharpened back, it approaches an exact match, on both axial pitch and normal circular pitch, just as the hob is nearing the scrap point. Also the area of contact increases along with the possibility of an edge contact.

Initial contact with this design is usually narrow and high, and if a little too much oversize is used, a top edge contact will result. As the hob is sharpened the contact will widen and drop towards the pitch line.

Fig. 27 shows the results for an axial pitch designed hob beginning with .200 oversize and decreasing down to .000 oversize. These computed maps show the contact progressing from the top edge down to a full bearing. The swivel settings in these examples were only 75% of that calculated. The patterns were optimized.

The Normal Pitch Design Concept

Using this approach the worm gear hob is made to match the normal circular pitch and the normal pressure angle of the worm at a preselected amount of oversize. Again the amount of oversize is selected by the hob manufacturer on the basis of experience to yield some planned amount of face contact. This design method, which has become an industry standard, presents the best contact pattern on the gear when the hob is new. As the hob is sharpened back the contact will broaden and drop.

Worm Gear Tooth Crowning

In other types of gearing it is not uncommon to apply some form of modification to the gear teeth surfaces to allow for minor errors, distortions or misalignment of axes. This crowning can also be applied to the worm gear tooth surface and is done in the longitudinal direction by the amount of hob oversize and in the radial direction by a hob profile modification to induce a tip and root flank easement.

Worm Gear Materials

With applications using a hardened steel worm and a bronze worm gear, the plasticity of the material permits a surface correction of some .002 inches during the initial break-in or loading period. Bronze has been called somewhat forgiving because of this. Other gear materials, such as cast iron or ductile iron, do not have this same property and can readily have material transfer or pick-up because of the iron against iron phenomenon. The initial contact area must be good for success with these latter materials.

(continued on page 47)

Internal Helical Gear Design

Fellows Corporation
Springfield, VT

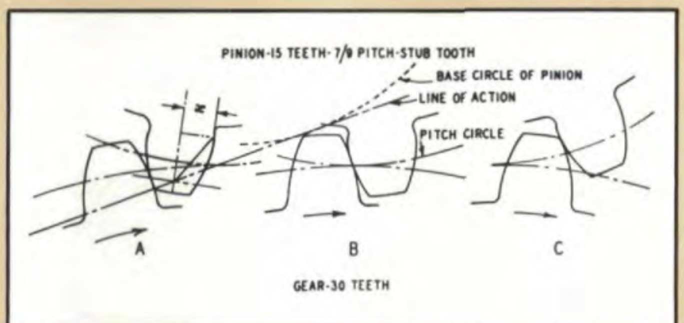


Fig. 1—Diagram showing progression of contact of spur gear teeth.

Helical gears operating on parallel axes are virtually spur gears with their teeth twisted. This twisting of the teeth makes helical gears operate more smoothly than spur gears. In a spur gear and pinion, there are three distinct phases of engagement as shown in Fig. 1. The flank of the (driving) gear tooth engages the tip of the pinion tooth; the teeth are in engagement in the vicinity of the pitch line; the tip of the (driving) gear tooth engages the flank of the pinion tooth. The teeth are only in contact at the pitch line for a small part of their engagement.

The Theory of Helical Gearing

Suppose we rivet together two spur gears, setting the teeth of one in advance of the other a distance equal to one half the circular pitch. We now have six instead of three phases of engagement. By carrying this still further and using a series of laminated gears, as shown in Fig. 2, the number of phases of engagement is greatly increased until, if it were possible to use a series of laminations of infinitesimal thickness, the pitch point of action of the gears would develop into a helix. If this helix had an advance equal to the circular pitch in the face width of the gear, we would have what is known as *continuous helical action*. It should be understood, however, that as far as the length of the line of action is concerned, this is the same as for two spur gears of the same pitch, tooth length, etc. The essential difference is that a helical gear, when properly designed and cut, is always in contact at the pitch line in some one plane.

Spur gear tooth action can be clearly seen by holding a pair of gears in the hands and rolling them together. As the teeth come into engagement, the sliding action is in one direction; then as the point of contact reaches the imaginary pitch line, the action is reversed and the teeth slide in the opposite direction. In other words, the direction of sliding is reversed on the teeth as the contact point passes the pitch point. In the helical gear, the so-called "pitch point rolling action" or pitch point contact takes place progressively from one plane, or imaginary lamination, to the adjacent plane, until the entire width of face has been covered.

"Diametral" and Normal Planes

There is one distinction between a spur and helical gear which sometimes causes confusion—it is that helical gear teeth are calculated in two planes, commonly known as the normal plane and the "diametral" plane, or plane of rotation.

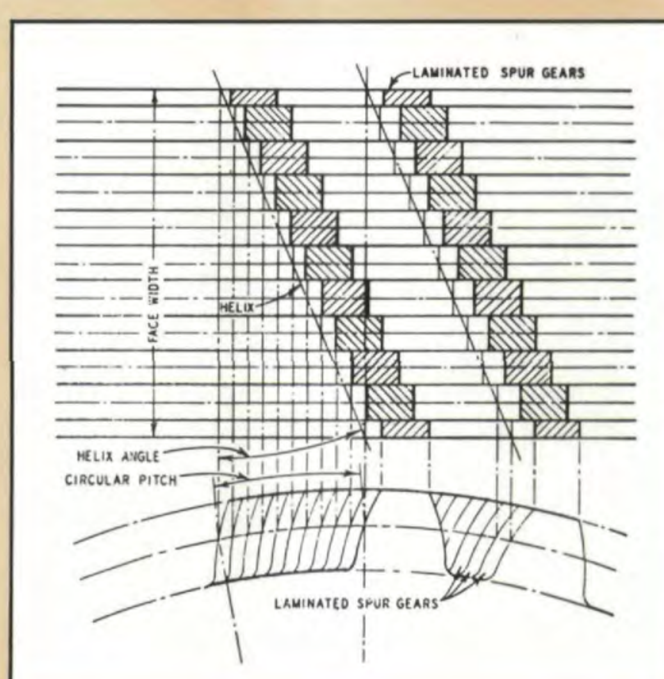


Fig. 2—Diagram showing how a series of laminated spur gears with the teeth set at an angle give similar tooth action to a helical gear.

The plane of rotation of a helical gear, as shown in Fig. 3, refers to the outline of the tooth as it is viewed from a position parallel with the face of the gear blank. The *normal plane* is the shape of the tooth lying in a plane at right angles to the helix angle of the tooth.

Relation Between Axial Thrust and Helix Angle

A certain mathematical relation exists between the helix angle and axial thrust of helical gears and is a measure of the tooth load. Reference to Fig. 4 will show that the tangential load c on the tooth—in the plane of rotation—can be resolved into two components, a and b. Component a represents the total tooth load, and b the axial thrust. By the law of triangles, a is the secant of the helix angle, and b the tangent. Therefore, the secant times the tangential load gives the total tooth load on the normal section of the tooth.

The tangent of the helix angle times the tangential load gives the axial thrust.

Example: Assume one wishes to find the total tooth load and the axial thrust on a pair of helical gears having a helix angle of 15° and carrying a tangential tooth load of 500 pounds. Total tooth load $=$ secant $15° \times 500 = 1.0353 \times 500 = 517.6$ pounds. Axial thrust $=$ tangent $15° \times 500 = 0.2679 \times 500 = 133.9$ pounds.

An increase in the helix angle, as shown at *B* and *C* in Fig. 4, increases the total tooth load and also the axial thrust, until, if the helix angle is increased to 45°, the axial thrust equals the tangential tooth load, and the total tooth load becomes 41½% greater than the tangential tooth load.

In addition to imposing a greatly increased load on the tooth, a steep helix angle also reduces the cross section of the tooth in the normal plane, and hence weakens it, making it less capable of carrying the greater load imposed. This can be clearly seen at *D*, *E* and *F*, Fig. 4, where the outlines of the tooth in the two planes (normal and "diametral" or plane of rotation) have been superimposed for comparison. A moderate helix angle is to be preferred, in order that the axial thrust may be kept within practical limits.

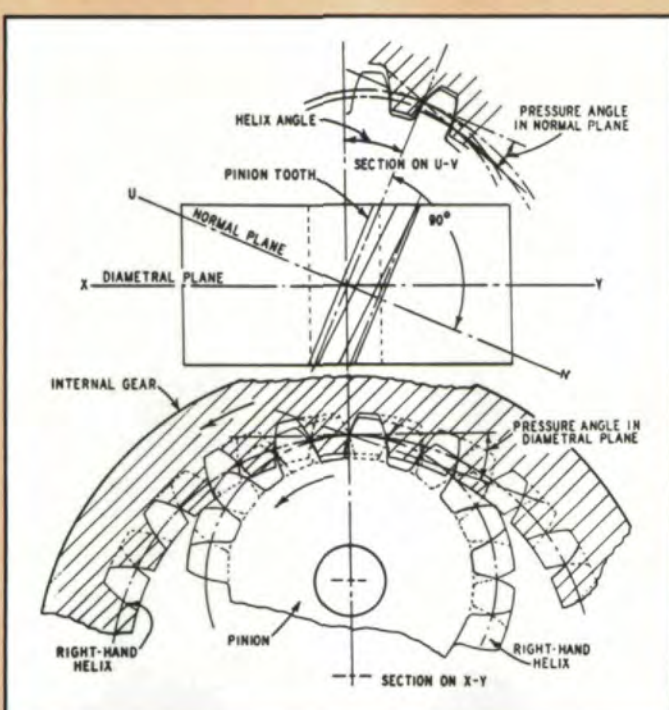


Fig. 3—Diagram giving notation of tooth parts of an internal helical gear and pinion.

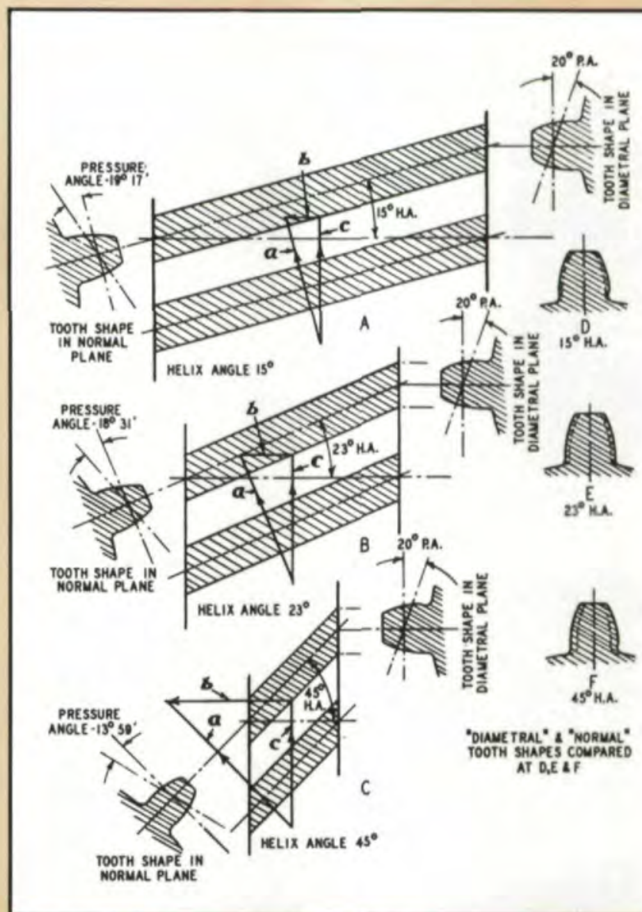


Fig. 4—Diagram showing how an increase in the helical angle increases the axial thrust. It also shows relationship between tangential tooth load, total tooth load and axial thrust.

In the foregoing analysis, the effect of friction between the teeth and its reaction to the axial thrust has not been considered. This factor is governed by so many other variables, such as lubrication, etc., that it is impossible to calculate with any degree of certainty how much friction between the teeth reduces the axial thrust.

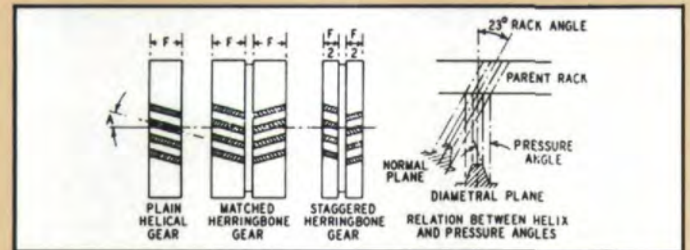
Width of Face for Herringbone Gears

The width of face for continuous helical action of helical and herringbone gears is given in Table 1. Reference to the illustration accompanying the table will show that for a herringbone gear in which the teeth are matched, the total width of face required is equal to twice the width of face of a plain helical gear, plus the width of the clearance groove. By staggering the teeth, the total width of face of a herringbone gear can be reduced, so that is equal to the width of a plain helical gear, plus the clearance groove.

Tooth Action of Herringbone Gears

In a herringbone gear, the tooth action is similar to that of a plain helical gear, with the exception that there is no end thrust exerted outside of the gear teeth themselves. This will be clearly seen in Fig. 5, where the arrows show the direc-

Table 1 – Minimum width of face for continuous helical action of helical and herringbone gears with helix angles of approximately 15 and 23°.



Diametral Pitch	Normal Pitch	Helix Angle "A"	Lead of Helix in Inches	Minimum Width of Face "F" in Inches
5/7	5.184	15°–20′	41.270	$2\frac{19}{64}$
5/7	5.456	23°–35′	25.904	$1\frac{7}{16}$
6/8	6.209	14°–55′	41.270	$1\frac{31}{32}$
6/8	6.518	23°	25.904	$1\frac{15}{64}$
7/9	7.254	15°–12′	41.270	$1\frac{21}{32}$
7/9	7.629	23°–25′	25.904	$1\frac{3}{64}$
8/10	8.279	14°–55′	41.270	$1\frac{31}{64}$
8/10	8.691	23°	25.904	$\frac{15}{16}$
9/11	9.324	15°– 9′	41.270	$1\frac{19}{64}$
9/11	9.801	23°–20′	25.904	$\frac{13}{16}$
10/12	10.349	14°–55′	41.270	$1\frac{3}{16}$
10/12	10.863	23°	25.904	$\frac{3}{4}$
12/14	12.418	14°–55′	41.270	$\frac{63}{64}$
12/14	13.036	23°	25.904	$\frac{5}{8}$

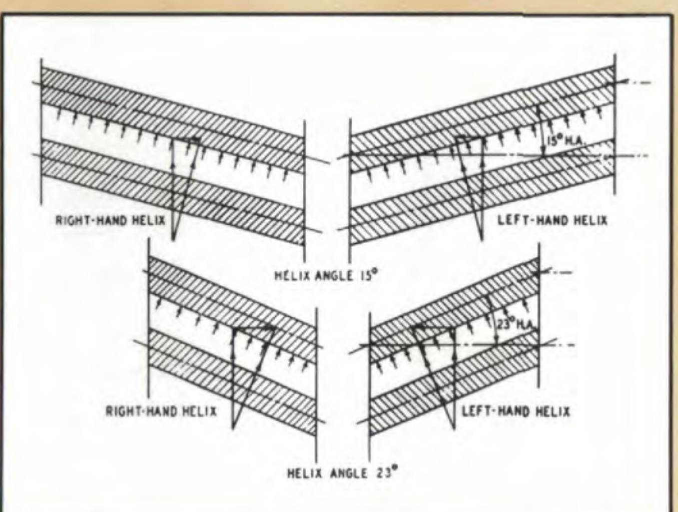


Fig. 5 – Diagram showing direction of thrusts on herringbone gear teeth.

tion in which the load thrusts are transmitted to gear teeth having helix angles of 15° and 23°.

Owing to the fact that properly designed herringbone gears are always in contact at the pitch line in some one plane, there is always rolling contact taking place at the same time as other parts of the teeth are slipping on each other.

It is advisable to keep the helix angle down to moderate angles, and practice has demonstrated that for the majority of cases, a helix angle not exceeding 23° gives the best results.

Calculating Tooth Parts of Helical Gears

The tooth parts of a helical gear, in the "diametral" plane or plane of rotation, are obtained by the rules and formulae used for ordinary spur gears. When it is necessary, however, to determine such dimensions as the normal tooth thickness, normal pressure angle, etc., the helix angle of the teeth must be considered. When helical gears are cut by the Fellows Gear Shaper method, it is not necessary to take the *normal* tooth shape into consideration, as all dimensions can be figured in the "diametral plane," the gear shaper cutter being designed to cut the correct tooth parts in the plane of rotation. When it is necessary to transfer dimensions from diametral to the normal plane for reasons such as finding size by either tooth caliper, ball or pin measurement, the following example is given.

Assume that it is necessary to determine the dimensions of a helical gear tooth in the normal plane having the following data: 15 teeth, 7/9 pitch, 20° stub tooth, 23° 25′ helix angle. The dimensions of the teeth in the "diametral plane" or plane of rotation are:

Number of Teeth, (N) 15
Diametral Pitch, (DP) 7/9
Pitch Diameter, (PD) .... 2.1428″
Pressure Angle, (PA) .... 20°
Circular Thickness, (CTh) .... 0.2244″
Helix Angle, (HA) .... 23°, 25′

Normal Plane

Number of Teeth, (Nn) =

$$\frac{N}{\cos^3 HA} = \frac{15}{0.7727} = 19.4125 \tag{1}$$

Normal Diametral Pitch, (NDP) =

$$\frac{DP}{\cos HA} = \frac{7}{0.91764} = \frac{7.6283}{9} \tag{2}$$

Pitch Diameter, (PDn) =

$$\frac{Nn}{NDP} = \frac{19.4125}{7.6283} = 2.5448'' \tag{3}$$

Normal Circular Thickness, (NCTh) =

$$\frac{1.5708}{NDP} = 0.2059'' \tag{4}$$

Normal Pressure Angle, (NPA) The Tangent of Normal Pressure Angle = Tan 20° x Cos 23° 25′ = Tan 18° 28′ (5)

The dimensions of the teeth in the normal plane are:

Number of Teeth, (Nn) = 19.4125

Normal Diametral Pitch, (NDP) = $\frac{7.6283}{9}$

Pitch Diameter, (PDn) = 2.5448 inches

Normal Circular Thickness, (NCTh) = 0.2059″

Normal Pressure Angle, (NPA) = 18° 28′

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CONTACT SURFACE TOPOLOGY. . .

(continued from page 42)

Aluminum bronzes with higher strength and a work hardening property will also run-in better with larger contact area and less oversize.

Real Life Experience

Many variables exist in the gear cutting process and before examining the contact pattern on a cut gear one must feel assured that the cutting machine is reasonably accurate, that tool accuracy, sharpening and mounting are correct, and that heating from the cut and its effect on distortion are minimal.

Frequently one does not see a neat full patch of contact but a spotty area. Surface finish and in some cases generating cuts will upset the expected results. Experience will quickly get one to focus on the essentials.

Hob Oversize and Hob Life

Using the axial pitch design procedure the oversize can be considered to be equal to the life of the hob. When the hob is sharpened down below the worm size, it is considered no longer usable. With the normal pitch design procedure the oversize can be substantially more than the sharpening life of the hob. While this is not likely on single thread hobs, it can be expected on hobs with higher lead angles. Because of the manufacturing methods used for the hob, a change in the profile cut on the gear will take place. The hob is terminated when the contact pattern is no longer acceptable.

Future Inspection Techniques

With the arrival of the multi-axis analytical gear checking machines it appears inevitable that the computed conjugate worm gear surface will be compared with the actual cut tooth surface. Computer control of the axes of rotation and travel make it possible to measure other than regular leads or involutes.

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EDITORIAL . . .

(continued from page 5)

research work for us. In spite of the constraints mentioned above, research and development by individual companies does need a higher priority than we have given it. We can no more afford to neglect it than we can afford to neglect capital investment or expanding sales markets.

One logical approach to this problem is to work more closely with universities, one of whose functions is to provide an atmosphere where basic research can be conducted. Unfortunately, our universities in general do not spend a great deal of time on gear related subjects. Only a handful of U.S. schools provide any kind of training related specifically to gearing. Yet two German universities, Munich and Aachen, offer doctoral programs in gearing, and judging from the number of research papers crossing my desk from Japanese universities, advanced gearing education is a priority there too. Seminars such as this one are certainly a step in the right direction; however, we need to encourage more cooperative ventures between the universities and industry.

The seeds of a brighter future are already being planted. AGMA has just successfully completed a combined Gear Expo and Technical Meeting in Cincinnati where the newest in products and research were presented to the industry. I have just received information from ASME/GRI of a program planned for December in Chicago which will be devoted entirely to the question of improving gear research in the United States. With imagination and continuing commitment on the part of business, government and the universities, this "training gap" could be closed.

Another challenge to the gear engineer brought about by the advent of the computer in the workplace is one that relates directly to the area of continuing education. One of the greatest time and labor saving innovations of the personal computer is the growth in the number of software programs available to help the gear design and manufacturing engineer. This software, filled with options and variables, can save the engineer enormous quantities of time and eliminate repetitive calculations. Some of these packages contain the experience of years and years of gear work done by their authors. This experience is available to the engineer for a few thousand dollars and the time it takes to call up the information on his own computer.

But this software contains a subtle trap. We must beware of the danger of using it as a crutch instead of a tool. If we allow such programs to become a substitute for our own personal design and engineering knowledge, we don't expand our own capabilities; we actually limit them. The time will eventually come when the engineer will be confronted by a problem for which the software has made no allowances. Then, only if he has not neglected his basic engineering and has not allowed himself to mentally stagnate, will he be able to confront the problem with innovative solutions of his own. There really is no short cut or substitute for solid, hands-on experience. As novelist Clarence Day says, "Information's pretty thin stuff unless mixed with experience." We need to remember that concepts are not produced by technology, only facilitated by them.

The most basic tool of every engineer has always been his or her curiosity. We must avoid the trap of letting marvelous labor saving devices, whose intention is to give us time for creative work, stifle that creativity and curiosity. An engineer should always be willing to take the time to stroll the shop floor, take the machines apart, and learn what makes them work. He or she should address the computer and its software the same way. A software package that solves current design problems should never stop an engineer from asking *why* a particular solution works.

The 19th century American writer, Henry Adams, wrote long before the development of the computer, but his comments about education and technology are still relevant. He suggested that a university education was beneficial, not because it could anticipate and solve all the problems of new technology, but that it provided the foundation for the lifetime of learning and study necessary to do so. Likewise, the new computer technology cannot eliminate and solve all our engineering problems. Only our own engineering education and experience will enable us to use the new technology to solve problems, to build and to create.

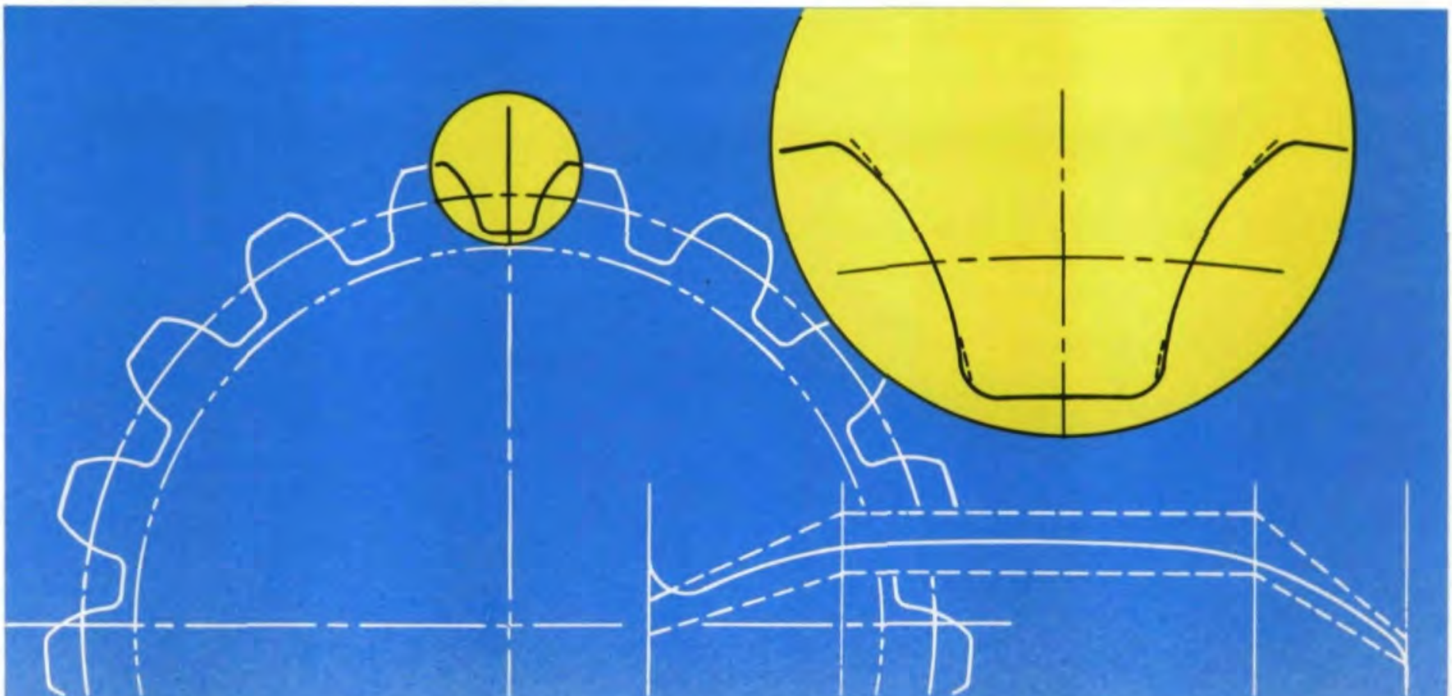
That, after all, is the real fun of engineering. Engineering, for all its emphasis on logic and mathematics and verifiable data, is a creative, inventive science. It would be indeed a tragic irony if we allowed one of the most innovative engineering developments of the 20th century to limit our ability to create and invent in the future.

So where does all this speculation about the computer in the work place lead us? Are we worse off than we were before? Has the invention been more trouble to us than it was worth? Of course not. No one wants to limit us to the possibilities of the days before the advent of the silicon chip, but we must look to the complex future that computers promise us with clear-eyed realism. They are neither the salvation nor the doom of our industry.

The ancient Greeks had a saying that seems appropriate. "The gods demand of us toil as the price of all good things." The computer as a design and manufacturing tool is certainly a good thing, but we are going to have to work to make it as useful and helpful to us as it can be. We will have to exercise thought, care and self-discipline to use it responsibly. We must look at it as a tool, not as an end in itself.

Ralph Waldo Emerson said, "Invention breeds invention." The computer and its implications for the gear industry—both the promises and the challenges—are nothing more than an opportunity to continue the basic functions of the engineer: to raise questions, to solve problems and, finally, to invent and create.

Michael Goldstein
Editor/Publisher

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