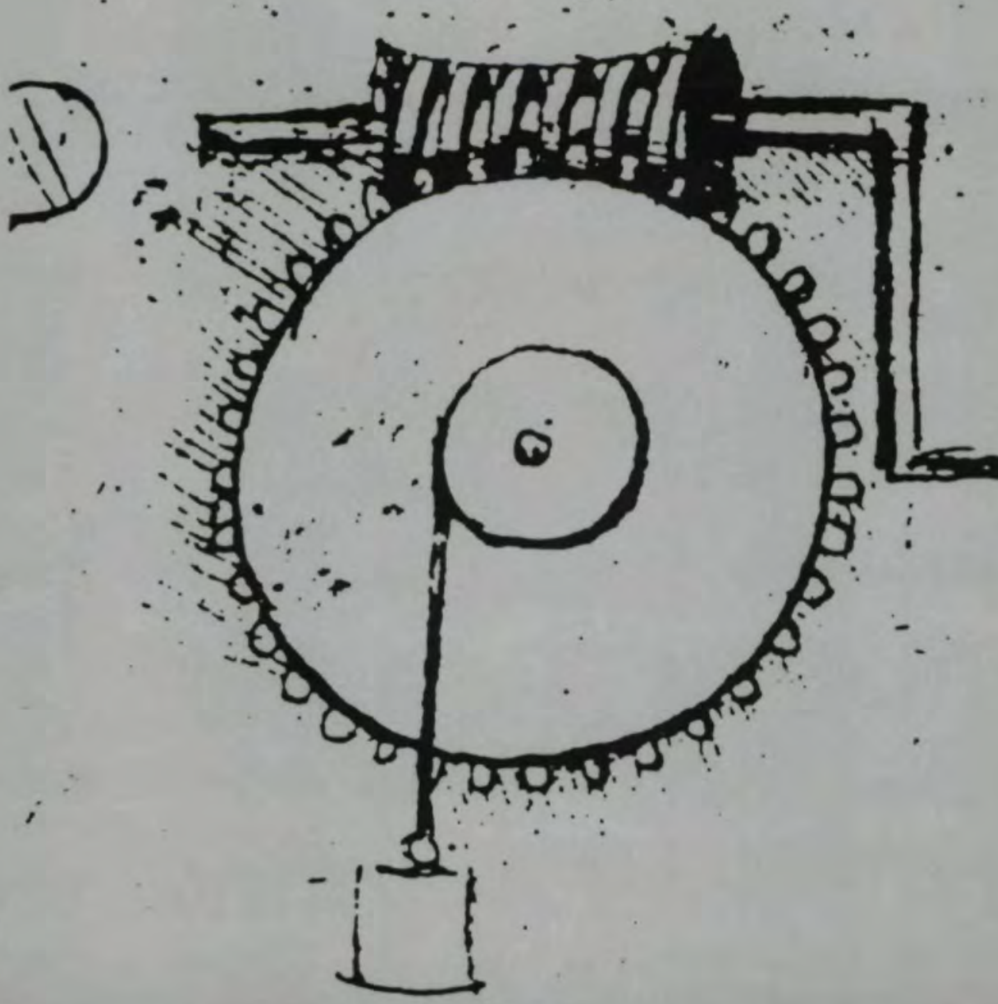
GEAR TECHNOLOGY

The Journal of Gear Manufacturing

MAY/JUNE 1987



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The Journal of Gear Manufacturing

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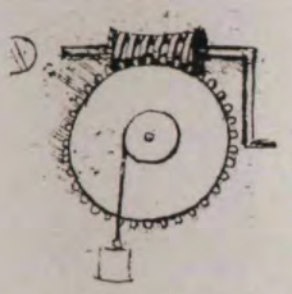

The Advanced Technology
of
Leonardo da Vinci
1452-1519

COVER

Horology, the science of clockmaking, was a major interest of Leonardo's. Among the timekeeping devices he invented was an alarm clock that worked by suddenly lifting the sleeper's feet into the air!

His unique worm gear design, illustrated on our cover, was also born of his interest in clockmaking. In his own words, "This lifting device has an endless screw which engages many teeth on the wheel. . . . the device is very reliable. Endless screws that engage only one of the teeth on the working wheel could cause great damage and destruction if the tooth breaks."

The design is called the Hindley worm gear, after the English clockmaker who "invented" it in the 18th century, 300 years after Leonardo's death.

May/June 1987 — Vol. 4, No. 3

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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

THE WORLD—OUR MARKET

As the time came to write this editorial, the replies to our survey from the last issue were just starting to pour in. We were gratified by the number of responses we received and by the amount of time many of you spent answering in great detail the text questions on the survey. Because of this unusually large response, it will take us some months to log, digest and respond to all the data. Thank you for this nice "problem."

In future editorials, I'll be discussing what we've learned, and we will reprint some of the most interesting comments. We're seeing some excellent suggestions and new ideas which we will start incorporating in future issues.

While reviewing those first surveys, there was one isolated comment that bothered me and kept recurring in my thoughts. A reader wrote, "Why are so many of your articles written by foreigners?"

It reminded me of a similar question that was addressed to me several years ago when GEAR TECHNOLOGY was still in its early months of publication. A marketing manager from what was then one of America's major gear machinery manufacturers asked, "Why are most all your advertisers foreign?"

Today, the marketing manager is no longer with that company, and the company itself is just a shadow of what it had formerly been. This manager and the leadership of his company had become so insular that they failed to see the "big picture."

I think that this lack of understanding that we are part of a world market is the exception. More and more American industries are being affected by this fact every day.

To deny what is taking place simply because we may not like it only increases our vulnerability.

A substantial amount of technical research and writing is being done outside our country. Although we get a majority of our articles from American authors, we are frequently told by American companies, "We'd like to write, but we just don't have the time." Apparently, people in the overseas gear industry find this research and writing important in their companies, their products, and to themselves. Rather than fault them or not take advantage of their knowledge, we should try to emulate them. In the long run, it will pay off.

Our first and strongest supporters, both with financial advertising commitments and encouragement, were foreign manufacturers or their American representatives. From their perspective, it was important that practical information be regularly disseminated to ensure the survival of the gearing industry.

We have tried to bring you a wide variety of information that we thought would be useful, exposing American industry to important articles no matter where they came from, keeping you apprised of what is going on around the world, what processes are being studied and what ideas are being discussed. We must get the tools to succeed from wherever necessary.

Standing with one's head in the sand tends to limit one's vision. You can't see what's approaching, making it difficult, if not impossible, to prepare for the future. We cannot afford the luxury of pretending that foreign markets, foreign competition and foreign research does not exist. Rather we must see these things as challenges and opportunities. The drive for excellence knows no boundaries and claims no special citizenship. As we have been the market for the world, now we must increasingly look at the world as our market.

Michael Goldstein

Michael Goldstein
Editor/Publisher

TECHNICAL CALENDAR

August 25-27
SME WORKSHOP:
International EDM
Lake Geneva, WI

Sept. 22-24
SME WORKSHOP:
Modern Grinding Technology
Detroit, MI

Call For Papers

SME Gear Processing & Manufacturing Clinic
Nov. 17-19, 1987
Detroit, MI

Proposals for papers must be submitted to the SME prior to **June 19, 1987**. They should be in the form of a title and an abstract of 100 words or less.

Suggested topics include, but are not limited to:

> CNC and gear manufacturing, Gear grinding techniques, Gear tool design, Gear tooth scoring, Gear tooth honing, High speed gearing, CNC and gear manufacturing, Hob sharpening, Hard gear finishing, Gear shaping, Gear grinding, Gear measuring, Broaching gears, Inspection techniques

Tabletop Exhibits

An additional feature of the Gear Processing and Manufacturing Clinic will be a series of tabletop exhibits scheduled for the first evening of the program. Gear designers and gear manufacturers and equipment suppliers are invited to display company products and literature at these exhibits.

For more information on SME events contact Joe Franchini at SME, 1 SME Dr., P.O. Box 930, Dearborn, MI 48121, (312) 271-1500.

Tour of Belgium, Switzerland & Germany

A trip to visit gear manufacturing and machine tool plants in Switzerland, Germany and Belgium is currently being planned by AGMA for **September 12-26,** providing the opportunity to see first hand the operation of these high-tech companies. There will be a visit to two Gear Research Institutes: one in Munich with Dr. Winter and a second in Aachen with Dr. Wech. The visit will include time at the October Fest in Munich and plenty of time for sightseeing. The trip is open to anyone who is interested. AGMA membership is not a requirement.

If you are interested or wish further information, contact Joe Arvin, Arrow Gear Co., 2301 Curtiss St., Downers Grove, IL 60515, (312) 969-7640.

VIEWPOINT

Editors Note: As this issue of GEAR TECHNOLOGY was almost ready to go to press, we received a copy of AGMA's March newsletter. It contained the following item by Mr. Joe Arvin, V.P. — Product Division of AGMA and Executive Vice President and General Manager of Arrow Gear. Because it provides another perspective on the subject of our editorial, we are reprinting it in entirety.

Touring Scandinavian Gear Plants

Joe Arvin
Arrow Gear
Downers Grove, IL

As I reflect back on my trip to Denmark, Finland, Norway and Sweden, the first thought that comes to mind is the excellent caliber of the dozen manufacturing companies that we visited. Most of them were very clean and well managed with the latest machine tools that are available anywhere in the world. The vast majority of their machining centers were Japanese built equipment.

The average top hourly wage at plants I visited was $8.00/hr. with an additional $4.00 for fringe benefits. In Finland, the government-owned companies have a depreciation write-off of one-half of the book value per year on new capital equipment. Privately held companies have an investment write-off of 30% per year up to seven years. There are extremely low interest rate loans available from the government for new manufacturing facilities and government subsidies for new manufacturing companies in rural areas.

During our visit to the ASEA Robotics plant in Sweden, we all were astonished by the money that is being put into research and development in what ASEA feels is one of the "up and coming" technologies. They sold 19 robots in 1974, whereas in 1985, over 2,200 were sold. And management proudly boasted that by 1990, they will be the leading robot manufacturer in the world.

Upon arrival at the Stockholm Sheraton in Sweden, I was contacted by Goran Lundstrom, Technical Editor of the Stockholm Press. He said that he had heard that I was V.P. of the American Gear Manufacturers Association, that he wanted very much to interview me and so I agreed. During my listing of the countries

(continued on page 8)

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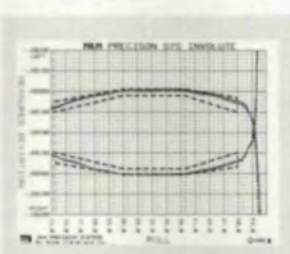
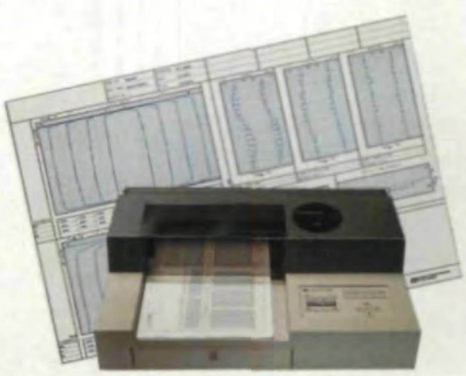


Graphics printer copies CRT graphics and tabular test data.



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and companies that we were visiting in Scandinavia, he periodically kept asking why I was there. I kept giving him the same answer: that I have visited gear and machine tool manufacturers in over a dozen countries during the past five years, and that I always found it interesting to see how other people were doing the same type of work that we do at Arrow Gear. I told him we also reciprocated by permitting foreign visitors to tour our plant, and that it was no longer an Asian/European/USA economy, but a worldwide economy that we are living in, and that we all have to get along together in this environment. I also told him that I was trying to help the AGMA to get its standards accepted in the world marketplace.

After the inverview was over, I asked Mr. Lundstrom why he kept questioning me as to why I was there. He said with a smile, the question wasn't why were we in Scandinavia and why did we visit these specific companies, but why weren't we also visiting their universities and their technical centers as the Japanese visitors do. Not only are the Japanese the No. 1 visitors to Scandinavia, but they spend literally months there visiting all the companies they can and spending weeks at their universities and tech centers. He also said that he and most Scandinavians thought America was a wonderful country, and that it was a shame that the U.S. manufacturing people were not spending more time in Sweden and in other corners of the world, absorbing their intellectual knowledge and bringing it back to the United States as the Japanese do for Japan.

One of the things that has continued to bother me a great deal in my numerous travels around the world, is the number of consumer products, complete assemblies, or component parts that are being made by foreign comanies and end up being sold in the United States. This is not only due to commercial overseas joint ventures by major U.S. corporations, but also by the U.S. government and branches of the Armed Forces buying foreign components and products.

For example, I discovered during this Scandinavian trip alone that the U.S. government buys component parts from Kongsburg Vaapenfabrik, Norway, for the F-16 fighter, ASN Pentagon missile, and that the U.S. Air Force gave them $8 million dollars for the development of an automated machining cell for jet engine rotors. When there are so many U.S. manufacturing companies going out of business daily, we should at least enjoy the business of our own government.

(continued on page 11)

Give Your Gears a Break — Select the Right Coupling!

Stan Jakuba,
S.R. Jakub Associates
West Hartford, CT

How important is the right choice of coupling in determining successful machine design? Consider the following example. A transmission of appropriate size was needed to transfer the speed of the engine driver to that of the driven generator. The transmission was properly selected and sized to endure the rated power requirements indefinitely, but after only a short time in operation, it failed anyway. What happened? The culprit in the case was a coupling. It provided the necessary power and protection against misalignment, but it lacked the ability to isolate the gears from the torque peaks of the diesel engine.

All reciprocating engines produce uneven torque with peaks much higher than the value of the rated torque. The torque fluctuations are caused by the variations in the tangential forces acting on the crankpins. The engine flywheel smooths the fluctuations to some extent; the more inertia it has, the better job it does. Unfortunately, the more inertia a flywheel has, the heavier it usually is, and therefore, invariably, the light and powerful engines of today are "jerky" in this respect. Their flywheels are light to keep the engine mass down. On the other hand, they often drive relatively heavier equipment. The rotating components of the driven equipment may very well have more inertia than the engine flywheel. We will explore the negative effects these features have on the load imposed on the gears in transmissions, power takeoffs, step-up gears, and similar mechanisms and discuss possible countermeasures.

Torque Fluctuations at the Prime Mover Output

One of the highest torque fluctuations is present in diesel engines. The size of the fluctuations depends on many fac-

AUTHOR:

MR. STAN JAKUBA *has over twenty years experience in the gear industry in the United States and overseas. President of S.R. Jakub Associates, Engineering and Training Consultants, Mr. Jakuba was educated in Czechoslovakia and holds a masters degree in mechanical engineering from MIT. He is the holder of several patents for engineering products and is a member of ASME and SAE. He is also secretary of the U.S. Metric Association.*

tors, the obvious ones being the load on the engine, which determines the torque peaks before the flywheel, and the flywheel polar inertia, which determines how much lower the peaks are on the output shaft. The relative size of the average torque, the maximum instantaneous torque, and the influence of the polar inertia of the flywheel are illustrated here in the case of a common prime mover, the four cylinder diesel engine.

Fig. 1 shows three traces of the instantaneous torque of such an engine sensed at three different locations. The high amplitude line represents the torque at the interface of the crankshaft with the flywheel. The straight line represents the torque that would be sensed after the flywheel if the flywheel had an infinitely large inertia, or if the driven equipment had infinitely low inertia. The line also represents the torque that would be displayed by a dynamometer gauge; i.e., the average torque on the output shaft. The medium amplitude line represents the torque that might be experienced on the output shaft in a real installation.

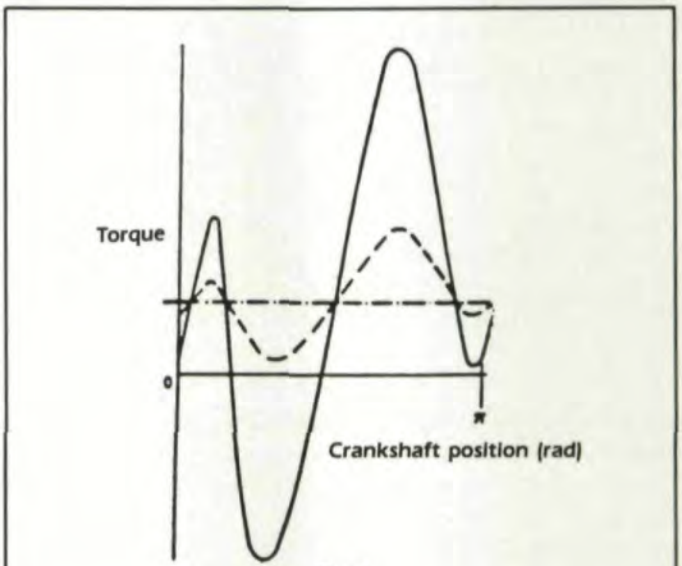


Fig. 1

It is apparent that at a steady load and speed, the peak of the medium amplitude will be higher with higher equipment inertia and lower with lower equipment inertia for a given engine. This consideration has a practical implication for the design of flanges: one ought to be skimpy with flanges on the equipment side and stay with as small diameters as possible. By contrast one may be generous with flanges on the engine side. Inertia added to the engine side benefits the transmission.

Coupling

There is no practical way to make engines with infinite polar inertia, nor can the equipment have zero inertia. Thus, some magnitude of the medium amplitude is inevitably input into the transmission. Obviously, the lowest possible amplitude is desired. Torsionally soft couplings have always been used to insure this condition. Today a wide selection of such couplings exists.

Ideally, the coupling should be so soft that it smooths out the torque fluctuation completely. It should store the excess energy during the time the peaks occur and release it during the time the valleys occur. Such a coupling is, of course, impossible to build. Furthermore, because of hysteresis, the amount of torque fluctuation the coupling can withstand is limited by its capacity to rid itself of heat.

The torsionally soft coupling is engaged to decrease the amplitude that reaches it to the level manageable by the driven equipment. Unfortunately, not all soft couplings are suitable in all applications. There are also applications where a soft coupling does more harm than good to the driven equipment, as in the case where the transmission is relatively small and the inertia of the gears can be neglected.

Resonance

Gear and transmission problems often stem from the overstress due to the occurrence of the torque peaks. The magnitude of the peaks is determined by the influences described so far and also by another factor. Consider that the peaks occur rhythmically. If the frequency of the peaks coincides with one of the natural frequencies of the system, the stress peaks grow with time until a component fails. More couplings and gears fail because of the overstress in resonance than from all other causes combined.

The task, then, is to analyze the whole engine-to-equipment system to predict its natural frequencies. Often, only the first natural frequency needs to be known, as the higher ones can be made to lie outside the operating range by the selection of the coupling stiffness. If the equipment operates at a fixed speed, it is relatively easy to select a coupling whose torsional stiffness is such that resonance does not occur in the vicinity of the operating speed. Many resilient couplings are on the market, and some manufacturers provide information on how to make the right selection. However, when it comes to the wide-speed-range operation, there is no coupling that can run without introducing resonance at some speeds. The task is to select the coupling that covers the widest speed range, learn the speeds where the system could resonate, and avoid operating at those speeds.

Transmission Protection

A torsionally soft coupling should be used wherever possible to protect the transmissions from overload due to excessive torque peaks. The softer the coupling is, the lower the transmitted peaks will be. Furthermore, the softer the coupling, the lower the highest resonant speed will be. A system with a low stiffness coupling will resonate only in the low speed range where the nominal torque is normally very low. Thus, resonance should be experienced only during start-up and shut-down, and the resulting amplitudes should be safe. For the protection of the coupling, these steps should be passed through quickly to limit the time available for the amplitudes to build up. The coupling will last indefinitely if the low speed range is passed through so fast that the stress amplitudes do not reach dangerous levels or the heat does not build up and destroy the resilient material.

A certain amount of inherent damping and nonlinear characteristics in a coupling help to prevent the growth of the amplitudes in resonance. A coupling with these two features should be selected if the operation close to resonance cannot be ruled out. The transmission will experience higher peaks at the steady state load but the increase is generally negligible in comparison to the substantial decrease in resonance.

There are applications where the full torque must be transmitted at a relatively low speed. The very low speed region is safely served by a very stiff coupling. With the stiff coupling, the system can be operated below all significant resonant speeds. The torque peaks transmitted to the transmission are then a function of the exciting pulses of the prime mover and, as discussed earlier, a function of the relative size of the inertias of the engine and the equipment. Operating a transmission in a system with a stiff coupling at higher speeds is not advised unless the resonant speeds are known and can be avoided, or at least crossed over fast at low load. The exact magnitude and the value of the resonant speeds is not easily predictable, and several resonant speeds may be present in the desired operating range.

The most common type of stiff coupling is the ordinary universal joint drive shaft. This type of coupling is often noisy. The noise is usually an indication of the operation in a resonant region. A backlash in the driveline makes the noise, and the stresses in the transmission are further increased with the backlash.

Generally speaking, the use of a torsionally stiff coupling requires that the coupling and the transmission are oversized in comparison to the case when the soft coupling is used. They have to withstand torque values much higher than the values predicted from the power absorbed by the equipment. How much higher depends largely on the relative sizes of the polar moments of inertia of the equipment rotating components with respect to the prime mover rotating components.

Conclusion

For maximum protection of the transmission preference should always be given to the use of a torsionally soft coupling. When selecting a soft coupling, the objective is to find one of the desired physical configuration, torque capacity and allowable speed, which also exhibits the lowest torsional spring rate in its class, and is designed to fail at a load safe for the transmission.

VIEWPOINT

(continued from page 8)

Dear Editor:

The article by Mr. Dale on "Gear Noise and the Sideband Phenomenon" contains some interesting test results, but may have given a slightly deceptive idea of what is currently possible.

The interaction of amplitude and frequency modification with resonances has been recognized for some time, although it is not the only possible way of obtaining highly asymmetric sidebands especially in epicyclic gears.[1] In the case quoted, the effects of pitch errors which only repeat every 559 teeth will give modulation at a vast range of frequencies, all multiples of the basic mesh cycle frequency.

Single flank testing is normally carried out slowly, and the article suggests that it is not possible to carry out transmission error checks at speed; this is, however, done at full speed and full torque on gearboxes. When testing under these conditions, it is easy to use time averaging techniques[2] which are more powerful than simple frequency analysis and have the advantages of separating out pitch errors on the two gears and increasing the accuracy of the results.

J. D. Smith
University of Cambridge
England

1. P. D. McFadden and J. D. Smith, "An Explanation for the Asymmetry of the Modulation Sidebands About Tooth Meshing Frequency in Epicyclic Gear Vibration." *Proc. Inst. Mech. Eng. 1985, 199* (C1), pp. 65-70.
2. J. D. Smith, *Gears and Their Vibration.* Marcel Dekker, New York and MacMillan, London. 1983.

Letters for this column should be addressed to Letters to the Editor, GEAR TECHNOLOGY, P.O. Box 1426, Elk Grove Village, IL 60007. Letters submitted to this column become the property of GEAR TECHNOLOGY. Names will be withheld upon request; however, no anonymous letters will be published. Opinions expressed by contributors are not necessarily those of the editor or publishing staff.

Gear Design Options

William L. Janninck
ITW-Illitron Automated Systems
Lincolnwood, IL

AUTHOR:

W. L. JANNINCK *is a gear geometrician and consultant to the Illinois Tools Division of ITW Inc. He has experience in the design and application of both generating and forming type metal cutting tools, specializing in the area of tooling and gaging for the manufacture of gears, splines and sprockets. Prior to entering private practice, he was manager of product engineering and chief engineer at ITD. He has served on various commitees of the AGMA and MCTI and is chairman of the AGMA Cutting Tool Committee. He also served on the SAE-ANSI Committee on Power Transmission Chains and Sprockets. He has written extensively on tool applications, gaging, gear design and gear inspection. He was educated at Northwestern University, Evanston, Illinois.*

An Introduction to a Spur Gearset Design

When specifying a complete gear design, the novice designer is confronted with an overwhelming and frequently confusing group of options which must be specified. This array of specifications range from the rather vague to the very specific.

There are many ways to narrow the selection and guide the designer in selecting the less easily defined parameters. Some values, especially the geometric ones, must be specified very exactly. Others can be selected rather loosely on the basis of choice or experience. (See Table 1.)

One effective method of controlling and organizing this set of specifications is based on laying out a group of possible choices for consideration that first meet the geometrical requirements of center distance and gear ratio, and then considering the physical requirements of transmitted power, input speed, gear material and gear accuracy. The availability of the personal or micro-computer has also made it quite simple to compute a group of different designs that would be possible candidates for a final design.

Table 2—Design Choices for Single Pair Set of External Spur Gears

Center distance	5.0 inches
Gear ratio	4 to 1 reduction
Pinion speed	1800 RPM
Face width	.80 inches
Gear material	Steel
Hardness	300 BHN

To simplify the example and also to eliminate a few of the choices we will give an example of a design for a single pair set of external spur gears with the specifications shown below.

The face width was arbitrarily chosen, but was based on a scale layout of the anticipated set using the operating pitch circles. A common sense of proportion prevailed. The gear material picked was a typical machineable gear steel that can be finish cut, preserving the cut gear accuracy. Expected accuracy would be in the range of AGMA 7 or 8.

Using the gear ratio and the center distance as input, a short computer program is used to arrive at a number of choices for gear and pinion teeth and related diametral pitch. This program, (Program 1) starts with the pinion teeth ranging from 10 to 55, sequentially, computing the mating gear teeth closest to the desired ratio, and prints out the associated diametral pitch for standard operating conditions. This gives 46 different possible sets to choose from. A sample output is shown in Table 3.

The number of choices may be further reduced by selecting some sets where currently available cutting tools can be used or a scattered group can be arbitrarily selected, ranging

Table 1—Basic Gear Design Specifications

Type of gear spur or helical	Helix angle & hand	Type of teeth internal or external
Diametral pitch	Pressure angle	Pinion & gear teeth
Gear ratio	Tooth form	Pinion enlargement
Fillet form	Center distance	Tooth proportion
Face width	Material & treatment	Input speed
Strength	Quality level	Life expectancy
Noise	Manufacturing methods	Lubrication

Table 3 – Available choices of teeth for spur gear sets.

Center Distance = 5
Target Gear Ratio = 4

Pinion T.	Gear T.	Act. Ratio	Diam. Pitch
10	40	4.0000	5.00000
11	44	4.0000	5.50000
12	48	4.0000	6.00000
13	52	4.0000	6.50000
14	56	4.0000	7.00000
15	60	4.0000	7.50000
16	64	4.0000	8.00000
17	68	4.0000	8.50000
18	72	4.0000	9.00000
19	76	4.0000	9.50000
20	80	4.0000	10.00000
21	84	4.0000	10.50000
22	88	4.0000	11.00000
23	92	4.0000	11.50000
24	96	4.0000	12.00000
25	100	4.0000	12.50000
26	104	4.0000	13.00000
27	108	4.0000	13.50000
28	112	4.0000	14.00000
29	116	4.0000	14.50000
30	120	4.0000	15.00000
31	124	4.0000	15.50000
32	128	4.0000	16.00000
33	132	4.0000	16.50000
34	136	4.0000	17.00000
35	140	4.0000	17.50000
36	144	4.0000	18.00000
37	148	4.0000	18.50000
38	152	4.0000	19.00000
39	156	4.0000	19.50000
40	160	4.0000	20.00000
41	164	4.0000	20.50000
42	168	4.0000	21.00000
43	172	4.0000	21.50000
44	176	4.0000	22.00000
45	180	4.0000	22.50000
46	184	4.0000	23.00000
47	188	4.0000	23.50000
48	192	4.0000	24.00000
49	196	4.0000	24.50000
50	200	4.0000	25.00000
51	204	4.0000	25.50000
52	208	4.0000	26.00000
53	212	4.0000	26.50000
54	216	4.0000	27.00000
55	220	4.0000	27.50000

Program 1 – Tooth Selector Program

```
10 REM – Toothsel.bas
20 REM – This program develops a series of choices of gear
  teeth sets
30 REM – for a specific center distance and ratio desired.
40 KEY OFF:CLS
50 DIM NP (55) ,NG(55) ,ACTR(55) ,DP(55)
60 INPUT "Center Distance  ";CD
70 INPUT "Ratio Desired "    ;RATIO
80 FOR I=10 TO 55
90 NP(I)=I:NG(I)=INT(I*RATIO+.5)
100 ACTR (I)=
  NG(I)/NP(I):DP(I)=(NP(I)+NG(I))/2/CD
110 NEXT I
120 LPRINT"     Available choices of teeth for spur gear
  sets."
130 LPRINT
140 LPRINT"     Center Distance = ",CD
150 LPRINT"     Target Gear Ratio = ",RATIO
160 LPRINT
170 LPRINT" Pinion T.  Gear T.    Act.
Ratio    Diam. Pitch"
180 FOR I=10 TO 55
190 Z$="    ##     ###     ###.####     ##.#####"
200 LPRINT USING Z$;NP(I) ;NG(I);ACTR(I);DP(I)
210 NEXT I
```

through various teeth numbers and pitches. For example, the sample list in this case contains commonly available pitches such as 5, 6, 8, 10, 12, 14, 16 and 20. If an odd target gear ratio or an uneven center distance is used, the possibility of standard diametral pitches is reduced.

If the teeth versus pressure angle chart for natural undercut shown in Fig. 1 is examined, the choice will be guided away from the lower pinion teeth numbers and lower pressure angles because natural undercutting causes poor gear operating conditions and also reduces strength.

In the example case the first likely choice is a 16-64 tooth gearset of 8DP and 20 or 25° pressure angle. The next choice would be 20-80 teeth of 10DP in 20 or 25° pressure angle, and so on thru 20DP.

Actually there is a great deal of flexibility in gear design and long addendum pinions and short addendum gears can be used to avoid or reduce undercutting. It is also possible to depart from the specified diametral pitch by a small amount, resulting in an oversize or undersize operating condition on center distance.

A flexible computer program was written to accomodate both standard and non-standard conditions and to freeze the total gear geometry for a particular gear set. The input values required to run the program are listed in Table 4.

The program has a standard basic rack embedded within it for 14.5 thru 25°PA as shown in Fig. 2. This basic rack is used, along with the circular tooth thickness on the pinion and the backlash desired in the set to do a complete gear design. If the pinion tooth thickness is set at one half of the

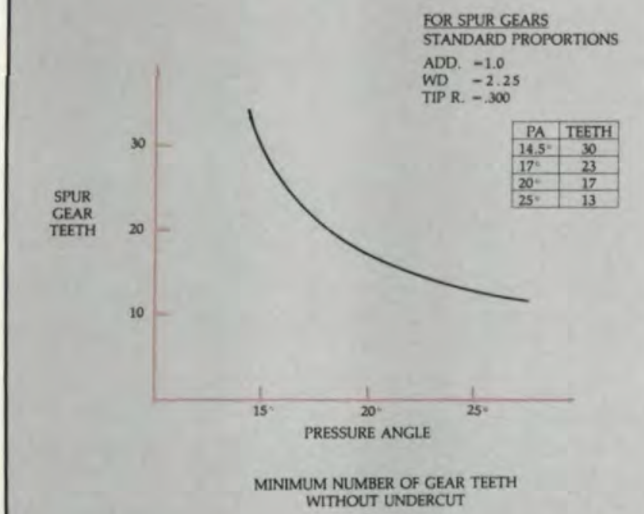


Fig. 1

Table 4

```
40 INPUT "Diametral Pitch      = ",DP
50 INPUT "Pressure Angle       = ",PA
60 INPUT "Center Distance      = ",CD
70 INPUT "Face Width           = ",F
80 INPUT "Steel Allow Tens.    = ",SAT
90 INPUT "Steel Allow Comp.    = ",SAC
100 INPUT "Pinion RPM          = ",RPM
110 INPUT "Pinion Teeth        = ",NP
120 INPUT "Gear Teeth          = ",NG
130 INPUT "Pinion CTT          = ",CTP
140 INPUT "Backlash            = ",BL
```

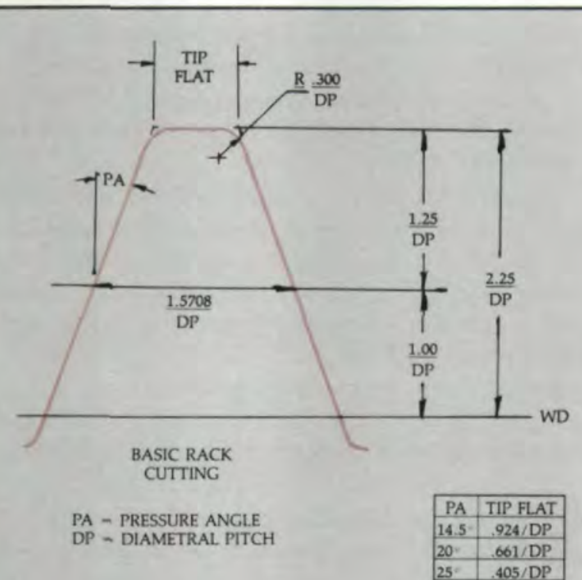


Fig. 2

circular pitch, a standard proportioned gear set is developed. If the pinion tooth thickness used is greater than half of the circular pitch, the pinion will be oversized.

There are many causes for gear tooth failure, as shown in AGMA 110.04 "Nomenclature of Gear Tooth Wear and Failure." The two principal modes for gear tooth failure are tooth breakage by bending fatigue and surface failure by pitting. More advanced failure considerations include scoring, spalling, rolling, peening, rippling, case crushing and various forms of wear.

The program segment for calculating the horsepower ratings uses the procedure outlined in AGMA 218.01 "For Rating the Pitting Resistance and Bending Strength of Spur and Helical Involute Gear Teeth" and is based on the stresses caused by the contact or compressive stresses where the gear teeth meet, and on the bending or tensile stresses which occur in the gear fillet area. See Fig. 3. Examples of these two failures are shown in Fig. 4.

The use of the superimposed parabola to determine the critical fracture or fatigue point on the root fillet is shown in Fig. 5 and is used to compute the interim values needed to establish the bending power rating.

The surface stresses caused by the rolling action of contacting cylinders results in elastic deformations. If the subsurface shear that develops exceeds the strength of the material, a crack occurs and propagates up to the surface developing a pit. See Fig. 6.

Fig. 7 illustrates the pattern of tooth pair contact for a gear with a contact ratio of about 1.5. In the central part of the tooth form a single pair of teeth are in contact. For shared load the highest point of single tooth contact is used in calculating the beam strength and the lowest point of single tooth contact is used in the surface strength calculation. The former is used for the "J" factor, and the latter is used for the "I" factor.

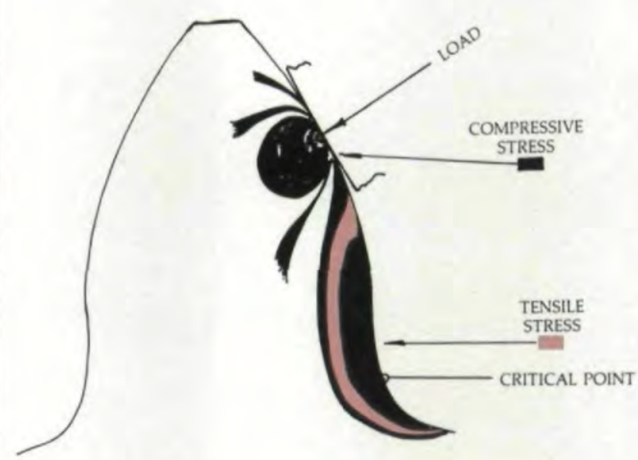


Fig. 3

Fig. 4 – Types of stress failures on a gear tooth

a – Destructive pitting

b – Spalling

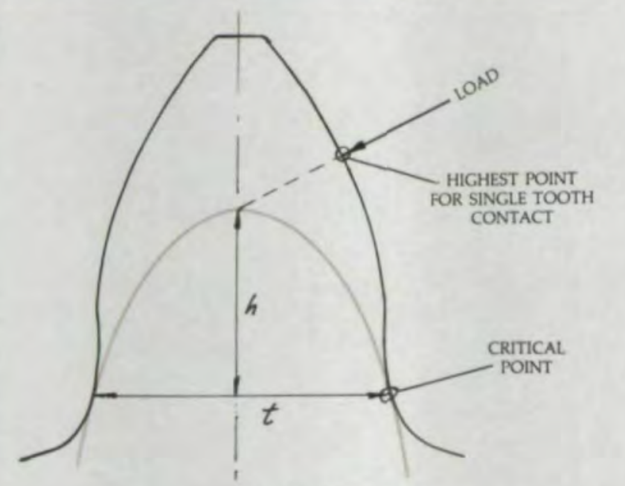


Fig. 5

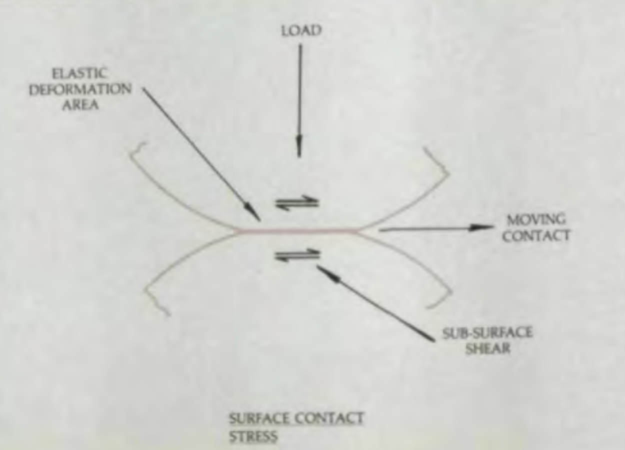


Fig. 6

The AGMA equations for the bending and pitting power ratings are shown below, first with all the K and C factors and, then, in simplified form without these factors.

$$P_{AT} = \frac{np\ d\ Kv}{126{,}000\ K_A} \bullet \frac{F}{Pd} \bullet \frac{J}{K_S K_M} \bullet \frac{S_{AT}\ K_L}{K_R\ K_T}$$

np = Pinion rpm
d = Pinion oper. p.d.
F = Gear face
Pd = Diametral pitch
J = Geom. factor-bending
S_{AT} = Tensile stress no.

$$P_{AT} = \frac{np\ d}{126{,}000} \bullet \frac{F}{Pd} \bullet J \bullet S_{AT} \tag{1}$$

$$P_{AC} = \frac{np\ F}{126{,}000} \bullet \frac{I\ Cv}{C_S\ C_M\ C_F\ C_A} \bullet \left(\frac{d\ S_{AC}}{Cp} \bullet \frac{C_L\ C_H}{C_T\ C_R}\right)^2$$

np = Pinion rpm
F = Gear face
I = Geom. factor – pitting
d = Pinion Oper. p.d.
S_{AC} = Surface comp. stress no.
Cp = Elastic coeff.

$$P_{AC} = \frac{np \bullet F \bullet I}{126{,}000} \bullet \left(\frac{d \bullet S_{AC}}{Cp}\right)^2 \tag{2}$$

The factors can generally be used as 1.0 and can mainly be considered as warning flags to induce some thought on the part of the designer. The flags are as follows:

- Ka & Ca – application factor which considers the even or shocky nature of the prime mover and the absorbing load;
- Kv & Cv – related to the effect that dynamically induced loads might cause, usually due to higher velocities;

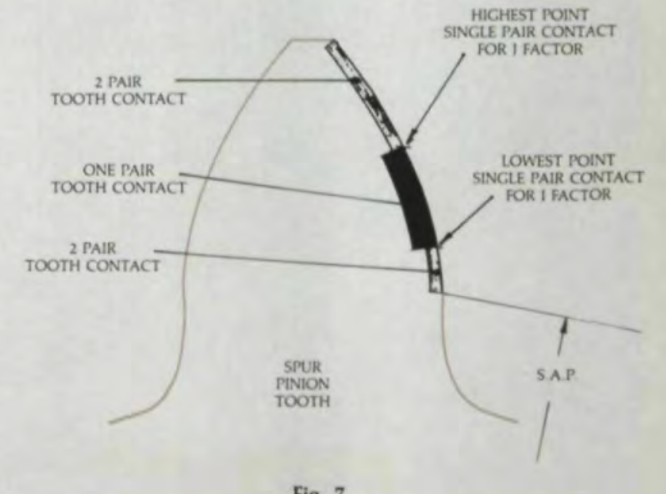


Fig. 7

- Ks & Cs — used in consideration of the effect that the actual physical size of the teeth might have;
- Km & Cm — warn about the effect of improper load distribution across the gear face;
- Kl & Cl — related to gear life. Normal life is planned for many millions of stress cycles and is modified for shorter life needs. See Fig. 8
- Kr & Cr — reliability factors normally expecting less than 1 failure in 100;
- Kt & Ct — temperature factors warning that gears normally do not exceed 250°F;
- Cf — consideration factor for the effect of surface finish on the surface strength;
- Ch — is a factor for the ratio of hardness between the gear and pinion.

Again, for simplicity the K & C factors will be used as 1.0, although some consideration has to be given to the application factor Ka & Ca and the reliability factor Kr & Cr in the final selection, as they are important and significant.

For the gear material chosen, steel at 300 BHN, the AGMA suggests the allowable tensile stress number as 40,000 and the allowable compressive stress number as 130,000 for input.

The complete computer output for one gear set design is shown in Table 5. At this point we can tell if the power capacity of these designs will meet the needs and with what reliability or safety factor.

Noise Considerations

Gear noise is complex and difficult to analyze, but is usually related to gear accuracy. For equivalent accuracy among gears with various pressure angles, it has been observed that lower pressure angle gears are quieter and, conversely, higher pressure angles are noisier. The relationship is shown in Fig. 9. If we have sufficient capacity in the gears designed, we could choose the lower pressure angle for a quieter set. This does not mean sets with higher pressure angles cannot be quiet, but they will probably require a higher level of gear accuracy.

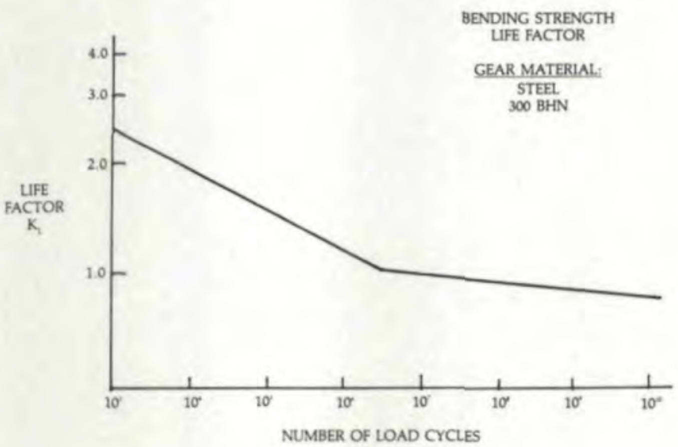
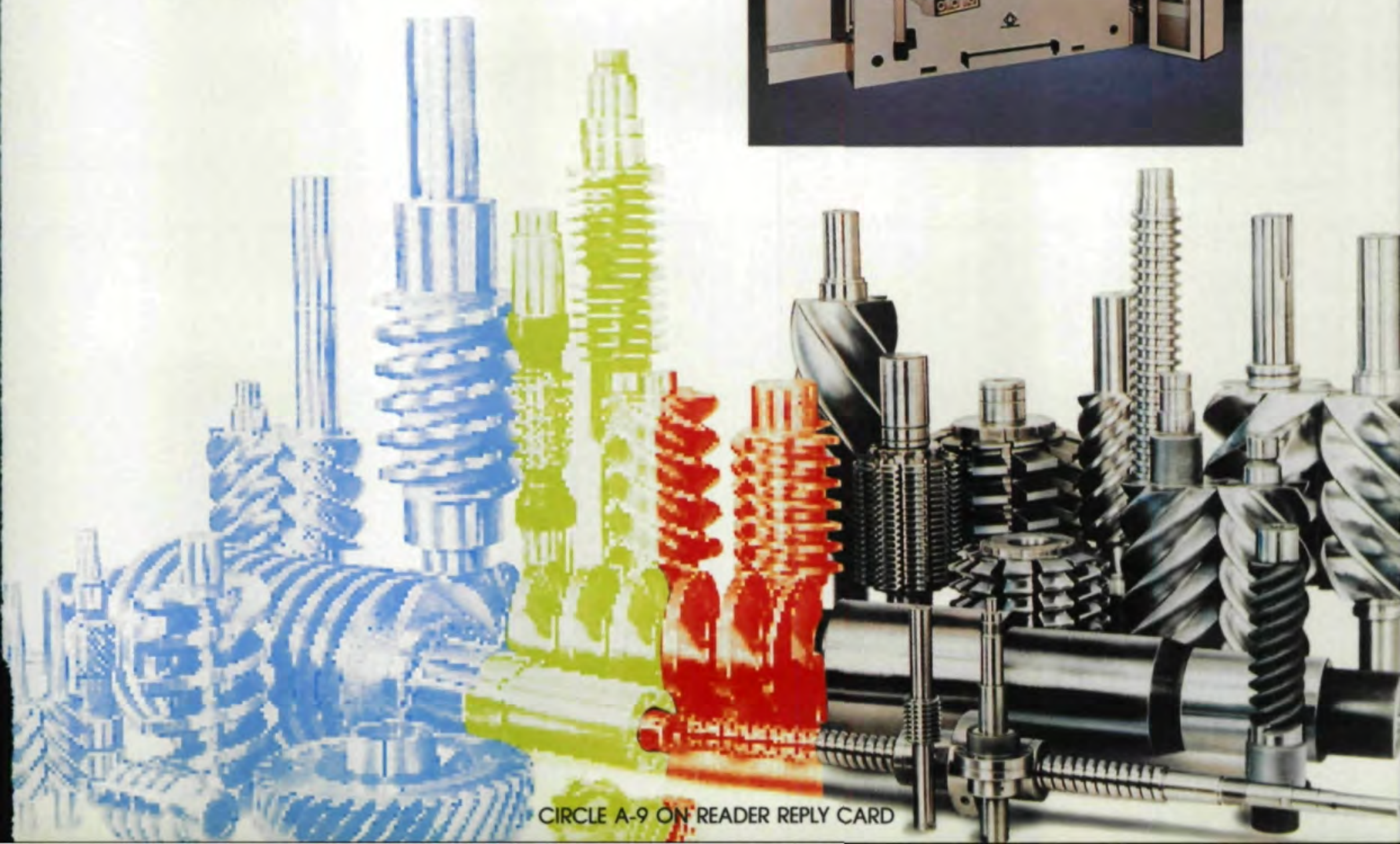


Fig. 8

Table 5 — Complete Computer Output for One Gear Set Design

Center Distance	5.0000	
Gear Ratio	4.0000	
Face Width	0.800	
Diametral Pitch	10.0000	
Pressure Angle	20.0000	
Pinion RPM	1800.00	
Tensile Stress No.	40000	
Compressive Stress No.	130000	
Backlash	0.0020	
Contact Ratio	1.691	
	Pinion	Gear
Teeth	20.0	80.0
Outside Diam.	2.200	8.200
Pitch Diam.	2.0000	8.0000
Oper. P.D.	2.0000	8.0000
Root Diam.	1.750	7.744
Base Diam.	1.8794	7.5175
Cir. Tooth Thick.	0.1571	0.1551
Tip Flat	0.0695	0.0778
T.I.F. Diam.	1.8850	7.8547
J FACTOR	0.368	
Horsepower Beam	33.6	
Horsepower Surf.	15.9	



Another area of consideration if noise reduction is of great importance is the use of non-standard working depth gears, such as stub or shorter working depth or extended working depth. See Fig. 10.

Stub teeth are very rigid and tend to be noisier than the longer extended tooth forms where more tooth flexure is present. *These tooth depth systems require separate geometrical* considerations for design especially in regard to the minimum number of teeth.

Strength Considerations

If maximum strength is of significant importance, then higher pressure angles would be chosen, as is shown in Fig. 11. Here the relationship is shown for strength versus pressure angle, where the strength increases with increasing pressure angle. This is also confirmed later using the AGMA horsepower capacity calculations.

For the gear material chosen, the computations for horsepower show the bending strength of the pinion is greater than the surface strength. This is quite typical of steel or other materials with insufficient surface strength. Typically such gears wear out rather than failing by fracture. It is preferable, obviously, to have a gear set fail progressively by wear rather than catastrophically by breakage.

The results of some 15 different program runs are summarized in Table 6. The horsepower based on bending and surface strength is tabulated. Several things can be noted. First, in the pitch range chosen all the sets have higher bending strength than surface strength. Second, the surface strength goes up as the teeth get smaller, and third, the bending strength goes down as the teeth get smaller. Fig. 12 presents a graphical comparison of the computations for the 20°PA group. In essence the surface strength is fairly level across the entire pitch range while the bending strength decreases with the finer pitches.

Reliability

To reduce the chance of failure in the gear set, the Kr & Gr factor as suggested by AGMA can be considered. The factor for high reliability, less than 1 in 1000 failures, is 1.25,

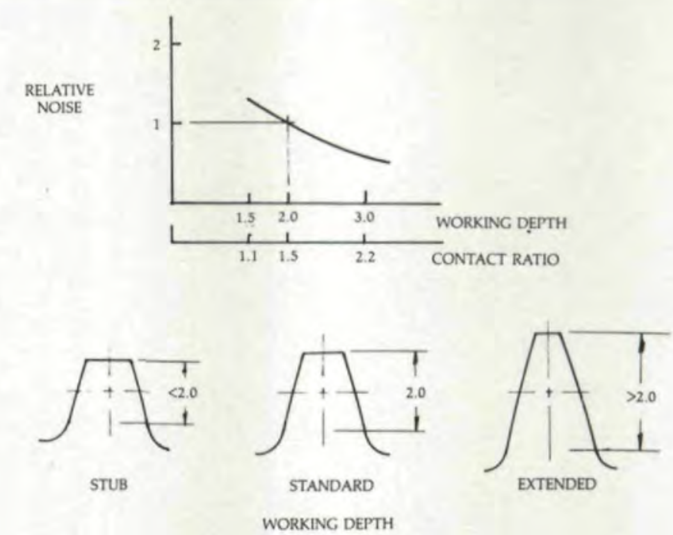
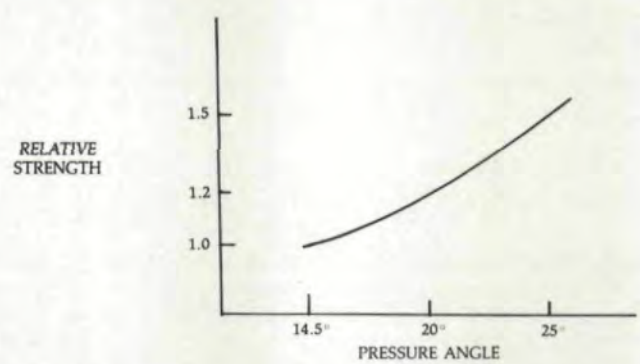
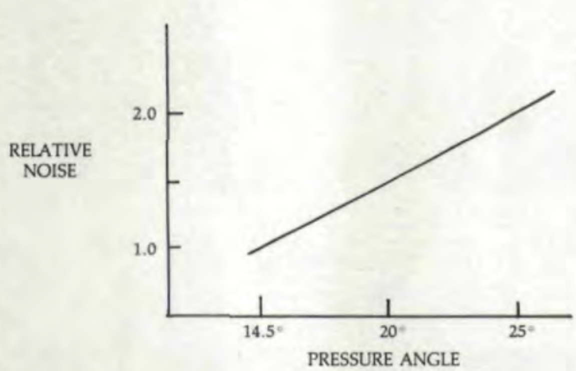


Fig. 10



COMPARISON OF STRENGTH WITH PRESSURE ANGLE

Fig. 11



COMPARISON OF NOISE WITH PRESSURE ANGLE

Fig. 9

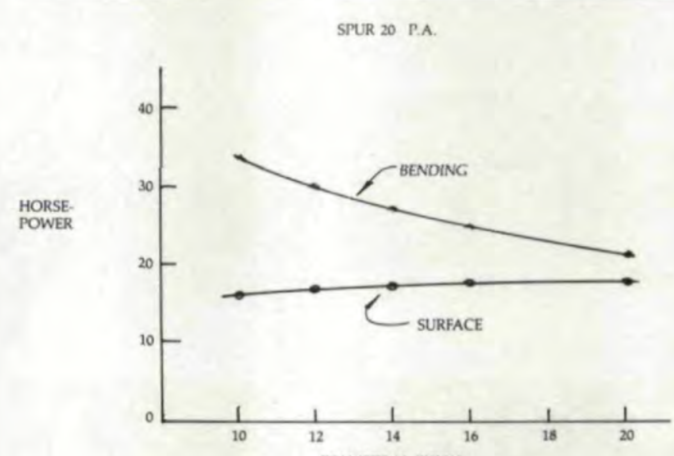


Fig. 12

(continued on page 42)

Influence of Geometrical Parameters on the Gear Scuffing Criterion – Part 2

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ABSTRACT

In Part 1 several scuffing* criteria were shown ultimately to converge into one criterion, the original flash temperature criterion according to Blok.

In Part 2 it will be shown that all geometric influences may be concentrated in one factor dependent on only four independent parameters, of which the gear ratio u, the number of teeth of the pinion z_1, and the addendum modification coefficient of the pinion x_1 are significant; whereas, the addendum modification coefficient of the wheel x_2 is of minor importance. Again, Blok's flash temperature criterion is confirmed. The low number of significant geometric parameters allows an examination of the influence of different shapes and values of the load sharing factor.

*Scuffing and scoring are synonyms for the same phenomenon. Since scoring may also have another meaning, the ISO Technical Committee 60 decided to apply the word scuffing in the ISO standards.

Determination of the Maximum Contact Temperature

For routine calculations and for the optimization of parameters the maximum contact temperature has to be determined by an iteration process or by a direct approximative expression.

The Equations (1) and (2), shown in Part 1, are rewritten for the maximum value

$$\Theta_{Bmax} = \Theta_M + \Theta_{flmax} \tag{1}$$

$$\Theta_{flmax} = \mu_{mC} X_{top} \frac{W_{Bt}^{3/4} v^{1/2}}{a^{1/4}} \tag{2}$$

where

$$X_{top} = X_M \{X_B X_r\}_{max} \tag{3}$$

For representative steels the thermal contact coefficient is

$$X_M = 50\ K.N^{-3/4}.s^{1/2}.m^{-1/2}.mm \tag{4}$$

The geometry factor is

$$X_B = 0.51\ (u+1)^{1/2} \frac{abs\ (\sqrt{1+\Gamma} - \sqrt{1-\Gamma/u})}{(1+\Gamma)^{1/4}(u-\Gamma)^{1/4}} \tag{5}$$

where

$$\Gamma = \frac{\tan \alpha_y}{\tan \alpha'_t} - 1 \tag{6}$$

The load sharing factor X_Γ is one of the trapezoid functions represented in Figs. 1 to 4.

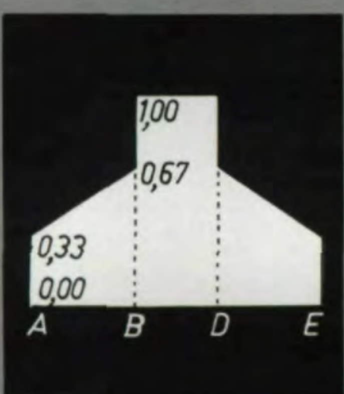

Fig. 1 – Traditional load sharing factor for a gear pair with unmodified tooth profiles.

Fig. 2 – Traditional load sharing factor for a gear pair with modified tooth profile, designed for high load capacity if the pinion is driver.

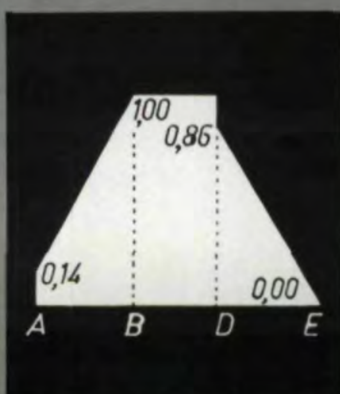

Fig. 3 – Traditional load sharing factor for a gear pair with modified tooth profile, designed for high load capacity if pinion is follower.

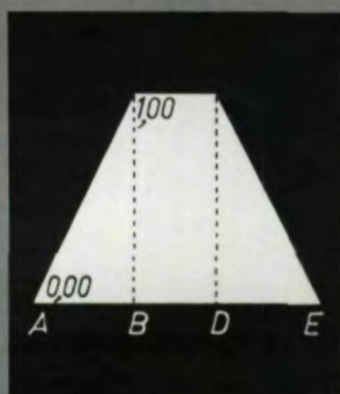

Fig. 4 – Traditional load sharing factor for a gear pair with modified tooth profile designed for smooth meshing.

The bulk temperature may be roughly approximated [1] by

$$\Theta_M \approx \Theta_{oil} + 0.47\ \Theta_{flmax} \text{ for sump lubrication} \quad (7)$$

$$\Theta_M \approx 1.2\ \Theta_{oil} + 0.56\ \Theta_{flmax} \text{ for jet lubrication} \quad (8)$$

The safety factor may be defined [1] by

$$S_B = \frac{\Theta_S - \Theta_{oil}}{\Theta_{Bmax} - \Theta_{oil}} \quad (9)$$

The iteration process for the maximum flash temperature depends solely on geometric influence factors which are concentrated in one single form factor. See Equation (3). That form factor depends on four parameters, u, z_1, x_1, x_2, of the seven independent parameters mentioned in Table 2. To examine the dependency on the four independent parameters, full calculations were made for 968 combinations. See Table 1.

Table 1 Parameters selected to cover the geometric field of application.

u	z_1	x_1	x_2
1	40 50 63 80 100 125 160	0,00 0,10 0,20 (z_1>16)	−0,20
1,6	32 40 50 63 80 100 125		0,00
2,5	25 32 40 50 63 80 100	0,30 0,40 0,50 0,60	0,20
4	20 25 32 40 50 63 80		0,40
6,3	16 20 25 32 40 50 63		

(Symbols, terms and units chosen in accordance with the international standard)

- a center distance (mm)
- A point of path of contact at tip of wheel
- b facewidth (mm)
- B lower point of transverse single contact
- C pitch point
- C_2 weight factor (value 1,5)
- D upper point of transverse single contact
- E point of path of contact at tip of pinion
- F_t tangential force at reference circle (N)
- GAM parameter on the line of action
- GAMA parameter on the line of action at point A
- GAMAB parameter on the line of action between A and B
- GAMB parameter on the line of action at point B
- GAMD parameter on the line of action at point D
- GAME parameter on the line of action at point E
- GAMED parameter on the line of action between E and D
- S_B safety factor, Equation (10)

NOMENCLATURE

TAA1 tangents of transverse tip pressure angle of pinion
TAA2 tangents of transverse tip pressure angle of wheel
TAT tangents of transverse working pressure angle
trapez number corresponding with figure number of
v pitch line velocity (m/s)
W_{Bt} specific tooth load(1)
x_1 addendum modification coefficient of pinion
x_2 addendum modification coefficient of wheel
X_B geometry factor, Equation (6)
X_{BE} geometry factor at point E
X_{Ca} tip relief factor (1)
XGAM load sharing factor
X_M thermal contact coefficient, Equation (4) $(K.N^{-3/4}\ .s^{1/2}\ .m^{-1/2}\ .mm)$
X_Q approach factor (1)
X_{top} form factor, Equation (3), (1) $(K.N^{-3/4}\ .s^{1/2}\ .m^{-1/2}\ .mm)$
X_ε contact ratio factor, (1)
X_Γ load sharing factor, Figs. 1 to 4
z_1 number of teeth of pinion
α_t transverse working pressure angle
α_y pressure angle of arbitrary point
β helix angle
Γ linear parameter on line of action, Equation (6)
Θ_B contact temperature, Equation (1) (°C), Part 1
Θ_{Bmax} maximum contact temperature, Equation (1), Part 2 (°C)
Θ_{fl} flash temperature, Equation (2) (°C), Part 1
Θ_{flaint} approximated mean value of the flash temperature, Equation (4) (°C), Part 1
Θ_{flmax} maximum flash temperature, Equation (2), Part 2 (°C)
Θ_{int} integral temperature, Equation (3) (°C), Part 1
Θ_M bulk temperature, Equation (8), (9) (°C)
Θ_{oil} oil temperature (°C)
Θ_S scoring temperature(1) (°C)
μ_{mC} mean coefficient of friction at pitch point(1)
μ_{my} mean local coefficient of friction(1)
π product of factors in comparison, Equation (7), Part 1

The essential part of the program, consisting of four procedures written in PASCAL, is presented in Table 2. The form factor X_{top} is the largest of four possible extreme values situated in point B, point D, located between A and B (marked by the parameter GAMAB), and located between D and E (marked by the parameter GAMED). (See Fig. 5)

$$X_{top} = \text{largest of} \begin{cases} \text{FMAX (GAMA, GAMB, GAMAB)} \\ \text{FLASHFACTOR (GAMB,trapez)} \\ \text{FLASHFACTOR (GAMD,trapez)} \\ \text{FMAX (GAME, GAMD, GAMED)} \end{cases} \quad (10)$$

The examination of the form factor produced two different conclusions; one about the confirmation of the flash temperature concept and the other about the influence of the load sharing factor.

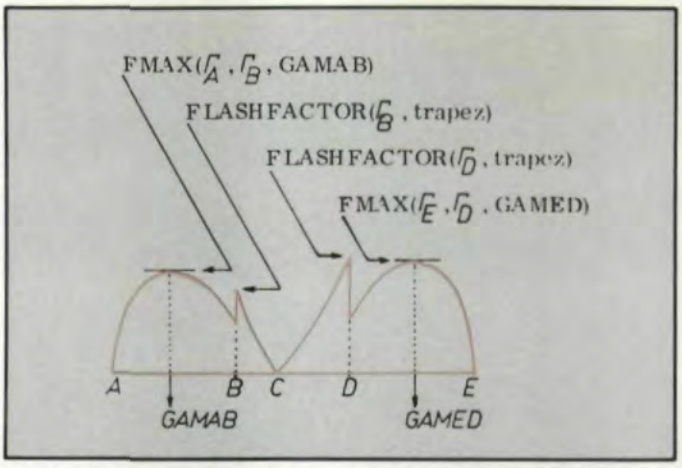

Fig. 5 – The form factor is the largest of four possible extreme values of the product $X_M X_B X_\Gamma$.

Table 2 Procedures for the determination of the maximum value of the product of thermal contact factor, geometric factor and load sharing factor.

```
procedure GAMMAPARAMETERS;
begin TAA1:-sqrt (sqr((1+2*(1+X1)/Z1)/0.93969262)-1);
      TAA2:-sqrt(sqr((1+2*(1+X2)/Z2)/0.93969262) -1);
      GAMA:=-Z2*(TAA2/TAW-1)/Z1; GAME:-TAA1/TAW-1;
      GAMD:-GAMA+6.283185/Z1/TAW; GAMB:-GAME-6.283185/Z1/TAW
end;

function FLASHFACTOR(GAM:real; trapez:integer):real;
var Q1,Q2:real;
begin U:-Z2/Z1; Q1:-1+GAM; Q2:-1-GAM/U; XGAM:-1;
  XB:-0.51*sqrt((U+1)/sqrt(Q1*Q2*U))*abs(sqrt(Q1) -sqrt (Q2));
  case trapez of
      1:begin Q1:-1/3; Q2:-1/3 end;
      2:begin if GAM<0 then Q1:-0 else Q1:-1/7; Q2:-6/7 end;
      3:begin if GAM>0 the Q1:-0 else Q1:-1/7; Q2:-6/7 end;
      4:begin Q1:-0; Q2:-1 end end;
  if GAM<GAMB then XGAM:-Q1+Q2*(GAM-GAMA)/(GAMB- GAMA);
  if GAM>GAMD then XGAM:-Q1+Q2*(GAME-GAM)/(GAME/-GAMD);
  FLASHFACTOR:-50*XB*XGAM
end;

function F(GAM:real):real;
begin F:-FLASHFACTOR(GAM,trapez)
end;

function FMAX(GA, GB:real; var G:real):real;
var A,B,C,D,FA,FB,FC,FD:real;
begin A:-GA; if GA<GB then B:-GB-0.01 else B:-GB+0.01; C:-A;
      FA:-F(A); FB:-F(B); FC:-FA;
      if FA>FB then begin C:-B; FC:-FB; B:-A; FB:-FA end;
      repeat A:-C; C:-B+0.382*(A-B); G:-B;
             FA:-FC; FC:-F(C); FMAX:-FB
      until (abs(B-A)<0.01) or (FB<FC); if FB<FC then
      repeat D:-C+0.382*(A-C); FD:-F(D);
         if FC>FD then begin A:-B; FA:-FB; B:-D; FB:-FD end
                  else begin B:-C; FB:-FC; C:-D; FC:-FD end;
         G:-C; FMAX:-FC
      until abs(B-A)<-0.01
end;
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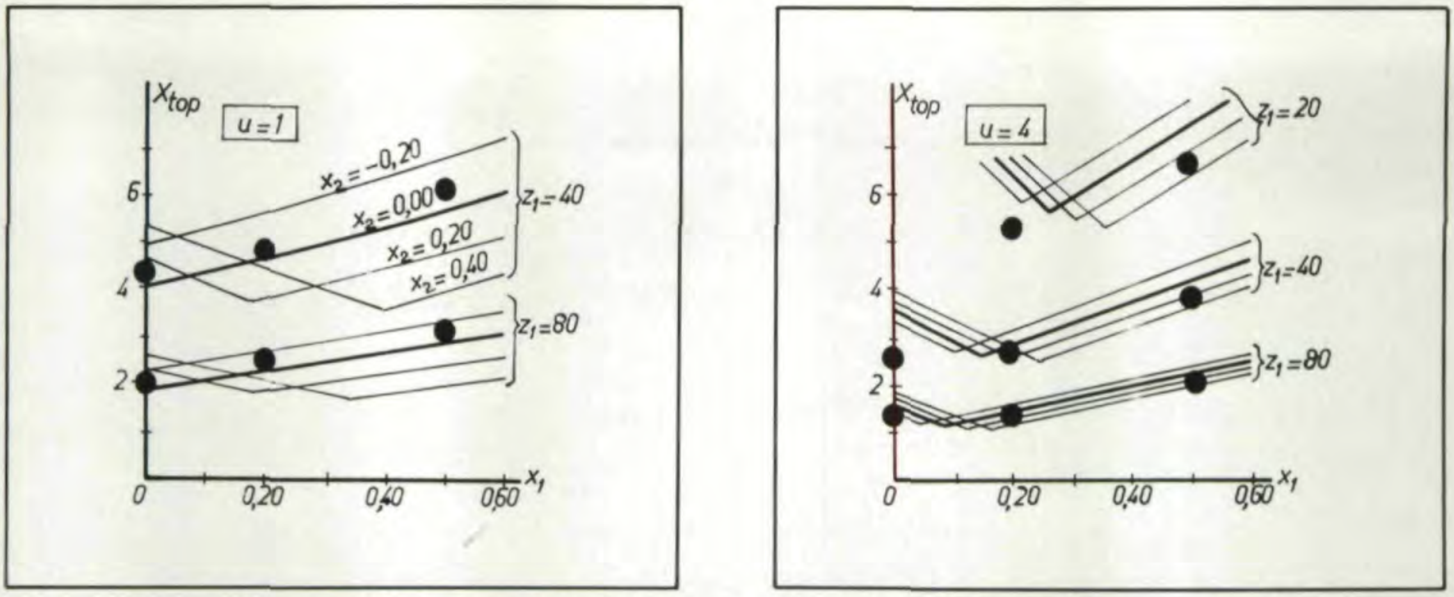

Figs. 6, 7 – Comparison of the form factor for the flash temperature with a similar one for the integral temperature (heavy dots).

Confirmation of the Flash Temperature Forumla

The curves of the form factor , Equation (3), for the determination of the maximum flash temperature coincide very well with similar curves for a quick approximation of the integral temperature, proposed by Hirt.[2] See Figs. 6 and 7. This confirms again that the integral temperature constitutes an approximation of the maximum contact temperature, in spite of all attempts to create a criterion with a different physical interpretation.

The empirical values characteristic of the integral temperature formula were determined by numerous tests. Hence, the integral temperature formula may be understood as a concise representation of gear tests, confirming the flash temperature criterion and, in addition, replacing the original tests of Blok, which were lost in May, 1940.

It is gratifying that the applicability of the flash temperature criterion for gears is confirmed in an unintentional way, leading to one criterion based upon the flash temperature concept, enriched with several practical influence factors gained in numerous gear tests and by field experience. See Fig. 8.

Field of Application

Without exception, the results of tests expressed in integral temperature can be applied to the flash temperature criterion. Similarly, if the flash temperature criterion is not applicable to a certain case, then the integral temperature criterion is not valid either.

The field of application of the flash temperature criterion covers straight and mild-extreme-pressure mineral oils, and, to a certain extent, extreme-pressure oils and perhaps synthetic oils. The theoretical basis of the flash temperature concept provides a boundary for its field of application, which prevents a dangerous situation similar to predicting pitting with a bending strength formula. Most likely, there are different physical causes for scuffing or related phenomena, and in the future additional criteria may be needed.

Diagrams of the Form Factor

Among the four parameters determining the shape and value of the form factor, the addendum modification coefficient of the wheel x_2 has been shown to be of less importance than the three other parameters u, z_1, and x_1. The curves shift to approximately the same curve if the ordinate x_1 is replaced by $x_1+(1,4/u)x_2$. See Figs. 6 and 7. Hence, the diagrams in Figs. 9 to 13, being exact for $x_2 = 0$, may be applied for any x_2. Moreover, the diagrams in Figs. 9 to 14 include an indication of the location of the maximum contact temperature on the path of contact.

The drastic reduction of several complicated geometric influence factors to only three parameters presents a convenient view on the consequences of the selection of geometric quantities. It may be helpful for the designer to achieve optimization.[3]

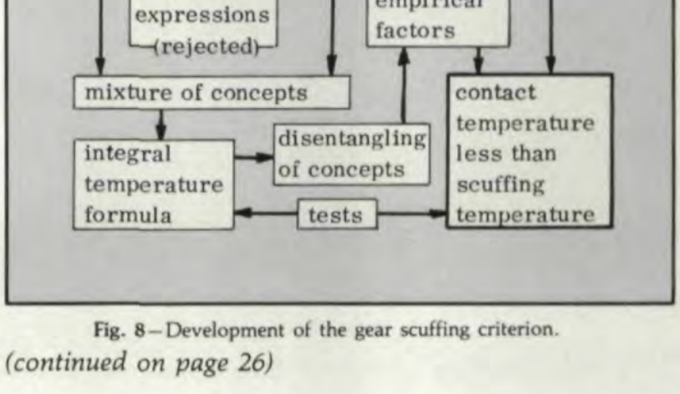

Fig. 8 – Development of the gear scuffing criterion.

(continued on page 26)

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(continued from page 23)

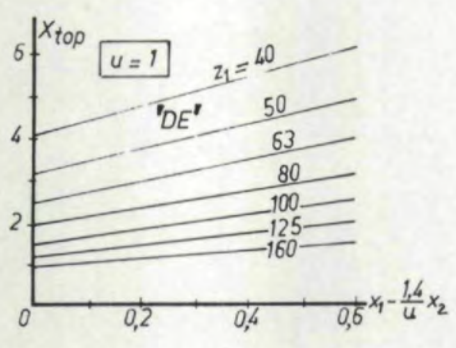


Fig. 9

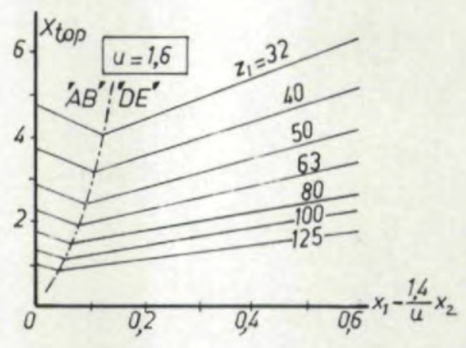


Fig. 12

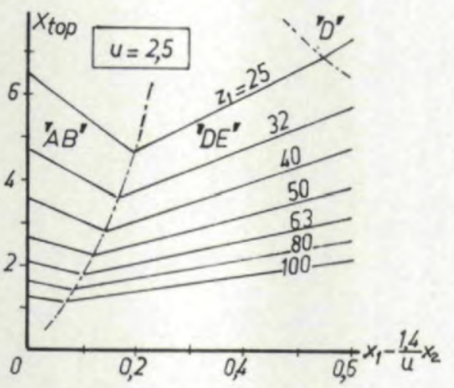


Fig. 10

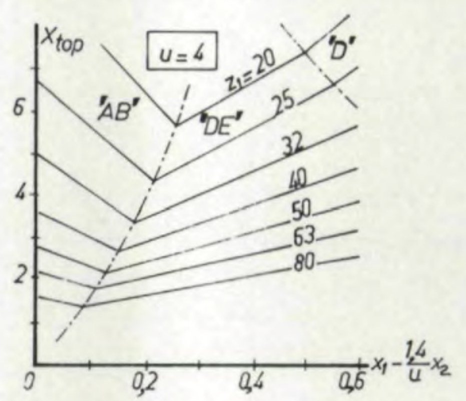


Fig. 13

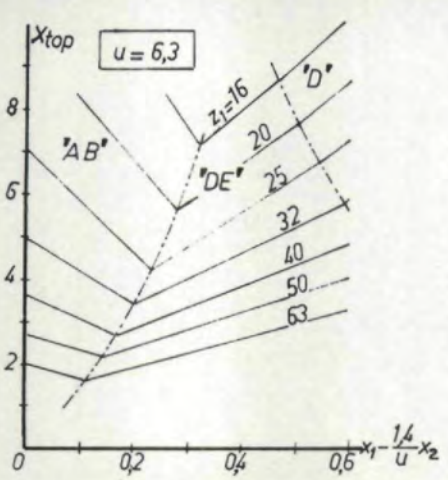
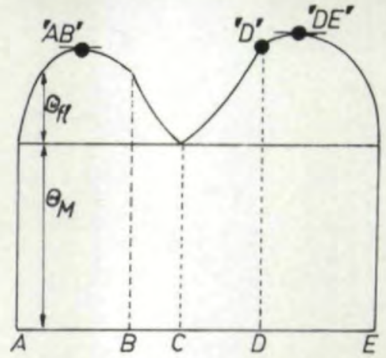


Fig. 11



Maximum contact temperature. For low values of x_1, the greatest maximum is between A an B, and X_{top} decreases with x_1. For larger values of x_1, the greatest maximum is between D and E, and X_{top} increases with x_1. The greatest maximum is at point D for largest values of x_1.

Fig. 14

Figs. 9-13 – Form factor X_{top} for the maximum flash temperature, valid for the load sharing factors for smooth meshing.

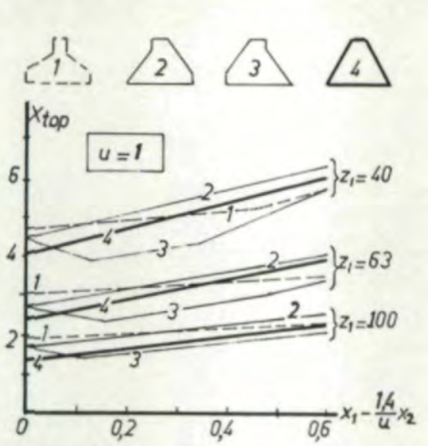


Fig. 15

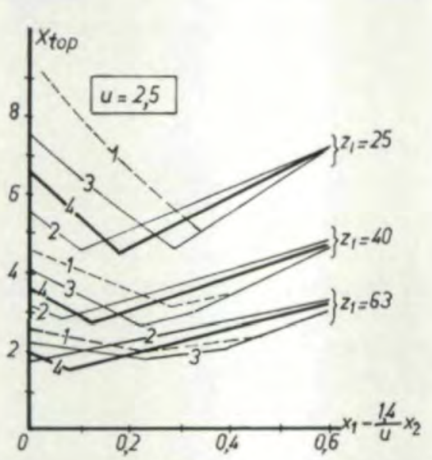


Fig. 16

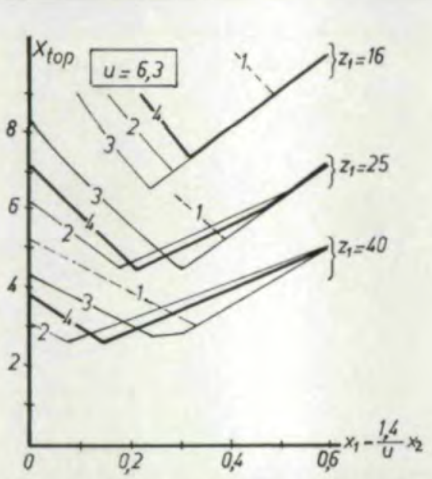


Fig. 17

Figs. 15-17 – Form factor X_{top} for the maximum flash temperature, valid for different load sharing factors, labeled with their figure numbers 1 to 4.

Another more or less unexpected result is shown in Figs. 15 to 17. The traditional load sharing factors, with distinct differences in shape and value, produce approximately the same form factors. See Figs. 1 to 4.

Conclusions

1. The scuffing (scoring) criterion is the flash temperature criterion according to Blok. It is based on the maximum contact temperature, being the sum of the bulk temperature and the maximum flash temperature. Enriched with some influence factors, it is presented in an international standard.
2. Several attempts to find a different criterion ultimately confirmed the applicability of the flash temperature criterion for engineering practice. The integral temperature, also presented in the same international standard, proved to be an approximation of the contact temperature.
3. For engineering practice the postulate of the maximum contact temperature (flash temperature criterion) was confirmed by numerous tests on gears, the results of which were expressed as a single value called integral temperature. All test results expressed in integral temperature data are fully applicable to the flash temperature criterion.
4. All geometrical influences can be concentrated in one form factor of the flash temperature formula. Diagrams of the form factor include an indication of the location of the maximum contact temperature on the path of contact. Such diagrams can be helpful for optimization of the design.
5. Different variants of traditional load sharing factors show nearly the same influence on the value of the flash temperature.
6. The theoretical basis of the flash temperature concept provides a basis for its field of application, preventing a misuse. To cover different physical causes for scuffing or related phenomena, additional criteria may be needed.

References:

1. ISO 6336/1: Introduction and general influence factors.
   ISO 6336/2: Calculation of surface durability (pitting).
   ISO 6336/3: Calculation of root stress (tooth breakage).
   ISO 6336/4: Calculation of scuffing load capacity.
   ISO 6336/5: Material qualities and endurance limits.
   In May, 1985, the above parts were approved by ISO/TC 60 to be published as Draft International Standards. These lengthy basic documents will be followed by simplified parts for several fields of application.
2. **HIRT, M.** "An Improved Method to Determine the Scoring Resistance of High Power Speed Gearing." Proc. 13th Turbomachinery Symposium.
3. **MENG, Huirong, LIU, Chong, CHEN, Qitai.** "Research Into the Increase of Scuffing Load Capacity by Optimization of Surface Flash Temperature of Gear Teeth." 2nd World Congress on Gearing, Paris, 1986.

Acknowledgement: This article was first presented at the 2nd World Congress on Gearing, March 3-5, 1986, Paris, France.

Hard Cutting – A Competitive Process in

Alfred Klein
Horsburgh & Scott,
Cleveland, Ohio

The higher load carrying capacities, compact dimensions and longer life of hardened gears is an accepted fact in industry today. However, the costs involved in case hardening and subsequent finishing operations to achieve these advantages are considerable. For example, in order to achieve desired running properties on larger gears, it has been necessary to grind the tooth flanks. This costly operation can now be replaced, in many cases, by a new Hard Cutting (HC) process which permits the cutting of hardened gears while maintaining extremely low tooling costs.

At the heart of this new process are new types of tools with especially hard inserts that enable case-hardened gears to be finished on the same heavy-duty gear cutting machines used for cutting the gears in the soft. The rotating table and generating slide of the machine perform a stepwise generating movement for cutting the teeth into the periphery of the hardened blank. Fig. 1 shows the principle of the HC process. The number of working cycles depends on the number of cutter teeth. Each cycle consists of a generating, reversing and indexing movement.

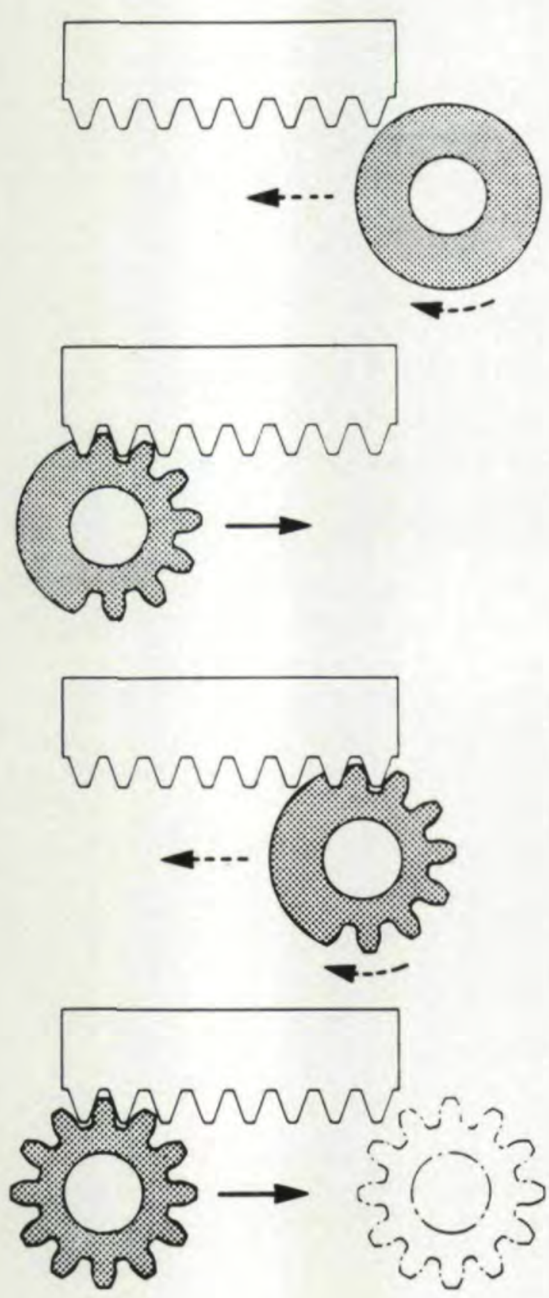


Fig. 1

Our experience indicates that AGMA 10 quality, coupled with a surface finish comparable to grinding, can be obtained with the HC process, replacing grinding for a wide variety of gears, including spur, single helical or narrow gap double helical design. When greater accuracy is required for turbine or marine propulsion drives, it can be obtained with a subsequent grinding operation. This grinding, however, is very

High Quality Gear Production

minimal since any distortion due to hardening has already been eliminated by the preceding HC process.

The tooling system is based on the single point tool. Such a tool is not restricted to a particular pitch and is, therefore, quite suitable for machining single gears. The cutting tool uses an arrangement of hard inserts on the body of the tool. Inserts consist of a 3 to 4 mm cemented carbide substrate to which a 1 mm layer of polycrystalline cubic boron nitride (CBN) is diffusion bonded. The main surface of the CBN layer forms the flank of the cutting edge, while the tool face is formed by the 1 mm thick edge of the CBN layer and the cemented carbide substrate. We see three major advantages result:

1. Hardened materials with a surface hardness up to HRC 65 can be accurately and cost-effectively cut because of the high mechanical and thermal shock resistance of the CBN tool material.
2. The advantageous arrangement of the insert means that regrinding is carried out on the narrow edge of the CBN layer, using diamond wheels to produce a perfectly sharp edge.
3. The design enables the tool to be reground on its face without affecting the profile, so that the tool can be utilized to its maximum without any sacrifice in accuracy.

Due to the tangential forces acting on the gear teeth during this process, high rigidity of the tool holder and machine parts is important. Because of this requirement, the machine used in the following tests uses hydrostatic bearing systems for the clapper box and ram guides, as well as hydraulic clamping on the rotating table and the generating slide during cutting.

Fig. 2

We recently conducted hard cutting trials on several large, but completely different gears and pinions. The specifications and tolerances for one of these test pieces are as follows:

Normal diametral pitch	1 DP
Number of teeth	23
Helix angle	20° 0′ 0″
Face width	9.0
Pressure angle	20°

Tolerance Type		AGMA Level	DIN Level
Profile:	4.000	14.5	2.6 (Ff)
Lead:	3.000	17.0	1.6 (Fb)
Pitch:	2.600	13.1	3.3 (fp)
Accumulated pitch error:	8.300	13.3	3.1 (Fp)

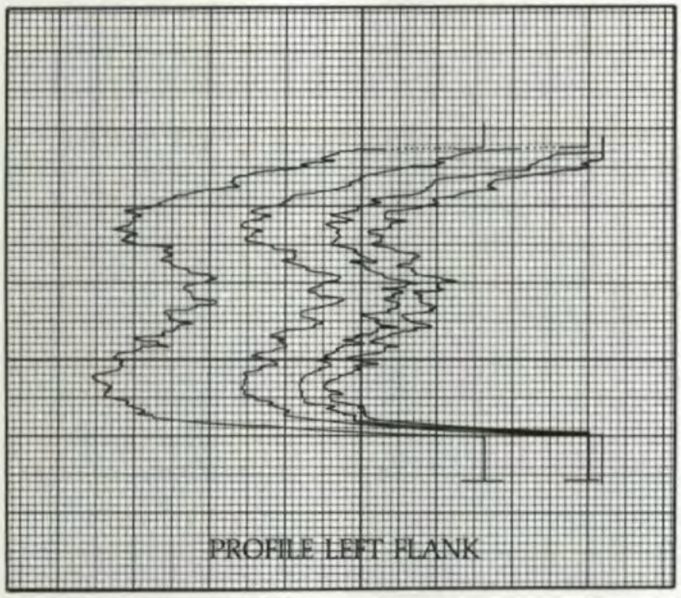
Note: Tolerances are specified in ten-thousandths of an inch. Surface finishes were in the 12 to 20 u range.

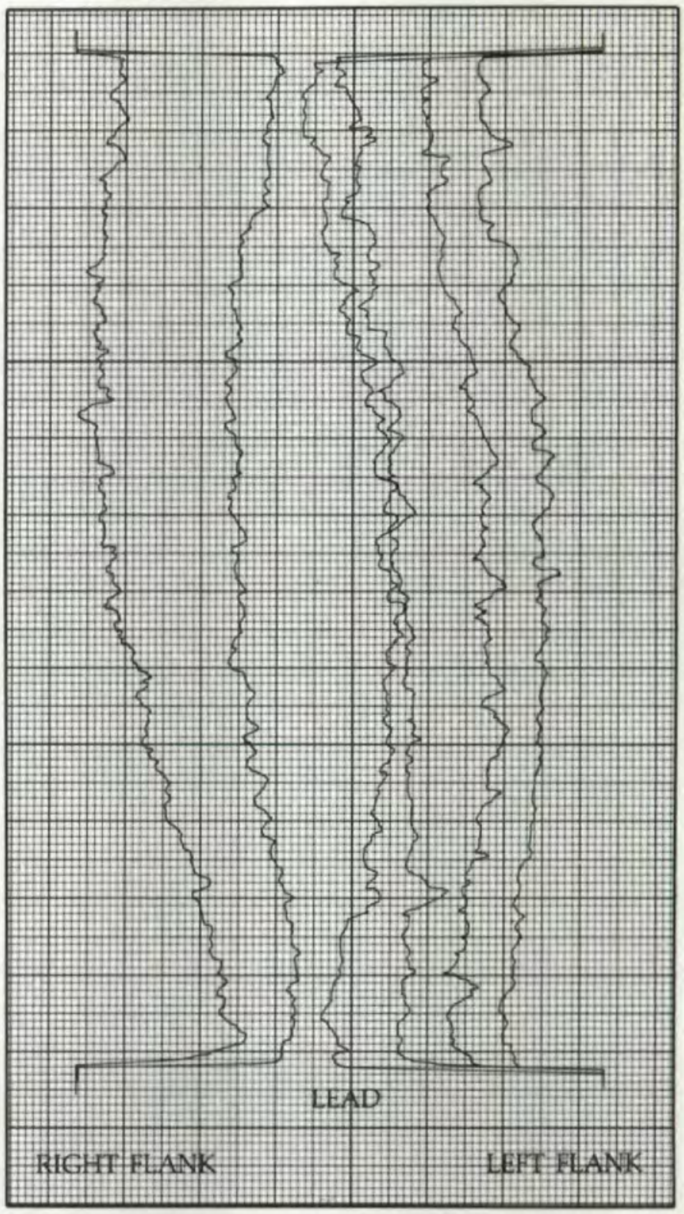
The test gear was cut by the single flank method. General setting data for all passes were

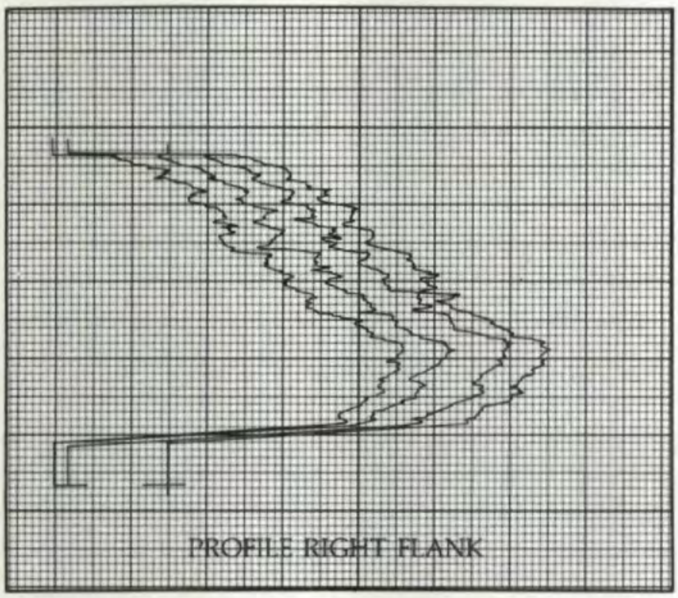
Ram stroke	13.4″	30 strokes/min.
Cutting speed		134 ft/min.
Generating path	2-2/3 pitches	
Cutter rake angle	−6°	

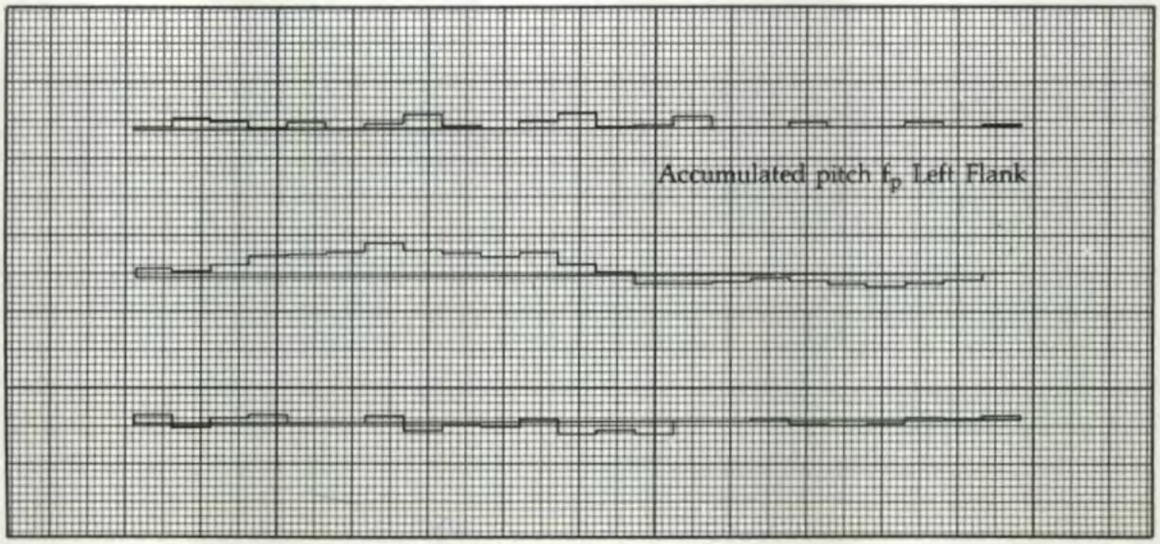
	L.H. Flank		R.H. Flank	
	Rough	Finish	Rough	Finish
Radial Infeed	.039″	.008″	.034″	.008″
Flank Layer Thickness	.0133″	.0027″	.0116″	.0027″
Total Strokes/Flank	104	88	104	88
(On Feed Setting)	(13)	(11)	(13)	(11)
Time per Flank	3.47′	2.93′	3.47′	2.93′
Indexing (Slow setting)	.60′	.60′	.60′	.60′
Total Time per Flank	4.07′	3.53′	4.07′	3.53′
Time per pass	81.4′	70.6′	81.4′	70.6′
Total Time per Gear	304 minutes			
Hardness of Test Gear	58 to 60 Rc			

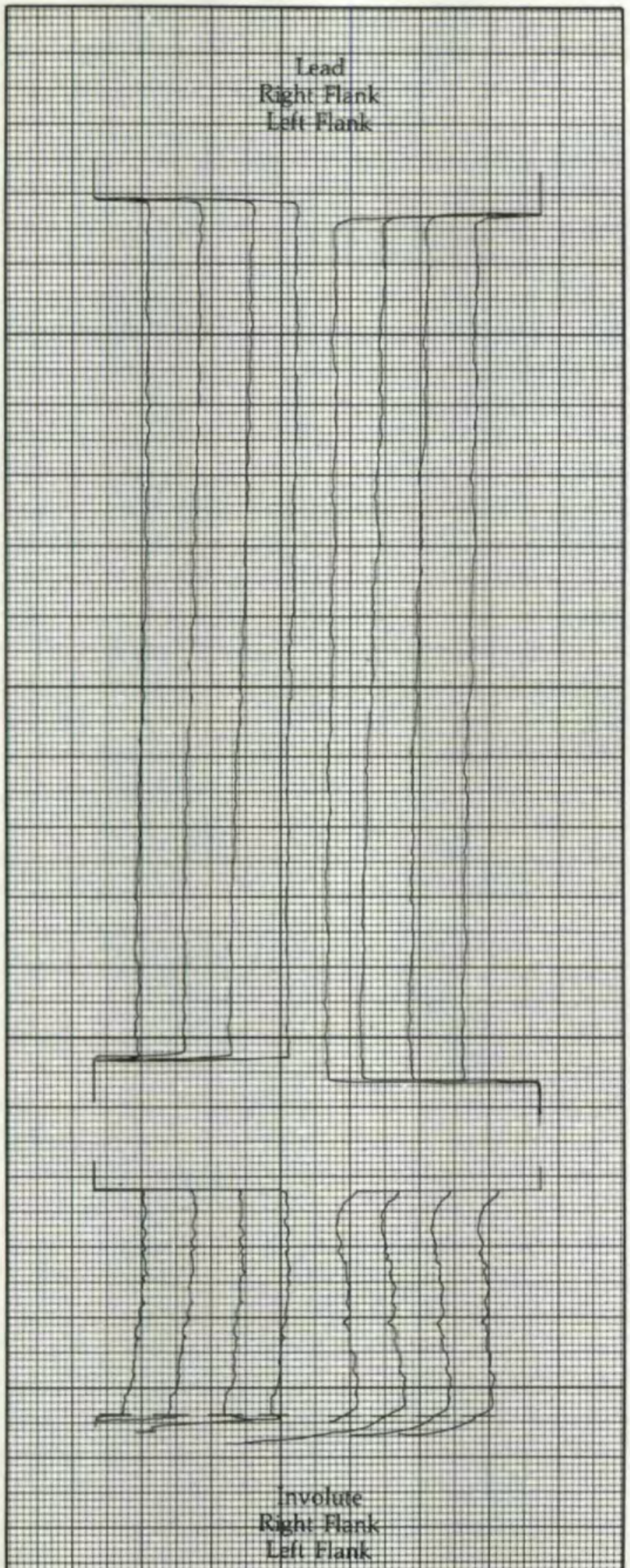
HEAT TREATED GEAR PRIOR TO HARD CUTTING PROCESS











FINISHED GEAR
AFTER HC PROCESS

The type of chip produced by the HC process is shown in Fig. 2. The tightly curled chips on the left are finishing chips and on the right are roughing chips. As our test samples show, .012"/.015" stock removal per flank per cutting pass is not unusual. The root fillet and diameter is not machined in this process; therefore, protuberance cutters should be used in the roughing or pre-heat treatment process.

Results and Conclusions

Our experience during these tests and in subsequent runs has shown that finishing times using the HC process are as much as nine times faster than traditional gear grinding methods. In many cases, it eliminates the need for a separate grinding step. It is important that the benefits of this process be recognized by gear designers as well as end-user applications engineers. The result will be new gear designs of compact dimension.

AUTHOR:

MR. ALBERT KLEIN *has nearly forty years of experience in the gear manufacturing industry. He received his early training with Hermann Pfauter Gear Hobbing in Ludwigsburg, West Germany, working with them until 1960. He has been with Horsburgh & Scott, producer of both custom gear designs and standard speed reducers, for 21 years.*

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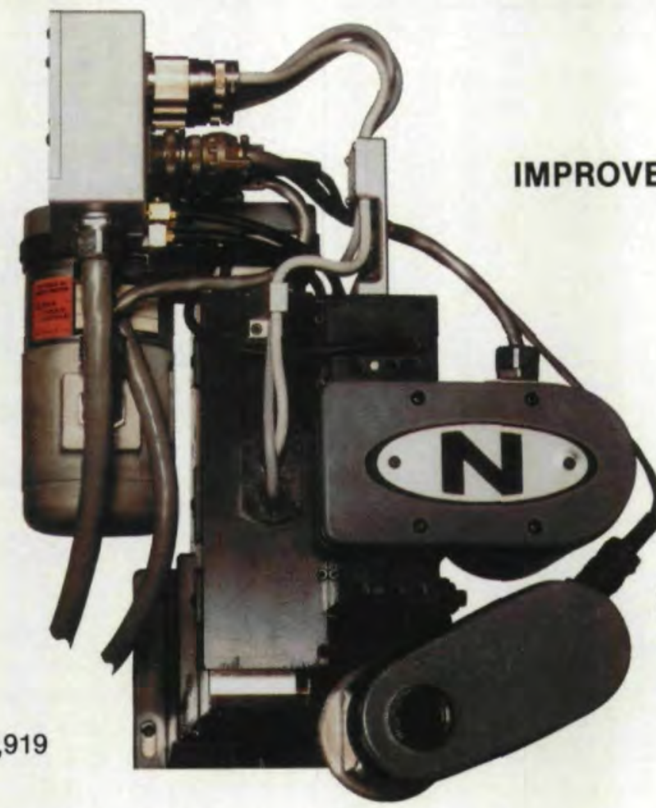

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RULES AND FORMULA *For Worm Gearing*

(Based on Lewis Formula)

TO FIND	RULE	FORMULA
Linear pitch	Divide the lead by the number of threads. (see "thread" below)	$P = \frac{L}{N}$
Addendum of worm tooth	Multiply the linear pitch by 0.3183.	$S = 0.3183P$
Pitch diameter of worm	Subtract twice the addendum from the outside diameter.	$d = 0 - 2S$
Pitch Diameter of Worm-Wheel	Multiply the number of teeth in the wheel by the Linear Pitch of the Worm, and divide the product by 3.1416.	$D = \frac{NP}{3.1416}$
Center Distance between worm and gear	Add together the Pitch Diameter of the Worm and the Pitch Diameter of the Worm-Wheel, and divide the sum by 2.	$C = \frac{D + d}{2}$
Whole depth of worm tooth	Multiply the Linear Pitch by 0.6866.	$W = 0.6866P$
Bottom Diameter of worm	Subtract twice the whole depth of tooth from the outside Diameter.	$b = o - 2W$

TO FIND	RULE	FORMULA
Helix Angle of worm	Multiply the Pitch Diameter of the Worm by 3.1416 and divide the product by the lead; the quotient is the cotangent of the Helix Angles of the Worm.	$\cot B = \frac{3.1416d}{L}$
Width of thread tool at end	Multiply the Linear Pitch by 0.31.	$T = 0.31P$
Minimum length of worm for complete action	Subtract four times the Addendum of the Worm thread from the outside Diameter of the wheel, square the remainder, and subtract the result from the square of the outside Diameter of the wheel. The square root of the result is the minimum length of Worm advisable.	$x = \sqrt{O^2 - (O - 4S)^2}$
Outside Diameter of worm	Add together the Pitch Diameter and twice the Addendum.	$o = d + 2S$
Pitch Diameter of worm	Subtract the Pitch Diameter of the Worm-wheel from twice the center distance.	$d = 2C - D$

P = Circular Pitch of Wheel and Linear Pitch of Worm;
L = Lead of Worm;
n = Number of Threads in Worm;
S = Addendum, or Height of Worm Tooth Above Pitch Line;
d = Pitch Diameter of Worm;
D = Pitch Diameter of Worm-Wheel;
o = Outside Diameter of Worm;
O = Outside Diameter of Worm-Wheel;
b = Bottom or Root Diameter of Worm;
N = Number of Teeth in Worm Wheel;
W = Whole Depth of Worm Tooth;
T = Width of Thread Tool at End;
E = Helix Angle of Worm and Gashing Angle of Wheel;
C = Distance Between Centers;
x = Threaded Length of Worm;
"Thread" = It is Understood that by the Number of Threads is Meant, Not Number of Threads per Inch, But the Number of Threads in the Whole Worm — One if It is Single Threaded, Four, if It is Quadruple Threaded, etc.

CONSIDERATIONS REGARDING STRENGTH OF WORM GEARING

The chief purpose of worm gearing is to reduce velocity. Hence, when designing worm drives it is essential that the diameter of the worm be kept as small as possible. Obviously if the diameter of worm is too large the worm gear may overheat and start undue wear.

"Industrial" Worm Gears are finished off so there are no sharp edges or corners on the teeth. This helps to eliminate friction and consequent heating. It also keeps from weakening the strength of the teeth.

Refer to table on page 198 for determining the strength of teeth. Table at right can be used for safe working unit stresses. The values are for single, double, triple and quadruple threads. In all cases the strength of the worm wheel is considered rather than the strength of the worm. It is safe practice to figure a worm gear as a spur gear insofar as strength of teeth is concerned.

TABLE OF WORKING STRESSES

For the Strength of Worm Gears

Used with "Lewis" Formula

Velocity in Feet Min. = V	Strength Factors = Y	Safe Working Unit Stress = S in Pounds per Sq. In.	
		Cast Iron	Phosphor Bronze
0	1.000	5300	8000
100	.857	4550	6800
200	.750	4000	6000
300	.666	3550	5350
450	.571	3000	4500
600	.500	2650	4000

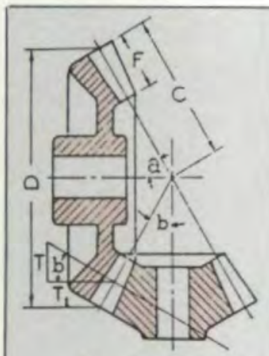
RULES AND FORMULAS *For the Strength of Bevel Gears (Based on the Lewis Formula)*

USE RULES AND FORMULAS (1) TO (4) IN ORDER GIVEN

No.	TO FIND	RULE	FORMULA
1	Velocity in feet per minute at the Pitch Diameter.	Multiply the product of the Pitch Diameter in inches; and the number of Revolutions per minute by 0.262.	$V = 0.262\ DR$
2	Allowable Unit stress at given Velocity.	Multiply the allowable static stress by 600 and divide the result by the Velocity in feet per minute plus 600.	$S = Ss \times \frac{600}{600 + V}$
3	Maximum safe Tangential load at Pitch Diameter.	Multiply together the allowable stress for the given velocity, the width of face, the tooth outline factor and the difference between the Pitch Cone Radius and the width of face: divide the result by the product of the Diametral Pitch and the Pitch Cone Radius.	$W = \frac{SFY\ (C\text{-}F)}{PC}$
4	Maximum safe Horsepower.	Multiply the safe load at the pitch line by the velocity in feet per minute, and divide the result by 33,000.	$H.P. = \frac{WV}{33{,}000}$

D = Pitch Diameter of Gear in Inches;
R = Revolutions per Minute;
V = Velocity in Feet per Minute at Pitch Diameter;
Ss = Allowable Static Unit Stress for Material; (or the allowable stress at zero velocity);
S = Allowable Unit Stress for Material at Given Velocity;
F = Width of Face;
N^1 = Number of Teeth in Equivalent Gear; (see diagram in table below);
Y = Outline Factor; (see table below);
P = Diametral Pitch (if circular pitch is given, divide 3.1416 by circular pitch to obtain diametral pitch);
C = Pitch Cone Radius;
W = Maximum Safe Tangential Load in Pounds at Pitch Diameter;
H.P. = Maximum Safe Horsepower.

Factors for Calculating Strength OF BEVEL GEARS



$$N^1 = \frac{\text{Number of Teeth}}{\text{COS.a}}$$

Table of Outline Factors (Y) for 14½° and 20° Involute

N^1	Outline Factor = Y		N^1	Outline Factor = Y	
	14½° Involute (Std.)	20° Involute		14½° Involute (Std.)	20° Involute
12	0.210	0.245	27	0.314	0.349
13	0.220	0.261	30	0.320	0.358
14	0.226	0.276	34	0.327	0.371
15	0.236	0.289	38	0.336	0.383
16	0.242	0.295	43	0.346	0.396
17	0.251	0.302	50	0.352	0.408
18	0.261	0.308	60	0.358	0.421
19	0.273	0.314	75	0.364	0.434
20	0.283	0.320	100	0.371	0.446
21	0.289	0.327	150	0.377	0.459
23	0.295	0.333	300	0.383	0.471
25	0.305	0.339	Rack	0.390	0.484

(Also see top of page 68)

END THRUST ON BEVEL AND MITRE GEARS

The Method of Calculation of End Thrusts is as Follows:

A = Pressure Angle of Gear Teeth.

K = Tooth Pressure at Middle of Tooth Face.

F = Separating Force = K x Tan. A.

B = Pitch Angle of Pinion.

T = Thrust on Pinion = K x Tan. A x Sin. B.

T_1 = Thrust on Gear = K x Tan. A x Cos. B.

The table at right gives the factors by which the tooth pressure is multiplied to find the thrust which give practically the same values found by solving the formulae for T and T_1 given above.

Gear Ratio	Pressure Angle A			
	14½°		20°	
	Gear	Pinion	Gear	Pinion
1 -1	.183	.183	.257	.257
1½-1	.215	.143	.303	.202
2 -1	.232	.116	.325	.163
2½-1	.240	.096	.338	.135
3 -1	.246	.082	.345	.115
3½-1	.249	.071	.350	.100
3¾-1	.250	.067	.352	.094
4 -1	.251	.062	.353	.088
4½-1	.253	.056	.355	.079
5 -1	.254	.051	.357	.072
5½-1	.255	.046	.358	.065

RULES AND FORMULAS *For the Strength of Gear Teeth*

(Based on the Lewis Formula)

USE RULES AND FORMULAS (1) TO (4) IN THE ORDER GIVEN

No.	To Find	Rule	Formula
1	Velocity in feet per min. at the pitch diameter	Multiply the product of the diameter in inches and the number of revolutions per minute, by 0.262	$V = 0.262\ DR$
2	Allowable unit stress at given velocity	Multiply the allowable static stress by 600 and divide the result by the velocity in feet per min. plus 600	$S = Ss \times \frac{600}{600 + V}$
3	Maximum safe tangential load at pitch diameter	Multiply together the allowable stress for the given velocity, the width of face, and the tooth outline factor; divide the result by the diametral pitch	$W = \frac{SAY}{P}$
4	Maximum safe horsepower	Multiply the safe load at the pitch line by the velocity in feet per minute, and divide the result by 33,000	$H.P. = \frac{WV}{33{,}000}$

D = Pitch Diameter of Gear in Inches;
R = Revolutions per minute;
V = Velocity in Ft. per Min. at Pitch Diameter;
Ss = Allowable Static Unit Stress for Material;
S = Allowable Unit Stress for Material at Given Velocity;
A = Width of Face in Inches;
Y = Outline Factor (see table below);
P = Diametral Pitch (if circular pitch is given, divide 3.1416 by circular pitch to obtain diametral pitch);
W = Maximum Safe Tangential Load in Lbs. at Pitch Diameter;
H.P. = Maximum Safe Horsepower.

Combining steps 2, 3 and 4 from above chart.

$$H.P. = \frac{Ss \times A \times Y \times V}{55 \times P \times (600 + V)}$$

FACTORS FOR CALCULATING *Strength of Gear Teeth*

No. of Teeth	Outline Factor = Y: 14½° Involute and Cycloidal	20° Involute	No. of Teeth	Outline Factor = Y: 14½° Involute and Cycloidal	20° Involute	No. of Teeth	Outline Factor = Y: 14½° Involute and Cycloidal	20° Involute
12	0.210	0.245	20	0.283	0.320	43	0.346	0.396
13	0.220	0.261	21	0.289	0.327	50	0.352	0.408
14	0.226	0.276	23	0.295	0.333	60	0.358	0.421
15	0.236	0.289	25	0.305	0.339	75	0.364	0.434
16	0.242	0.295	27	0.314	0.349	100	0.371	0.446
17	0.251	0.302	30	0.320	0.358	150	0.377	0.459
18	0.261	0.308	34	0.327	0.371	300	0.383	0.471
19	0.273	0.314	38	0.336	0.383	Rack	0.390	0.484

WORKING STRESSES *For the Strength of Gear Teeth*

Used in the Lewis Formula

Velocity in Feet per Minute = V	Strength Factors	Safe Working Unit Stress = S, in Pounds Per Square Inch					
		Cast Iron		Phosphor Bronze		Steel	
		Ordinary Workmanship	High-Grade Workmanship	Ordinary Workmanship	High-Grade Workmanship	Ordinary Workmanship	High-Grade Workmanship
0	1.000	6,000	8,000	9,000	12,000	15,000	20,000
100	0.857	5,150	6,850	7,700	10,300	12,800	17,100
200	0.750	4,500	6,000	6,750	9,000	11,200	15,000
300	0.666	4,000	5,350	6,000	8,000	10,000	13,300
450	0.571	3,400	4,550	5,150	6,850	8,550	11,400
600	0.500	3,000	4,000	4,500	6,000	7,500	10,000
900	0.400	2,400	3,200	3,600	4,800	6,000	8,000
1,200	0.333	2,000	2,650	3,000	4,000	5,000	6,650
1,800	0.250	1,500	2,000	2,250	3,000	3,750	5,000
2,400	0.200	1,200	1,600	1,800	2,400	3,000	4,000

(continued on page 48)

BACK TO BASICS...

Rotary Gear Honing

John P. Dugas
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Rotary gear honing is a hard gear finishing process that was developed to improve the sound characteristics of hardened gears by:

1. Removing nicks and burrs.
2. Improving surface finish.
3. Making minor corrections in tooth irregularities caused by heat-treat distortion.

The process was originally developed to remove nicks and burrs that are often encountered in production gears because of careless handling. Further development work with the process has shown that minor corrections in tooth irregularities and surface finish quality improvement can be achieved. These latter improvements can add significantly to the wear life and sound qualities of both shaved and ground hardened gears.

Gear honing does not raise tooth surface temperature, nor does it produce heat cracks, burned spots or reduce skin hardness. It does not cold work or alter the microstructure of the gear material, nor does it generate internal stresses.

Honing can be applied to both external and internal spur and helical gears utilizing a variety of specialized types of honing machine tools. Both taper and crown honing operations can be carried out if desired.

How the Process Works

The process uses an abrasive-impregnated, helical gear-shaped tool. This tool is generally run in tight mesh with the hardened work gear in crossed-axes relationship under low, controlled center distance pressure.

The work gear is normally driven by the honing tool at speeds of approximately 600 surface ft. per minute. During the work cycle, the work gear is traversed back and forth in a path parallel to the work gear axis. The work gear is rotated in both directions during the honing cycle. The process is carried out with conventional honing oil as a coolant.

The honing tool is a throw-away type that is discarded at the end of its useful life. The teeth are thinned as the tool wears. This tooth thickness reduction can continue until root or fillet interference occurs with the work gear. Then the O.D. of the hone can be reduced to provide proper clearance.

Eventually, thinning of the hone teeth also results in root intereference with the outside diameter of the work gear. When this condition occurs, the hone is generally considreed to be at the end of its useful life. In some isolated cases, it has been found practical to re-cut the hone root diameter with a grinding wheel to provide additional hone life.

Usually the amount of stock removed from the gear tooth by honing ranges from 0.0005" to 0.002" measured over pins.

The production rate at which honing operations can be carried out depends on the pitch diameter and face width of the work. A gear 1" diameter by 1" width can be honed in approximately 15 seconds. A gear 24" in diameter by 3" face width will require approximately 10 minutes honing time. Of course, honing of salvage gears requires longer cycles.

External Gear Honing Machines

A typical 24" external gear honing machine has the motor-driven honing tool mounted at the rear of the work spindle. The work spindle is mounted on a tilting table that can be positioned to provide four selective modes of operation.

The first mode is called loose backlash, where the hone and work gear are positioned in loose backlash operation on a fixed center distance. This method is sometimes utilized to slightly improve surface finish only, primarily on fine pitch gears with minimum stock removal.

The second mode of operation is called zero backlash. Here the work gear is positioned in tight mesh with the honing tool. The table is locked in fixed center distance location with a pre-selected hone pressure. This method is sometimes used to provide maximum gear tooth runout correction with minimum stock removal.

The third and most generally applied mode of operation is called constant pressure. The work gear is held in mesh with the honing tool at a constant pressure. This method removes nicks and burrs and provides maximum surface finish improvement in a minimum time.

The fourth mode of operation is called differential pressure. A pre-selected low pressure is present between the hone and the low point of an eccentric gear, and a pre-selected increased amount of pressure is present between the hone and the high point of eccentricity. This method has all of the desirable features of the constant pressure method plus the ability to slightly correct eccentricity. The amount of eccentricity in the gears with differential pressure honing may cause the hone to wear faster than the constant pressure method.

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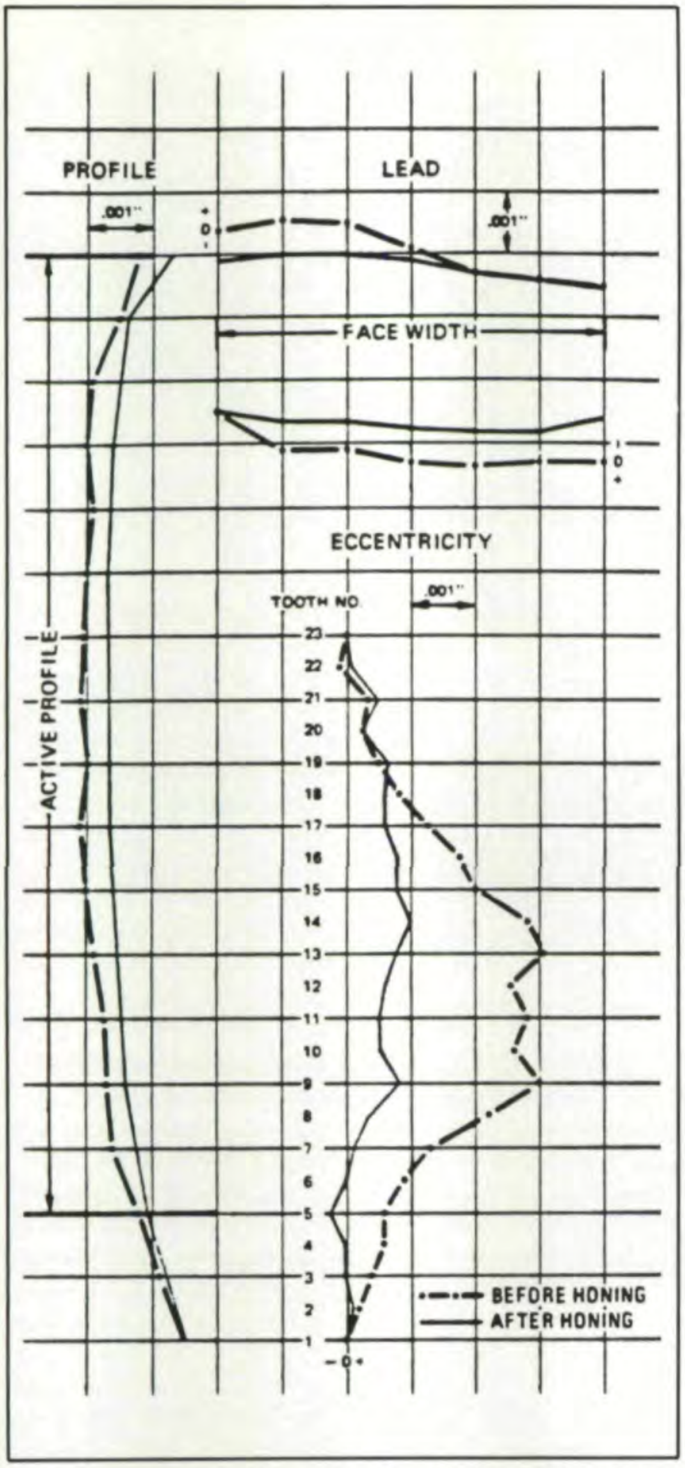

Fig. 1 – Improvement in tooth accuracy achieved by honing a shaved helical truck transmission gear.

Rotary Gear Honing Tools

Honing tools are a mixture of plastic resins and abrasive grains such as silicon carbide, that is formed in a precision mold. They are made in a wide variety of mix numbers with grits ranging from 60 to 500, to suit special production and part requirements.

Honing tools are divided into three different types for three different applications.

1. The standard type honing tool is made in a variety of resin and abrasive mixes for gears that have been shaved and heat treated. It removes nicks and burrs, improves tooth spacing and runout and can provide surface finishes down in the 8 to 14 μ range.

2. The "AA" ground honing tool, similar to the standard type tool, has precision tooth forms. All critical dimensions on this tool are held within .0002". It is used on ground hardened gears to improve spacing and runout and provide surface finishes in the 8 to 10 μ range.

3. The polishing type honing tool is a flexible, porous polyurethane tool that will bring the surface finish down to the 4 to 6 μ range for total contact on ground or shaved gear teeth that have been previously honed. Polishing action is achieved by using an abrasive liquid compound during the finishing process.

Honing tools are made in diameters ranging from 3½" (for internal gears) to 14" with face width from 1/2" to 2". Most gear hones are made with approximately 9" diameter.

Selection of the proper honing tool depends on the tooth finishing method, the gear material, the honing application requirements and machine capacities.

Honing Shaved Gears

Traditionally, tooth surface finishes in the range of 25 to 40 μ have been provided by the rotary gear shaving operation. The honing process, because it is not basically a heavy stock removal or tooth correction process, *cannot* substitute for gear shaving, which is performed on the soft gear. In fact, the tendency of a hone to charge a gear under 40 Rockwell C hardness with abrasive particles makes honing of soft gears a questionable application.

However, because a gear has to be heat treated, a process that usually roughens the tooth surface to a degree, the honing process tends to restore the hardened tooth surface finish to its original shaved condition and actually improves it. In all cases, the honed surface finish is better than the surface finish before honing. (Fig. 1).

To hone production gears, economy dictates that one grit
(continued on page 48)

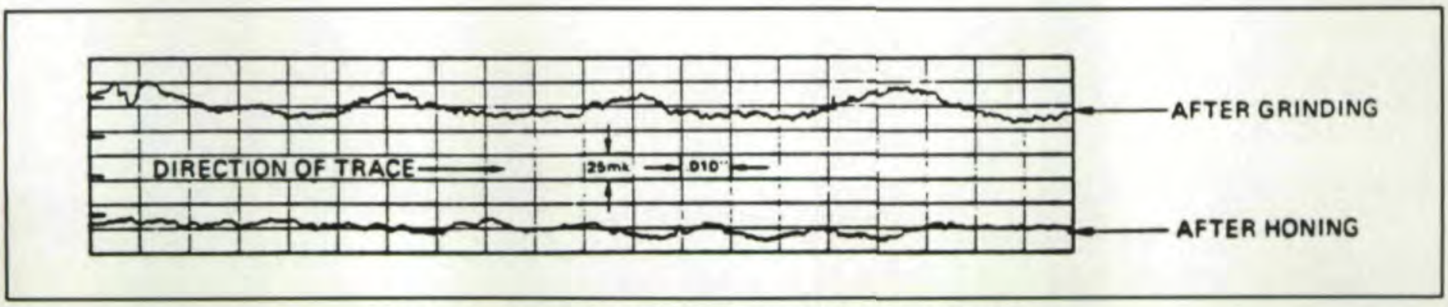

Fig. 2 – Proficorder checks of ground gear teeth before and after honing.

BACK TO BASICS...

Gear Roll-Finishing

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National Broach & Machine,
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In this discussion of gear roll-finishing particular attention is called to the special tooth nomenclature resulting from the interaction between the rolling die teeth and the gear teeth. To eliminate confusion the side of a gear tooth that is in contact with the "approach" side of a rolling die tooth is also considered to be the approach side. The same holds true for the "trail" side. Thus, the side of the gear tooth that is in contact with the trail side of a rolling die is also considered to be the trail side.

Gear roll-finishing (Fig. 1) is much different from gear shaving in that a flow of material rather than a removal of material is involved. A study of gear tooth action is required to analyze the material flow in the rolling process. In Fig. 2 as a gear rolling die tooth engages the approach side of a workpiece tooth, sliding action occurs along the line of action in the arc of approach, in a direction from the top of the gear tooth toward the pitch point where instantaneous rolling action is achieved. As soon as the contact leaves the pitch point, sliding action occurs again, but in the opposite direction toward the pitch point in the arc of recession.

What is more interesting, however, is that the contact between the die and work gear teeth on the trail side produces exactly the opposite direction of sliding to that on the approach side. (See Fig. 3.) The result of these changing directions of sliding is that material is being compressed toward the pitch point on the approach side and extended away from the pitch point on the trail side. (See Fig. 4.)

This action causes a greater quantity of material to be displaced on the trail side than on the approach side by a ratio of about three to one. On the approach side, the tendency is to trap the material rather than permit it to flow toward the top and root of the teeth as on the trail side. Thus, unlike the situation in a metal removal process like gear shaving, the quantity of material flow during the rolling process, as well as the hardness of that material, have a significant effect on the accuracy of the produced form.

Fig. 1 (center) – Operating principle of double-die gear roll-finishing.

In successful roll-finishing, an undercut near the root section, such as is found conventional preshaved tooth forms, is desirable. Since most production gears are also provided with a tip chamfer, the material will tend to be pulled up into the chamfer on the trail side and down away from the chamfer on the approach side.

As a result, some adjustments in hobbed tooth tip chamfer depths and angles are required to balance out the opposed metal flow conditions on each tip side. These chamfer depths and angles have to be held to close tolerances.

If too much stock is left for gear roll-finishing, or if the gear material is too hard (above approximately 20Rc), several conditions may result. The sliding action on the approach side of the tooth may cause a "seaming" of material that builds up in the area of the pitch point. On the trail side, the flow of excess material may result in a burr on the tip of the gear tooth and a "slivering" of material into the root area. Fig. 5 shows the condition of a roll-finished gear tooth when too much stock is flowed or high hardness conditions are encountered.

In Fig. 6 photomicrographs show the conditions encountered when stock removal is excessive and material hardness is too high. A seam is evident in the approach side of the tooth at the left in

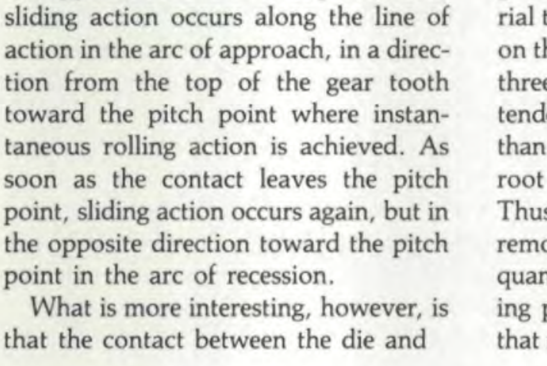
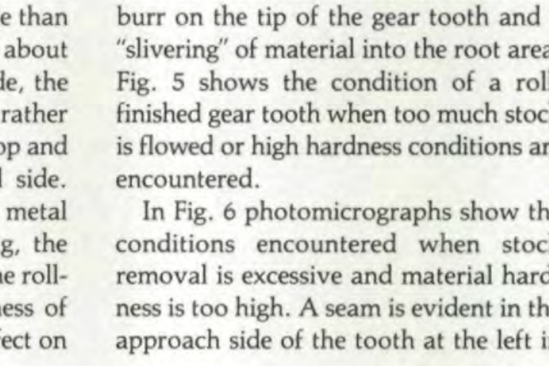


Fig. 2 (left) – Contact action between one tooth of a workpiece and the approach side of a rolling die tooth.

Fig. 3 (right) – Contact action between the tooth of a workpiece and the trail side of a rolling die tooth.



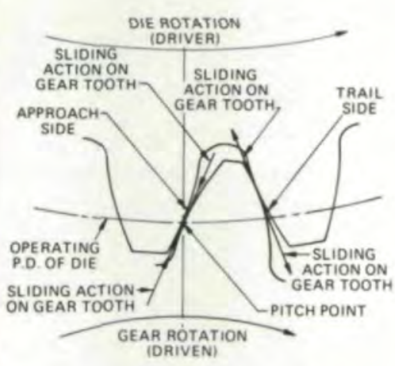


Fig. 4 (left) – Differing flow directions induced by each side of a die tooth with gear roll-finishing.

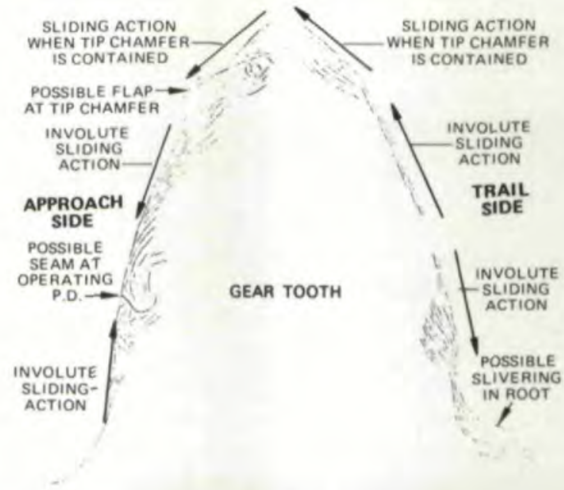


Fig. 5 (right) – Tooth flow pattern that results when too much stock is left for roll-finishing, or when material hardness is excessive.

the area of the operating pitch diamater. The trail side photomicrograph at the right in this figure shows slivering in the root portion with about 0.004-in. of lapped-over metal, and about 0.002-in. deep surface cold-working of the material.

In contrast, photomicrographs in Fig. 7 show the excellent tooth structure that can be achieved with roll-finishing if stock reduction is held to a minimum and material is not too hard. No evidence of cold-working or seaming is seen in the approach side at the left. In the trail side at the right in this figure, no evidence of slivering or cold-working is seen.

The amount of stock reduction with roll-forming should be held to about one half of that normally associated with shaving if seaming and slivering are to be avoided. The burr condition on the tip of the trail side of the tooth can be improved by close control of the angle and location of the protective tooth chamfer generated by the hob in the tooth generating operation.

Gear Rolling Dies

Since roll-finishing involves material

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flow rather than metal removal, it should be expected that the tooth form on the die would not be faithfully reproduced on the workpiece tooth because of minute material springback and material flow conditions.

Even with gear shaving it has been found necessary to modify the shaving cutter teeth profiles somewhat to produce a desired form on the work gear teeth. Experience to date has shown that a different type of tooth form modification is required for gear roll dies than for gear shaving cutters. As with gear shaving cutters, the correct amount of gear rolling die tooth form modification is determined from an extensive development program. Less rigid gear roll-finishing machines usually require greater and varying die form modifications.

Gear roll dies are made from a special fatique and impact resistant high speed steel to the tolerances shown in Table 1.

Gear Rolling Machines

Several important design considerations have be met in a roll-finishing machine. These include rigidity, strength, high speed loading, die phasing, and independent adjustment for die axis and die positioning.

The force required to roll-finish a gear depends upon its width, diametral pitch, tooth shape, cycle time material, and hardness.

Double-Die Gear Rolling

The double-die machine shown in Fig. 8 is a vertical design with the dies mounted one above the other. Such a design provides maximum rigidity, requires minimum floor space and also gives max-accessibility for a hinged automatic work loader as well as die head positioning adjustments.

Table 2 and Fig. 9 illustrate the range of gearing for which gear rolling dies have been produced for finish-rolling production applications.

Single-Die Gear Rolling

A single-die gear rolling process is ideally adapted for low and medium production finishing operations where both roll-finishing and shaving operations or roll-finishing only are done economically.

A single gear rolling die is mounted in a heavy-duty gear head above the workpiece in Fig. 10. The die is driven by an electric motor to provide rotation of the workpiece that meshes with it. Normally, semi-automatic loading methods are utilized on single-die roll finishing machines whose work cycles are somewhat longer than those of the fully automatic, double-die machines.

The workpiece is mounted on an arbor between head and tailstock. In operation, the table supporting the head and tail-

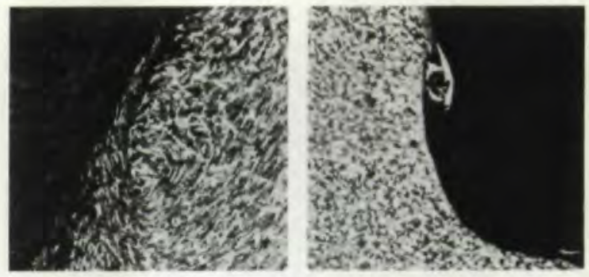

Fig. 6 – Photomicrographs of a gear tooth with high hardness. The approach side, left, has a seam in the area of the pitch diameter. The trail side, right, shows where excessive stock has caused cold-working and a sliver near the root.

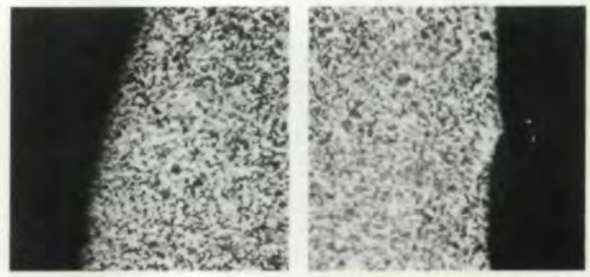

Fig. 7 – Photomicrographs of a properly roll-finished gear tooth. The approach side, left, has no seaming. The trail side, right, shows no slivering or cold-working.

Table 1 – Tolerances for Gear Roll-Finishing Dies

Die Specification	Tolerance-In.
Involute Profile (True Involute Form)— Active Length, tiv	
Through 0.177-in. Working Depth	0.00015
0.178 Through 0.395-in. Working Depth	0.00020
Lead—(Uniformity-tiv Per Inch of Face)	0.0003
Parallelism— (Opposite Sides of Same Tooth Alike Within)	0.0002
Helix Angle— (Deviation From True Angle—Per Inch of Face)	0.0005
Tooth Spacing— (Adjacent Teeth at Pitch Diameter)	0.00015
Circular Pitch—(Variation-tiv)	0.0002
Spacing Accumulation— (Over Three Consecutive Teeth)	0.00025
Runout—(tiv at Pitch Diameter)	0.0004
Face Runout—(tiv Below Teeth)	0.0002
Tooth Thickness	Minus 0.0010
Hole Diameter	Plus 0.0002

Note: Dies can be made in pairs alike within 0.0005-in. measured over pins, if necessary

Fig. 8 – Operating components of a double-die gear roll-finishing machine.

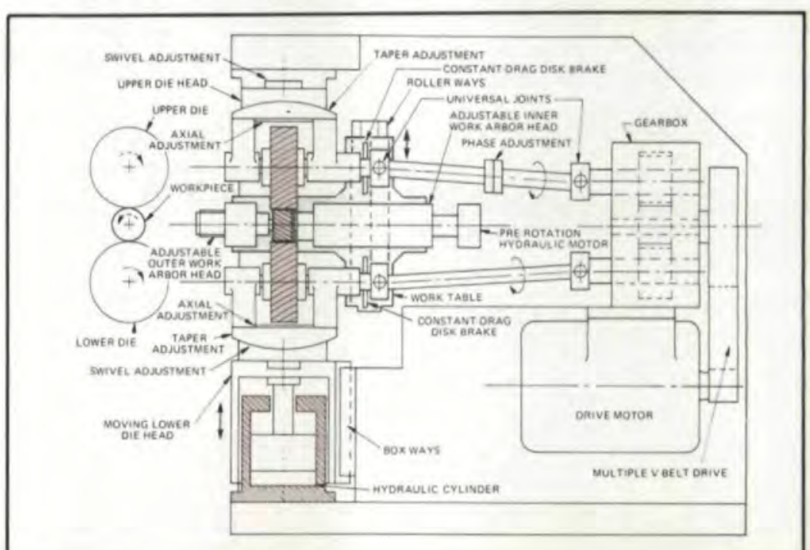

Table 2—Data on Roll-Finished Gears

No. Teeth	Pitch Diameter (in.)	Normal Diametral Pitch	Normal Pressure Angle	Helix Angle	Hand	Face Width (in.)	Material
26	4.6666	6.539	18° 28′	23° 25′	L	1.380	8620
25	3.3667	8.8709783	16° 30′	33° 10′	L	0.918	8620
14	1.0711	14	20°	21°	L	0.727	5140H
17	1.2143	15.1535	18° 35′ 09″	22° 30′	L	0.758	4024
28	2.0000	15.1535	18° 35′ 09″	22° 30′	R	3.04	4024
18	1.2542	15.5	17° 30′	22° 11′ 30″	R	1.935	4620
16	0.9621	18	18° 30′	22° 30′	R	0.728	5130, Fine Grain (5-8)
34	2.0445	18	18° 30′	22° 30′	L	0.860	5130, Fine Grain (5-8)
20	1.1580	18.5	18°	21°	R	0.874	4027H
19	1.0549	19.3	20°	21° 03′ 42″	R	0.705	4027H

Fig. 9—Gears that have been successfully roll-finished.

stock is fed upward by a unique, air-powered, heavy-duty radial feed system. The continuous upfeed of the table provides the large force necessary to roll-finish the gear teeth.

During the work cycle, the workpiece can be rotated in one direction for one part of the cycle, then reversed and rotated in the other direction for the balance of the cycle. This double-rotation sequence tends to balance the metal flow action on the approach and trail sides of the work gear teeth.

Tooth thickness size of the workpiece is controlled by adjusting the height of the table with a handwheel-controlled elevating screw.

Acknowledgement: *This article was presented at the SME Gear Processing Technology Clinic, Nov., 1986, Schaumburg, IL.*

GEAR DESIGN OPTIONS . . .
(continued from page 18)

Table 6

PA	DP	HPC	HPB
14.5°	10	12.7	26.7
	12	13.4	24.1
	14	13.8	22.0
	16	14.0	20.2
	20	14.3	17.3
20°	10	15.9	33.6
	12	16.7	30.0
	14	17.1	27.1
	16	17.5	24.6
	20	17.9	20.8
25°	10	18.8	40.3
	12	19.6	35.8
	14	20.1	32.1
	16	20.5	29.0
	20	21.0	24.4

and that for very high reliability or less than 1 in 10,000 failures is 1.5. The computed capacity would be reduced in accordance with the degree of reliability desired.

Comments

The opportunity to use a computer to assist in gear design can present the designer with a large choice of possible candidates. Exercising options and choices can help in zeroing in on a final selection, but obviously there is no one design that fits all requirements. Many different gear sets can be suitable.

Designers are not restricted to standard pressure angles or pitches nor to standard tooth forms. Appropriate basic data can be placed in the program if desired, for example, to use existing tooling even if metric module, to reduce costs or time delays.

No gear design is really considered final or so perfect that the parts can go into production without some model testing or pilot manufacturing. It is at this point that the unforseen factors can be dealt with.

References:

1. AGMA 110.04-1980, "Nomenclature of Gear Tooth Failure Modes." American Gear Manufacturers Association, Arlington, Virginia.
2. AGMA 218.01-1982, "For Rating the Pitting Resistance and Bending Strength of Spur and Helical Involute Gear Teeth." American Gear Manufacturers Association, Arlington, Virginia.

This article was first presented at the SME Gear Processing and Manufacturing Clinic, Schaumburg, IL Nov. 11-13, 1986.

Gear Design Program

```
10 REM spurgear.bas
20 CLS:KEY OFF
30 PI=3.141592654#:RA=180/PI
40 INPUT "Diametral Pitch          =",DP"
50 INPUT "Pressure Angle"          =",PA
60 INPUT "Center Distance"         =",CD
70 INPUT "Face Width               =",F
80 INPUT "Steel Allow Tens.        =",SAT
90 INPUT "Steel Allow Comp.        =",SAC
100 INPUT "Pinion RPM              =",RPM
110 INPUT "Pinion Teeth            =",NP
120 INPUT "Gear Teeth              =",NG
130 INPUT "Pinion CTT              =",CTP
140 INPUT "Backlash                =",BL
150 PRINT"     Program in progress, turn printer on, please."
160 HADD=1.25/DP:TIPR=.3/DP:HDED=1/DP:CLR=.25/DP
170 HTF=1.5708/DP-2*HADD*TAN(PA/RA)
180 TCD=(NP+NG)/2/DP
190 PDP=NP/DP:PDG=NG/DP:PRP=PDP/2:PRG=PDG/2
200 BDP=PDP*COS(PA/RA):BDG=PDG*COS(PA/RA)
210 OPPDP=2*CD*NP/ (NP+NG):OPPDG=2*CD*NG/(NP+NG)
220 OPPRG=OPPDG/2:OPPRP=OPPDP/2
230 CP=PI/DP:CSP=CP-CTP:BRG=BDG/2:BRP=BDP/2:BP=CP*COS(PA/RA)
240 PDED=(CSP-HTF)/2/TAN(PA/RA)
250 RDP=PDP-2*PDED
```

(continued on page 44)

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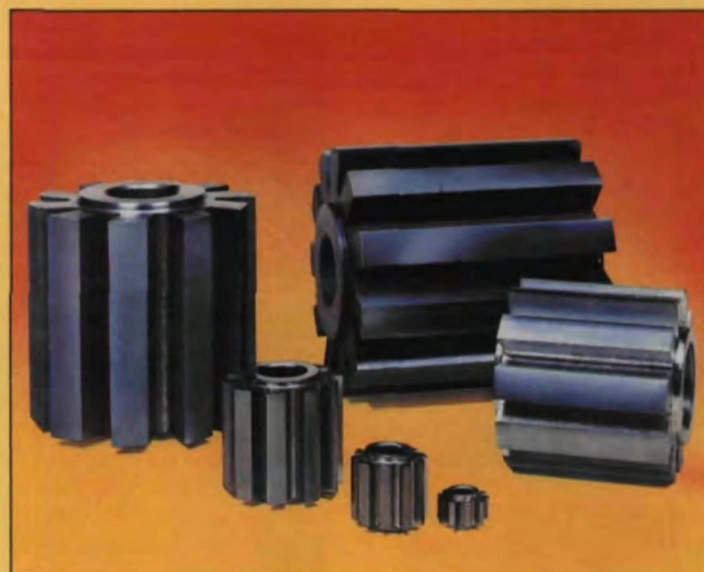

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16–35	1.875	1.875	.750	M42
8–20	3.000	3.000	1.250	M3
4–10	4.000	4.000	1.250	M3
3–6	5.000	5.000	1.500	M3

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GEAR DESIGN PROGRAM

```
260 ODG=2*CD-RDP-2*CLR:ORG=ODG/2
270 ODP=RDP+2*(HADD+HDED):ORP=ODP/2
280 COSOP=TCD*COS(PA/RA) /CD
290 OPPA=RA*ATN(SQR(1-COSOP^2)/COSOP)
300 INVDP=TAN(OPPA/RA)-(OPPA/RA)
310 INVPA=TAN(PA/RA)-(PA/RA)
320 CTG=CP+2*TCD*INVDP-2*TCD*INVPA-CTP-BL
330 GDED=(CP-CTG-HTF)/2/TAN(PA/RA)
340 RDG=PDG-2*GDED
350 WDG=(ODG-RDG)/2:WDP=(ODP-RDP)/2
360 PX=BDP/ODP:XA=ATN(SQR(1-PX^2)/PX)
370 INVXA=TAN(XA)-XA
380 GX=BDG/ODG:XB=ATN(SQR(1-GX^2)/GX)
390 INVXB=TAN(XB)-XB
400 TFP=ODP*(CTP/PDP-INVXA+INVPA)
410 TGF=ODG*(CTG/PDG-INVXB+INVPA)
420 LA=SQR(CD^2-(BRG+BRP)^2)
430 LAP=LA-SQR(ORG^2-BRG^2)
440 LAG=LA-SQR(ORP^2-BRP^2)
450 WLRP=SQR((LAP+BP)^2+BRP^2)
460 TIFRP=SQR(BRP^2+LAP^2)
470 TIFRG=SQR(BRG^2+LAG^2)
480 CR=(LA-LAP-LAG)/BP
490 Y=PDED-TIPR
500 X=CTP/2+Y*TAN(PA/RA)+TIPR/COS(PA/RA)
510 Q=BRP/WLRP:WLPA=ATN(SQR(1-Q^2)/Q):INVWL=TAN(WLPA)-WLPA
520 B=CTP/PDP-INVWL+INVPA:A=WLPA-B
530 DL=BDP/COS(A)
540 PRINT"     Program in progress."
550 AL=PI/4
560 E=(X+Y/TAN(AL))/PRP
570 BA=AL-E
580 KS=Y/SIN(AL):KE=KS+TIPR
590 TE=PRP*SIN(E)-(KE*COS(BA))
600 NE=PRP*COS(E)-(KE*SIN(BA))
610 HH=DL/2-NE
620 YY=2*HH*TAN(BA)-TE
630 IF ABS(YY)<.0001 Then 670
640 YYY=(2*HH/(COS(BA)^2)-KE*SIN(BA))*(1-KS/PRP/SIN(AL))+KS*SIN(BA)
650 AL=AL-YY/YYY
660 GOTO 560
670 H=HH:TH=TE*2
680 X1=TH^2/4/H
690 Y2=DP/((COS(A)/COS(PA/RA))*((1.5/X1)-TAN(A)/TH))
700 RM=Y^2/(Y+PRP)+TIPR
710 H5=.18-.008*(PA-20)
720 J5=H5-.03
730 L5=.45+.01*(PA-20)
740 K5=H5+(TH/RM)^J5*(TH/H)^L5
750 J=Y2/K5
760 HPB=RPM*OPPDP*F*J*SAT/126000!/DP
770 R1=LA-LAG-BP:R2=LA-R1:R3=SIN(OPPA/RA)*OPPRP:R4=LA-R3
780 CX=R1*R2/R3/R4
790 CG=NG/(NG+NP)
```

GEAR DESIGN PROGRAM

```
800 CC=COS(OPPA/RA)*SIN(OPPA/RA)*CG/2
810 I=CC*CX
820 HPS=RPM*F*I*(OPPDP*SAC/2300)^2/126000!
830 IF LAP<0 THEN CR=0:TIFRP=0
840 LPRINT"           Gear Design Summary"
850 LPRINT
860 LPRINT
870 A$="Center Distance              ###.####"
880 LPRINT USING A$;CD
890 B$="Gear Ratio                ##.####"
900 LPRINT USING B$;NG/NP
910 C$="Face Width                ##.###"
920 LPRINT USING C$;F
930 D$="Diametral Pitch           ###.####"
940 LPRINT USING D$;DP
950 E$="Pressure Angle            ##.####"
960 LPRINT N USING E$;PA
970 F$="Pinion RPM                #####.##"
980 LPRINT USING F$;RPM
990 G$="Tensile Stress No.        ######."
1000 LPRINT USING G$;SAT
1010 H$="Compressive Stress No.  ######."
1020 LPRINT USING H$;SAC
1030 I$="Backlash                #.####"
1040 LPRINT USING I$;BL
1050 J$="Contact Ratio           #.###"
1060 LPRINT USING J$;CR
1070 LPRINT
1080 LPRINT"            Pinion              Gear"
1090 K$="Teeth            ###.#             ###.#"
1100 LPRINT USING K$;NP;NG
1110 L$="Outside Diam.    ###.###           ###.###"
1120 LPRINT USING L$;ODP;ODG
1130 M$="Pitch Diam.      ###.####          ###.####"
1140 LPRINT USING M$;PDP;PDG
1150 N$="Oper. P.D.       ###.####          ###.####"
1160 LPRINT USING N$;OPPDP;OPPDG
1170 0$="Root Diam.       ###.###           ###.###"
1180 LPRINT USING 0$;RDP;RDG
1190 P$="Base Diam.       ###.####          ###.####"
1200 LPRINT USING P$;BDP;BDG
1210 Q$="Cir. Tooth Thick.  #.####            #.####"
1220 LPRINT USING Q$;CTP;CTG
1230 R$="Tip Flat           #.####            #.####"
1240 LPRINT USING R$;TFP;TFG
1250 S$="T.I.F. Diam      ###.####          ###.####"
1260 LPRINT USING S$;TIFRP*2;TIFRG*2
1270 T$="J FACTOR          #.### "
1280 LPRINT USING T$;J
1290 U$="Horsepower Beam  ####.#            ####.#"
1300 LPRINT USING U$;HPB
1310 V$="Horsepower Surf.  ####.#           ####.#"
1320 LPRINT USING V$;HPS
1330 IF CR=O THEN LPRINT" Warning! Possible undercut on pinion. Suggest design change."
```

CLASSIFIED

Rates: Classified Display—$110 per inch (minimum 3") Type will be set to advertiser's layout or *Gear Technology* will set type at no extra charge. **Word Count:** 35 characters per line, 7 lines per inch.

Payment: Full payment must accompany classified ads. Mail copy to *Gear Technology*, P.O. Box 1426, Elk Grove Village, IL 60007. **Agency Commission:** No agency commission on classifieds.

Closing date: Ads must be received by the 25th of the month, two months prior to publication. **Acceptance:** Publisher reserves the right to accept or reject classified advertisements at his discretion.

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DESIGN AND TOOLING

- Custom gear design including non-standard pressure angles for more strength.
- Programs to enter tooling data into computer files and search for existing cutters to cut a new gear or spline.
- Gearing computer software for sale.
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Cedar Falls, Iowa 50613
(319) 266 4674

CIRCLE A-38 ON READER REPLY CARD

GEAR ESTIMATING

The COSTIMATOR® computer aided cost estimating system insures speed and consistency in the difficult task of estimating the costs of all types of gears.

Used by small shops and Fortune 500 companies throughout the country.

For complete information contact us today.

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West Springfield, MA 01089
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CIRCLE A-29 ON READER REPLY CARD

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GEAR TOOTH GRINDING & HONING ONLY

Production Quantities
3/4" P.D. to 27.5" P.D.;
3.5 D.P. and 11" Face

Gear Tooth Finishing is our Only Business

We have no turning, hobbing or shaping capability

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716-684-3811

CIRCLE A-42 ON READER REPLY CARD

There's still time . . .
closing date for a classified ad in the July/August issue is **May 25.**

Coniflex bevels up to 34" diameter.
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BREAKDOWN WORK IS OUR SPECIALTY!!

Complete machining, heat treat, lapping & testing.

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1-800-238-0651

CIRCLE A-34 ON READER REPLY CARD

REBUILDING

HOBBER REBUILDING SPECIALISTS

Having trouble meeting today's demand quality control tolerances?

Let our factory trained and experienced staff return your machine to optimum operating condition.

We specialize in repairing, rebuilding and modernizing all makes of hobbers.

- Cleveland Rigidhobbers
- Gould & Eberhardt
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CIRCLE A-41 ON READER REPLY CARD

BUSINESS OPPORTUNITIES WANTED

WANTED

High quality gear manufacturer operation. Small to medium (25") gears. With or without product line. All inquiries held in strictest confidence. Principles only.

Box GM
c/o Gear Technology
P.O. Box 1426
Elk Grove Village, IL 60007

ROTARY GEAR HONING . . .
(continued from page 37)

of tool and a relatively short honing cycle be used. What is produced in the way of surface finish, then, represents a compromise. First, the honing tool must remove nicks and burrs; then it should make minor tooth corrections that will improve sound level and wear life. The improvement in surface finish, which is in reality a by-product of the honing process, is a valuable adjunct which will help promote long wear life as well as improving sound characteristics.

Honing Gound Gears

In the aerospace industry, gears are traditionally operated at high speeds under heavy loads. They are usually cut, heat treated and ground to provide tooth surfaces (usually of sophisticated modified forms) of the highest order of accuracy. However, tests with exotic surface measuring equipment have shown that ground surfaces have a jagged, wavy profile that will not support heavy loads or wear long unless costly break-in procedures are carried out.

Ground tooth surfaces usually have a surface finish in the 16 to 32 μ range. Honing with type "AA" honing tools can bring this surface finish down to the 8 to 10 μ range (Fig. 2). In one 39-tooth, 5-D.P., 20° P.A., 7.800" P.D. spur helicopter drive gear, honing of the gear teeth down to 8 μ surface finish increased wear life by 1,000% and increased load carrying capacity by 30%. Other tests by the gearing industry have shown 100% load carrying capacity increases by honing ground layers.

Acknowledgement:
Presented at the SME Gear Processing and Manufacturing Clinic, November 11-13, 1986, Schaumburg, Illinois

ENGINEERING CONSTANTS . . .
(continued from page 33)

	Reference diameter											
Adjacent pitch error and difference between adjacent pitches	up to 15.8 in. (up to 400 mm)	DIN	2	2	3	3	4	5	5	6	7	8
		AGMA			15	15	13	12	12	11	10	9
	over 15.8 in. (over 400 mm)	DIN	2	2	2–3	2–3	3–4	4–5	4–5	5–6	6–7	7–8
		AGMA	15	15	14–15	14–15	13	12	12	11	10	9
Total profile error		DIN	2	2	3	3	4	5	5	5	6	6
		AGMA						14–15	14–15	14–15	13–14	13–14
Maximum accumulated pitch error		DIN	3	3	4	5	6	7	5	6	7	8
Radial run-out		DIN	3	4	5	5–6	6	7	5	6	7	8
Total composite error (double flank)		DIN			5	6	7	8	6	7	8	9
		AGMA				12	11–12	10–11	12	11–12	10–11	9
Tooth-to-tooth composite error	up to 15.8 in. (up to 400 mm)	DIN			5	5	6	7	6	7	8	9
	over 15.8 in. (over 400 mm)	DIN			4–5	4–5	5–6	6–7	5–6	6–7	7–8	8–9
Total tooth alignment error	up to 15.8" (400 mm), $P \geqq 4$ ($m \leq 6$)	DIN	1	1	1	2	2	3	3	3	4	4
	up to 15.8" (400 mm), $P < 4$ ($m > 6$)	DIN	2	2	2	3	3	4	4	4	5	5
	over 15.8" (400 mm), $P \geqq 4$ ($m \leq 6$)	DIN		2	2	3	3	4	4	4	5	5
	over 15.8" (400 mm), $P < 4$ ($m > 6$)	DIN		3	3	4	4	5	5	5	6	6

Comparison of DIN and AGMA Qualities

A closer comparison between the various tolerance systems is beyond the scope of a simple table, for the corresponding quality fields never coincide exactly; moreover the various national standards do not make comparable adjustments to gear error tolerances to allow for the influence of the gear diameter and size of pitch: a given class in one standard can, in certain ranges of diameter and pitch, cover several classes of another standard.

A comparison between AGMA run-out tolerances and the equivalent DIN has been omitted, due to the differences in definition. AGMA Quality Classes for lead tolerances have been included in spite of minor differeces in definition.

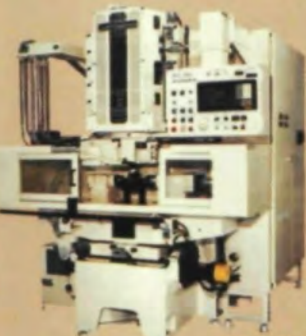
When it comes to gear machining needs, Mitsubishi cuts the leading edge.


GA15CNC


GA25CNC


GA63CNC


SA25CNC


FA30CNC


Gear Cutting Tools

Mitsubishi mechanical and electronic technologies come together today in a full lineup of the most advanced gear cutting machines on the market. Whether your requirements call for hobbing, gear shaping or gear shaving, including flexible gear manufacturing systems, Mitsubishi offers the ideal solution for any need. All machines are unsurpassed in flexibility, speed, accuracy and rigidity — resulting in new standards of productivity. For full information on the entire lineup, along with available tools, contact us today.

SPECIFICATIONS

Model		Max. Workpiece Dia. (in)	Max. Workpiece Pitch (D.P.)	Main Motor (HP)	Machine Weight (Lbs)
Gear Hobbers	GA15CNC	5.9/7.9	6.35	10	8,800
	GA25CNC	9.8	4	10	17,600
	GA40CNC	15.7	4	10	18,700
	GA63CNC	24.8	1.8	20	24,200
Gear Shapers	SA25CNC	9.8	4	7.5	11,000
	SA40CNC	15.7	4	10	15,600
	SA63CNC	25.6	3.2	24.7	21,100
Gear Shaver	FA30CNC	12.2	3.2	7.5	11,500

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CIRCLE A-8 ON READER REPLY CARD


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