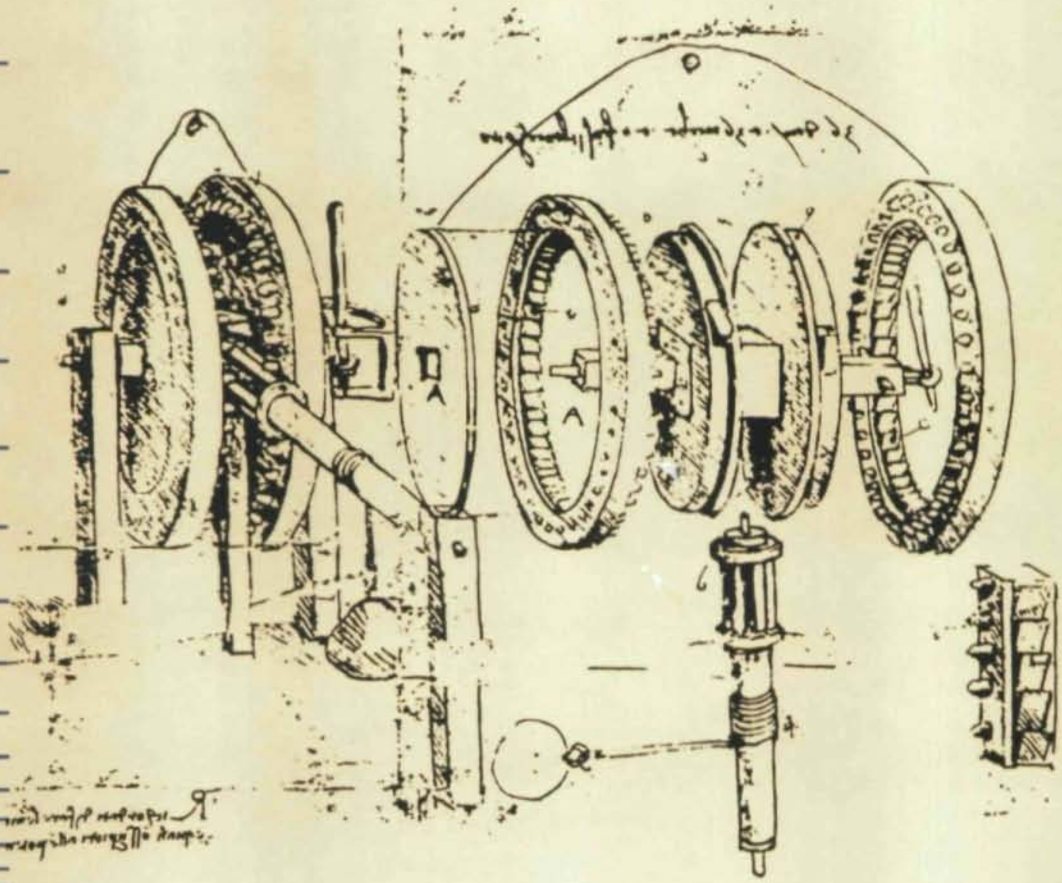
GEAR TECHNOLOGY

The Journal of Gear Manufacturing

MAY/JUNE 1984



Single Flank Testing of Gears

Advantages of Nitride Coated Gear Tools

CNC Controlled CBN Form Grinding

Gear Design

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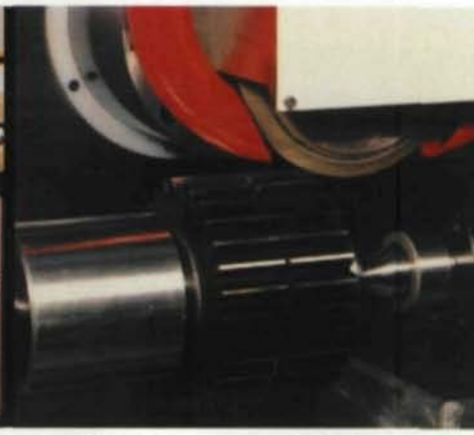
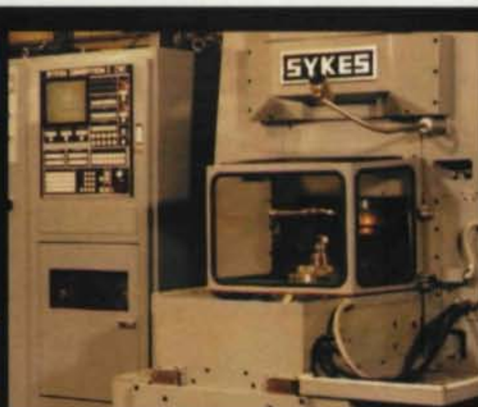
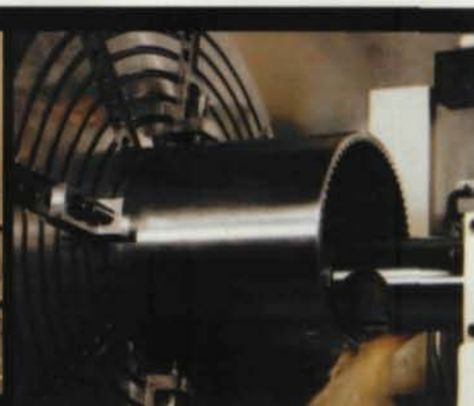


▲ CNC Borazon gear grinding





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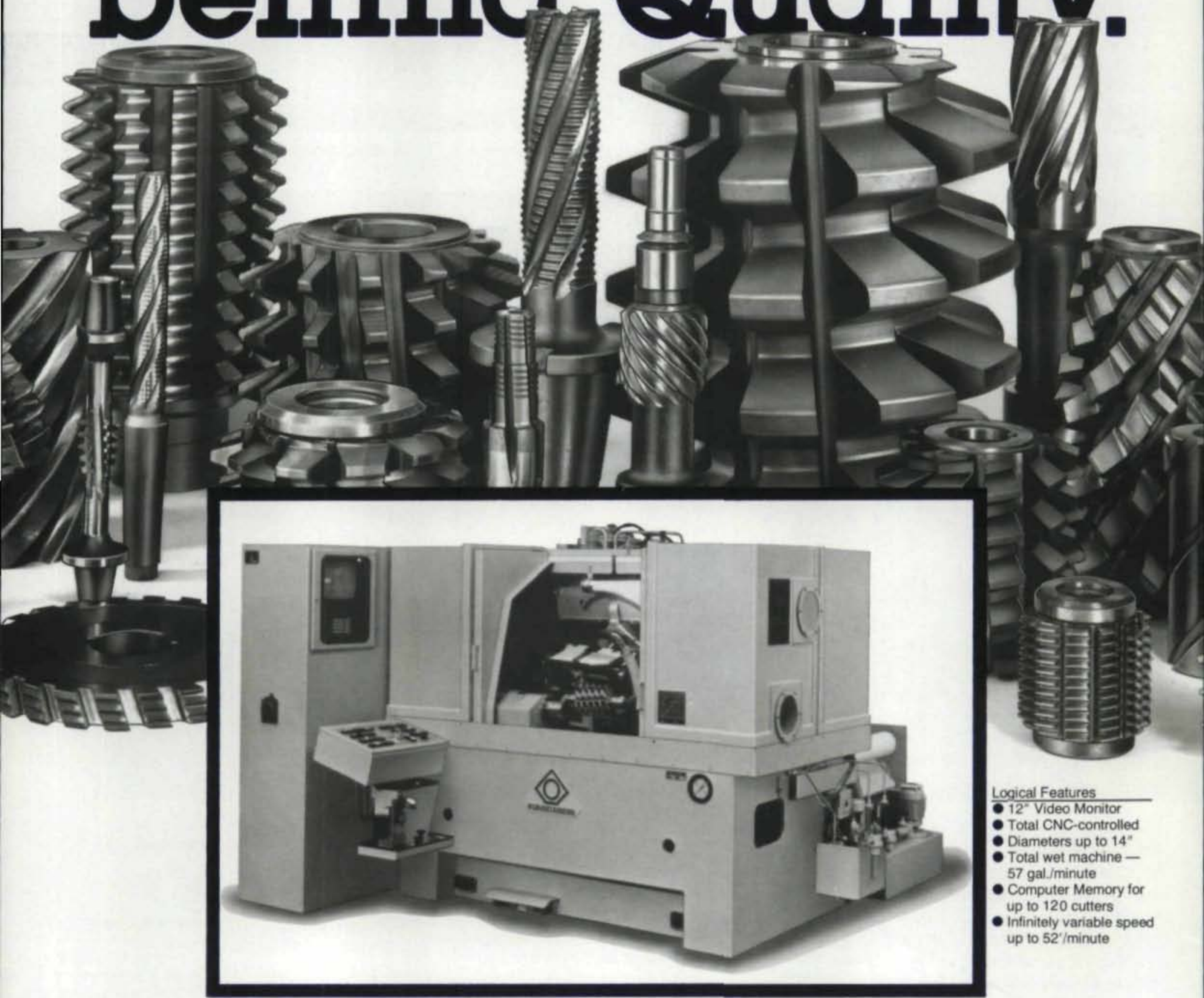

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GEAR TECHNOLOGY

The Journal of Gear Manufacturing

SPECIAL THANKS to **Martha Jordt** for her invaluable service in helping to change GEAR TECHNOLOGY from idea to reality.

MANUSCRIPTS: We are requesting technical papers of every sort from manufacturers of gear making machinery and related equipment, universities, and engineers. Articles should be of an educational and training nature with general appeal to anyone having anything to do with the purchase of materials or machinery, or 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.

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DEPARTMENTS

June/July 1984 — Vol. 1, No. 1

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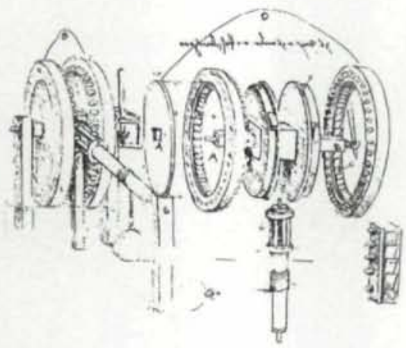

COVER:

The advanced technology of LEONARDO DA VINCI 1452-1519.

During the sixteen years Leonardo was in the service of Duke Ludovico Sforza, he produced numerous sketches on optics, mechanical engineering and anatomy. It appears he was searching for identical mechanisms in nature and man. In all areas of industry and engineering, he displayed his prophetic genius.

Leonardo sketched dozens of devices aimed at translating movement back and forth between rotary motion and piston-like reciprocating action. Windlases, cranes, winches and pulley-blocks are seen through the pages of his notebooks. They each try, through sophisticated reducing gears, to outdo all the others in efficiency. The drawing above is a windlass and is one of the most perfect and precise designs of which working models have been built in modern times.

The cover drawing displays on the left an assembled mechanism for changing the rocking motion of an upright lever to the rotary motion of a shaft, so as to lift a heavy weight. The illustration on the right is an exploded view of the same mechanism. As the operating lever is rocked back and forth, the stone suspended by a rope is wound upwards around the horizontal shaft. The rocking lever swings a square shaft, upon which are two fixed wheels. There are pawls on the outside edges of the wheels, which engage ratchets in the bores of two outer rings. These ratchet wheels also have gear teeth engaging a common lantern gear on the final shaft. When the operating lever is pushed one way, it engages the ratchet wheel. Pushed the other way, and the other pawl engages the ratchet wheel; however, the shaft revolves in the same direction. One application of this principle was found in Da Vinci's own work in an illustration of a paddle-driven boat.

Guest Editorial

At a time when there are many pressures on the Gear Industry and its representative Association, the American Gear Manufacturers Association, it seems particularly appropriate that **Gear Technology — The Journal of Gear Manufacturing** appears. AGMA is particularly pleased to have the opportunity to write the first editorial for this magazine. Over many years, the Gear Industry has had a need for an educational and training journal as well as a resource and reference document on gears, gear products and flexible couplings.

It is further hoped that this magazine will reproduce some of the many papers that are published worldwide on gearing. There is much technological development work that is being carried on throughout the world which should be brought to the attention of everyone in this industry. In addition, there must be a larger, continuing effort on the part of American Manufacturers in the area of gear research. As an advocate and supporter of this and as a forum for the exchange of such knowledge, this journal can be of great value to the Gear Industry. AGMA salutes Michael Goldstein, Editor of **Gear Technology — The Journal of Gear Manufacturing** and wishes him God-Speed in the years to come with this exciting new venture.

William W. Ingraham

William W. Ingraham

Executive Director
AGMA

EDITORIAL

Over the years, we have traveled extensively throughout the industrialized world, and became increasingly aware of the availability of enormous amounts of technical writing concerning research, experiments, and techniques in the gear manufacturing field. New manufacturing methods, materials, and machines were continuously being developed, but the technical information about them was not readily available to those that could best use it. There was no central source for disseminating this knowledge. Thus, the idea for "GEAR TECHNOLOGY, The Journal of Gear Manufacturing" was born.

In this and future issues, we will provide a forum of discovery and innovation for you, the gear manufacturing industry. Every other month, we will be bringing to you technical papers and articles from around the world: solving specific problems, explaining new technology, and describing new techniques. Future issues will contain an index of past articles to provide you with a single reference source. We will be an ongoing gear clinic, ranging from the basics to the lead-edge of technology.

The enthusiastic response to the concept of this magazine from the machine manufacturers, research engineers, and people within the gear manufacturing community has been very gratifying. The advertisers in this, and in future issues, are demonstrating their commitment to excellence. Their belief that providing education and information to you, their customers, is a very necessary ingredient in your success, and ultimately, their success too. These advertisers are offering the technology, productivity and price/performance that you will need to succeed in the world market.

Our goals are lofty and we are dedicated to becoming a part of the changes that are going to be taking place. We will be listening to you, our readers, and will continuously search for the information that you will need to succeed and prosper in the 1980's.

Michael Goldstein
Editor/Publisher

Single Flank Testing of Gears

by
Robert E. Smith
Senior Manufacturing Technology Engineer
Gleason Machine Division

Presumably, everyone who would be interested in this subject is already somewhat familiar with testing of gears by traditional means. Three types of gear inspection are in common use: 1) measurement of gear elements and relationships, 2) tooth contact pattern checks and 3) rolling composite checks. Single Flank testing falls into this last category, as does the more familiar Double Flank test. Figure 1.

As an introduction to the basic understanding of the subject, most of this article relates to the simple case of inspecting spur gears. The interpretation of data, relative to helical gears, is a little more complex, but the general principles apply.

With Single Flank testing, mating gears roll together at their proper center distance with backlash and with only one flank in contact. Testing gears in this manner more closely simulates operation of the gears in their application than any other means of evaluation. Gears can be tested by pairs, or with master gears.

The Single Flank test is run using optical encoders, which measure rotational motion (angular displacement error). Encoders may be attached to the input and output shafts of a special machine for testing pairs of gears. The encoders may also be used portably by attaching them directly to the input and output shafts of an actual gear box so as to inspect the quality of a complete train of gears.

Data from the encoders is processed in an instrument that shows the accuracy or smoothness of rotational motion resulting from the meshing of the gears (transmission errors). This data can be directly related to portions of involute or profile errors, pitch variation, runout and accumulated pitch variation. Probably the most important aspect of Single Flank testing is that it permits measurement of profile conjugacy, which is the parameter that most closely relates to typical gear noise.

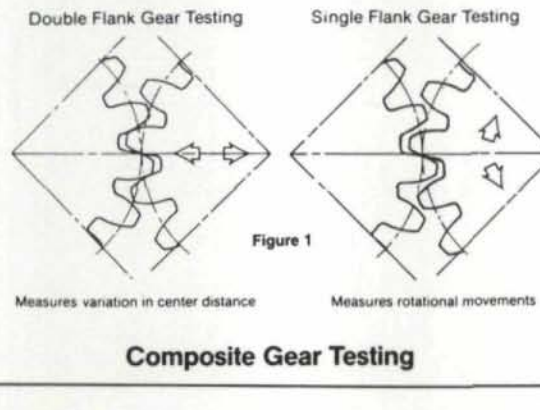

Figure 1

Composite Gear Testing

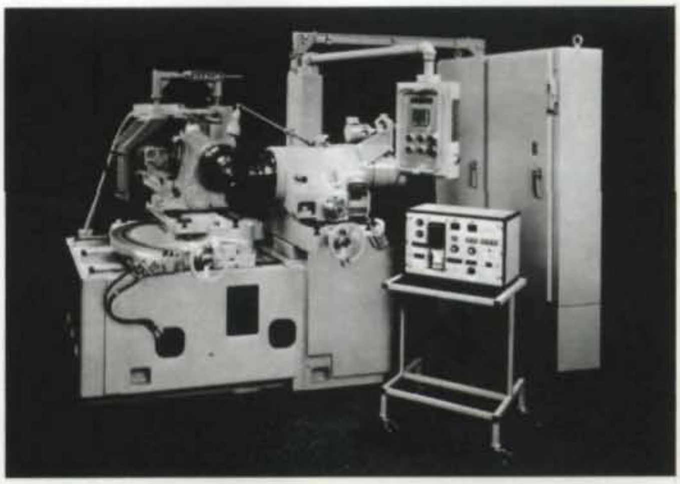

Figure 2

Single Flank testing is not a panacea. Lead or tooth alignment variation of spur and helical gears cannot be measured directly by this method. Lead errors do, however, influence motion transmission errors. These result from profile variations, due to the influence of overlap or increased contact ratio. Likewise, in the case of bevel or hypoid gears, tooth contact pattern checks are important to the development of tooth shape to allow for deflection characteristics under load. Lead or spiral is best measured by elemental checks or by tooth contact pattern checks.

Figure 2 shows a typical Single Flank measuring machine. Figure 3 shows its principle of operation. The two motions which are to be compared are monitored by circular optical gratings. Each grating produces a train of pulses having a frequency which is a measure of the angular movement of each corresponding shaft and hence of each gear mounted thereon.

Pulse frequencies from each grating are usually different because the gear ratio is not normally 1:1. It is, therefore, necessary to modify the frequency from shaft Z1 based upon the frequency from shaft Z2, which is hereby established as the reference frequency. The signal from shaft 2 has a frequency of f2, which is equal to:

$$f2 = f1 \times \frac{Z1}{Z2}$$

where: Z1 = the number of teeth in the gear on shaft 1 and
Z2 = the number of teeth in the gear on shaft 2.

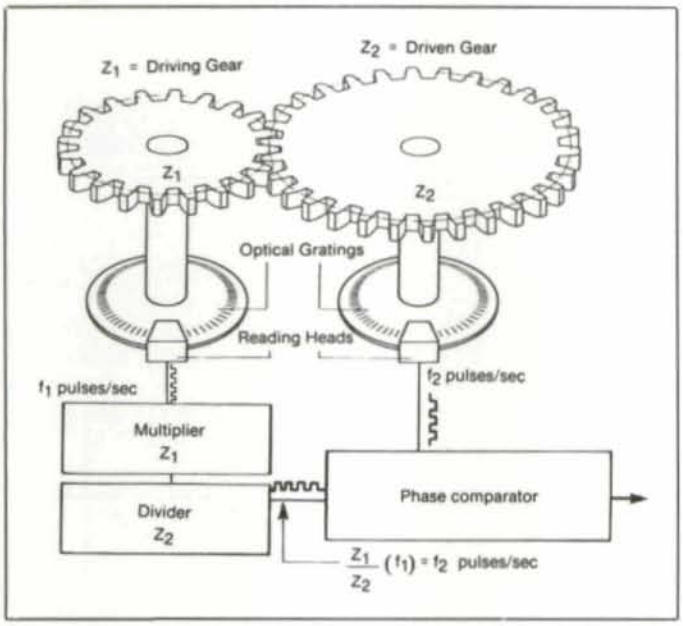

Figure 3

However, f2 has superimposed on it a frequency modulation due to transmission errors of the gears under test. Therefore, the pulse train coming from the grating on shaft 2 will have small differences in phase from the pulse train for shaft f1. This phase difference between the two represents the amount of error in the gears being tested.

Phase differences of less than one arc second can be detected. This difference is recorded as an analog waveform and comes out of the instrument on a strip chart, as shown in Figure 4.

Gears with perfect involute tooth forms will roll together with uniform motion. When pitch errors or involute modifications (intentional or otherwise) exist in a gear, non-uniform motion or transmission errors will result.

In some lightly loaded applications, perfect involutes are desirable for noise control. However, profiles are often modified to obtain a compromise between load carrying capabilities and smoothness of roll or transmitted motion. Such modifications produce predictable, intentional variations on graphic analysis outputs. These variations must be acknowledged when interpreting the graphs. Figure 5 shows three typical tooth shapes and their resulting motion curve. Figure 5a is a perfect involute showing zero angular displacement error. Figure 5b shows typical top and root relief, resulting in the parabola-like motion curve. Figure 5c shows a tooth with pressure angle error and the resulting saw tooth motion curve.

Figure 6 is another way to show a direct relationship between involute shape and a single flank graph. Such curves are a graphic representation of some of the types of non-uniform motion that gears are likely to transmit. It is this non-uniform motion that creates the exciting force that will shake a structure and cause noise.

There are other areas of gear quality that are important besides profile conjugacy and noise. These become more apparent as the graph is run for at least one test gear revolution. All tooth meshes will be added together to generate the results, as shown in Figure 4. The graph in Figure 4 shows additional information; adjacent pitch error, total accumulated pitch error and total transmission error.

The ability to check accumulated pitch error is an important attribute of single flank testing. First of all, there is a difference between "runout" and "accumulated pitch variation". A gear with runout does have accumulated pitch variation. A gear with accumulated pitch variation does not necessarily have runout.

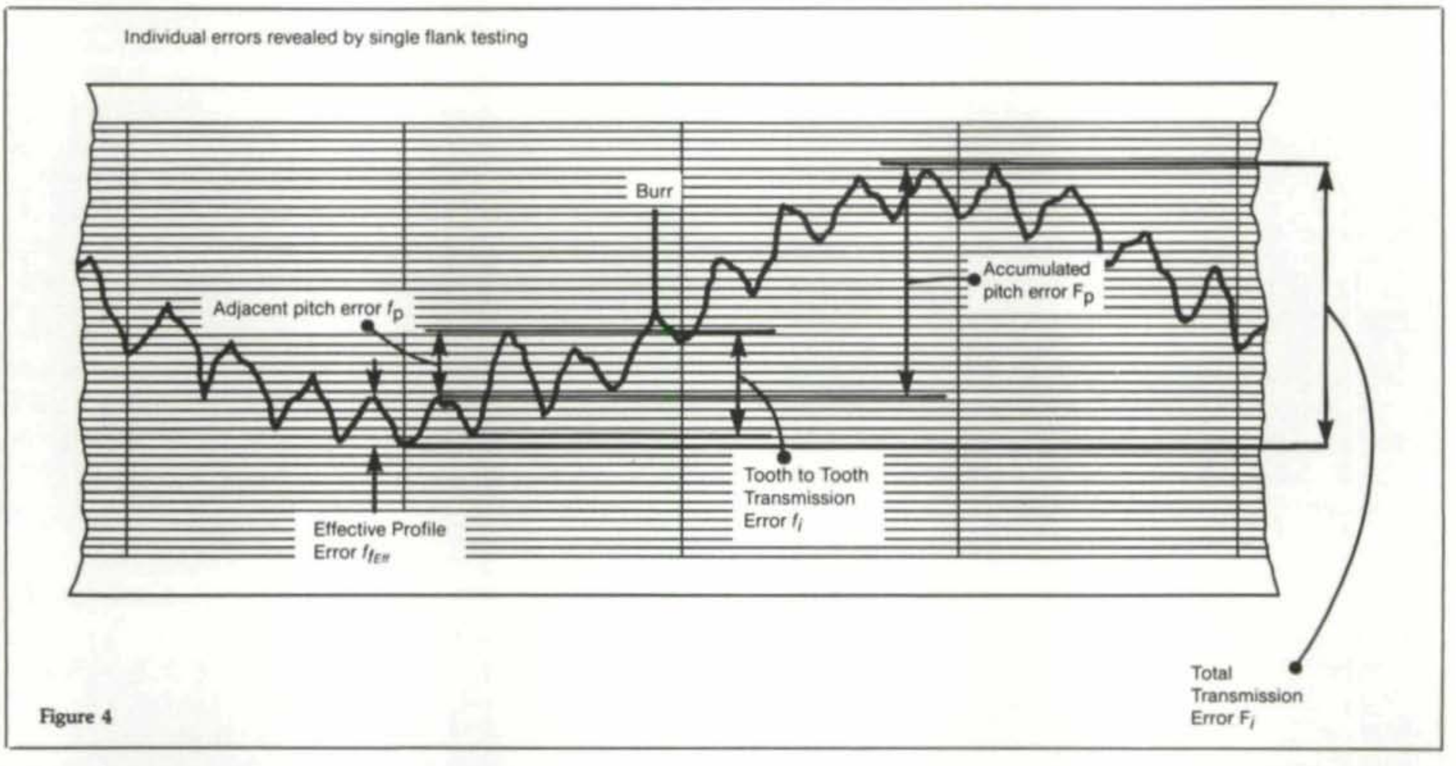

Figure 4

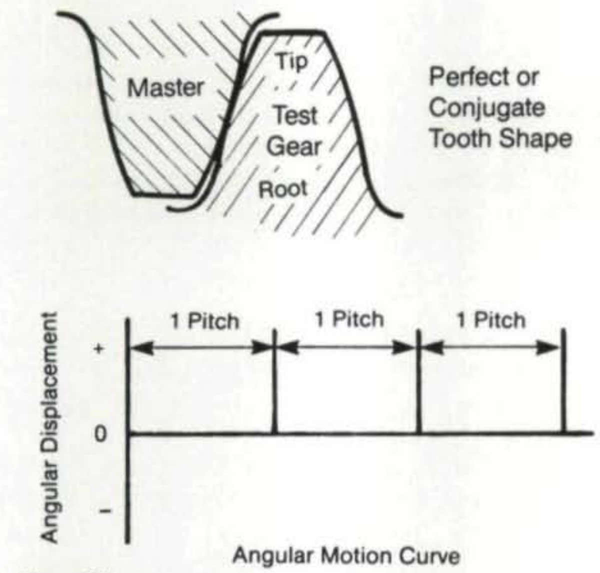

Figure 5A

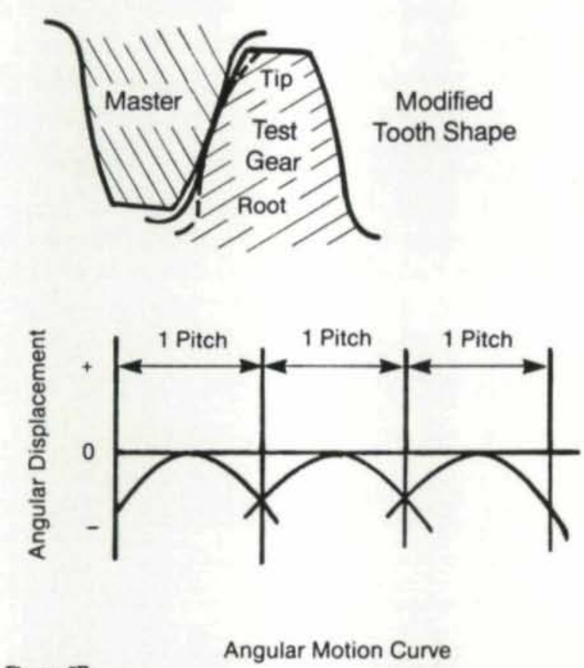

Figure 5B

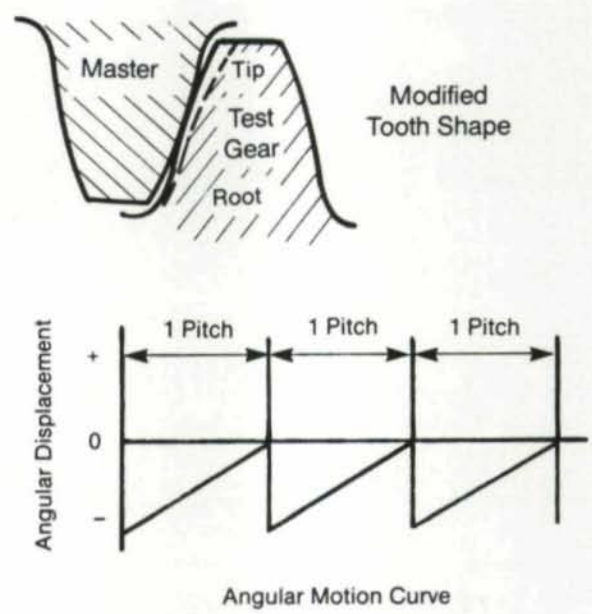

Figure 5C

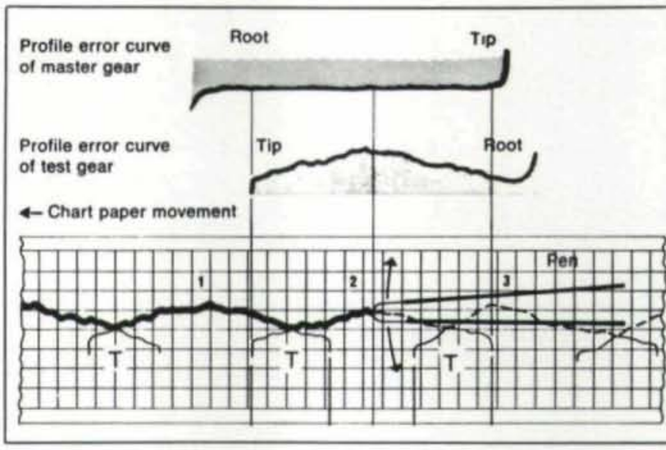

Figure 6

Runout occurs in a gear with a bore or locating surface that is eccentric from the pitch circle of the teeth. Runout is shown as a variation in depth of a ball type probe as it engages each successive tooth slot. Or, it can be a large total composite error if observed on a double flank tester.

A gear can be produced, by various means, that will have no runout, as described above, and will show little or no reading by the ball check. It could, however, have large accumulative pitch errors. This can happen when a gear is hobbed with runout and then shaved or ground on a machine that does not have a rigid drive coupling the tool to the workpiece.

When the gear is hobbed with an eccentric pitch circle, the slots are at different radii and angular positions. When the gear is shaved, it is run with a tool that maintans a constant, rigid center distance, but is not connected to the workpiece by a drive train. Therefore, all slots are now machined to the same radius, from the center of rotation, and are displaced from true angular position by varying small amounts. The resulting gear has very small amounts of individual pitch errors, but has a large accumulated pitch error, which the single flank tester responds to.

These accumulative pitch errors have all the undesirable effects of a gear with traditional runout. It would check "good" by either a ball check or a double flank composite test. Accumulative pitch errors can only be found or properly evaluated by a precision index/single probe spacing checker, or by a single flank composite test.

Figures 7 and 8 are intended to help illustrate the advantages of single flank vs. double flank composite tests.

Figure 9 is an extreme example, whereby the wrong number of teeth are cut in the part. Double flank composite testing will indicate that the part is acceptable, but single flank testing will reject it.

Figure 7

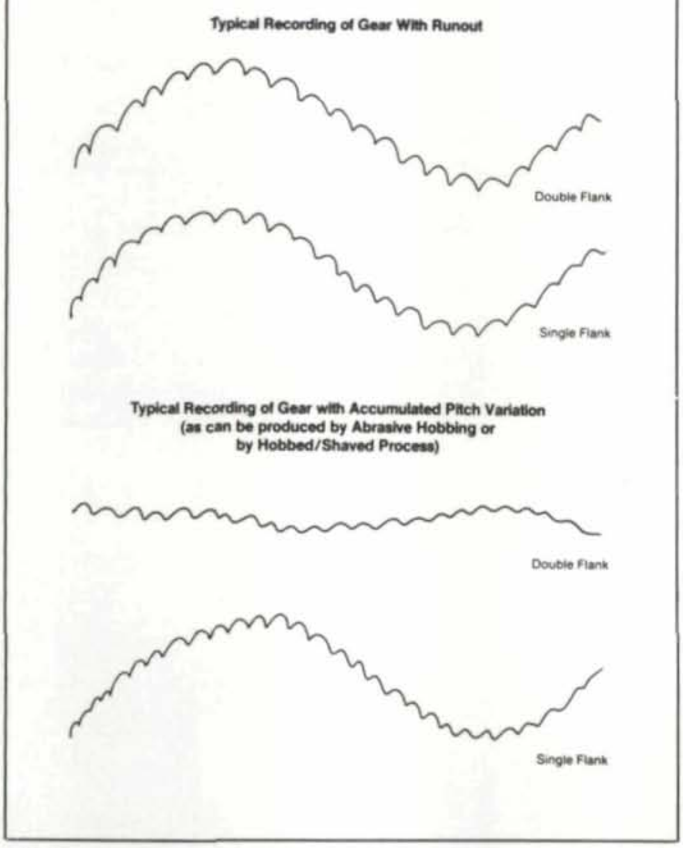

Figure 8

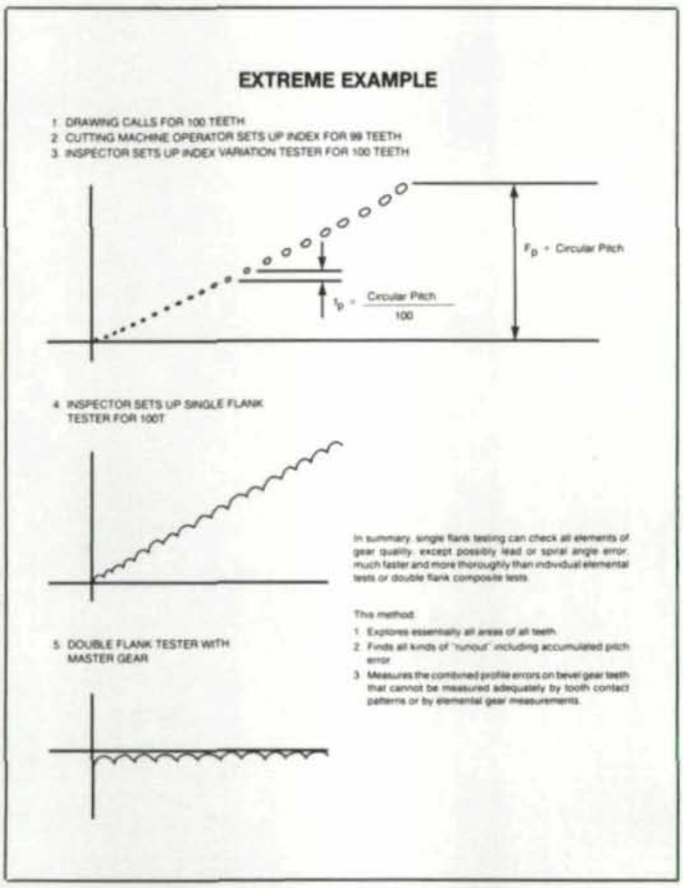

Figure 9

In summary, single flank testing can check all elements of gear quality, except possibly lead or spiral angle error, much faster and more thoroughly than individual elemental tests or double flank composite tests. This method:

1. Explores, essentially, all areas of all teeth.
2. Finds all kinds of "runout" including accumulated pitch error.
3. Measures the combined profile errors on bevel gear teeth that cannot be measured adequately by tooth contact patterns or by elemental gear measurements.

AUTHOR:

ROBERT SMITH, Senior Manufacturing Technology Engineer at Gleason Machine Division, has over thirty years experience in the Gear Industry. Mr. Smith received his training from Rochester Institute of Technology. While at Gleason, Mr. Smith's engineering assignments have included gear methods, manufacturing, research and gear quality. These assignments involved the use and application of instrumentation for the study of noise, vibration, and structural dyamics. From these assignments, he expanded his ideas relating to gear metrology. Currently, Mr. Smith is chairman of the Measuring Methods and Practices and Master Gear Subcommittee in the American Gear Manufacturers Association, and is also a member of the Rochester Industrial Engineering Society and Society of Experimental Stress Analysis.

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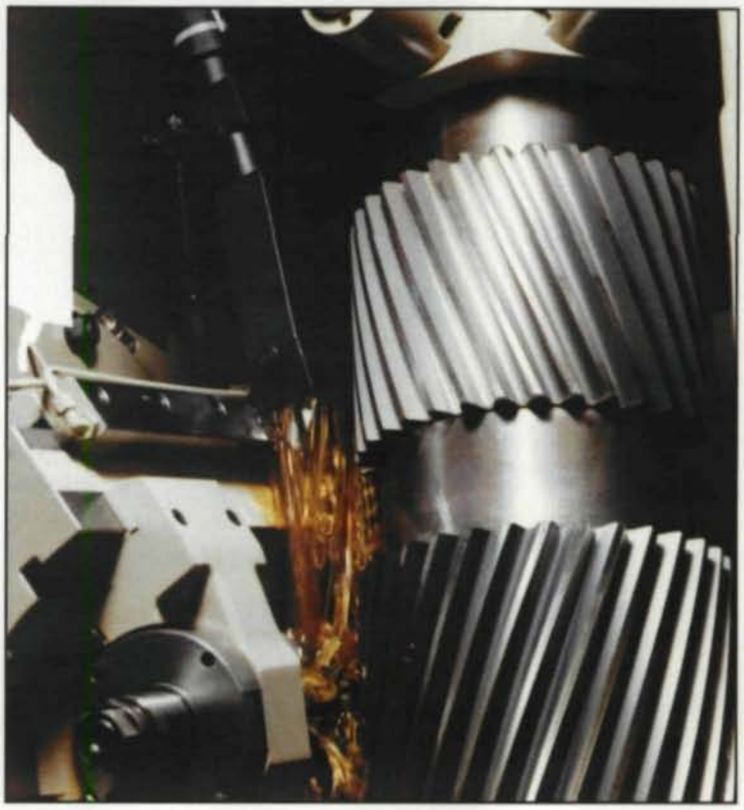


Advantages of Titanium Nitride Coated Gear Tools

Peter W. Kelly
Barber-Colman Co.

Abstract

A brief introduction to the subject of Thin Film Coatings and their application to gear hobs and shaper cutters is followed by a detailed description of the Chemical Vapor Deposition Process and the Physical Vapor Deposition Process. Advantages and disadvantages of each of these processes is discussed.

Emphasis is placed upon application engineering of coated gear tools based on laboratory and field test results.

Recommendations are suggested for tool design improvements and optimization of gear cutting operations using coated tools. Productivity improvements potentially available by properly utilizing coated tools are considered in terms of both tool cost and machining cost.

Introduction

Gear cutting tools are among the most complex and costly tools used in the metal working industry. This is especially true of the more accurate tools which normally require form grinding of the critical gear tooth form generating surfaces. Because the tools are expensive, as well as the gear cutting operation itself, means have long been sought to improve tool life and to increase the productivity of the gear cutting process. Many methods have been applied in the past to obtain these ends including, special designs of the gear tools, use of higher alloy high speed steels, and various coating or treatment methods which were applied to the tool surfaces in an attempt to increase their wear life. These methods have included various nitriding processes, chrome or nickel or other plating processes, as well as others, which may even have fallen within the realm of black art. There have been some processes which have proven viable for certain unique or special applications. In these cases, significant improvements have been obtained and the methods have been economically justified. However, most proposals which appeared to yield significant life or productivity improvements in a laboratory or closely controlled production environment have not, in fact, proven to be practical manufacturing alternatives for use on gear tools in the long run.

Titanium nitride (TiN) coatings have been successfully applied to carbide inserts since 1969. The chemical vapor deposition (CVD) process, which is a high temperature process, has been applied to carbide inserts used in turning, boring, and milling applications and proven highly successful. The high temperature required by the CVD process does not distort, nor does it change the carbide substrate, therefore, the process had proven applicable to carbide inserts. This high temperature method had not been applied until recently to high speed steel gear cutting tools, because their complex geometry was distorted by the high temperature involved in the process.

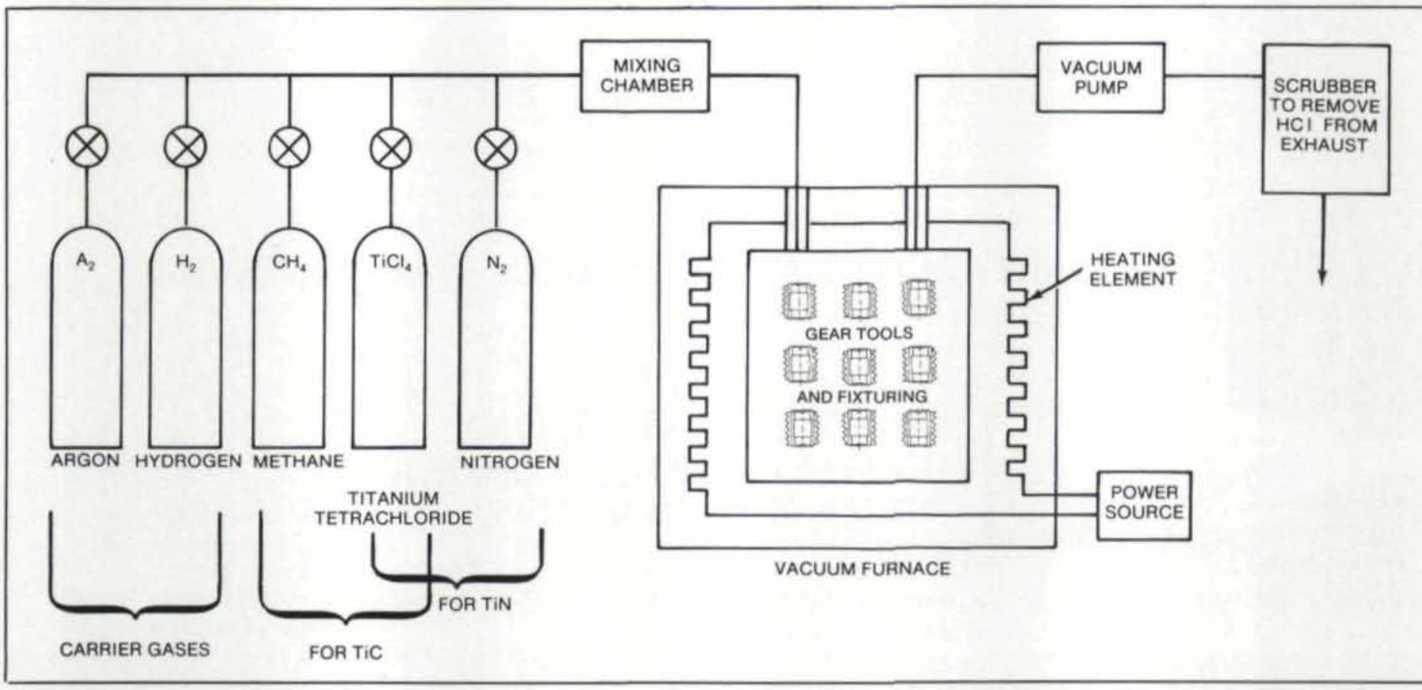


FIG. 1 — Schematic Diagram of the CVD process.

In spite of the above, experimentation and development of titanium carbide (TiC) and titanium nitride (TiN) coatings of gear hobs and gear shaper cutters was being carried on in this country using the CVD process in late 1979 and early 1980. The physical vapor deposition (PVD) process, which is conducted at a low enough temperature such that no annealing or thermal distortion of hardened high speed steel occurs, was successfully applied to gear cutting tools early in 1980 in Japan. The very rapid growth and acceptance of the PVD process are illustrated by the following. There was only one coated gear hob displayed at the International Machine Tool Builders Show in September of 1980 in Chicago; but within the next year, there was a multitude of high speed steel cutting tool applications including many gear tools at the EMO Show held in Hanover, Germany in September of 1981.

The tool coatings mentioned above, titanium carbide and titanium nitride, produce very hard, thin, and wear resistant layers. These characteristics combine to make those coatings ideally suited to the critical and complex geometries involved in gear cutting tools. Since titanium carbide is the somewhat harder of the two, it is recommended when the type of service encountered is highly abrasive. On the other hand, titanium nitride, with its low coefficient of friction and anti-weld characteristics, is applicable in cases in which gauling or welding are encountered.

Chemical Vapor Deposition Process

Chemical vapor deposition (1) is carried out in a vacuum furnace at the relatively higher temperatures of 1750 to 1930°F. In the process, illustrated schematically in Fig. 1, the vapors of titanium and nitrogen (TiN) or the vapors of titanium and carbon (TiC) are chemically combined and deposited on the surface of the cutting tool.

The high temperature involved require that these coatings be applied to a tool in the soft condition or that subsequent rehardening would be necessary due to the fact that such high temperatures would anneal a hardened tool. In either case, the necessity to vacuum harden the coated tool, after the final machining operations on the cutting teeth, results in distortion, which is excessive for all but the lower quality classes of gear cutting tools.

CVD coatings range from 3000 to 3500 Knoop hardness for titanium carbide and 2500 to 3000 Knoop hardness for titanium nitride. These numbers compare to maximum extrapolated Rockwell C hardnesses of 90 and 85 respectively. When compared to hardened high speed steel ranging from 64 to 68 Rockwell C, it can be seen that very significant improvements in hardness are achievable. TiN and TiC CVD coatings are also relatively thick at .0003 inches. While this can be a detriment to the accuracy of high precision gear tools, it does provide the potential of substantial life improvment.

The CVD process also has the potential of producing more economical coatings. One of the reasons this is true is that the equipment used is less complex, in that lower vacuums are employed. This means that not only is the equipment itself more economical, but the energy costs involved are also lower. Additionally, since small amounts of relatively inexpensive materials are utilized, the overall material cost of the process is also modest. With the high temperature involved in the process, which inherently produces improved adherence of the coating, work preparation and cleanliness are not as critical as in other methods.

The major applications for CVD are to provide wear resistant coatings for carbide inserts and lower precision classes of high speed steel tools. Tool life improvement of as much as six times on higher speed steel cuttings tools has resulted in a high degree of customer satisfaction.

Physical Vapor Deposition Process

The highest precision grades of high speed steel tools have geometric tolerances which are less than the distortions which occur during even the most closely controlled hardening operations. Therefore, the critical dimensions of such tools must be finished subsequent to hardening, and no additional high temperature operations are permitted. In order to coat precision high speed steel tools successfully, the low temperature range of 530 to 890°F of the PVD process is necessary to prevent annealing of the high speed steel substrate. Because of the lower temperature levels, there is no appreciable distortion of the tools. Fig. 2 is a schematic diagram of the PVD Reactive Ion Plating process employed by Barber-Colman Company to produce TiN Gear Tool coatings.

PVD is an entirely different process in which ions of titanium react with ions of nitrogen or carbon at the substrate surface resulting in a compound (TiN or TiC) which is physically deposited on the tool surfaces. These coatings range in maximum hardness from 2000 to 2800 Knoop for TiN and TiC respectively which correspond to approximately 80 and 85 Rockwell C. Comparing these values to the normal range of hardened high speed steel, 64 to 68 Rockwell C, it is seen that a very significant improvement in tool life is possible. The coating is extremely thin, only .000120 inch which makes it ideally suited for use o the most precise grades of high speed steel gear tools.

The coating process is carried out in a high vacuum furnace, with pressures ranging from 10^{5} to 10^{6} torr. This very

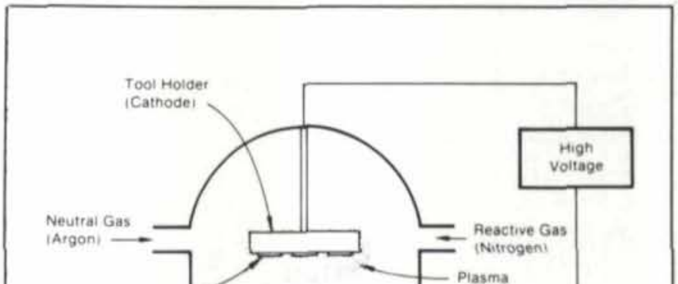


FIG. 2 — Schematic diagram of the PVD reactive ion plating process in which the tool is exposed to ion bombardment cleaning both before and during deposition.

high vacuum requires expensive equipment and substantial energy costs. The material costs, however, are quite nominal.

There are various physical vapor deposition processes in use.

The three main methods are:

Evaporation — Not applicable to high speed steel precision tools due to low film adhesion and problems in obtaining uniform coverage of complex geometries. An additional difficulty associated with TiN coating is the necessity to melt the compound titanium nitride which has a very high melting point of 5828°F.

Sputtering — This process is utilized by a number of domestic, as well as foreign companies. However, the process has low coating rates, uses much energy, and is somewhat difficult to control and quite susceptible to contamination.

A refinement of this method is called Reactive Sputtering in which the atoms of a gas introduced into the deposition chamber react with the pure metal atoms to form a metallic compound on the work.

Ion Plating — The basic methods applied are essentially off-shoots of work done by D. M. Mattox [2, 3] dating back to 1963. Ion Plating has an inherently high degree of controlability and is the basic process used by a number of foreign equipment builders.

The reactive gas refinement noted above under sputtering can similarly be applied providing Reactive Ion Plating to produce metallic compound coatings.

Although current PVD processes employ the latest techniques of plasma physics and thin film technology, the basic methods are not new. Applications in which PVD has been used for many years include depositing of refraction index improvement coatings on optical equipment, such as the lenses in eye glasses, telescopes, binoculars, microscopes and cameras. Decorative coatings have been applied to various metals and plastics for a wide range of industrial and consumer products. Watch cases and other items of jewelry have been enhanced with the beauty and durability of TiN. The electronics industry has utilized the PVD thin film technology for application of very critically controlled coatings to semi-conductors, vacuum tubes, cathode ray tubes and other components.

The characteristics of TiN which have the potential of providing wear and abrasion resistance are its high hardness, low coefficient of friction, and inertness which results in an anti-weld surface.

These characteristics also allow use of cutting tools at higher speeds and feeds. The high quality coatings, producing up to six times life improvement at no distortion or dimensional changes in the high speed steel, make the PVD Reactive Ion Plating process ideally suited to gear cutting tools.

Table 1. Equivalent Hardness Of Various Materials

Material	Knoop Hardness 1000g Load	Extrapolated Hardness Rc
Diamond	7000	--
Borazon	4500	--
Titanium Carbide	3500	90
Titanium Nitride	3000	85
Tungsten Carbide	1870	80
High Speed Steel	870	66

Application Engineering Of Coated Gear Tools

Advantages of TiN for HSS Tools

Titanium nitride possess a number of physical characteristics which are particularly effective in enhancing the life of high speed steel tools. Probably the most important of these characteristics is its extremely **high hardness**. It can be seen in Table 1 that the extrapolated equivalent hardness of titanium nitride at Rockwell C 85 is much higher than that of the high speed steel upon which it will be deposited. It should also be noted that the hardness of titanium nitride is considerably higher than that of tungsten carbide and is even approaching the hardness of Borazon (Cubic Boron Nitride).

The high hardness of titanium nitride provides excellent wear and abrasion resistance to heavy and prolonged cutting loads.

Titanium nitride rubbing against soft steel has a considerably **lower coefficient of friction** than hardened high speed steel would have. This reduces the tool loads and, therefore, the heat generated during cutting.

Actual tests have proven conclusively that due to the lower frictional characteristics of the TiN coatings that machine loads and the resultant power required have been reduced. Documented records are available indicating reductions of 25% to 30% in power requirements. There have been numerous instances in which the amount of oil coolant smoking due to high temperature in gear cutting operations has been reduced significantly by the use of coated tools.

The resultant lower temperature contributes to extended tool life and also improves the accuracy of the gear generating process. The major benefit of the lower heat generated as a result of reduced friction would be to allow the use of higher speeds and possibly feeds which of course would reduce the machining time and cost of the gear cutting operation.

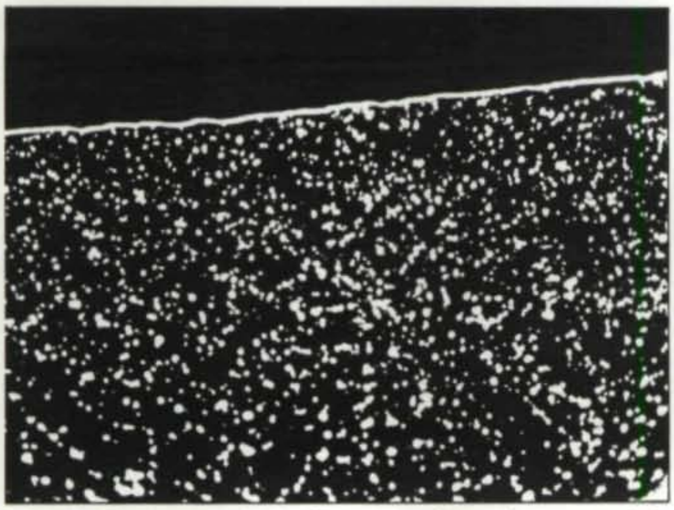


FIG. 3 — Photomicrograph of ASP23, 3-thread ground automotive hob: 400X, 5% nital etch with .000120 inch thick TiN Coating.

One of the reasons for wide spread use of titanium nitride as a protective coating on jewelry is that it is **extremely inert.** Because of this characteristic, coatings will produce no chemical reaction with the normally encountered gear materials these tools will cut, nor will they suffer any adverse effects in conjunction with gear cutting fluids.

Fig. 3 is a photomicrograph of a TiN coated powdered metal high speed steel gear hob tooth. Because of the very **thin** titanium nitride layer, .000120 inch, there is no appreciable dimensional change to the tool on which it was applied.

In addition, the maximum temperatures encountered in the PVD reactive Ion Plating Process are low enough to prevent annealing of hardended high speed steels. Since there is no significant dimensional build-up or thermal distortion the process is applicable to the highest precision grades of gear cutting tools.

Application Data

Because a wide range of accurate application engineering data for TiN coated gear cutting tools is not yet available, it is necessary to carefully consider each case individually. However, test results from a reasonable number of applications are available to allow realistic estimates and projections of potential advantages. The following information can be used, recognizing that it is based on a limited amount of overall experience.

Tool Wear & Performance Improvements

Fig. 4 illustrates the various types of wear often found on gear tools. Both chipping and gouging are types of wear which are due to deficiencies in the gear cutting operation which should be corrected. These types of wear produce damage deeply into the teeth and therefore will not be improved by the very thin layer of the Titanium Nitride coatings.

For the purposes of this discussion, each of the three (3) types of wear which occur on the relieved surfaces of the tooth directly in back of the cutting edge will be grouped together and referred to as "Flank Wear". These are periphery wear, corner wear, and lower flank wear, each occurring at a tooth location defined by its name. Flank wear is the type which is normally predominant in properly applied gear cutting tools. Since it occurs on the relieved tooth surfaces, those surfaces which are not removed by subsequent sharpening of the tool, TiN coating will be most effective in reducing flank wear and thus increasing tool life. In addition, the low frictional and anti-weld characteristics of tianium nitride can be utilized to allow higher speeds and feeds thus providing increased productivity.

Cratering is the eroding of material away from the cutting face of a hob or shaper cutter. The erosion of that surface is a result of the abrasive action of the chip curling over the face of the tooth, or a chemical reaction of the chip and coolant combination or welding in which the high pressures and heat encountered in the cutting action tend to weld the chip to the cutting face. As it is broken away a small particle of the face is removed with the chip. Since this cratering occurs on the cutting face of the tooth, only the first use of the tool before sharpening would be improved by the TiN coating. When cratering is a problem, it would generally be advisable to consider appropriate coolant changes or additives, or possibly the use of a higher alloy tool steel which would be resistant to the abrasive action which caused the cratering.

A series of in-house tests was conducted to determine life improvement ratios available through TiN coating of high speed steel gear hobs and shaper cutters. Since it is always necessary to at least maintain or hopefully improve existing standards of accuracy when applying new types of tools, the amount of acceptable wear before sharpening a preshave protuberance type hob was determined by the number of pinions cut before the amount of undercut produced or the true involute form (TIF) diameter violated established tolerances.

The results of these tests, conducted at fixed feed and speed rates, are shown in Fig. 5 which indicates that the coated flank wear was reduced to about 1/3 of the uncoated wear when the effective cutting edge dullness was held constant for both the uncoated and the coated tools. However,

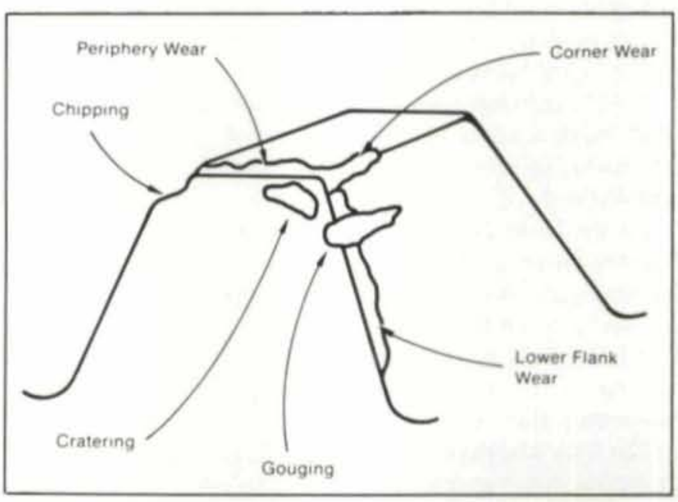


FIG. 4 — Six types of wear encountered on gear tool teeth.

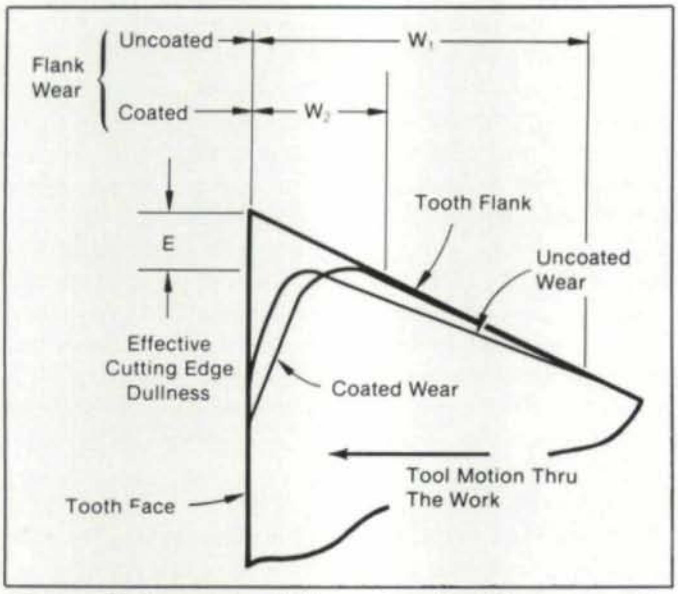


FIG. 5 — Flank wear when holding cutting edge dullness constant for coated vs. uncoated tools.

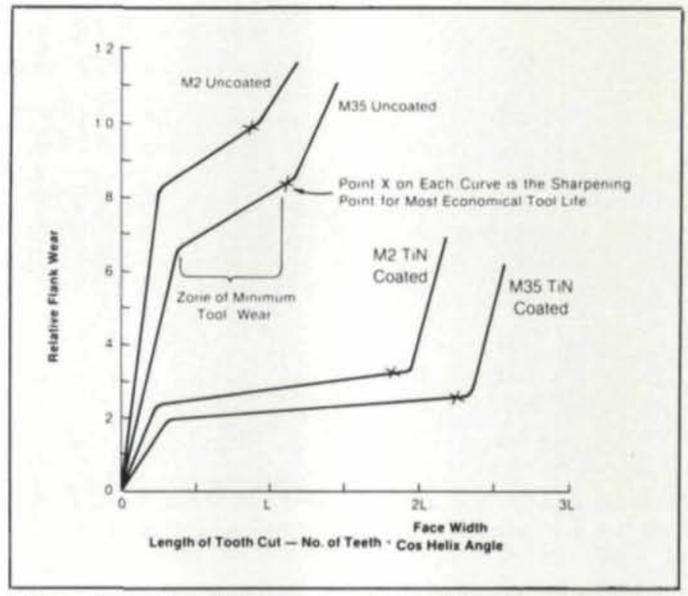


FIG. 6 — Potential tool life improvement with TiN coating under identical cutting conditions.

at the time at which the amount of undercut and the TIF diameter exceeded established tolerances, the number of pinions cut with the coated tool was twice the number cut with the uncoated tool. The increase in tool life was three (3) times in terms of flank wear and two (2) times in the number of parts cut. Therefore, the overall tool life was increased by a factor of six (6) times. Subsequent tests, both in-house and at various customer manufacturing operations, have shown that improvements up to 10 to even 15 times are possible.

Fig. 6 is a graph of the Relative Flank Wear vs. the Length of Tooth Cut for uncoated and coated tools of M-2 and M-35 tool steel. The curves show that, as normally expected in high speed steel tools, there is a high initial rate of wear after which a zone of minimum tool wear occurs. As the tool continues to dull during the zone of minimum wear, a point will eventually be reached at which time the cutting edge breaks down rapidly due to increased head and load. Subsequent failure of the tool is very rapid and catastrophic if the cutting action is not stopped at this time. In order to optimize the economy of the cutting tool application it would be necessary to sharpen the tool at the point marked "X" on each curve.

This point is at the end of the low wear rate portion of each curve just prior to the point at which rapid breakdown of the cutting edge is imminent. Currently these points must be established by test procedures in specific instances. However, after TiN coatings have been used for a sufficient length of time to accumulate accurate application data it will be possible to reasonably predict the economical life for low production runs.

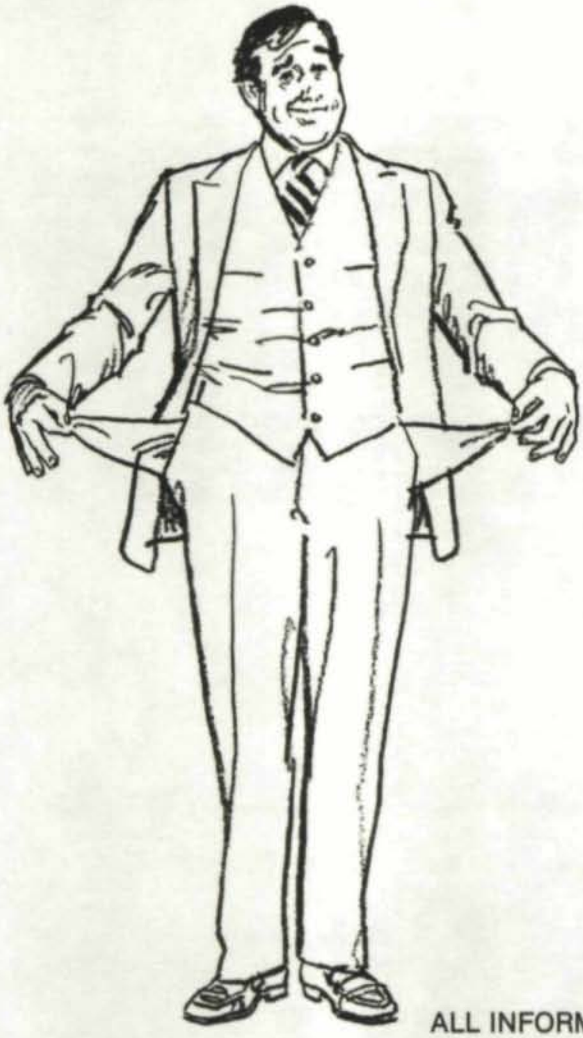
It should be pointed out that because the amount of wear on coated tools will be much less that that encountered in previous practice, it will be important to assure that sharpening practices are revised. Obviously the amount of stock removed in sharpening must be reduced to approximately 1/3 that of prior practice. Also the amount of stock removed in sharpening in excess of the actual wear must be held to a minimum. For example, if prior practice was such that tools were sharpened after .025 of wear and .030 stock was removed, the amount of tool life removed in excess of actual wear would be 17% (5/30) of the total sharpening stock. However, if the wear encountered with a coated tool was only .008 and an additional .005 was removed in sharpening, the amount of the tool life removed, in excess of actual wear, would be approaching 40% (5/13) of the total sharpening stock or over two (2) times that amount in previous practice.

Another extremely important point to bear in mind is, that even though cratering had not been a problem when using an uncoated tool, if the same tool is TiN coated and more pieces per sharpening are being produced, then cratering is very likely to become a problem.

This is because cratering is a function of the amount of chips which have been curled over the cutting face and it is that face which does not have the benefit of TiN protection. A critical characteristic of cratering is that a point is eventually reached when the workpiece chip loads in the eroded craters cause sections of the tool cutting edges to break away. These broken sections usually exceed the amount of normal flank wear by a considerable amount. In fact, in severe cases when cratering is excessive, the whole top of a tooth can be broken away. Fig. 7 is an example of a hob which had been used to cut an excessively long linear length of work tooth resulting in catastrophic failure. The cratering shown in the close-up view of the tooth tips was the cause of the breaking out of entire hob teeth as seen in the overall view. Fig. 8 is a similar example of a severely cratered shaper cutter tooth. Good practice would forbid use of a tool to this extent due to the high risk of tooth breakage.

Optimized Productivity

In order to assure that the maximum potential of TiN coatings is obtained, it is necessary to consider both the tool itself and the machine in which it is being used. In most gear



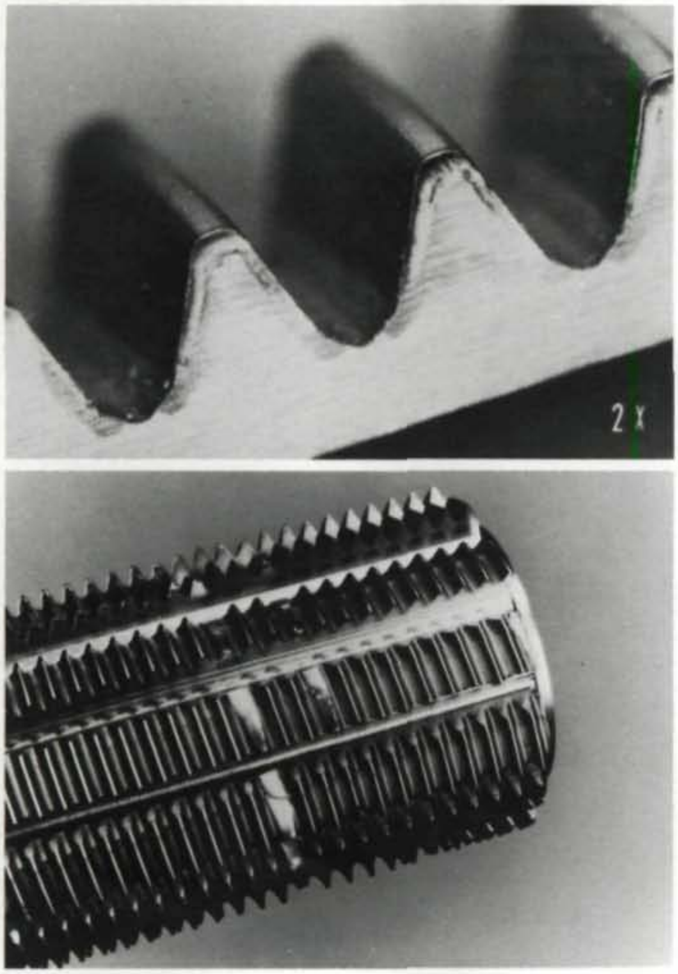


FIG. 7 — 15 DP hob illustrating castastrophic tooth failure due to excessive cratering caused by hobbing too many linear inches.

cutting production operations, the tool cost per piece cut is usually in the range of 5% to 25% of the total cost of the gear cutting operation. This means, that even if the entire tool cost could be eliminated, only 5% to 25% of the gear cutting cost could be saved. However, to save that much in the cost of machining time would require a change of speed or feed, or feed and speed in combination, of only about 30% at the most.

Actual tests of TiN coated drills and end mills have shown amazing results, In the case of drills an increase of feed actually increased the tool life, and in the case of end mills, an increase of speed significantly increased the tool life. At this time, most gear tool testing at increased speeds or feeds indicates that a range of 30% to 50% increase of speed and approximately 10% to 20% increase of feed is possible at no adverse loss of tool life.

Therefore, in order to obtain the optimum potential of TiN coating, it is important to increase the speed, and possibly the feed, to the maximum amount, depending upon a number of factors which would have to be evaluated for each specific application. Among these factors are consideration of the work piece material, its hardness and its machinability. It is also necessary to consider the material used in the cutting tool, the rigidity of the machine and the work holding fixtures, as well as, the condition of the machine itself in terms of wear, tightness adjustments, etc. Obviously, the machine must have sufficient power to operate at the increased metal removal rates, and must have sufficient speed range or feed range available. The rigidity of the structure of the workpiece itself must also be adequate to withstand the possible increased loads due to higher feeds or speeds.

The coolant also becomes more critical at elevated speeds. It may be necessary to change coolants to accommodate the more adverse conditions due to increased speed and load. Limited in-house and field tests have shown substantial tool life improvement when using TiN coated tools with borate additives, in some cases, and with chlorinated cutting oils, in others.

Because of the improved cutting efficiency it may be possible to increase the cutting depth of multiple cut operations. This could reduce the number of cuts or passes and yield significant productivity improvements.

In the overall, it would be expected that more pieces per sharpening and reduced tool wear will both be available thru the use of coated tools. It may be required that heavy duty machinery be applied to obtain the maximum benefits.

A side effect of TiN coated tools will be the improved accuracy, which results because of the lower cutting loads and the fact that sharper tools are used during a higher percentage of the overall production run. This will result in less scrap, reduced lead and involute errors, and improved surface finish.

Production Test Results

To illustrate many of the foregoing remarks and recommendations, consider the actual production test results

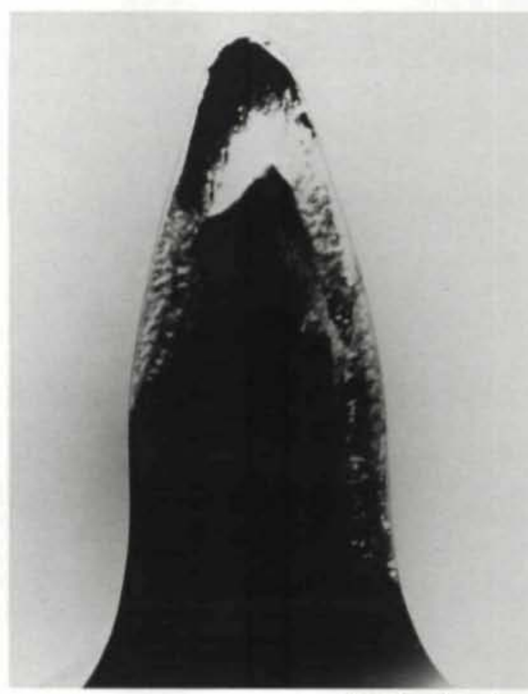


FIG. 8 — Excessive cratering on DP shaper before sharpening cutter tooth due to shaping too many parts.

shown in Tables 2 through 5 and summarized in Table 6. These tests were conducted on the production floor selected, high volume gear or pinion manufacturers.

The data at the top of Tables 2 through 5 includes the specifications of the tool, the workpart being cut, and the machine in which the gear cutting operation took place. The remaining information is a tabulation of the operating data, the gear cutting results and the gear cutting cost analysis for the conditions of: 1) The uncoated tool operating under previously established production conditions, 2) The TiN coated tool operating under the same conditions as had been established for the previously applied uncoated tool, and 3) Assumed increased speed or depth of cut, the resulting wear, and the number of pieces cut when applying the TiN coated tool in order to obtain increased productivity.

To gain the maximum potential available from TiN coated gear tools, it is necessary to increase the speed, and/or increase the feedrate, and/or reduce the number of cuts in a multiple cut operation. For the purpose of illustrating the large savings available by reducing the machining time, it was assumed that the tools tabulated in the tables could be operated at increased speed, and in one case at a reduced number of cuts. It was further assumed that they would produce only half the number of pieces when worn the same amount as the TiN coated tools at the originally established lower speed. These are conservative estimates based on results of other gear tool tests and on experimental work done with other types of high speed steel cutting tools.

Note that the costs tabulated in Tables 2 through 6 include only the cost of the gear cutting tool and the cost of the actual hobbing or shaping operation itself. The machining costs are based on the calculated cutting time at a conservative estimate of $25.00 per hour. In the simplified analysis presented, the tool sharpening cost and the cost of the gear cutting machine downtime for tool changing have not been included. When these costs are considered, the economic advantages of applying TiN coated tools are increased substantially because the amount of sharpening stock and the frequency of sharpening are greatly reduced.

The detailed information contained in the tables can be examined to the depth desired by the reader. A few comments will be provided here relative to significant improvements in the overall operation due to application of TiN coated tools.

Table 2 lists the condition of hobbing a 16 tooth automotive transmission pinion with a single thread, unground Class C, TiN coated hob. Comparing the results of the uncoated and the TiN coated hob under the same operating conditions, and the assumed results for the TiN coated hob at a 30% increase in hob surface speed, it can be seen that, under the same operating conditions, the coated hob produced the same number of pieces with only approximately half the hob wear of the uncoated tool. This provided an overall savings per part of 1.7%

However, when operating at the 30% increase of surface speed, the overall savings per part would increase over 10 times to 20.7%. This, of course, is due to the fact that for the original uncoated tool application the percentage of the hob cost per part, to the total cost per part, was only 5.2%. When cutting under the same conditions with a coated hob, that ratio fell to 3.5% and increased to 8.7% when operating at the increased surface speed. Also note that the tool cost at the higher productivity conditions increased over and above that of the uncoated tool, 8.7% vs. 5.2%, but that overall savings per piece, when including machining time, was 20.7%.

The hob test in Table 3 was for a coarser pitch, double thread, Class C accurate unground hob, in which the surface speed was increased by 30% in the assumptions made, for comparing economic advantages at higher productivity rates. Note that the number of pieces per sharpening and the number of pieces per life of the hob are approximately 2 times and 3 times, respectively, for the coated hob at increased productivity over the uncoated hob. This large increase in tool life, combined with the approximate 30% reduction in machining time, yields a very substantial 27.1% savings in overall hobbing costs per part.

The shaper cutter test data for the 51 tooth automative gear of Table 4 is for a shaping application which produced exceptional tool wear reduction. When the TiN coated cutter was initially used at previously established production rates, the number of gears produced prior to its first sharpening, that is with TiN coating on the shaper cutter tooth face, was an astonishing 906 pieces or 12 times the number of pieces produced with an uncoated cutter. At that time, the amount of flank wear on the cutter was only .015 compared to .025 for the uncoated cutter. After sharpening the cutter, thus removing the TiN coating from the tooth face, it was found that 225 pieces could be cut before cratering had progressed to the point that it was feared cutting additional parts would cause tooth breakage. At that time, the flank wear on the cutter was only .005.

Continuing to operate under those productivity conditions and running lots of 225 pieces per .005 of sharpening, the cutter produced 22,500 parts during its life which changed the percent of cutter cost per part to the total cost per part from 21.3% for the uncoated cutter to 2.9% for the TiN coated cutter. This resulted in an 18.9% total cost savings per part cut.

However, an even greater savings in total gear shaping cost would be available by increasing the stroke rate by 20% from 750 to 900 strokes per minute. At the assumed increased wear of .010 per 225 pieces cut, the cutter cost per part would increase to $0.06 which would still be $0.21 less than the cost per part of the uncoated cutter. As shown in Table 4, the resulting machining cost reduction to $0.81 would result in a total savings per part cut of 31.5%.

Another shaping test of an 8.5 NDP 20 tooth 22° right hand helical gear is listed in Table 5. In this test it was intended to utilize as nearly as possible the same speeds and feeds with the coated cutter as were being applied in the existing production line with an uncoated cutter. However, some slight changes in stroke rate and depth of cut were necessary, and their values were adjusted in order to approximate the same cycle time. As can be seen in Table 5, this resulted in an increase of almost 12 times in cutter life (17550/1488) and a total savings per part of 22.6%.

Table 2. TiN Coated Hob Test

Hob Data		
Tool No.		T-368530
ID No.		67041-80-01
OD x Length	(in. x in.)	2.75 x 4.00
NDP		15.58
NPA	(deg)	20
No. Threads		1
Class	(AGMA)	C
No. Gashes		12
Sharpenable Tooth Length	(in.)	.370
Material		M2
Coating		TiNite™

Part Data		
Part No.		8631142
Material		5140H
Hardness		98-100 R_B
No. Teeth		16
Face Width	(in.)	1.016
Linear Inches/Part	(in)	17.09

Machine Data		
Make		Cleveland
Model No.		1886
Cap'y: Dia x Lg	(in. x in.)	
Age	(Yr.)	

Operating Data		Uncoated Hob	TiN Coated Hob	
			Same Speed, Feed &/or Depth	Incr'd. Speed Feed &/or Depth
Hob Speed 1st Cut	(rpm/sfm)	200/144	200/144	260/187*
2nd Cut	(rpm/sfm)			
Feed 1st Cut	(in/rev.)	.120	.120	.120
2nd Cut	(in/rev.)			
Hob Shift	(in/pc.)	.006	.006	.012*
Conventional/Climb		Climb	Climb	Climb
Coolant		Oil	Oil	Oil
Parts Per Load		1	1	1
Hob Travel	(in.)	2.00	2.00	2.00
No. Cuts		1	1	1
Hobbing Time	(min/pc.)	1.33	1.33	1.02
Machining Time Hourly Rate	($/Hr.)	25.00	25.00	25.00
Hobbing Results				
Av. No. Pcs. Per Sharpening	(pcs/Shr'g)	315	315	157*
Av. Stock Per Sharpening	(in.)	.030	.014	.014*
No. Sharpenings/Hob		12	26	26
Total Pieces Per Life of Hob	(pcs/Hob)	4095	8505	4239
Gear Cutting Cost Analysis				
Hob Price	($)	120	120	120
Price of TiN Coating	($)		48	48
Total Hob Price	($)	120	168	168
Hob Cost Per Part	($)	.03	.02	.04
Hob Savings/Part	($)		.01	-.01
Machining Cost/Part	($)	.55	.55	.42
Machining Savings/Part	($)		0	.13
Total Cost/Part	($)	.58	.57	.46
Total Savings/Part	($)		.01	.12
Hob Cost/Part to Total Cost/Part	(%)	5.2	3.5	8.7
Savings/Part	(%)		1.7	20.7

*Assumed

Table 3. TiN Coated Hob Test

Hob Data		
Tool No.		AD-587326
ID No.		76258-81-01
OD x Length	(in. x in.)	3 x 4
NDP		10.5
NPA	(deg)	22°30′
No. Threads		2
Class	(AGMA)	C
No. Gashes		13
Sharpenable Tooth Length	(in.)	
Material		M2
Coating		TiNite™

Part Data		
Part No.1		6835566
Material		5130H
Hardness		140 BHN
No. Teeth		19
Face Width	(in.)	1.44
Linear Inches/Part	(in.)	27.4

Machine Data		
Make		Lees-Bradner
Model No.		
Cap'y Dia x Lg	(in. x in.)	
Age	(Yr.)	

Operating Data		Uncoated Hob	TiN Coated Hob	
			Same Speed, Feed &/or Depth	Incr'd. Speed Feed &/or Depth
Hob Speed 1st Cut	(rpm/sfm)	290/228	290/228	377/296*
2nd Cut	(rpm/sfm)			
Feed 1st Cut	(in/rev.)	.060	.060	.060
2nd Cut	(in/rev.)			
Hob Shift	(in/pc.)	.010	.0025	.005*
Conventional/Climb				
Coolant		OIL	OIL	OIL
Parts Per Load		1	1	1
Hob Travel	(in.)	2.27	2.27	2.27
No. Cuts		1	1	1
Hobbing Time	(min/pc.)	1.24	1.24	.95
Machining Time Hourly Rate	($/Hr.)	25.00	25.00	25.00
Hobbing Results				
Av. No. Pcs. Per Sharpening	(pcs/Shr'g)	250	959	480*
Av. Stock Per Sharpening	(in.)	.018	.012	.012*
No. Sharpenings/Hob		8	12	12
Total Pieces Per Life of Hob	(pcs/Hob)	2000	11511	5760
Gear Cutting Cost Analysis				
Hob Price	($)	138.50	138.50	138.50
Price of TiN Coating	($)		55.00	55.00
Total Hob Price	($)	138.50	193.50	193.50
Hob Cost Per Part	($)	.07	.02	.03
Hob Savings/Part	($)		.05	.04
Machining Cost/Part	($)	.52	.52	.40
Machining Savings/Part	($)		0	.12
Total Cost/Part	($)	.59	.54	.43
Total Savings/Part	($)		.05	.16
Hob Cost/Part to Total Cost/Part	(%)	11.9	3.7	7.0
Savings/Part	(%)		8.5	27.1

*Assumed

Table 4. TiN Coated Shaper Cutter Test

Shaper Cutter Data		
Tool No.		
ID No.		65356-80-01
PD x Length	(in. x in.)	4.0886 x .975
NDP		18.7773
NPA	(deg)	20
No. Teeth		66
Class	(B-C)	3
Helix Angle/Hand	(deg)	33/RH
Sharpenable Tooth Length	(in.)	.500
Material		M2
Coating		TiNite™

Part Data		
Part No.		
Internal/External		External
Material		
Hardness		
No. Teeth		51
Face Width	(in.)	
Helix Angle	(deg)	33
Linear Inches/Part	(in.)	

Machine Data		
Make		
Model No.		
Cap'y: Dia x LG	(in. x in.)	
Age	(Yr.)	

Operating Data			Uncoated Cutter	TiN Coated Cutter: Same Speed, Feed &/or Depth	TiN Coated Cutter: Incr'd. Speed Feed &/or Depth
Total Depth Of Cut		(in.)	.160	.160	.160
Infeed/Stroke		(in./Stk.)	.001	.001	.001
Work Pitch Diameter		(in.)	3.238	3.238	3.238
Infeed Time		(min.)	.347	.347	.289
1st Cut	Depth	(in.)	.090	.090	.090
	Rotary Feed	(in./Stk.)	.025	.025	.025
	Stroke Rate	(Stk./min.)	750	750	900*
	Cutting Time	(min.)	.542	.542	.452
2nd Cut	Depth	(in.)	.055	.055	.055
	Rotary Feed	(in./Stk.)	.025	.025	.025
	Stroke Rate	(Stk./min.)	750	750	900*
	Cutting Time	(min.)	.542	.542	.452
3rd Cut	Depth	(in.)	.015	.015	.015
	Rotary Feed	(in./Stk.)	.015	.015	.015
	Stroke Rate	(Stk./min.)	750	750	900*
	Cutting Time	(min.)	.978	.978	.753
Machining Time		(min.)	2.41	2.41	1.95
Parts Per Load			1	1	1
Machining Time Hourly Rate		($/Hr.)	25.00	25.00	25.00
Coolant			Sol Oil	Sol Oil	Sol Oil
Shaping Results					
Av. No. Pcs. Per Sharpening		(pcs/Shpg.)	75	225	225*
Av. Stock Per Sharpening		(in.)	.025	.005	.010*
No. Sharpenings/Cutter		(Shpg./Ctr.)	20	100	50
Total Pieces/Life of Ctr.		(pcs./Ctr.)	1500	22500	11250
Gear Cutting Cost Analysis					
Shaper Cutter Price		($)	399.00	399.00	399.00
Price of TiN Coating		($)		239.00	239.00
Total Shaper Cutter Price		($)	399.00	638.00	638.00
Cutter Cost Per Part		($)	.27	.03	.06
Cutter Savings/Part		($)		.24	.21
Machining Cost/Part		($)	1.00	1.00	.81
Machining Savings/Part		($)		0	.19
Total Cost/Part		($)	1.27	1.03	.87
Total Savings/Part		($)		.24	.40
Cutter Cost/Part-to Total Cost/Part		(%)	21.3	2.9	6.9
Savings/Part		(%)		18.9	31.5

*Assumed

Table 5. TiN Coated Shaper Cutter Test

Shaper Cutter Data		
Tool No.		FGS-4749
ID No.		80238-82-00
PD x Length	(in. x in.)	4.40 x 1.25
NDP		8.5
NPA	(deg)	22°30'
No. Teeth		36
Class	(B-C)	3
Helix Angle/Hand	(deg)	22/LH
Sharpenable Tooth Length	(in.)	.625
Material		M2
Coating		TiNite™

Part Data		
Part No.		10-22-080-002
Internal/External		External
Material		4027H
Hardness		
No. Teeth		20
Face Width	(in.)	1.18
Helix Angle	(deg)	22/RH
Linear Inches/Part	(in.)	25.4

Machine Data		
Make		B-C
Model No.		HD 150
Cap'y Dia x Lg	(in. x in.)	6 x 1.57
Age	(Yr.)	1

Operating Data			Uncoated Cutter	TiN Coated	Cutter
				Same Speed, Feed &/or Depth	Incr'd. Speed Feed &/or Depth
Total Depth Of Cut		(in.)	.333	.333	.333
Infeed/Stroke		(in./Stk.)	.00136	.00136	.00136
Work Pitch Diameter		(in.)	2.5377	2.5377	2.5377
Infeed Time		(min.)	.790	.706	.706
1st Cut	Depth	(in.)	.166	.166	.261*
	Rotary Feed	(in./Stk.)	.0326	.0326	.0326
	Stroke Rate	(Stk./min.)	400	448	448
	Cutting Time	(min.)	.611	.546	.546
2nd Cut	Depth	(in.)	.095	.095	.072*
	Rotary Feed	(in./Stk.)	.0326	.0326	.012
	Stroke Rate	(Stk./min.)	400	448	672
	Cutting Time	(min.)	.611	.546	.988
3rd Cut	Depth	(in.)	.072	.072	—*
	Rotary Feed	(in./Stk.)	.0115	.009	—*
	Stroke Rate	(Stk./min.)	600	672	—*
	Cutting Time	(min.)	1.155	1.318	—*
Machining Time		(min.)	3.167	3.116	2.240
Parts Per Load			1	1	1
Machining TimeHourly Rate		($/Hr.)	25.00	25.00	25.00
Coolant			Oil	Oil	Oil
Shaping Results					
Av. No. Pcs. PerSharpening		(pcs/Shpg.)	48	225	112*
Av. Stock Per Sharpening		(in.)	.020	.008	.008*
No. Sharpenings/Cutter		(Shrg./Ctr.)	31	78	78
Total Pieces/Life of Ctr.		(pcs./Ctr.)	1488	17550	8775
Gear Cutting Cost Analysis					
Shaper Cutter Price		($)	676.48	676.48	676.48
Price of TiN Coating		($)		136.00	136.00
Total Shaper Cutter Price		($)	676.48	812.48	812.48
Cutter Cost Per Part		($)	.45	.05	.09
Cutter Savings/Part		($)		.40	.36
Machining Cost/Part		($)	1.32	1.32	.93
Machining Savings/Part		($)		0	.39
Total Cost/Part		($)	1.77	1.37	1.02
Total Savings/Part		($)		.40	.75
Cutter Cost/Part To Total Cost/Part		(%)	25.4	3.6	8.8
Savings/Part		(%)		22.6	42.4

Table 6. Tool Cost Vs. Machining Cost per piece of Uncoated and TiN Coated Tools

From Table	Tool / Work Material	Costs/Savings		Uncoated	TiN Coated: Same Speed, Feed & Depth	TiN Coated: Increased Speed, Feed &/or Depth
2	2.75 x 4 15.58 NDP UNG HOB M2 / 514OH 100 R_B	Tool Cost/PC	$	.03	.02	.04*
		Machining Cost/PC	$	.55	.55	.42
		Tool Cost of Total	%	5.2	3.5	8.7
		Savings/PC	%	—	1.7	20.7
3	3 x 4 10.5 NDP UNG HOB M2 / 5130H 140 BHN	Tool Cost/PC	$	.07	.02	.03*
		Machining Cost/PC	$	.52	.52	.40
		Tool Cost of Total	%	11.9	3.7	7.0
		Savings/PC	%	—	8.5	27.1
4	4.0886x.975 18.7773 NDP Shaper Cutter M2	Tool Cost/PC	$	.27	.03	.06*
		Machining Cost/PC	$	1.00	1.00	.81
		Tool Cost of Total	%	21.3	2.9	6.9
		Savings/PC	%	—	18.9	31.5
5	4.40x1.25 8.5 NDP Shaper Cutter M2 / 4027H	Tool Cost/PC	$	.45	.05	.09*
		Machining Cost/PC	$	1.32	1.32	.93
		Tool Cost of Total	%	25.4	3.6	8.8
		Savings/PC	%	—	22.6	42.4

*Assumed

The above summary combined with similar tests from a number of other applications indicates the following values are realistic.

Tool Condition	Production Rates	$\frac{\text{Tool Cost}}{\text{Total Cost}}$ %	Total Savings %
Uncoated	Existing	5 to 25	—
TiN Coated	Existing	2 to 5	2 to 20
TiN Coated	Increased	5 to 10	20 to 40

During the course of testing, the TiN coated cutter it was suggested that it would be appropriate to reduce the number of cuts from two to three in order to increase overall productivity.

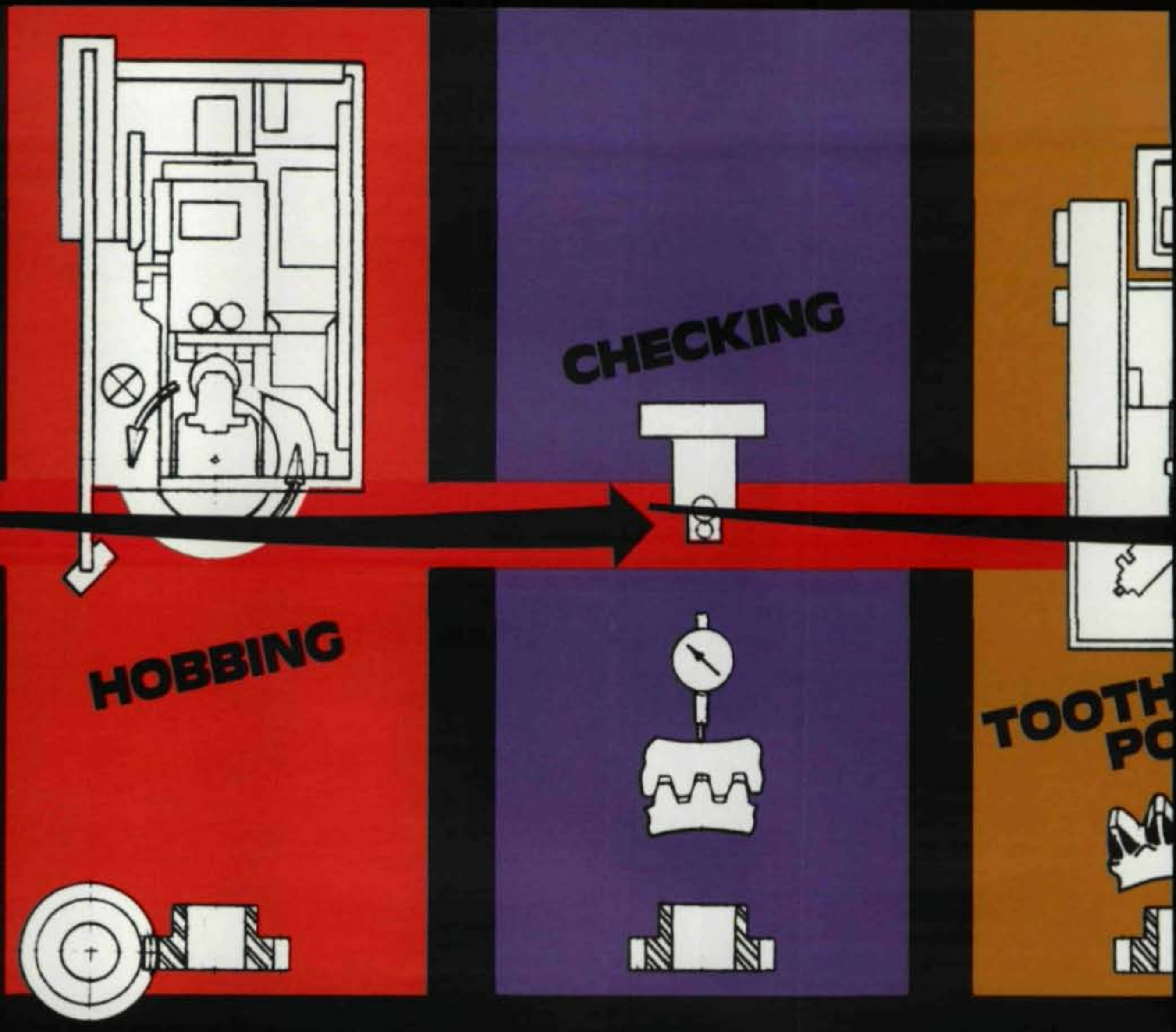
Therefore, the assumptions in establishing the calculations for Table 5 were that combining the depth for the original first and second cut and holding the average sharpening stock to .008 would result in being able to cut 112 gears per sharpening. This increase in productivity would reduce the machining cost per part by $0.39 and provide a total savings per part 42.4%.

The economic results of Tables 2 through 5 are summarized in Table 6 for easy reference where it can be seen that for 8 to 20 DP high production gear tools the maximum savings potential is available in utilizing TiN coated tools at increased production rates. It is possible to save 20 to 40% of the gear cutting costs.

New Coating Developments

As noted earlier, the application of titanium nitride coatings to gear cutting tools is relatively new. The major coatings which have proven advantageous to-date are titanium carbide and titanium nitride. Currently, titanium carbide is being applied primarily by the CVD process. Work is being done to enable titanium carbide coating by the PVD process. Titanium carbide has the advantage in that its hardness is approximately Rockwell C 90 which is somehwat harder than the titanium nitride coating at Rockwell C 85. However, titanium carbide is more brittle and must be utilized with more care. It also has a higher coefficient of friction which would generate more heat and higher tool loads. At present, the CVD process often is

(Continued on page 48)



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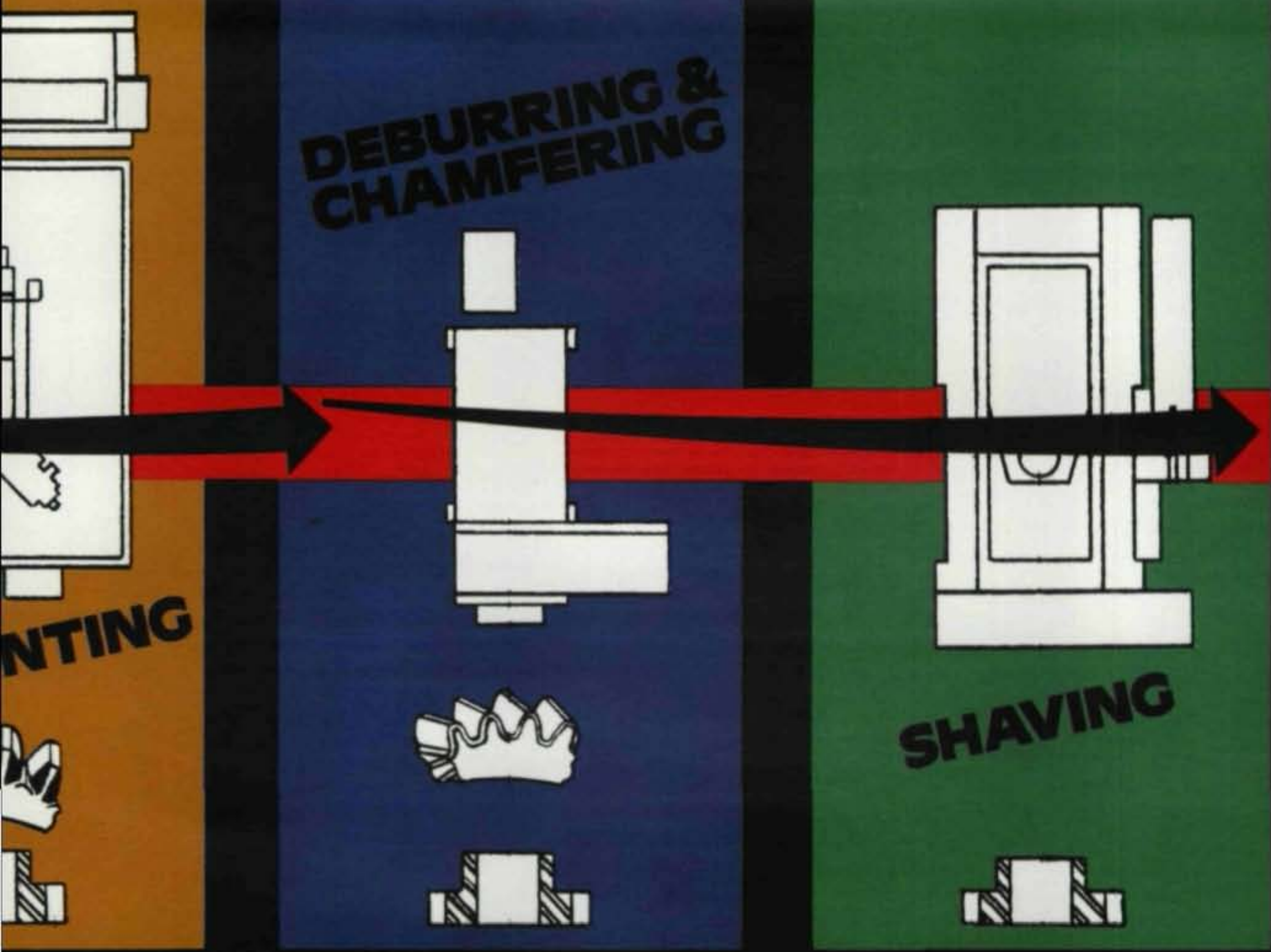
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CNC Controlled CBN Form Grinding

by Dennis R. Gimpert
American Pfauter Limited

Introduction to CBN

Borazon is a superabrasive material originally developed by General Electric in 1969, (Figure 1). It is a high performance material for machining of high alloy ferrous and super alloy materials. Borazon CBN — Cubic Born Nitride — is manufactured with a high temperature, high pressure process similar to that utilized with man-made diamond. Borazon is, next to diamond, the hardest abrasive known; it is more than twice as hard as aluminum oxide. It has an extremely high thermal strength compared to diamond, (Figures 2 & 3). It is also much less chemically reactive with iron, cobalt or nickel alloys.

Traditionally, aluminum oxide wheels have been utilized in production grinding operations. In practice, CBN wheels have advantages due to their hardness and thermal stability. In general, CBN wheels are 3000 to 4000 more wear resistant than aluminum oxide. Since aluminum oxide wheels wear much more rapidly, this wheel must be dressed and/or sized frequently. Further, aluminum oxide wheels must be changed more often with higher machine tool downtime.

In April of 1981, newer CBN materials were introduced by General Electric with their 550, 560 and 570 series. These CBN crystals are micro-crystalline in structure versus conventional CBN which has a single crystal structure. What does the micro-crystalline structure provide? This newer CBN material is composed of micron size (1 um, .000040") crystals. Thus, as the CBN cutting edges becomes dull, any fracturing that occurs to the crystal is microscopic in nature versus large scale fracturing as in the traditional CBN. The 550, 560 & 570 type CBN is self sharpening and will remain sharp throughout its life. A new range of workpiece

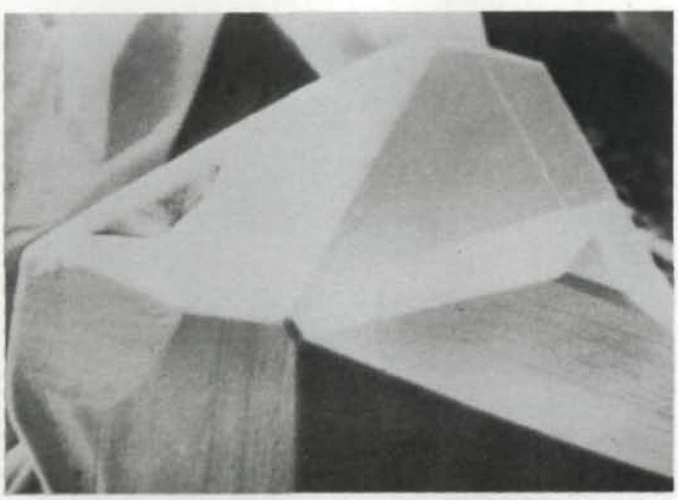


Figure 1 — Cubic Boron Nitride crystals

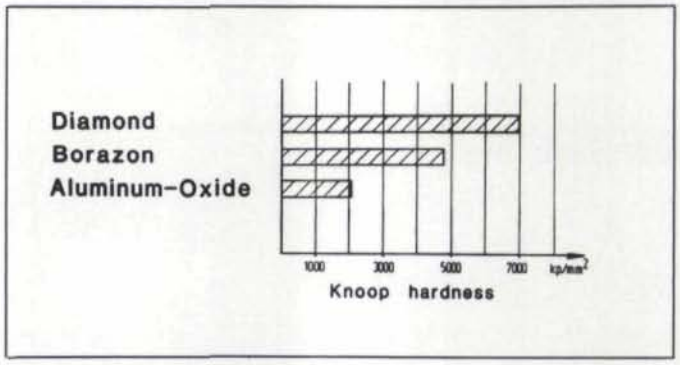


Figure 2 — Knoop hardness scale

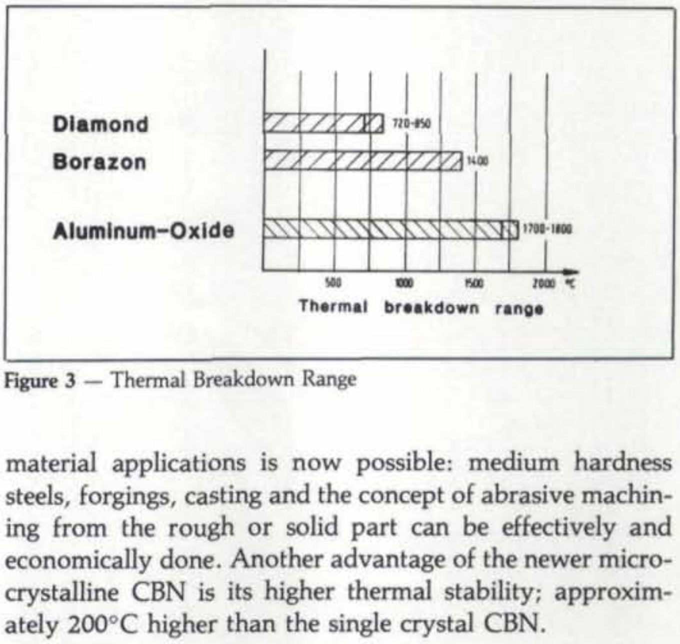


Figure 3 — Thermal Breakdown Range

material applications is now possible: medium hardness steels, forgings, casting and the concept of abrasive machining from the rough or solid part can be effectively and economically done. Another advantage of the newer micro-crystalline CBN is its higher thermal stability; approximately 200°C higher than the single crystal CBN.

CBN Machining

To understand how CBN "grinds", or more correctly machines, it is useful to think of it as a milling process rather than as a grinding process. Each exposed CBN crystal is, in effect, like the tooth of a milling cutter. Consider then, the thousands of consistently sharp cutting edges which are exposed to the workpiece and making chips, (Figure 4). Creep feed, (without stickslip effect), and heavy infeed are applied to give the CBN time to cut much in the same way as a milling cutter is fed through its cut, based upon chip load per cutting tooth. Swarf collected from CBN machines consists of tiny chips similar to those from a milling operation. The chip making action also reduces heat generation which is a

Figure 4 — Micro Photograph of CBN plated form wheel

constant problem with dull aluminum oxide wheels. Aluminum oxide also tends to burnish rather than cut a material, which increases heat and reduces metal removal rates.

The effective application of coolant to the workpiece and CBN wheel is critical. Coolant is utilized for two purposes; as a lubricant for the cutting process and to dissipate heat. When CBN is properly utilized, the workpiece will remain cool and show no thermal damage. Any heat that is generated during the process is transferred to the chip and coolant medium. Lubricity is important to prevent workpiece chips from adhering to the wheel and reducing the CBN machine efficiency. Many different coolants have been utilized from straight mineral oil to a 10% soluble oil. The Kapp Company recommends Texaco Transultex Type A cutting oil which has a sulfur chlorinated base. In high metal removal conditions, flow rates of 150 1/min (40 gal./min) are recommended.

A stiff rigid machine tool is required to successfully apply CBN techniques. The machine should be equipped with such modern items as ball-screw drives for feed slides, way materials with low coefficients of friction and high dampening qualities and intensive coolant flooding of the grinding zone. The grinding spindle must be both a precision running spindle and of a rigid design. Although horsepower required is not high due to the efficient CBN chip making process, it is normally greater than on traditional grinding machines. Lower horsepower will not allow full advantage of CBN capabilities. The General Electric Company recommends a minimum of 2 to 3 spindle horsepower for each inch of CBN wheel width on a 6 to 10 inch diameter wheel. Another good machine feature is an electric load meter. As the CBN wears down, the cutting efficiency decreases which results in a higher spindle power load. With a load meter, it is possible to monitor this cutting efficiency and to replace (and replate) the CBN wheel at the desired point.

CBN Electroplated Wheels

Until recently, Borazon has been primarily utilized in resin or metal bond wheels. This bonded wheel was dressed and/or conditioned to expose the grain. Further, the bonded wheel required periodic redressing to maintain correct form due to the bond failing. Worth repeating, is the fact that this redressing was required since the CBN crystal simply was no longer supported by the bonding material. If the form had a complex or highly accurate geometry, costly dressing rolls or equipment were needed. In some traditional aluminum oxide form grinding applications, 25% of the total grinding time is spent on dressing. In small diameter form wheel applications, the problems of traditional dressing and truing becomes even greater.

A solution to the problem of maintaining a true form is the electroplated CBN wheel, (Figure 5). The high form accuracy is achieved by the use of a precision machined form wheel with a single layer of galvanically bonded CBN. This electroplated bonding is nickel with sufficient plating to capture approximately 50% of the crystal. Since the form wheel is precisely controlled, the need for wheel dressing is completely eliminated. Further, after the CBN form wheel has worn, it can be stripped of the remaining CBN crystals and recoated. A prime example of successful plated CBN wheel application is the grinding of internal ball tracks of constant velocity joints. In one case, an electroplated wheel was able to grind 200 parts. Previous aluminum oxide wheels ground less than 3 parts before dressing was required.

Future developments with electroplated CBN wheels look promising. Present wheel surface speeds are limited, to an extent, by present machine and grinding spindle design. With higher wheel speeds of up to 100 m/second (20,000 sfpm), wheel life and material removal rates can both be increased. Work is presently underway with magnetic bearing designs to allow higher wheel speeds with small diameter wheels. Another problem to be solved with increased wheel speeds is the critical requirement of proper coolant application. As wheel speeds increase, the phenomenon of high-speed air barrier occurs. This phenomenon makes the application of coolant at the wheel and workpiece more difficult. Wheel manufacturers are now experimenting with slotted wheel designs and direct coolant application through integrated coolant holes in the wheel itself.

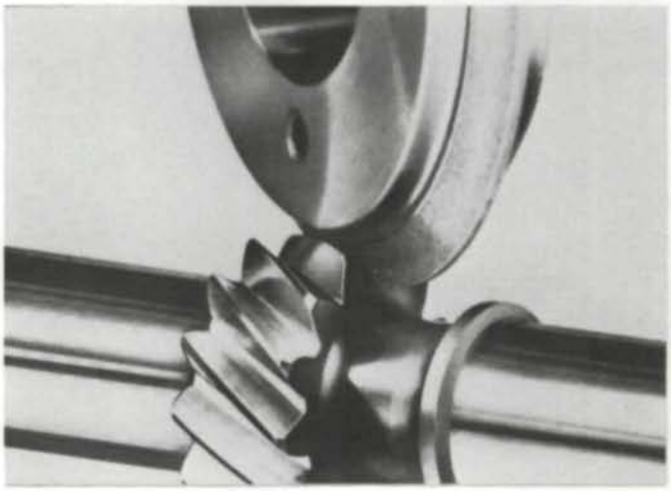


Figure 5 — CBN plated form wheel

Conclusion

The use of Borazon CBN form grinding represents one of the most significant process developments for soft and hard machining in the last 50 years. With multi-axis CNC machine tools, the process is repeatable and precise with setups a simple matter of recalling axis and feed data from memory as a coded parts program. CBN is no longer an "expensive" method done only when no other way is possible. It now competes with milling, shaping and other finishing processes by metal cutting and, in some instances, even against rough cutting. Any form machinable by a disc or pencil type milling cutter or grinding wheel can be ground by CBN form grinding. On tough materials such as stainless, inconel and the high nickel content types, CBN may be the only "practical" method of finishing.

Advantages of Electroplated CBN Form Wheels

1) Accurate control of form
2) Maintain modified profile due to form incorporation in wheel
3) Possible to grind finer forms from the solid where heat treatment allows
4) Non-standard forms can be ground, i.e. gerotors or compressor screw rotor
5) Can grind full form, including root
6) No dressing is required
7) CBN wheels can be stripped and recoated hence, low cost per workpiece
8) Free cutting without heat checking
9) More economical than vitrified wheels on a production basis
10) Long life. CBN wheel life has an average of 10,000 times the diameter of the wheel equal to linear units of form length ground
11) Reduced grinding time
12) Low machine wear due to creep feeds and absence of grinding dust
13) High precision

Disadvantages of Electroplated CBN Form Wheels

1) Requires CBN form wheel for each distinct form
2) Lead time required to design and manufacture form wheel

Production Examples

Hob Sharpening

Wheel Data
- 15 degree CBN electroplated wheel
- 200 mm, (7.87") diameter

Hob Data
- 5 Pitch
- 145 mm, (5.75") face width
- 14 gashes
- Material M3
- .80 mm, (.032") wear

Machine Data
- Kapp Model AS204GT Hob Sharpener

Grinding Wheel
- 1370 surface meters (4500 surface feet)
- 2030 mm, (80") stroke per minute
- .20 mm, (.008") stock removal per pass

Total Grinding Time
- 12.5 minutes

Quality
- MCTI Class AA

Internal Gear Grinding

Wheel Data
- CBN Electroplated Wheel
- 65 mm (2.56") Diameter

Part Data
- 20/40 Pitch
- 114 Teeth
- 30 mm (1.18") Face
- 1.37 mm (.054") Whole Depth
- Material — High Heat Resistant

Machine Data
- Kapp VIG335CNC Internal Gear Grinder

Grinding Data

Pieces Per Load	1
Stock Removal	Roughing 1.0 mm (.039") Finishing .3 mm (.012")
Number of Passes	2
Feed Rate	Roughing 300 mm (11.81") Finishing 300 mm (11.81")

Total Grinding Time
- 90 Minutes Per Piece

External Gear Grinding

Wheel Data
- CBN electroplated wheel
- 250 mm, (9.84") diameter

Part Data
- 6.35 Pitch
- 30 Teeth
- 40 mm, (1.61") face
- 8.9 mm, (.350") whole depth
- AISI E52100 material
- Rc 62 hardness

Machine Data
- Kapp Model VAS481CNC External Gear Grinder

Grinding Data
- Pieces per load — 2
- Stock removal per flank .1mm (.004")
- Number of passes — 1
- 1370 surface meters (4500 surface feet) per minute
- 600 mm (23.6") per minute feed rate
- 6000 mm (236.0") per minute rapid return

Total Grinding Time
- 6.0 minutes per load
- 3.0 minutes per piece

(Continued on page 46)

BACK TO BASICS...

Gear Design

National Broach and Machine Division of
Lear Siegler, Inc.

A gear can be defined as a toothed wheel which, when meshed with another toothed wheel with similar configuration, will transmit rotation from one shaft to another. Depending upon the type and accuracy of motion desired, the gears and the profiles of the gear teeth can be of almost any form.

Gears come in all shapes and sizes from square to circular, elliptical to conical and from as small as a pinhead to as large as a house. They are used to provide positive transmission of both motion and power. Most generally, gear teeth are equally spaced around the periphery of the gear.

The original gear teeth were wooden pegs driven into the periphery of wooden wheels and driven by other wooden wheels of similar construction. As man's progress in the use of gears, and the form of the gear teeth changed to suit the application. The contacting sides or profiles of the teeth changed in shape until eventually they became parts of regular curves which were easily defined.

To obtain correct tooth action, (constant instantaneous relative motion between two engaging gears), *the common normal of the curves of the two teeth in mesh must pass through the common point, or point of contact, of the pitch circles of the two wheels*, Fig. 1-1. The common normal to a pair of tooth curves is the line along which the normal pressure between the teeth is exerted. It is not necessarily a straight line. Profiles of gear teeth may be any type or types of curves, provided that they satisfy the law of contact just defined. However, manufacturing considerations limit the profiles to simple curves belonging to the circle group, or those which can be readily generated or form cut, as with gear cutters on standard milling machines.

Because of inherent good properties and easy reproducibility, the family of cycloid curves was adopted early (1674) and used extensively for gear tooth profiles. The common

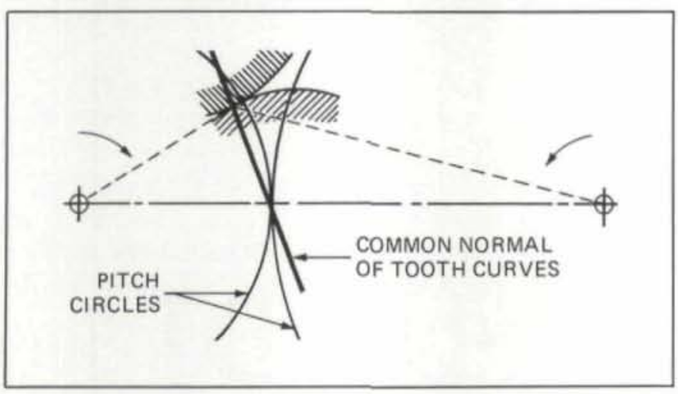


FIG. 1-1—For constant instantaneous relative motion between two engaging gears, the common normal of the curves of the two teeth in mesh must pass through the common point, or point of contact, of the pitch circles of the two gears.

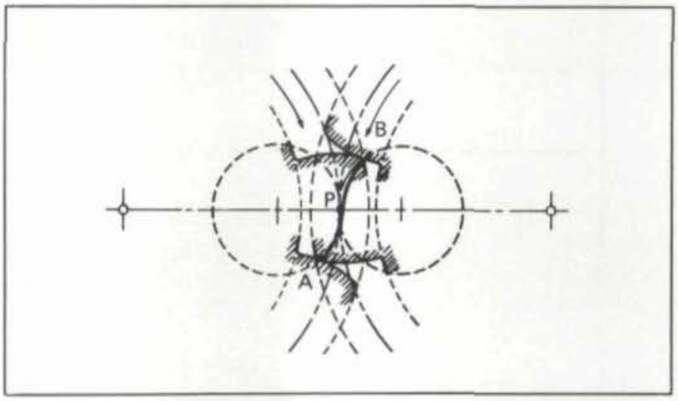


Fig. 1-2—The common normal of cycloidal gears is a curve which varies from a maximum inclination with respect to the common tangent at the pitch point to coincidence with the direction of this tangent. For cycloidal gears rotating as shown here, the arc *BP* is the*Arc of Approach*, and the arc *PA*, the *Arc of Recess*.

normal of cycloidal gears is a curve, Fig. 1-2, which is not of a fixed direction, but varies from a maximum inclination with respect to the common tangent at the pitch point to coincidence with the direction of this tangent. Cycloidal gears roll with the direction of this tangent. Cycloidal gears roll with conjugate tooth action providing constant power with uninterrupted rotary motion. One disadvantage of this type of gear is that the center distance between mates must be held to fairly close tolerances, otherwise mating gears will not perform satisfactorily.

The involute curve was first recommended for gear tooth profiles in the year 1694 but was not commonly used until 150 years later. The curve is generated by the end of a taut line as it is unwound from the circumference of a circle, Fig. 1-3. The circle from which the line is unwound is commonly

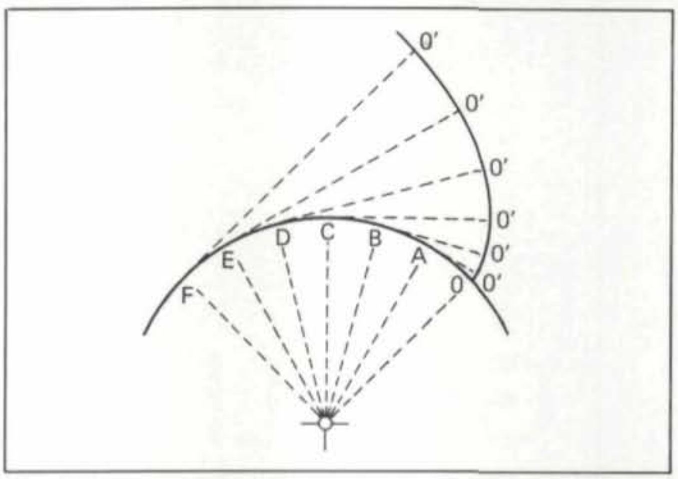


Fig. 1-3—The involute tooth form used for virtually all gearing today is generated by the end of a taut line as it is unwound from the circumference of a circle. The circle from which the line is unwound is the *Base Circle*.

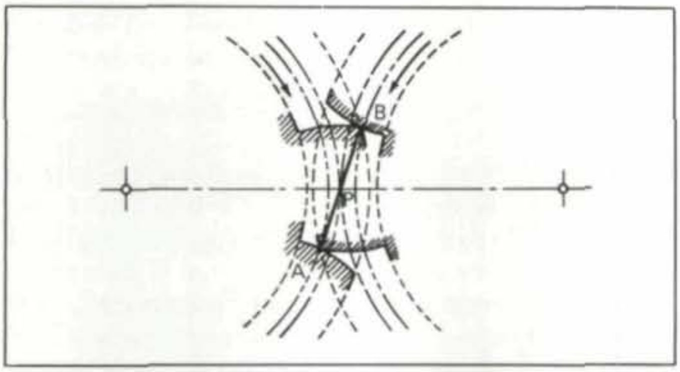


Fig. 1-4—The common normal of involute gear teeth is a straight line *AB*.

known as the "base circle". The common normal of involute gear teeth is a straight line (AB in Fig. 1-4). Gears of this type satisfy all the requirements for smooth, accurate and continuous motion. Gears with involute tooth profile are very flexible in both geometric modification and center distance variation.

There have been many other types of gear tooth forms, some related to the involute curve. One particular type of recent interest is the "circular arc" gear (where the profile is an arc from the circumference of a circle). First proposed in this country by Ernest Wildhaber in the 1920's, the circular arc gear was recently introduced by the Russians as the "Novikov" tooth form. These profiles are not conjugate. Gears with this tooth form depend upon helical overlapping of the teeth in order to roll continuously. This can and does create face width size and end thrust problems.

At the present time, except for clock and watch gears, the involute curve is almost exclusively used for gear tooth profiles. Therefore, except for an occasional comment, the following discussion will cover some of the basic elements and modifications used in the design of involute tooth form gears.

Ratio

The primary purpose of gears is to transmit motion and at the same time, multiply either torque or speed. Torque is a function of the horsepower and speed of the power source. It is an indication of the power transmitted through a driving shaft and from it the gear tooth loads are calculated. The loads applied to gear trains can vary from practically

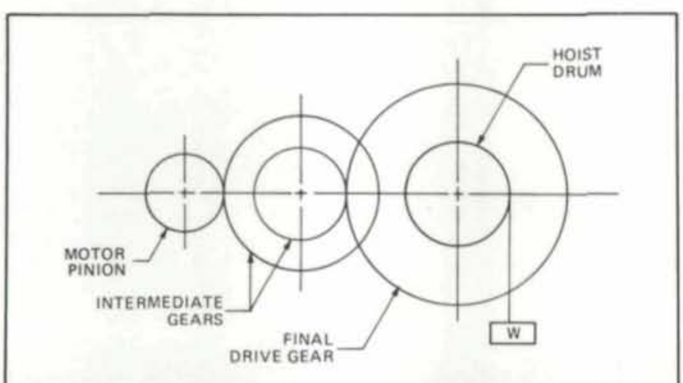


Fig. 1-5—Ratios of the gear teeth in this hypothetical hoist drive would depend upon weight (*W*) to be lifted and torque (*T*) available from the motor.

nothing to several tons or more. Gears, properly designed and meshed together in mating pairs, can multiply the torque and reduce the higher rotational speed of a power producing source to the slower speeds needed to enable the existing power to move the load. Where application requires speed rather than torque, the process if reversed to increase the speed of the power source.

Rotational speeds of the shafts involved in power transmission are inversely proportional to the numbers of teeth (**not** the pitch diameters) in the gears mounted on the shafts. With the relative speed of one member of a pair of gears known, the speed of the mating gear is easily obtained by the equation:

$$n_G = \frac{n_p N_p}{N_G}$$

Where N_p and N_G = Number of teeth in pinion and gear.
n_p and n_G = Revolutions per minute (rpm) of pinion and gear respectively.

The ratio of speed to torque is of the utmost importance in the design of gear teeth to transmit and use the power. A typical case would involve the design of the gearing for a hoist to raise a certain weight (W) at a uniform speed, when making use of a motor with a given horsepower (hp) running at a given speed (rpm) and driving through a pinion with number of teeth N_p, Fig. 1-5.

Obviously, the ratio of the gear teeth and the number of gears needed depend entirely upon the application and the power source.

Velocity

Circumferential velocity is an important factor present in all running gears. Its value is obtained by multiplying the circumference of a given circle by the rpm of the shaft. In reference to the pitch circle it is generally referred to as "pitch line velocity" and expressed as "inches per minute" or "feet per minute".

Circumferential velocities in a complex gear train have a direct effect on the loads to be carried by each pair of gears. As the load *W*, in Fig. 1-5, is shown tangent to the perephery of the final cylinder, so the loads on gear teeth are applied tangent to the pitch diameters and normal to the gear tooth profile. Since the rpm's of mating gears are inversely proportional to the numbers of teeth, it can be shown that the pitch line velocities of the two gears are equal and the loads carried by their respective teeth will also be equal.

Elements of Gear Teeth

A very excellent reference for the names, description and definition of the various elements in gears is the American Gear Manufacturers Association (AGMA) Standard entitled "Gear Nomenclature".

Pitch

Pitch is generally defined as the distance between equally spaced points or surfaces along a given line or curve. On a cylindrical gear it is the arc length between similar points on

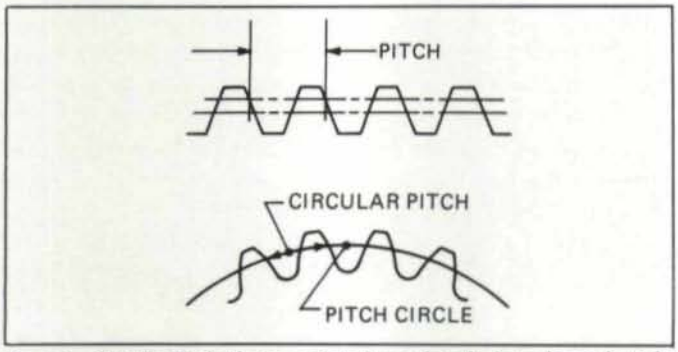


Fig. 1-6—*Circular Pitch* of gear teeth is the arc length along the pitch circle between identical points on successive teeth.

successive teeth and is known as **circular pitch** (p). See Fig. 1-6. Therefore, by definition, circular pitch of gear teeth is a function of circumference and numbers of teeth, varying with diameter and evolving into straight line elements as shown in Figs. 1-7 and 1-8. In Fig. 1-7 the teeth are shown as **helical,** or at an angle to the axis of the gear cylinder. If the teeth were parallel to the axes they would be straight or **spur** teeth as they are more commonly called. With **spur** teeth, Fig. 1-7, the **normal circular pitch** and the **transverse circular pitch** would be equal and the **axial pitch** (a straight line element) would be infinite.

One of the most important pitch classifications in an involute gear is the one termed **base pitch**, in Fig. 1-8. Primarily, it is the circular pitch on the perimeter of the base circle, but by definition of the involute curve the arc distance becomes the linear normal distance between corresponding sides of adjacent teeth when raised to position as part of the **taut line.** In spur gears there is only one base pitch to consider. On the other hand, in helical gears, base pitch is definable in the section normal to the helix angle (**normal base pitch**), parallel to the gear axis (**axial base pitch**) and perpendicular to the gear axis (**transverse base pitch**), Fig. 1-9. Since the gear teeth are equally spaced, it becomes apparent that in order to roll together properly, two gears must have the same base pitch. More specifically, two mating involute gears must have the same **normal base pitch.**

Originally gears were classified and calculated beginning with circular pitch. With the number of teeth (N) and the

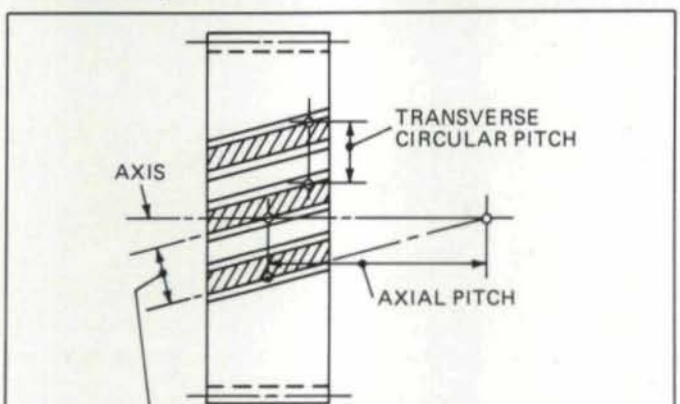


Fig. 1-7—For helical gear teeth, pitch may be measured along a line normal to the gear teeth (*Normal Circular Pitch*), in a direction perpendicular to the axis of rotation (*Transverse Circular Pitch*), and in a direction parallel to the axis of rotation (*Axial Pitch*).

circular pitch (p) given, the circumference of the circle and consequently the **pitch diameter** (D) can be calculated from

$$D = \frac{N \times p}{\pi}$$

For simplification, developers of gear design techniques created a separate term for the value of π divided by circular pitch (π/p). This is **diametral pitch** (P) Fig. 1-10 which is the ratio of teeth to the pitch diameter in inches. It is a number, it cannot be seen or measured. However, the system developed since the inception of diametral pitch is used almost exclusively wherever the decimal system of measuring is used.

Diametral pitch regulates the proportions or size of the gear teeth. The number of gear teeth and the diametral pitch regulate the size of the gear. Therefore, for a known load to be transmitted, the pitch is chosen which in turn determines the number of teeth to suit the desired ratio and size of gear. The number of teeth divided by the diametral pitch produces the diameter of the gear pitch circle, Fig. 1-9. The part of the tooth above the pitch circle is called the **addendum** and the lower part **dedendum,** Fig. 1-11. Two addendums added to the pitch diameter equal the outside diameter of the gear.

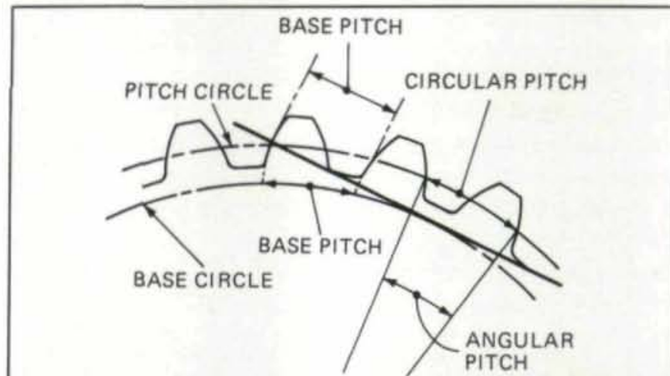


Fig. 1-8—*Base Pitch and Angular Pitch* as defined by this drawing are important gear terms. In order to roll together properly, involute gears must have the same *Normal Base Pitch*.

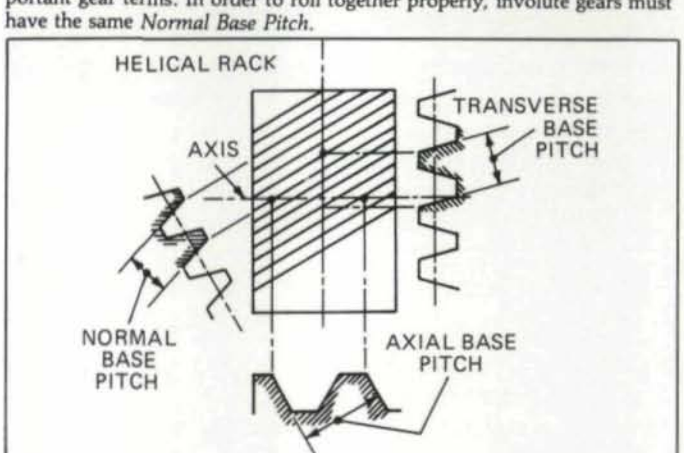


Fig. 1-9—This drawing defines *Transverse Base Pitch, Normal Base Pitch* and *Axial Base Pitch* for a helical rack.

Pressure angles

Pressure angles in involute gears are generally designated by the greek letter phi (ϕ), with subscripts to denote the various sections and diameters of the gear tooth, Fig. 1-12.

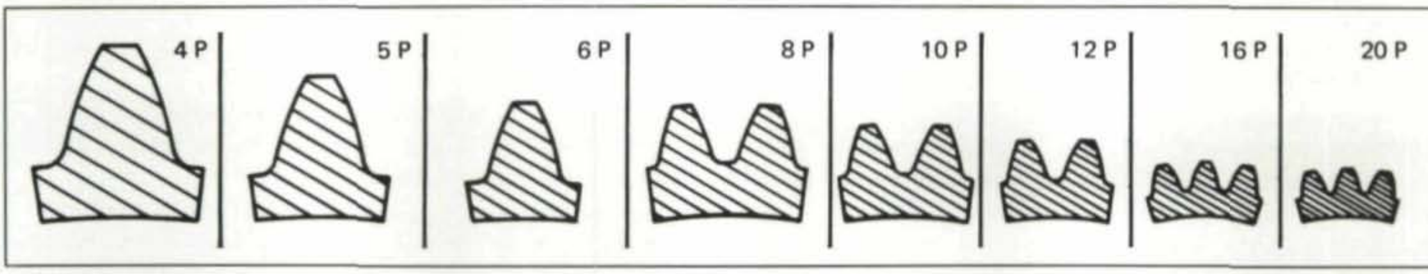


Fig. 1-10—Gear teeth of different diametral pitch, full size, 20-Deg. pressure angle.

An involute curve is evolved from origin point A on a base circle. The point P on a taut line containing point B describes the curve. The taut line is tangent to the base circle at point B, and normal to the involute curve at P. This line segment BP is known as the **radius of curvature** of the involute curve at point P and is equal in length the arc AB. The angle ϵ subtended by the arc AB is the roll angle of the involute to the point P. The angle between OP (radius r) and OB (base radius r_b) *is the pressure angle* ϕ at point P. Angle θ between the origin OA and radius OP is the polar angle of point P. (The polar angle θ and the radius r are the **polar coordinates** of point P on the involute curve). When given in radians, angle θ is known as the **involute function** of the pressure angle θ and is used extensively in gear calculations.

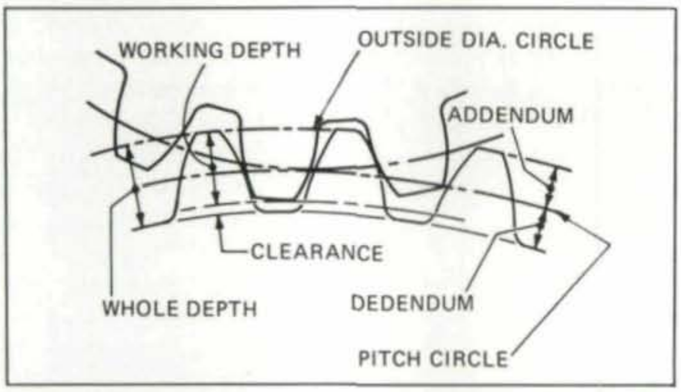


Fig. 1-11—The portion of a gear tooth above the pitch circle is called the *Addendum;* the portion of the tooth below the pitch circle is called the *Dedendum.*

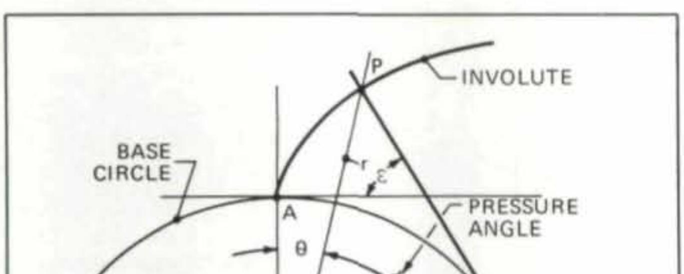


Fig. 1-12—This drawing defines *Roll Angle* (ϵ), *Pressure Angle* (ϕ) and *Polar Angle* (Θ).

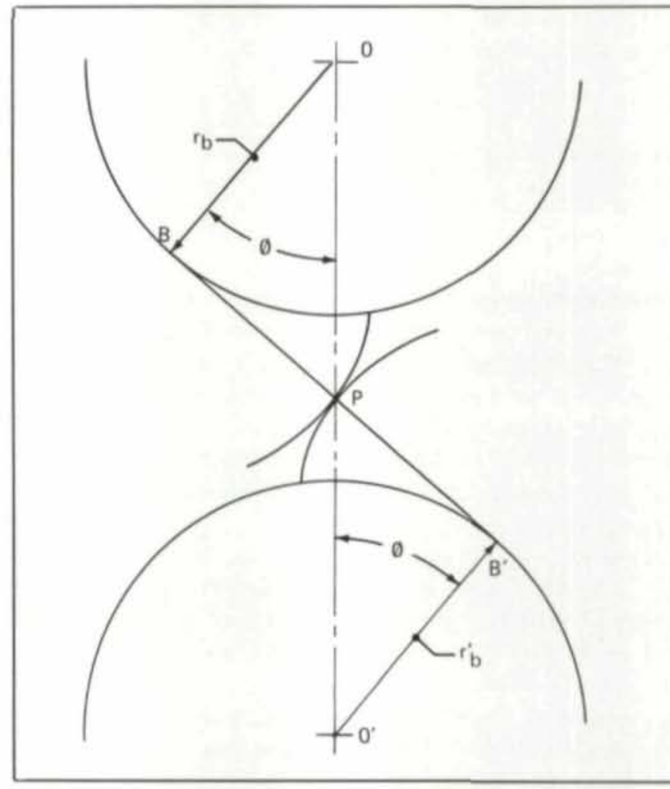


Fig. 1-13—When two involute teeth are brought into contact and made tangent at a point P, pressure angle ϕ is equal to both.

When two involute curves are brought together as profiles of gear teeth and are made tangent at a point P, the pressure angle θ is equal on both members, Fig. 1-13. The line BB' is the common normal passing through the point of contact P and is tangent to both base circles. All contact and tooth action will take place along the common normal. If one member is rotated, the involute curves will slide together and drive the other member in the opposite direction.

The pressure angle through the point of contact of a pair of involute curves is governed by regulating the distances between the centers of their respective base circles. A gear does not really have a pressure angle until its involute curved profile is brought into contact with a mating curve as defined in Fig. 1-13. At that time the pressure angle θ becomes the **operating** or **rolling** pressure angle between the mating gears. For a given center distance, C, and base circle diameters, the rolling pressure angle is determined by the expression.

$$\cos \phi = \frac{r_b + r_b'}{C}$$

Similar to the pitch element, the pressure angles of a spur gear are only in a plane normal to the gear axis. In helical gears, pressure angles are defined in three planes. The **transverse** pressure angle is normal to the gear axis or parallel to the gear face. **Normal** pressure angle is in the plane or section which is normal or perpendicular to the helix. In the plane of the gear axis the pressure angle is termed **axial.** This plane is used mostly in reference to involute helicoids with very high helix angles such as worms or threads.

As at point *P* the pressure angle at any radius greater than the given base radius may be defined as

$$\cos\phi = \frac{r_b}{r}$$

The actual rolling or operating pressure angle of a pair of gears is chosen by the designer as the most practical for his application. Several things should be considered, among which is the strength of the resulting tooth and its ability to withstand the specified load. Another important item is the rate of profile sliding, as mentioned earlier. However, the majority of involute gears are in a **standard** use class which can be made using methods and tooth proportions which are well proven. Generally, involute gears roll at pressure angles ranging from 14½° to 30°. Standard spur gears for general use are usually made with 20° pressure angle. The **normal** pressure angle of standard helical gears ranges from 14½° to 18½° and sometimes 20°. The higher pressure angles (25°-30°) are generally used in gear pumps. In standard gears these pressure angles are generally (but not always) the operating angle between mates. Usually the given pressure angle is the same as derived from the normal base pitch and selected normal diametral pitch, or

$$\cos\phi_n = \frac{P_{bn}P_n}{\pi}$$

Diametral Pitch, Numbers of Teeth and Pitch Circles

The number of factors which control or are controlled by diametral pitch would probably confound the inexperienced gear designer. Among these are: strength required of the gear teeth, the number of teeth to provide a given ratio, and size of the pitch circles to satisfy center distance or space requirements. It becomes obvious that pitch, number of teeth and pitch diameters are dependent upon and regulate each other.

Load to be transmitted by gear teeth will most certainly dictate tooth thickness which is regulated by diametral pitch. Choice of a pitch to handle a given load is one of the more difficult tasks for the gear designer. The inexperienced will probably design more than one set of gears for a given load before finalizing the design with the proper power rating. Actually, there is no method of choosing a pitch in advance which will carry a given load.

Once the torque is established, tables are available to aid in selection of a diametral pitch. After the basic gear design is completed, there are standard equations to "rate" the gears with the maximum load carrying capacity, to be compared against the orginal torque. If they are then under- or over-designed, corrections must be made. Sometimes a complete new design is required. Tooth load is not the only criterion in choosing the diametral pitch and consequently the number of teeth. Very often the tooth load is not a critical factor at all and is replaced by problems of correct speed, ratio and center distance.

Quite often when the load is small, or not even an important factor, consideration should be given to using fine pitches such as 12 to 20, or even finer. When applicable, the fine pitch gear offers longer service, greater capacity, and better production control. This is because there are more teeth for a given pitch diameter with finer pitch. This means there will be more teeth in mesh at any given instant and less load on each individual tooth. Since the fine pitch gear has a relatively shorter active profile, there is less sliding between profiles than with coarser pitch gears. This reduces the possibility of fatigue failure.

The diametral pitch referred to is usually the pitch of the tool producing the gear teeth and is known as the generating pitch. The chosen pressure angle is considered at the diametral pitch, or more specifically, at the pitch diameter, *D*.

$$D = \frac{N}{P}$$

By definition, the pitch diameter is proportional to the diametral pitch and the number of teeth, and the base diameter is the product of the pitch diameter and cosine of the pressure angle, Fig. 1-14.

$$2r_b = D_b = D\cos\phi$$

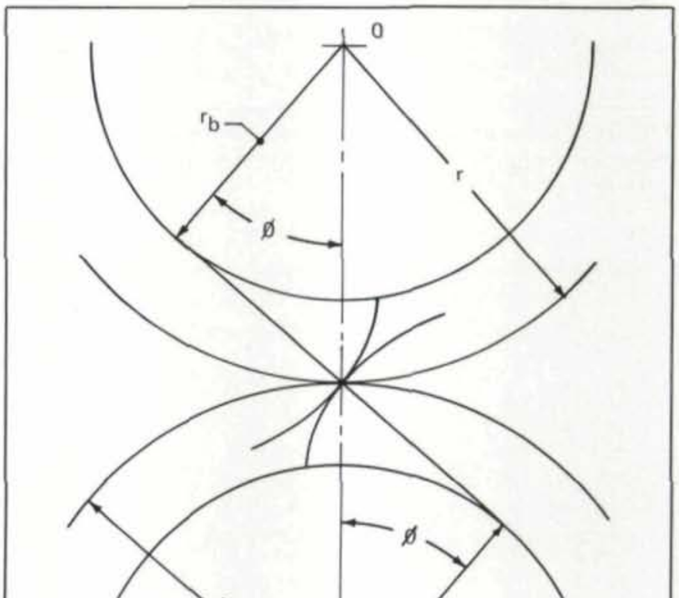


Fig. 1-14—*Base Diameter (D_b)* is twice the base radius (r_b) or the product of the pitch diameter (D) and the cosine of the pressure angle (ϕ).

In most cases a ratio controlling the amount of increase or decrease of rotational speeds is determined before the design of a gear set. The center distance $(r + r')$ is usually defined approximately by space limitations or may be given as a previously set dimension. Nominal rolling pitch circle diameters can be approximated through use of the given ratio and center distance. For example, if the desired ratio is 3 to 1, one pitch diameter will be three times larger than the other. The numbers of teeth to be used in the gears are products of the pitch diameter and the chosen diametral pitch.

$$N = PH$$

With the number of teeth defined, the next step is the design of the gear tooth and its mate.

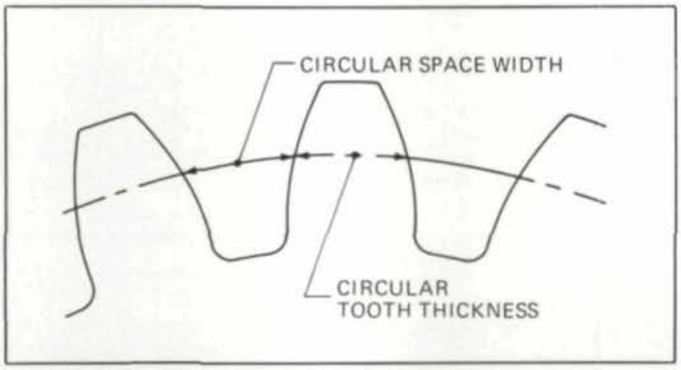


Fig. 1-15—Circular space width and circular tooth thickness defined. When dealing with these measurements, the diameter at which they are measured must be specified.

The Teeth

The elements which describe and regulate the gear tooth proportions are more or less standardized under the diametral pitch system. Most of the circular dimensions are shown in Fig. 1-8 with the radial dimensions shown in Fig. 1-11. In Fig. 1-15, the tooth and space thicknesses are illustrated. These dimensions may be given or calculated at any defined diameter.

Gear teeth having involute profiles are very versatile and adaptable to the many variations that may be required. A large percentage of involute gears have **standard** teeth.

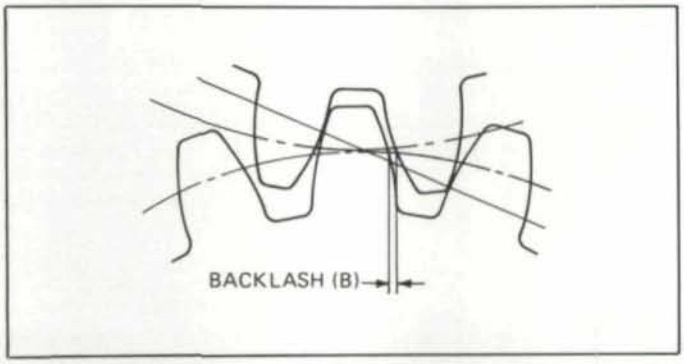


Fig. 1-16—Practical gear teeth usually are thinned to provide a certain amount of *Backlash* (*B*) to prevent tooth interference under the worst conditions of manufacturing tolerances and expansion due to temperature increase.

Standard toothed gears generally run in mating pairs on standard center distance, usually with the teeth *thinned* for backlash, Fig. 1-16. This backlash is essentially an angle, measurable in various ways and very necessary in any pair of mating gears. It must be sufficient to permit the gears to *turn freely under the worst conditions of manufacturing* tolerances and temperature variations. The amount of backlash used must be regulated and controlled within practical limits. It is possible to have too much backlash. This could be detrimental to the operation of the gear train. Basically, backlash is a factor in determining the final thickness of the gear tooth. In standard gears, the tooth thickness of one gear of a pair is determined by subtracting one half of the total desired backlash from one half of the circular pitch:

$$t = \frac{p}{2} - \frac{B}{2}$$

There are a few applications which call for no backlash. In these cases, the other dimensional elements must be held to extremely close tolerances. The required backlash depends upon where, when and how the gear set will be used. For example, in cases of tooth deflection due to heavy loading, or when extreme temperature variations are present, the amount of backlash required must be determined from experience. For general application, tables recommending backlash limits with reference to diametral pitch are available in AGMA standards and many other texts.

The whole depth of tooth, Fig. 1-11, must provide sufficient clearance for the tip of the mating tooth to swing through and make proper flank contact. Corners at the bottom, or root, of the tooth space are rounded rather than sharp. Depending on the method of manufacture, the rounding can be a true radius or a trochoid-type curve tangent to both flank and root of the tooth space. This rounding is usually expressed as **root corner radius** or more specifically as **root corner fillet.**

The expression $D + f$ is sometimes used when referring to the whole depth of gear teeth. The term has an early beginning and, for a considerable length of time, was the expression for whole depth of tooth. It was originally derived from the sum of working depth, D, and the desired root fillet, f, Fig. 1-17. Since the mating teeth make contact somewhat above the working depth circle, this method of obtaining the whole depth assured clearance between the mating tooth

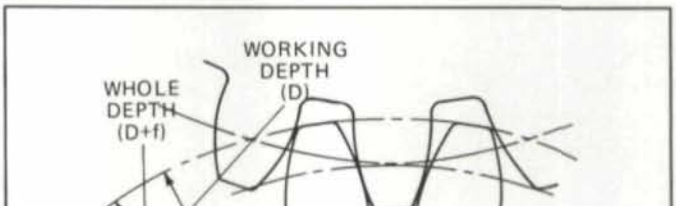


Fig. 1-17—*Whole Depth* $(D + f)$ is the sum of working depth and root fillet depth.

tip and the root fillet. The working depth is flexible and changeable within certain limits to be discussed later. With these two determinants, it was a simple matter to obtain a practical whole depth of tooth which allowed mating gears to rotate freely. The term $D + f$ is specified on most gear cutting tools to define the depth of tooth which will be produced by the tool when the tooth thickness of the gear is one-half the circular pitch.

Despite use of the term, $D + f$, whole depth remains addendum plus dedundum because the root fillet is part of the dedendum. However, with new innovations in manufacture and finishing of gear teeth, the establishment of whole depth has become more complicated in some instances.

Rack Teeth and the Basic Rack

An involute gear with an infinite pitch diameter will have straight profiles angled at the chosen pressure angle, Fig. 1-18. Such teeth are known as **rack teeth** and when several are put together in an elongated toothed member it is known as a **gear rack.** The rack tooth has the same properties as a gear tooth with the involute curve for its profile. Gear racks are driven by cylindrical involute gears and are used extensively throughout general industry.

The basic rack tooth is a special case of the involute rack tooth form. Generally, it defines the basis for a system or family of involute gears. The general tooth proportions of the basic rack show the standard design conditions of the system defined. Any involute gear of the basic rack system should be designed to roll freely with the basic rack, regardless of modification, so that it can also roll freely with any other gear of the same system, Fig. 1-19.

By definition, a tangent to any curve is perpendicular to the radius of curvature at the point of contact. Therefore, as shown in Fig. 1-20, the involute curve from a base circle is generated by tangents and radii of curvature. Accordingly, if the profile of the basic rack tooth is considered tangent to an involute curve from the base circle, it becomes the generator of an involute gear with a given number of teeth, Fig. 1-20. A cutting tool, with teeth in the form of the basic rack and used to produce gear teeth from a solid blank, is a hob, Fig. 1-21.

The generating action of a hob, as described previously, is unique in that as long as the base pitch (p_b) is maintained (see Fig. 1-19), any number of pitch and pressure angle combinations will produce the same gear tooth. However, the

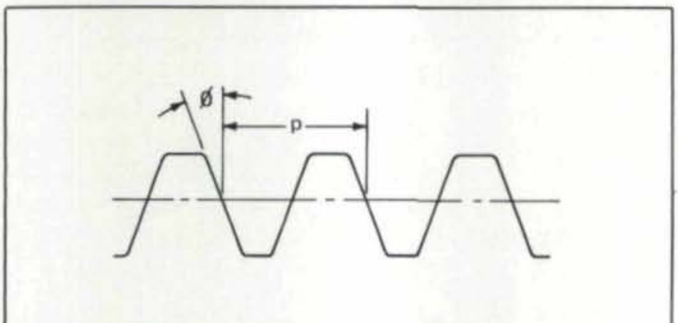


Fig. 1-18—An involute tooth form gear of infinite pitch diameter is called a *Rack*. The teeth have straight sides whose angle equals the chosen pressure angle.

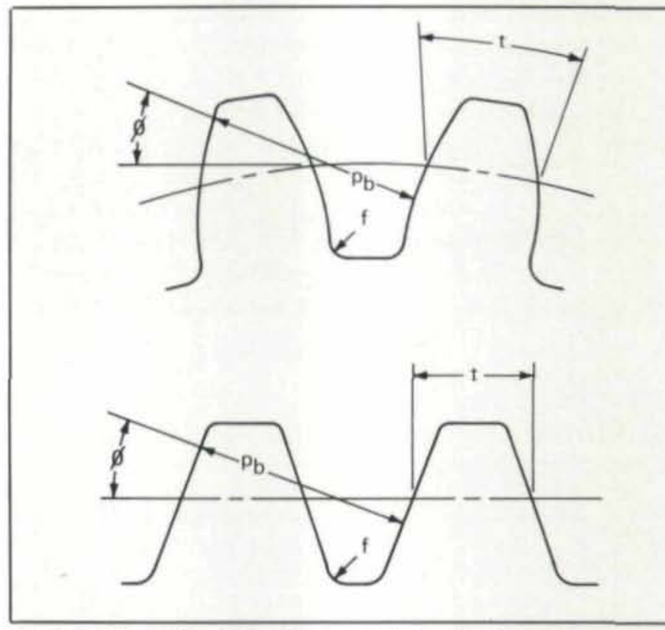


Fig. 1-19—Since a circular involute gear is designed to the same parameters as the basic rack, it will roll freely with the basic rack and with any other circular gear using the same design system.

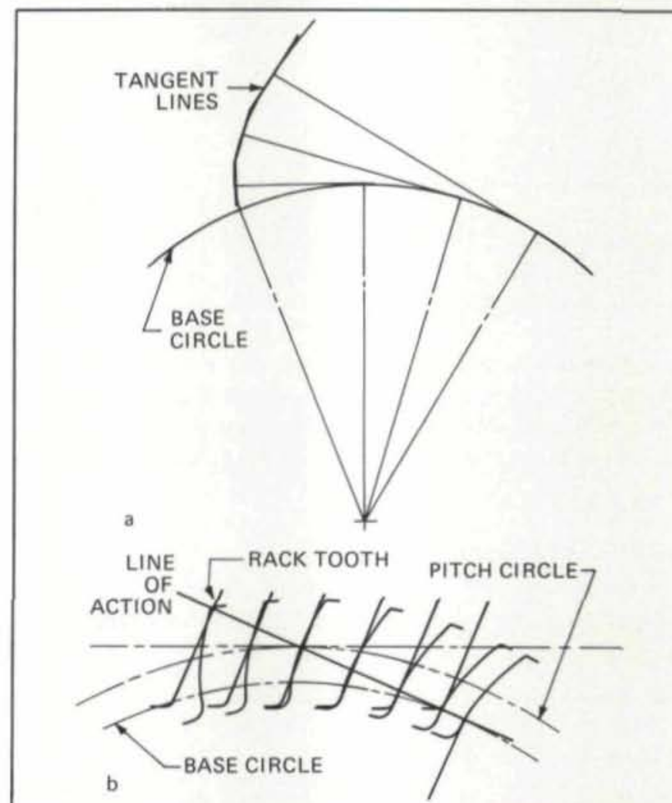


Fig. 1-20—An involute curve may be generated by a series of tangents, *a*. Therefore, if the profile of the basic rack tooth is considered to be tangent to an involute curve from a base circle, the rack becomes the generator of an involute gear with a given number of teeth, *b*.

large majority of hobs are **standard generating** with respect to the gears they produce. By this is meant that the gear **generating** pitch diameter (sometimes called **theoretical**) is obtained by dividing the number of gear teeth by the hob diametral pitch, and the pressure angle of the hob becomes the generating pressure angle of the gear.

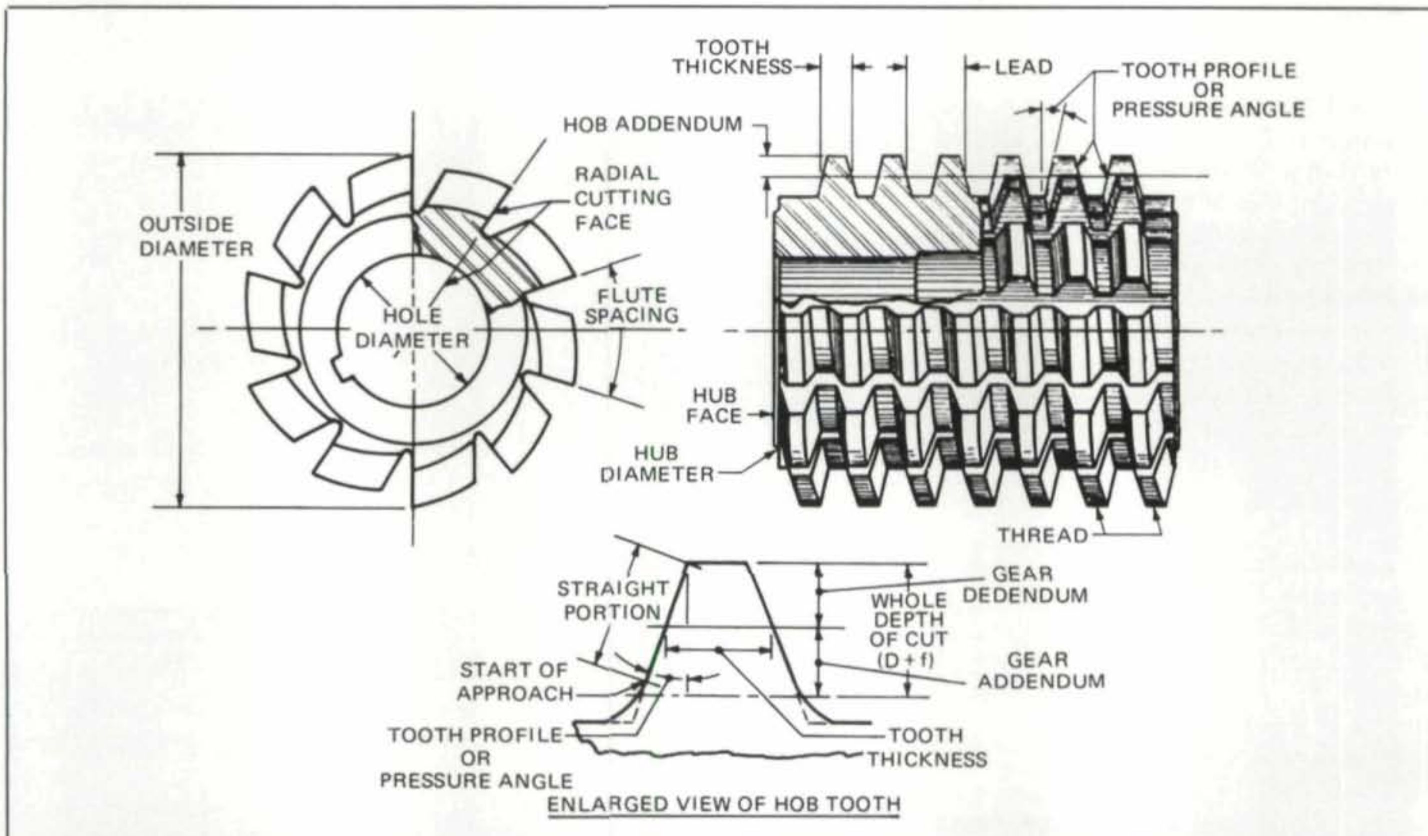


Fig. 1-21—A tool with teeth in basic rack form, which cuts gear teeth from a solid blank, is a hob. Details of a typical hob for production of involute tooth form gears are shown here.

Roots and Fillets

The gear root corner fillet is part of a trochoidal curve, generated by the corner of the hob tooth which is generally rounded. It is not practical to have a sharp corner on a hob tooth. First, the root fillet produced will be the very minimum and second, the corner will wear away—producing an inconsistent and unknown root fillet. In many cases, the root fillet produced by a standard tip corner radius of approximately 1/20 of normal circular pitch would suffice. Very often the load to be carried by the gear teeth will dictate the type and size of root fillet. Therefore, the gear designer should be aware of his control over the profile and root conditions to be produced in gears of his design.

One of the more common methods of root fillet control is the use of **short-lead** (sometimes called **short-pitch**) hobs. The term, **short-pitch**, describes and defines both tool and process. Linear lead and pitch of a hob (or rack) are one and the same as circular pitch of a cylindrical gear. Therefore, the reference **short-lead** indicates a smaller pitch. In order to maintain the base pitch, see Fig. 1-19, it is necessary to reduce the pressure angle accordingly. The difference in rolling diameters between standard generating and short lead is shown in Fig. 1-22.

The advantage of short-lead hobs is in the shape of the root corner fillets produced in the gear tooth space. A very important factor in gear tooth design is control of the root fillet tangent point in relation to the lower end of the active profile on the tooth flank. Since the tooth form is not altered, the short-lead hob allows the designer to relate root fillet magnitude with the whole depth and beam strength of the tooth. It is obvious that the closer to the center of the hob tip radius one moves the **generating** pressure angle, the closer the hob tip will come to reproducing itself, and the root fillet tangency approaches its lowest point of contact. However, there are limiting factors.

The hobbing tool designers contend that the lowest practical generating pressure angle for a hob is approximately 12 deg. for purposes of both manufacture and use. As the pressure angle of the hob is reduced, the tool has a tendency to undercut the gear tooth flank. This undercut can become excessive on smaller numbers of teeth and is a problem for the gear designer. He must decide the importance of the root fillet condition to his design.

It is possible to use hobs rolling at pressure angles greater than the theoretical pressure angle of the gear. Such hobs are known as **long-lead** and their effect on the root fillet shape is opposite that of the short-lead type. However, since conditions requiring the use of long-lead hobs are unusual, they are seldom used.

As higher performance is demanded of gears, the shape of the entire gear root space, not just the fillet alone, has become more and more important. Relative position of the gear root fillet with respect to the form diameter is a critical factor in the load carrying capacity of the gear and is also most important to the rolling conditions with its mate and with the finishing shaving cutter.*

*The comments here and in the next four paragraphs apply to other types of gear finishing tools as well as to shaving cutters.

As mentioned previously, the hobbed or shapercut root fillet is a generated curve in the trochoid family. Although often very close, it is not part of a true circle. Shaper cutters produce higher fillets than hobs for the same depth of tooth. Consequently, it is usually necessary to cut shaper-cut gears deeper than hobbed gears to maintain a good relationship between profile form and fillet diameters.

Root fillets should never extend above the form diameter. The shaving cutter must finish the gear profile for the desired length without contacting the gear root fillet, Fig. 1-23. Therefore, gear designers must regulate form diameter, root fillet shape, and root diameter of the tooth for ease of manufacture and satisfactory performance while allowing for the required load.

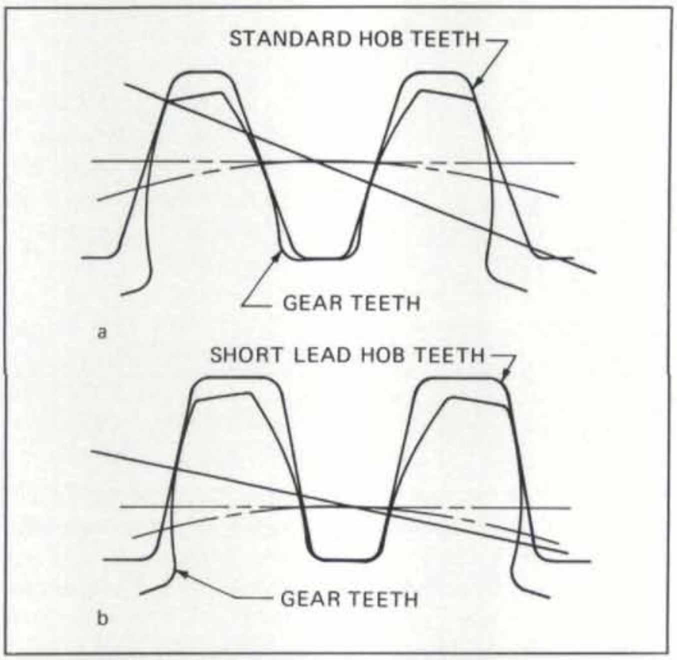
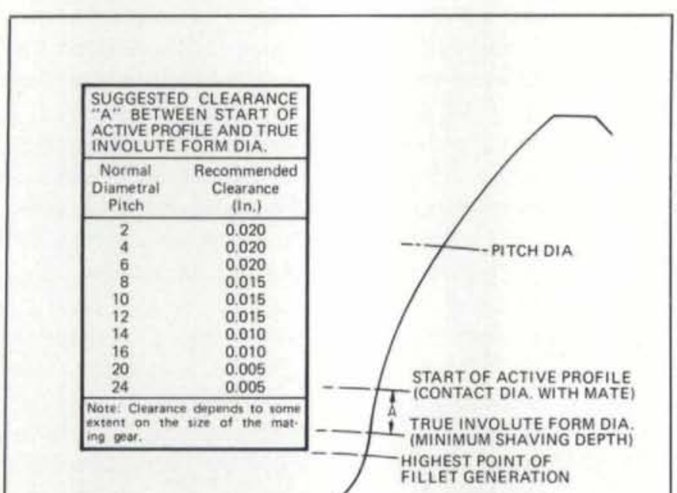


Fig. 1-22—The difference in rolling diameters for *Standard Generating, a,* and *Short Lead* hobs, *b.*



SUGGESTED CLEARANCE "A" BETWEEN START OF ACTIVE PROFILE AND TRUE INVOLUTE FORM DIA.

Normal Diametral Pitch	Recommended Clearance (In.)
2	0.020
4	0.020
6	0.020
8	0.015
10	0.015
12	0.015
14	0.010
16	0.010
20	0.005
24	0.005

Note: Clearance depends to some extent on the size of the mating gear.

Fig. 1-23—When gear teeth are to be finished by shaving, teeth must be cut to allow for this as shown.

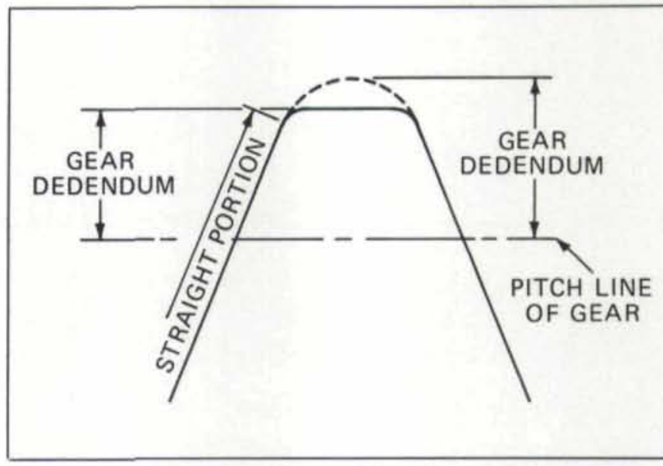


Fig. 1-24—Tooth form of preshaving hob. If full rounded root fillets are desired, additional whole depth becomes necessary.

The form diameter, as defined, is the lowest point on the gear tooth where the desired profile is to start. Often referred to as TIF (true involute form), it can also be represented in degrees roll or inches along the line of action. It is an important control point on the gear profile. Some establish the form diameter at the point of actual mating gear contact and others choose an arbitrary distance below the actual start of active profile. From a practical standpoint, the extension of the active profile should be determined by the extreme tolerances of center distance, mating gear outside diameter, size and runout.

In determining the actual whole depth and root fillet shape, the designer must consider the load which the tooth is to bear. These whole depths are based on the use of preshaving hobs and shaper cutters with a standard tip radius of 1/20 of the normal circular pitch.

If full rounded root fillets are desired, additional whole depth becomes necessary. It is advisable to maintain the same length of profile portions on the preshaving tool and determine the size of tip radius and extra depth from that point as indicated in Fig. 1-24. In this manner, the same amount of profile between form and fillet diameters is available for shaving cutter contact. When the designer must establish the fillet diameter at the maximum possible point because of load carrying demands, depending on the diametral pitch, a radial amount of 0.020/0.040 in. below the form diameter can be used safely.

During the shaving operation a small amount of material is removed from the gear tooth thickness. If the profile in the vicinity of the fillet diameter is not relieved in any manner, a step in the tooth flank will result, Fig. 1-25. This step, caused by the shaving cutter digging in, is detrimental to shaving action. It not only causes excessive wear of the shaving cutter teeth, but also affects the accuracy of the shaved profile. Thus, some amount of undercut must be provided to minimize the shaving cutter contact with the gear tooth flank.

The basic problem is to move the fillet and a short portion of tooth profile out of the path of the shaving cutter tip. In some cases with small numbers of teeth, a natural undercutting of the tooth flank occurs. Sometimes this will provide

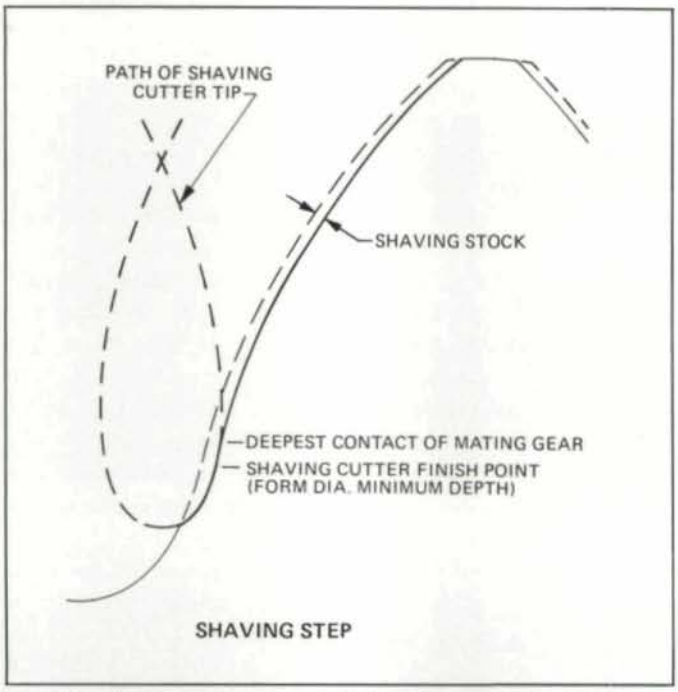


Fig. 1-25—A step in the gear tooth flank will result if the profile in the vicinity of the fillet diameter is not relieved in any way.

sufficient clearance. However, since natural undercut only occurs in specific cases, the necessary clearance must usually be provided by other means.

By putting a high point or protuberance on the flank of the preshaving tool tooth at the tip, a controlled undercut can be generated into the lower gear profile. The magnitude and shape of the undercut portion can be regulated by altering the amount of protuberance, tool tip radius, the total length of the protuberance section and/or the rolling pressure angle of the tool. Without control, profile undercut can be detrimental to the basic tooth form of the gear. If it is allowed to run up too high on the profile, it could cut away profile needed to maintain involute contact ratio with its mate. On the other hand, it might be so low that it cannot be reached by the shaving cutter and, therefore, serve no useful purpose.

Natural undercut, usually occurring in small pinions, can be controlled to some extent by changing the tool tip radius or by designing the pinion oversize from standard proportions. However, with the protuberance type tool full control of a profile undercut can be maintained.

Theoretically, the protuberance type tool should be designed for a specific gear and in accordance with the number of teeth. This becomes impractical when one desires to use the least number of tools for a range of gears with varying numbers of teeth.

Generally, the amount of undercut should be from 0.0005 to 0.0010 in. greater than the shaving stock being removed from each flank of the gear tooth. If the gear tooth flank is to be crown shaved, the depth of undercut should be increased by the amount of crown specified for each side of the tooth. The position of the undercut should be such that its upper margin meets the involute profile surface at a point below its form diameter. Sometimes, it is not possible to construct a preshave tool tooth from which will keep all generated undercut below this profile control point. In such cases, it is permissible to allow the tool to undercut the preshaved profile slightly above the form diameter, providing at least 0.0005 in. of stock is left for removal by the shaving cutter, see Fig. 1-26. However, any amount of profile removed by undercutting will reduce the involute overlap with the shaving cutter. For best control of gear tooth profile form, tooth contact ratio with the shaving cutter should never be less than 1.0 (preferably 1.2). Since the tip radius and high point of the preshave tool determine the fillet diameter and the height of undercut, it is sometimes possible to use slightly larger root fillets.

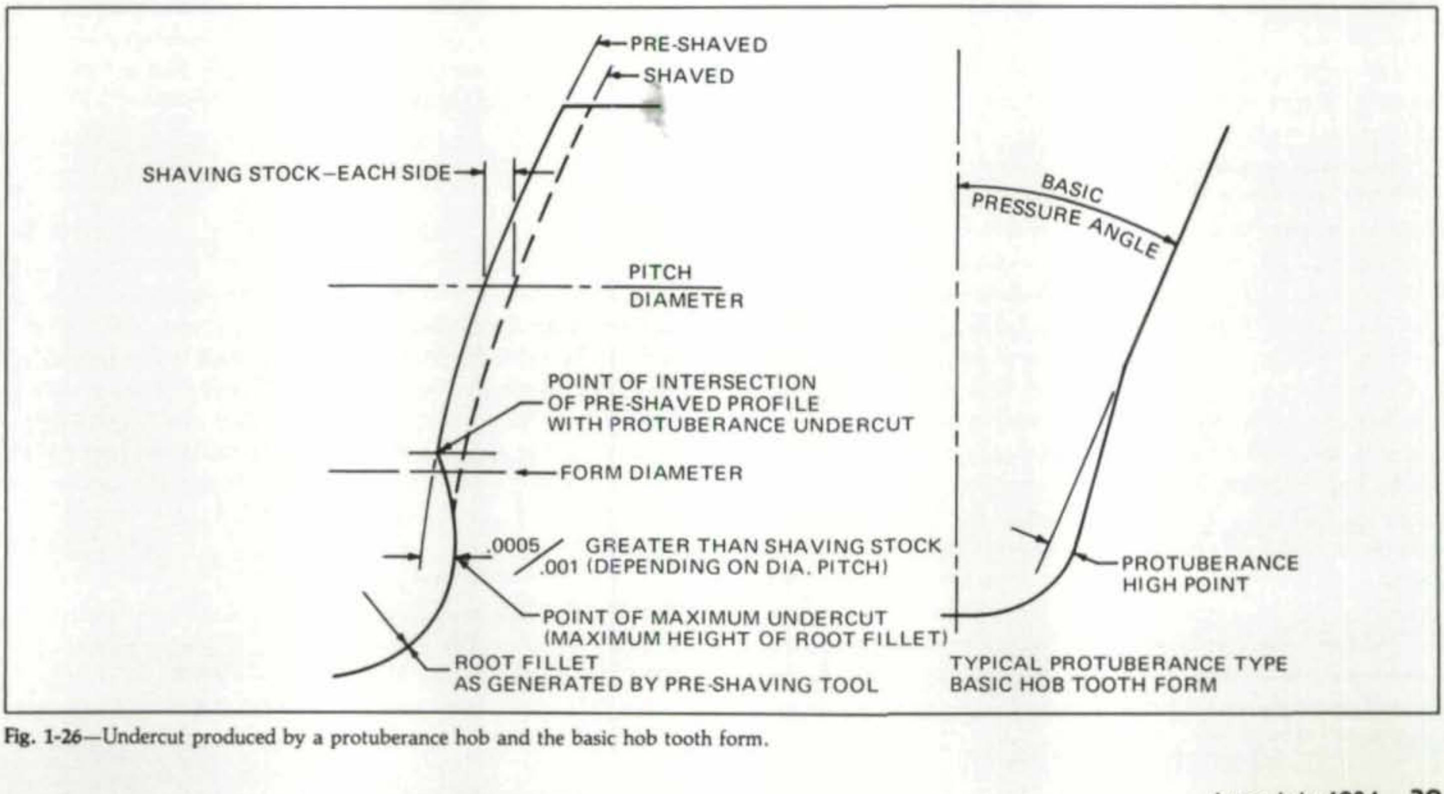


Fig. 1-26—Undercut produced by a protuberance hob and the basic hob tooth form.

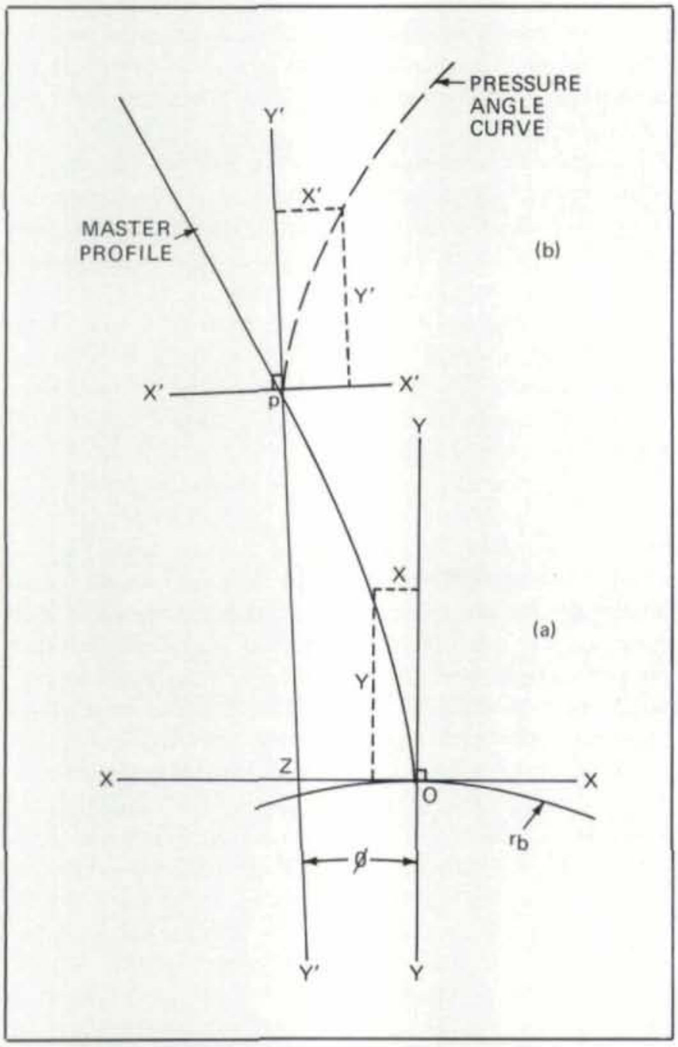


Fig. 1-27—Construction of, *a*, the master involute profile, and, *b*, the pressure angle curve.

When considering the root fillet area of the gear tooth space, it is advisable to simulate as closely as possible the conditions which will be prevalent in the gear. Some of the elements are readily calculable, such as: fillet diameter (or SRP) from a given preshave tool tip radius; the reverse calculation of a tool tip radius needed to produce a desired fillet height above a given root diameter, and the necessary increase in gear dedendum when changing from a corner radius to a full tip radius on the tool. However, proper analysis and study of the root fillet shape and capabilities can only be achieved by simulating the actual generated condition on the drawing board or in a computer.

Computer users can program the rolling action of hob or shaper cutter and receive results in approximately the same amount of time. However, by graphic or layout method, the shaper cutter presents more of a problem than the hob. Since the hob is the most prevalent of all preshaving tools, a graphic method of generating the root fillet area of gear teeth with or without the use of protuberance will be detailed.

These layouts can be made on an average size drafting board at scales of 50 to 150 times size. Scales of the layouts are limited only by normal diametral pitch and gear base diameter.

Generating Gear Root Fillets

First step is to make a master profile chart on some transparent material. The chart has a master involute profile curve which is laid out by means of rectangular co-ordinates. The base circle diameter must be large enough to produce desirable scales for the sizes of gears being used. The scale of the layout is determined by the ratio of the base circle diameters of the master profile and the work gear. Depending upon the scale desired, a single master involute profile could be used for a number of rolling layouts.

In addition to the master involute curve, the master profile chart has opposing involute curves radiating from the original involute curve. These curves represent the loci of the pitch point between the profiles of the gear teeth and the generating tools. Their number, position, and points of origin are determined by the various pressure angles of the tools to be rolled out.

The master profile chart also has a series of straight lines which are tangent to both convolutions of the involute curve. These lines represent the position of the side or profile of the generating hob or rack tooth cutter at the various degrees of pitch diameter roll.

The second step in the graphical method is to lay out the profile of the hob or rack tooth cutter on transparent material to the proper scale.

To lay out a fillet, a copy of the master profile chart is made on a white background. Then, the hob layout is laid over the master profile in various generating positions, and the fillet generated by pricking the master profile chart print. A line connecting these prick points on the print gives the true generated root fillet.

Master Profile Chart Layout

In the development of the initial master involute profile, Fig. 1-27a, it is best to choose a base radius which will provide a fair enlargement of the tooth profiles in general use. The X and Y axes are laid off on transparent material. The intersection point O is the origin of the involute curve lying on the circumference of the base circle radius, r_b, The general equations for the involute curve are:

$$Y = \cos\epsilon + \epsilon \sin\epsilon;\ X = \sin\epsilon - \epsilon\cos\epsilon$$

where ϵ is the roll angle for various points on the curve. By using a series of roll angles starting from 0 degrees, values for x and y (basic co-ordinates) may be tabulated. Tables of these rectangular co-ordinates, covering a wide range of roll angles, have been published. Subsequent co-ordinates of points on the curve are determined by the product of the chosen base radius and the values of the co-ordinates for the various roll angles:

$$Y' = r_b(\cos\epsilon' + \epsilon'\sin\epsilon' - 1)$$
$$X' = r_b(\sin\epsilon' - \epsilon'\cos\epsilon')$$

Sufficient points are calculated to form an accurate curve, which represents the involute profile of a gear tooth from the base radius.

In Fig. 1-27b, the origin of the X and Y axes for the pressure angle curve lies at point P on the master profile. To establish point P, the new Y axis is drawn through point Z on the original X axis at an angle θ from the original axis. The intersection of this new Y axis and the master profile curve is point P. Therefore,

$$\text{arc}\ \theta = \text{inv}\ \theta = \tan\theta - \text{arc}\ \theta$$
$$OZ = r_b \tan\theta$$

where θ is the pressure angle of the hob used in generating the gear tooth form. The new X axis is drawn perpendicular to the new Y axis through point P.

The radius r to the point P becomes the base radius of the pressure angle curve which is of involute form and opposed to the curve of the master profile. Radius r is determined through use of the master profile base radius and the chosen pressure angle:

$$r = \frac{r_b}{\cos\theta}$$

Substituting r for r_b in the equations for Y' and X' and using the same roll angles and basic co-ordinates as used for the master involute, the co-ordinates for the points on the pressure angle curve are calculated and plotted. Subsequent curves are established and plotted through the use of various pressure angles.

The straight lines shown in Fig. 1-28 are the control lines representing the straight side or profile of the hob tooth generating the gear tooth form. They are drawn tangent to the master involute profile and intersect all pressure angle curves. Their number depends only upon the desired accuracy of the gear fillet layout and their position relative to the original Y axis is of no consequence. Since the flank of a hob tooth is always tangent to the profile of the generated gear tooth, it is important that these straight lines are always tangent to the master involute profile.

The steps illustrated in Figs. 1-27 to 1-29 complete the involute profile chart. Actual generating layout work is done on prints of the chart, not on the original drawing. In this manner, gears of different diametral pitches and pressure angles may be generated on the same form. The scale of each layout is determined by the ratio of the base radius of the master involute form and the base radius of the gear to be generated:

$$s = \frac{r_{bl}}{r_{bG}}$$

where s is the layout scale, r_{bl} is the base radius of the master layout, and r_{bG} is the base radius of the gear. The scale is used to determine the dimensions of the hob profile which generates the gear tooth fillet and undercut. The height of the fillet and undercut is then reconverted to a radial distance above the gear base circle.

Hob Tooth Layout

The hob tooth form is laid out as shown in Fig. 1-29. This layout is made on transparent paper using the scale calculated from the equation of s. Construction lines KK, MM and NN are laid off first. The distance a is the radial dedendum of the gear tooth space from the pitch radius r to the desired or produced root radius r_r:

$$a = r - r_r$$

Point P' is the pitch point on the hob profile which coin-

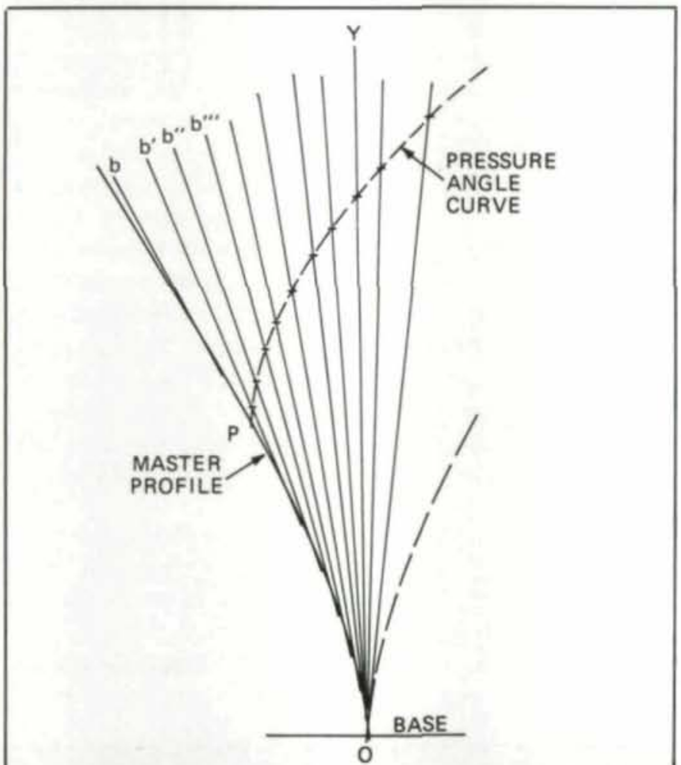


Fig. 1-28—Control lines representing positions of straight side of hob in generation of gear tooth form.

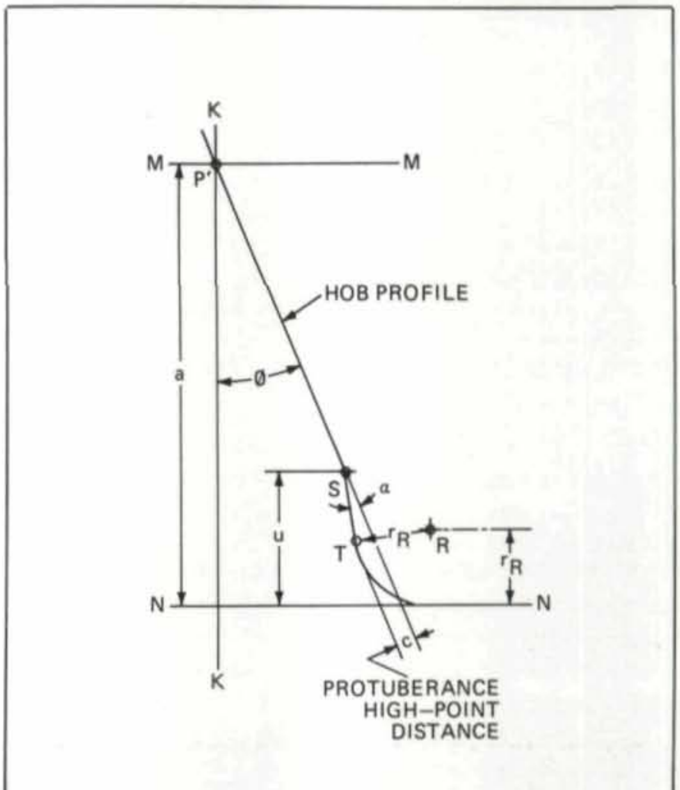


Fig. 1-29—Hob tooth profile.

cides with point P on the master profile. A line representing the main profile of the hob is laid off through P' at pressure angle ϕ to line KK and extending to the tip of the tooth on line NN. The amount of protuberance high point is measured-off parallel to the hob profile, and the desired tip radius of the hob is laid-in tangent to the distance c and the construction line NN. The approach to the protuberance is made tangent to tip radius r_R *at point* T and intersecting the hob profile at point S. The distance u to the start of approach is usually given on hob tool prints supplied by the vendors. However, 5 degrees is a good approximation of the angle a for the approach. Points P', S, T and R are encircled for future use. In order to regulate or control the amount and position of undercut produced, it may be necessary to change the tip radius, protuberane and points of intersection several times before completing a satisfactory tooth form.

When no protuberance formed under cut is desired, it is only necessary to make the tip radius tangent to the tip and profile of the hob. Then point T will lie on the hob profile and point S will cease to exist.

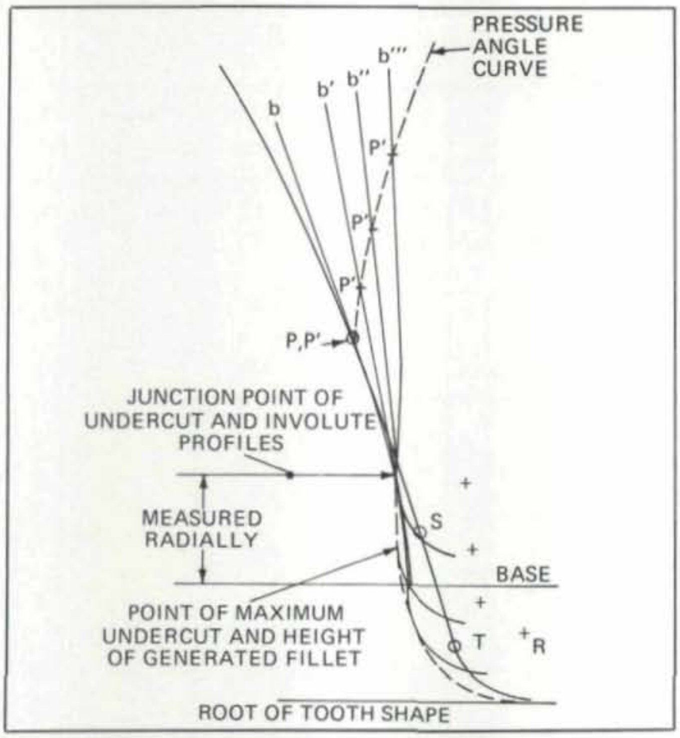


Fig. 1-30—Generation of root fillet by hob protuberance.

Using the Master Profile

For most efficient use, prints or duplications of the master profile chart should be of the type which show the lines of the layout on a white background. Then as shown in Fig. 1-30, the layout of the hob is laid over the master profile with point P' on the hob profile directly over P on the master profile and with the main profile of the hob coincident with the initial control line b which is tangent to the profile at point P. Using the point of a pricker, point P' is aligned with P as perfectly as possible and then points S, T and R on the hob form are punched through to the print below. A compass set to the scaled tip radius of the hob is centered at the transferred point R and the radius is inscribed through the transferred point T. Points S and T are joined by a straight line. In all cases, point S should fall on a control line coinciding with the main profile of the hob. Subsequent positions of the hob tip are determined by the same procedure, using the intersection points between the control lines and the pressure angel curve for locating P'. As P' proceeds along the pressure angle curve, the fillet and the undercut are formed to the root of the gear space by the hob tip radius, protuberance and approach.

Modifications of Standard Gear Tooth Forms

Quite often gears with teeth of standard proportions are either ill-suited or inadequate for the purpose intended. The versatility of the involute system makes it applicable in such cases. As long as a few basic rules are observed, the possible types of modified tooth forms are quite extensive.

Stud Tooth Gears

One of the most common types is the **Stub Tooth Form.** This tooth form differs from the standard type in tooth depth, Fig. 1-31. The shorter height makes a stronger tooth and minimizes undercut produced in small pinions. However, the length of contact between mating gears is shortened, which tends to offset the increase in tooth strength as well as raise the noise level of running gears.

The actual amount of reduction in tooth height depends upon the gear application and has definite limits. The length of the line of contact should never be less than one base pitch long. This, of course, is to maintain continuous action from tooth to tooth. Through experience, development and use, several standard stub tooth form systems have been established. Among these are: American Standard 20-Degree, Fellows Stub-Tooth and the Nuttall Stub-Tooth systems. These standard systems all have definite formulas for arriving at tooth proportions. They are very well defined in AGMA Standards, *Machinery's Handbook* and other texts.

A special case of stub tooth design is in its use as non-running involute spline teeth. Involute splines have maximum strength at the base; they can be accurately spaced and are self-centering which equalizes the bearing and stresses. *The teeth can be measured and fitted accurately.* Normally the tooth height is standardized at 50% of that based on the diametral pitch. For example, a 5 diametral pitch spline tooth would have a 10 pitch addendum and whole depth. Usually the pressure angle is 30 deg. However, this is not mandatory and quite often is changed to suit design conveniences. Involtue spline teeth may be either helical or spur.

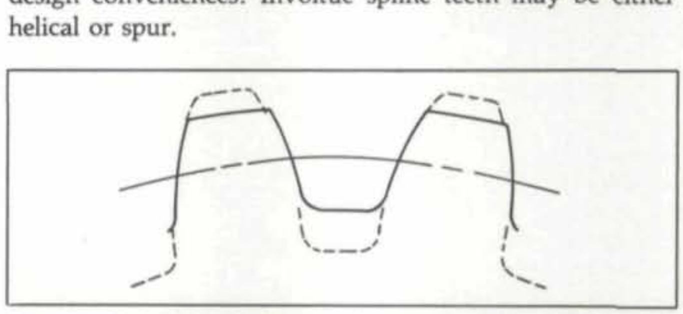


Fig. 1-31—*Stub Teeth* (solid line) are shorter than *Standard Teeth* (dashed line). They are stronger and minimize undercut in small pinions.

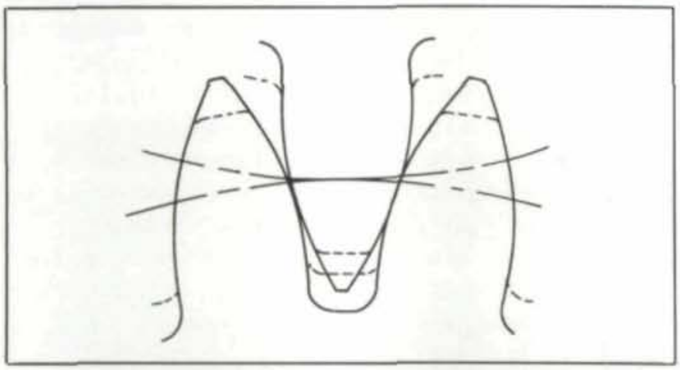


Fig. 1-32—*Extended Addendum Gears* (solid line) have longer period of contact to provide a smoother roll between adjacent mating teeth.

Extended Addendum Gears

Sometimes it is desirable to have the maximum possible length of contact between mating gears. The reasons for gear designs of this type are specialized and will vary with each application. However, when properly designed and accurately manufactured, it can be assumed that the longer period of contact will provide a smoother roll between adjacent mating teeth.

Normally, gears of this type are designed to roll on standard center distance with standard tooth thickness and backlash requirements. The exception to full standard tooth proportions is the extended addendum on both mating gears. This results in a longer radial working depth which requires lower root diameters, see Fig. 1-32. The length of the addendum is limited by the minimum allowable top land of the tooth and loss of beam strength due to the tooth length and possible undercut of the tooth flank.

Long and Short Addendum Gears

Occasionally a design will require a gear set wherein one member is considerably smaller than the other. If the tooth proportions are made standard, root fillet conditions produced in the small pinion and mating gear contacts may result in a poorly operating set of gears. As an example, Fig. 1-33 shows a pair of mating gears with the standard outside diameter of the larger member extended beyond the limit of the involute curved profile of the pinion. Since all profile action stops at the base circle, the mating gear addendum extending beyond the line of action represents a loss of contact between mates. Further, if the root of the pinion flank had not been excessively undercut, thus weakening the tooth, the mating gear tip would have found fillet metal as interference to its trochoidal sweep. The **Long and Short Addendum Method** was devised to alleviate this type of situation.

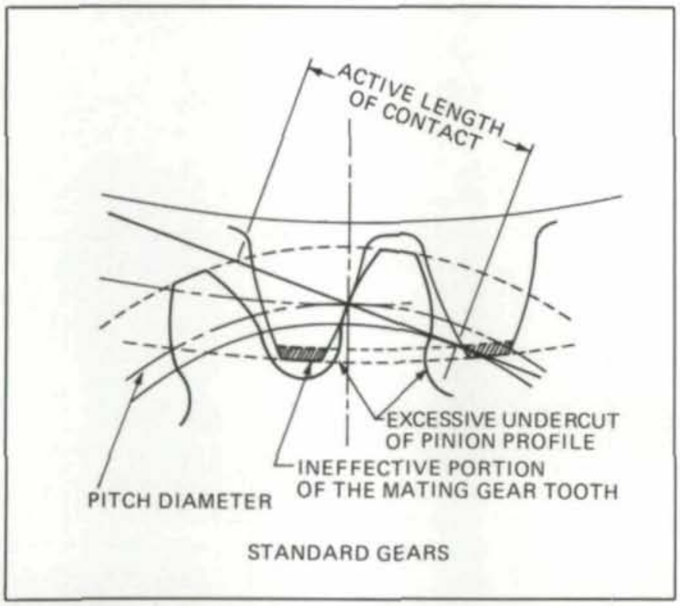


Fig. 1-33—A pair of mating gears with the standard outside diameter of the larger member extending below the limit of the involute profile of the pinion.

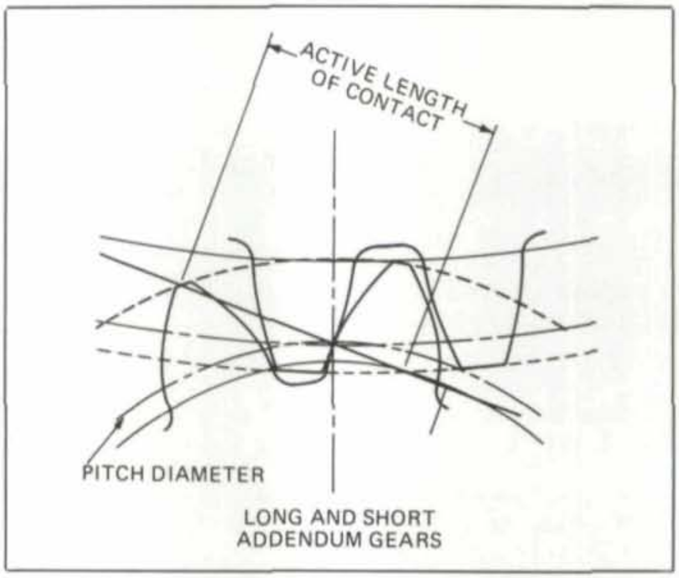


Fig. 1-34—*Long and Short Addendum Gears* minimize the problems where the pinion is very small compared to the mating gear.

Normally, the method is applied after a gear set is already designed as *standard* and the undesirable conditions have been discovered. The pinion outside diameter can be increased (long addendum), and the gear outside diameter decreased (short addendum) by an equal amount, so that the numbers of teeth, ratio, standard working depth and center distance remain unchanged, see Fig. 1-34.

Dependent on the pressure angle and the numbers of teeth, the amount of diameter revision can be varied. Limiting factors include the minimum acceptable top land on the pinion tooth, excessive profile sliding and the beam strength requirements for the gear. These design items must be checked after the preliminary tooth proportions have been established.

The actual amount of *profile shift*, as it is known, can be determined in various ways, including an outright guess. However, there are more exact methods. The difference between this new outside diameter and the original standard design divided by two is the *profile shift*. An approximate maximum shift would be one-half of standard addendum for the given diametral pitch. In any event, changing the diameters by two times the profile shift results in an oversize (long addendum) pinion and an undersize (short addendum) gear compared with the original standard units. Consequently, it becomes necessary to revise the tooth thicknesses.

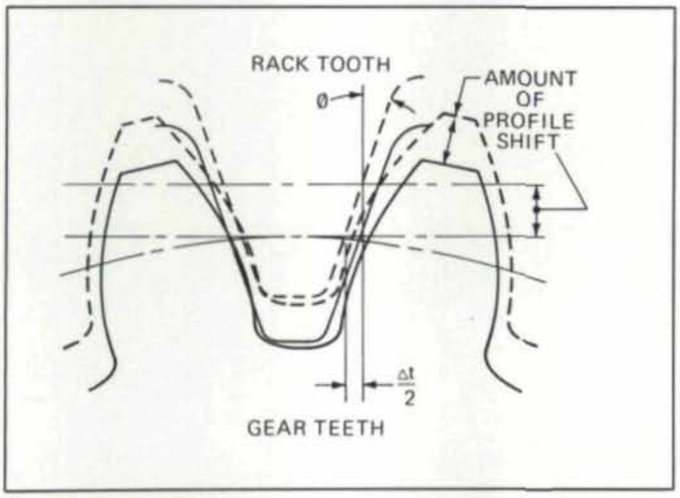


Fig. 1-35—Long addendum pinion showing profile shift.

Consider the basic rack tooth and its relationship to the gear tooth space. When the rack (hob) tooth is fed radially in towards the gear center a thinner gear tooth results. If the rack tooth is pulled out away from the center, the gear tooth becomes thicker, Fig. 1-35.

Therefore,

$$\frac{\Delta t}{2} = \text{Profile Shift} \times \text{Tangent } \phi$$

$$\text{Pinion tooth thickness} = t_p = \frac{p}{2} + (\Delta t) - \frac{B}{2}$$

$$\text{Gear tooth thickness} = t_G = \frac{p}{2} - (\Delta t) - \frac{B}{2}$$

These calculations can be used for both spur and helical gears. In helical gears, the dimensions used are normal to the helix angle at the generating pitch diameter.

The preceding equations and discussions are based on maintaining standard center distance, where the generating pitch diameters are the actual rolling pitch diameters. Therefore, the calculations for the tooth thicknesses are both simple and exact.

Non-Standard Center Distance Gears

Sometimes it is necessary to operate a pair of gears on center distances other than standard. Although the involute gear system lends itself readily to either an increased or decreased center distance, it is usually more expedient to consider a center distance greater than standard for a more efficient gear set. From the previous discussion, it is obvious that to operate on a spread center distance, the gear teeth themselves can no longer be standard and must be specially computed. The calculations involved are a little more tedious than those for long and short addendum gears, but they are not difficult.

Equations are also available for obtaining the outside diameters, size control and other dimensions necessary to complete the gear set design.

Over- and under-sized gear designs of this type are quite common. In the speed reducer and automotive transmission fields, it is much more expedient to change the design of the gear teeth for a ratio change than to incur the larger tooling cost for changing the gear housing center distance.

This ability, to modify physically the shape of the teeth and still have an efficiently operating gear set, is one of the tremendous assets of the involute gear system.

Internal Gear Teeth

Until now, the discussion has dealt exclusively with external-type gears where the teeth protrude outwardly from a cylindrical body and have convexly curved profiles. *The counterpart to the external is the internal gear where the* teeth protrude inwardly from the inside diameter of a ring, Fig. 1-36. Similar to external-type gears, internal gears can heave almost any tooth shape as long as it will roll continuously with or conform to the mating tooth profile. Internal gears are largely used in reduction gear trains and as spline teeth in non-running units.

Involute internal gear teeth have concave curved profiles. Theoretically, an external and internal gear with the same number of teeth and tooth proportions would conform exactly, except for root clearances. Internal gears, with involute curve profiles, have the same basic fundamentals as the externals. In some of the equations, it will be necessary to subtract instead of adding. However, the same forms and modifications used with external gears may be used with internal gears.

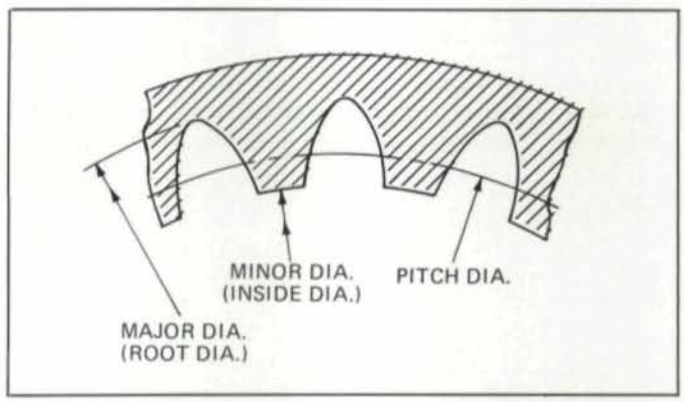


Fig. 1-36—Internal gear teeth protrude inwardly from the inside diameter of a ring.

Modification of Tooth Profile and Lead

Quite often errors in manufacture, deflections of mountings, deflections of teeth under load, and distortion of materials in heat treatment all combine to prevent the attainment of true involute contact between mating teeth. Besides contributing to objectionable noise, these undesirable meshing conditions are inefficient and lead to premature failure of the teeth. Some of these factors are illustrated in Fig. 1-37.

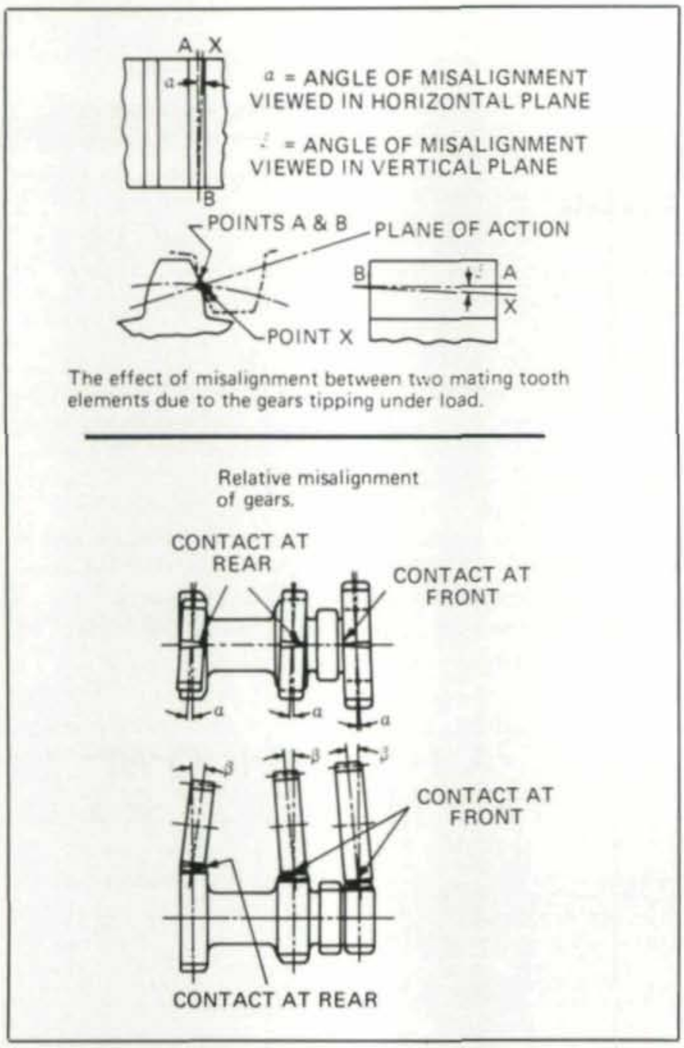


Fig. 1-37—Certain undesirable meshing conditions lead to premature failure of gear teeth. Some of these are shown here.

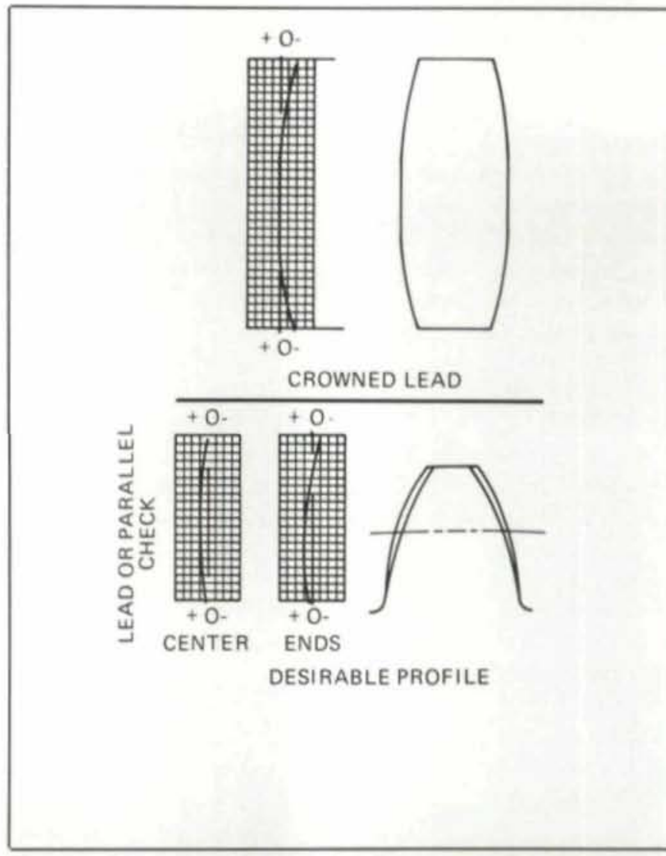


Fig. 1-38—Crowned gear teeth minimize the undesirable effects of departures from theoretical tooth form.

To alleviate and minimize the effect of these errors, the profiles and leads of the gear teeth are modified. The modifications are departures from the true theoretical form and designed to offset the undesirable contacts caused by the original errors, Fig. 1-38.

Normally the errors are discovered after design and manufacture of the prototype or initial lot of gears. Modifications are then developed to suit the conditions. Sometimes the designer may be experienced enough to predict, at least approximately, the distortions to take place and, from his experience, be able to order initial modifications.

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VIEWPOINT

The staff of GEAR TECHNOLOGY would like to thank you for the warm and enthusiastic response we have received, both by phone and letter, regarding our new publication. Excerpts from some of the letters received have been included in our first column. For future editions, we welcome your thoughts and comments as a regular part of our magazine. Let us here from you.

Dear Editor:

We share your interest in promoting the existence of a periodical dealing with the subject of gear manufacture. It seems that the scope of the new magazine fits our objectives for process and product exposure.

Lew Wallace
Gleason Machine Division

Your new magazine GEAR TECHNOLOGY certainly does sound exciting. All I can say is: "It's about time!" Best of luck in your adventure.

Joseph W. Coniglio
Vice President — Engineering
Gould & Eberhardt Gear Machinery Corp.

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.

CNC CONTROLLED CBN . . .
(Continued from page 28)

Accuracy

Profile Tolerance	.010 mm (.0004")
Lead Tolerance	.008 mm (.0003")
Pitch Variation +/−	.007 mm (+/− .0003")
Surface Finish	.0005 mm, 20 CLA

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24.4 mm (.960") whole depth
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Rc 55-60 hardness

Machine Data
Pfauter P1000 FSNC External Gear Grinder

Grinding Data
Pieces per load — 1
Stock removal .15 mm, (.006")
Number of passes — 1
650 mm (25.6") per minute feed rate

Total Grinding Time
38.8 minutes per piece

AUTHOR:

DENNIS GIMPERT is one of American Pfauter's two Sales Managers. Mr. Gimpert was educated at Michigan Technological University and Rockford College. He worked for Clark Equipment Corp. from 1971-1974, and for Barber-Colman Co. from 1974-1976 as an application engineer for gear hobbing and shaping equipment. In 1976, he joined American Pfauter. Mr. Gimpert has served on the Vehicular Gear Committee of American Gear Manufacturers Association and is active in various activities of the Society of Manufacturing Engineers.

. . . AND FROM THE INDUSTRY

What's new . . . as a service to our readers, GEAR TECHNOLOGY will offer space to announce personnel and corporate changes that would be of interest to the Gear Industry. If you have an announcement you wold like to make, please send it to **GEAR TECHNOLOGY,** P. O. Box 1426, Elk Grove, IL 60007.

BOURN & KOCH
William (Bill) Beal has joined **Bourn & Koch Machine Tool Co.,** Rockford, Illinois as manager of Hobber Remanufacture. Beal brings 36 years of gear manufacturing experience to Bourn and Kock, a rebuilder and retrofitter. Beal was formerly responsible for rebuilding at the Barber-Colman Co.

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ADVANTAGES OF TITANIUM NITRIDE . . .
(Continued from page 23)

utilized to apply titanium nitride over an initially applied titanium carbide surface in order to provide improved frictional characteristics.

Among the currently anticipated gear tool coating improvements, the following appear to be the most promising:

1. Certain developments are being pursued in titanium nitride coatings with the PVD process to provide reduced processing costs.

2. Titanium Carbo-nitride (TiCN) is most likely the next high speed steel coating which will be developed to provide higher hardness and improved abrasion resistance. The major disadvantage of Titanium Carbo-nitride is its brittleness.

3. Titanium Carbon Oxynitride (TiCON) is generating interest in certain quarters, but its total characteristics and advantages are not well known at this time.

4. Hafnium Nitride (HfN) and Zirconium Nitride (ZN) are candidates for future test work. Some basic testing has been done to date indicating that Hafnium Nitride is not as hard as Titanium Nitride, but has higher thermal stability.

5. Titanium Boride (TiB) has a serious disadvantage due to the extreme toxicity of gaseous boron which is used during the coating process. Therefore, not much work has been done with this possible coating.

6. Silicon Carbide (SiC), Silicon Nitride (SiN) and Tungsten Carbide (WC) are additional possibilities for future coatings applicable to high speed steel, but currently not much work is being done.

7. There are also possibilities of obtaining enhanced characteristics from combinations of the above coatings through the use of multi-layer coatings. This technique is used extensively in coating lenses in the optical industry. However, the boundaries between layers offer potential adhesion problems which would have to be solved before the process would be completely applicable to HSS cutting tools.

Until any of the above possibilities of improved coatings are proven realities, the most advantageous tool cost and gear cutting productivity improvements available will be through application of TiN coated tools.

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AUTHOR:

P. W. KELLY is Gear Tool Products Manager for the Barber-Colman Co. His responsibilities include marketing, design and application engineering and pricing of Gear Tool Products comprised of gear hobs, gear shaper cutters and specialized milling cutters. Mr. Kelly earned his BSME from Illinois Institute of Technology and MSME from the University of Wisconsin. Since 1975, he has held Engineering and Managerial positions at Barber-Colman. Currently he is a member of Society of Manufacturing Engineers, American Society of Manufacturing Engineers, American Engineering Society, and the American Gear Manufacturers Association. From 1973-1983, Mr. Kelly served as Chairman of the Gear Manufacturing Committee for the AGMA.

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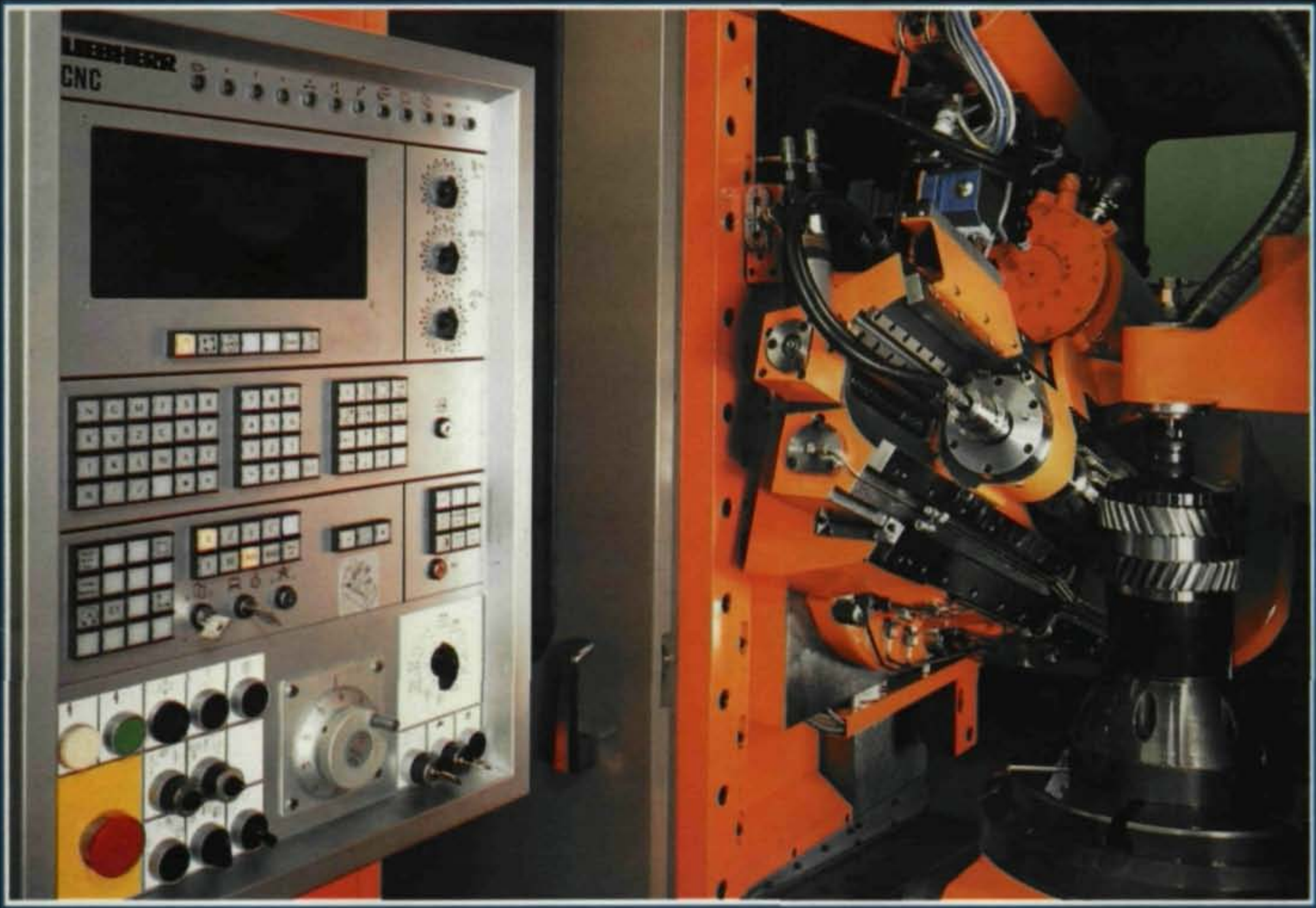
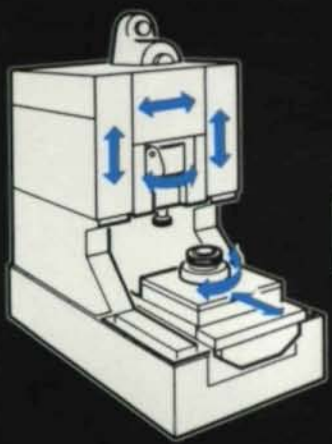


6-AXIS CNC
GEAR SHAPER



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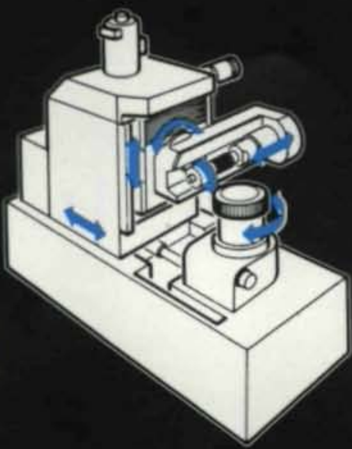

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