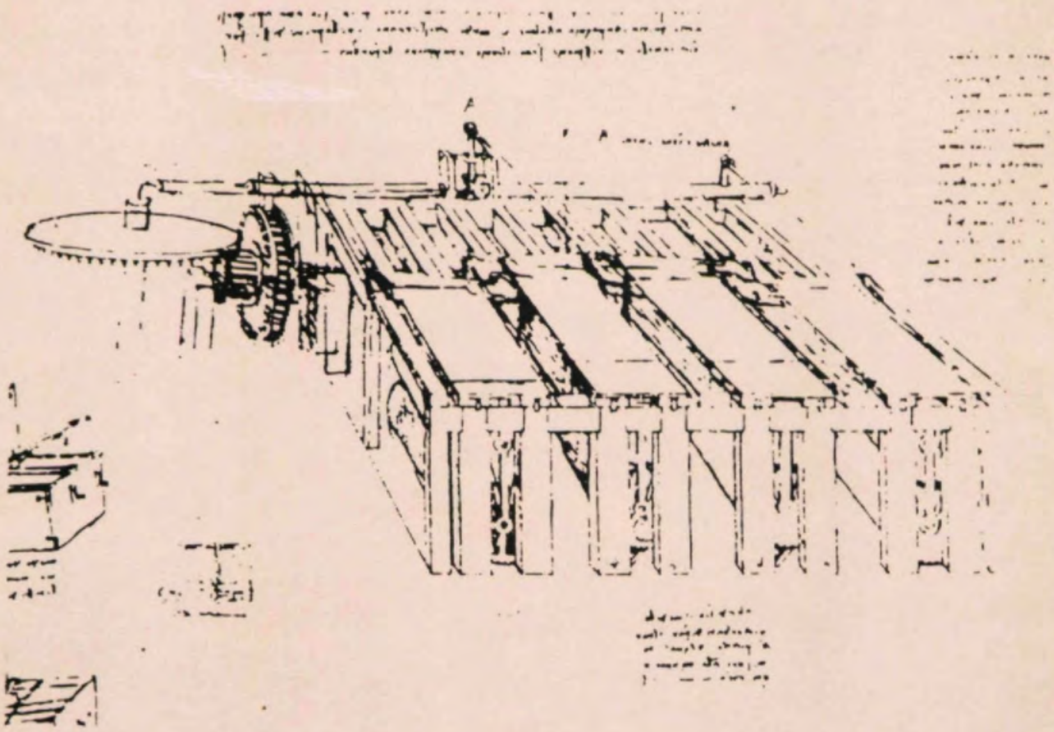
GEAR TECHNOLOGY

The Journal of Gear Manufacturing

NOVEMBER/DECEMBER 1987



Computer Aided Design for Gear Shaper Cutters

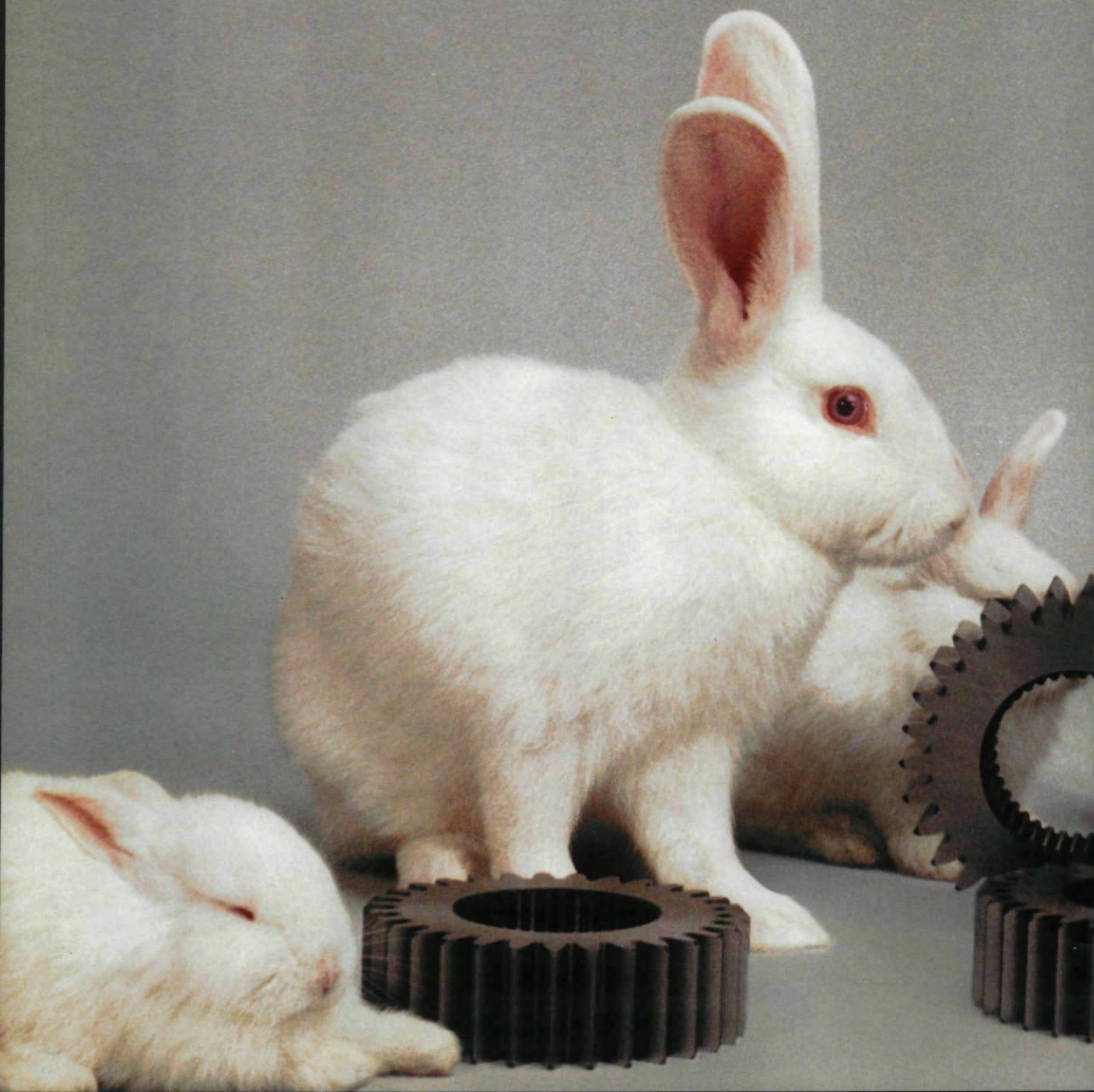
KHV Planetary Gearing

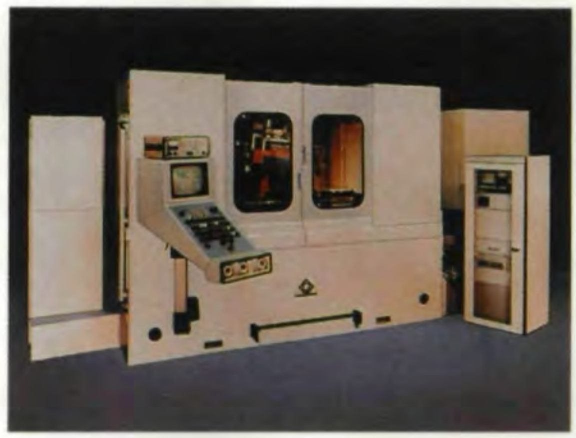
Finishing Gears by Ausforming

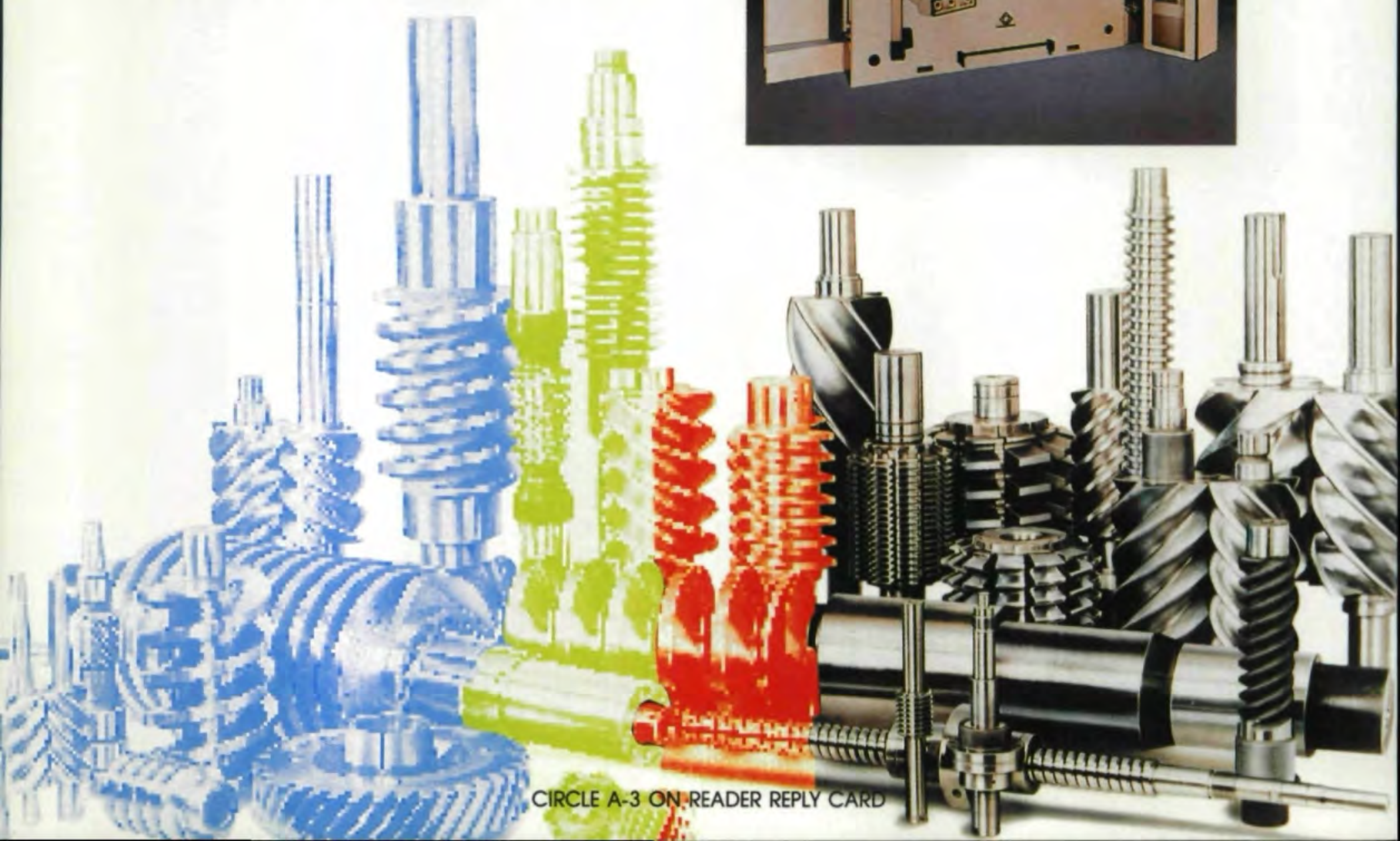
Selection of Hobbing Data

Good Gears Start with Good Blanks

Kapp makes fast work of high







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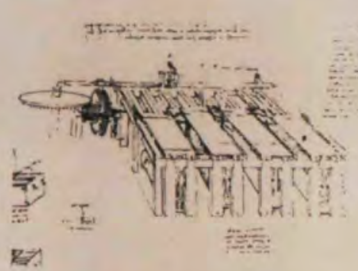

The Advanced Technology
of
Leonardo Da Vinci
1452-1519

COVER

Leonardo designed a great many machines for use in the textile industry. The design on our cover is a shearing machine for removing the nap from woolen cloth, a job previously done by hand with enormous scissors. The cloth was stretched on a travelling frame which was pulled through the machine. A set of gear works activated the second blades of the scissors devices. Leonardo also designed a gig mill for raising woolen nap that was so effective that when one of its successors was incorporated into textile machines in the early 19th century, widespread unemployment and rioting among textile workers resulted.

FEATURES

DEPARTMENTS

November/December, 1987 Vol. 4, No. 6

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

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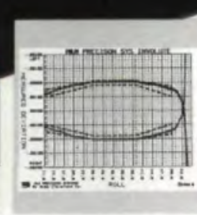

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EDITORIAL

DON'T SELL US SHORT!

How is it that we woke up one day in the early 1980s to find that apparently American industry was suddenly inefficient, our workforce unproductive and our management inept? Almost overnight industry found its sales dropping dramatically, while for many companies foreign competition became excruciatingly intense. This sudden change in the economic climate proved fatal for many companies and has been nearly as hard on our collective morale. In a country used to winning, we began to hear ourselves talked of as losers.

Unfortunately, this grim diagnosis was right in some cases. The tough times of the early 1980s were, perhaps, a necessary, if cruel, awakening for American business. But these first assessments were only partial truths, and the post mortems, premature. As with Mark Twain's obituary, the reports of the death of American manufacturing have been greatly exaggerated.

While we had, perhaps, been careless and allowed ourselves to be blindsided by foreign competition, not all our troubles could be laid at the door of a soft, aging economy that had lost its edge. Industry was in a bind that no amount of competitive edge or effort could entirely overcome.

The early 1980s was a time that the American dollar and interest rates rose to their highest levels in many years. Prices for goods manufactured for export suddenly increased in terms of foreign currencies, while dollar prices of foreign products dropped, cutting off markets and making American products uncompetitive through no fault of our manufacturers. While some American companies failed because of competitive softness or bad management, most others were victims of economic circumstances over which they had no control. Even the most astute businessperson cannot overcome these kinds of economic conditions, or, for that matter, a trade or fiscal policy on the part of the government that undercuts his or her best efforts.

But American industry has always responded well under pressure. Historically, it has been pragmatic and flexible. It has learned from its own mistakes and has taken advantage of good ideas from everywhere. Its attitude in the early 1980s was no exception. In spite of the foreign exchange disadvantages, a great many companies took the steps necessary to survive. They invested in updated machines, processes and practices. They designed better, more saleable products, and they developed a savvier approach to their role in the global economy. They took some of their competitors' best ideas and used them to their own advantage. At the same time, businesspeople and, hopefully, their elected representatives, became more aware of the need for sensible government trade and economic policies.

All these hard-learned lessons are beginning to have an effect. The recent "weaker dollar" is actually a more "competitive dollar," giving American business a fairer shot at global markets. As a result, more and more foreign companies are finding it profitable to buy from American suppliers. American companies are again becoming competitive in terms of price, quantity and delivery.

Just as important, American quality is again becoming a standard by which products can be judged. Some of our toughest competitors from overseas now buy parts from us, not because they're cheap or because it's politically advantageous to do so, but because our products are the best they can get. American ingenuity, resourcefulness and flexibility are again making a difference in our competitive stance.

Of course, things will never be the same as they were two decades ago when American products dominated every market. That era is over, probably never to return. We as a business community have had some serious problems, not all of which are yet solved, and all our troubles are not over. Because of our size, response time to changes can seem painfully slow, and current global economic realities will demand every bit of technical, manufacturing and business expertise we can muster. This country is unmatched in terms of human and physical resources and market size. Historically, we have always been able to summon the will and the means to get any job done that we really wanted done. Times and economic conditions have changed, but American resourcefulness has not. In evaluating winners and losers in the economic arena over the next few years, we should not sell ourselves short.

Michael Goldstein

Michael Goldstein
Editor/Publisher

VIEWPOINT

Dear Editor:

Thank you for the very good magazine on gears. It is very informational, contains a wealth of trade information, but still maintains a high level of papers, unfortunately not typical for other trade publications.

Eugene I. Rivin
Professor
Wayne State University

Questions From the Industry

Dear Sirs:

We are interested in purchasing a computer program which would enable us to undertake the following:

1. Plot involute gear tooth profile forms.
2. Indicate the interference patterns generated with pinion cutters.
3. Show the action of undercutting for varying numbers of teeth; all produced from mathematical input data.

We shall be pleased to learn if you can be of help to us in this respect.

Llewellin's Machine Co. Ltd.
M.S. Horlick
15 Kings Square
Bristol BS2 8JJ
England
(0272) 424026/9

Gentlemen:

We have a small job shop type gear company. We have recently begun pursuing a computer system to help us in our gear calculations. Due to the time involved in figuring formulas pertaining to various gears on paper, we are looking for programs to use on a computer system.

We are asking if you know of any computer program that may be available pertaining to gear formulas.

Also, we have a Gould & Eberhardt 48H Gear Hobber and we are interested in a computer program for changing gear calculations for hobbing helical gears non-differentially. This process is often time consuming to do on paper, and we think that somewhere, someone has a program to simplify this process.

Any help or suggestions you can offer would be greatly appreciated.

Bethlehem Gear & Machine Company
Robert A. Schrum
P.O. Box 157
Moundsville, WV 26041
(304) 845-9050

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

Gear Couplings

Re the article of Mr. Stan Jakuba: "Give your Gears a Break—Select the Right Coupling!" May/June, '87 issue and the contributions in July/August, '87 issue by Mr. Michael M. Calistrat and Mr. Stan Jakuba.

The following comments of experts in the field of torsional vibration calculation (T.V.C.) will prove that the selection of the right torsionally soft coupling (right elasticity, spring characteristic and damping) is not "black magic" anymore, but an easy task for an engineer having a highly sophisticated computer program and the right input data for all components of the transmission. The right coupling for a geared diesel generator set can be selected in a few minutes. Provided the governor of the engine is stable in the system, the coupling will last for ten or more years. A fluctuating torque monitoring device as developed by the signer can be helpful, especially for larger sized transmissions, as shown below even for rather complicated systems with several branches. The signer has done the T.V.C. for 13 US Navy support vessels with high powered diesel geared installations (300,000. HP total).

As a practical example, Fig. 1 shows a main and auxiliary drive for a large sized shuttle tanker (125,000. tdw). The following components are used. The main diesel two-stroke, slow-speed engine (MAN - B&W, 5 cylinder, 6526 kW at 102 rev) is driving a propeller through a reduction gear, which also includes a PTO (Power Take Off) for a hydro power package (5200 kW at 1200 rev) for discharging the crude oil. Between the main engine and the reduction gear a very soft coupling is arranged. On the forward end of the engine a generator drive (1200 kW at 1800 rev) via a speed increasing gear is attached, and a turbine driven by the exhaust gas (550 kW at 1800 rev) is incorporated. The generator is runnng at constant speed even at variable engine speed using a RCF (Renk Constant Frequency) gearbox. For eight different modes of operation for this vessel, an optimized coupling has to be

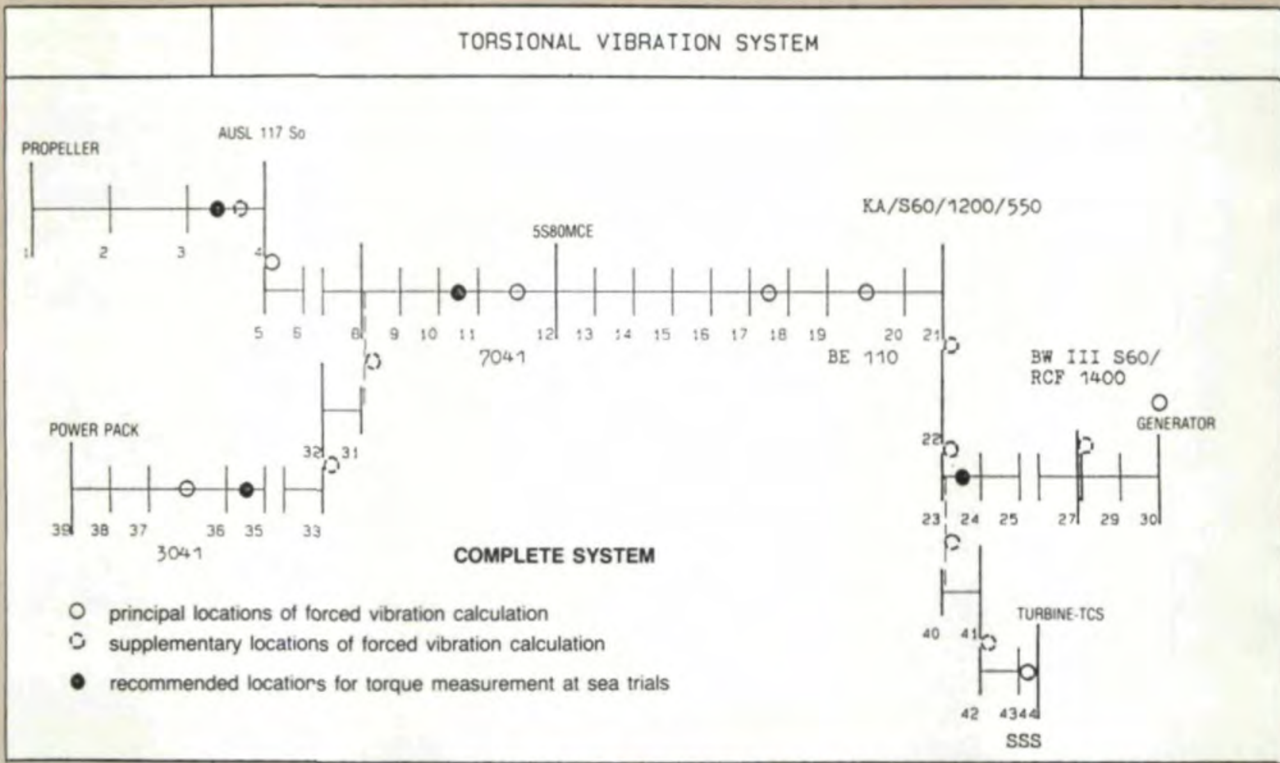


Fig. 1 – Scheme of the torsional vibration system for a crude oil shuttle tanker.

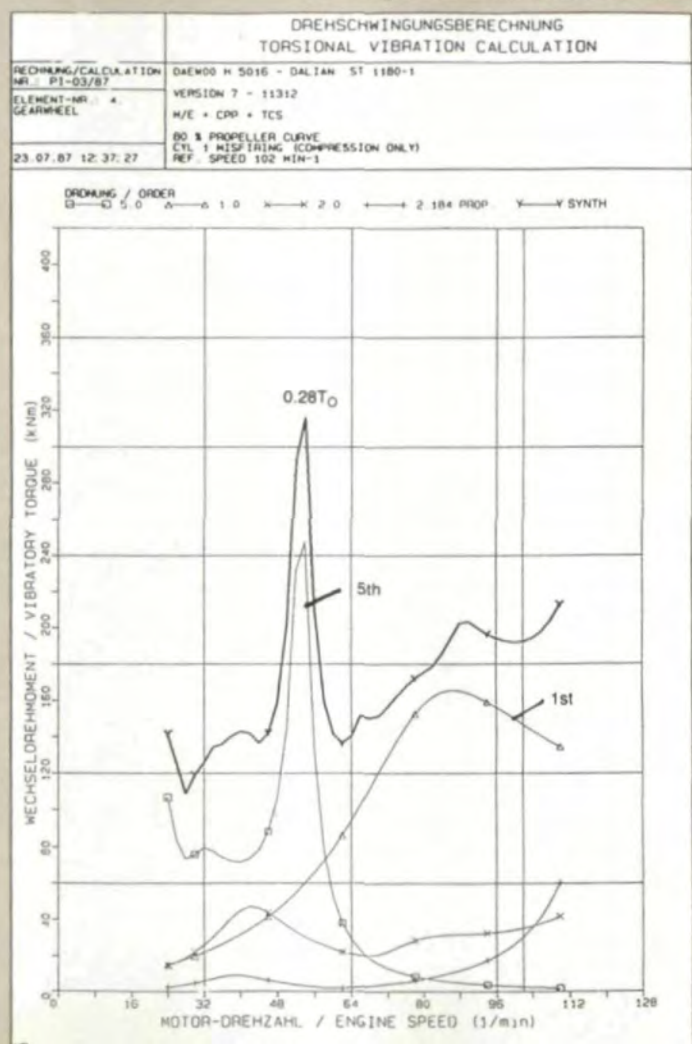


Fig. 2 – T.V.C. for misfiring situation. Cylinder 1 has only compression. Main engine is driving propeller and turbine. The 5th order and 1st order of engine are dominant.

selected, it being impossible to shift all critical resonances out of the speed range.

Fig. 2 shows a typical example for a misfiring situation (only combustion), which is defining the worst situation for the installation. The first order dominant at misfiring and the fifth order dominant for this five cylinder engine create fluctuating torques which at certain modes of operation may exceed given permissible values for either resilient coupling material or the gears. Twenty years of experience in T.V.C. have built up so much knowledge in selection of the right components for transmisions in marine and industrial application that the points discussed by Mr. Jakuba and Mr. Calistrat are not a serious subject in my daily routine any more.

Dr. Wilhelm Schaeffer
Renk Tacke GmbH
Augsburg, West Germany

(continued on page 48)

TECHNICAL CALENDAR

NOVEMBER 17-19
SME GEAR PROCESSING & MANUFACTURING CLINIC
Michigan Inn, Southfield, MI

Three days of presentations and discussions. Topics to be covered include gear manufacturing methods and machines, hob design and selection, gear inspection and chart evaluation, gear finishing, contour gear hardening, gear grinding, cold orbital forming, coolants and CNC gear shaping. Tuesday evening, Nov. 17, will feature tabletop exhibits of the latest gearing products. For more information about attendance or exhibition space, contact Joe Franchini at SME, (313) 271-1500, x394.

CALL FOR PAPERS
INTERNATIONAL CONFERENCE ON GEARING
Zhengzhou, China

Next year, Nov. 5-10, 1988 there will be a three-day conference which will include discussions on theory of tooth form, gear strength and durability, gear materials and heat treatment, dynamics of gear systems, vibration and noise analysis, lubrication, non-cylindrical and non-involute gears, power transmissions and standards. The deadline for abstracts (no more than 400 words) of papers to be presented is Nov. 1, 1987. The working language of the conference will be English. A gear products exhibition and tours to local factories, research institutes and universities will be included. This meeting is sponsored by the Machine Design & Mechanisms Institution of the Chinese Mechanical Engineering Society. Co-sponsores are the International Federation for the Theory of Machines & Mechanisms, Japan Society of Mechanical Engineers, Verin Deutscher Ingenieure and ASME/GRI. *For further information, contact: Inter-Gear* '88 Secretariat, Zhengzhou Research Institute of Mechanical Engineering, Zhongyuan Rd., Zhengzhou Henan China. Tel: 47102, Cable: 3000, Telex 46033 HSTEC CN

CHANGE OF SCHEDULE — CALL FOR PAPERS

ASME's International Power Transmission & Gearing Conference will be held April 5-7, **1989**, in Chicago, IL. Please note this change of date.

The conference will focus on major technical aspects of power transmission component design, manufacture and application, including gear efficiency, geometry, rating, lubrication, noise, dynamics, chains, belts, drives, bearings, shafts, and clutches and brakes for heavy, off-road equipment, industrial, automotive, marine and other applications.

The deadline for papers for this conference is **DECEMBER 31, 1987.**

For further information, contact: Donald L. Borden, P.O. Box 502, Elm Grove, WI 53122.

SME GEAR CLINIC & TABLETOP SHOW IN SUBURBAN DETROIT

November 17-19, 1987, SME will present its Gear Processing and Manufacturing Clinic and Tabletop Exhibits at the Michigan Inn, Southfield, MI.

Some of the papers to be presented include "Methods and Machines of Gear Manufacturing," "Design and Selection of Hobs," "Basic Gear Design," "Gear Inspection and Chart Evaluation," "Selection of High-Speed Steel for Gear Cutting Tools," "Orbital Forming in the Manufacture of Gears" and "Deburring and Finishing Gears With Power Brushes."

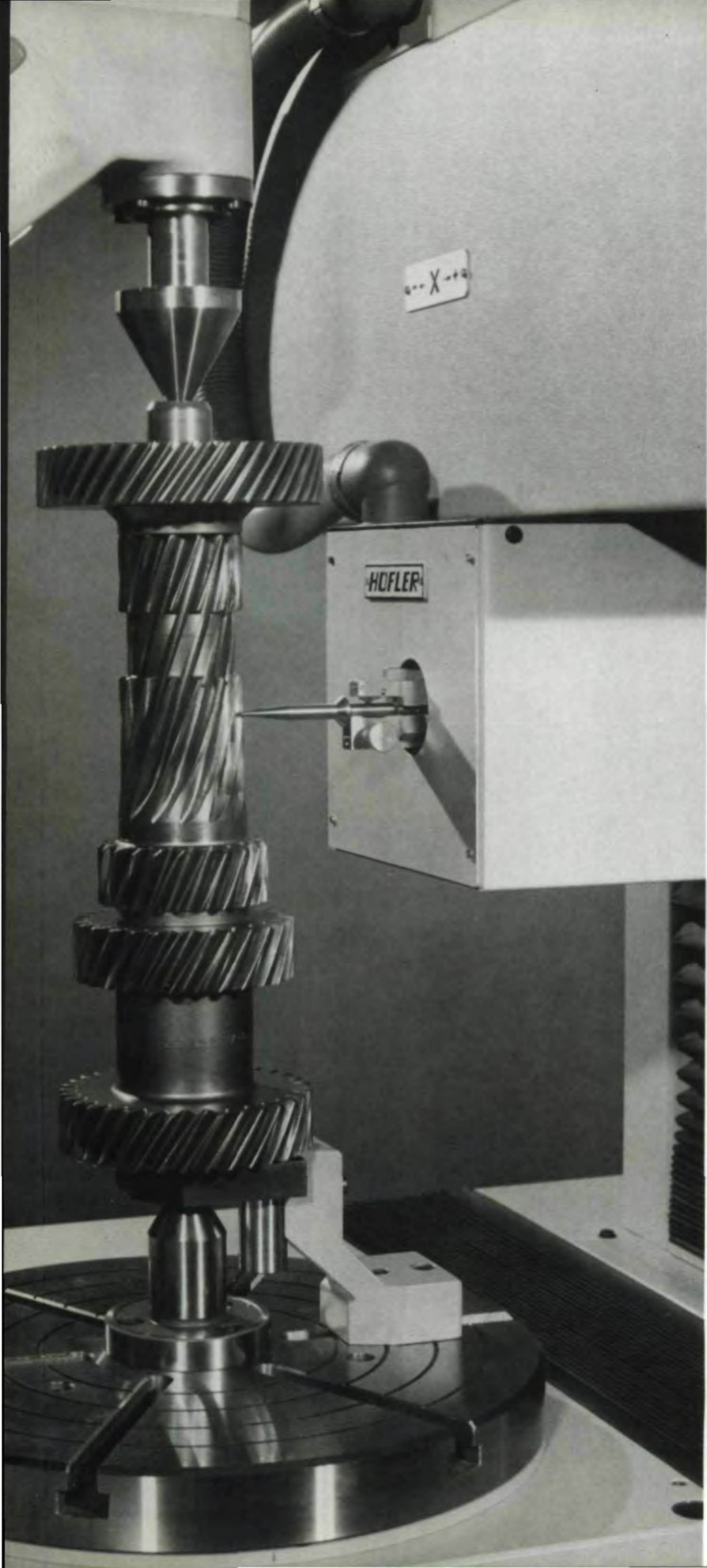
In addition to these presentations, there will be opportunity for attendees to meet and discuss the presentations with individual speakers on a one-on-one basis.

The Manufacturing Engineering Certification Institute has approved this clinic for 18 professional credits toward the SME Recertification Program.

Tuesday evening of the conference has been reserved for an SME reception and for tabletop exhibits of some of the latest products available in the gear industry. The exhibitors include Gleason Works, ITW Illinois Tools/ITW Illitron, Dexter Gear & Spline, M&M Precision Systems, Bourn & Koch Machine Tool and Abrasive Technology.

Southfield, MI, is a suburb of Detroit. The Michigan Inn is located off U.S. 10 between Eight Mile and Nine Mile Roads, approximately 30 minutes from Detroit's Metropolitan Airport.

For more information about the clinic and exhibit, contact Joe Franchini at SME headquarters, (313) 271-1500, x394.


HÖFLER

Computer Aided Design For Gear Shaper Cutters

Donald S. Whitney
Fellows Corporation
Springfield, VT

Computer programs have been developed to completely design spur and helical gear shaper cutters starting from the specifications of the gear to be cut and the type of gear shaper to be used. The programs generate the working drawing of the cutter and, through the use of a precision plotter, generate enlarged scaled layouts of the gear as produced by that cutter and any other layouts needed for its manufacture.

The gear data which should appear on the gear drawing to insure that a cutter correct for the part can be designed are shown.

Part information should include the following items:

1. Number of teeth.
2. Diametral pitch or module. (If helical, specify normal or transverse.)
3. Pressure angle (normal or transverse).
4. Outside or inside diameter. (Include limit if topping.)
5. Tooth thickness (normal or transverse at some given diameter or dimension over or between specified pins).
6. Helix angle and hand.
7. Depth of cut. (Specify root diameter and tolerance.)
8. Material and hardness at time of cutting.
9. Lowest point of contact. (Specified as true involute form diameter. Mating part and center distance information will answer this requirement.)
10. Root fillet radius specification.
11. Stock allowance for preshave, pregrind, or roughing operations (amount and position of undercut if required).
12. Amount of chamfer measured radially with limit angle of chamfer if necessary.
13. AGMA quality number.
14. Type and serial number of machine.

A computer program works through an optimizing process to produce the best cutter design possible for the part in question. A good program considers the many constraints and variables involved and investigates many designs before it arrives at the optimum one. Because of these many variables, it is not possible to write a direct formula or algorithm that will result in a cutter that will meet all of the requirements. For this reason many designs may be investigated. The following is a step by step description of how such programs are arranged.

Fig. 1 illustrates the three basic cutter blanks that will be discussed here: disk, deep counterbore for internals and parts with interfering shoulders, and taper shank (usually for internals).

The type of blank is an input parameter with the blank dimensions, thickness, face width, life, etc. being supplied by program libraries of standard dimensions for the pitch and cutter size involved. If special blank dimensions are required, they can be substituted for the standard ones. The basic size of the cutter (number of teeth) will be computed to fit the machine being used if this is known, or it may be input by the number of teeth or pitch diameter wanted on the cutter.

In addition to cutter blank dimensions, all other information required for the

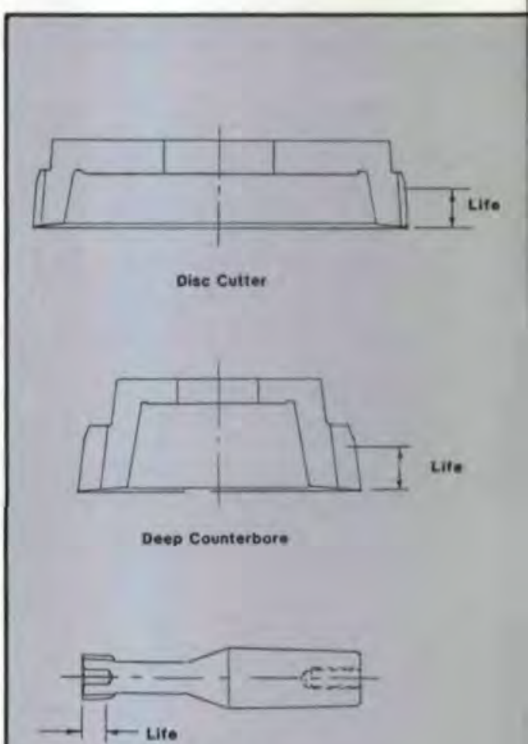


Fig. 1 – Typical blank designs.

AUTHOR:

DONALD S. WHITNEY *worked for Fellows Corporation for 46 years until his retirement in early 1987. He began with the company as a gear shaper cutter designer and then specialized in computer programming, writing various programs for the design of gear shaper cutters and shaving tools. Mr. Whitney is a member of the AGMA Gear Inspection Handbook Committee and a Life Member of the Society of Manufacturing Engineers.*

design, such as, clearance angles, tolerances, material, etc. is supplied by the program libraries and need not be entered as input parameters, unless special, non-standard values are required.

Once the cutter size (pitch diameter) has been fixed by one of these methods, the program computes the cutter tooth dimensions for the given part and then checks all of the following points, ascertaining that it produces the part specifications and meets the following good cutter design criteria:

1. Does the cutter have sufficient tip land?
2. Will the true involute form (TIF) diameter be held?
3. Will the cutter have a suitable size radius on the tip corners?
4. Will the cutter modify the tips of the work teeth?
5. Will the undercut produced by protuberance be positioned correctly?
6. Will chamfer produced be to specifications?
7. Will too much rough side rub be present on cutters for internal work? Will cutter have finish rub? Will it trim the tips of the work teeth?
8. Is the cutter barrel strong enough on taper shank cutters? Does it need flutes?
9. Will flank rub be present on internal or external work?

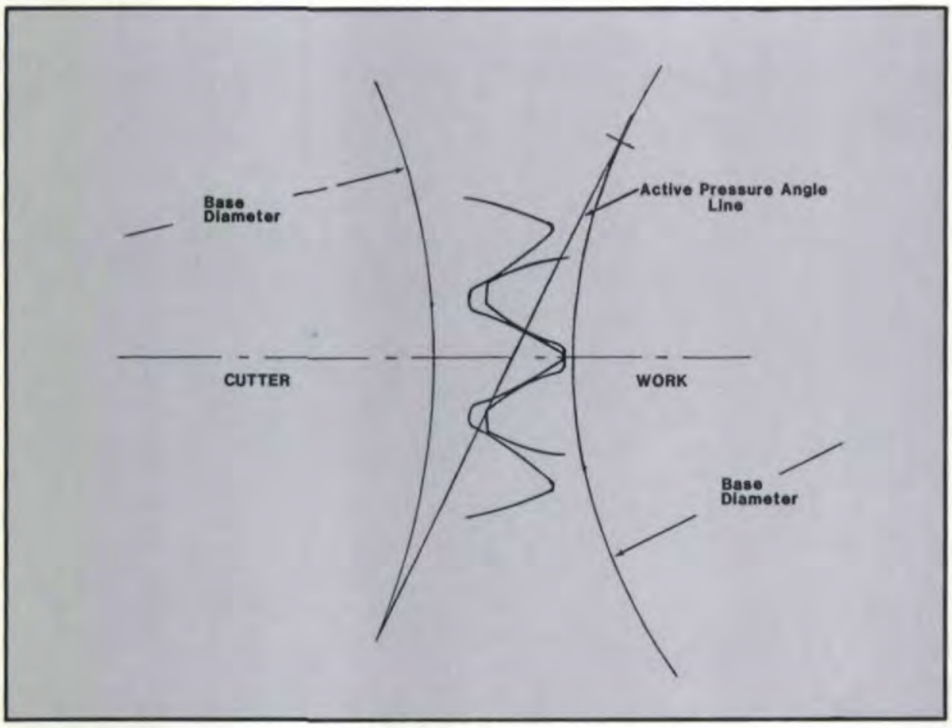


Fig. 2 – Enlarged cutter – too little tip land.

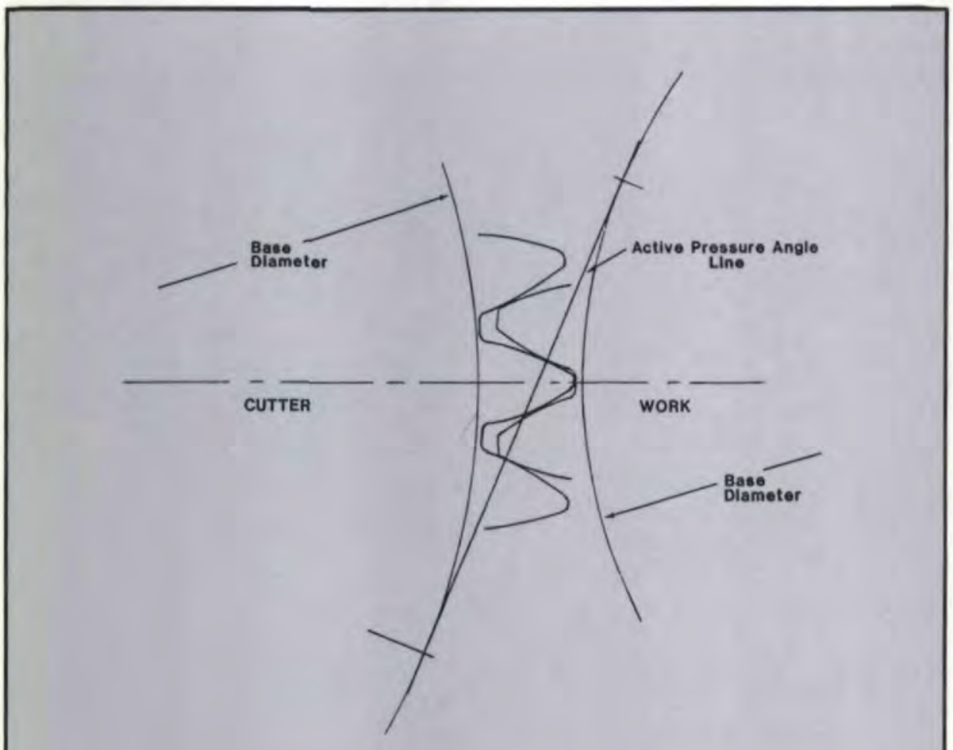


Fig. 3 – Reduced cutter – sufficient tip land.

Items 1, 2, 3, 4, 5 and 7 are dependent on the active pressure angle (APA) between the cutter and work, so the program must determine by trial and error the best starting APA to produce best results when the cutter is new and throughout its life. The APA of a cutter changes as it is sharpened back, so we must investigate what happens through its whole life. The APA discussed here originates from a cutter that operates at the transverse pressure angle of the work. When the cutter size (center distance) is increased from this point, the APA increases on external work and decreases when the size (center distance) is decreased. The active pressure angle change is in the opposite direction on internals, but the effect of enlarging or reducing the cutter is similar to external work with respect to cutter land and the true involute form (TIF) produced. This article will use the term "enlargement" or "reduction" of the cutter to indicate that the APA is being changed to satisfy requirements. The following examples will indicate situations that dictate this type of change.

Each time the cutter size (enlargement or reduction) is changed to improve or satisfy one of the above conditions, the new design is completely computed. Each feature is then checked to be sure that some other design requirement has not been violated.

Fig. 2 shows a cutter with too little land on the tips of the teeth. Fig. 3 is the same cutter reduced to a point where satisfactory land is attained. This is an external part; therefore, the APA is reduced with cutter size. Standards for the minimum allowable land for good cutter life are stored in the program library and are dependant on pitch and pressure angle of the part.

Fig. 4 illustrates how cutter enlargement affects the TIF produced. A reduced cutter will lower the trochoid height produced, resulting in a design that will hold the required TIF.

Fig. 5 shows a condition where the involute at the tip of the work tooth will be modified because of contact below the base diameter of the cutter. Shown on this same figure is the condition of cutter contact below the base circle of the work, resulting in "undercut", which takes away some of the involute near the base diameter of the work.

Undercut is most likely to happen on pinions of 20 teeth or less, while tip modification is more common on large gears and internal gears. Enlarging the cutter will improve or eliminate both the undercut or modification conditions.

From the foregoing, it can be seen that some situations require cutter enlargement, while others require reduction of the same cutter to remedy the problem. Obviously we cannot do both, so a compromise must be made by the program. The final design will always be one that at least has minimum land and produces the specified true involute form.

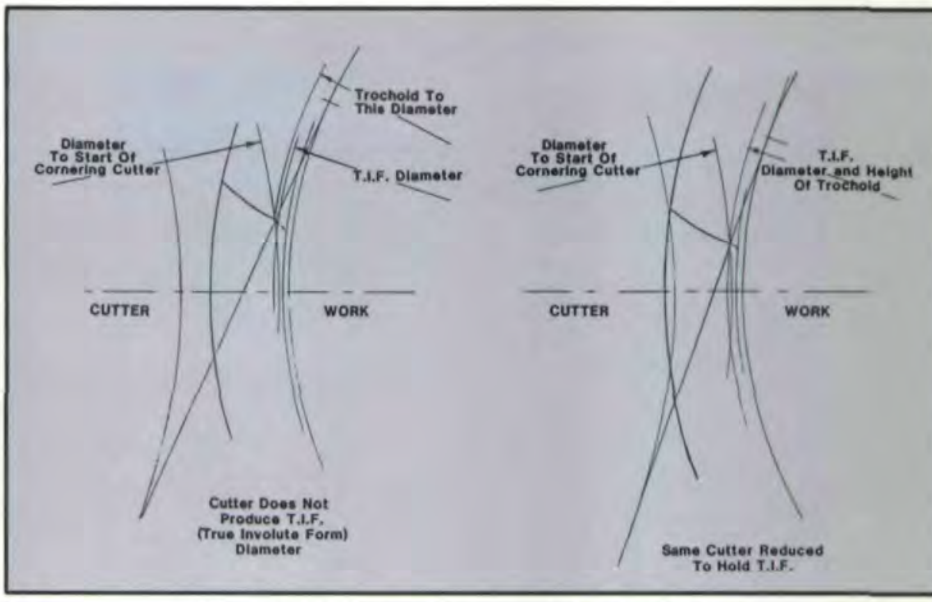


Fig. 4 – Effect of cutter enlargement on true involute form.

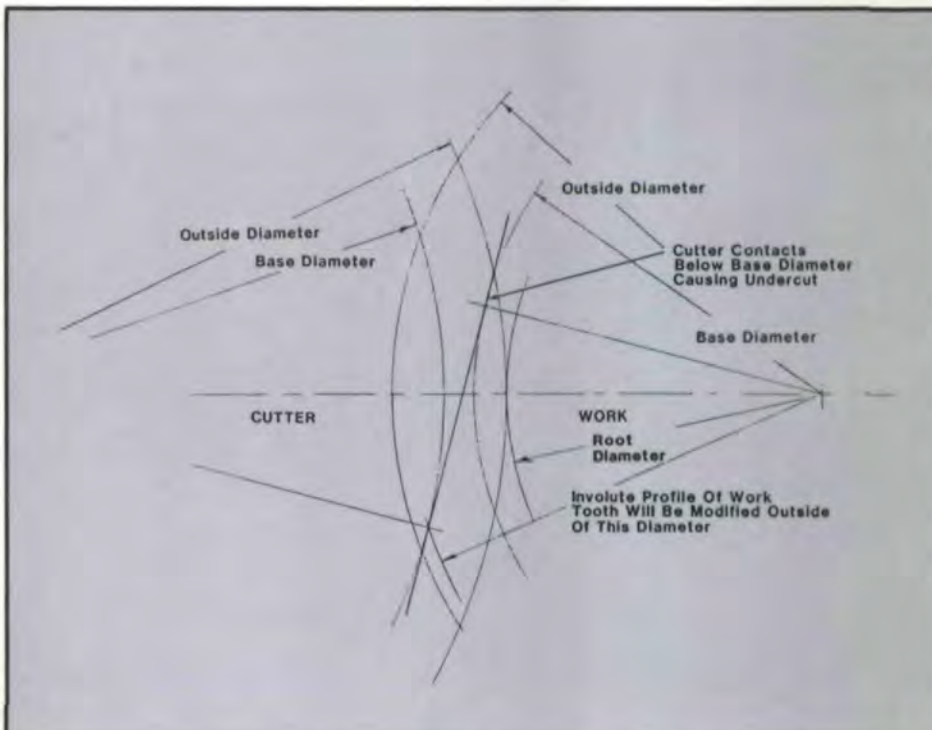


Fig. 5 – Example of tip modification and undercut.

Cutter Rub (Internal Gears)

This condition is almost always present to some degree when cutting internal gears. All gear shaping machines relieve either the cutter or the work on the return stroke to avoid scuffing the cutting edges. This relief is generally accomplished by separating the cutter and work by approximately .010" (.25mm) to .032" (.81mm), depending upon the make and type of machine. When cutting some internal parts, a wrapping effect of the internal part around the cutter results, causing the cutter to interfere or rub with the internal teeth when it is pulled back to give cutting edge relief. Three types of rub can occur and are described as follows:

Rough Side Rub. This interference, as shown in Fig. 6, is caused by the hooking action of the cutter teeth relative to the work profile as they enter on the uncut side of the work near the inside diameter. At this time, no generating action has taken place. Spaces produced are an image of the cutter teeth, requiring the cutter to be pulled back at an angle away from the hook side to provide relief on the return stroke. The dotted lines show the position of the cutting edges after the relieving motion has taken place. In the example shown, cutter rub exists even though the cutter was relieved on an angle known as the backoff angle. The amount of backoff angle is limited, as will be shown later. The conventional angle is normally about 5°, which is sometimes increased to as much as 10-12° to correct extreme rough side rub conditions. A small amount of rough side rub [less than .002" (.05mm)] can be tolerated and is usually present when cutting small internals. Excessive rough side rub will show up as an excessive burr on the face of the gear and the inside diameter from the trailing side of the cutter teeth. This problem causes excessive wear and/or load on the trailing side near the tips of the cutter teeth, resulting in a deterioration of tool life. Aside from an increase in the backoff angle, with limitations as shown later, the only way to reduce this condition is to decrease the

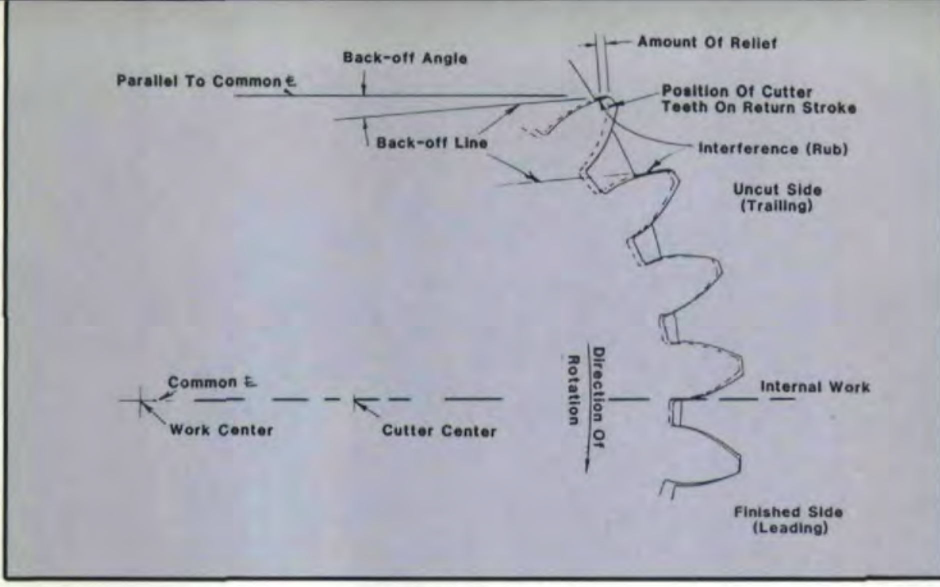


Fig. 6 – Rough side rub.

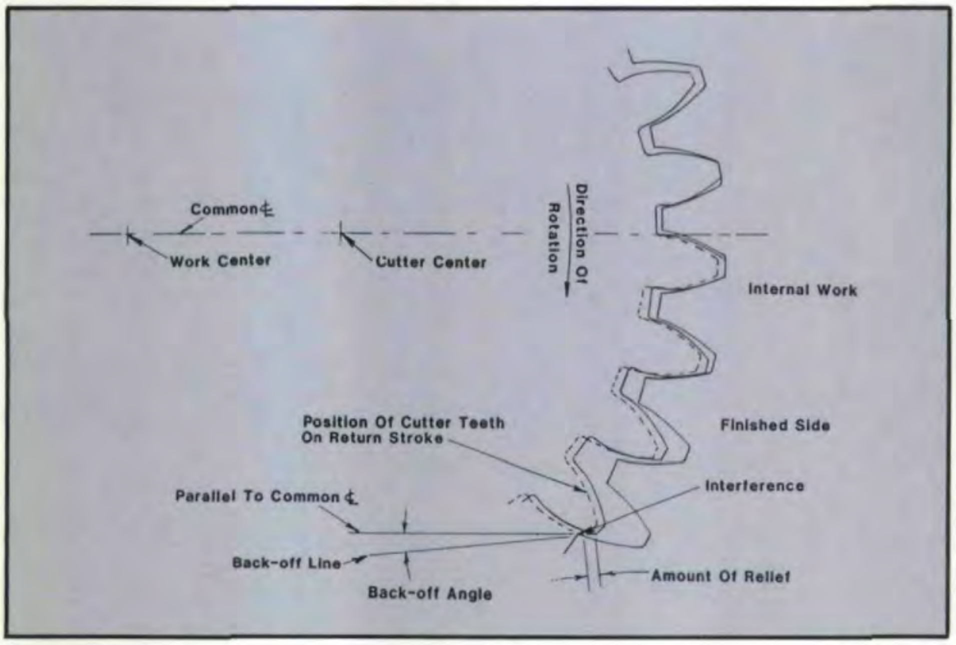


Fig. 7 – Finish side rub.

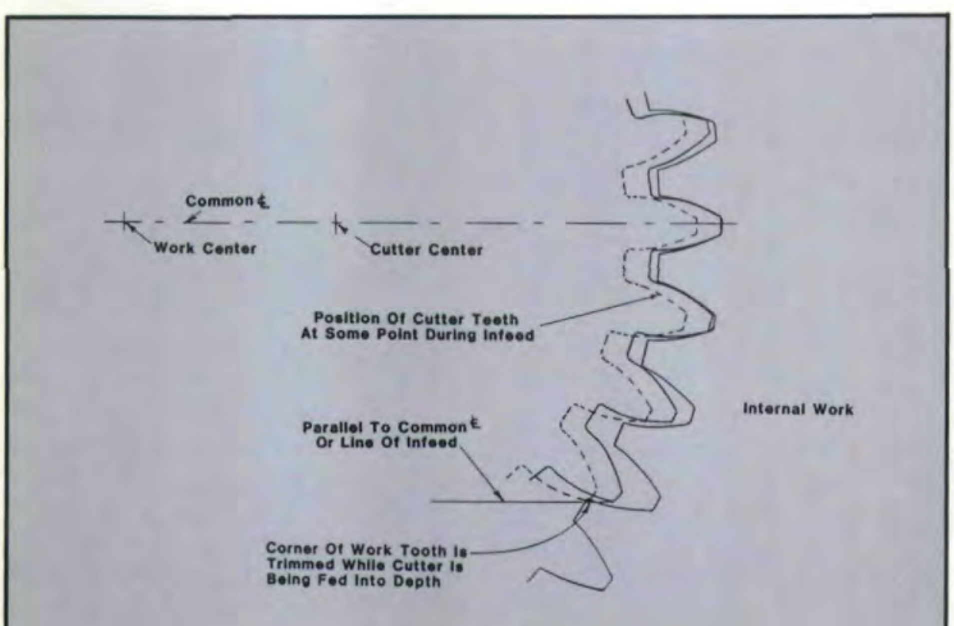


Fig. 8 – Infeed trim.

amount of backoff, usually to not less than .010" (.25mm), reduce the number of teeth in the cutter, increase the inside diameter of the part or use multiple cuts as described later.

Finish Side Rub. As shown in Fig. 7, finish side rub occurs when the cutter size is large relative to the work size, and it appears on a fully finished side of the work teeth. Finish rub will show up in the form of an excessive burr left at the inside diameter of the work teeth on the leading edges. The cutter will show excessive wear near the tips of the teeth on the corresponding side. The direction of backoff angle is almost always taken as shown and is dictated by the requirements of the rough side. Decreasing this angle would improve the finish side rub, but this usually cannot be done for the previously mentioned reasons. An increase in the operating pressure angle by reducing the cutter diameter and/or reducing the number of teeth in the cutter will decrease the amount of finish rub. This type of program will not allow any amount of finish rub to be present on the final design.

Infeed Rub. This is rub on the leading side of the teeth caused by the same conditions as finish side rub and occurring only while the cutter is feeding into depth. For this reason, the burr left at the inside diameter while infeed rub takes place will be cut away when the cutter reaches full depth and cannot be seen on the finished part. Excessive infeed rub will show up as abnormal wear on the leading side of the cutter teeth. The same remedies listed previously for improving finish side rub hold true for infeed rub. The multiple cut method, described later, will also reduce or eliminate infeed rub.

Infeed Trim. As shown in Fig. 8, infeed trim can exist regardless of the amount of backoff and occurs when the number of teeth in the cutter is too large in relation to the number of teeth in work. The angle of backoff can affect this trimming condition if it is accomplished by offsetting the cutting head upright (which is common on some of the newer types of gear shapers) and not by the swiveling method. Infeed trim will show up on the finished part as a modification or trimming at the inside diameter of one or more work teeth. This trimming is done by a cutting action, as opposed to rub, and thus does not harm the cutting tool. This condition can be reduced or eliminated by increasing the operating pressure angle of the cutter and work by decreasing the outside diameter of the cutter or by decreasing the number of teeth in the cutter.

Maximum Cutter Sizes (Internal Gears)

Fig. 9 shows the relationship between recommended teeth numbers in cutters and work pieces for the more common pressure angles. The data given here is to be used as a guideline in the selection of a cutter that will not produce excessive rough side rub, infeed rub or any amount of finish side rub or infeed trim. When the number of teeth in the work is less than the cross bars on the chart indicate for the pressure angle involved, the inside diameter should be increased by shortening the addendum. On such parts or on marginal ones, the amount of rub present should be checked by the computer program before cutting. Variations occurring from machine to machine in the amount and angle of backoff also make a precutting check desirable. Of course, internal parts of fewer teeth than shown on this chart can be cut successfully, but they should be treated as individual cases. The majority of parts in this category must be cut on machines with reduced backoff. The program stores the equivalent of this chart, which is used as a starting point to determine the number of teeth in a cutter for an internal part. If the starting number of teeth results in excessive rub, this number will be reduced by the program until the rub is eliminated.

Flank Rub (External or Internal Gears)

This type of rub can occur on the portion of the cutter tooth below its base circle on the trailing side of the cutter and will show up as excessive wear at this point. Fig. 10 illustrates this condition on an internal gear. If the previous cut leaves stock no greater than that shown by the dotted line, flank rub will not be present. For this reason, a common method for eliminating flank rub is to take multiple cuts or use the multipass method described later. The amount to feed in for each pass is determined by computer. Anything that can be done to increase the distance from the cutter base diameter to the outside diameter, such as, cutter enlargement or an increase in the number of teeth, will help eliminate this condition.

Multiple Cut Or Multipass Cutting Method

Most modern gear shapers are equipped to take multiple cuts (3 or 4 cuts in the normal manner) or to use the multipass (high rotary feed at low infeed rates) to provide a way of eliminating or reducing rough side rub and infeed rub on internals. These techniques also eliminate flank rub on both internal and external parts. Fig. 10 shows how multiple cuts can eliminate flank rub on an internal if the previous cut stock is as shown by the dotted line. This process can also be used to eliminate rough side

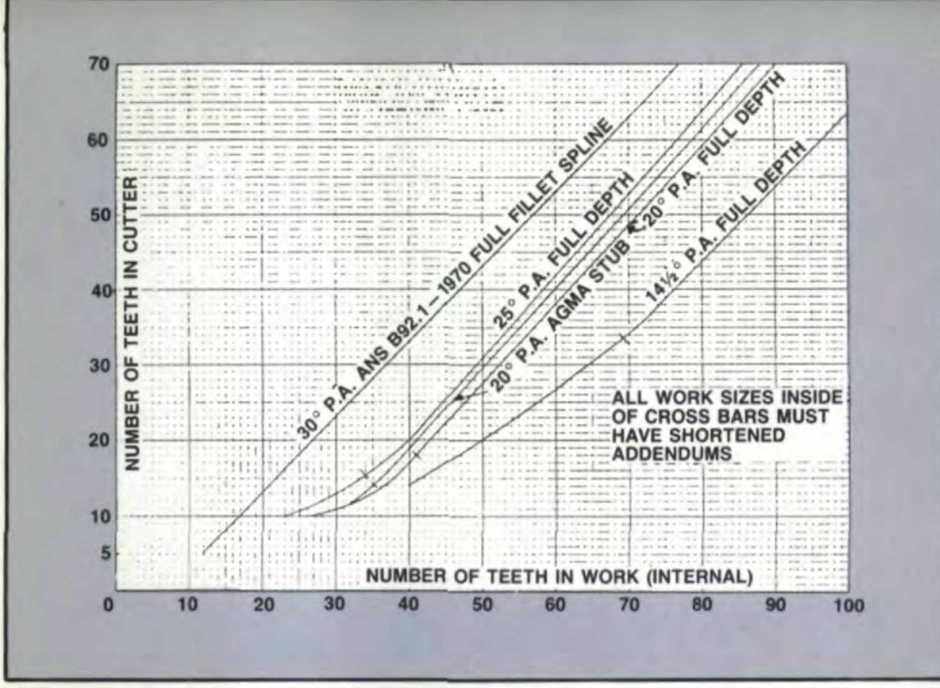


Fig. 9 – Relationship between recommended teeth numbers in cutters and work pieces for common pressure angles.

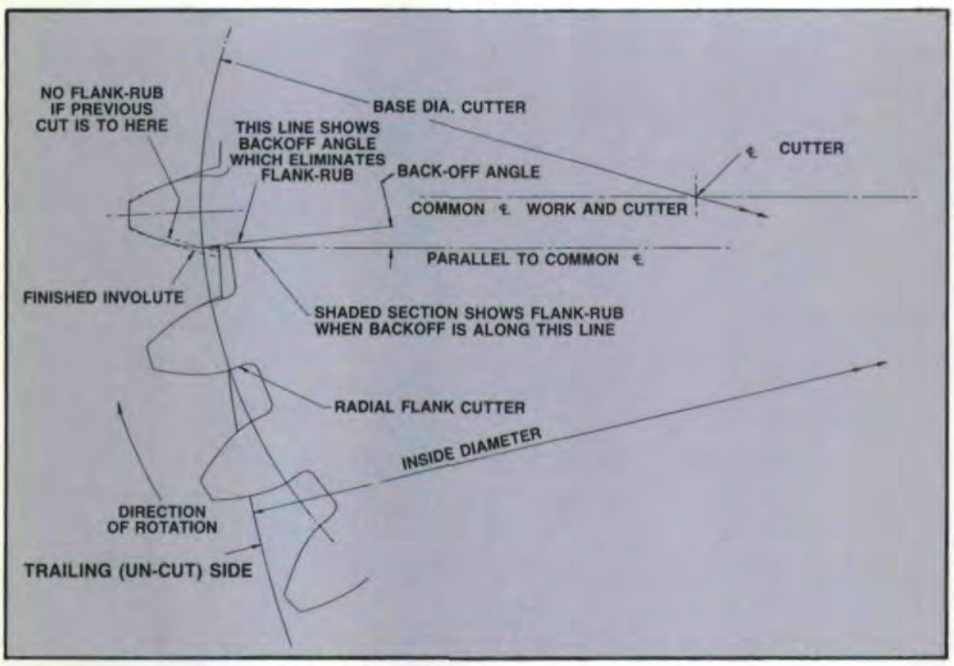


Fig. 10 – Flank rub diagram – internal work.

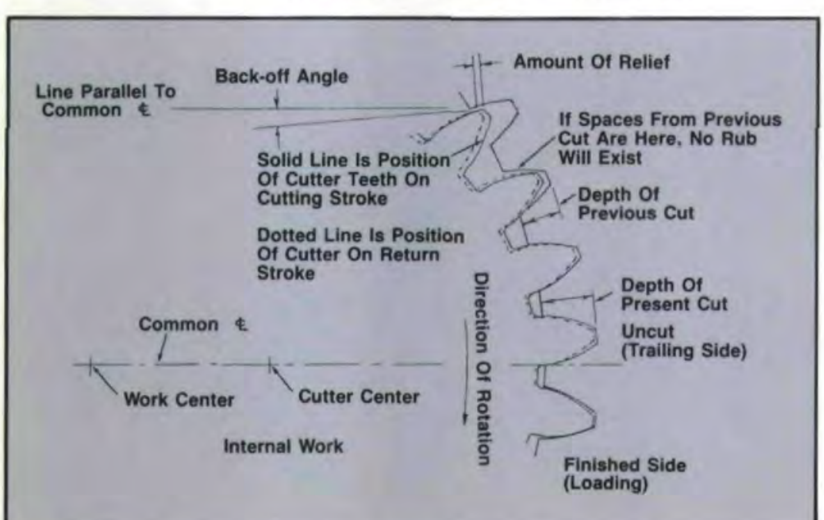


Fig. 11 – Multiple cut or multipass method of eliminating rub.

rub on internal gears. Fig. 11 shows that if the stock where rub takes place is removed by a previous cut, it will no longer rub. In this example, possibly more than one cut had to be taken to get to the "previous" depth, to attain that depth without rub. The amount of infeed between cuts for both flank rub and rough side rub elimination is determined by the computer program.

Another important feature of the multiple cut/multipass feature is to allow the use of a taper shank cutter larger than normal when cutting small internal splines. Because of reduced rub, a cutter with one or two teeth more may be used, resulting in a cutter with a stronger barrel which will be less apt to deflect.

Cutter Deflection of Small Taper Shank Cutters

The long slim cutters necessary for small diameter, wide face width, internal splines require special attention. A good program computes the relative beam strength of the barrel of the cutter, and if it comes below a previously determined minimum, it should change the cutter to a fluted barrel design. Fig. 12 illustrates the difference in the fluted and nonfluted design.

Finished Cutter Design

When the final optimized design has been determined using the methods outlined above, computer and plotter create a fully detailed working drawing as shown on Fig. 13. In addition to this, any enlarged layouts required for the fabrication of the cutter are plotted along

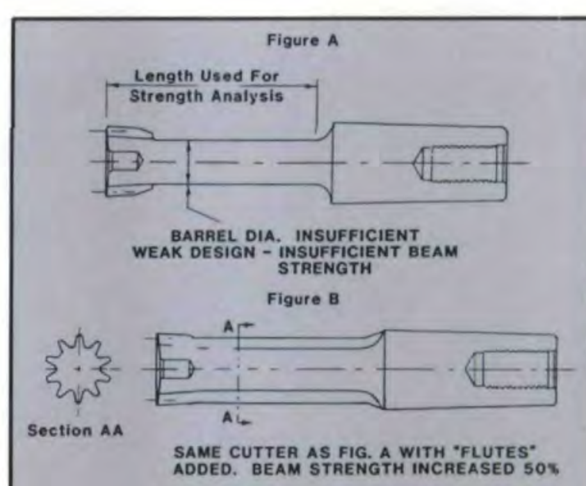


Fig. 12 – Improvement of beam strength by use of fluted barrel design.

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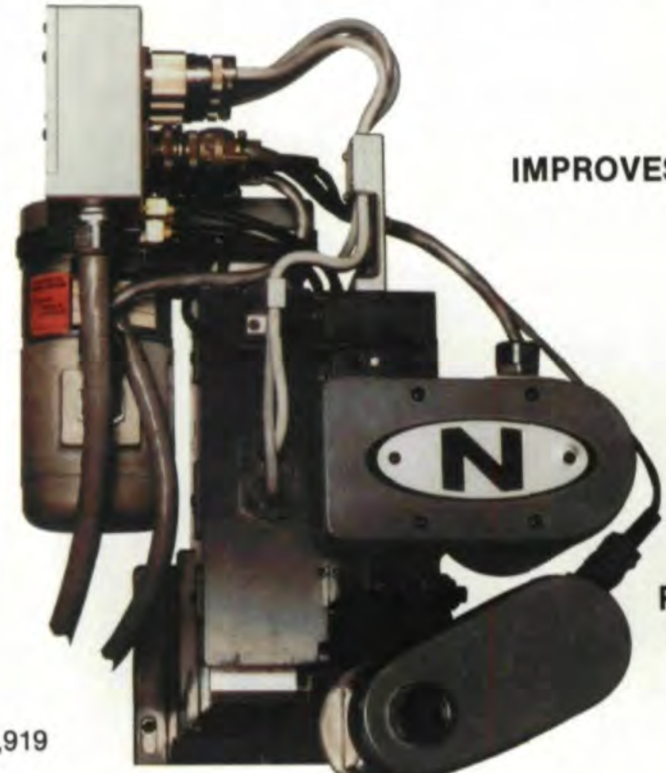

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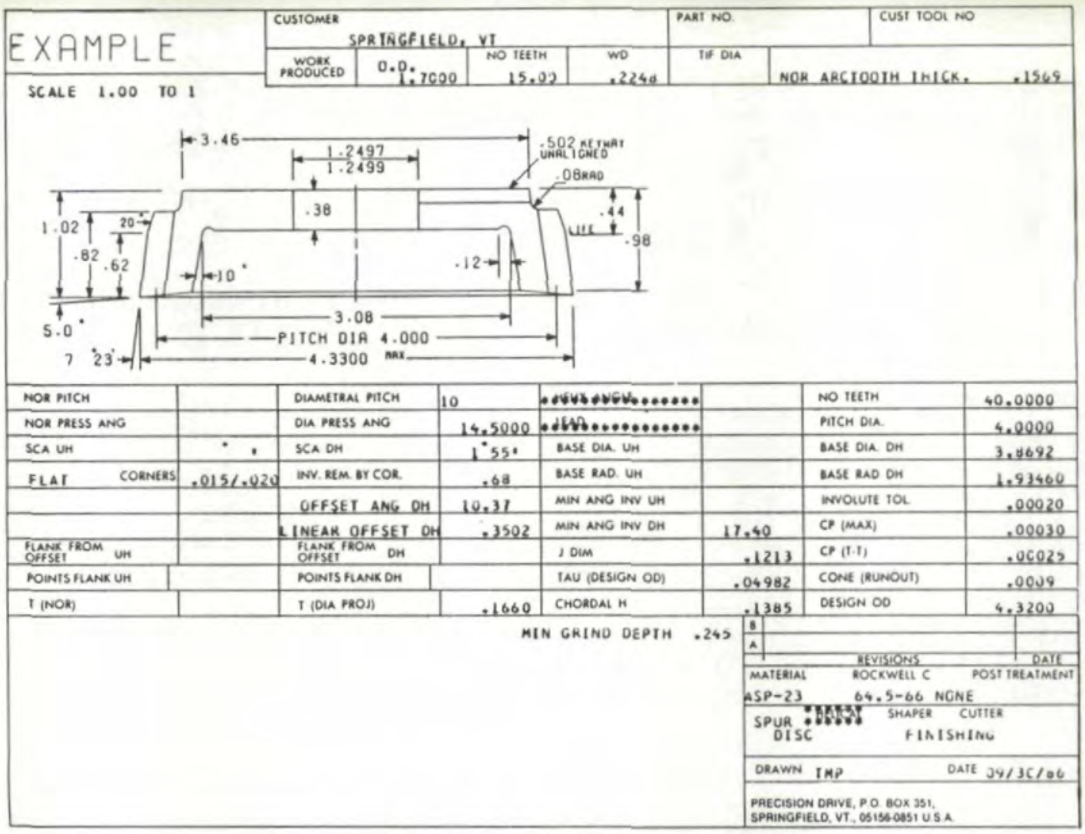
CIRCLE A-19 ON READER REPLY CARD



NOR PITCH		DIAMETRAL PITCH	10			NO TEETH	40.0000
NOR PRESS ANG		DIA PRESS ANG	14.5000			PITCH DIA.	4.0000
SCA UH	° '	SCA DH	1°55'	BASE DIA. UH		BASE DIA. DH	3.8692
FLAT CORNERS	.015/.020	INV. REM. BY COR.	.68	BASE RAD. UH		BASE RAD DH	1.93460
		OFFSET ANG DH	10.37	MIN ANG INV UH		INVOLUTE TOL.	.00020
		LINEAR OFFSET DH	.3502	MIN ANG INV DH	17.40	CP (MAX)	.00030
FLANK FROM OFFSET UH		FLANK FROM OFFSET DH		J DIM	.1213	CP (T-T)	.00025
POINTS FLANK UH		POINTS FLANK DH		TAU (DESIGN OD)	.04982	CONE (RUNOUT)	.0009
T (NOR)		T (DIA PROJ)	.1660	CHORDAL H	.1385	DESIGN OD	4.3200

B		
A		
REVISIONS		DATE
MATERIAL	ROCKWELL C	POST TREATMENT
ASP-23	64.5-66	NONE
SPUR DISC	SHAPER CUTTER FINISHING	
DRAWN IMP		DATE 09/30/86
PRECISION DRIVE, P.O. BOX 351, SPRINGFIELD, VT., 05156-0851 U.S.A.		

Fig. 13 – Sample working drawing.

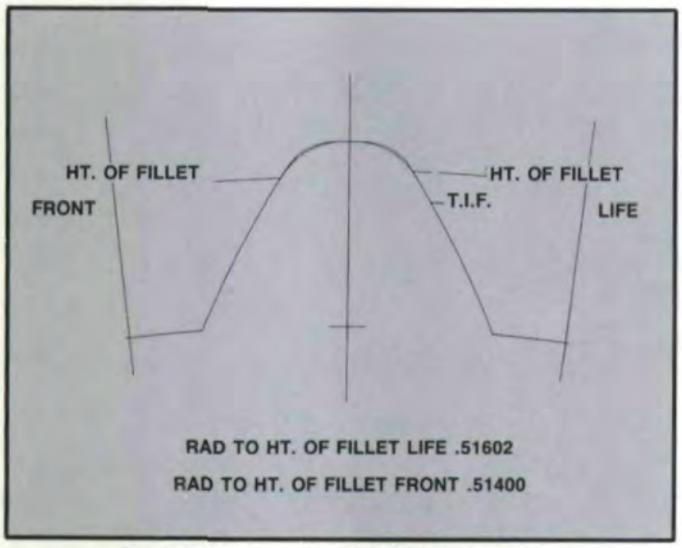


Fig. 14 – American National Standards Institute (ANSI) standard full fillet internal spline, 1.0 module, 30° p.a.

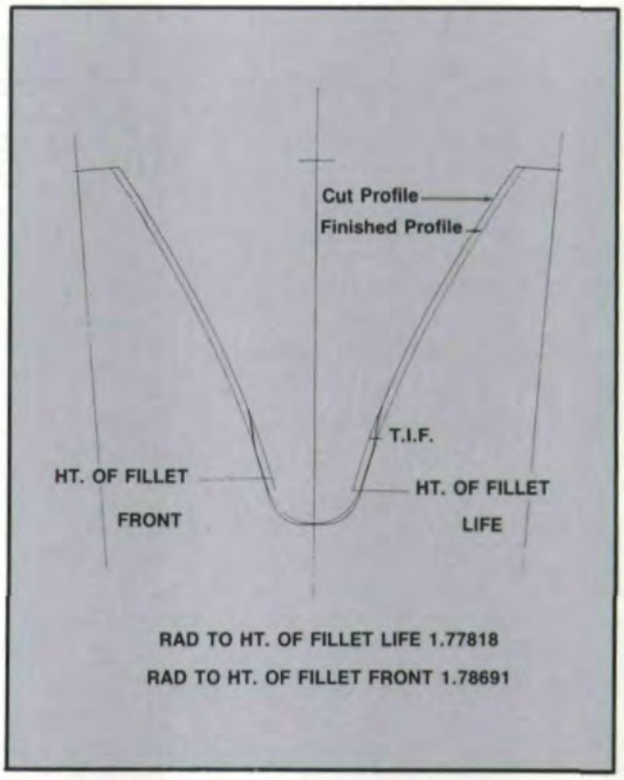


Fig. 15 – Example of undercut for preshaving or pregrinding, 10 d.p., 25° p.a.

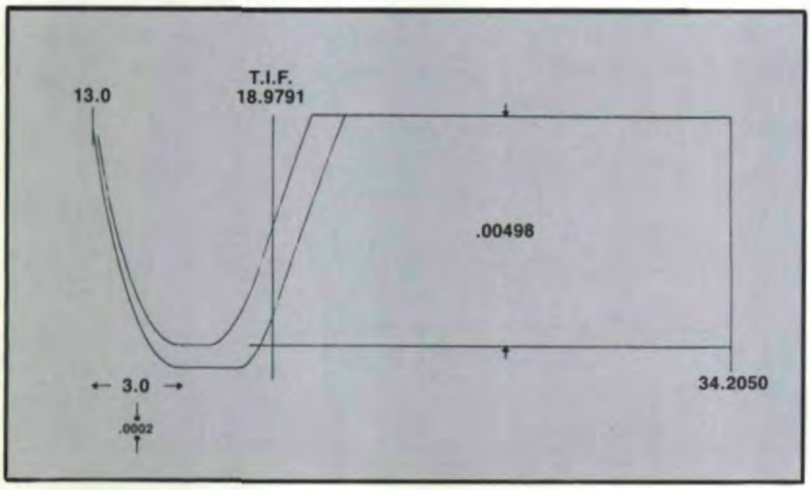


Fig. 16 – (left) Plotted involute chart of part shown in Fig. 15. Note stock left at TIF diameter.

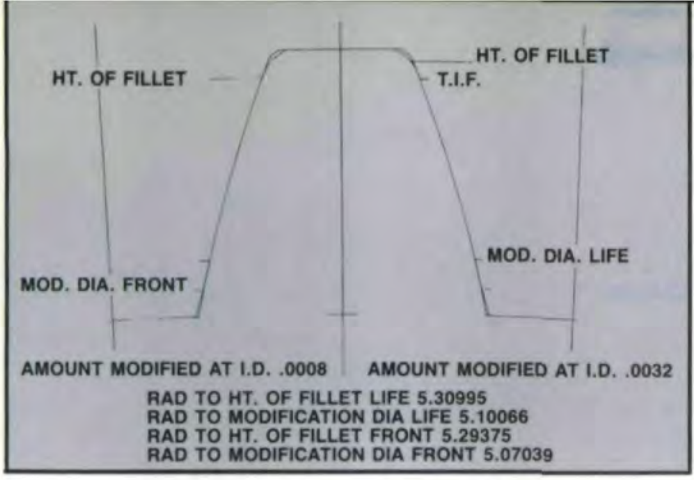


Fig. 17 – Tip modification – 6/8 d.p., 20° p.a. internal gear. See also Fig. 5.

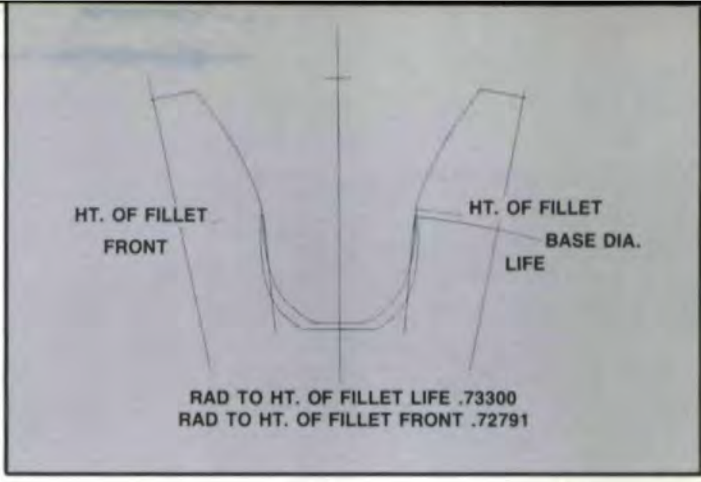


Fig. 18 – Extreme undercut, 10 d.p., 14½° p.a. pinion.

with a greatly enlarged layout from 10 to 100 x size of the work tooth profile. Examples of these work layouts are shown in Figs 14-18.

NOTE: These layouts show two values of fillet height and modification diameters, one for "front" (new cutter), and one for the life position. This change is due to the reduction of the outside diameter of the cutter as it is sharpened back, resulting in a change in the active pressure angle (APA), thus changing the fillet and modification diameters.

The computer programs also compute and print other important information for the designer and user, including the variation in depth of cut throughout the life of the cutter, variation in chamfer produced by a chamfering cutter and variation in the outside diameter produced by a topping cutter.

Each final cutter design, along with all of the input gear data, is stored in a data base on the computer system and is available to any computer aided manufacturing (CAM) programs used in the fabrication of the cutter.

Conclusion

The development of this type of software greatly enhances the ability to rapidly design a gear shaper cutter to do the best job possible. The many critical points that must be checked are done so without fail, with appropriate action in each case. The precise plotted layouts of the work are invaluable in communicating the exact profile that will be produced on the workpiece.

Acknowledgement:
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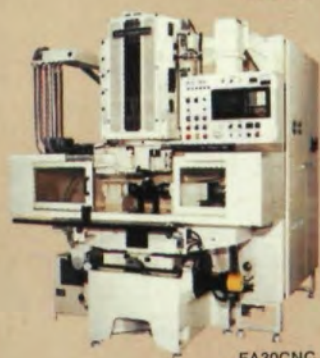
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SPECIFICATIONS

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

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CIRCLE A-8 ON READER REPLY CARD

KHV Planetary Gearing

Dr. David Yu
MPC Products Corporation
Skokie, IL

Abstract:

When evaluating modern machinery, space and weight are often indices as important as efficiency. Planetary gearing may have many of the desired characteristics: light weight, small size, large speed ratio, high efficiency, etc. The question is can a single type of planetary gearing possess all the above advantages combined. In order to give a correct answer, the basic types of the planetary gearing will presented first; then, through comparison of different types, a relatively new type of planetary gearing, the KHV planetary, will be recommended. In a later article, an optimum programming for the design of the KHV planetary gear will be introduced.

Introduction

Traditionally, a worm or a multi-stage gear box has been used when a large speed ratio is required. However, such boxes will become obsolete as size and efficiency become increasingly important considerations for a modern transmission. The single-enveloped worm gear has a maximum speed ratio of only 40 to 60. Its efficiency is only 30 to 60 per cent. The necessity of using bronze for the worm gear and grinding nitroalloy steel for the worm drives up material and manufacturing costs. Large axial forces (that on the worm are equal to the tangential force on the worm gear) require bulky bearings and shafts. Double-enveloped worm gearing can obtain a higher efficiency, but the required technology is complicated with no interchangeability. The maximum speed ratio is about 5 to 7 for a single stage gear box and 25 to 50 for a double-stage. The multi-stage gear box is complex in structure with many gears, shafts, bearings, etc., and its efficiency decreases when power passes gears and bearings in each stage. There are many advantages to epicyclic or planetary gearing, such as, light weight, small size, large speed ratio, high efficiency and the capability of providing a differential motion. However, each type of planetary gearing has its own particular features, and whether or not there is a type that can possess all the features and all the above merits combined is debatable.

Most methods for classifying epicyclic or planetary gearing are based on structure, but sometimes the structure alone cannot represent the other important features. Hence, it is suggested that not only structure, but also other indices (such as, speed ratio, efficiency, etc.) be expressed in the classification. The classification and some strict definitions used in this article might be quite different from those in other books. For example, according to *Gear Handbook*,[1] the "single epicyclic gear" has three types: "planetary gear" (Fig. 1a), "star gear" (Fig. 1b) and "solar gear" (Fig. 1c). The "compound epicyclic gear" also has three types: "compound planetary gear" (Fig. 2a), "compound star gear" (Fig. 2b) and "compound solar gear" (Fig. 2c). But in this article, four of the six different categories are considered one type, 2K-H (-), and the other two types might not be considered epicyclic gears at all. It

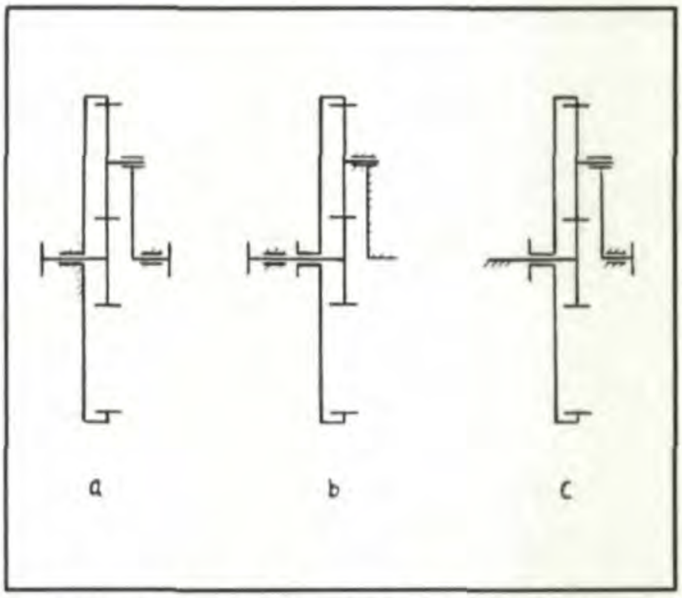

Fig. 1 – Three types of "Single Epicyclic Gear" according to Reference 1.

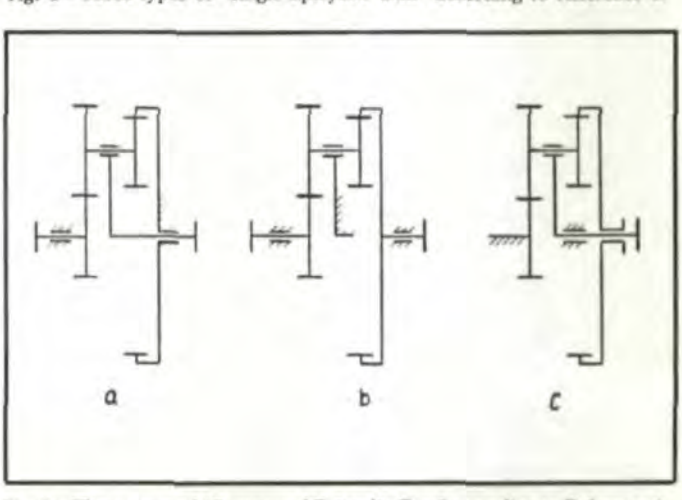

Fig. 2 – Three types of "Compound Epicyclic Gear" according to Reference 1.

AUTHOR:

DR. DAVID YU *is a gearing specialist for MPC Products Corporation. Since 1982 he has been an Honorary Fellow of the University of Wisconsin at Madison. In the academic arena, he has served as the Deputy Head of the Mechanical Engineering Department, and as Professor of Machine Design at a chinese university. Professor Yu is the author of numerous articles on gearing. He is a member of CMES, JSME and ASME Gear Research Institute.*

is better to use the term "gearing" instead of the above "gear," since "gear" means only one gear, and "gearing" implies a pair of gears or a gear train or a gear set. An epicyclic gearing must have a planet gear which orbits a common central axis when it is revolving its own axis. The locus of any point except its center is an epicycloid or epitrochoid, which is where the term "epicyclic gearing" comes from. The "star gear" (Fig. 1b) or the "compound star gear" (Fig. 2b) should not be considered a type of "epicyclic gearing" because the carrier is fixed, and there is not any epicycloidal motion. Practically, it is only a conventional gear train.

In order to give a strict definition, the author suggests using the term "moving-axis gearing." *A moving-axis gearing consists of planets with moving axes and the other gears that directly mesh with the planets.* For example, in Fig. 3, the moving-axis gearing includes a-H-p-b, where p is the planet, and H is the carrier supporting the planet. Both gear a and gear b are in direct mesh with the planet p. Either the axis of gear a or the axis of gear b coincides with the common central axis 0-0; therefore, a and b are called central gear K. Neither gear c nor gear d has a moving axis nor does either one directly mesh with the planet. Therefore, they are not members of moving-axis gearing, and they form a conventional gearing. Since in conventional gearing, all the axes of the gears are fixed, they can be named *fixed-axis gearing.*

In a moving-axis gearing, as in Fig. 3, if gear b is fixed, we only need to know the motion of either H or a. Then

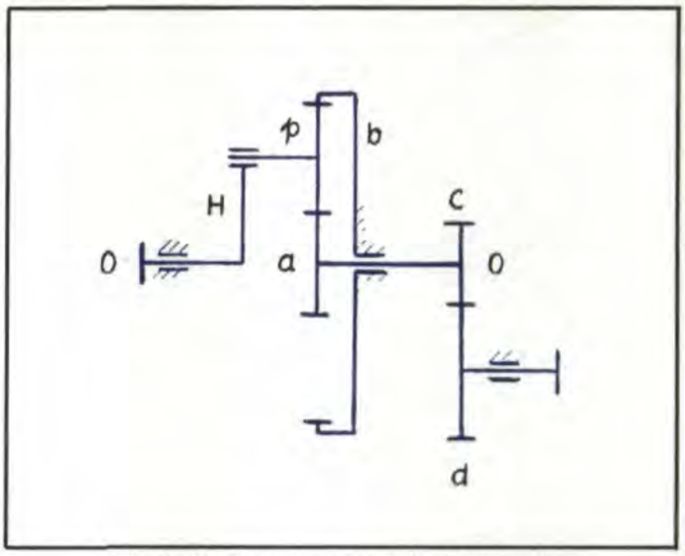

Fig. 3—Moving-axis gearing & fixed-axis gearing.

we can determine the other. In other words, there is only one constraint or one degree of freedom. Since the motion of this kind of gearing is something like the motion of the planet orbiting a fixed star, the author suggests using the term "planetary gearing." *A planetary gearing is a type of moving-axis gearing that has only one degree of freedom.*

In the planetary gearing, one of the main members (except the carrier) must be of zero angular velocity. Usually, zero angular velocity implies fixed, but not always. Exceptions will be explained later. The speed ratio of the planetary gearing is a constant and can be determined by the relation of the numbers of teeth of the gears.

Another type of moving-axis gearing is differential gearing, which is defined as follows: *A differential gearing is a moving-axis gearing that has two degrees of freedom.* For example, if in Fig. 3, neither a, b nor H is fixed, two motions of the three members should be given; then, the third can be determined. Therefore, the speed ratio of the differential may not be a constant if the motion of any of the three members is changed. Through a differential, two motions can be composed into one motion (two input to one output), and vice versa. The term "fixed-differential" used in Reference 1 will not be used in this article. Because this type of gearing has only one degree of freedom, it is not a differential, but a planetary gear.

In order to have a clear comparison among different types, it is necessary to introduce the calculations for speed ratio and efficiency.

Speed Ratio

In moving-axis gearing, the sign of a speed ratio should be taken into account, and some definitions should be given.

The angular speed ratio of two members, a and b, is defined as

$$r_{ab} = N_a/N_b$$

where the subscript ab means from a to b, and N_a and N_b are the angular velocities (or number of revolutions per time) of a and b.

If the speed ratio is from b to a, then

$$r_{ba} = N_b/N_a = 1/r_{ab}$$

If $r_{ab} > 0$, *a and b rotate in the same direction*, and if $r_{ab} < 0$, in opposite directions.

If $|r_{ab}| > 1$, from a to b is a speed reduction, and if $|r_{ab}| < 1$, from a to b is a speed increase.

A high speed ratio can be obtained when either the absolute value of a reduction speed ratio is very large, or the absolute value of an increase speed ratio is very small. For example, both $|r_{ab}| = |N_a/N_b| = 100$ and $|r_{ba}| = |N_b/N_a| = 0.01$ are considered high speed ratios. The lowest speed ratio is $|r| = 1$.

If a speed ratio of two members is relative to a third member or a relative speed ratio, a superscript is used. For example, the relative speed ratio of a and b to H is defined as

$$r^h_{ab} = (N_a - N_h)/(N_b - N_h) = 1/r^h_{ba} \tag{1a}$$

In the same way,

$$r^b_{ah} = (N_a - N_b)/(N_h - N_b) = 1/r^b_{ha} \tag{1b}$$

$$r^a_{bh} = (N_b - N_a)/(N_h - N_a) = 1/r^a_{hb} \tag{1c}$$

Then we can obtain the following three basic equations:

$$r^h_{ab} + r^b_{ah} = 1 \tag{2a}$$

$$r^h_{ba} + r^a_{bh} = 1 \tag{2b}$$

$$r^b_{ha} + r^a_{hb} = 1 \tag{2c}$$

There is only one speed ratio (or its reciprocal) that can be related to the size and can also provide some characteristics of the planetary. It is the relative speed ratio to the carrier H and is named the basic speed R_o.

$R_o = r^h_{ab}$ (The reciprocal r^h_{ba} can also be used as an *alternative.*)

The relative speed ratio of a and b to H is based on considering the carrier H as relatively fixed. It, therefore, can be determined in the same manner as that for a fixed-axis gear train, and the relation of teeth or diameters can be used.

Then

$$R_o = r^h_{ab} = (-1)^P (Z_b\, Z'_b \,.\,.\,.\,.\,.\,.\,.\,)/(Z_a\, Z'_a \,.\,.\,.\,.\,.\,.\,.\,) \tag{3}$$

where: Z_a, Z'_a, ... are numbers of teeth of the driving gears from a to b; Z_b, Z'_b, ... are numbers of teeth of the driven gears from a to b; and p is the number of pairs of external grearing from a to b.

Efficiency

Input power P_{in} is defined as positive, and output power P_{out} and frictional loss power P_f are negative. Conventionally, efficiency is used as a positive value; therefore, sometimes a minus sign or an absolute sign should be added as follows:

$$\eta = -P_{out}/P_{in} = (P_{in} - |P_f|)/P_{in} = 1 - |P_f|/P_{in} \tag{4a}$$

and

$$\eta = P_{out}/(P_{out} + P_f) = 1/(1 + P_f/P_{out}) \tag{4b}$$

If the frictional loss power can be evaluated, it is not difficult to calculate the efficiency. The method for calculating the frictional loss power is based on "latent power" or "gearing power." Although the procedures of derivation are different,the resulting formulae are similar in References 2-7.

Frictional power is caused by frictional force and relative sliding velocity and is a function of either force and relative velocity or torque and relative angular velocity. Suppose there are two systems, planetary gearing and its modified system, in which the carrier H is assumed to be relatively fixed. Torque is independent of motion, and the relative angular velocity of any two members remains unchanged whether the carrier is relatively fixed or not. Therefore, the two systems have the same frictional loss power. Through the modified system, find out the frictional loss power, then calculate the efficiency of the planetary. We define

$$P^h_a = T_a\,(N_a - N_h) \tag{5a}$$

and

$$P^h_b = T_b\,(N_b - N_h) \tag{5b}$$

T_a and T_b are the torques. P^h_a and P^h_b are the latent powers of a and b, respectively. Since they are not real powers, but have the same dimension as power, it is convenient to refer to them as the latent powers.

In the modified system, if $P^h_a > 0$ ($P^h_b < 0$), from Equation 4a, then

$$\eta^h_{ab} = -P^h_b/P^h_a = 1 - |P_f|/P^h_a \tag{6a}$$

and

$$|P_f| = (1 - \eta^h_{ab})\,P^h_a \tag{7a}$$

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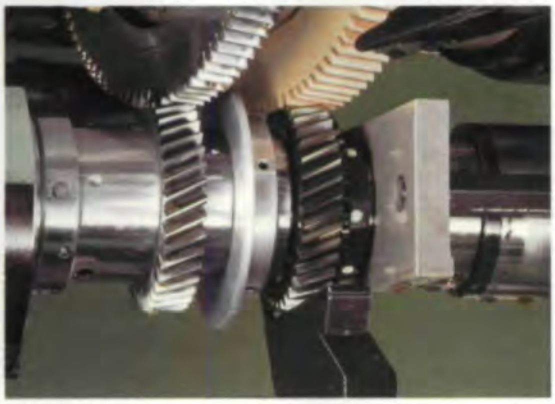

This illustration shows the tool finishing the work-piece gear and the external master gears engaged to control the operation.

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In the modified system, if $P_a^h<0$ ($P_b^h>0$), from Equation 4b, then

$$\eta_{ba}^h = -P_a^h/P_b^h = P_a^h/(P_a^h + P_f) \qquad (6b)$$

and

$$P_f = P_a^h\,(1 - \eta_{ba}^h)/\eta_{ba}^h \qquad (7b)$$

The superscript h indicates that the carrier is relatively fixed. Some feature of the modified system is similar to that of a fixed-axis or conventional gearing. Therefore, the data or formulae for calculating the efficiency of a conventional gearing can be loaned to η_{ab}^h or η_{ba}^h.

In a planetary system, one of the basic members should be of zero angular velocity. For example, $N_b = 0$, then H and a will be the input and the output.

Let $S_a = P_a^h/P_a = T_a(N_a - N_h)/(T_a\,N_a) = 1 - N_h/N_a = 1 - r_{ha}$ (8)

In the planetary, a is the input or driving member; i.e., $P_{in} = P_a = T_a\,N_a > 0$

If $S_a>0$, then $P_a^h>0$. Using Equations 4a and 7a, obtain $\eta_{ah} = -P_h/P_a = 1 - |P_f|/P_a = 1 - (1 - \eta_{ab}^h)\,P_a^h/P_a = 1 - (1 - \eta_{ab}^h)\,(1 - r_{ha})$ (9)

If $S_a < 0$, then $P_a^h<0$. Using Equations 4a and 7b, obtain $\eta_{ah} = -P_h/P_a = 1 - |P_f|/P_a = 1 - (1 - \eta_{ba}^h)\,|P_a^h|/(P_a\,\eta_{ba}^h) = 1 - |1 - r_{ha}|\,(1 - \eta_{ba}^h)/\eta_{ba}^h$ (10)

In the planetary, H is the input or driving member; i.e., $P_{out} = P_a = T_a\,N_a < 0$.

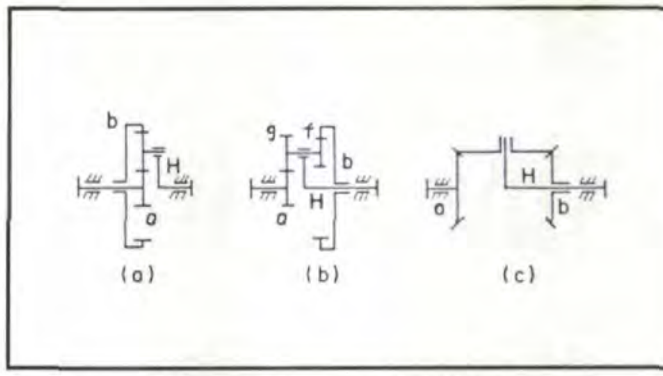

Fig. 4 – Basic arrangements of 2K-H (–) type.

If $S_a < 0$, then $P_a^h > 0$. Using Equations 4b and 7a, obtain $\eta_{ha} = -P_a/P_h = P_a/(P_a + P_f) = 1/[1 + (1 - \eta_{ab}^h)\,|1 - r_{ha}|]$ (11)

If $S_a > 0$, then $P_a^h < 0$. Using Equations 4b and 7b, obtain $\eta_{ha} = \eta_{ba}^h/[\eta_{ba}^h + (1 - \eta_{ba}^h)\,(1 - r_{ha})]$ (12)

As mentioned before, there are different methods for classifying the planetary. The classification suggested in this article can specify some important features of each type and might be better than the others. Since the primary subject of this article is not the classification system, only some basic types will be introduced for a comparison between the KHV and the others.

2K-H (–) TYPE

2K means two central gears, H denotes one carrier, and (–) means that the basic speed ratio is negative; i.e., $R_o < 0$. The basic arrangements of the 2K-H (–) are shown in Fig. 4.

Speed Ratio

$R_o = r_{ab}^h$ is the basic speed ratio which can be determined by the numbers of teeth. For example, in Fig. 4a and 4c, $R_o = r_{ab}^h = -Z_b/Z_a<0$; and in Fig. 4b, $R_o = r_{ab}^h = -Z_bZ_g/(Z_a Z_f)<0$.

If b is fixed, or $N_b = 0$, then $r_{ah}^b = (N_a - N_b)/(N_h - N_b) = N_a/N_h = r_{ah}$.

From Equation 2a, $r_{ah}^b = 1 - r_{ab}^h = 1 - R_o$, then

$$r_{ah} = N_a/N_h = 1 - r_{ab}^h = 1 + |r_{ab}^h| = 1 + |R_o| \qquad (13)$$

If, instead of the ring gear b, the sun gear a is fixed, just exchange the symbols a and b in Fig. 4, and f and g in Fig. 4b.

Efficiency

Since $r_{ah} > 1$, $r_{ha} = 1/r_{ah} < 1$. From Equation 8, $S_a = 1 - r_{ha} > 0$. Therefore, if a is input, Equation 9 should be used, and if a is output, Equation 12 should be used. For example, if $R_o = r_{ab}^h = -3$, then $r_{ah} = 1 + |R_o| = 1 + 3 = 4$. The efficiencies are listed in Table 1.

If η_{ab}^h or η_{ba}^h is a constant, the relation between efficiency and speed ratio is as in Table 2.

Important Features

Since 2K-H (–) type has a $R_o<0$, as for cylindrical gears, it must consist of one pair of internal gearing (+) and one pair of external gearing (–), so that (+)(–) = (–). For bevel

Table 1. Efficiency of a 2K-H (−) planetary with constant speed ratio and different η^h_{ab} (or η^h_{ba})

Reduction Speed Ratio $r_{ah} = 4$		Increase Speed Ratio $r_{ha} = 1/4$	
η^h_{ab} %	η_{ah} %	η^h_{ba} %	η_{ha} %
99	99.25	99	99.25
97	97.75	97	97.73
95	96.25	95	96.20
80	85.00	80	84.21

Table 2. Efficiency of a 2K-H (−) planetary with respect to the speed ratio

Speed Reduction & $\eta^h_{ab} = 95\%$		Speed increase & $\eta^h_{ba} = 95\%$	
r_{ah}	η_{ah} %	r_{ha}	η_{ha} %
7	95.71	1/7	95.68
10	95.50	1/10	95.48
50	95.10	1/50	95.10

gears, when H is relatively fixed, members a and b should have opposite directions of rotation as in Fig. 4c.

From Equation 13, $r_{ah} = N_a/N_h = 1 + |R_o|$. For example, if $R_o = -Z_b/Z_a$, then $r_{ah} = 1 + Z_b/Z_a$. It reveals the fact that the speed ratio of a planetary r_{ah} is only one plus the absolute value of the basic speed ratio, R_o, which represents the speed ratio of a fixed-axis gearing or a conventional gearing. Usually, the larger the R_o, the larger the diameter of the biggest gear, which determines the overall size. Therefore, the 2K-H (−) type cannot be used for high speed ratios, because its size will be too large. The maximum speed ratio for the 2K-H (−) type is about 12. For example, the "ROSS" reducer with a structure similar to Fig. 4a has a maximum speed ratio of only 9,[8] which sometimes can also be obtained by a pair of conventional gears. When a high speed ratio is required, multi-stage gearing should be used, which results in a complicated structure with many gears, shafts and bearings.

Since $r_{ah} > 0$, the input and the output always have the same direction of rotation. Since $r_{ah} > 1$, from a to H must be a speed reduction, and from H to a must be a speed increase.

From Tables 1 and 2, we can observe that the change of speed ratio has little effect on the efficiency, and the efficiency of the 2K-H (−) is higher than the efficiency when H is relatively fixed. This is a very important feature. For example, if the efficiency of a gearing transmitting 10,000 hp drops 1%, this might be a small relative amount. But $10{,}000 \times 1\% = 100$ hp, which means an increase of 100 hp of power loss or the generation of a large amount of heat. Then the temperature of the lubricant would increase and its viscosity would decrease, which might cause the failure of the gearing. Therefore, for the large power gearing, the efficiency should be kept as high as possible, and the 2K-H (−) is a good candidate.

2K-H (+) Type

This type also consists of two central gears and one carrier, but has a positive basic speed ratio $R_o > 0$. Its basic arrangements are as shown in Fig. 5.

If b is fixed, from Equation 2 and the fact that $R_o > 0$, we find

$$r_{ah} = N_a/N_h = 1 - r^h_{ab} = 1 - R_o = 1 - |R_o| \qquad (14)$$

When R_o approaches 1, the r_{ah} approaches zero, which means r_{ah} will become a very high speed increase ratio and its reverse r_{ha} will be a very high speed reduction ratio. For example, in Fig. 5a if $Z_a = 100$, $Z_b = 101$, $Z_g = 99$, and $Z_f = 100$, then $R_o = r^h_{ab} = Z_g\ Z_b/(Z_a\ Z_f) = 99 \times 101/(100 \times 100) = 9999/10000 > 0$, and $r_{ah} = N_a/N_h = 1 - R_o = 1/10000$ and $r_{ha} = N_h/N_a = 1000$.

The main features of 2K-H (+) type are as follows:

Since $R_o > 0$, usually, the 2K-H (+) consists of two pairs of internal gears or two pairs of external gears. As for the bevel gearing, a and b should have the same direction of rotation when H is relatively fixed.

When R_o approaches 1, r_{ah} approaches zero and r_{ha} approaches infinity. Therefore, a very high speed ratio can be obtained and the size remains small.

When $0 < R_o < 2$, then $|r_{ah}| < 1$. This means from a to H is a speed increase. If $R_o > 2$, then $|r_{ha}| > 1$. It means from a to H is a speed reduction. Usually, a large R_o requires a

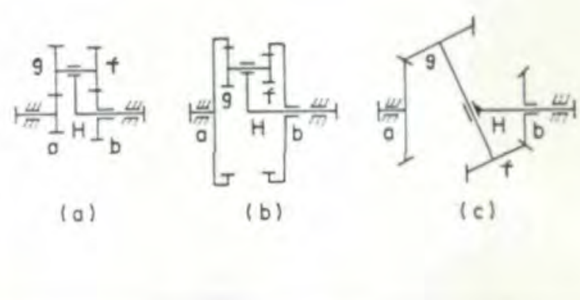

Fig. 5 – Basic arrangements of 2K-H (+) type.

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- C. Skew Axis

3. Gear Ratios

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- A. Nomenclature
- B. Involuntary — Contact Ratio, etc.
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  - b) Circular Type
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  - d) Schematic — Principles
  - e) Speeds — Feeds
  - f) Machine Cutting Conditions
- B. Hobbing
  - a) The Hobbing Machine
  - b) Types and Manufacturers
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Table 3. Efficiency of 2K-H (+) Type (%)

$R_o = r^h_{ab} = 11/12 < 1$				$R_o = r^h_{ab} = 64/63 > 1$			
$r_{ha} = 12$	$r_{ah} = 1/12$			$r_{ha} = -63$	$r_{ah} = -1/63$		
η^h_{ba}	η_{ah}	η^h_{ab}	η_{ha}	η^h_{ab}	η_{ah}	η^h_{ba}	η_{ha}
99.9	98.90	99.9	98.91	99.9	93.60	99.9	93.98
99.0	88.89	99.0	90.09	99.0	36.00	99.0	60.74
98.0	77.55	98.0	81.97	98.0	−28	98.0	43.36
97.0	65.98	97.0	75.18	97.0	<0	97.0	33.56
90.0	−22.2	90.0	47.62	90.0	<0	90.0	12.33

large size, therefore, $R_o > 2$ is seldom used.

When $R_o < 1$, then $r_{ah} > 0$, which means that a and H are in the same direction of rotation. When $R_o > 1$, then $r_{ah} < 0$, which means that a and H are rotating in opposite directions.

When R_o lies between 0 and 1, $r_{ah} = 1 - R_o = 1$ to 0, therefore, $r_{ha} = 1/r_{ah} > 1$ and $S_a = 1 - r_{ha} < 0$. If a is the input, then Equation 10 is used for calculating η_{ah}. If H is the input, Equation 11 is used for η_{ha}.

When R_o is larger than 1 then $r_{ah} < 0$; therefore, $r_{ha} < 0$, and $S_a > 0$. If a is the input, Equation 9 is used for calculating for η_{ah}. If H is the input, Equation 12 is used for η_{ha}.

The efficiencies of $R_o = 64/63$ and $R_o = 11/12$ are listed in Table 3 as examples.

From the above, three important conclusions can be drawn:

a). The 2K-H (+) has a very useful feature, the capability of providing a high speed ratio with a small size. However, the larger the speed ratio, the lower the efficiency. For example, the "ANDANTEX" reducer is similar to the one in Fig. 5a. When $r_{ha} = 3.27$, $\eta_{ha} = 95\%$, but when $r_{ha} = 67$, η_{ha} is only about 45%.[9] Therefore, effort should be made to increase the efficiency.

b). When the gearing is for speed increase, the efficiency may become negative, which is called self-lock and is normally undesirable.

c). The efficiency is very sensitive to the change of η^h_{ab} (or η^h_{ba}). Therefore, how to obtain a high η^h_{ab} is a crucial problem for increasing the efficiency of the 2K-H (+) type.

KHV Type

From the previous discussion, it is clear that the 2K-H (−) type can obtain a high efficiency, but its maximum speed ratio is small (not much larger than that of a pair of conventional gears), and the 2K-H (+) type can provide a high speed ratio with a small size, but its efficiency is low. One feasible method of overcoming these disadvantages is to combine these two types. The combination (which can provide a medium speed ratio with a medium efficiency,) is named 3K type. Owing to space limit, the details of the 3K will not be discussed here. However, its structure must be more complicated than a single 2K-H (−) or 2K-H (+), and sometimes, the gain may be less than the loss. An interesting question may arise: whether there is a type that can obtain both high efficiency and high speed ratio with a small size and a compact structure.

First, for a large speed ratio, it is better to choose 2K-H (+) type which is made up of two pairs of either internal gearing, (+) (+) = +, or external gears (−) (−) = +. Can we just use one pair of internal gears (+) to obtain a more compact structure?

Secondly, for increasing efficiency when other conditions are the same, the most important factor is η^{h}_{ab} (or η^{h}_{ba}); i.e. the efficiency when the carrier is relatively fixed or the efficiency of a conventional gearing. (See Table 3.) This figure can be obtained from some empirical data (for example, 0.95 to 0.98 for a pair of external gears and 0.97 to 0.99 for a pair of internal gears) or from some formulae as in References 5, 6 and 10. One of them for a pair of gears is as follows:

$$\eta_{12} = 1 - 3.14159\ \mu\ K_e\ (Z_2 \pm Z_1)\ /\ (2\ Z_2\ Z_1) \quad (15)$$

where: Z_2 and Z_1 are numbers of teeth, μ is coefficient of friction, "+" stands for external gearing; "−" for internal gearing; and K_e is meshing coefficient.

K_e has a relation to contact ratio and the position of meshing. μ bears relation to materials, surface finishing, lubrication,etc. Besides K_e and μ, the important factors are the numbers of teeth and arrangement. For example, in a pair of external gears where $Z_2 = 100$ and $Z_1 = 99$, the $(Z_2 + Z_1)/(Z_2Z_1) = 199/9900$. In an internal gear pair, $(Z_2 - Z_1)/(Z_2Z_1) = 1/9900$. Therefore, internal gearing is much better than external, and the best arrangement is $Z_2 - Z_1 = 1$.

Thirdly, in a planetary along the power flow, the fewer the pairs of gears, the higher the efficiency. If it consists of only one pair of gears, the power loss would be the smallest.

It is the KHV type that can satisfy the above requirements. Consisting of only a central gear (ring gear b) meshing with planets (a), a carrier (H), and an equal angular velocity mechanism (V), the KHV (as shown in Figs. 6,7) is compact in structure, light in weight, and capable of providing both large speed ratio and high efficiency. The speed ratio for a single stage KHV is from 40 to 200. Its weight and size are about one quarter those of a multi-stage conventional gear box. Through optimum design, its efficiency can reach 92% when the speed ratio is up to 200.

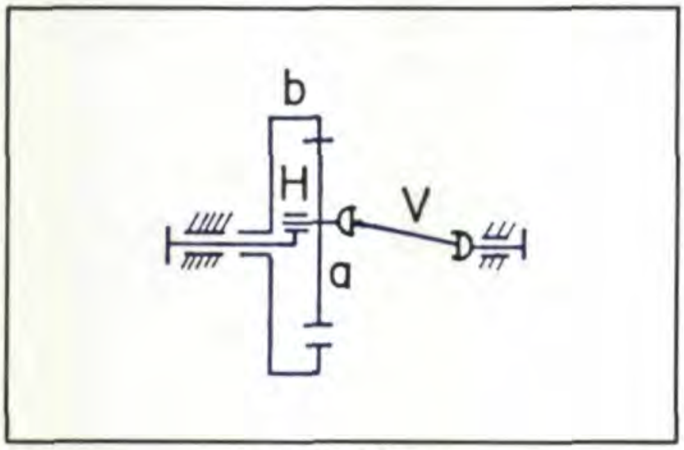

Fig. 6 The KHV type.

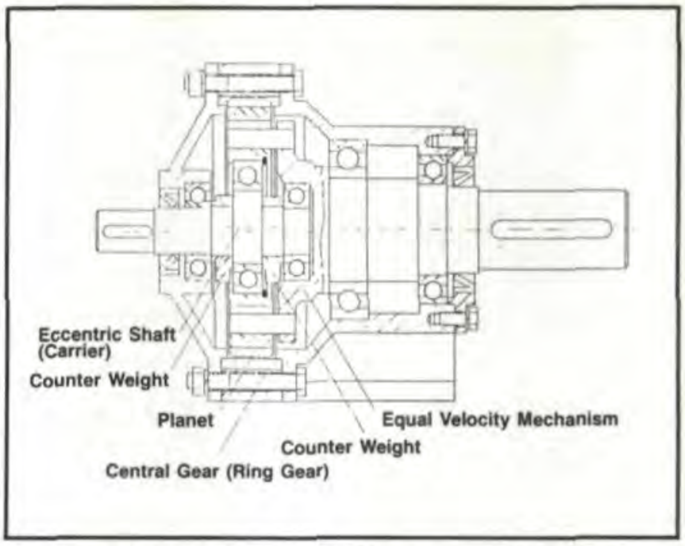

Fig. 7 – The KHV reducer with H as input and V as output.

The KHV has a basic speed ratio $R_o = Z_b/Z_a > 0$, which is similar to the 2K-H (+). Therefore, the KHV can provide a high speed ratio. Since it consists of only one pair of internal gears, it might be possible for the KHV to obtain a high efficiency.

There are only three main members in the KHV: the ring gear b, the carrier H, and the planet a. If one is fixed, the other two should be the input and the output.

If the ring gear b is fixed, or $N_b = 0$, then

$r^{b}_{ah} = N_a/N_h = 1 - R_o = 1 - r^{h}_{ab} = 1 - Z_b/Z_a = (Z_a - Z_b)/Z_a$ (See Equation 2a.)

$$\text{Then } r_{ha} = N_h/N_a = -Z_a/(Z_b - Z_a) \quad (16)$$

If the angular velocity of the planet is zero, or $N_a = 0$, $r^{a}_{bh} = N_b\ /N_h = 1 - r^{h}_{ba} = 1 - Z_a/Z_b = (Z_b - Z_a)/Z_b$

$$\text{Then } r_{hb} = N_h/N_b = Z_b/(Z_b - Z_a) \quad (17)$$

The equal angular velocity mechanism V is used to transmit the motion and power of the planet to a shaft whose axis coincides with the central axis. Any coupling that can transmit motion between two parallel shafts, such as, universal joints or Oldham couplings, might be used as the V. However, these kinds of couplings are too large and heavy, which would tend to nullify the advantages of the KHV. Therefore, special designs for the V should be used. One of them is the plate shaft type, as shown in Fig. 8. On the planet there are several holes of the same diameter d_2 equally spaced on a circle of D. On the plate shaft there is a plate having several pins of the same diameter d_1 equally spaced on a circle of D. Each pin fits into one hole. The dimensions are so arranged that $d_2 - d_1 = 2\ 0_10_2$, where 0_10_2 is the center distance of b and a. Then 0_10_2M N can form a parallelogram four-bar linkage, in which the opposite bars always have the same angular motion. Since M and 0_2 are on the plate, $M0_2$ can represent the plate shaft. Since N and 0_1 are on the planet, $N0_1$ can represent it. $M0_2$ and $N0_1$

therefore, are opposite bars of the same angular motion. As mentioned before, $N_a = 0$ does not mean that the planet a is fixed, otherwise the KHV cannot work because it is jammed.

In Fig. 8, if ring gear b is rotatable, and the plate shaft (MO_2) is fixed, the planet (NO_1) can move when carrier H (O_1O_2) rotates. Since (NO_1) should have the same angular velocity as the opposite bar (MO_2), the angular velocity of the planet (NO_1) is zero $N_a=0$, and the movement is a translation with no rotation. There are also other types of the V, such as floating plate type, zero teeth difference type, etc.

Fig. 9 is a KHV for driving a five ton gantry. The plate shaft is fixed, the carrier is the input and the ring gear is the output. The speed ratio is +74. From outside only the 5.5 hp motor can be seen. The old design used to be a bulky 3-stage gear box parallel to the wheel. The KHV cuts down weight and size to the maximum extent, since there is no increase in weight and size for the KHV. All the parts of the KHV are built inside the wheel. In the same way, the KHV can be built inside the drum of a winch, the pulley of a conveyor or the adapter of a gearmotor.

The KHV was first patented in Germany in 1925. Its tooth form was epicycloid. In the 1930's involute tooth form was also used for the KHV. However, its development was hindered for a long time by the problems of interference and efficiency. Having studied those problems for decades, the Japanese Sumitomo Company developed the CYCLO reducer. Improved by using hardened rollers, high accuracy and modified epicycloid (or epitrochoid) tooth form, the CYCLO has reached an efficiency of 95% with speed ratio up to 87[11] and has become a successful gearing in the world market. Some businessmen predict that the CYCLO will replace multi-stage gear and worm boxes in the future. However, it is difficult for other companies to design and make the particular gear cutting and grinding machines for the modified epitrochoid teeth and to know how to modify the form.

The basic speed ratio of the KHV is positive (+), similar to the 2K-H (+) type, which can provide a high speed ratio, but the higher the speed ratio, the lower the efficiency.

To avoid interference, the minimum teeth difference between the ring gear and the pinion of an internal pair of gears is usually limited to 10 or 12 for 20° pressure angle involute tooth form.[12, 13] But the KHV cannot keep to this restriction. Since the 1960's KHVs with teeth differences of 2 to 6 have also been used in China, Japan and Russia.

However, the author suggests using the smallest teeth difference; i.e., 1, and a high speed ratio in order to bring the advantages of the KHV into full play. The question is how to avoid interference and increase efficiency. The interference calculation of internal gearing is very complicated and is separately introduced in another work by this author.[14] From Equation 15 the speed ratio $r_{ha} = -Z_a/(Z_b - Z_a)$, therefore, the more teeth and the smaller the teeth difference, the higher the speed ratio. For example, if $Z_b = 50$, when teeth difference, $Z_d = Z_b - Z_a = 1$, $r_{ha} = -49/(50\text{-}49) = -49$, and when $Z_d = 6$, $r_{ha} = -44/(50\text{-}44) = -7.3$ the same result can also be provided by a pair of simple conventional gears.

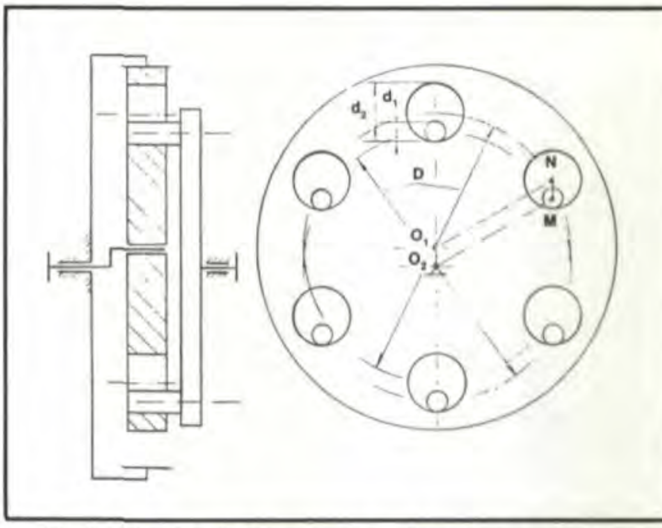

Fig. 8—The plate shaft type equal angular velocity mechanism.

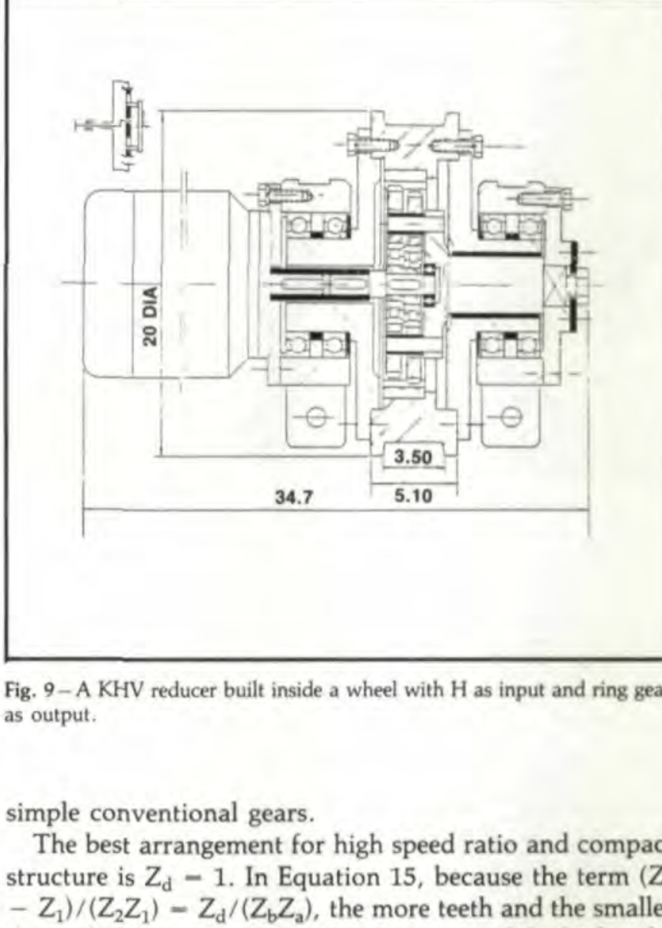

Fig. 9—A KHV reducer built inside a wheel with H as input and ring gear as output.

The best arrangement for high speed ratio and compact structure is $Z_d = 1$. In Equation 15, because the term $(Z_2 - Z_1)/(Z_2Z_1) = Z_d/(Z_bZ_a)$, the more teeth and the smaller the teeth difference, the smaller this term and the higher the η^h_{ab}(or η_{12}), which is an important factor determining efficiency of the KHV. The best arrangement is also $Z_d = 1$ and a large number of teeth or high speed ratio. Therefore, it might be possible for the KHV with $Z_d = 1$ to obtain both high efficiency and high speed ratio. Through optimum programming, the KHV can reach an efficiency of 92% with speed ratio up to 200.

The maximum speed ratio of a single stage CYCLO is 87, which might be limited by the table of the grinding machine

(continued on page 48)

Finishing of Gears by Ausforming

Maurice F. Amateau, Pennsylvania State University and Raymond A. Cellitti, R.C. Associates

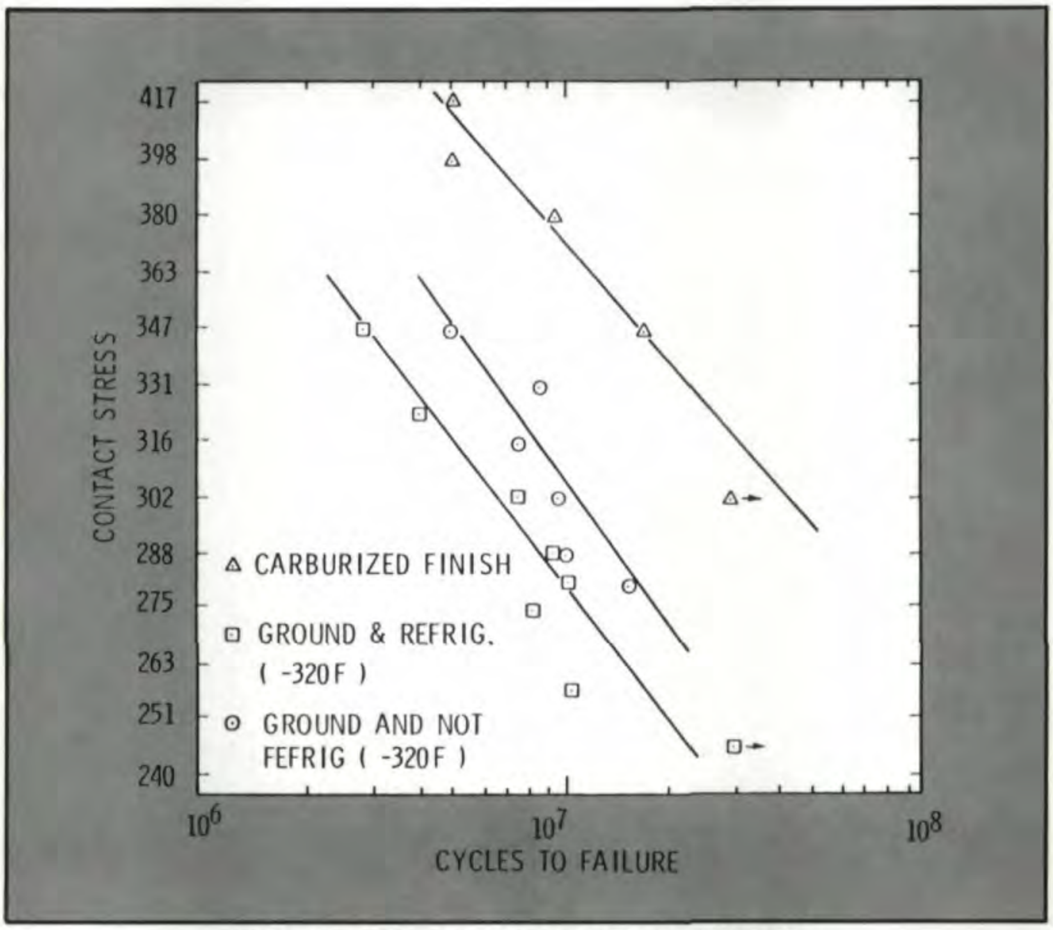

Fig. 1—Pitting fatigue of 9310 steel carburized to 0.90% carbon.

Abstract:

A low temperature thermomechanical finishing process now under development has the potential for producing longer gear life and stronger, more reliable and lower cost gears. This process ausforms the carburized surface of gear teeth while in the metastable austenitic condition during a gear rolling operation. A high degree of gear accuracy is possible by the combined use of interactive forming, low temperature processing and precision gear rolling dies. Thermomechanical processing can eliminate the need for hard gear grinding, thus offering substantial reductions in finishing costs as well as improvements in pitting fatigue life.

Introduction

Almost all machines or mechanical systems contain precision contact elements such as bearings, cams, gears, shafts, splines and rollers. These components have two important common requirements: first, they must possess sufficient mechanical properties, such as, high hardness, fatigue strength and wear resistance to maximize their performance and life; second, they must be finished to close dimensional tolerances to minimize noise, vibration and fatigue loading. As these requirements become more stringent, the cost of processing can increase significantly. The combination of mechanical properties and dimensional control is normally accomplished by separate manufacturing steps, each of which may partially nullify one or more of the desirable characteristics induced in the prior operation. For example, highly loaded precision gears are generally carburized to minimize plastic deformation and wear. They are then ground to achieve the necessary dimensional control, remove high temperature transformation products and improve surface finish. The optimum mechanical properties are developed in the as-carburized condition; thus, the finish grinding operation, which is required for achieving dimensional requirements, precludes achieving maximum strength and reliability. This point is illustrated by examining the pitting fatigue behavior of carburized 9310 steel subjected to various surface treatments.[1] (See Fig. 1.) The unground, as-carburized finish results in the best fatigue performance compared to the as-ground surfaces. Maximum dimensional stability and resistance to plastic deformation are achieved when the microstructure of the steel is totally martensitic. Diffusional transformation products, such as ferrite, lead to low hardness, as does untransformed austenite. The untransformed or "retained" austenite is also detrimental to dimensional stability, as it can eventually transform under mechanical stress to martensite, resulting in a volume change. Cryogenic treatments after normal quenching are often used to control the amount of retained austenite, as seen in Fig. 1. Such treatments can also have a detrimental effect on fatigue life.

Improvements in performance, reliability and manufacturing costs could be realized if surface finishing and strengthening were accomplished in a single process. Certain low temperature thermomechanical treatments, such as ausforming, have this potential. Ausforming treatment is applied to the steel while it is in the metastable austenitic condition prior to quenching to martensite. The principal benefit of this treatment is that it can produce significant improvements in strength without degrading toughness and ductility. In some steels, toughness is actually improved simultaneously with strength. The low temperature nature of this treatment produces very little thermal distortion, thus making it ideally suited for precision finishing operations. The ausforming process can be directly substituted for groups of conventional finishing processes, saving considerable cost while optimizing mechanical performance. Fig. 2 compares the processing steps for finishing gears by conventional methods with those required for ausforming. Ausform finishing can eliminate grinding, shaving, shot peening, honing and cryogenic treatments. The importance of eliminating grinding for optimized rolling contact fatigue life is illustrated in Fig. 3. Removal by grinding of more than 1 mil of surface material from 8620 steel carburized to 1.2C can severely reduce fatigue strength. This phenomenon can originate from two sources. First, the fatigue resistant compressive layers of carburized material are removed, and, second, incipient grinding damage in the form of microcracks and transformation products is generated.

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In addition to ausforming, other low temperature thermomechanical treatments for precision finishing are possible. Low temperature deformation during the decomposition of metastable austenite to bainite, known as isoforming, will result in significantly different toughness and strength characteristics than ausforming. Numerous other low temperature treatments are also possible, providing a large range of processing options and resulting effects.

Current Status of the Technology

The strengthening effect of ausforming is attributed to the inheritance of much of the dislocation structure and carbide

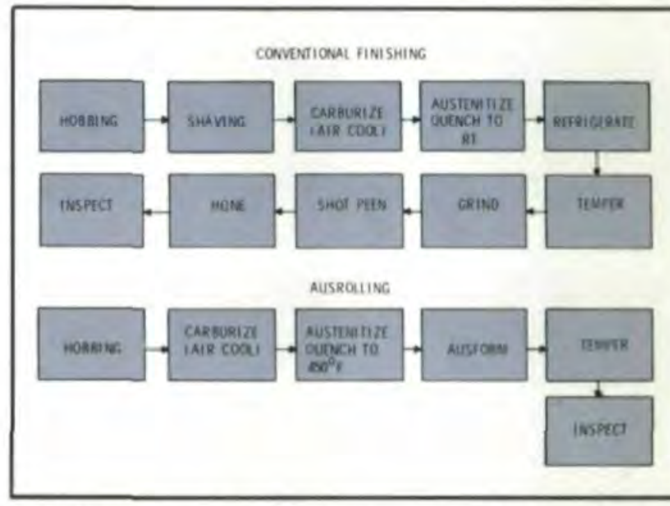

Fig. 2 – Comparison of manufacturing steps – conventional versus ausforming finishing of precision gears.

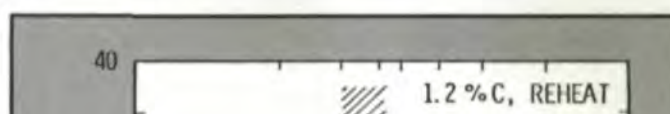

Fig. 3 – Fatigue life as a function of the amount of material removed. 8620 steel carburized to 1.2%C and reheat treated from 1550°F. All tests at 347 ksi.

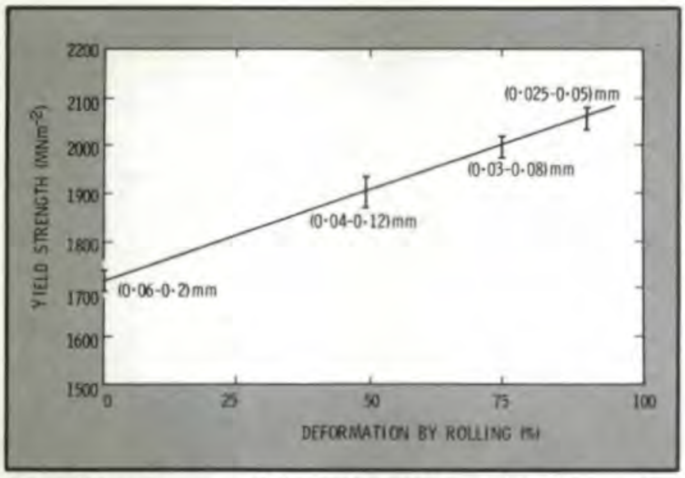

Fig. 4 – Effect of amount of deformation at 510°C on the yield strength of a 0.32C-3.0Cr-1.5Ni-1Si -0.5Mo steel. A range of martensite plate sizes (in brackets) was investigated at each deformation.[5]

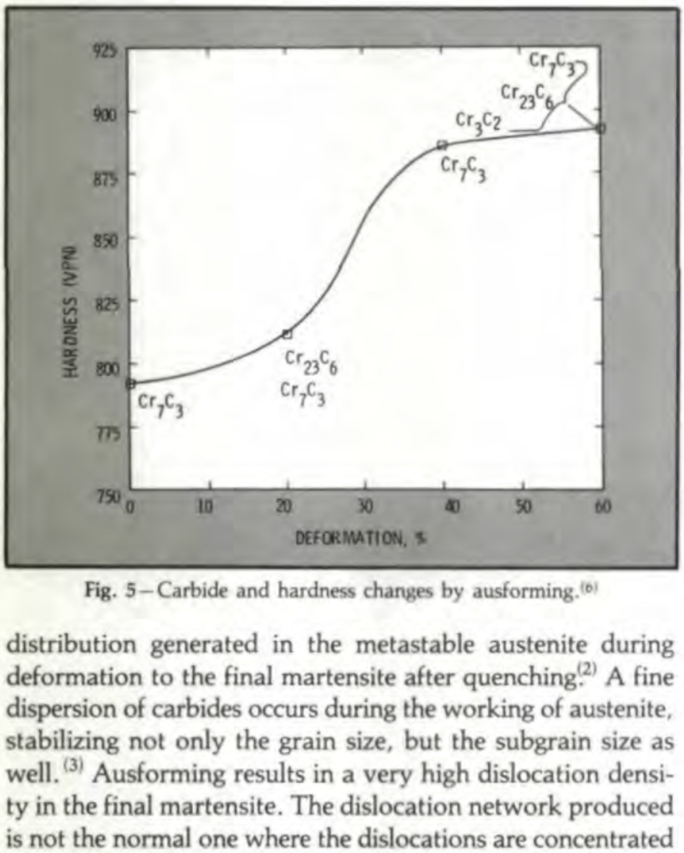

Fig. 5 – Carbide and hardness changes by ausforming.[6]

distribution generated in the metastable austenite during deformation to the final martensite after quenching.[2] A fine dispersion of carbides occurs during the working of austenite, stabilizing not only the grain size, but the subgrain size as well.[3] Ausforming results in a very high dislocation density in the final martensite. The dislocation network produced is not the normal one where the dislocations are concentrated at the cell walls; rather, they are more uniformly dispersed. Larger scale microstructure effects also play an important role in the strengthening process. The low working temperatures (in the range from room temperature to 1100°F, depending on alloy composition) tend to restrict austenite grain growth and, ultimately, the martensite plate size.[4-5] These effects combine to result in an increase in yield strength with increasing amounts of plastic deformation. The effect of the plastic deformation of metastable austenite on resulting martensitic plate size and strength for 0.32C-3.0Cr-1.5Ni-1.5Si-0.5Mo steel is shown in Fig. 4.[5] Each particular steel composition has a saturation limit above which very little further strengthening occurs. For a high-carbon, high-chromium cold-worked die steel (D2 tool steel), this saturation effect is

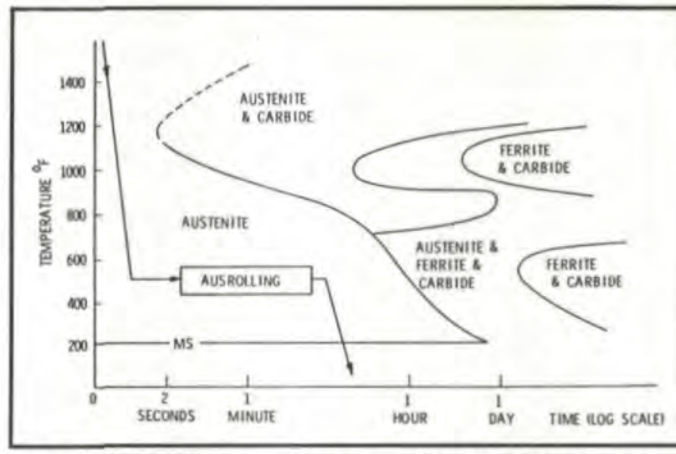

Fig. 6 – Schematic illustration depicting ausrolling time-temperature regime quench from austenitizing temperature.

associated with a change in carbide composition.[6] The change in hardness and carbide form over the range between 0 to 60% deformation is seen in Fig. 5.

The two principal requirements for a steel to respond effectively to the ausforming treatment are the presence of some carbide forming elements and the persistence of the austenite in the metastable form – that is, the existence of a deep austenite bay region in the isothermal time-temperature-transformation (T-T-T) curve – for a sufficient period of time to permit the required amount of deformation. Fig. 6 is the T-T-T curve for a 1.0C-1.2Cr-3.25Ni-0.13Mo steel (carburized 9310) exhibiting such a bay region. This steel has been successfully surface ausformed. Numerous highly alloyed steels are potentially ausformable; however, for low alloy steels, alloy modification or careful selection may be required. Some existing through-hardened steels that may prove to be satisfactory are 51B70, 4370 and 1580. There are also several low-alloy steels specifically developed for ausforming, one of which is the Si-Mn-Mo-V-Cu steel developed at RARDE over ten years ago.[7]

At the present time, it is not clear that thermomechanical forming near the martensite start temperature (M_s) is detrimental to the final properties. For conventional ausforming, working close to the M_s is not common practice since deformation can raise the effective martensite transformation temperature and promote the transformation reaction. In this case, a considerable volume of strong and brittle martensitic material would undergo deformation. If ausforming is confined to only the superficial layers, the formation of some martensite may be tolerated; and, indeed, may even be desirable.

Surface ausforming of carburized steel has been already demonstrated. Similar success may be possible for other surface conditions such as nitriding and carbonitriding. There is no experience available to suggest the effect of surface composition, case thickness or gradient on the properties of termomechanically treated surfaces.

Ausforming can have a significant impact on the tempering response of steel. These effects could be utilized advantageously to minimize processing operations and maximize mechanical properties. Two consequences of ausforming are the low carbon content and low degree of tetragonality of the subsequent martensite. These result from the precipita-

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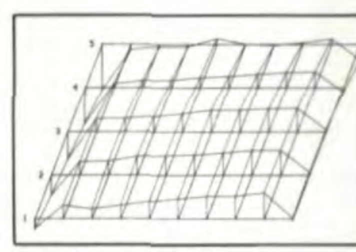

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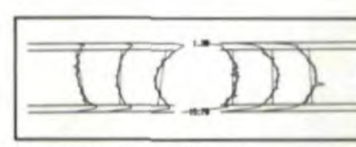

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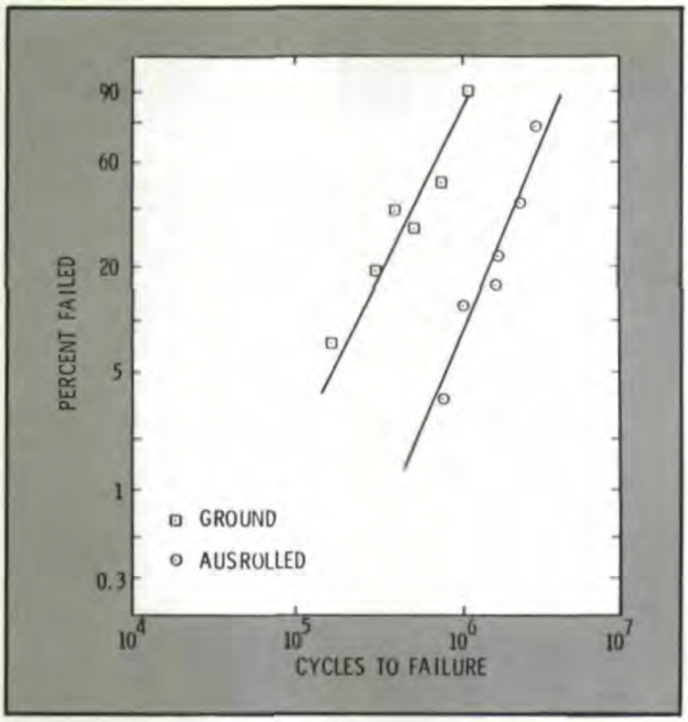

Fig. 7—Rolling contact fatigue of EX15 steel at 450,000 psi contact stress.

tion of carbides during working. Ausforming steels of carbon contents above .3% can eliminate Stage I tempering, which would occur during the tempering of conventionally quenched martensite; thus, autotempering appears to be promoted by ausforming. For surface ausforming, optimum tempering treatments are most likely not those used for conventionally developed martensite.

At the present time, low temperature thermomechanical treatments have very few industrial applications. The reason is that substantial amounts of plastic deformation are required; thus, making it difficult to achieve uniform deformation in complex parts and requiring high forging presures and considerable energy expenditure. Applying low temperature thermomechanical treatments only to the surface of parts overcomes both of these limitations, rendering ausforming an ideal processing method for precision machine elements.

Ausforming significantly improves the rolling contact pitting fatigue strength of EX15 steel. Fig. 7 shows the Weibull distribution of rolling contact fatigue failures at a constant contact stress of 450 ksi for both ausformed and conventionally quenched and tempered steel. A significant improvement in fatigue life can be seen.

The principal features of the apparatus for precision ausforming of gears are interactive forming utilizing closed-loop control and a high accuracy forming die. A schematic diagram of this system is shown in Fig. 8. In this apparatus, the gear rolling die is driven by a hydraulic motor, and the workpiece gear is engaged and driven by the die. A special lead profile is generated on the die gear to produce a swagging action on the workpiece as it is being fed into the die. This geometry, illustrated in Fig. 9, shows a crown section on the bottom half of the lead and is used in the final finishing operation.

The control architecture allows for control signals for the vertical and horizontal feed to be either independent or dependent on each other. Command signals can be dynamically modified based on feedback signals from each of the four transducers. The current system is supervised through a 9600 baud serial link to the external microprocessor. The system is under actual position control at all times, but is in virtual load control during the deformation operation by use of the external microprocessor which supervises the generation of the command signal. The surface austenitization is accomplished with a 10 KHz AF induction system. Thermal monitoring is performed with a noncontacting fiber optic IR sensing pyrometer. This system is capable of measuring temperature over the range from 600 to 1800°F using a zinc sulfide detector. The system response time is one millisecond, permitting continuous reading of the temperature over gear

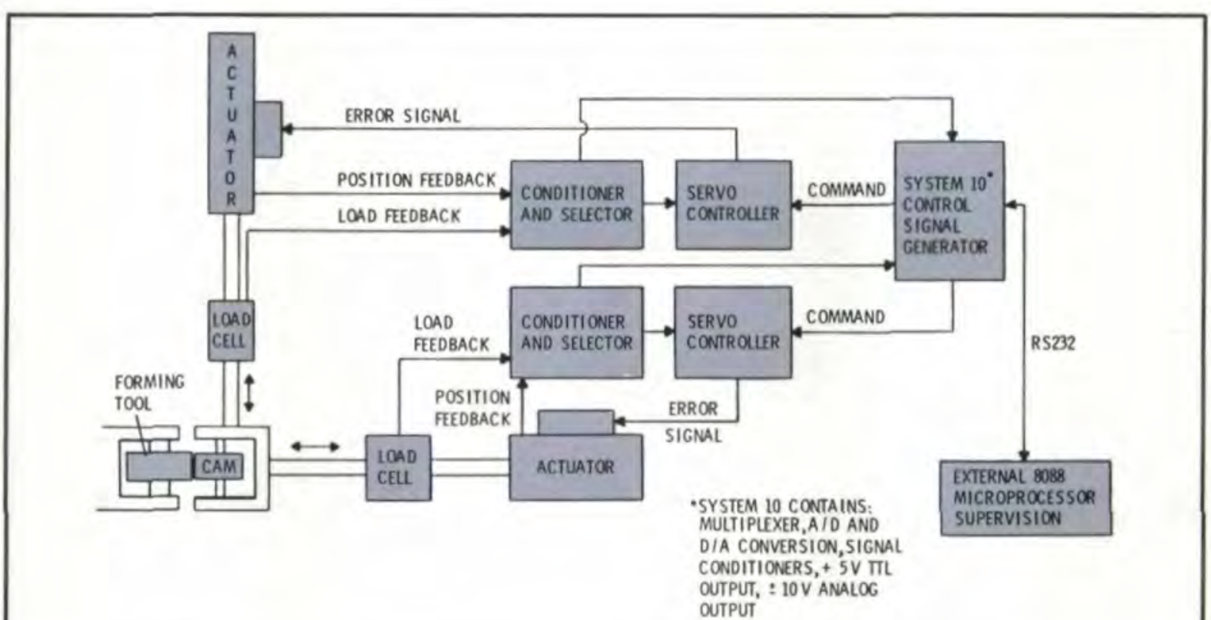

Fig. 8—Control system for interactive gear forming.

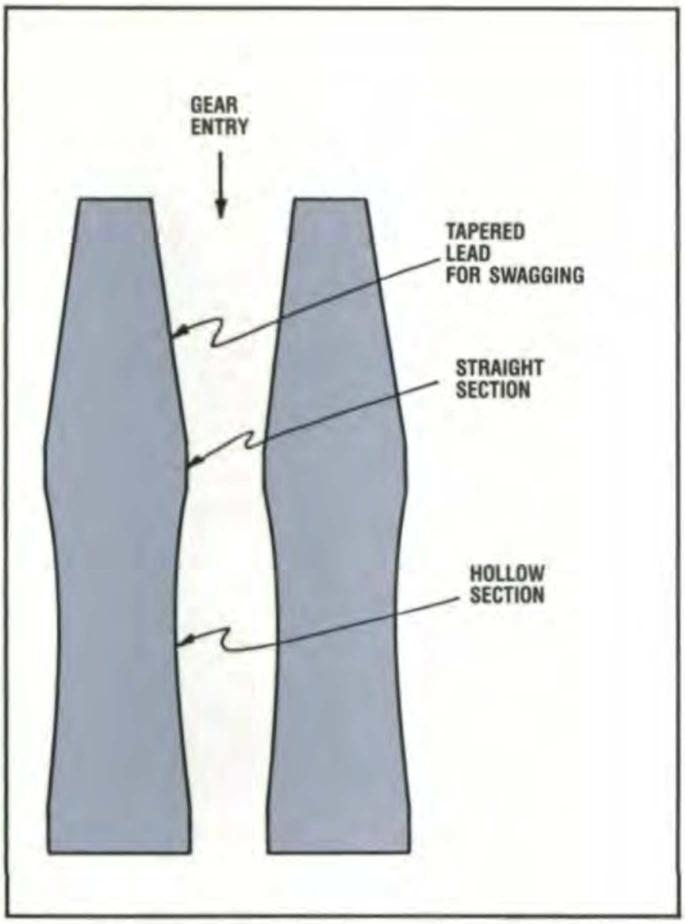

Fig. 9 – Special lead design for swagging and crowning.

tooth profile. A 0-10 volt analog output is used as a control input for real time process control.

Preliminary results on the finishing of gears by rolling in the metastable austenitic condition have indicated a potential for substantial improvements in gear accuracy. Processing has been performed on a hobbed gear with an initial gear quality of approximately AGMA 8. The microstructural examination of the finished gear indicated that approximately ½ to ¾ mil of material was severely worked by the treatment. Considerable improvements in surface finish have also resulted. An average starting surface roughness of 43.2 microinches CLA has been transformed to an average roughness of 11.7 microinches after processing. The retained austenite at the surface of an ausformed gear is about 7%, which is similar to that achieved by conventional finishing that includes both cryogenic treatment and shot peening. The hardness profile of the surface ausformed gear tooth was identical to the conventionally treated gear below the depth of ausforming.

The initial experiments provided great insight into the potential economic advantages of thermomechanical finishing. The ausforming gear rolling operation takes 51 seconds to complete, during which time a crown contour can also be developed on each tooth. This compares to approximately 38 minutes of grinding time to finish this same gear without the crown. If crown grinding were performed, the finishing time would be closer to one hour. Since significant deformation can be achieved by thermomechanical working, additional cost savings can be realized by the elimination of shaving operations or by requiring less dimensional control during rough cutting or hobbing operations. In the case of gear finishing by rolling, a significant cost savings may be realized in the inspection phase by eliminating the need for match setting of gears.

Conclusion

Low temperature thermomechanical processing has considerable potential for the finishing of precision machine elements to net shape. The feasibility of applying ausforming, one such process, to the finishing of spur gears has already been demonstrated. In this process, the final dimensions and finish quality are achieved by thermomechanically working the carburized cases of gear teeth while they are still in the metastable austenitic condition prior to quenching to martensite. The results of such processing methods are (1) the elimination of several manufacturing steps including grinding and hard finishing, (2) the generation and retention of surface compressive residual stresses, and (3) the achievement of ausform strengthening in the surface layers subjected to the high operating stresses. The following benefits are derived from these effects: significantly lower manufacturing costs, greater yield strength, improved fracture resistance, greater pitting and bending fatigue strength and greater product reliability.

The encouraging results from previous studies justify additional research and development to refine and implement the technology and to extend it to the other machine elements such as bearings, splines, cams, rollers, clutch surfaces and shafts.

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Selection of Hobbing Data

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Introduction

The art of gear hobbing has advanced dramatically since the development and introduction of unique machine and tool features such a no backlash, super rigidity, automatic loading of cutting tools, CNC controls, additional machine power and improved cutter materials and coatings. It is essential to utilize all these features to run the machine economically.

The following guideline is an attempt to systemize the modern knowledge of hobbing and to assist gear process engineers in selection and determination of various unknowns for cutting gears most efficiently. Obviously, it is an impossible task to take into consideration all the variables influencing machine, cutter, speed and feed selection. There are also differences of opinion regarding the best ways to hob gears. Nevertheless, this guideline can serve as a starting point for a large variety of applications and sizes. The recommendations are based on the assumption of using adequate hobbing machines, rigid fixtures and TiN coated hobs made of M35 or similar material.

Cutter Selection

Selection of number of starts. The number of starts of the hob depends on pitch, number of teeth, stage of manufacturing, gear quality and divisibility with number of teeth of the gear. Generally, the productivity increases when hobs with greater number of starts are employed.

Table 1 shows multi-start hobbing limitations depending on pitch and number of teeth.

Additional limitations can be set for finish hobbing in reference to AGMA quality.

In the case of non-hunting ratio combinations between the number of teeth of the gear and number of starts of the hob, the maximum number of starts should not exceed two.

Additional limitations for pre-shave hobbing:

The number of starts should be tested for hunting ratio combination, and in case of a non-hunting ratio, the number of starts should be reduced until a hunting condition is reached.

For AGMA quality less than 11, the number of starts can equal 2, even in the case of non-hunting ratio combinations.

For AGMA quality greater than 10, the number of starts should not exceed 3. (See Table 2.)

Selection of number of hob gashes. The more numbers of gashes, the more cutting edges are engaged, resulting in less cutter load and wear. Greater numbers of gashes also increase the number of enveloping cuts, which make the involute smoother. This is especially important when cutting gears with small numbers of teeth. Table 3 shows minimum recommended number of gashes depending on number of hob starts.

Selection of hob diameter. Smaller hob diameter usually results in less cutting time since greater hob RPM can be achieved. Limiting factors for reducing further hob diameter are usually minimum number of gashes and lead angle. Maximum lead angle for hobs with straight gashes is usually within 6 to 8 degrees. Sometimes for high production, hob diameter has to be increased to enable utilization of a longer hob, since very high length over diameter ratio may undermine the system rigidity.

Hob accuracy class selection. Hob accuracy selection is based on AGMA gear quality, stage of manufacturing and number of starts. Class accuracy should be equal or better than the accuracies shown in Tables 4 and 5.

Selection of the Machining Data

The selection of the number of cuts is influenced by pitch, number of gear teeth, stage of manufacturing, surface finish and gear quality. A two-cut cycle can be selected for gears with a module greater than 4.5. For small numbers of teeth (less than 12) and a module greater than 4, a two-cut cycle should also be selected. The number of cuts for finish hobbing can be selected from Fig. 1.

Hob speed is usually selected based on cutter material and coating as well as workpiece material, hardness and pitch. Fig. 2 can be used to determine machinability in reference to workpiece material and Brinell hardness, assuming the cutting tool is made of M35 or the equivalent.

Based on machinability and module, one can determine hob surface speed from Fig. 3.

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Table 1 – Multi-start hobbing limitations.

	LIMITATIONS	
No. of Hob Starts	Max. Module (NDP)	Min. No. of Teeth
2	5.5 (4.61)	13
3	4.5 (5.64)	17
4	3.0 (8.46)	20
5	2.5 (10.16)	25

Table 2 – Number of starts in relation to AGMA quality level.

AGMA Quality Level	12	11	10	9	8	7	6
Max. No. of Starts	1	1	1	2	2	3	3

Table 3 – Minimum gashes in relation to hob starts.

Number of Starts	1	2	3	4	5
Min. No. of Gashes	9–12	12–13	13–17	17–19	17–21

Table 4-5 – AGMA class accuracy vs. hob starts.

Selection for 1 or 2 Start Hob			**Selection for 3 to 5 Start Hob**		
	Finish	Pre-Shave		Finish	Pre-Shave
Class C	3	4	Class C	3	4
	4	5		4	5
	5	6		5	6
Class B	6	7	Class B	6	7
	7	8			
Class A	8	9	Class A	7	8
	9	10		8	9
	10	11			10
					11
Class AA	11	12			
	12				

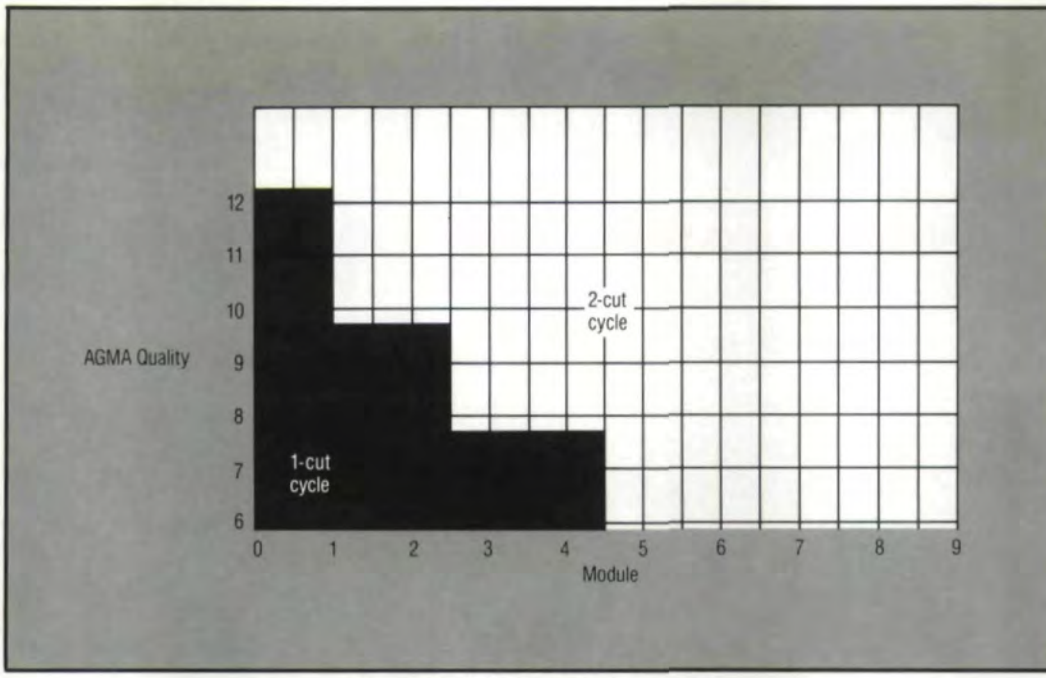

Fig. 1—Finish hob cut selection chart.

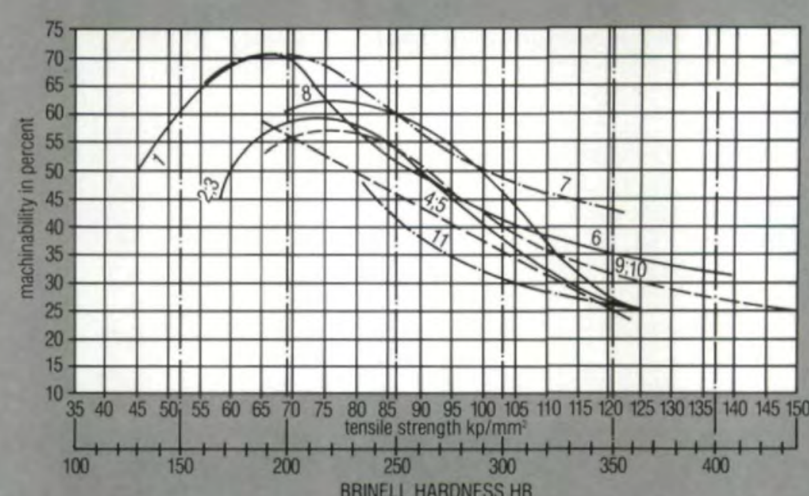

Table of Machinability of Material

(AISI Designations)

1. Carbon steels 10xx, 11xx, 12xx, 15xx
2. Low alloy chrome-nickel steels.
3. Nickel steels 23xx, 25xx.
4. Chrome-nickel steels.
5. High alloy chrome-nickel steels, 31xx, 33xx, 303xx stainless steel.
6. Chrome-nickel-molybdenum steels 43xx, 47xx, 86xx, 87xx, 93xx, 98xx.
7. Nickel-molybdenum steels 46xx, 48xx.
8. Chrome-molybdenum steels 41xx.
9. Chrome steels 50xx, 51xx, 511xx, 521xx, 514xx, 515xx.
10. Chrome-vanadium steels 61xx.
11. Silicon-manganese steels 92xx.
12. Ductile iron.

Fig. 2—Machinability in reference to workpiece material.

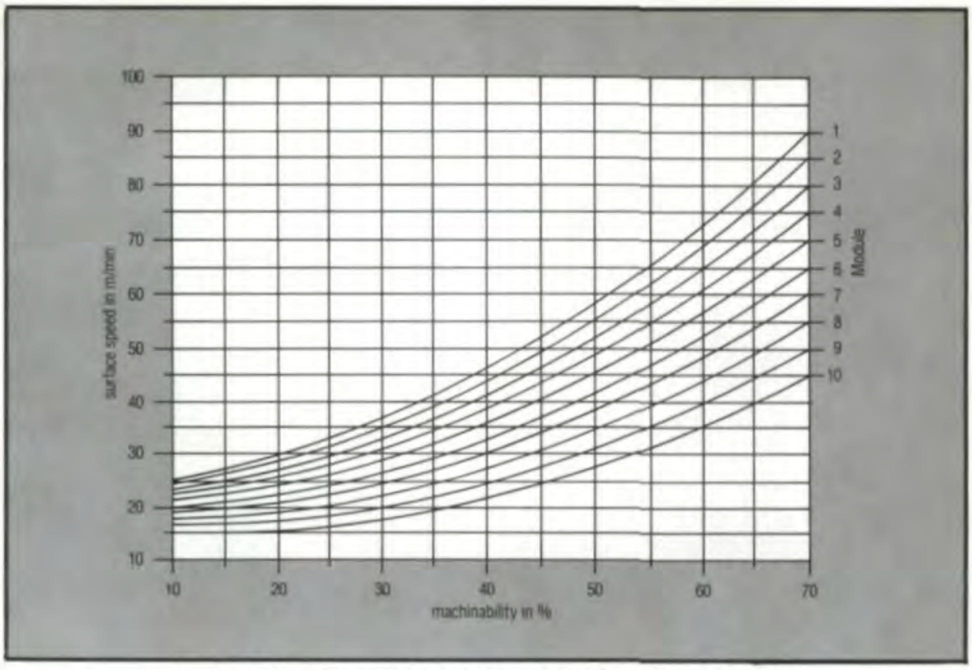

Fig. 3—Hob surface speed in relation to module and machinability.

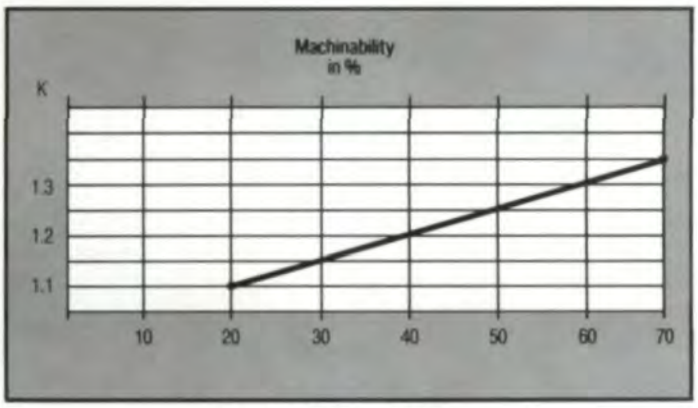

Fig. 4—Multiplier K chart.

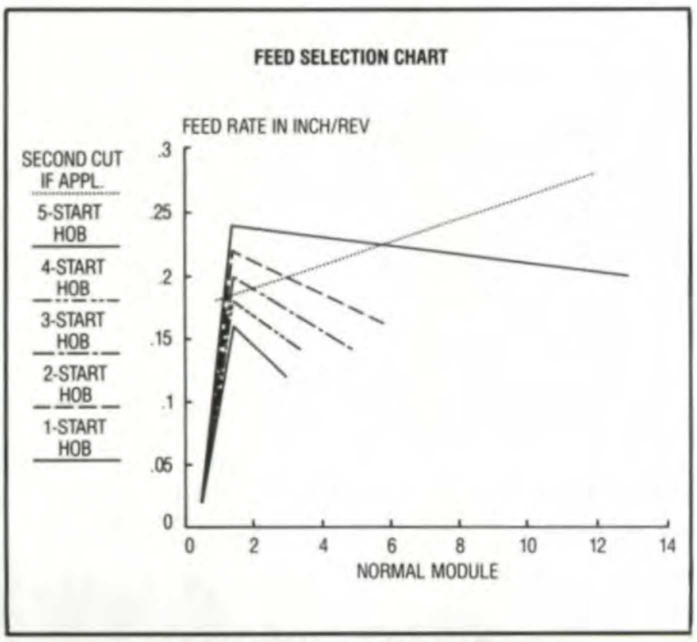

Fig. 5—Roughing cut feed rate chart.

For TiN coated hobs, the speed can be increased depending on the machinability for all materials (except ductile iron) according to the multiplier K selected from Fig. 4 (*K-multiplier*).

Feed Selection

Feed rate for roughing cut can be selected according to Fig. 5. The selected rate should be multiplied by the machinability factor C1 and the factor C2 for small number of gear teeth.

$$C1 = \left(\frac{\text{Machinability}}{70}\right)^{.4}$$

Gear teeth factor should apply only if number of teeth is less than 25.

Gear teeth factor

$$C2 = \left(\frac{\text{Number of teeth}}{25}\right)^{.3}$$

Feed rate for finish cut should be limited by allowed feed scallop depth. Maximum allowed feed rate for finish cut can be calculated with formula:

$$\text{Feed rate} = \cos\beta \bullet \sqrt{\frac{\delta \bullet 4 \bullet \text{DAO}}{\text{SIN } \lambda}}$$

Where

β = helix angle
δ = allowed feed scallop depth
DAO = hob diameter
λ = pressure angle.

For pre-shave cut, the feed scallop depth should not exceed .0008". If approach feed control rate is available, it can be set to roughing feed rate.

BACK TO BASICS...

Good Gears Start With Good Blanks

Wilford E. Maples
Star Cutter Company
Farmington Hills, MI

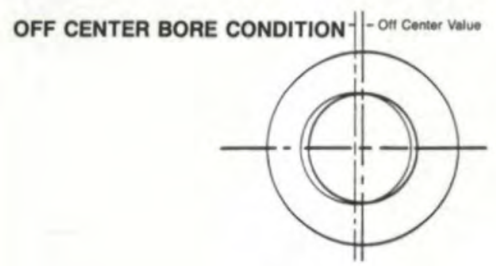

Fig. 1

Introduction

The quality of the finished gear is influenced by the very first machining operations on the blank. Since the gear tooth geometry is generated on a continuously rotating blank in hobbing or shaping, it is important that the timed relationship between the cutter and workpiece is correct. If this relationship is disturbed by eccentricities of the blank to its operating centerline, the generated gear teeth will not be of the correct geometry. During the blanking operations, the gear's centerline and locating surfaces are established and must be maintained as the same through the following operations that generate the gear teeth. **This centerline of the manufacturing operations must also be the same as the operating centerline of the gear as it is used.** (See Fig. 1.)

Gear Blank Design

The gear design engineer can assist in assuring good quality finished parts by designing gear teeth on well proportioned blanks. The configuration of the blank should be one that allows the workpiece to be supported just inside the root diameter of the gear teeth during cutting operations. Another consideration in the blank is to avoid thin cross section areas between the gear teeth and the operating bearing area that may warp or distort unevenly during heat treatment. A well proportioned blank with the adequate locating area and uniform cross section compared to a poor blank design is shown in Fig. 2. The poor design has very little support material behind the gear teeth and a thin web cross section. This blank is an extreme example that would be difficult to locate accurately for hobbing and shaving. It would also be distorted considerably during hardening.

Gear Blank Inspection

During the blanking operation, adequate gaging must be done to insure that the locating surfaces of the blank are maintained perpendicular and concentric to the centerline of the gear. **It is good manufacturing practice to do finish turning and facing operations on the blank prior to cutting the gear teeth.** In the case of a hole type gear, the blank should be finish cut on the locating face or faces, and the bore finish cut perpendicular to the face and concentric to the outside diameter. Typical tolerances for an automotive blank 25 mm thick and 100 mm outside diameter would be:

1. Locating face squareness to bore 0.01 mm to 0.02 mm.
2. Outside diameter concentricity to bore 0.12 mm maximum.
3. Bore size tolerance for 25 mm bore 0.01 mm to 0.02 mm.
4. Bore size taper tolerance for 25 mm bore 0.005 mm to 0.008 mm.

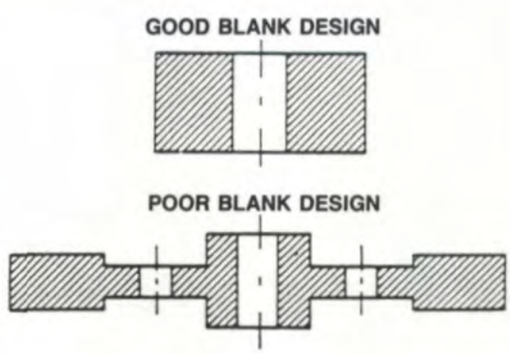

Fig. 2

AUTHOR:

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Table 4-1—Typical Gear Blank Tolerances*

Blank Dia. In.	Face Runout In.	Hole Size In.	Hole Taper In./In.	Hole Roundness In.-Max	O.D. In.-Max.	O.D. Runout In.
Up to 1, 1-in. Thick	0.0003-0.0005	0.0003-0.0006	0.0002-0.0003	0.0002-0.0003	0.003	0.003
1 to 4, up to 1-in. Thick	0.0004-0.0008	0.0005-0.001	0.0002-0.0003	0.0003-0.0005	0.005	0.005
4 to 8	0.0006-0.0012	0.0008-0.0012	0.0002-0.0003	0.0004-0.0006	0.005	0.007
8 to 12	0.001-0.002	0.001-0.0015	0.0002-0.0003	0.0005-0.0007	0.005	0.008

*Tolerances for Specific Gears Should be Selected in Accordance with Quality Requirements.

Fig. 3

SAMPLE PINION GEAR BLANK CHECK

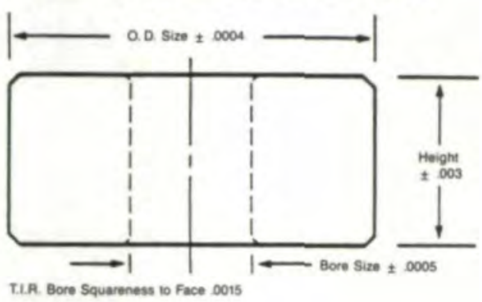


Fig. 4

INTERNAL GEAR BLANK SAMPLE CHECK

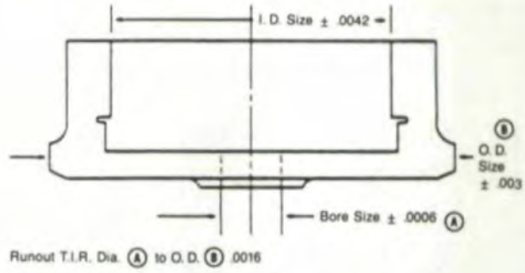


Fig. 5

BASIC ARBOR FOR CHECKING PINION BLANK

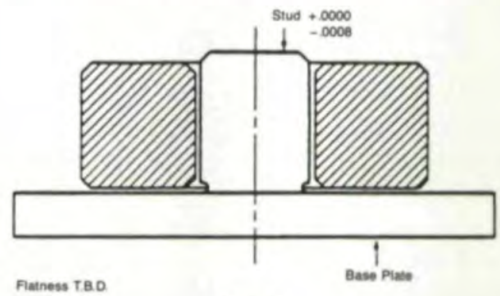


Fig. 6

Fig. 3 is a table of different gear blank tolerances. Fig. 4 is a sample pinion gear blank, while Fig. 5 is a sample internal gear blank.

The blank tolerances should be specified by the manufacturing engineer separately from the finished part print tolerance. Gaging fixtures with precision indicators should be supplied for the machine operator's use to check alignments, concentricities and perpendicular accuracies of locating surfaces to the centerline of the blank. The gaging fixtures should be placed at each finish turning machine for constant use by the machine operator.

Fig. 6 illustrates basic arbor for checking pinion blanks, while Fig. 7 is a photograph of a fixture to check face runout and O.D. runout.

The bores or bearing journals should be inspected at their finish machining stages using air gages or precise indicator gages. It is important to check both the size of the bore or bearing journal and its configuration. A tapered or out-of-round condition will result in poor workholding efficiency if the blank is driven by clamping in the bore or on the bearing journal during the gear generating process. Fig. 8 illustrates various bore configurations to check.

When production rates are high, automatic gaging is used to check each blank for size, concentricity and perpendicular locating surfaces. The automatic gaging of each blank is the most precise method of gear blank gaging. The operator's influence is removed from the gaging operation and all decisions are made by the instrument. There is also the advantage with automatic gaging that all elements are checked by the gage at a high rate of speed.

Metallurgical Considerations

Various types of materials and metal preparation are used for gear blanks; specifically, forgings, castings, cold forming or basic bar stock. In the case of cold forming, sometimes an in-process heat treatment may be necessary to improve part machinability. Also, if a part has been turned on very high speed equipment, work hardening may be experienced, and again, in-process heat treatment may be required. There are various other examples, such as making a blank readily machinable for hole broaching, but having poor tool life when the gear teeth are generated. Each individual process will have to be experimented with, and experience will ultimately determine if either metallurgical change is called for or an in-process heat treatment should be employed.

Gear Blank Related Errors

The errors caused by gear blank inaccuracies can be illustrated by assuming that a perfectly aligned hobbing machine with accurate tooling is used to cut gears from random blanks. The gear blanks are simple hole type flat gears that are clamped in the bore and located on the faces. Presenting a gear blank to the precision hobbing machine with the bore accurate for size and shape, but the locating faces not perpendicular to the bore centerline, will result in the gear blank wobbling during the hobbing operation. The inaccuracy of the locating faces causes the centerline of the bore to be skewed from the generating centerline of the hobbing machine. The resulting lead inspection would produce a chart that shows high lead variation. If tooling, specifically the hob, is improperly mounted to an extreme runout condition, a part with excessive involute error could be produced, as illustrated in Fig. 9.

Presenting a gear blank to the hobbing machine with excessive concentricity error between the outside diameter and the bore centerline will result in uneven cutting load around the blank. The resulting gear teeth will then show a like concentricity error when a red line check is performed on the teeth; that is, when the teeth are checked from the centerline of the bore and/or rolled with a master gear.

Presenting a blank to the hobbing machine with an oversize bore will result in poor clamping in the bore. The blank will then have a tendency to slip on the arbor, resulting in mutilated or off lead gear teeth as shown in Fig. 10.

In the case where the blank is centralized on the bore and clamped on its faces, the blank centerline will be shifted off the centerline of the hobbing machine. The subsequent concentricity check of the gear teeth to the centerline of the bore will show excessive concentricity error.

In the high volume production of gears, such as in the automotive industry, different locating areas are used, specifically in pinion hobbing. Here the fixture holds the piece

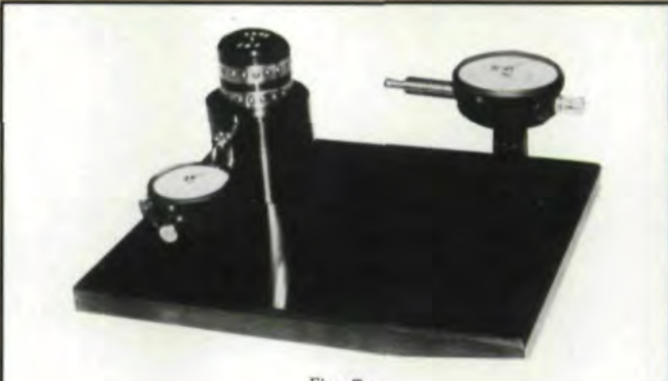


Fig. 7

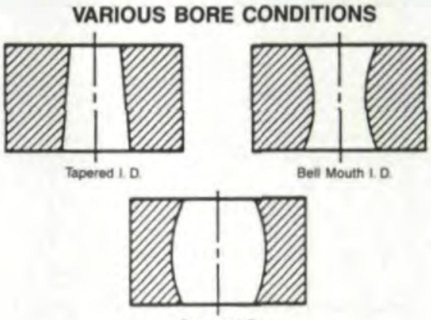


Fig. 8

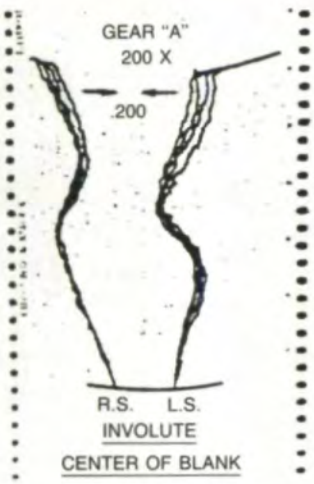


Fig. 9

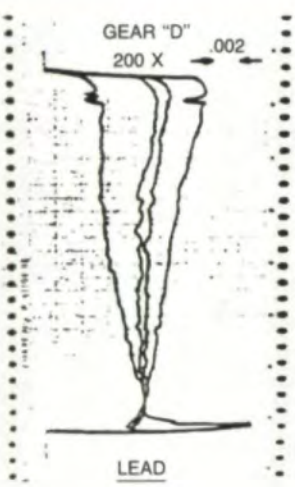


Fig. 10

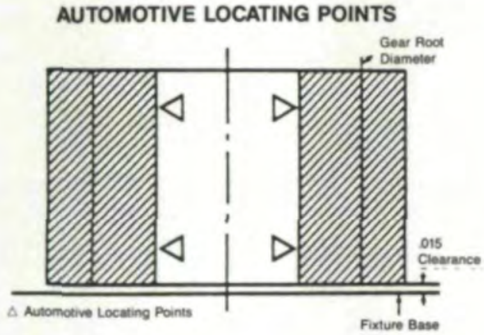


Fig. 11

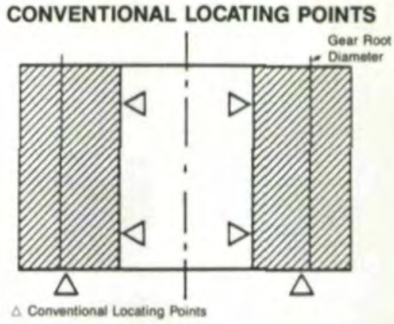


Fig. 12

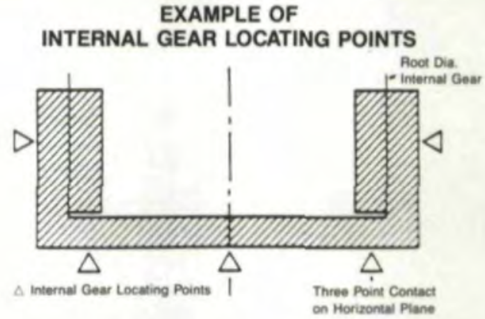


Fig. 13

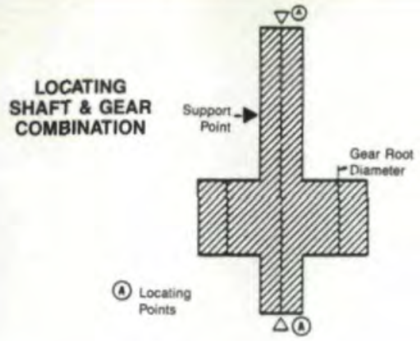
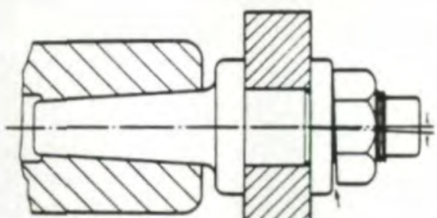
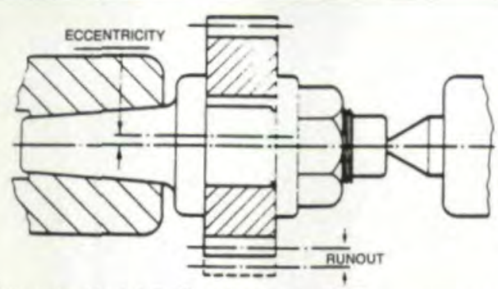


Fig. 14



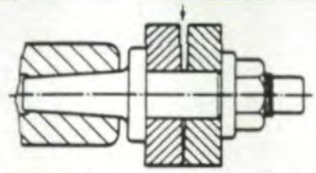
Lock-up nut is not square to holding plate. Therefore, arbor is bent.

Fig. 15



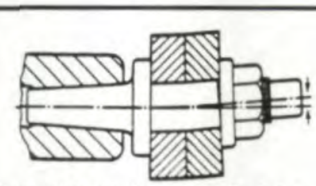
Oversize blank I. D. Therefore, part will have gear runout.

Fig. 16



Blanks not squarely mounted. Results in lead, involute and concentricity errors.

Fig. 17a



Bent arbor results in lead, involute and concentricity errors.

Fig. 17b

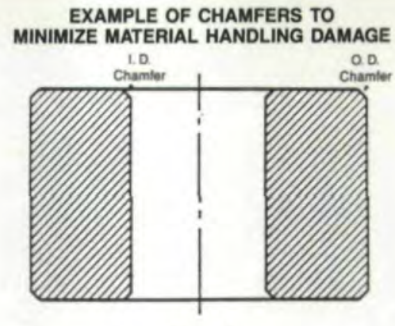


Fig. 18

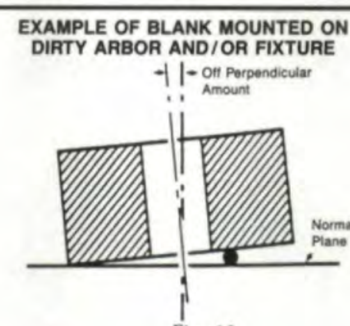


Fig. 19

part in the bore only and does not locate on the face area. (See Fig. 11.) The conventional method is to locate the piece part on the bore as well as the lower face, as illustrated in Fig. 12. Locating points for internal gears are shown in Fig. 13, while locating points for shaft and gear combination are shown in Fig. 14.

Several examples of improperly mounted gear blanks are illustrated in Figs. 15 through 17.

Another point that must be considered is material handling abuse. The gear blanks should be designed with adequate chamfers at the outside diameter edges and locating surface corners. (See Fig. 18.) Nicks, burrs, dirt and any other foreign material on the locating and clamping surfaces will cause the part centerline to skew, resulting in lead variations and excessive concentricity errors. (See Fig. 19.)

Summary

The quality of the finished gear is affected by the quality of the blank on which the gear teeth are cut. The operating centerline is the centerline from which the gear is designed to run; therefore, once a manufacturing centerline is established, it must remain the same as the operating centerline.

Good gear practice by the design engineer is helpful to the manufacturing engineer in producing high quality gears at lower manufacturing costs. The manufacturing engineer must remember that all errors in the locating surfaces and clamping surfaces of the blank are reflected in the quality of the finished gear. Good inspection and quality control procedures will result in early detection of bad blanks. The poor blanks can then be removed from the system before subsequent expensive operations are performed on them.

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(continued from page 31)

and the strength of the equal velocity mechanism. Most materials for the CYCLO are high carbon chrominum steel with high hardness, which requires an elaborate heat treatment and an accurate grinding. The KHV has involute teeth with a concave to convex contact (internal gear contacts external gear), which results in a high contact strength. The internal gear has a very strong bending strength, and the external gear is modified, which increases the bending strength. Therefore, the KHV can use conventional gear materials and common accuracy. Usually, the ring gear is carbon steel without grinding, and the planet is either carbon or alloy steel. All the gears of the KHV can be generated by standard gear cutters and conventional equipment.

Compared to the CYCLO, the KHV is easier to manufacture and less expensive. Therefore, it is a very promising gearing.

Many thanks to Professor Ali A. Seireg of the University of Florida and the University of Wisconsin for his review of this article.

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VIEWPOINT

(continued from page 7)

ERRATA

Editors Note: We apologize that several errors appeared in recent issues of GEAR TECHNOLOGY. We regret any inconvenience these errors may have caused.

In the article **"Longitudinal Load Distribution Factor for Straddle- and Overhang-Mounted Gears"** by Toshimi Tobe, et al, appearing in our Jul/Aug issue, the following errors occurred:

Equation 17 should read

$$k \equiv (P_n\ /b)/[w_1 + w_2 + w^P]_{max}$$
$$k - K_{H\beta}{}^{-0.96}[k] -_{K_{H\beta}}$$

Equation 19 was ommitted. It should read,

When both tooth surfaces just come into contact at $\eta - \eta^*$, *the position* η^* is obtained from

$$\frac{ds_o(\eta)}{d\eta}\ _{\eta=\eta^*} - 0$$

as follows:

$$\eta^* - \frac{1}{e_1c_2^2+e_2c_1^2}\left\{e_1(f_1+c_1)\ c_2^2+e_2\ (f_2+c_2)c_1^2-\frac{c_1^2\ c_2^2}{2_{b2}}\ e_{eq}\right\}$$

In Equation 20, the figure $e_2\ c_2^1$ should read $e_2\ c_1^2$

The letter ϵ in Figs. 9 and 15b should read ξ

The first equation in the footnote on the bottom of page 16 should read $K_{F\beta} - K_{H\beta}^N$

In the note in Table B.1 the symbol lG/in should read lb/in.

In the caption for Fig. 14, the last equation should read $c_1/b_1 - 0.5,\ e_o^* - 2\ 0\mu m$.

In the last two paragraphs on page 46, Fig. B.1 is incorrectly labelled A.1.

In the Sept/Oct issue, on page 50 of Stan Jakuba's **"SI Units — Measurements and Equivalencies"** the figures 1kg-m under "Moment of force . . ." 1kg/dm under "Specific force of gravity" and 1 kg-m/rad under "Spring rate: torsional" all refer to kg force, a symbol sometimes written as kp.

In this same issue, page 47 was incorrectly laid out. The table of terms at the top of the page and Equations 3 and 11 immediately below it both are part of the article, **"Selection of a Proper Ball Size. . ."** by Van Gerpen and Reece, continued from page 34.

Equation 3 should read:

$$\text{PACB}$$
$$- K^* \left[\frac{\pi}{Z} - \left[\frac{(\text{ BTN}}{\text{Cos (BHA) *BD)}}\right] + K^*\text{Inv(PACP)}\right]$$
$$+ \text{PACP)} \qquad (3)$$

The remainder of the equations on page 47 (Nos. 12-19) belong to Paul Dean's article, **"Interrelationship of Tooth Thickness Measurements as Evaluated by Various Measuring Techniques,"** continued from page 23.

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