GEAR
CUTTING PRACTICE

The quality of the materials used in the manufacture of this book is governed by continued postwar shortages.

GEAR CUTTING PRACTICE

Methods of Producing Gears for Commercial Use Including Wartime Data Supplement

BY

FRED H. COLVIN

Editor Emeritus of American Machinist; Author of "American Machinists' Handbook"; Member, American Society of Mechanical Engineers, Franklin Institute

AND

FRANK A. STANLEY

Consulting Engineer; Formerly Editor of Western Machinery and Steel World; Formerly Associate Editor of American Machinist; Author of "American Machinists' Handbook," "Punches and Dies," etc.

SECOND EDITION
FIFTH IMPRESSION

McGRAW-HILL BOOK COMPANY, INC.
NEW YORK AND LONDON
1943

GEAR CUTTING PRACTICE

PRINTED IN THE UNITED STATES OF AMERICA

THE MAPLE PRESS COMPANY, YORK, PA.

PREFACE TO THE SECOND EDITION

In the second edition, a Wartime Data Supplement has been added. Much of the development shown in this section is a natural outcome of the steady progress made in the field of gear cutting. But the wartime demand for better and faster methods has played its part in bringing improvements in this and other lines.

Airplane instruments use a large number of small gears and pinions and will be aided by the new type of gear shaper illustrated in this section. For this reason several details of this machine, the methods of holding small work, and the data on the tooth proportions of fine-pitch gears are shown.

In the same way space has been given the new methods of flame-hardening the teeth of both bevel and spur gears by machines developed especially for that work. All these developments are doing their share in war production.

THE AUTHORS.

NEW YORK,
February, 1943.

PREFACE TO THE FIRST EDITION

Both the increasing use of gearing and the greater demands made for strength, efficiency, and quietness make it necessary for the mechanic to be familiar with the latest methods used by gear makers. Only enough of the theory of gearing is given in this book to make the terms used clear to the reader without previous gear experience.

All of the data given have been secured from what are believed to be the best sources of information and the authorities are given, both as a source of reference and as an acknowledgment of the assistance received. The recommendations and the standards of the American Gear Manufacturers Association are quoted freely as these represent standard practice in many shops. Automobile practice is also given, both in the selection and heat treatment of the steels used and in the methods used, because this represents advanced methods of making gears that are strong and quiet.

Small-shop gear cutting practice also receives careful attention, including the cutting of spur, bevel, and worm gears on the standard milling machine. Much of the general information as to cutters and cutting coolants or lubricants will be found useful in shops of any size. The same is true of the tables of tooth parts and of feeds and speeds that have been found satisfactory in well-known plants.

THE AUTHORS.

NEW YORK, N. Y.
April, 1937.

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CHAPTER I

GEARS AND GEAR CUTTING PRACTICE

Gears have become a most important factor in machine construction. While they have always been a necessary element in machine design, the demand for gears that will transmit large amounts of power in small spaces, and that will be practically unbreakable and at the same time noiseless, has greatly increased the problems of the gear maker. Automobile transmissions, in which from 65 to 250 hp. are transmitted within a very limited space and where minimum noise is also demanded, are excellent examples of modern gear making. Not only must narrow-faced gears transmit large power, but the gear teeth must stand the abuse incident to being shifted and clutched by unskilled hands. Clashing of gears that would have meant disaster a comparatively few years ago has almost no effect on modern gears. The gears used in the power units of many machines must have similar qualities as to power transmitted, freedom from breakage, and lessening of noise.

Steel Gears.—While the machine shop is chiefly concerned with the machining of gears, it must be remembered that much of the success or failure of gears depends on the material of which they are made and on the heat treatment they receive from the time the steel leaves the steel mill until the gears are ready for assembly. Excellent results are obtained in many places by the use of low-carbon steel with the teeth carburized and made fairly hard. These gears can be heat treated with a minimum of distortion. Where the load is excessive or the treatment in use is severe, as in automobile gears, alloy-steel gears are used almost exclusively. Some of the steels used for these gears, together with the heat treatment recommended, are given later. While some steels are harder to machine than others, modern tools handle them all without difficulty.

Other Materials.—While the present demand for great strength and toughness has led to the general use of steel for gears, cast iron is still a good material for many purposes. Bronze is also used for gears in special work where corrosion is a factor, and monel metal and the aluminum alloys also have their place. Rawhide gears, which formerly were used largely to reduce noise, have given way to those made of Formica, Textolite, Micarta, and other products of the bakelite type. These are made up of layers of paper or cloth, impregnated with resins of the bakelite order. They are stronger than might be supposed and fill a large field where reduction of noise is important. These materials are treated in the same way as metals so far

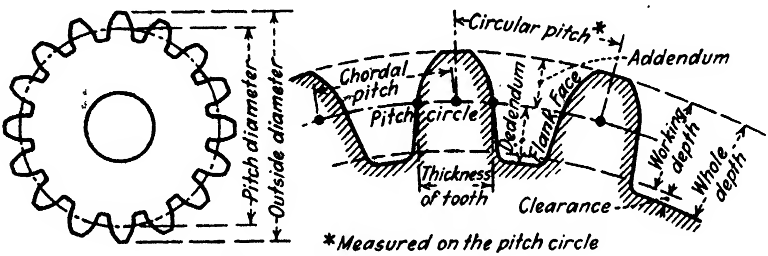


Fig. 1.—Names of gear-tooth parts.

as the cutting of the teeth is concerned, except that cutter speeds may vary according to the material. As they mate with metal gears, they are machined in the same manner.

Tooth Curves and Pressure Angles.—Nearly all gears now used are made with a modification of the involute curve, which is explained on page 5. Varying conditions and the differing ideas of engineers cause modifications with which the shop has little to do. It is, however, advisable to become familiar with the terms used and the names of the different parts of the gear teeth, as shown in the diagram, Fig. 1. Simple definitions of tooth parts are also given. On later pages the very complete nomenclature of the American Gear Manufacturers Association will be found for reference where needed. It is especially advisable to understand the difference between diametral pitch and circular pitch, and to bear in mind that the pitch diameter is more important than, and entirely different from, the outside diameter. For, although the pitch diameter is not visible to the eye any more than in the case of a screw thread, it must always be considered in gear cutting. While, as can be seen in Fig. 1, the pitch diameter is usually above the center of the

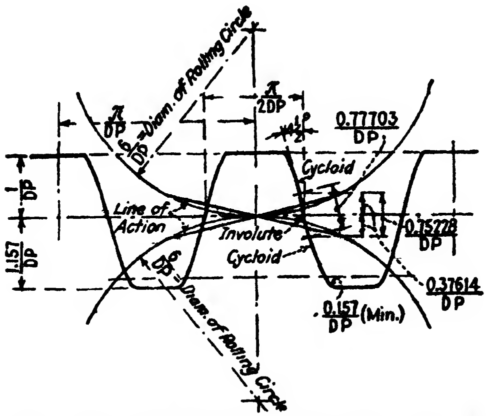


FIG. 2.—Basic rack for 14½-deg. full-depth tooth—composite system.

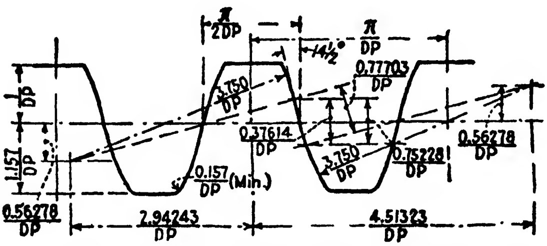


FIG. 3.—Approximation to basic rack for 14½-deg. full-depth tooth—composite system.

tooth depth by the amount of the clearance, this is not always the case, as some gear teeth are designed with the pitch line much nearer the top of the teeth in order to give more metal at the base of the tooth. This is usually done in the case of gears having comparatively few teeth. Some of the composite tooth forms approved by the American Standards Association are also shown in Figs. 2 to 6.

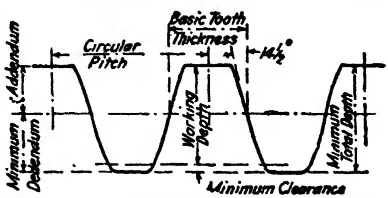


FIG. 4.—Proportions of full-depth spur-gear teeth; 14½-deg. composite system.

Shapes of Gear Teeth. *Cycloidal or Epicycloidal.*—A curved tooth generated by the point of a circle rolling away from the gear wheel or rack.

Involute.—A curved tooth generated by unwinding a tape or string from a cylinder. The rack tooth has straight sides.

Involute Standard.—The standard gear tooth has a 14½-deg. pressure angle, which means that the teeth of a standard rack have straight sides 14½ deg. from the vertical.

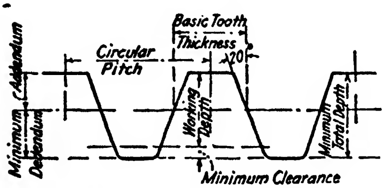


Fig. 5.—Basic rack for 20-deg. stub-tooth involute system for spur gears.

Involute—Stubbed.—A tooth shorter than the standard and usually with a 20-deg. pressure angle.

Teeth and Parts of Gears.

Addendum.—Length from pitch line to outside diameter.

Chordal Pitch.—Distance from center to center of teeth in a straight line.

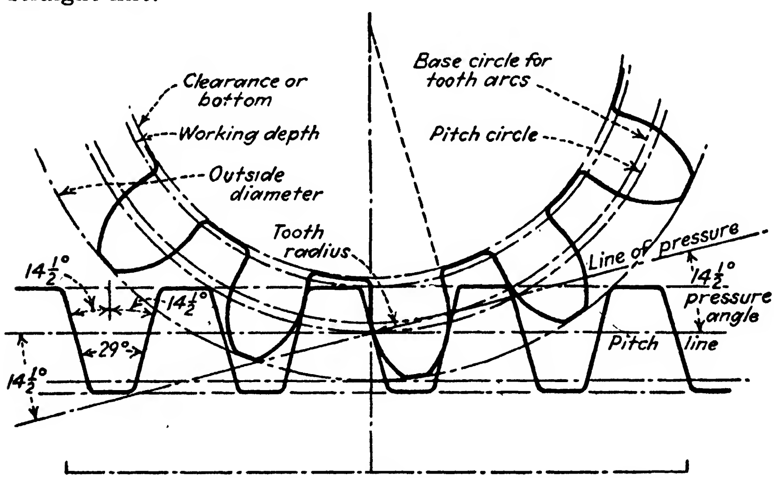


Fig. 6.—Action of standard 14½-deg. tooth gear and rack.

Circular Pitch.—Distance from center to center of teeth measured on the pitch circle.

Clearance.—Extra depth of space between teeth.

Dedendum.—Length from pitch line to base of tooth.

Diametral Pitch.—Number of teeth divided by the pitch diameter or the number of teeth to each inch of diameter.

Face.—Working surface of tooth outside of pitch line.

Flank.—Working surface of tooth below pitch line.

Outside Diameter.—Total diameter over teeth.

Pitch Diameter.—Diameter at the pitch line.

Pitch Line.—Line of contact of two cylinders which would have the same speed ratios as the gears.

Linear Pitch.—Sometimes used in rack measurement. Same as circular pitch of a gear.

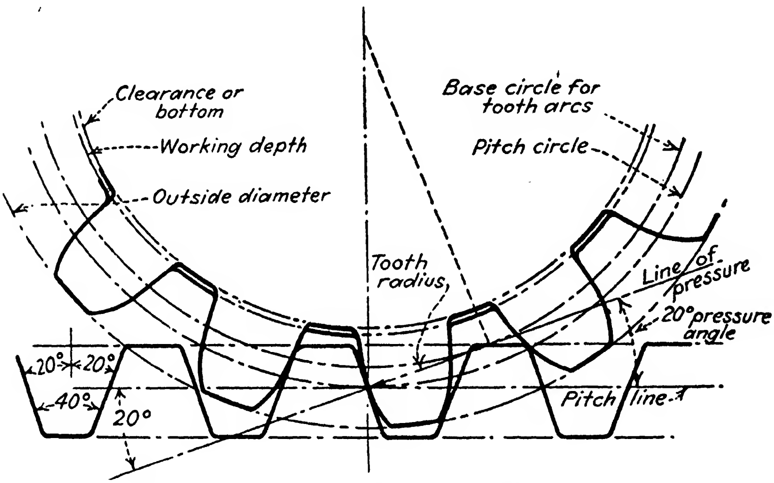


FIG. 7.—Action of 20-deg. stub-tooth gear and rack.

Profiles of Gear Teeth.—These illustrations contain very complete data as to part names and proportions of teeth. It is stated that in Fig. 3 the maximum profile error is 0.0022 in. divided by the diametral pitch. The reason for modifying the curve slightly is that by so doing it is more easily reduced to template or cutter form.

Data on page 4 show the practice recommended by the American Gear Manufacturers Association for a basic 14½-deg. rack are proportions for spur gears of this pressure angle. Figure 5 shows basic rack for 20-deg. teeth.

Pressure angles may be a bit confusing. These are the angles at which one tooth presses against another, as in Figs. 6 and 7. The standard pressure angle is generally considered as 14½ deg., but 20 deg. and other angles are also commonly used. Gear teeth are also made of different lengths. Standard tooth dimensions are given in Tables 4 to 6 and the length of stub teeth in Table 9. It will be seen that, instead of being a fixed proportion of the depth of the standard tooth, the stub tooth is given a definite relation to the standard tooth. This gives the diametral pitch in fractional numbers as 3/4, which means that if the diametral pitch is three, it has only the depth of a four standard pitch.

Terms Commonly Used.—In order to avoid confusion regarding terms used in connection with gearing it seems advisable to quote some suggestions of the National Twist Drill & Tool Co. written to avoid misunderstandings in the shop. These apply to both spur and bevel gearing. The calculations are based on gears having 60 teeth and 5 diametral pitch.

To say or write 5 pitch, would mean 5 diametral pitch.

To say or write 5″ or 5-in. pitch, means 5-in. circular pitch.

The diametral pitch of a gear is the number of teeth to each inch of its pitch diameter.

The circular pitch of a gear is the distance from the center of one tooth to the center of the next tooth, measured along the pitch circle.

To find diametral pitch, divide the number of teeth by the pitch diameter—$60 \div 12 = 5$ pitch.

Having diametrical pitch, to find circular pitch. Divide the decimal 3.1416 by the diametral pitch—$3.1416 \div 5 = 0.628$ circular pitch.

Having circular pitch, to find diametral pitch. Divide the decimal 3.1416 by the circular pitch—$3.1416 \div 0.628 = 5$ diametral pitch.

To find pitch diameter, divide the number of teeth by the diametral pitch—$60 \div 5 = 12$-in. pitch diameter.

To find outside diameter, add to the pitch diameter 2 parts of the pitch—$12 + 2/5 = 12\frac{2}{5}$-in. outside diameter.

To find number of teeth, multiply the pitch diameter by the pitch—$12 \times 5 = 60$ teeth.

To find the distance between the centers of two spur gears, divide half the sum of the teeth of both gears by the diametral pitch—

$$60 + 60 = 120 \div 2 = 60 \div 5 = 12\text{-in. centers.}$$

To find the thickness of a tooth at pitch line, divide 1.57 by the diametral pitch, or divide the circular pitch by 2.

The whole depth of a tooth is 0.6866 of the circular pitch for standard depth teeth.

Having properly sized a gear for cutting, note that the cutters are made to cut an additional depth of one-tenth the thickness of the cutter at the pitch line, for clearance. Thus, in a 5-pitch cutter the thickness would be 0.3142 in. The clearance provided would, therefore, be one-tenth of this thickness, or 0.031 in.

Example: $\frac{1}{10}$ of 0.3142 in. = 0.031 in. clearance.

The gears must be set to run with this amount of clearance to work well.

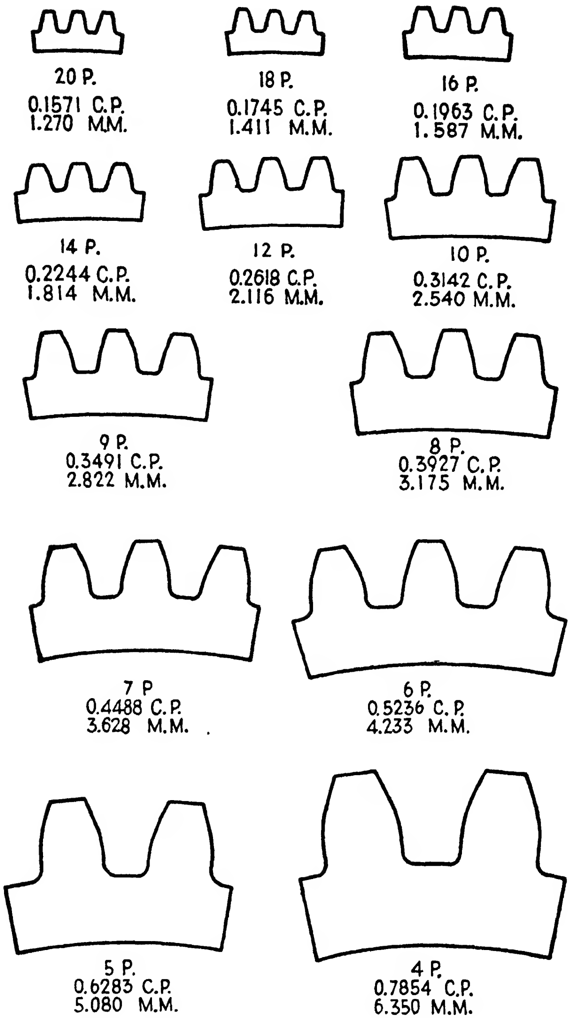
In order to make it easy to visualize the different sizes of gear teeth, Fig. 8 shows the outline of the sizes most commonly used. By remembering that the distance between teeth is approximately 317 times the spacing of a thread of similar pitch, it may be easier to compare one with the other.

Pitch Diameter.—One of the features about gearing most confusing to the beginner is to understand just what the pitch diameter really is. Perhaps the easiest way is to take two perfectly smooth disks and roll their edges together. Where the disks touch is the pitch or contact line. If, now, we cut shallow teeth in both disks and fasten pieces equivalent to the metal we have cut away to the tops of the teeth of each disk, we have two gears with the teeth meshing into each other as in Fig. 9. The distance between the centers of the disks is the same as before, but instead of smooth-friction disks we now have teeth that drive with a positive motion. The point of contact of the teeth remains the same as it was before the teeth were formed; this contact is called the pitch line. We have, however, added to the outside of the diameters of both gears, which shows why the outside diameter or O. D., as it is commonly called, is larger than the pitch-circle diameter.

With teeth of standard length the outside diameter is always "two pitches larger than the pitch diameter," or P. D.

Gears are generally designated by the number of teeth *per inch of diameter at the pitch line.* A 10-pitch gear has 10 teeth for every inch of gear diameter at the pitch line. If it is 4 in. in pitch diameter, the gear has 40 teeth and the outside diameter is 4 in., plus 2 parts of the pitch, or 4 in. plus 0.2 in., or $4\frac{1}{5}$ in.

This refers to gears where the working length of the tooth is the same both above and below the pitch line. There is also a clearance at the bottom of the tooth space, generally equal to $\frac{1}{10}$ of the thickness of the tooth at the pitch line. The clearance can also be found by dividing the constant 0.157 by the diametral pitch. For the 10-pitch gear just mentioned the tooth clearance at the bottom is 0.0157 in., below the lower half



C.P. = *Circular pitch* M.M. = *m/m module*
P = *Diametral pitch*

Fig. 8.—Some commonly used gear teeth, full size.

of the working tooth, or *dedendum.* The upper half of the working tooth is called the *addendum.* In the Fellows stub-tooth gear the clearance is 1/4 the addendum or dedendum, where they are equal, or 1/8 of the depth of the working tooth. See Table 7.

Measuring Tooth Depth.—Measuring tooth depth and thickness is comparatively simple in the case of spur or straight-tooth gears. But both bevel gears and helical gears add complications. In the bevel gear the tooth depth and thickness vary as the

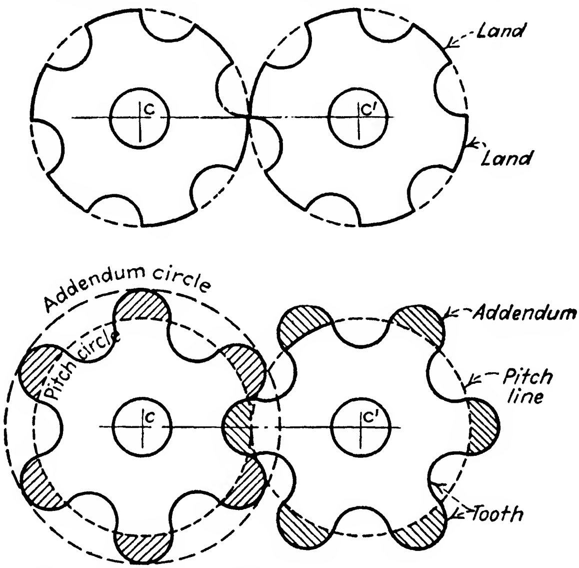


FIG. 9.—Diagram showing pitch line and tooth action.

diameter changes, while in the helical gear the teeth form a helix around the surface, in the same way that a screw thread winds around a bolt or shaft, except that the helix angle is very different. The helix angle in the gear is measured from the axis or center line of the bore in the hub, while in the screw thread the helix angle is measured at right angles to the axis. Helical gears were formerly called spiral gears, which is not correct, and the term helical is now almost universally used.

Types of Gears.—While there are no truly spiral gears, unless we call the scroll of a lathe chuck by this name, the name spiral-bevel gear is used to designate the type of bevel gear designed by the Gleason Works for use in automobile rear axles. The different types of gears are shown in Fig. 10, which includes the

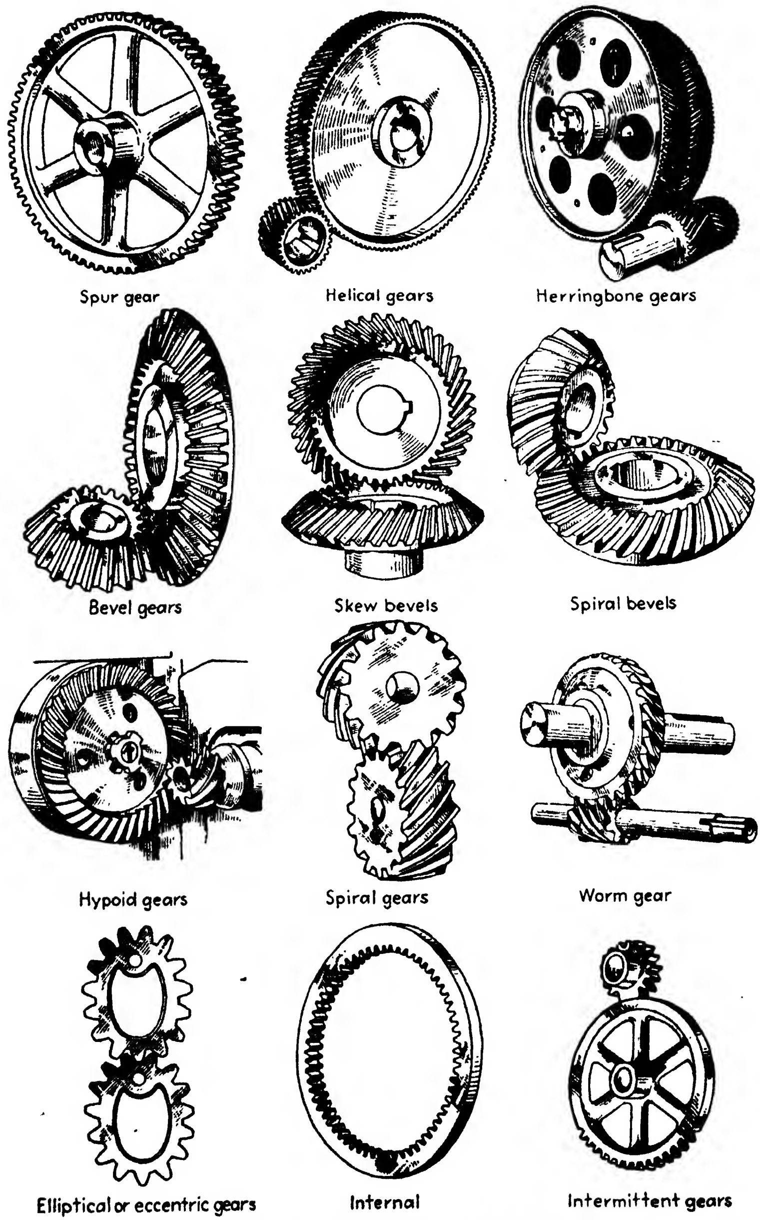


FIG. 10.—Some of the different kinds of gears.

hypoid type of bevel. This is simply a spiral-bevel gear with the teeth so arranged that the driving pinion can be set above or below the center of the large bevel, or ring gear, as it is usually called.

Helical gears are now so widely used that it is advisable to pay particular attention to their characteristics and the way in which they are made. Their advantage is self-evident. Owing to the angle of the teeth, two or more teeth are always in contact, which gives a very smooth drive and reduces noise to a considerable extent. Because of the angle at which the teeth work on

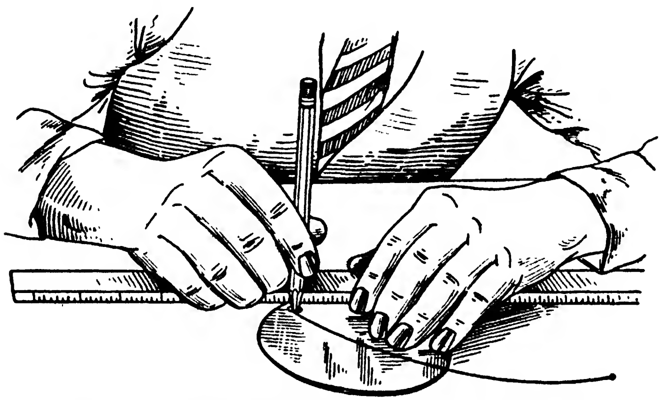


FIG. 11.—Drawing a cycloidal curve for a rack tooth.

each other the helical gear produces a side thrust on the bearings and there is more of rubbing action on the teeth than with the straight spur gear. This is, however, more than offset by the constant drive and the quietness of the helical gear.

How Gear Teeth Are Laid Out.—While it is the duty of the production or shop man to make gears as ordered by the drafting room or engineering department, he should know something of the way in which gear-tooth curves are produced. The following illustrations from the Fellows Gear Shaper Co. clarify this matter: In Fig. 11 a disk with a hole near the edge is placed against a straight edge on the drawing board, with the hole next to the straight edge. Placing a pencil point through the hole and rolling the disk along the ruler draws a curved line as shown. This is known as a cycloid, and was the curve formerly used for gear teeth. A similar curve is being drawn in Fig. 12, except that instead of the straight edge the disk is rolled around another disk. The curve in Fig. 11 gave the curve used in the cycloidal rack and the curve in Fig. 12 the tooth curve for gears of the pitch diameter of the disks. These curves vary with the size

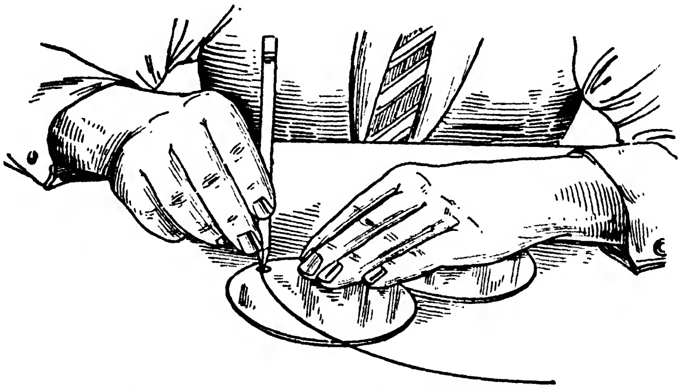


FIG. 12.—Drawing a cycloidal curve gear tooth.

of the disks and also the location of the hole in the rolling disk. The pencil point should be just at the edge of the disk.

The curve now generally used for gear teeth is the involute, which is traced or generated as in Fig. 13. A cord is wrapped around the disk and the pencil point placed in the loop at the

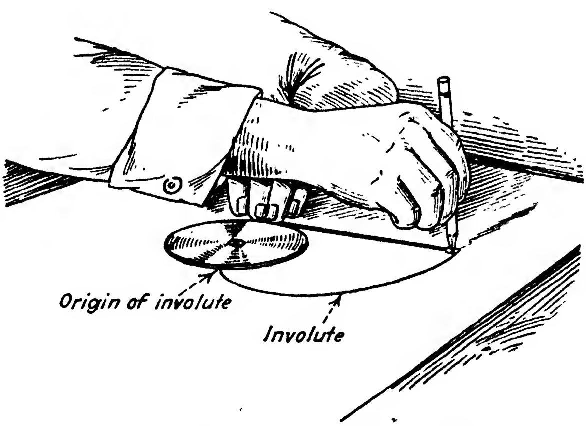


FIG. 13.—How an involute curve is drawn.

end of the cord. The curve traced as the pencil moves away from the disk is called an involute and is shown in Fig. 13. The rack tooth in the involute system has straight sides instead of curved as in the cycloid, in Fig. 11, the angle of the sides with the base depending on the pressure angle desired. The standard pressure angle is 14½ deg., but 20 deg. is also common, and other angles are used in some places. Pressure angle means the angle at which the pressure is applied by the tooth of the driving gear, as it turns the driven gear. This was shown in Figs. 6 and 7.

✓**Gear Cutting Methods.**—Spur and helical gears are produced by three methods: milling, planing or shaping, and hobbing. Hobbing is, however, a form of milling, but is continuous instead of intermittent. The oldest is the use of the milling cutter formed to the shape of the tooth to be cut. After each tooth was cut, the gear blank was indexed and the next tooth cut at the proper distance from the first. Sometimes the indexing was not consecutive, but the blank was moved several teeth to reduce the distortion due to concentrating the heat of cutting in one place.

Although the hobbing method was proposed many years ago, it was not introduced until after the Fellows method of gear shaping was well established. This method uses a cutter which is virtually a hardened gear with proper tooth clearance. The cutter spindle is fed toward the work spindle until the cutter has reached its proper depth and then they both revolve at the proper ratio to have the cutter generate the teeth as they both revolve. The cutter reciprocates in the same way as a tool in the ram of a shaper; hence its name. The gear is completed in one revolution unless a finishing cut is desired. In some of the later machines this second cut is taken automatically. The shaping method works equally well on spur or helical gears.

Hobbing is also a continuous method in which the hob and gear blank revolve together and the cutter is fed across the face of the gear blank as it revolves. As with the gear shaper, this machine can cut both spur and helical gears. Both the hob and the work must be properly geared together to produce the desired results. While some of the originators of this process felt that the method must produce a series of flats on the gear teeth, the system has grown far beyond their dreams and is now widely used.

Bevel gears are cut both with rotary cutter and by planing. Hobbing has also been used for bevel gears but is not common practice. As the width of the tooth of a bevel gear varies from one end to the other, it is clear that a rotary cutter can only be as wide as the narrow end of the tooth. To secure the proper width at the large end it is necessary to take a second cut. To get the proper shape of tooth, or as near an approximation as can be had in this way, the gear blank must be shifted, as will be shown in the section devoted to bevel gears.

It is generally considered that correctly shaped bevel-gear teeth can be secured only by the planing process. Here the planing or shaping tool moves in a path that is radial with the center of the gear blank. The tool is sometimes guided by a hardened-steel form made to the correct tooth form, and several times as large as the tooth itself. In this way any errors in the form become negligible in the tooth itself. The tool automatically makes the tooth of proper width at all points. In other bevel-gear planers the tools generate a true tooth form by moving in the correct paths while the gear blank turns. This is shown in the section on bevel gears. There are, of course, many gears cut with a rotary cutter that are giving satisfaction in their place. But it is evident that they cannot be so nearly correct as where the teeth are planed radially.

Spiral-bevel gears are cut with a special face milling cutter in a special machine. This machine is most ingenious and combines a variety of motions that produce remarkably efficient gears. The machine and the methods used will be shown in Sec. V.

Where gears are cut in fairly large quantities they are now made by the shaping or planing method, as in the Fellows or Sykes machines (neither the Maag nor Sunderland machines are well known in this country) or by the hobbing process. Both of these methods generate the teeth as work and cutter rotate together. For jobbing work and where the cost of hobs or shaping cutters would be prohibitive, the formed cutter is still used, either singly or in small gangs. Gear cutting with these cutters is simply a milling job in which the teeth are spaced by the indexing head.

With the gear blank turned to the right diameter and the cutter centered properly, the cutter is set to cut the blank to the correct depth for the tooth. The cutter is then fed through the blank, which is indexed for each tooth. Where it is feared that the heat generated in the cut might distort the gear blank, it is indexed for several teeth before the next tooth is cut. This distributes the heat around the blank and is called "block indexing." Both spur and helical gears are cut in this way, details of which will be shown in Chap. II. Bevel gears are also cut in this way as will be seen later.

The hobbing process vies with the shaping method in popularity and is used in both spur and helical gear work, as well as in cutting splines, as is the shaping method. Bevel gears have also been cut by hobbing, but this method is not as yet in general use.

Planing or shaping has been used in producing both spur and bevel gears but is now principally confined to the latter type. The Bilgram and Gleason machines are excellent examples of this method.

How Gears Are Measured.—Gears are measured by the pitch diameter and by the number of teeth per inch of pitch diameter. No attention is paid to the outside diameter of the gear except when the gear blanks are being turned. The pitch designates spacing of the teeth with regard to the diameter but does not indicate the distance from one tooth to the next, as is the case with screw threads, except as the diameter is related to the circumference. A six-pitch gear has six teeth for each inch of diameter. If the gear has 60 teeth, it is 10 in. in diameter on the pitch line.

Gears are also measured by the circular pitch, which is the distance between the center of one tooth to the center of the next, on the pitch line, as shown in Fig. 1. This method of measurement is not common, except on large gears of coarse pitch.

Chordal pitch, which is the distance between tooth centers in a straight line instead of on the pitch circle, is now rarely used.

A fourth method is by module, or metric pitch. A module is the pitch diameter in millimeters divided by the number of teeth in the gear. This bears the same relation to diametral pitch that the millimeter bears to the inch. As there are 25.4 mm. to the inch, a 10-module gear corresponds to a diametral pitch of 2.54 teeth.

Tables that follow show the relation between diametral and circular pitch and also between the more commonly used pitches in French modules. The larger tables show the proportions of the various parts of the teeth.

Tables giving the outside diameter of gear blanks are also given to avoid the necessity of making calculations for turning the blanks ready for the cutting of the teeth.

TABLE 1.—DIAMETER INCREMENTS, PLUS AND MINUS

In order to obtain full involute action when pinion is in contact with a 14½-deg. basic rack, the outside diameter of the pinion should be increased and the outside diameter of the mating gear decreased the same amount.

A tabulation of the amount of increase and decrease and the corresponding tooth thickness on the pitch line for both the gear and pinion is given.

CTh = circular tooth thickness.

14½ deg. *PA*. 1 *DP*.

Teeth	Diameter* increment	Pinion *CTh*	Gear *CTh*
8	1.4985	1.9583	1.1833
9	1.4358	1.9421	1.1995
10	1.3731	1.9259	1.2157
11	1.3104	1.9097	1.2319
12	1.2477	1.8935	1.2481
13	1.1850	1.8773	1.2643
14	1.1223	1.8611	1.2805
15	1.0597	1.8449	1.2967
16	0.9970	1.8286	1.3130
17	0.9343	1.8124	1.3292
18	0.8716	1.7962	1.3454
19	0.8089	1.7800	1.3616
20	0.7462	1.7638	1.3778
21	0.6835	1.7476	1.3940
22	0.6208	1.7314	1.4102
23	0.5581	1.7151	1.4265
24	0.4954	1.6989	1.4427
25	0.4328	1.6827	1.4589
26	0.3701	1.6665	1.4751
27	0.3074	1.6503	1.4913
28	0.2447	1.6341	1.5075
29	0.1820	1.6179	1.5237
30	0.1193	1.6017	1.5399
31	0.0566	1.5854	1.5562

* Diameter increment equals amount pinion diameter is to be increased over standard diameter and gear diameter decreased.

Proportions of Gear Teeth.—It is important to know the relation of the different parts of the gear teeth as well as to have tables showing the various dimensions. The proportions adopted by the American Gear Manufacturers Association for 14½-deg. full-depth teeth shown in outline in Fig. 4 (p. 3) are as follows:

PROPORTIONS FOR SPUR GEARS

	For Diametral Pitch	For Circular Pitch
1. Addendum	$= \frac{1}{DP}$	$0.3183 \times CP$
2. Minimum dedendum	$= \frac{1.157}{DP}$	$0.3683 \times CP$
3. Working depth	$= \frac{2}{DP}$	$0.6366 \times CP$
4. Minimum total depth	$= \frac{2.157}{DP}$	$0.6866 \times CP$
5. Pitch diameter	$= \frac{N}{DP}$	$0.3183 \times N \times CP$
6. Outside diameter	$= \frac{N + 2}{DP}$	$0.3183 \times (N + 2) \times CP$
7. Basic tooth thickness on pitch line	$= \frac{1.5708}{DP}$	$0.5 \times CP$
8. Minimum clearance	$= \frac{0.157}{DP}$	$0.05 \times CP$
9. Radius of fillet	$= 1\frac{1}{3} \times$ Clearance	

N = number of teeth. DP = diametral pitch. CP = circular pitch.

TABLE 2.—DIAMETER INCREMENTS, PLUS AND MINUS

In order to obtain full involute action when pinion is in contact with a 20-deg. basic rack, the outside diameter of the pinion should be increased and the outside diameter of the mating gear decreased the same amount.

A tabulation of the amount of increase and decrease and the corresponding tooth thickness on the pitch line for both the gear and pinion is given.

CTh = circular tooth thickness.

20 deg. *PA*. 1 *DP*.

Teeth	Diameter* increment	Pinion *CTh*	Gear *CTh*
8	1.0642	1.9581	1.1835
9	0.9472	1.9156	1.2260
10	0.8302	1.8730	1.2686
11	0.7132	1.8304	1.3112
12	0.5963	1.7878	1.3538
13	0.4793	1.7453	1.3963
14	0.3623	1.7027	1.4389
15	0.2453	1.6601	1.4815
16	0.1284	1.6175	1.5241
17	0.0114	1.5749	1.5667

* Diameter increment equals amount pinion diameter is to be increased over standard diameter, and gear diameter decreased.

For 20-deg., full-depth involute teeth the diameter increments are as shown in Table 2.

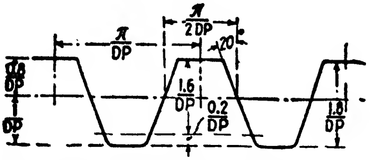


FIG. 14.—Proportions for spur gears.

Proportions of Stub Teeth.—The proportions of 20-deg. stub teeth, as adopted by the American Gear Manufacturers Association, are

STUB-TOOTH STANDARD

	Diametral Pitch	Circular Pitch
Addendum	$= \frac{0.8''}{DP}$	$0.2546'' \times CP$
Dedendum	$= \frac{1''}{DP}$	$0.3183'' \times CP$
Working depth	$= \frac{1.6''}{DP}$	$0.5092'' \times CP$
Total depth	$= \frac{1.8''}{DP}$	$0.5729'' \times CP$
Pitch diameter	$= \frac{N}{DP}$	$0.3183'' \times N \times CP$
Outside diameter	$= \frac{N + 1.6''}{DP}$	$PD + (2 \times \text{addenda})$

NOTE 1.—The above proportions are identical with those of the recommended practice for Herringbone gears.

NOTE 2.—A minimum root clearance of 0.2 in. $\div$ *DP* is recommended for new cutters and gears. There is correct tooth action, however, between gears cut to this new system and those cut to the older Nuttall system, the only dimensions affected being the clearance. Where the proposed gear runs with the Nuttall there is a clearance of 0.1425 in./*DP*, and where the Nuttall gear runs with the proposed gear the clearance is 0.2146 in./*DP*.

NOTE 3.—Diametral pitch used up to 1 *DP* inclusive. Circular pitch used for 3 in. *CP* and over.

Proportion of Metric Gear Teeth.—The methods of using the metric involute gear formulas for determining the dimensions of gears by metric pitch are:

Module is the pitch diameter in millimeters divided by the number of teeth in the gear.

Pitch diameter in millimeters is the module multiplied by the number of teeth in the gear.

M = module.
D' = the pitch diameter of gears in millimeters.
D = the whole diameter of gear in millimeters.
N = the number of teeth in gear.
D'' = the working depth of teeth.

t = thickness of tooth on pitch line.
f = amount added to depth for clearance.

TABLE 3.—MODULE SYSTEM OF TOOTH PARTS

Module, mm.	Circular pitch, mm.	Thickness of tooth at pitch line, mm.	Addendum, mm.	Dedendum, mm.	Clearance, mm.	Working depth of tooth, mm.	Total height of tooth, mm.	Width of thread tool at end, mm.	Width of thread at end, mm.	Nearest English diametral pitch
M	Pc	t	a	d	c	a^2	H	T	T	Pd
0.50	1.57	0.78	0.50	0.58	0.08	1	1.08	0.48	0.52	50.800
0.75	2.36	1.18	0.75	0.87	0.12	1.50	1.62	0.73	0.78	33.866
1	3.14	1.57	1	1.16	0.16	2	2.16	0.97	1.05	25.400
1.25	3.93	1.96	1.25	1.45	0.20	2.50	2.70	1.21	1.31	20.320
1.50	4.71	2.36	1.50	1.73	0.23	3	3.23	1.46	1.57	16.933
1.75	5.50	2.75	1.75	2.02	0.27	3.50	3.77	1.70	1.84	14.514
2	6.28	3.14	2	2.31	0.31	4	4.31	1.94	2.10	12.700
2.25	7.07	3.53	2.25	2.60	0.35	4.50	4.85	2.19	2.37	11.288
2.50	7.85	3.93	2.50	2.89	0.39	5	5.39	2.43	2.63	10.160
2.75	8.64	4.32	2.75	3.18	0.43	5.50	5.93	2.68	2.89	9.236
3	9.42	4.71	3	3.47	0.47	6	6.47	2.92	3.15	8.466
3.50	10.99	5.50	3.50	4.05	0.55	7	7.55	3.40	3.68	7.257
4	12.57	6.28	4	4.63	0.63	8	8.63	3.89	4.20	6.350
4.50	14.14	7.07	4.50	5.21	0.71	9	9.71	4.38	4.73	5.644
5	15.71	7.85	5	5.78	0.78	10	10.78	4.87	5.26	5.080
5.50	17.28	8.64	5.50	6.36	0.86	11	11.86	5.35	5.79	4.618
6	18.85	9.42	6	6.94	0.94	12	12.94	5.84	6.31	4.233
7	21.99	10.99	7	8.10	1.10	14	15.10	6.81	7.36	3.628
8	25.13	12.57	8	9.26	1.26	16	17.26	7.79	8.42	3.175
9	28.27	14.14	9	10.41	1.41	18	19.41	8.76	9.47	2.822
10	31.42	15.71	10	11.57	1.57	20	21.57	9.73	10.52	2.540
11	34.56	17.28	11	12.73	1.73	22	23.73	10.71	11.57	2.309
12	37.70	18.85	12	13.88	1.88	24	25.88	11.68	12.63	2.116
14	43.98	21.99	14	16.20	2.20	28	30.20	13.63	14.73	1.814
16	50.26	25.13	16	18.51	2.51	32	34.51	15.58	16.83	1.587

Then

$$M = \frac{D'}{N} \quad \text{or} \quad \frac{D}{N+2}$$

$$D' = NM$$

$$D = (N+2)M$$

$$N = \frac{D'}{M} \quad \text{or} \quad \frac{D}{M-2}$$

$$D'' = 2M$$

$$t = M1.5708$$

$$f = M1.5708/10 = 0.157M$$

The module is equal to the addendum, measured in millimeters and parts of millimeters.

Module System Translation Formulas.—Module is the pitch diameter in millimeters divided by number of teeth in the gear.

Or, the outside diameter in millimeters divided by the number of teeth in gear plus 2.

To find circular pitch in millimeters from module multiply 3.142 mm. by the number of module, therefore

$$2 \text{ module} \times 3.142 \text{ mm.} = 6.284 \text{ mm.}$$

To find circular pitch in inches from the circular pitch in millimeters, multiply the number of millimeters by 0.03937 in., therefore

$$6.284 \text{ mm.} \times 0.03937 = 0.2474 \text{ in.}$$

To find diametrical pitch from circular pitch in inches, divide 3.1416 by the circular pitch in inches, therefore

$$3.1416 \div 0.2474 \text{ in.} = 12.6999 \text{ diametral pitch.}$$

Tables of Tooth Parts.—The tables of tooth parts which follow will be found useful in many ways. Those which give diametral pitch in the first column will be consulted more frequently because most gears are designated by diametral pitch. The two tables in which circular pitch is given first will be used more by those making worm wheels and worms. These tables have two extra columns which give the width of the thread tool for cutting the worm, at both top and bottom.

The other tables, giving the corrected addenda and thickness, and the chordal thickness of stub teeth, will also be found useful. Two other tables, which are standard practice of the Fellows Gear Shaper Co., may also save calculation.

TABLE 4.—GEAR-TOOTH PARTS

Diametral Pitch in First Column

Diametral pitch	Circular pitch	Thickness of tooth on pitch line	Addendum and module	Working depth of tooth	Depth of space below pitch line	Whole depth of tooth
P	P'	t	s	D''	$s + f$	$D'' + f$
$\frac{1}{2}$	6.2832	3.1416	2.0000	4.0000	2.3142	4.3142
$\frac{3}{4}$	4.1888	2.0944	1.3333	2.6666	1.5428	2.8761
1	3.1416	1.5708	1.0000	2.0000	1.1571	2.1571
$1\frac{1}{4}$	2.5133	1.2566	0.8000	1.6000	0.9257	1.7257
$1\frac{1}{2}$	2.0944	1.0472	0.6666	1.3333	0.7714	1.4381
$1\frac{3}{4}$	1.7952	0.8976	0.5714	1.1429	0.6612	1.2326
2	1.5708	0.7854	0.5000	1.0000	0.5785	1.0785
$2\frac{1}{4}$	1.3963	0.6981	0.4444	0.8888	0.5143	0.9587
$2\frac{1}{2}$	1.2566	0.6283	0.4000	0.8000	0.4628	0.8628
$2\frac{3}{4}$	1.1424	0.5712	0.3636	0.7273	0.4208	0.7844
3	1.0472	0.5236	0.3333	0.6666	0.3857	0.7190
$3\frac{1}{2}$	0.8976	0.4488	0.2857	0.5714	0.3306	0.6163
4	0.7854	0.3927	0.2500	0.5000	0.2893	0.5393
5	0.6283	0.3142	0.2000	0.4000	0.2314	0.4314
6	0.5236	0.2618	0.1666	0.3333	0.1928	0.3595
7	0.4488	0.2244	0.1429	0.2857	0.1653	0.3081
8	0.3927	0.1963	0.1250	0.2500	0.1446	0.2696
9	0.3491	0.1745	0.1111	0.2222	0.1286	0.2397
10	0.3142	0.1571	0.1000	0.2000	0.1157	0.2157
11	0.2856	0.1428	0.0909	0.1818	0.1052	0.1961
12	0.2618	0.1309	0.0833	0.1666	0.0964	0.1798
13	0.2417	0.1208	0.0769	0.1538	0.0890	0.1659
14	0.2244	0.1122	0.0714	0.1429	0.0826	0.1541

Diametral pitch	Circular pitch	Thickness of tooth on pitch line	$1/P$ or the addendum and module	Working depth of tooth	Depth of space below pitch line	Whole depth of tooth
P	P'	t	s	D''	$s + f$	$D'' + f$
15	0.2094	0.1047	0.0666	0.1333	0.0771	0.1438
16	0.1963	0.0982	0.0625	0.1250	0.0723	0.1348
17	0.1848	0.0924	0.0588	0.1176	0.0681	0.1269
18	0.1745	0.0873	0.0555	0.1111	0.0643	0.1198
19	0.1653	0.0827	0.0526	0.1053	0.0609	0.1135
20	0.1571	0.0785	0.0500	0.1000	0.0579	0.1079
22	0.1428	0.0714	0.0455	0.0909	0.0526	0.0980
24	0.1309	0.0654	0.0417	0.0833	0.0482	0.0898
26	0.1208	0.0604	0.0385	0.0769	0.0445	0.0829
28	0.1122	0.0561	0.0357	0.0714	0.0413	0.0770
30	0.1047	0.0524	0.0333	0.0666	0.0386	0.0719
32	0.0982	0.0491	0.0312	0.0625	0.0362	0.0674
34	0.0924	0.0462	0.0294	0.0588	0.0340	0.0634
36	0.0873	0.0436	0.0278	0.0555	0.0321	0.0599
38	0.0827	0.0413	0.0263	0.0526	0.0304	0.0568
40	0.0785	0.0393	0.0250	0.0500	0.0289	0.0539
42	0.0748	0.0374	0.0238	0.0476	0.0275	0.0514
44	0.0714	0.0357	0.0227	0.0455	0.0263	0.0490
46	0.0683	0.0341	0.0217	0.0435	0.0252	0.0469
48	0.0654	0.0327	0.0208	0.0417	0.0241	0.0449
50	0.0628	0.0314	0.0200	0.0400	0.0231	0.0431
56	0.0561	0.0280	0.0178	0.0357	0.0207	0.0385
60	0.0524	0.0262	0.0166	0.0333	0.0193	0.0360

TABLE 5.—GEAR-TOOTH PARTS
Circular Pitch in First Column

Circular pitch	Threads or teeth per inch linear	Diametral pitch	Thickness of tooth on pitch line	Addendum and module	Working depth of tooth	Depth of space below pitch line	Whole depth of tooth	Width of thread tool at end	Width of thread at top
P'	$1''/P'$	P	t	s	D''	$s+f$	$D''+f$	$P' \times 0.3095$	$P' \times 0.3354$
2	1/2	1.5708	1.0000	0.6366	1.2732	0.7366	1.3732	0.6190	0.6707
1 7/8	8/15	1.6755	0.9375	0.5968	1.1937	0.6906	1.2874	0.5803	0.6288
1 3/4	4/7	1.7952	0.8750	0.5570	1.1141	0.6445	1.2016	0.5416	0.5869
1 5/8	8/13	1.9333	0.8125	0.5173	1.0345	0.5985	1.1158	0.5029	0.5450
1 1/2	2/3	2.0944	0.7500	0.4775	0.9549	0.5525	1.0299	0.4642	0.5030
1 7/16	16/23	2.1855	0.7187	0.4576	0.9151	0.5294	0.9870	0.4449	0.4821
1 3/8	8/11	2.2848	0.6875	0.4377	0.8754	0.5064	0.9441	0.4256	0.4611
1 1/3	3/4	2.3562	0.6666	0.4244	0.8488	0.4910	0.9154	0.4127	0.4471
1 5/16	16/21	2.3936	0.6562	0.4178	0.8356	0.4834	0.9012	0.4062	0.4402
1 1/4	4/5	2.5133	0.6250	0.3979	0.7958	0.4604	0.8583	0.3869	0.4192
1 3/16	16/19	2.6456	0.5937	0.3780	0.7560	0.4374	0.8154	0.3675	0.3982
1 1/8	8/9	2.7925	0.5625	0.3581	0.7162	0.4143	0.7724	0.3482	0.3773
1 1/16	16/17	2.9568	0.5312	0.3382	0.6764	0.3913	0.7295	0.3288	0.3563
1	1	3.1416	0.5000	0.3183	0.6366	0.3683	0.6866	0.3095	0.3354
15/16	1 1/15	3.3510	0.4687	0.2984	0.5968	0.3453	0.6437	0.2902	0.3144
7/8	1 1/7	3.5904	0.4375	0.2785	0.5570	0.3223	0.6007	0.2708	0.2934
13/16	1 3/13	3.8666	0.4062	0.2586	0.5173	0.2993	0.5579	0.2515	0.2725
4/5	1 1/4	3.9270	0.4000	0.2546	0.5092	0.2946	0.5492	0.2476	0.2683
3/4	1 1/3	4.1888	0.3750	0.2387	0.4775	0.2762	0.5150	0.2321	0.2515
11/16	1 5/11	4.5696	0.3437	0.2189	0.4377	0.2532	0.4720	0.2128	0.2306
2/3	1 1/2	4.7124	0.3333	0.2122	0.4244	0.2455	0.4577	0.2063	0.2236
5/8	1 3/5	5.0265	0.3125	0.1989	0.3979	0.2301	0.4291	0.1934	0.2096
3/5	1 2/3	5.2360	0.3000	0.1910	0.3820	0.2210	0.4120	0.1857	0.2012
4/7	1 3/4	5.4978	0.2857	0.1819	0.3638	0.2105	0.3923	0.1769	0.1916
9/16	1 7/9	5.5851	0.2812	0.1790	0.3581	0.2071	0.3862	0.1741	0.1886
1/2	2	6.2832	0.2500	0.1592	0.3183	0.1842	0.3433	0.1547	0.1677
4/9	2 1/4	7.0685	0.2222	0.1415	0.2830	0.1637	0.3052	0.1376	0.1490
7/16	2 2/7	7.1808	0.2187	0.1393	0.2785	0.1611	0.3003	0.1354	0.1467
3/7	2 1/3	7.3304	0.2143	0.1364	0.2728	0.1578	0.2942	0.1326	0.1437
2/5	2 1/2	7.8540	0.2000	0.1273	0.2546	0.1473	0.2746	0.1238	0.1341
3/8	2 2/3	8.3776	0.1875	0.1194	0.2387	0.1381	0.2575	0.1161	0.1258
4/11	2 3/4	8.6394	0.1818	0.1158	0.2316	0.1340	0.2498	0.1125	0.1219
1/3	3	9.4248	0.1666	0.1061	0.2122	0.1228	0.2289	0.1032	0.1118
5/16	3 1/5	10.0531	0.1562	0.0995	0.1989	0.1151	0.2146	0.0967	0.1048
3/10	3 1/3	10.4719	0.1500	0.0955	0.1910	0.1105	0.2060	0.0928	0.1006
2/7	3 1/2	10.9956	0.1429	0.0909	0.1819	0.1052	0.1962	0.0884	0.0958
1/4	4	12.5664	0.1250	0.0796	0.1591	0.0921	0.1716	0.0774	0.0838
2/9	4 1/2	14.1372	0.1111	0.0707	0.1415	0.0818	0.1526	0.0688	0.0745
1/5	5	15.7080	0.1000	0.0637	0.1273	0.0737	0.1373	0.0619	0.0671
3/16	5 1/3	16.7552	0.0937	0.0597	0.1194	0.0690	0.1287	0.0580	0.0629
2/11	5 1/2	17.2788	0.0909	0.0579	0.1158	0.0670	0.1249	0.0563	0.0610
1/6	6	18.8496	0.0833	0.0531	0.1061	0.0614	0.1144	0.0516	0.0559
2/13	6 1/2	20.4203	0.0769	0.0489	0.0978	0.0566	0.1055	0.0476	0.0516
1/7	7	21.9911	0.0714	0.0455	0.0910	0.0526	0.0981	0.0442	0.0479
2/15	7 1/2	23.5619	0.0666	0.0425	0.0850	0.0492	0.0917	0.0413	0.0447
1/8	8	25.1327	0.0625	0.0398	0.0796	0.0460	0.0858	0.0387	0.0419
1/9	9	28.2743	0.0555	0.0354	0.0707	0.0409	0.0763	0.0344	0.0373
1/10	10	31.4159	0.0500	0.0318	0.0637	0.0368	0.0687	0.0309	0.0335
1/16	16	50.2655	0.0312	0.0199	0.0398	0.0230	0.0429	0.0193	0.0210
1/20	20	62.8318	0.0250	0.0159	0.0318	0.0184	0.0343	0.0155	0.0168

TABLE 6.—GEAR-TOOTH PARTS*

14½-deg. and 20-deg. Involute Full-length Tooth Form

Diametral pitch	Dimensions, in.				
	Circular thickness	Addendum	Dedendum plus clearance	Whole depth of tooth	Double depth of tooth
6	0.2618	0.1667	0.2083	0.3750	0.7500
7	0.2244	0.1429	0.1786	0.3215	0.6430
8	0.1964	0.1250	0.1563	0.2813	0.5626
9	0.1745	0.1111	0.1389	0.2500	0.5000
10	0.1571	0.1000	0.1250	0.2250	0.4500
11	0.1428	0.0909	0.1136	0.2045	0.4090
12	0.1309	0.0833	0.1042	0.1875	0.3750
14	0.1122	0.0714	0.0893	0.1607	0.3214
16	0.0982	0.0625	0.0781	0.1406	0.2812
18	0.0873	0.0556	0.0705	0.1261	0.2522
20	0.0785	0.0500	0.0650	0.1150	0.2300
22	0.0714	0.0455	0.0604	0.1059	0.2118
24	0.0654	0.0417	0.0566	0.0983	0.1966
26	0.0604	0.0386	0.0533	0.0919	0.1838
28	0.0561	0.0357	0.0507	0.0864	0.1728
30	0.0524	0.0333	0.0483	0.0816	0.1632
32	0.0491	0.0313	0.0462	0.0775	0.1550
34	0.0462	0.0294	0.0394	0.0688	0.1376
36	0.0436	0.0278	0.0377	0.0655	0.1310
38	0.0413	0.0263	0.0363	0.0626	0.1252
40	0.0393	0.0250	0.0350	0.0600	0.1200

Fellows Gear Shaper Company Standard.

* Note that this table gives circular tooth thickness instead of circular pitch.

TABLE 7.—GEAR-TOOTH PARTS*
20-deg. Pressure Angle, Stub-tooth Form

Diametral pitch	Dimensions, in.				
	Circular thickness	Addendum	Dedendum plus clearance	Whole depth of tooth	Double depth of tooth
6/8	0.2618	0.1250	0.1563	0.2813	0.5626
7/9	0.2244	0.1111	0.1389	0.2500	0.5000
8/10	0.1964	0.1000	0.1250	0.2250	0.4500
9/11	0.1745	0.0909	0.1136	0.2045	0.4090
10/12	0.1571	0.0833	0.1042	0.1875	0.3750
11/14	0.1428	0.0714	0.0893	0.1607	0.3214
12/14	0.1309	0.0714	0.0893	0.1607	0.3214
14/18	0.1122	0.0556	0.0705	0.1261	0.2522
16/21	0.0982	0.0476	0.0626	0.1102	0.2204
18/24	0.0873	0.0417	0.0566	0.0983	0.1966
20/26	0.0785	0.0386	0.0533	0.0919	0.1838
22/29	0.0714	0.0345	0.0494	0.0839	0.1678
24/32	0.0654	0.0313	0.0462	0.0775	0.1550
26/35	0.0604	0.0286	0.0435	0.0671	0.1342
28/37	0.0561	0.0270	0.0420	0.0641	0.1282
30/40	0.0524	0.0250	0.0350	0.0600	0.1200
32/42	0.0491	0.0238	0.0338	0.0576	0.1152
34/45	0.0462	0.0222	0.0322	0.0544	0.1088
36/48	0.0436	0.0208	0.0309	0.0517	0.1034
38/50	0.0413	0.0200	0.0300	0.0500	0.1000
40/54	0.0393	0.0185	0.0285	0.0470	0.0940

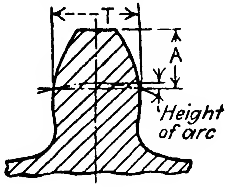
Fellows Gear Shaper Company Standard.
* Note that this table gives circular tooth thickness instead of circular pitch.

Designation of diametral pitch on stub teeth shows the relation of pitch to depth as compared with standard length teeth. A 6/8 pitch, for example, indicates 6 diametral pitch but that the tooth only has the depth of an 8-pitch tooth of standard length.

TABLE 8.—CORRECTED ADDENDUM AND THICKNESS OF GEAR TEETH

Adjust gear tooth caliper according to this table to obtain corrected addendum and thickness for the exact number of teeth being cut in gear.

This table is figured for 1 diametral pitch. For any other diametral pitch divide the number in the table by the required pitch.

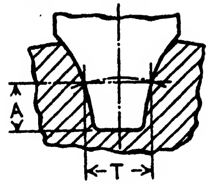


Teeth	Corrected addendum A	Corrected thickness of tooth T	Teeth	Corrected addendum A	Corrected thickness of tooth T
6	1.10222	1.55291	46	1.01336	1.57050
7	1.08777	1.55764	47	1.01311	1.57051
8	1.07686	1.56072	48	1.01285	1.57052
9	1.06836	1.56283	49	1.01258	1.57053
10	1.06155	1.56435	50	1.01233	1.57054
11	1.05598	1.56533	51	1.01209	1.57055
12	1.05133	1.56631	52	1.01187	1.57056
13	1.04739	1.56698	53	1.01165	1.57057
14	1.04401	1.56752	54	1.01143	1.57058
15	1.04109	1.56794	55	1.01121	1.57058
16	1.03852	1.56827	56	1.01102	1.57059
17	1.03625	1.56856	57	1.01083	1.57060
18	1.03425	1.56880	58	1.01064	1.57061
19	1.03244	1.56899	59	1.01046	1.57061
20	1.03083	1.56918	60	1.01029	1.57062
21	1.02936	1.56933	61	1.01011	1.57062
22	1.02803	1.56948	62	1.00994	1.57063
23	1.02681	1.56956	63	1.00978	1.57063
24	1.02569	1.56967	64	1.00963	1.57064
25	1.02466	1.56977	65	1.00947	1.57064
26	1.02371	1.56986	66	1.00933	1.57065
27	1.02234	1.56994	67	1.00920	1.57065
28	1.02194	1.56998	68	1.00907	1.57066
29	1.02121	1.57003	69	1.00893	1.57066
30	1.02055	1.57008	70	1.00880	1.57067
31	1.01990	1.57012	71	1.00867	1.57067
32	1.01926	1.57016	72	1.00855	1.57067
33	1.01869	1.57019	73	1.00843	1.57068
34	1.01813	1.57021	74	1.00832	1.57068
35	1.01762	1.57025	75	1.00821	1.57068
36	1.01714	1.57028	76	1.00810	1.57069
37	1.01667	1.57032	77	1.00799	1.57069
38	1.01623	1.57035	78	1.00789	1.57069
39	1.01582	1.57037	79	1.00780	1.57069
40	1.01542	1.57039	80	1.00772	1.57070
41	1.01504	1.57041	81	1.00762	1.57070
42	1.01471	1.57043	82	1.00752	1.57070
43	1.01437	1.57045	83	1.00743	1.57070
44	1.01404	1.57047	84	1.00734	1.57071
45	1.01370	1.57048	85	1.00725	1.57071

TABLE 8.—CORRECTED ADDENDUM AND THICKNESS OF GEAR TEETH.—(*Continued*)

Teeth	Corrected addendum A	Corrected thickness of tooth T	Teeth	Corrected addendum A	Corrected thickness of tooth T
86	1.00716	1.57071	136	1.00454	1.57076
87	1.00708	1.57071	137	1.00451	1.57076
88	1.00700	1.57071	138	1.00447	1.57076
89	1.00693	1.57072	139	1.00444	1.57076
90	1.00686	1.57072	140	1.00441	1.57076
91	1.00679	1.57072	141	1.00439	1.57076
92	1.00672	1.57072	142	1.00435	1.57076
93	1.00665	1.57072	143	1.00432	1.57076
94	1.00658	1.57072	144	1.00429	1.57076
95	1.00651	1.57073	145	1.00425	1.57077
96	1.00644	1.57073	146	1.00422	1.57077
97	1.00637	1.57073	147	1.00419	1.57077
98	1.00630	1.57073	148	1.00416	1.57077
99	1.00623	1.57073	149	1.00413	1.57077
100	1.00617	1.57073	150	1.00411	1.57077
101	1.00611	1.57074	151	1.00409	1.57077
102	1.00605	1.57074	152	1.00407	1.57077
103	1.00599	1.57074	153	1.00405	1.57077
104	1.00593	1.57074	154	1.00402	1.57077
105	1.00587	1.57074	155	1.00400	1.57077
106	1.00581	1.57074	156	1.00397	1.57077
107	1.00575	1.57074	157	1.00394	1.57077
108	1.00570	1.57074	158	1.00391	1.57077
109	1.00565	1.57075	159	1.00389	1.57077
110	1.00560	1.57075	160	1.00386	1.57077
111	1.00556	1.57075	161	1.00383	1.57077
112	1.00551	1.57075	162	1.00380	1.57077
113	1.00546	1.57075	163	1.00378	1.57077
114	1.00541	1.57075	164	1.00376	1.57077
115	1.00537	1.57075	165	1.00373	1.57077
116	1.00533	1.57075	166	1.00370	1.57077
117	1.00529	1.57075	167	1.00368	1.57077
118	1.00524	1.57075	168	1.00366	1.57077
119	1.00519	1.57075	169	1.00364	1.57077
120	1.00515	1.57075	170	1.00362	1.57077
121	1.00511	1.57075	171	1.00359	1.57077
122	1.00507	1.57075	172	1.00357	1.57077
123	1.00503	1.57076	173	1.00355	1.57077
124	1.00499	1.57076	174	1.00353	1.57077
125	1.00495	1.57076	175	1.00351	1.57077
126	1.00491	1.57076	176	1.00349	1.57077
127	1.00487	1.57076	177	1.00347	1.57077
128	1.00483	1.57076	178	1.00345	1.57077
129	1.00479	1.57076	179	1.00343	1.57078
130	1.00475	1.57076	180	1.00342	1.57078
131	1.00472	1.57076	181	1.00340	1.57078
132	1.00469	1.57076	182	1.00339	1.57078
133	1.00466	1.57076			
134	1.00462	1.57076			
135	1.00457	1.57076			

Table 9.—Chordal Thickness and Addenda of Gear Cutters



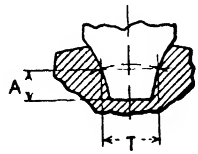
T = Chordal thickness of cutter at pitch line.
A = Perpendicular distance from chord to outside circumference of cutter (corrected addendum).

Diametral pitch	Dimension	Number of gear cutter							
		No. 1 135 teeth	No. 2 55 teeth	No. 3 35 teeth	No. 4 26 teeth	No. 5 21 teeth	No. 6 17 teeth	No. 7 14 teeth	No. 8 12 teeth
1	*T*	1.5707	1.5706	1.5702	1.5698	1.5694	1.5686	1.5675	1.5663
	A	1.1525	1.1459	1.1395	1.1334	1.1277	1.1209	1.1131	1.1057
1½	*T*	1.0471	1.0470	1.0468	1.0465	1.0462	1.0457	1.0450	1.0442
	A	0.7683	0.7639	0.7596	0.7556	0.7518	0.7472	0.7420	0.7371
2	*T*	0.7853	0.7853	0.7851	0.7849	0.7847	0.7843	0.7837	0.7831
	A	0.5762	0.5729	0.5697	0.5667	0.5638	0.5604	0.5565	0.5528
2½	*T*	0.6283	0.6282	0.6281	0.6279	0.6277	0.6274	0.6270	0.6265
	A	0.4610	0.4583	0.4558	0.4533	0.4511	0.4483	0.4452	0.4423
3	*T*	0.5235	0.5235	0.5234	0.5232	0.5231	0.5228	0.5225	0.5221
	A	0.3841	0.3819	0.3798	0.3778	0.3759	0.3736	0.3710	0.3685
3½	*T*	0.4487	0.4487	0.4486	0.4485	0.4484	0.4481	0.4478	0.4475
	A	0.3292	0.3274	0.3255	0.3238	0.3222	0.3202	0.3180	0.3159
4	*T*	0.3926	0.3926	0.3925	0.3924	0.3923	0.3921	0.3919	0.3915
	A	0.2881	0.2864	0.2848	0.2833	0.2819	0.2802	0.2783	0.2764
5	*T*	0.3141	0.3141	0.3140	0.3139	0.3138	0.3137	0.3135	0.3132
	A	0.2305	0.2291	0.2279	0.2266	0.2255	0.2242	0.2226	0.2211
6	*T*	0.2617	0.2617	0.2617	0.2616	0.2615	0.2614	0.2612	0.2610
	A	0.1921	0.1910	0.1899	0.1889	0.1879	0.1868	0.1855	0.1842
7	*T*	0.2243	0.2243	0.2243	0 2242	0.2242	0.2240	0.2239	0.2237
	A	0.1646	0.1637	0.1627	0.1619	0.1611	0.1601	0.1590	0.1579
8	*T*	0.1963	0.1963	0.1962	0.1962	0.1961	0.1960	0.1959	0.1957
	A	0.1440	0.1432	0.1424	0.1416	0.1409	0.1401	0.1391	0.1382
9	*T*	0.1745	0.1745	0.1744	0.1744	0.1743	0.1742	0.1741	0.1740
	A	0.1280	0.1273	0.1266	0.1259	0.1253	0.1245	0.1236	0.1228
10	*T*	0.1570	0.1570	0.1570	0.1569	0.1569	0.1568	0.1567	0.1566
	A	0.1152	0.1144	0.1139	0.1133	0.1127	0.1120	0.1113	0.1105
11	*T*	0.1427	0.1427	0.1427	0.1427	0.1426	0.1426	0.1425	0.1423
	A	0.1047	0.1041	0.1035	0.1030	0.1025	0.1019	0.1012	0.1005
12	*T*	0.1308	0.1308	0.1308	0.1308	0.1308	0.1307	0.1306	0.1305
	A	0.0960	0.0954	0.0949	0.0944	0.0939	0.0934	0.0927	0.0921
14	*T*	0.1122	0.1122	0.1121	0.1121	0.1121	0.1120	0.1119	0.1118
	A	0.0823	0.0818	0.0814	0.0809	0.0805	0.0800	0.0795	0.0789
16	*T*	0.0981	0.0981	0.0981	0.0981	0.0981	0.0980	0.0979	0.0979
	A	0.0720	0.0716	0.0712	0.0708	0.0705	0.0700	0.0695	0.0691
18	*T*	0.0872	0.0872	0.0872	0.0872	0.0872	0.0871	0.0871	0.0870
	A	0.0640	0.0636	0.0633	0.0629	0.0626	0.0622	0.0618	0.0614
20	*T*	0.0785	0.0785	0.0785	0.0785	0.0784	0.0784	0.0783	0.0783
	A	0.0576	0.0573	0.0569	0.0566	0.0564	0.0560	0.0556	0.0553
22	*T*	0.0714	0.0714	0.0714	0.0713	0.0713	0.0713	0.0712	0.0712
	A	0.0521	0.0521	0.0518	0.0515	0.0512	0.0509	0.0506	0.0502
24	*T*	0.0654	0.0654	0.0654	0.0654	0.0654	0.0653	0.0653	0.0652
	A	0.0480	0.0477	0.0472	0.0470	0.0470	0.0467	0.0463	0.0460

TABLE 10.—CORRECTED T AND A OF STUB-TOOTH GEAR CUTTERS

T = Chordal thickness of cutter at pitch line.

A = Perpendicular distance from chord to outside circumference of cutter, corrected addendum.



Diametral pitch	Dimension	No. 1 135 *T*	No. 2 55 *T*	No. 3 35 *T*	No. 4 26 *T*	No. 5 21 *T*	No. 6 17 *T*	No. 7 14 *T*	No. 8 21 *T*
4/5	*T*	0.3926	0.3926	0.3925	0.3924	0.3923	0.3921	0.3919	0.3915
	A	0.2302	0.2285	0.2269	0.2254	0.2240	0.2223	0.2204	0.2185
5/7	*T*	0.3141	0.3141	0.3140	0.3139	0.3138	0.3137	0.3135	0 3132
	A	0.1644	0.1630	0 1618	0.1605	0.1594	0.1581	0.1565	0.1550
6/8	*T*	0.2617	0.2617	0.2617	0.2616	0.2615	0.2614	0.2612	0.2610
	A	0.1439	0.1428	0.1417	0.1407	0.1397	0.1386	0.1373	0.1360
7/9	*T*	0.2243	0.2243	0.2243	0.2242	0 2242	0.2240	0.2239	0.2237
	A	0.1279	0.1270	0.1261	0.1252	0.1244	0.1234	0.1223	0.1212
8/10	*T*	0.1963	0.1963	0.1962	0.1962	0.1961	0.1960	0.1959	0.1957
	A	0.1151	0.1143	0.1135	0.1127	0.1120	0.1112	0.1102	0.1093
9/11	*T*	0.1745	0.1745	0.1744	0.1744	0.1743	0.1742	0.1741	0.1740
	A	0.1046	0.1039	0.1032	0.1025	0.1019	0.1011	0.1002	0.0994
10/12	*T*	0.1570	0.1570	0.1570	0.1569	0.1569	0.1568	0.1567	0.1566
	A	0.0959	0.0951	0.0946	0.0940	0.0934	0.0927	0.0920	0.0912
12/14	*T*	0.1308	0.1308	0.1308	0.1308	0.1308	0.1307	0.1306	0.1305
	A	0.0822	0.0816	0.0811	0.0806	0.0801	0.0796	0.0789	0.0783

Standard Backlash Practice.—In order to secure uniformity of product the American Gear Manufacturers Association has adopted a standard allowance for backlash in spur gears. This backlash is measured with the gears at the standard center distance, using a feeler between the teeth as indicated in the diagram accompanying the table. To secure this backlash the teeth should be cut thinner than the standard thickness by an amount equal to one-half the backlash desired. The table gives the thickness of the feeler to be used on diametral pitches from 1 to 24.

TABLE 11.—STANDARD BACKLASH

The values given in the table are based on dividing the following constants by the diametral pitch:

Minimum, 0.03″ Average, 0.04″ Maximum, 0.05″

Thickness of feeler, inch			DP
Minimum	Average	Maximum	
0.002	0.003	0.004	24
0.002	0.003	0.004	16
0.003	0.004	0.005	12
0.003	0.004	0.005	10
0.004	0.005	0.006	8
0.005	0.007	0.008	6
0.006	0.008	0.010	5
0.008	0.010	0.013	4
0.010	0.013	0.017	3
0.012	0.016	0.020	2½
0.015	0.020	0.025	2
0.020	0.027	0.033	1½
0.030	0.040	0.050	1

Measuring Backlash.—There are two methods of measuring backlash, one by the use of a feeler placed between the teeth, when the gears are located at the proper center distance as already stated, and the other by placing the gears on studs on a gear-testing fixture and measuring the difference between the standard center distance and the distance obtained when the gears are brought into close mesh with each other.

Of course, this last test does not represent backlash directly, and it is necessary to compute it by means of the following formula:

$$b = 2 \tan a \times d_1$$

in which

b = backlash in inches.

d_1 = difference between standard and measured center distance.

a = pressure angle.

Backlash can be obtained either by feeding the cutter in beyond standard depth, or by a special cutter having thicker teeth than standard. The second method is recommended when a large number of gears are to be cut.

The amount that the cutter should be fed in beyond standard depth is equal to one-half the difference between the standard and measured center distance.

For 0.008 in. backlash, we would feed the cutter in approximately 0.0055 in. beyond standard depth for both gear and pinion, and when the gears are placed at the standard center distance, a feeler 0.008 in. thick could be placed between the teeth.

The accompanying table gives the amount of backlash obtained by feeding the cutter in to different depths, and also the difference between the measured and actual center distance indicating a certain amount of backlash. This table applies both to 14½-deg. full-length teeth and 20-deg. stub-tooth gears.

Table 12.—Backlash, Together with Amount That Cutter Should Be Fed In, and Different Center Distance Obtained

Backlash, in.	14½-deg. full-length tooth		20-deg. stub tooth	
	Amount to feed cutter in beyond standard depth per gear	Difference between measured and standard center distance	Amount to feed cutter in beyond standard depth per gear	Difference between measured and standard center distance
0.0010	0.0009	0.0018	0.0008	0.0017
0.0015	0.0014	0.0028	0.0010	0.0020
0.0020	0.0019	0.0038	0.0014	0.0028
0.0025	0.0024	0.0048	0.0017	0.0034
0.0030	0.0029	0.0058	0.0020	0.0040
0.0035	0.0034	0.0068	0.0024	0.0048
0.0040	0.0038	0.0076	0.0027	0.0054
0.0045	0.0043	0.0086	0.0030	0.0060
0.0050	0.0048	0.0096	0.0034	0.0068
0.0060	0.0058	0.0116	0.0041	0.0082
0.0070	0.0067	0.0134	0.0048	0.0096
0.0080	0.0077	0.0154	0.0055	0.0110
0.0090	0.0087	0.0174	0.0062	0.0124
0.0100	0.0096	0.0192	0.0068	0.0136
0.0110	0.0106	0.0212	0.0075	0.0150
0.0120	0.0116	0.0232	0.0082	0.0164
0.0130	0.0125	0.0250	0.0089	0.0178
0.0140	0.0135	0.0270	0.0096	0.0192
0.0150	0.0145	0.0290	0.0100	0.0200
0.0160	0.0154	0.0308	0.0110	0.0220

TABLE 13.—TABLE FOR TURNING AND CUTTING GEAR BLANKS FOR STANDARD LENGTH TOOTH

Pitch	16	12	10	8	Pitch	16	12	10	8
Depth of tooth	0.135	0.180	0.216	0.270	Depth of tooth	0.135	0.180	0.216	0.270
No. of teeth	Outside diameter				No. of teeth	Outside diameter			
10	3/4	1	1 2/10	1 1/2	60	3 7/8	5 2/12	6 2/10	7 3/4
11	13/16	1 1/12	1 3/10	1 5/8	61	3 15/16	5 3/12	6 3/10	7 7/8
12	7/8	1 2/12	1 4/10	1 3/4	62	4	5 4/12	6 4/10	8
13	15/16	1 3/12	1 5/10	1 7/8	63	4 1/16	5 5/12	6 5/10	8 1/8
14	1	1 4/12	1 6/10	2	64	4 1/8	5 6/12	6 6/10	8 1/4
15	1 1/16	1 5/12	1 7/10	2 1/8	65	4 3/16	5 7/12	6 7/10	8 3/8
16	1 1/8	1 6/12	1 8/10	2 1/4	66	4 1/4	5 8/12	6 8/10	8 1/2
17	1 3/16	1 7/12	1 9/10	2 3/8	67	4 5/16	5 9/12	6 9/10	8 5/8
18	1 1/4	1 8/12	2	2 1/2	68	4 3/8	5 10/12	7	8 3/4
19	1 5/16	1 9/12	2 1/10	2 5/8	69	4 7/16	5 11/12	7 1/10	8 7/8
20	1 3/8	1 10/12	2 2/10	2 3/4	70	4 1/2	6	7 2/10	9
21	1 7/16	1 11/12	2 3/10	2 7/8	71	4 9/16	6 1/12	7 3/10	9 1/8
22	1 1/2	2	2 4/10	3	72	4 5/8	6 2/12	7 4/10	9 1/4
23	1 9/16	2 1/12	2 5/10	3 1/8	73	4 11/16	6 3/12	7 5/10	9 3/8
24	1 5/8	2 2/12	2 6/10	3 1/4	74	4 3/4	6 4/12	7 6/10	9 1/2
25	1 11/16	2 3/12	2 7/10	3 3/8	75	4 13/16	6 5/12	7 7/10	9 5/8
26	1 3/4	2 4/12	2 8/10	3 1/2	76	4 7/8	6 6/12	7 8/10	9 3/4
27	1 13/16	2 5/12	2 9/10	3 5/8	77	4 15/16	6 7/12	7 9/10	9 7/8
28	1 7/8	2 6/12	3	3 3/4	78	5	6 8/12	8	10
29	1 15/16	2 7/12	3 1/10	3 7/8	79	5 1/16	6 9/12	8 1/10	10 1/8
30	2	2 8/12	3 2/10	4	80	5 1/8	6 10/12	8 2/10	10 1/4
31	2 1/16	2 9/12	3 3/10	4 1/8	81	5 3/16	6 11/12	8 3/10	10 3/8
32	2 1/8	2 10/12	3 4/10	4 1/4	82	5 1/4	7	8 4/10	10 1/2
33	2 3/16	2 11/12	3 5/10	4 3/8	83	5 5/16	7 1/12	8 5/10	10 5/8
34	2 1/4	3	3 6/10	4 1/2	84	5 3/8	7 2/12	8 6/10	10 3/4
35	2 5/16	3 1/12	3 7/10	4 5/8	85	5 7/16	7 3/12	8 7/10	10 7/8
36	2 3/8	3 2/12	3 8/10	4 3/4	86	5 1/2	7 4/12	8 8/10	11
37	2 7/16	3 3/12	3 9/10	4 7/8	87	5 9/16	7 5/12	8 9/10	11 1/8
38	2 1/2	3 4/12	4	5	88	5 5/8	7 6/12	9	11 1/4
39	2 9/16	3 5/12	4 1/10	5 1/8	89	5 11/16	7 7/12	9 1/10	11 3/8
40	2 5/8	3 6/12	4 2/10	5 1/4	90	5 3/4	7 8/12	9 2/10	11 1/2
41	2 11/16	3 7/12	4 3/10	5 3/8	91	5 13/16	7 9/12	9 3/10	11 5/8
42	2 3/4	3 8/12	4 4/10	5 1/2	92	5 7/8	7 10/12	9 4/10	11 3/4
43	2 13/16	3 9/12	4 5/10	5 5/8	93	5 15/16	7 11/12	9 5/10	11 7/8
44	2 7/8	3 10/12	4 6/10	5 3/4	94	6	8	9 6/10	12
45	2 15/16	3 11/12	4 7/10	5 7/8	95	6 1/16	8 1/12	9 7/10	12 1/8
46	3	4	4 8/10	6	96	6 1/8	8 2/12	9 8/10	12 1/4
47	3 1/16	4 1/12	4 9/10	6 1/8	97	6 3/16	8 3/12	9 9/10	12 3/8
48	3 1/8	4 2/12	5	6 1/4	98	6 1/4	8 4/12	10	12 1/2
49	3 3/16	4 3/12	5 1/10	6 3/8	99	6 5/16	8 5/12	10 1/10	12 5/8
50	3 1/4	4 4/12	5 2/10	6 1/2	100	6 3/8	8 6/12	10 2/10	12 3/4
51	3 5/16	4 5/12	5 3/10	6 5/8	101	6 7/16	8 7/12	10 3/10	12 7/8
52	3 3/8	4 6/12	5 4/10	6 3/4	102	6 1/2	8 8/12	10 4/10	13
53	3 7/16	4 7/12	5 5/10	6 7/8	103	6 9/16	8 9/12	10 5/10	13 1/8
54	3 1/2	4 8/12	5 6/10	7	104	6 5/8	8 10/12	10 6/10	13 1/4
55	3 9/16	4 9/12	5 7/10	7 1/8	105	6 11/16	8 11/12	10 7/10	13 3/8
56	3 5/8	4 10/12	5 8/10	7 1/4	106	6 3/4	9	10 8/10	13 1/2
57	3 11/16	4 11/12	5 9/10	7 3/8	107	6 13/16	9 1/12	10 9/10	13 5/8
58	3 3/4	5	6	7 1/2	108	6 7/8	9 2/12	11	13 3/4
59	3 13/16	5 1/12	6 1/10	7 5/8	109	6 15/16	9 3/12	11 1/10	13 7/8

TABLE 13.—TABLE FOR TURNING AND CUTTING GEAR BLANKS FOR STANDARD LENGTH TOOTH.—(*Continued*)

Pitch	16	12	10	8	Pitch	16	12	10	8
Depth of tooth	0.135	0.180	0.216	0.270	Depth of tooth	0.135	0.180	0.216	0.270
No. of teeth	Outside diameter				No. of teeth	Outside diameter			
110	7	9 4/12	11 2/10	14	142	9	12	14 4/10	18
111	7 1/16	9 5/12	11 3/10	14 1/8	143	9 1/16	12 1/12	14 5/10	18 1/8
112	7 1/8	9 6/12	11 4/10	14 1/4	144	9 1/8	12 2/12	14 6/10	18 1/4
113	7 3/16	9 7/12	11 5/10	14 3/8	145	9 3/16	12 3/12	14 7/10	18 3/8
114	7 1/4	8 8/12	11 6/10	14 1/2	146	9 1/4	12 4/12	14 8/10	18 1/2
115	7 5/16	9 9/12	11 7/10	14 5/8	147	9 5/16	12 5/12	14 9/10	18 5/8
116	7 3/8	9 10/12	11 8/10	14 3/4	148	9 3/8	12 6/12	15	18 3/4
117	7 7/16	9 11/12	11 9/10	14 7/8	149	9 7/16	12 7/12	15 1/10	18 7/8
118	7 1/2	10	12	15	150	9 1/2	12 8/12	15 2/10	19
119	7 9/16	10 1/12	12 1/10	15 1/8	151	9 9/16	12 9/12	15 3/10	19 1/8
120	7 5/8	10 2/12	12 2/10	15 1/4	152	9 5/8	12 10/12	15 4/10	19 1/4
121	7 11/16	10 3/12	12 3/10	15 3/8	153	9 11/16	12 11/12	15 5/10	19 3/8
122	7 3/4	10 4/12	12 4/10	15 1/2	154	9 3/4	13	15 6/10	19 1/2
123	7 13/16	10 5/12	12 5/10	15 5/8	155	9 13/16	13 1/12	15 7/10	19 5/8
124	7 7/8	10 6/12	12 6/10	15 3/4	156	9 7/8	13 2/12	15 8/10	19 3/4
125	7 15/16	10 7/12	10 7/10	15 7/8	157	9 15/16	13 3/12	15 9/10	19 7/8
126	8	10 8/12	12 8/10	16	158	10	13 4/12	16	20
127	8 1/16	10 9/12	12 9/10	16 1/8	159	10 1/16	13 5/12	16 1/10	20 1/8
128	8 1/8	10 10/12	13	16 1/4	160	10 1/8	13 6/12	16 2/10	20 1/4
129	8 3/16	10 11/12	13 1/10	16 3/8	161	10 3/16	13 7/12	16 3/10	20 3/8
130	8 1/4	11	13 2/10	16 1/2	162	10 1/4	13 8/12	16 4/10	20 1/2
131	8 5/16	11 1/12	13 3/10	16 5/8	163	10 5/16	13 9/12	16 5/10	20 5/8
132	8 3/8	11 2/12	13 4/10	16 3/4	164	10 3/4	13 10/12	16 6/10	20 3/4
133	8 7/16	11 3/12	13 5/10	16 7/8	165	10 7/16	13 11/12	16 7/10	20 7/8
134	8 1/2	11 4/12	13 6/10	17	166	10 1/2	14	16 8/10	21
135	8 9/16	11 5/12	13 7/10	17 1/8	167	10 9/16	14 1/12	16 9/10	21 1/8
136	8 5/8	11 6/12	13 8/10	17 1/4	168	10 5/8	14 2/12	17	21 1/4
137	8 11/16	11 7/12	13 9/10	17 3/8	169	10 11/16	14 3/12	17 1/10	21 3/8
138	8 3/4	11 8/12	14	17 1/2	170	10 3/4	14 4/12	17 2/10	21 1/2
139	8 13/16	11 9/12	14 1/10	17 5/8	171	10 13/16	14 5/12	17 3/10	21 5/8
140	8 7/8	11 10/12	14 2/10	17 3/4	172	10 7/8	14 6/12	17 4/10	21 3/4
141	8 15/16	11 11/12	14 3/10	17 7/8	173	10 15/16	14 7/12	17 5/10	21 7/8

Gear Blanks.—Gear blanks, of whatever material, should be turned of the correct diameter to secure teeth of the proper length when the gears are finished. The relation of the outside diameter to the pitch diameter has been explained on page 7. Tables showing the outside diameter, for gears having different pitches and numbers of teeth, are given herewith. Table 13 gives the outside diameter for standard tooth gears and Table 14 for gears having stub teeth. These apply only to spur gears, not to helical gears.

How the Design of Gears Affects Production.—Although we are considering mainly the production of gears and not the shape of the tooth or other features of design, some of these factors affect production to such an extent that they should not be overlooked. Accurate preparatory operations, such as the size of

TABLE 14.—STUB-SPUR-GEAR DIAMETERS WITH WORKING CLEARANCE OR RUNNING ALLOWANCE MADE

Number of teeth	6/8 Pitch			8/10 Pitch			10/12 Pitch		
	Root diameter	Pitch diameter	Outside diameter	Root diameter	Pitch diameter	Outside diameter	Root diameter	Pitch diameter	Outside diameter
12	1.678	2 000	2.241	1 243	1.500	1 693	0.986	1 200	1.361
13	1.845	2.167	2 408	1.368	1 625	1 818	1.086	1.300	1.461
14	2.011	2.333	2 574	1 493	1 750	1 943	1.186	1 400	1.561
15	2 178	2.500	2.741	1 618	1.875	2.068	1 286	1.500	1.661
16	2.345	2.667	2.908	1 743	2.000	2 193	1.386	1.600	1.761
17	2 511	2 833	3.074	1 868	2 125	2 318	1.486	1.700	1.861
18	2.678	3.000	3 241	1.993	2 250	2 443	1.586	1 800	1 961
19	2.845	3 167	3 408	2 118	2 375	2 568	1.686	1.900	2.061
20	3.011	3.333	3 574	2 243	2 500	2 693	1.786	2.000	2.161
21	3 178	3 500	3 741	2 368	2 625	2.818	1.886	2 100	2 261
22	3 345	3 667	3.908	2 493	2 750	2 943	1.986	2 200	2.361
23	3.511	3 833	4.074	2 618	2 875	3 068	2 086	2.300	2.461
24	3 678	4.000	4.241	2 743	3.000	3 193	2 186	2 400	2.561
25	3.845	4 167	4 408	2 868	3 125	3 318	2.286	2.500	2 661
26	4.011	4 333	4 574	2 993	3 250	3.443	2 386	2.600	2.761
27	4.178	4.500	4.741	3 118	3 375	3 568	2 486	2.700	2.861
28	4.345	4 667	4 908	3 243	3 500	3.693	2 586	2.800	2.961
29	4 511	4 833	5.074	3.368	3 625	3.818	2.686	2 900	3 061
30	4.678	5.000	5 241	3.493	3 750	3 943	2.786	3 000	3.161
34	5.345	5 667	5.908	3 993	4 250	4 443	3.186	3 400	3.561
38	6.011	6.333	6 574	4 493	4 750	4.943	3 586	3 800	3 961
42	6 678	7.000	7 241	4 993	5 250	5 443	3.986	4.200	4.361
46	7 345	7 667	7 908	5 493	5 750	5.943	4.386	4.600	4.761
50	8 011	8.333	8.574	5.993	6 250	6.443	4.786	5.000	5.161
54	8.678	9 000	9.241	6.493	6.750	6.943	5 186	5.400	5 561
58	9.345	9.667	9.908	6.993	7 250	7.443	5 586	5 800	5.961
62	10.011	10 333	10.574	7.493	7 750	7.943	5.986	6.200	6.361
66	10 678	11.000	11.241	7.993	8.250	8 443	6.386	6.600	6.761
70	11.345	11.667	11.908	8.493	8.750	8.943	6 786	7.000	7.161
74	12.011	12.333	12.574	8.993	9.250	9.443	7.186	7.400	7.561
78	12.678	13.000	13.241	9.493	9.750	9.943	7.586	7.800	7.961
82	13.345	13.667	13.908	9.993	10.250	10.443	7.986	8.200	8.361
86	14.011	14.333	14 574	10.493	10.750	10.943	8.386	8 600	8 761
90	14.678	15.000	15.241	10.993	11.250	11.443	8 786	9.000	9.161

TABLE 14.—STUB-SPUR-GEAR DIAMETERS WITH WORKING CLEARANCE OR RUNNING ALLOWANCE MADE.—(*Continued*)

Number of teeth	12/14 Pitch			14/16 Pitch			16/18 Pitch		
	Root diameter	Pitch diameter	Outside diameter	Root diameter	Pitch diameter	Outside diameter	Root diameter	Pitch diameter	Outside diameter
12	0.816	1.006	1.138	0.697	0.857	0.978	0.607	0.750	0.857
13	0.899	1.083	1.221	0.769	0.929	1.050	0.670	0.813	0.920
14	0.983	1.167	1.305	0.840	1.000	1.121	0.732	0.875	0.982
15	1.066	1.250	1.388	0.911	1.071	1.192	0.795	0.938	1.045
16	1.149	1.333	1.471	0.983	1.143	1.264	0.857	1.000	1.107
17	1.233	1.417	1.555	1.054	1.214	1.335	0.920	1.063	1.170
18	1.316	1.500	1.638	1.126	1.286	1.407	0.982	1.125	1.232
19	1.399	1.583	1.721	1.197	1.357	1.478	1.045	1.188	1.295
20	1.483	1.667	1.805	1.269	1.429	1.550	1.107	1.250	1.357
21	1.566	1.750	1.888	1.340	1.500	1.621	1.170	1.313	1.420
22	1.649	1.833	1.971	1.411	1.571	1.692	1.232	1.375	1.482
23	1.733	1.917	2.055	1.483	1.643	1.764	1.295	1.438	1.545
24	1.816	2.000	2.138	1.554	1.714	1.835	1.357	1.500	1.607
25	1.890	2.083	2.221	1.626	1.786	1.907	1.420	1.563	1.670
26	1.983	2.167	2.305	1.697	1.857	1.978	1.482	1.625	1.732
27	2.066	2.250	2.388	1.769	1.929	2.050	1.545	1.688	1.795
28	2.149	2.333	2.471	1.840	2.000	2.121	1.607	1.750	1.857
29	2.233	2.417	2.555	1.911	2.071	2.192	1.670	1.813	1.920
30	2.316	2.500	2.638	1.983	2.143	2.264	1.732	1.875	1.982
34	2.649	2.833	2.971	2.269	2.429	2.550	1.982	2.125	2.232
38	2.983	3.167	3.305	2.554	2.714	2.835	2.232	2.375	2.482
42	3.316	3.500	3.638	2.840	3.000	3.121	2.482	2.625	2.732
46	3.649	3.833	3.971	3.126	3.286	3.407	2.732	2.875	2.982
50	3.983	4.167	4.305	3.411	3.571	3.692	2.982	3.125	3.232
54	4.316	4.500	4.638	3.697	3.857	3.978	3.232	3.375	3.482
58	4.649	4.833	4.971	3.983	4.143	4.264	3.482	3.625	3.732
62	4.983	5.167	5.305	4.269	4.429	4.550	3.732	3.875	3.982
66	5.316	5.500	5.638	4.554	4.714	4.835	3.982	4.125	4.232
70	5.649	5.833	5.971	4.840	5.000	5.121	4.232	4.375	4.482
74	5.983	6.167	6.305	5.126	5.286	5.407	4.482	4.625	4.732
78	6.316	6.500	6.638	5.411	5.571	5.692	4.732	4.875	4.982
82	6.649	6.833	6.971	5.697	5.857	5.978	4.982	5.125	5.232
86	6.983	7.167	7.305	5.983	6.143	6.264	5.232	5.375	5.482
90	7.316	7.500	7.638	6.269	6.429	6.550	5.482	5.625	5.732

Contributed by A. Wasbauer, Standardisation Engineer, The Ingersoll Milling Machine Company.

holes in the gears and the gear faces and bosses, must be considered. Unless the hole is large enough to permit a mandrel of sufficient size to be used, the gear cannot be held rigidly for either cutting or grinding the teeth, if that be the finishing operation. The bosses of the gears must be parallel and at right angles with the holes. Holes must be held to close limits and mandrels must fit them accurately. Figure 15 shows a group of gears from a British automobile plant, all designed with a view of making it easy to hold them while the teeth are being cut or ground. In only two cases, *E* and *G*, are the gear faces thicker than the hubs, which permits continuous cutting. In all the other gears shown the thickness of the hubs leaves more or less space between the teeth where the wheel or cutter is doing no work. These are fairly typical of automotive practice where cluster gears are not used.

Cutting Oils and Compounds.—Use of cutting lubricant for gears varies as with other shop operations. Cutting oils are generally considered best where a smooth finish is desired. Where high speeds are used, however, commercial soluble compounds will usually carry the heat away more rapidly than oil. These coolants are largely used for rough cuts in steel, sometimes with a little mineral oil added to improve the finish.

Mineral lard oil usually gives good results in finishing, but there is no universal rule that can be relied on. Soft, low-carbon steel, for example, is more apt to tear than to cut freely. This can sometimes be overcome by adding 10 per cent of kerosene, and powdered sulphur frequently makes a satisfactory addition to the cutting oils.

Regular brass and bronze gears are usually hobbed dry, but some of the special bronze mixtures are so hard and tough that a lubricant is advisable. Some use a strong solution of soapsuds while others prefer a mixture of mineral lard oil and kerosene.

Cooling compounds are sometimes used in roughing cuts on cast iron but are rarely necessary. In any case the finishing cut is usually more satisfactory when no compound is used.

When to Use Cutting Compounds.—Decision as to whether a cooling compound or a cutting oil is to be used is dependent entirely upon the nature of the material being cut. If the amount of metal being removed generates considerable heat, it will require a better cooling medium than can be found in the list of

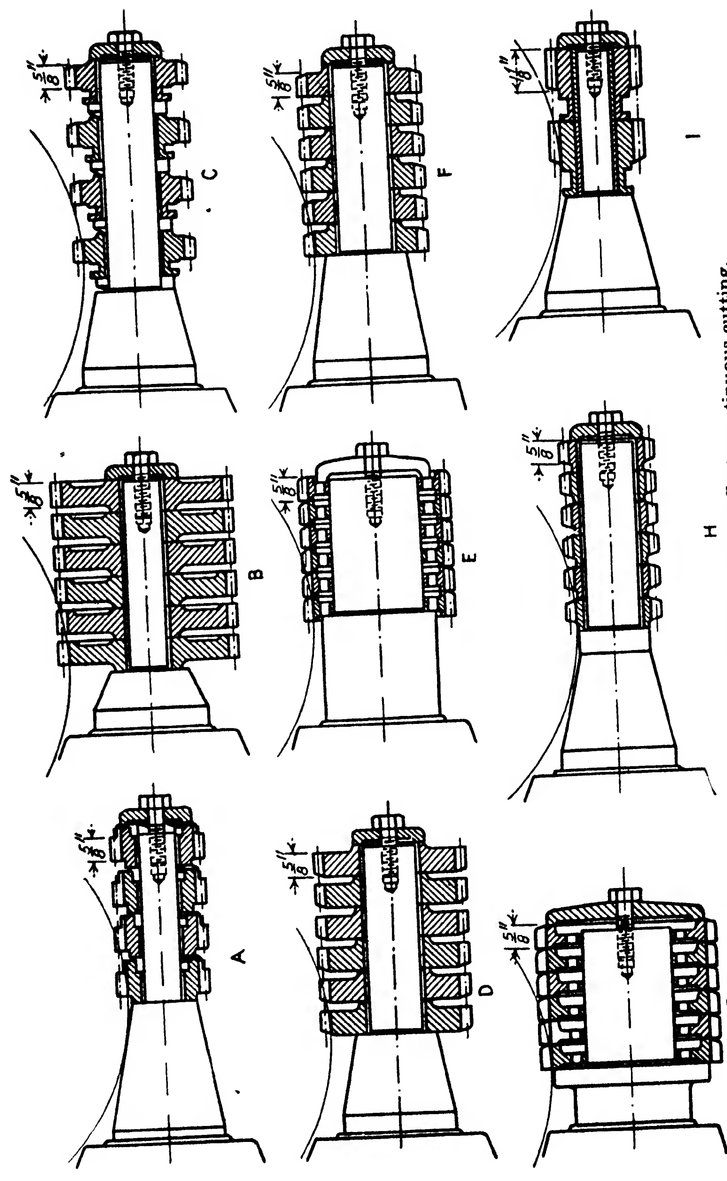


FIG. 15.—Showing how design of gears affects continuous cutting.

oils. On the other hand, it is fairly safe to say that the regular soluble oils or cutting compounds, which have a large percentage of water in their make-up, are well suited to conditions where considerable heat is generated. The better the finish required, the more oil will be required in the compound.

Some materials, such as cast iron, are generally cut dry, for the reason that when a cooling compound is used, the fine dust is formed into a paste by the action of the coolant; this adheres to the cutting edges of the cutter and soon dulls its keen edge. If it were possible to prevent the formation of a paste, the speed of cutting could be considerably increased when cutting cast iron over that used when cutting steel. As a matter of fact, however, practically the same speed is used for cutting cast iron and steel. When a coolant is used, it is preferable to apply it only on the roughing operation and to do the finishing operation dry. Various combinations are made to obtain cutting compounds for machinery and alloy steels, depending upon whether the operation is a roughing or a finishing one. It might be stated, however, that conditions vary so much that it is impossible to state definitely just when a certain compound or cutting oil is satisfactory until it has been tried out under actual working conditions.

Cutting Oils for Machinery and Soft Steel.—Owing to the "stringy" nature of low-carbon or machinery steel, it is sometimes found that the chips cling to the edge of the cutting tool and do not cut clean, but tear from the side of the gear tooth. This trouble is more noticeable in the case of fine-pitch gears, due, of course, to the lack of chip clearance. When a good finish cannot be secured with commercial cutting compounds, the trouble may be overcome, in most cases, by using mineral-lard oil to which sulphur is added. These sulphurized mineral oils can now be easily obtained. Cases have been found when the steel was so soft that even this compound did not bring satisfactory results, and when all other methods of preventing extreme tearing have failed, a commercial grade of olive oil has proved effective.

Cutting Oils for Alloy and High-carbon Steels.—Alloy steels vary so much in their chemical constituents that it is difficult to give any definite information as to the cutting oil or compound that should be used. For automobile gears, an alloy steel that is very common is chrome-nickel, the nickel content varying from 0.3 to 0.5 per cent. Practice varies considerably in regard to whether a cooling compound or a cutting oil should be used. In general, however, it will be found that for roughing, a cooling compound is used, and for finishing, a cutting oil; generally mineral-lard oil or a comparatively cheap grade of motor oil gives the best results, although there are exceptions to this rule. When the steel contains from 0.25 to 0.35 per cent carbon, a good commercial finish can be secured by the use of commercial soluble cutting compounds; these are sometimes mixed with mineral-lard oil when a good smooth finish is desired.

Commercial soluble cutting compounds are generally used in a diluted form, the amount of water added being governed by the nature of the material being cut. A reduction of 20 per cent is generally found satisfactory, although the amount of dilution is also dependent upon the constituents of the compound used. For finishing, an oil commercially known as No. 1 lard oil is quite extensively used, and this is sometimes diluted with kerosene. Even on alloy steels, trouble is sometimes experienced with tearing, especially when the carbon content is low, and when this is the case, an addition of powdered sulphur to the lubricant will generally bring relief.

For cutting very hard or high-carbon steel, a water compound is not satisfactory, and when trouble is experienced with lard oil or mineral-lard oil, it is sometimes possible to improve the cutting action by the addition of kerosene, or, as a last resort, a mixture of kerosene and turpentine. A mixture of kerosene and turpentine is used when finishing steel gears that have been heat-treated after the roughing operation.

Cooling Compounds for Brass, Bronze, and Aluminum.—Ordinary commercial cast brass is cut dry, especially when taking a finishing cut, but for roughing, a soluble oil or compound will help to keep the cutter cool; although it will add little to its cutting efficiency. The same is true of ordinary bronze; although when cutting naval or tobin bronze it is generally found necessary to use mineral-lard oil and kerosene to prevent tearing.

For aluminum a soluble compound is used and instead of adding water, kerosene is used. Sometimes the proportion of kerosene to the compound is about 50 per cent, the percentage, of course, being governed by the constituents of the compound.

Cutting Fibrous Materials.—Materials, such as fiber, rawhide, cloth, Micarta, and similar materials are generally cut dry, and owing to the lack of strength of these materials, it is advisable to use a dummy gear to support the material against the thrust of the cutter. When the cutter is cutting on the "pull" stroke, the gear should be located on top of the fiber blanks, and when cutting on the "push" stroke, on the bottom.

Automobile and Other Gears.—R. B. Schenck, of the Buick Motor Car Company, in a paper before the Society of Automotive Engineers in April, 1936, enumerated 10 steels that were being used by 20 automobile builders for transmission gears. These were 2515, 3145, 4615, 4620, 4640, 5135, 5140, 5145, 5150, and 6150. These include two distinct groups with respect to carbon content, heat treatment, and physical properties. The low-carbon group includes steels 2515, 4615, and 4620, the last two numbers denoting the points of carbon. The other steels are in the high-carbon group and represent about 90 per cent of the gears used.

Transmission gears are forged either in an upsetter or under a hammer. Small gears, such as the reverse idler, are sometimes machined from bar stock. With forged gears the design of the dies is very important as the flow lines about the axis of the gear must be uniform and uninterrupted. The flow can be seen by the process of etching the surface and examining with the microscope.

Forging temperature is also important as there is a temperature range for each steel that gives best results. While the old-style forge furnace which is manually operated and depends on the skill of the operator is still in general use, there is a tendency to adopt continuous-heating furnaces in which the temperature is controlled by thermostat.

The characteristics of the steels mentioned are as follows:

No. 2115. Yield point when annealed from 30,000 to 37,000 lb. When heat-treated from 37,500 to 65,000 lb. This steel is recommended for ring gears on the axle.

No. 2315. Yield point when annealed from 35,000 to 45,000 lb. When heat-treated from 40,000 to 80,000 lb. This steel is recommended for drive pinion gears.

No. 2520. Yield point when annealed from 40,000 to 50,000 lb. When heat-treated from 60,000 to 100,000 lb. This steel is recommended for pinion gears, steering worm sectors, and other high-duty requirements.

No. 3145. Yield point when annealed from 50,000 to 65,000 lb. When heat-treated from 115,000 to 240,000 lb. This steel is recommended for transmission gears and shafts.

No. 3250. Yield point when annealed from 55,000 to 70,000 lb. When heat-treated from 150,000 to 265,000 lb. This steel is recommended for transmission gears and high-duty work.

No. 5150. Yield point when annealed from 60,000 to 80,000 lb. When heat-treated from 150,000 to 275,000 lb. This steel is recommended for transmission gears and other parts where hardness and high elastic limit are needed.

The recommended heat treatment for these steels will be found in Chap. X, on pages 255 to 270.

Steels Used.—Gear steels used by Buick Motor Company in 1936 were as follows:

The steel is made by the open-hearth process and is purchased to specification G.M. 3145-A, which is identical with S.A.E. 3145 except for the carbon range, the suffix "A" denoting a five-point range—in this case 0.43–0.48. This material, with minor changes in carbon range from time to time, has been in use almost continuously for the past 17 years.

In addition to chemistry, the requirements cover grain size, normality, banding, inclusions, and macrostructure. Samples are taken from each heat of steel as it is received and the material is held in the steel yard until released by the Metallurgical Department.

Physical Properties.—Low-carbon gears are characterized by a heavy case and a soft core, and high-carbon gears by a light case and a hard core.

The case and core characteristics of low-carbon gears usually fall within the following ranges: case depth, 0.030 to 0.050 in.; case hardness, Rockwell C-55 to 62; core hardness, C-30 to 40. A rule for depth of case, which has been found quite satisfactory for low-carbon gears in a number of instances, specifies "twice as much core as case." Correctly interpreted, this means a case depth equal to one-sixth of the thickness of the tooth at its base. Thus, a tooth 0.240 in. thick at the base would require a case of 0.040.

With high-carbon gears, the case and core characteristics usually range as follows: case depth, 0.001 to 0.010; case hardness, C-50 to 58; core hardness, C-48 to 56. As core hardness increases, less case is required until a point is reached where the slightly higher hardness of the case over that of the core is insufficient to justify more than a very slight depth, such as that obtained with the "dip" method of hardening. It seems probable that, with a core hardness of C-50 or more, a cyanide case does little more than improve wear resistance and has only a slight effect upon fatigue.

For purposes of stress analysis, a gear tooth may be regarded as an intermittently loaded cantilever beam in which the bending stress varies from a maximum at the surface to zero at the center. Since the bending stress decreases from the surface inward, the outer portions of the core are stressed to a degree dependent upon the depth of case. A relation, therefore, exists between depth of case and the resistance of case and core to fatigue. In high-carbon gears, where the hardness of case and core is so nearly the same, this relation is usually ignored, but in low-carbon gears, where large differences in hardness exist, case depth becomes a factor of major importance. Fatigue may start in either case or core, depending upon which is overstressed.

The case and core characteristics of Buick transmission gears cover the following ranges: case depth, 0.001 to 0.003; case hardness, C-50 to 55; core hardness C-50 to 55. The same ranges are given for both case and core, since the extremely light case has almost no effect on the Rockwell hardness. The tensile strength of the core lies between 275,000 and 300,000 lb. per square inch.

Gear Materials and Blanks.—The following data of American recommended practice, as approved by the American Standards Association July 31, 1933, will be of service in connection with the other information concerning steels recommended for gears by the Society of Automotive Engineers. This gives the chemical composition of the steels used, showing the different alloys now being used. These alloys and the proportions are of course, liable to change from time to time.

American Recommended Practice

Forged and Rolled Carbon Steel. 1. *Material Covered.*—This specification covers steel for gears in three groups, according to heat treatment, as follows: (*a*) case-hardened gears. (*b*) unhardened gears, not heat-treated after machining, and (*c*) hardened and tempered gears.

2. *Basis of Purchase.*—Forged or rolled gear steels shall be purchased on the basis of the requirements as to chemical composition specified in Table 15. Class N steel will normally be ordered in 10 point carbon ranges within these limits. Requirements as to physical properties have been omitted, but when they are called for the requirements as to carbon shall be omitted.

3. *Process.*—The steels may be made by either or both the open-hearth and electric-furnace processes.

4. *Discard.*—A sufficient discard shall be made from each ingot to secure freedom from injurious piping and undue segregation.

5. An analysis of each melt of steel shall be made by the manufacturer to determine the percentages of the elements specified. This analysis shall be made from drillings taken at least 1/8 in. beneath the surface of a test ingot obtained during the pouring of the melt. The chemical composition thus determined shall be reported to the purchaser or his representative and shall conform to the requirements specified.

6. Analysis may be made by the purchaser from one or more bars or forgings representing each melt. The chemical composition thus determined shall conform to the requirements specified in Table 15. Drillings for analysis shall be taken at any point not closer to the center than midway between the center and the surface, but not within 1/8 in. of the surface of the bar or forging.

7. The material shall be free from injurious defects and shall have a workman-like finish. Cold-finished bars shall have a bright, smooth surface.

8. (*a*) The melt number shall be legibly stamped on each bar or forging 4 in. or over in thickness, and on those of smaller section when so specified. (*b*) The identification marks specified on the order to be stamped on gear blanks shall be placed on the web or in such a position that they will not be obliterated in machining.

9. The inspector representing the purchaser shall have free entry at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manufacture of the material ordered. The manufacturer shall afford the inspector, without charge, all reasonable facilities to satisfy him that the material is being furnished in accordance with these specifications.

10. (*a*) Unless otherwise specified, any rejection based on tests made in accordance with this specification shall be reported within 10 working days from the receipt of samples. (*b*) Material which shows injurious defects while being finished by the purchaser will be rejected, and the manufacturer shall be notified.

11. Samples tested in accordance with this specification which represent rejected material, shall be preserved for two weeks from date of test

report. In case of dissatisfaction with the results of the tests, the manufacturer may make claim for a rehearing within that time.

Forged and Rolled Alloy Steel. 1. *Material Covered.*—This specification covers alloy-steel for gears, in two classes according to heat treatment, as follows: (*a*) case-hardened gears, and (*b*) hardened and tempered gears.

2. *Basis of Purchase.*—Forged and rolled alloy gear steels shall be purchased on the basis of the requirements as to chemical composition specified in Table 16. Requirements as to physical properties have been omitted.

6. Analyses may be made by the purchaser from one or more bars or forgings representing each melt. The chemical composition thus determined shall conform to the requirements specified above. Drillings for analysis may be taken at any point midway between the center and the surface of solid forgings or bars, or cuttings may be taken off the entire end surface; when sampling, the surface material for a depth of at least ⅛ in. shall be discarded.

Note: Paragraphs 3, 4, 5, 7, 8, 9, 10, and 11 are identical with those of the specifications for forged and rolled carbon steel.

Steel Castings.—1. It is recommended that steel castings for cut gears be purchased on the basis of chemical analysis, that only two types of analysis be used, one for case-hardened gears and the other for both untreated gears and those which are to be hardened and tempered.

2. *Process.*—The steel is to be made by the open-hearth, crucible or electric-furnace processes. The converter process is not recognized.

3. *Discard.*—Sufficient risers shall be provided to secure soundness and freedom from undue segregation. Risers shall not be broken off the unannealed castings by force. Where risers are cut off with a torch the cut shall be at least ½ in. above the surface of the castings, and the remaining metal removed by chipping, grinding, or other noninjurious method.

4. Steel for use in gears shall conform to the requirements, as to chemical composition as indicated in Table 17.

5. All steel castings for gears must be thoroughly normalized or annealed, using such temperature and time as will eliminate entirely the characteristic structure of unannealed castings.

6. An analysis of each melt of steel shall be made by the manufacturer to determine the percentages of the elements specified. This analysis shall be made from drillings taken at least ⅛ in. beneath the surface of a test ingot obtained during the pouring of the melt. The chemical composition thus determined shall be reported to the purchaser or his representative, and shall conform to the requirements specified.

TABLE 15.—CHEMICAL COMPOSITION OF FORGED AND ROLLED CARBON STEEL

Use	Class	Carbon	Manganese	Phosphorus	Sulphur
Case-hardened...	C	0.15 to 0.25	0.40 to 0.70	Max. 0.045	Max. 0.055
Untreated...	N	0.25 to 0.50	0.50 to 0.80	Max. 0.045	Max. 0.055
Hardened (or untreated).	H	0.40 to 0.50	0.40 to 0.70	Max. 0.045	Max. 0.055

TABLE 16.—CHEMICAL COMPOSITION OF FORGED AND ROLLED ALLOY STEEL

No.	Carbon	Manganese	Phos. Max.	Sulph. Max.	Nickel	Chrome	Vanadium		Molybdenum
							Min.	Desired	
2315	0.10–0.20	0.30–0.60	0.04	0.05	3.25–3.75				
2350	0.45–0.55	0.50–0.80	0.04	0.05	3.25–3.75				
2512	0.17 max.	0.30–0.60	0.04	0.05	4.75–5.25				
3115	0.10–0.20	0.30–0.60	0.04	0.05	1.00–1.50	0.45–0.75			
3215	0.10–0.20	0.30–0.60	0.04	0.045	1.50–2.00	0.90–1.25			
3250	0.45–0.55	0.30–0.60	0.04	0.045	1.50–2.00	0.90–1.25			
3312	0.17 max.	0.30–0.60	0.04	0.045	3.25–3.75	1.25–1.75			
3340	0.35–0.45	0.30–0.60	0.04	0.045	3.25–3.75	1.25–1.75			
6120	0.15–0.25	0.30–0.60	0.04	0.045	...	0.80–1.10	0.15	0.18	
6150	0.45–0.55	0.50–0.80	0.04	0.045	...	0.80–1.10	0.15	0.18	
4615	0.10–0.20	0.30–0.60	0.04	0.050	1.50–2.00	...	..	..	0.20–0.30

TABLE 17.—CHEMICAL COMPOSITION OF STEEL CASTINGS

Use	Class	Carbon	Manganese	Phosphorus		Sulphur
				Acid	Basic	
Case-hardening ...	C	0.15 to 0.25	0.40 to 0.60	0.06 max.	0.05 max.	0.06 max.
Untreated or hardened	H	0.30 to 0.40	0.40 to 0.60	0.06 max.	0.05 max.	0.06 max.

7. Analyses may be made by the purchaser from one or more castings representing each melt. The chemical composition thus determined shall conform to the requirements specified above. Drillings for analysis shall be taken at any point not closer to the center than midway between the center and the surface, but not within ⅛ in. of the surface of the casting.

8. (*a*) The castings shall conform substantially to the shapes and sizes indicated by the patterns or drawings submitted by the purchaser. When dimensioned drawings are provided, the foundry shall take all responsibility for correctness as to shrinkage.

(*b*) The castings shall be free from injurious defects and have a workman-like finish. Defects which do not impair the strength of the castings may be welded by an approved process in a manner satisfactory to the purchaser. The metallic-electrode method of electric welding is an approved process. No welding shall be done in a manner to conceal defects. The defects shall be cleaned out to solid metal before welding, and when so required by the inspector, shall be submitted to him in this condition for his approval. All steel castings welded by the foundry shall be heat-treated before delivery.

9. The melt number and foundry symbol shall be legibly stamped or cast on each casting 6 in. or over in thickness or diameter and on castings of smaller section when so specified.

10. (*a*) The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manufacture of the castings ordered. The manufacturer shall afford the inspector, without charge, all reasonable facilities to satisfy him that the castings are being furnished in accordance with these specifications.

(*b*) If in the case of important castings for special purposes, surface inspection in the green state is required, this shall be so specified in the order.

(*c*) All tests (except check examination for analysis and annealing) and inspection shall be made at the place of manufacture prior to shipment, unless otherwise specified, and shall be so conducted as not to interfere unnecessarily with the operation of the works.

11. Castings which show injurious defects or fail to pass check examinations as to chemistry or heat treatment subsequent to their acceptance at the manufacturer's works will be rejected and the manufacturer shall be notified.

12. Samples tested by the purchaser, which represent rejected castings, shall be preserved for two weeks from the date of the test report. In case of dissatisfaction with the results of the tests the manufacturer may make claim for a rehearing within that time.

Bronze and Brass Castings. 1. *Material Covered.*—These specifications cover nonferrous metals for spur, bevel, and worm gears, bushings, and flanges for composition gears.

2. *Basis of Purchase.*—This material shall be purchased on the basis of chemical composition.

3. *Process.*—The alloys may be made by any approved method.

Use and Chemical Composition

4. (*a*) For spur and bevel gears, hard cast bronze, A. S. T. M. B-10-18, S. A. E. No. 62, and the well-known 88-10-2 mixture to the following limits:

Copper	86-89%
Tin	9-11%
Zinc	1- 3%
Lead (max.)	0.20%
Iron (max.)	0.06%

Good castings made from this bronze should give the following minimum physical characteristics:

Ultimate strength	30,000 lb. per sq. in.
Yield point	15,000 lb. per sq. in.
Elongation in 2 in	14%

(*b*) For bronze worm gears, two alternative analyses of phosphor bronze are recommended, S. A. E. 65 and 63.

S. A. E. No. 65, phosphor gear bronze.

Copper	88 to 90%
Tin	10 to 12%
Phosphorus	0.1 to 0.3%
Lead, zinc and impurities (max.)	0.5%

Good castings made of this alloy should give the following minimum physical characteristics:

Ultimate strength	35,000 lb. per sq. in.
Yield point	20,000 lb. per sq. in.
Elongation in 2 in	10%

S. A. E. No. 63, called leaded gun metal.

Copper	86 to 89%
Tin	9 to 11%
Lead	1 to 2.5%
Phosphorus (max.)	0.25%
Zinc and impurities (max.)	0.50%

Good castings made of this alloy should give the following minimum physical characteristics:

Ultimate strength	30,000 lb. per sq. in.
Yield point	12,000 lb. per sq. in.
Elongation in 2 in	10%

These alloys, especially No. 65, are adapted to chilling for hardness and refinement of grain. No. 65 is to be preferred for use with worms of great hardness and fine accuracy. No. 63 is to be preferred for use with unhardened worms.

(*c*) For bronze bushings for gears, S. A. E. No. 64 is recommended, of the following analysis:

Copper	78.5 to 81.5%
Tin	9 to 11%
Lead	9 to 11%
Phosphorus	0.05 to 0.25%
Zinc (max.)	0.75%
Other impurities (max.)	0.25%

Good castings of this alloy should give the following minimum physical characteristics:

Ultimate strength	25,000 lb. per sq. in.
Yield point	12,000 lb. per sq. in.
Elongation in 2 in.	8%

(*d*) For brass flanges for composition pinions, A. S. T. M. B-30-32T and S. A. E. No. 40 are recommended. This is a good cast red brass, of sufficient strength and hardness to take its share of load and wear under conditions of style No. 2 of the A. G. M. A. standards for composition gearing, a design in which the flanges mesh with the mating gear, and therefore ample for style No. 1. The composition is as follows:

Copper	83 to 86%
Tin	4.5 to 5.5%
Lead	4.5 to 5.5%
Zinc	4.5 to 5.5%
Iron (max.)	0.35%
Antimony (max.)	0.25%
Aluminum	None

Good castings made from this alloy should give the following minimum physical characteristics:

Ultimate strength	27,000 lb. per sq. in.
Yield point	12,000 lb. per sq. in.
Elongation in 2 in	16%

5. An analysis of each melt shall be made by the manufacturer. The chemical composition thus determined shall be reported to the purchaser or his representative and shall conform to the above specifications.

6. (*a*) The sample for chemical analysis may be taken either by sawing, drilling, or milling the casting, or test coupon, and shall represent the average cross section of the piece.

(*b*) The saw, drill, cutter, or other tool used shall be thoroughly cleaned. No lubricant shall be used in the operation, and the saw dust or metal chips shall be carefully treated with a magnet to remove any particles of iron.

7. (*a*) Inspection may be made at the manufacturer's works where castings are made, or at the point at which they are received, at the option of the purchaser.

(*b*) If the purchaser elects to have inspection made at the manufacturer's works, the inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manufacture of the material ordered. The manufacturer shall afford the inspector, free of cost, all reasonable facilities to satisfy him that the material is being furnished in accordance with these specifications. All tests and inspection shall be so conducted as not to interfere unnecessarily with the operation of the works.

8. Castings which show injurious defects revealed by machining operations subsequent to acceptance may be rejected, and, if rejected, shall be replaced by the manufacturer free of cost to the purchaser. The full weight of the original material rejected shall be returned to the manufacturer.

Non-metallic Gears and Pinions.—This is a brief outline of the practice recommended by the American Gear Manufacturers Association.

Preferred Pitch.—The pitch of the gear or pinion should bear a reasonable relation to the transmitted horsepower and speed, or to the applied torque.

Two preferred pitch tables are given, their choice being optional. Table 18 is based on the horsepower load at a given pitch-line velocity, while Table 19 is based on the applied torque.

The torque used in Table 19 is the pounds torque at 1-ft. radius, which for any given horsepower and speed can be obtained from the formula:

$$T = \frac{5252 \text{ hp.}}{\text{r.p.m.}}$$

These preferred pitch tables are applicable to both rawhide and phenolic laminated materials.

Bore Sizes.—For plain phenolic laminated pinions, *i.e.*, without metal end plates, a drive fit of 0.001 in. per inch of shaft diameter should be used. Above 2.5 in. diameter shaft, the fit should be constant at 0.0025 to 0.003 in.

Where metal reinforcing end plates are used, the drive fit should be of the same standard as used for metal.

Relation of Bore to Pinion Diameter.—The root diameter of the pinion of phenolic laminated type should be such that the minimum distance from the edge of the keyway to the root diameter shall be at least equal to the depth of tooth.

For rawhide pinions, this point is covered under the American Gear Manufacturers Association "Adopted Standard for Rawhide Gears," revision of 1933.

Keyway Stresses.—On a plain phenolic laminated gear or pinion, the keyway stress should not exceed 3,000 lb. per square inch.

The keyway stress is calculated by the formula:

$$S = \frac{33{,}000 \times \text{hp.}}{PLS \times A}$$

If the keyway stress formula be expressed in terms of shaft radius and r.p.m. it would read:

$$S = \frac{63{,}000 \times \text{hp.}}{\text{r.p.m.} \times r \times A}$$

S = unit stress, pounds per square inch.
hp. = horsepower transmitted.
PLS = peripheral speed of shaft, foot per minute.
A = area of keyway in pinion, square inch. Length by height.
r = shaft radius.

When the design is such that the keyway stresses exceed 3,000 lb. per square inch, metal reinforcing end plates may be used. Such end plates should not extend beyond the root diameter of the teeth. The distance from the outer edge of the retaining bolt to the root diameter of the teeth shall not be less than a full tooth depth.

The use of drive keys should be avoided, but if required, metal end plates should be used on the pinion to take the wedging action of the key.

Mating Gear.—The mating gear should be of cast iron or hard steel. Soft steel, brass, or soft bronze should be avoided. The teeth should be cut and in good condition for best results.

For phenolic laminated pinions, the face of the mating gear should be the same or slightly greater than the pinion face.

Standard Keyways for Gears.—In connection with the bores of gears the question of keyways is equally important. Therefore the American Gear Manufacturers Association has adopted standard sizes of keys for different bores and also tolerances that will be observed. These standards bear the date of July, 1934,

TABLE 18

Hp. rating	Up to 1,000 P.L.V.* D.P.	1,000 to 2,000 P.L.V. D.P.	Over 2,000 P.L.V. D.P.
¼ to 1	8 to 10	10 to 12	12 to 16
1 to 2	7 to 8	8 to 10	10 to 12
2 to 3	6 to 7	7 to 8	8 to 10
3 to 7½	5 to 6	6 to 7	7 to 8
7½ to 10	4 to 5	5 to 6	6 to 7
10 to 15	3 to 4	4 to 5	5 to 6
15 to 25	2½ to 3	3 to 4	4 to 5
25 to 60	2 to 2½	2½ to 3	3 to 4
60 to 100	1¾ to 2	2 to 2½	2½ to 3
100 to 150	1½ to 1¾	1¾ to 2	2 to 2½

* Pitch-line velocity.

TABLE 19

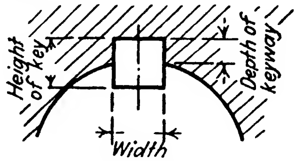
D.P.	Torque, ft.-lb. Min. Max.
16	1–2
12	2–4
10	4–8
8	8–15
6	15–30
5	30–50
4	50–100
3	100–200
2½	200–450
2	450–900
1½	900–1,800
1	1,800–3,500

and the sizes now conform to the recommended practice of the American Standards Association. The standards are shown in accompanying tables with their explanatory diagrams.

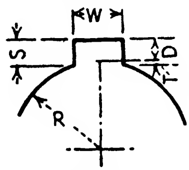
Standard Limits and Tolerances on Holes in Gears.—The importance of having the hole in the hub of a gear of correct size in order to secure proper fit on the shaft and satisfactory operation in service, led the American Gear Manufacturers Association to adopt a standard of recommended practice as to tolerances and limits. Both the allowance and the tolerance are divided into three general classes according to the kind of work for which the gears are intended. These classes are:

Depth of Keyways.—The depth of keyways shall be one-half of the key height measured at the edge according to the diagrams on page 51.

The accompanying table is also used for plain- and gib-head taper keys with the standard taper of 1/8 in. per foot where the depth shown is the deep end of the keyway.



Where the depth of keyway is taken from the center line of bore instead of the vertical wall as recommended, use the formulas as follows:



$$T = R - \sqrt{R^2 - \left(\frac{W}{2}\right)^2}$$

$$S = D + T$$

TABLE 20.—RECOMMENDED KEYSTOCK AND KEYWAYS FOR HOLES IN GEARS

Diameter of holes inclusive, in.	Recommended standard		
	Keyways		Keystock
	Width	Depth	
5/16 to 7/16	3/32	3/64	3/32 × 3/32
1/2 to 9/16	1/8	1/16	1/8 × 1/8
5/8 to 7/8	3/16	3/32	3/16 × 3/16
15/16 to 1 1/4	1/4	1/8	1/4 × 1/4
1 5/16 to 1 3/8	5/16	5/32	5/16 × 5/16
1 7/16 to 1 3/4	3/8	3/16	3/8 × 3/8
1 13/16 to 2 1/4	1/2	1/4	1/2 × 1/2
2 5/16 to 2 3/4	5/8	5/16	5/8 × 5/8
2 13/16 to 3 1/4	3/4	3/8	3/4 × 3/4
3 5/16 to 3 3/4	7/8	7/16	7/8 × 7/8
3 13/16 to 4 1/2	1	1/2	1 × 1
4 9/16 to 5 1/2	1 1/4	7/16	1 1/4 × 7/8
5 9/16 to 6 1/2	1 1/2	1/2	1 1/2 × 1
6 9/16 to 7 1/2	1 3/4	5/8	1 3/4 × 1 1/4
7 9/16 to 8 15/16	2	3/4	2 × 1 1/2
9 to 10 15/16	2 1/2	7/8	2 1/2 × 1 3/4
11 to 12 15/16	3	1	3 × 2
13 to 14 15/16	3 1/2	1 1/4	3 1/2 × 2 1/2
15 to 17 15/16	4	1 1/2	4 × 3
18 to 21	5	1 3/4	5 × 3 1/2

Recommended Specials*

Keyways		Keystock
Width	Depth	
$\frac{1}{8}$	$\frac{3}{64}$	$\frac{1}{8} \times \frac{3}{32}$
$\frac{3}{16}$	$\frac{1}{16}$	$\frac{3}{16} \times \frac{1}{8}$
$\frac{1}{4}$	$\frac{3}{32}$	$\frac{1}{4} \times \frac{3}{16}$
$\frac{5}{16}$	$\frac{1}{8}$	$\frac{5}{16} \times \frac{1}{4}$
$\frac{3}{8}$	$\frac{1}{8}$	$\frac{3}{8} \times \frac{1}{4}$
$\frac{1}{2}$	$\frac{3}{16}$	$\frac{1}{2} \times \frac{3}{8}$
$\frac{5}{8}$	$\frac{7}{32}$	$\frac{5}{8} \times \frac{7}{16}$
$\frac{3}{4}$	$\frac{1}{4}$	$\frac{3}{4} \times \frac{1}{2}$
$\frac{7}{8}$	$\frac{5}{16}$	$\frac{7}{8} \times \frac{5}{8}$
1	$\frac{3}{8}$	$1 \times \frac{3}{4}$

* These sizes are suggested as special when conditions make it necessary to deviate from standard.

It is understood that these keys are to be cut from cold-finished stock and are to be used without machining, as this American Gear Manufacturers Association standard is for general industrial practice. The keystock is to be cold-rolled steel 0.10 to 0.20 carbon.

Keystock to vary from the exact nominal size in width and thickness to a negative tolerance is as follows:

	Inch
Keys $\frac{3}{32}$ square to $\frac{3}{8}$ in. square inclusive......	0.002
Keys $\frac{1}{2}$ square to $\frac{3}{4}$ in. square inclusive.......	0.0025
Keys $\frac{7}{8}$ square to $1\frac{1}{2} \times 1$ in. flat inclusive.....	0.003
Keys $1\frac{3}{4} \times 1\frac{1}{4}$ to 3×2 in. flat inclusive......	0.004
Keys $3\frac{1}{2} \times 2\frac{1}{2}$ to $5 \times 3\frac{1}{2}$ in. flat inclusive...	0.005

Keyways to be cut from exact nominal size to plus 0.002 in width, and depth shall be nominal to plus $\frac{1}{64}$ for straight keys. For taper keys depth shall be from nominal to $\frac{1}{64}$ minus.

For heat-treated pinions the depth shall be $\frac{1}{32}$ to $\frac{3}{64}$ in. over nominal size with $\frac{1}{32}$-in. radius in corners of keyway.

Recommended Practice for Holes.—

Class 1—Precision gears for aircraft, printing machinery, etc.

Class 2—Automobile, machine tool, etc.

Class 3—Pumps, hoisting machines, and general-jobbing gears.

These groupings are primarily for the use of the gear shop, and the order of the groups has no relation whatever to the quality of the gear. In all cases the user of the finished gear is to receive a first-class gear, for the work in hand whether for an aeroplane, an automobile, a machine tool or a pump; or whether a gear is cut, punched, or cast.

TABLE 21.—TABLE OF LIMITS FOR HOLES FOR THREE CLASSES OF WORK

Work to Be within Limits of the Class Specified. First-class Job in All Cases

If Plugs or Gages Are Made, Large End of Gage to Be Marked "Not Go," and Small End "Go"

Class 1. Precision—aircraft, printing machines, etc.

	Nominal size	To ½ in. inc.	To 1 in. inc.	To 2 in. inc.	To 3 in. inc.	To 4 in. inc.	To 5 in. inc.	To 6 in. inc.	To 7 in. inc.	To 8 in. inc.	To 9 in. inc.	To 10 in. inc.	To 11 in. inc.	To 12 in. inc.
Not go	+ Limit	0.000	0.000	0.000	0.00025	0.0005	0.0005	0.0005	....	....	0.00075	....	....	0.001
Go	− Limit	0.00025	0.0005	0.00075	0.00075	0.00075	0.001	0.001	....	....	0.001	....	....	0.001
	Tolerance	0.00025	0.0005	0.00075	0.001	0.00125	0.0015	0.0015	....	....	0.00175	....	....	0.002

Class 2. Automobile, machine tools, etc.

	Nominal size	To ½ in. inc.	To 1 in. inc.	To 2 in. inc.	To 3 in. inc.	To 4 in. inc.	To 5 in. inc.	To 6 in. inc.	To 7 in. inc.	To 8 in. inc.	To 9 in. inc.	To 10 in. inc.	To 11 in. inc.	To 12 in. inc.
Not go	+ Limit	0.00025	0.0005	0.00075	0.001	0.00125	0.0013	0.00175	....	....	0.002	....	....	0.0025
Go	− Limit	0.00025	0.0005	0.00075	0.001	0.001	0.0012	0.00125	....	....	0.002	....	....	0.0025
	Tolerance	0.0005	0.001	0.0015	0.002	0.00225	0.0025	0.003	....	....	0.004	....	....	0.005

Class 3. Standard jobbing gears

	Nominal size	To ½ in. inc.	To 1 in. inc.	To 2 in. inc.	To 3 in. inc.	To 4 in. inc.	To 5 in. inc.	To 6 in. inc.	To 7 in. inc.	To 8 in. inc.	To 9 in. inc.	To 10 in. inc.	To 11 in. inc.	To 12 in. inc.
Not go	+ Limit	0.0005	0.00075	0.001	0.00125	0.0015	0.00175	0.002	....	....	0.003	....	....	0.004
Go	− Limit	0.0005	0.00075	0.001	0.00125	0.0015	0.00175	0.002	....	....	0.003	....	....	0.004
	Tolerance	0.001	0.0015	0.002	0.0025	0.003	0.0035	0.004	....	....	0.006	....	....	0.008

The present recommendation is for hole diameters ranging from ½ to 12 in. Sizes outside of this range can be called special, and be made to special specifications.

Gears, in general, are either made to special specifications furnished by the customer, or to the best judgment of the maker. In all cases the maker furnishes the workman with gages, or sizes to limits.

There are limits of *allowance* and limits of *tolerance*. All work must be within certain allowable limits to function properly. This limit is called the allowance, and depends on the kind of fit desired; whether

Force fits, where hydraulic pressure is used.

Shrink fits, using heat.

Driving fits, using reasonable bumping force.

Snug fits or *wringing fits*, using manual force.

Running fits of the various kinds, close or easy.

These different kinds of fits are best obtained by gages, made to suit the definite requirements of the various shops.

The amounts to allow for the different fits have been tabulated in various handbooks and may, or may not, be closely followed to suit different materials and equipment. Changes also are always to be expected due to the evolution of machine-shop processes. One of the best tables for the above different fits has been compiled by the Newall Engineering Co., and is printed in the American Machinist Handbook.

At the present period of gear standardization we believe the limits of tolerance are of first importance. There is a constant unavoidable variation in work, from the exact size or allowance necessary to produce a given fit or a given specification due to wear of cutting tools and gages, to errors of workmanship, and also to the human element especially when setting tools to gages.

As a matter of economy in production it is advisable to tolerate as large a variation as will function properly. When that limit is overstepped, the work must be scrapped.

To obtain this economy in quantity production it was customary to have a set of tolerances for the operator slightly closer than those of the inspector, the desire being to increase the skill of the worker and reduce rejections of work. The operators soon learn and take advantage of the wider limits actually allowed the inspectors, so the better plan now used, is to have both sets of gages the same size.

The accompanying table of tolerances can also be called protection limits. That is, this table should be representative of standard American practice, and gears made to these limits would be of standard quality. It will be noted that the tolerance of Class 2 is about twice that of Class 1, and that of Class 3 about twice that of Class 2 or four times that of Class 1.

CHAPTER II

SPUR GEARS AND CIRCULAR CUTTERS

Cutting Spur Gears.—Although both the shaping and hobbing methods of cutting gears are used almost exclusively in producing gears in large quantities, as in the automotive field, there are

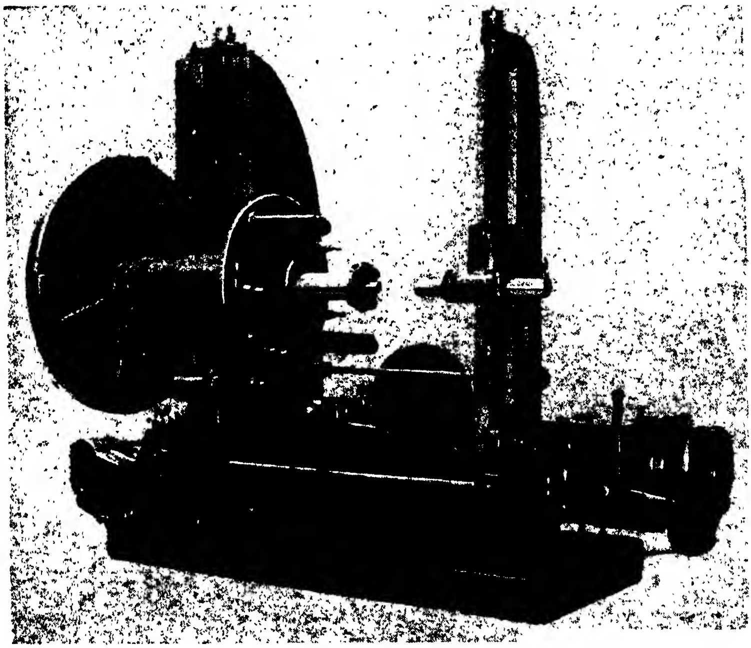


FIG. 16.—Newark 100-in. gear hobber.

many uses for the older type of gear cutter using the single milling cutter and indexing from tooth to tooth, as in the milling machine. This is especially true in large-gear work. A typical example of such a machine is shown in Fig. 16, which is a 100-in. machine, meaning that it will cut teeth in a gear blank 100 in. in diameter and a 24-in. face. The gear blank is carried on a horizontal mandrel, which is supported at the outer end and is carried in a spindle with the large indexing worm wheel at the left. The cutter is mounted on a rigid arbor in a long cutter carriage, or saddle, which feeds along the bed and advances the cutter into the work. In many cases a number of gear blanks can be mounted side by side on the work mandrel so that several gears can be cut at the same time.

The indexing mechanism is an important part of such a machine and particular attention has been paid to both design and construction. This mechanism is positive but is arranged with a safety device so that it is protected in case of accidental collision, for example, when the rim, or a lug on the gear blank, might bind against the rim rest that takes the thrust of the cutter in the work. The large worm wheels used for indexing are made in two parts, each being generated in place on the machine so that it cannot contain errors from another wheel. The generating process is continued until all errors are minimized in the worm wheel. Change gears are provided for cutting all numbers of teeth from 4 to 500, except prime numbers above 100 and their multiples. (Prime numbers are those which cannot be divided by any whole number except these numbers themselves.) An automatic stop prevents the cutter from feeding into the work during the indexing period.

These spur-gear cutters can also be used for roughing out the teeth of bevel gears by use of a special attachment. Either spur or bevel gears up to one-diametral or 3-in. circular pitch can be cut in steel, or 3/4-diametral pitch and 4-in. circular pitch in cast iron.

These machines have a feed screw of heat-treated steel that operates on the draw-cut principle, with thrust bearings at each end of the screw. This keeps the feed screw constantly in tension, the object being to provide a smooth feeding motion to the work.

Formed Cutters.—When formed cutters are used, they must be selected according to the number of teeth in the gear to be cut. There are eight standard cutters in the regular set, represented by whole numbers from 1 to 8. There are also seven intermediate cutters, all of these being shown in the list which follows. In some special cases cutters are made for the exact number of teeth in the gear to be cut. The list of cutters and the data in Table 22 will make it easy to select and use these standard cutters with good results.

Brown and Sharpe Involute Gear-tooth Cutters

No. 1	will cut wheels from	135 teeth to	a rack.
No. 1½	will cut wheels from	80 teeth to	134 teeth.
No. 2	will cut wheels from	55 teeth to	134 teeth.
No. 2½	will cut wheels from	42 teeth to	54 teeth.
No. 3	will cut wheels from	35 teeth to	54 teeth.
No. 3½	will cut wheels from	30 teeth to	34 teeth.
No. 4	will cut wheels from	26 teeth to	34 teeth.
No. 4½	will cut wheels from	23 teeth to	25 teeth.
No. 5	will cut wheels from	21 teeth to	25 teeth.
No. 5½	will cut wheels from	19 teeth to	20 teeth.
No. 6	will cut wheels from	17 teeth to	20 teeth.
No. 6½	will cut wheels from	15 teeth to	16 teeth.
No. 7	will cut wheels from	14 teeth to	16 teeth.
No. 7½	will cut wheels from	13 teeth to	14 teeth.
No. 8	will cut wheels from	12 teeth to	13 teeth.

Table 22.—Depth of Space and Thickness of Tooth in Spur Wheels When Cut with These Cutters

Pitch of cutter	Depth to be cut in gear, in.	Thickness of tooth at pitch line, in.	Pitch of cutter	Depth to be cut in gear, in.	Thickness of tooth at pitch line, in.
1¼	1.726	1.257	11	0.196	0.143
1½	1.438	1.047	12	0.180	0.131
1¾	1.233	0.898	14	0.154	0.112
2	1.078	0.785	16	0.135	0.098
2¼	0.958	0.607	18	0.120	0.087
2½	0.863	0.628	20	0.108	0.079
2¾	0.784	0.570	22	0.098	0.071
3	0.719	0.523	24	0.090	0.065
3½	0.616	0.448	26	0.083	0.060
4	0.539	0.393	28	0.077	0.056
5	0.431	0.314	30	0.072	0.052
6	0.359	0.262	32	0.067	0.049
7	0.308	0.224	36	0.060	0.044
8	0.270	0.196	40	0.054	0.039
9	0.240	0.175	48	0.045	0.033
10	0.216	0.157			

To measure a gear cutter use the distance from the pitch line to the bottom of the space. Cutters for bevel gears are made approximately to the width of the space at the small end of the teeth where the face is one-third the apex.

The Newark Gear Cutting Machine Co. also makes hobbing machines of large capacity. Some of the attachments permit interesting operations to be performed. Such unusual jobs as the hobbing of a multiple-thread worm, with the worm held in a collet chuck can be done on these machines. Another attachment makes it possible to generate worm wheels by the use of fly cutters, star cutters, or tapered hobs.

Herringbone gears can also be cut with an end-milling attachment, or either spur or helical gears cut with a single cutter. These operations will be shown in the chapters devoted to the different kinds of gears.

Fig. 17.—Dividing head arranged to cut gears in a milling machine.

Cutting Gears in the Milling Machine.—Cutting gears in the milling machine is largely a matter of selecting the right cutter and operating the index head so as to secure the correct number of divisions. These are obtained by plain indexing, if the number of teeth in the gear to be cut can be indexed directly from the plates supplied with the head. If this is not the case, the number probably can be secured by compound or differential indexing.

For work of this kind there is no better method than to follow the instructions of Brown and Sharpe for use on their milling machines. These, together with the tables used to find the proper gearing, are given herewith. Figure 17 shows a dividing head geared for differential indexing, and the tables, pages 325 to 333, give the gearing for different numbers of teeth from 2 to 258.

In order to select the proper change gears, it is first necessary to find the ratio of the required gearing between the spindle and plate. After this has been done, the correct gears can be found. The following formulas show the manner in which this gearing is calculated.

N = number of divisions required.

H = number of holes in index plate.

n = number of holes taken at each indexing.
V = ratio of gearing between index crank and spindle.
x = ratio of the train of gearing between the spindle and the index plate.

$$\left.\begin{array}{l} S = \text{gear on spindle.} \\ G_1 = \text{first gear on stud.} \end{array}\right\} \text{Drivers.}$$

$$\left.\begin{array}{l} G_2 = \text{second gear on stud.} \\ W = \text{gear on worm.} \end{array}\right\} \text{Driven.}$$

$$x = \frac{HV - Nn}{H} \text{ if } HV \text{ is greater than } Nn.$$

$$x = \frac{Nn - HV}{H} \text{ if } HV \text{ is less than } Nn$$

$x = S/W$ (for simple gearing).
$x = SG_1/G_2W$ (for compound gearing).

V is equal to 40 on the Brown and Sharpe Universal Index Centers, and the index plates furnished have the following numbers of holes: 15, 16, 17, 18, 19, 20, 21, 23, 27, 29, 31, 33, 37, 39, 41, 43, 47, 49.

The gears furnished have the following numbers of teeth: 24 (two gears), 28, 32, 40, 44, 48, 56, 64, 72, 86, 100.

In selecting the index circle to be used, it is best to select one with a number having factors that are contained in the change gears on hand, for if H contains a factor not found in the gears, x cannot usually be obtained, unless the factor is canceled by the difference between HV and Nn, or unless N contains the factor.

When HV is greater than Nn and gearing is simple, use 1 idler.

When HV is greater than Nn and gearing is compound, use no idlers.

When HV is less than Nn and gearing is simple, use 2 idlers.

When HV is less than Nn and gearing is compound, use 1 idler.

Select "n" so that the ratio of gearing will not exceed 6:1 on account of the excessive stress upon the gears.

A few examples are given herewith to illustrate the application of the above formulas:

Example 1.—*To Cut a Gear with* 59 *Teeth.* $N = 59$. Required H, n and x. Assume

$$H = 33, \quad n = 22.$$

Then

$$x = \frac{(33 \times 40) - (59 \times 22)}{33} = \frac{22}{33} = \frac{2}{3}.$$

We now select gears giving this ratio, as 32 and 48, the 32 being the gear on spindle and the 48 the gear on worm. HV is greater than Mn, and the gearing is simple, requiring one idler.

Example 2.—*To Cut a Gear with* 319 *Teeth.* $N = 319$. Required H, n and x. Assume

$$H = 29, \qquad n = 4.$$

Then

$$x = \frac{(319 \times 4) - (29 \times 40)}{29} = \frac{116}{29} = \frac{4}{1}.$$

When the ratio is not obtainable with simple gearing, try compound gearing.

4/1 can be expressed as follows:

$$\frac{3 \times 4}{1 \times 3} \quad \text{or} \quad \frac{72 \times 64}{24 \times 48}$$

for which there are available gears. HV is less than Nn and the gearing compound, requiring one idler.

To Cut a Gear with 271 *Teeth.*

Figure 18 shows the headstock geared, simple gearing, for 271 divisions. Referring to the table, the gears called for are: C, 56 teeth and E, 72 teeth, with one idler D. The idler D serves to rotate the index plate in the same direction as the crank, thus in making 280 turns of the crank, nine divisions are lost, giving the correct number of divisions, 271. The sector should be set to indicate one-seventh turn, or three holes in the 21 hole circle, and the head is ready for 271 divisions, the indexing being made the same as for plain indexing.

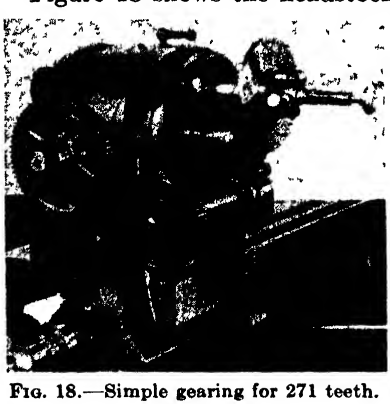


FIG. 18.—Simple gearing for 271 teeth.

To Cut a Gear with 250 *Teeth.*

Figure 19 shows the headstock geared for 250 divisions. This requires two idlers. As given on page 331, the necessary gears are: C, 24 teeth and E, 40 teeth, with the two idlers serving to connect the gears and rotate the index plate in the opposite direction to the crank.

The Principles of Hobbing.[1]—The hobbing process is the generating action that takes place between the hob and the work when cutting gears in a hobbing machine.

This generating action is best demonstrated by noting the action of a rack and a gear in mesh, when the rack is moved forward and backward, rotating the gear. If the gear were made

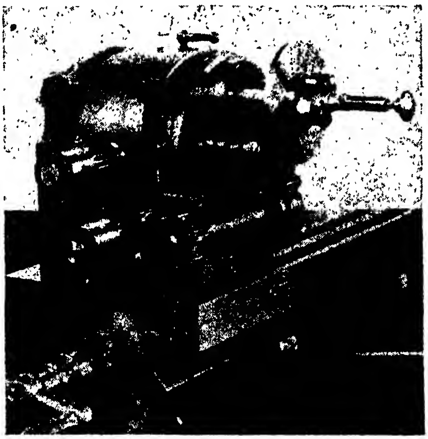


FIG. 19.—Head geared for a 250 toothed gear.

from plastic material, the rack would produce involute-shaped teeth in the gear when so moved.

On a worm, we have such a rack form wound spirally around a cylinder in the shape of a lead or thread. The lead is also called the pitch, and is the circular distance from tooth to tooth measured on the pitch line of the gear. A worm, when revolved while in mesh with a gear, will cause the gear to rotate because of the lead on the worm. This lead causes the rack form to move from one end of the worm to the other, an amount equal to one pitch for each revolution of the worm. The result is similar to moving a rack as described above. If we now add gashes similar to those in form cutters, we have a typical hob for involute or worm gears.

Hobs are made with single or multiple threads. A single-thread hob, when in action, makes one revolution for each tooth

[1] For much of this outline of the principles involved in hobbing we are indebted to Carl J. Oxford of the National Twist Drill Co.

in the gear being cut. A two-threaded or double-thread hob makes one revolution for every two teeth. This relation holds for hobs having any number of threads.

When a spur-gear hob is set up in the hobbing machine, it is swung to the helix angle of the thread, in other words, the paths of the rotating hob teeth are parallel with the teeth in the gear to be cut. By rotating the hob and gear in the proper ratio, a gear having involute form teeth will be produced, at one point only; but by feeding the hob along the axis of the gear blank, a complete gear will have been produced when the hob has traveled across the full width of the gear face.

When cutting worm gears, it is necessary only to set the hob with its axis at right angles to that of the gear, to rotate hob and gear blank in the proper ratio, and to feed the hob into the gear blank to the proper tooth depth or pitch diameter. The hob, being the same diameter and form as the worm, will generate the proper shape so that worm and worm gear will run together.

✓**Hobbing vs. Milling of Gear Teeth.**[1]—There is some confusion regarding the relative merits of these two methods of cutting gear teeth, so that a brief, nontechnical discussion of their main differences, and their respective fields of usefulness, should be of value. This discussion relates only to gear teeth having involute contours, as this system of gear teeth is in almost universal use today.

Hobbing.—When gear teeth are cut with hobs, the contours of the teeth are generated by the relative motion of the hob and the work. If a single-thread hob is used, the hob must make one revolution while the gear being cut rotates the distance of one tooth and one space, measured on the periphery of the pitch circle of the gear. The teeth in the hob usually have straight sides of a predetermined pressure angle, but the rolling action of the gear automatically produces an involute contour on the gear teeth. This is generally spoken of as "generating action."

Hobs having two or more threads may also be used. The only difference of operation is that the gear must rotate a number of teeth equal to the number of threads in the hob for every revolution of the latter, but the contour of the gear teeth will not be affected. With a hob of the proper pitch and shape, it is possible to cut gears having any number of teeth within the

[1] Extracts from a *Bulletin* of National Twist Drill Tool Co.

practical range of the hobbing machine, and to give these teeth the correct involute contour. It should be noted, however, that this contour will change as the number of teeth in the gears change, because of the difference in diameters of the base circles from which the involutes are formed.

Milling.—When gear teeth are milled with a single formed cutter there is this important difference, that no generating action takes place and that the contours of the gear teeth are determined by the contour of the cutter used for milling them.

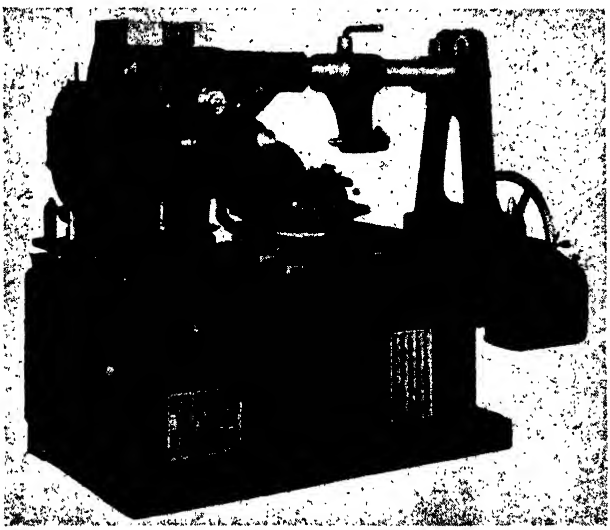


Fig. 20.—Barber-Colman gear-hobbing machine.

Applications.—Where a large number of gears of any given pitch are to be produced, and where suitable hobbing machines are available, the hobbing method is usually preferred to form cutters both from the standpoint of production capacity, and for strict adherence of the tooth contours to the desired involute shape. Using hobs with ground form insures extreme accuracy of tooth spacing and correctness of contour.

On the other hand, where a variety of gears are to be produced in small quantities, or where it is desired to cut only one or two gears of a given size, single cutters are usually more economical to use, because of their lower first cost. A further advantage is that with such cutters gears can be cut on the ordinary milling machine. But it should be remembered that a range cutter, even with ground form, only closely approximates the true involute gear tooth contour.

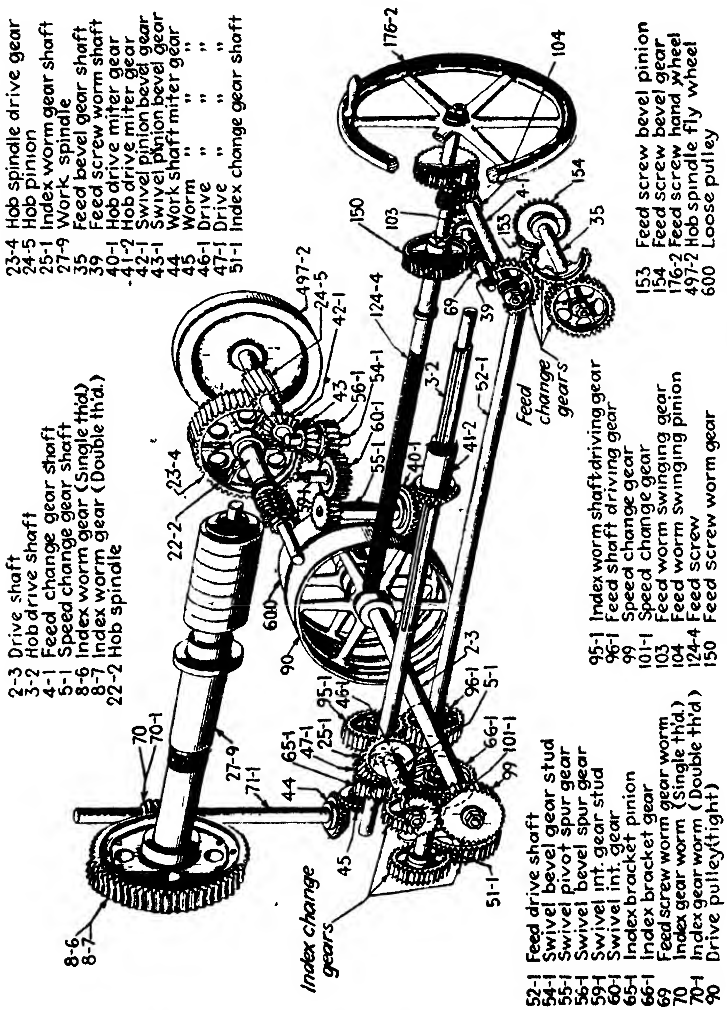


FIG. 21.—Gear-train mechanism of Barber-Colman hobber.

Cutting Gears by the Hobbing Process.—Cutting gears by the hobbing process differs from the older formed-cutter process in two ways: The cutting and indexing are continuous and the teeth are generated. It also differs from the hobbing of worm wheels since, instead of the hob being fed down into the work, it feeds across the face of the gear at the same time that both cutting and indexing are taking place. Either straight spur teeth or helical teeth can be cut by the hobbing process, the only difference being in the angle at which the hob is set to the work. In the case of spur gears this angle depends on the pitch and diameter of the hob while, with helical gears, the helix angle must be added to (or subtracted from in some cases) that of the hob itself. The hob, the work, and the feed must all work in proper ratio to

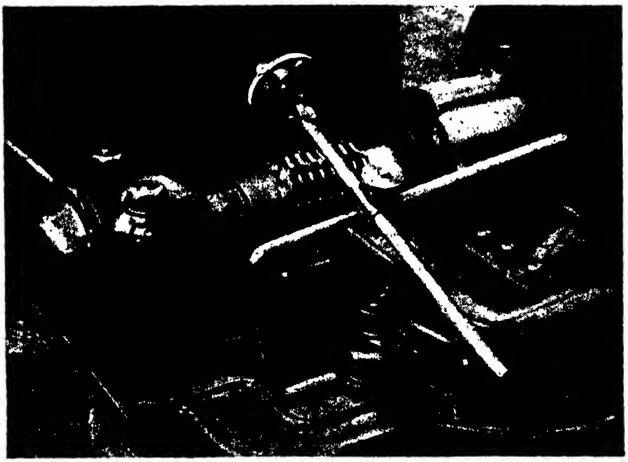


Fig. 22.—Checking concentricity of a hob.

secure good gears. A general view of the Barber-Colman No. 12, hobber is shown in Fig. 20. The gear train is shown in Fig. 21.

One of the first essentials of good gearing is cleanliness of both the hob and gear mountings so that no small particle of dirt may throw either out of alignment. It is necessary also to check both hob and work for concentricity, as in Fig. 22. For finishing fine gears the maximum eccentricity of the hob on the spindle should not exceed 0.00025 in., or 0.000125 in. actual eccentricity.

Proper work-holding fixtures are also necessary, some approved methods being seen in Figs. 23 to 31. It is particularly necessary that the gear blanks be supported as near the bottom of the teeth as possible, especially in the case of spoked gears with a light rim. The general setup is seen in Fig. 23 and details in Figs. 24 and 25. The gears in Fig. 24 have straight sides and support

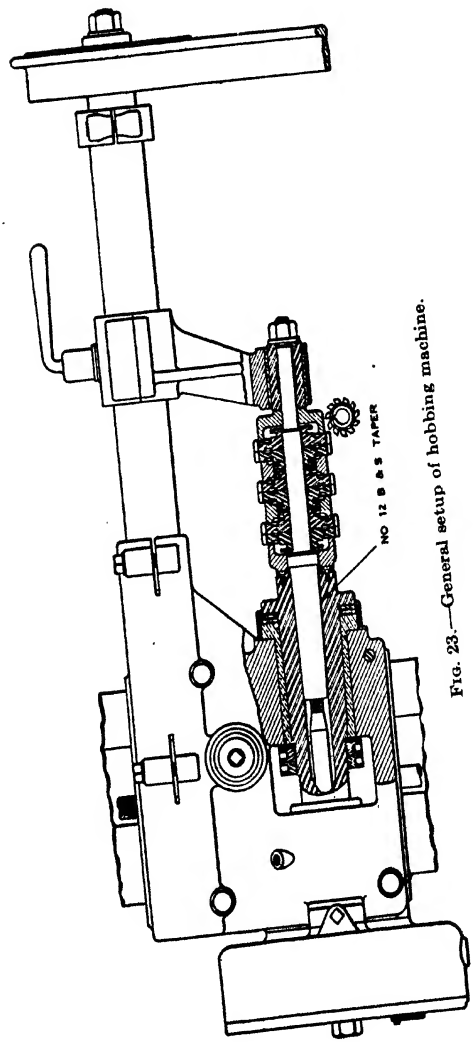


Fig. 23.—General setup of hobbing machine.

themselves, but there must be no burs or dirt to throw them out of line. The ends should be supported by cupped or recessed washers or end plates, as shown. In both Figs. 23 and 25, however, the gear hubs are wider than the face and it is necessary to provide solid supports between the gear blanks. In Fig. 23 one side of the gear has a straight face so that two gears can be

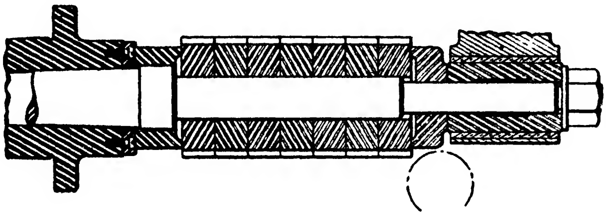


FIG. 24.—A plain gear-blank setup.

made to support each other on one side. But in Fig. 25 it is necessary to provide supports between each gear. It will be noted that all the arbors shown are supported at the outer end by a hardened- and ground-steel bushing that fits in the overarm. This bushing or sleeve is a close sliding fit as no shake is permissible.

In the examples shown the work mandrel has fitted the holes in the gears, but this is not always advisable, as it is sometimes

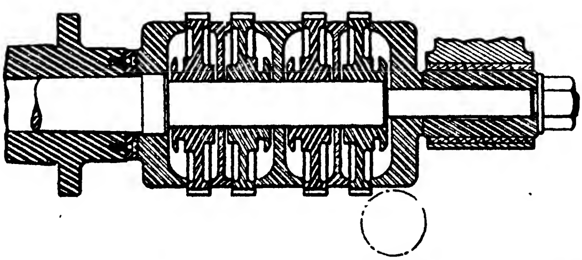


FIG. 25.—Holding clamps and separators for gears with hubs.

more economical to have sleeves for the work as in Fig. 26. Another change that is particularly advisable on work where the cutting time is short is to reduce handling time by using a dead center instead of the sleeve in the overarm and to have the driving end of the work mandrel quickly removable. A method of doing this is seen in Fig. 27. Here the work mandrel is held on centers at both ends, but the positive drive is secured by a flattened end at *AA*, as shown in the detail illustration. With this arrangement the work is easily and quickly removed and another mandrel, which has been loaded during the cut, is put in its place.

Work mandrels should be of steel, hardened and ground, unless they are for a temporary job. The hardened mandrels are better than soft mandrels and much to be preferred in every way where the quantity warrants. They should be carefully finished to run within 0.00025 in. of true and they should be carefully handled, in both the shop and tool crib.

Clamping collars need not fit the mandrels tightly but can be $\frac{1}{64}$ in. large for ease in loading and unloading. They must,

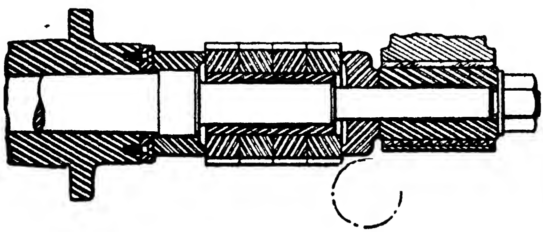


FIG. 26.—Supporting gear blanks on a sleeve.

however, have parallel faces that are very close to being square with the holes. It is best to harden these also if they are for standard equipment, but this is less important than with the work mandrels.

It is necessary in many cases to make the holding fixtures to conform to the piece on which the gear is to be cut. Sometimes, as in Fig. 28, it is convenient to use a collet type of chuck on the finished end of the gear shaft. The collet is operated by the draw in rod through the work spindle. If there is danger of the collet not driving the work positively, a pin or key can be put in the chuck to prevent turning. Still another method, somewhat similar, is shown in Fig. 29, where an expanding mandrel takes the place of the collet. This expansion takes place inside the hollow end of the transmission shaft and both centers and drives it while the teeth are being cut.

When, as sometimes happens, splines are to be cut on the end of a rough shaft, and one that is too long to be held between centers, a larger, hollow work spindle is used, and a center placed at the back end as in Fig. 30. Still another modification of the

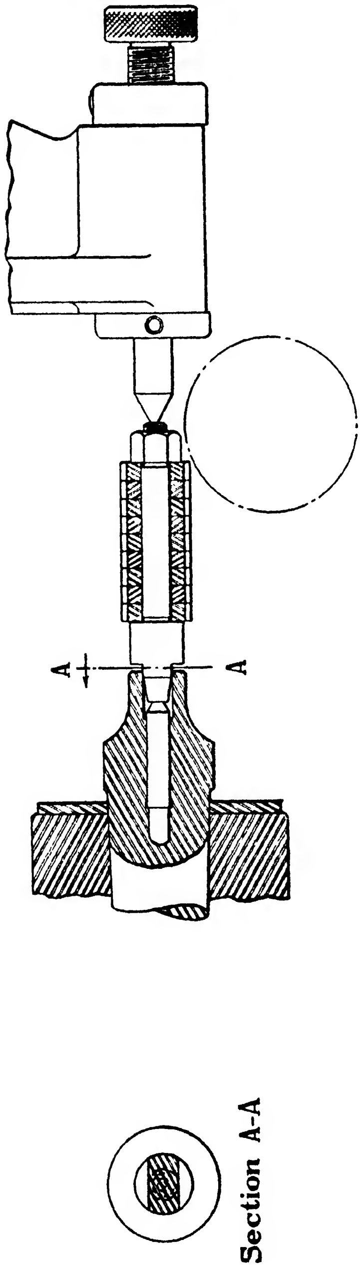


FIG. 27.—A mandrel that can be changed quickly.

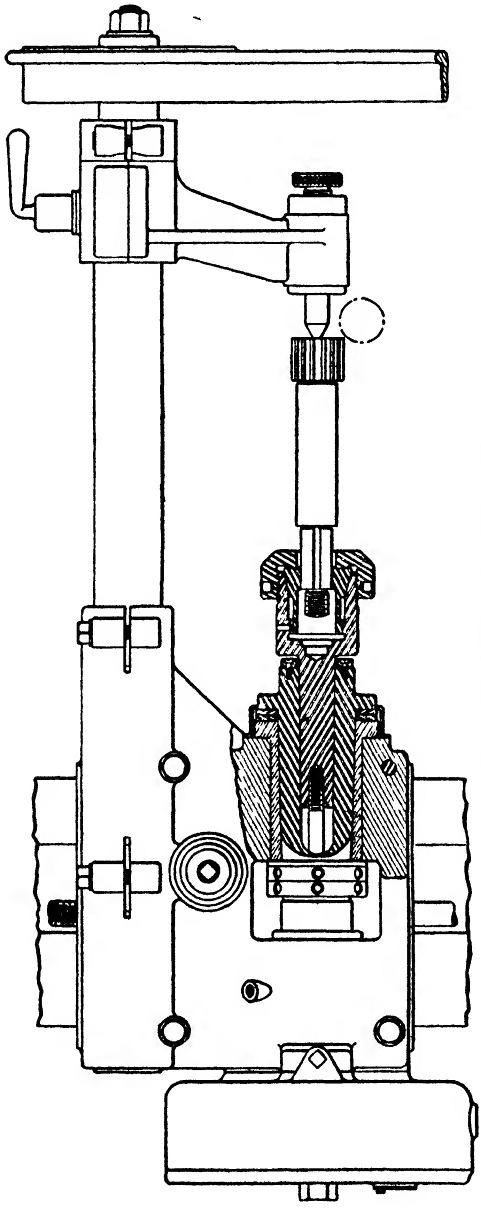


FIG. 28.—Using a collet-type chuck.

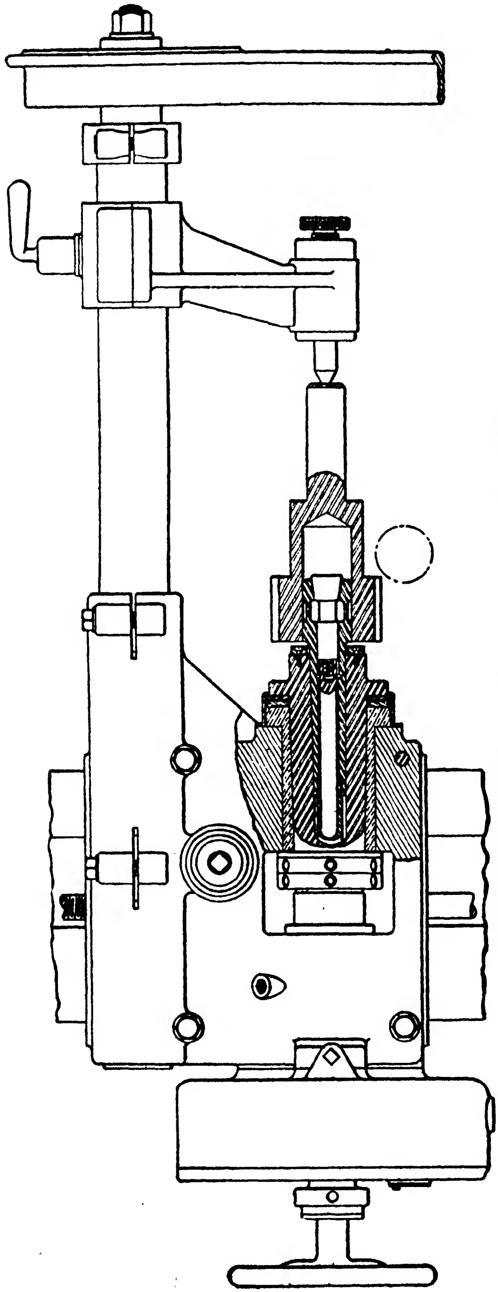


FIG. 29.—Where an expanding mandrel is necessary.

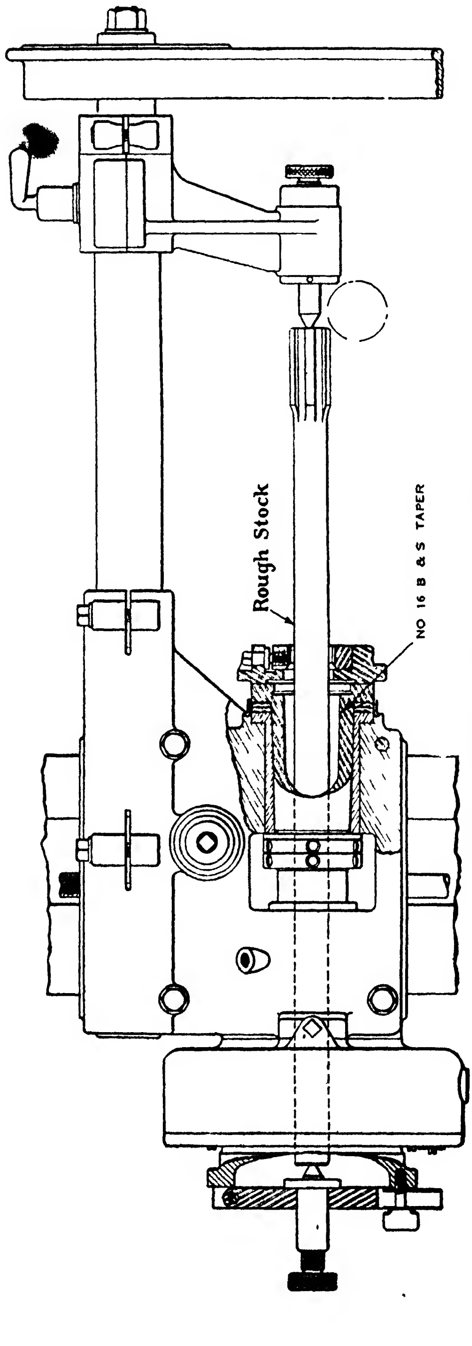


FIG. 30.—Using a back center on long work.

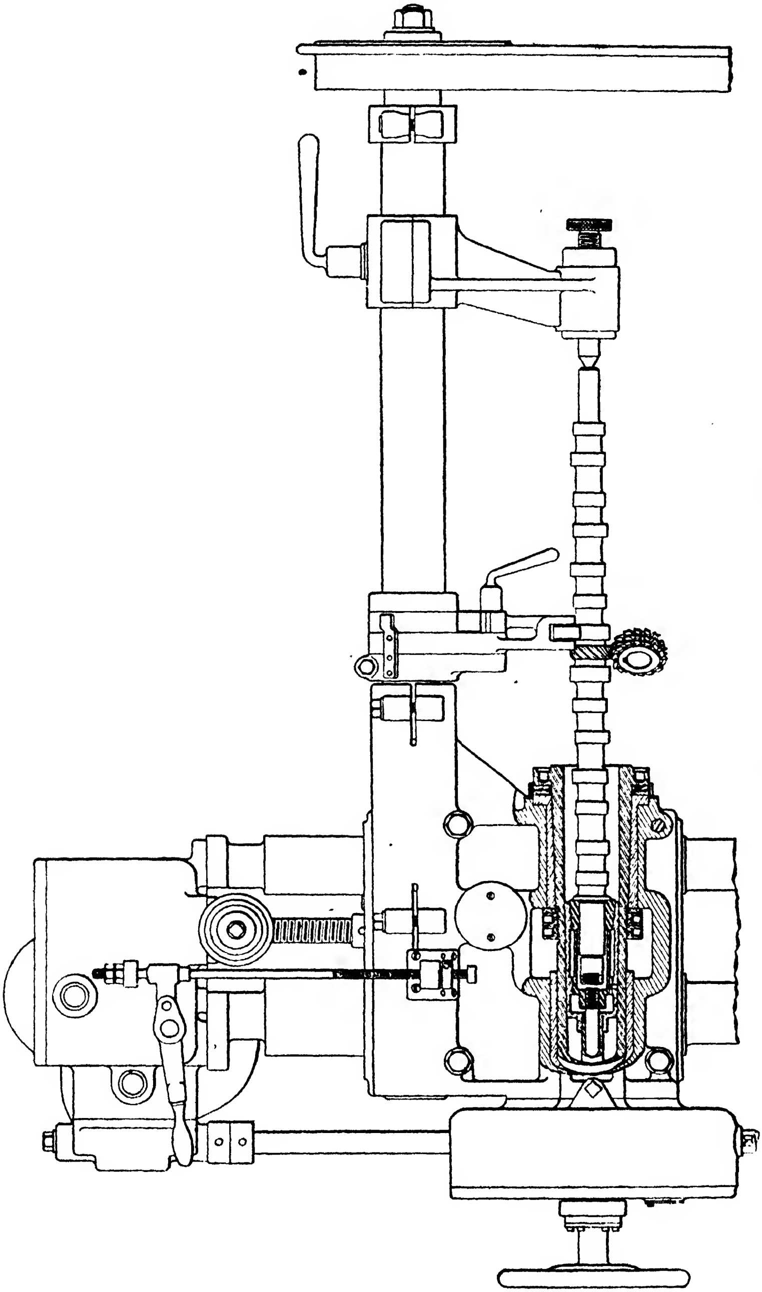


Fig. 31.—Where a backrest is used to advantage.

methods already shown may be seen in Fig. 31 where a camshaft is being held in a collet that is well inside the work spindle. In addition the camshaft is supported against the thrust of the hob by a roller backrest on the overarm. The hob is cutting the gear in the center of the camshaft, from which the timer or oil pump is to be driven.

These examples give a good general idea of the way in which work is held for hobbing.

Setting Up Hobbing Machines.—Owing to the differences in construction of hobbing machines and the fact that all builders

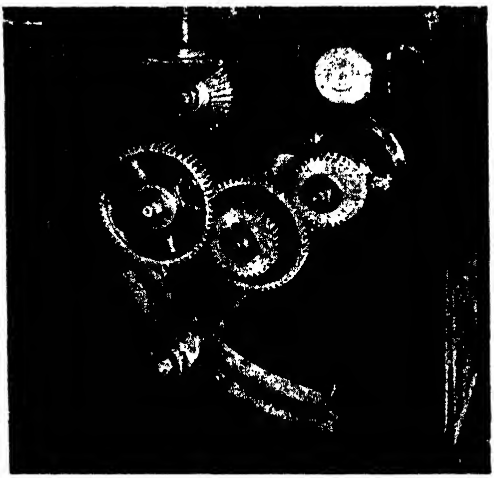
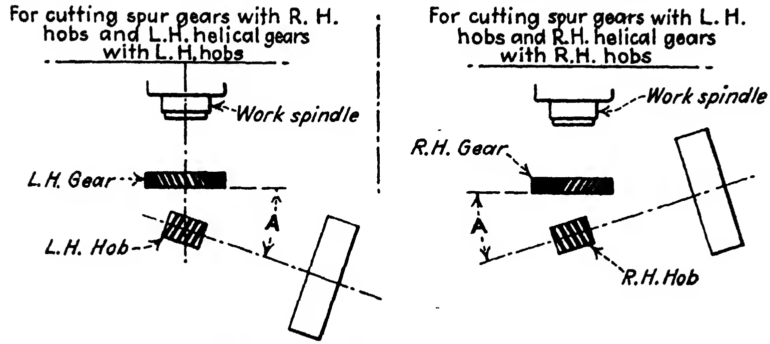


FIG. 32.—Gearing used on a hobbing machine.

supply detailed instructions for the operation of their machines, it does not seem advisable to attempt such instructions in this volume. With an understanding of the principles involved from the information already given, there will be no trouble experienced in handling these machines. The selection of the proper change gears is similar to that involved in the cutting of threads in a lathe, the ratio of the particular machine used being taken instead of the the pitch of the lead screw as in the case of the lathe. This ratio is marked on the machine itself.

We take a No. 12 Barber-Colman machine as an example, where the ratio is 30 to 1. The ratio between the gear to be cut and 30 gives the ratio of the driving and driven gears in the index gears. If we wish to cut a 27 gear, the ratio is 30 to 27, 30 representing the driving gears and 27 the driven gears. In Fig. 32 there are two driving and two driven gears, the second driving gear being on the intermediate stud. Both pairs of gears must be in this ratio, the first pair being 30 and 27 and the second driving gear being 60 and the last driven gear, 54, both of these being double the first pair, simply for convenience. This refers to a



I For spur gears A = lead angle marked on hob
II For R.H. helical gears with R.H. hobs, or L.H. helical gears with L.H. hobs.
A = helical angle minus lead angle marked on hob.
III For R.H. helical gears with L.H. hobs, or L.H. helical gears with R.H. hobs.
A = helical angle plus lead angle marked on hob.

FIG. 33.—Setting right- and left-hand hobs.

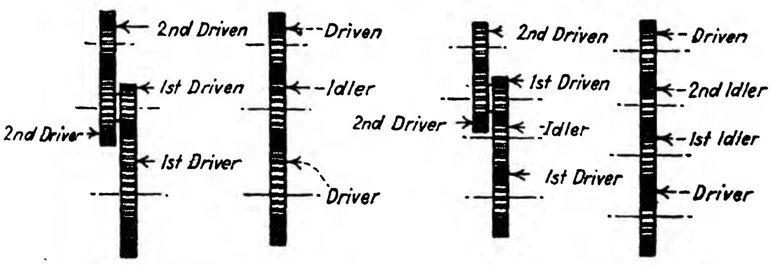


FIG. 34.—Gear trains used for hobs of both "hands."

hob having only a single lead. When it has more than one lead, the ratio changes accordingly. These are not the gears shown in Fig. 32. In hobbing gears the indexing is tied up with the rate of feed, for which tables are necessary. These are given in the instruction books for the different machines.

The swiveling of the hob is a simple matter as shown in Fig. 33. This diagram shows the settings for cutting spur gears with both right- and left-hand hobs, and for cutting helical gears of both hands with hobs of either hand, limited by the factors given on page 142. These diagrams should make the principles clear to those interested. Figure 34 shows the gear trains for both right- and left-hand hobs.

ESTIMATING PRODUCTION

The actual time required for hobbing any piece of work is calculated by the following formula:

$$T = \frac{N \times [L + O]}{F \times \text{r.p.m.}},$$

in which

T = cutting time in minutes.
N = number of teeth in gear to be cut.
L = distance between the outside faces of the work.
O = overrun of the hob, or the distance traveled from the time it first touches the work until it reaches full depth
($L + O$ therefore represents the total distance traveled by the hob while cutting.)
F = feed of the hob in inches per revolution of the work.
R.p.m. = revolutions per minute of the hob.

If a multiple-thread hob is used, the number of teeth (N) in this formula must be divided by the number of threads in the hob.

For example, let us assume that we wish to figure the time for hobbing a 45-tooth, 10-pitch gear without hubs, having a face width of $1\frac{1}{4}$ in., on the No. 12 hobbing machine, holding four gears on the arbor and using a feed of 0.060 in. per revolution of the work with a hob speed of 130 r.p.m. The total face of the gears is then $1\frac{1}{4}$ in. $\times$ 4, or 5 in., while the overrun of a 10-pitch hob, 3 in. in diameter, is approximately $\frac{25}{32}$ in. $L + O$ is therefore equal to $5\frac{25}{32}$ in. Then $T = \dfrac{45 \times 5.78 \text{ in.}}{0.060 \text{ in.} \times 130} = 33.3$ min. cutting time. If $1\frac{1}{2}$ min. are required to remove the finished gears and reload the arbor, the total floor-to-floor time for four gears is about 35 min. equivalent to a production of approximately seven gears per hour.

If the gears have hubs on one or both sides, L must, of course, include the distance between the faces, in addition to the actual face width of the gears.

Selecting Proper Feeds and Speeds.—The conditions governing the choice of appropriate feeds and speeds for hobbing are so many and varied that it is impossible to make any general recommendations which can be relied upon to give the best results in every case.

No recommendations which might be made can have a universal application to all classes of work, or can be followed indiscriminately without careful consideration of the peculiar requirements of each different job. Judgment gained from experience in the operation of the hobbing machine is the best

TABLE 23.—APPROXIMATE FEEDS AND SPEEDS FOR HOBBING SPUR GEARS

Material	Finishing, 6 to 16 diam. pitch, single-thread hob: Feed per rev. of work, in.	Finishing, 6 to 16 diam. pitch, single-thread hob: Speed, r.p.m.	Roughing, 6 to 16 diam. pitch, single-thread hob: Feed per rev. of work, in.	Roughing, 6 to 16 diam. pitch, single-thread hob: Speed, r.p.m.	Roughing, 6 to 16 diam. pitch, double-thread hob: Feed per rev. of work, in.	Roughing, 6 to 16 diam. pitch, double-thread hob: Speed, r.p.m.
Fiber	0.060 to 0.120	180				
Rawhide, Micarta, etc	0.060 to 0.120	180				
Soft brass	0.060 to 0.120	180				
Malleable iron, soft cast iron	0.060 to 0.100	150				
Hard brass, bronze	0.050 to 0 090	150				
Semisteel	0.050 to 0.090	150				
Low-carbon machinery steel	0.045 to 0.075	150	0.090 to 0.120	130	0.075 to 0.100	130
0.35–0.45 carbon steel	0.040 to 0.060	130	0 075 to 0.090	104	0.070 to 0.080	104
Case-hardening alloy steel	0.040 to 0.060	130	0 075 to 0.090	104	0.070 to 0 080	104
0.45–0.60 carbon steel	0 040 to 0.050	104	0.065 to 0.075	77	0.060 to 0 070	77
High-carbon alloy steel	0.030 to 0.045	77	0.045 to 0.060	77	0.040 to 0 055	77

basis for determining suitable feeds and speeds, and may be supplemented where necessary by experiment, until a result is found which gives a satisfactory quality of work combined with the most economical rate of production. Tables 23 and 24 suggesting feeds and speeds are therefore offered only as a guide to what may be considered ordinary practice.

Accuracy and smoothness of finish are usually the factors which decide whether gears shall be completed at one cut or have two operations. This must be decided by those most familiar with the work and the conditions under which it operates.

Tables 23 and 24 represent average conditions for spur gears and shafts. The feeds must be modified to some extent for helical gears, depending on the angle of the teeth. Multiplying the feed selected for a spur gear of corresponding size by the cosine of the angle of the helix will give a good approximation of the feed for the helical gear.

TABLE 24.—APPROXIMATE FEEDS AND SPEEDS FOR HOBBING SPUR GEARS

Material	Finishing		Finishing		Roughing	
	12 to 32 diam. pitch single-thread hob		12 to 32 diam. pitch double-thread hob		12 to 32 diam. pitch double-thread hob	
	Feed per rev. of work, in.	Speed r.p.m.	Feed per rev. of work, in.	Speed r.p.m.	Feed per rev. of work, in.	Speed r.p.m.
Fiber	0.050 to 0.100	533	0.040 to 0 070	453		
Rawhide, Micarta, etc.	0.050 to 0.100	453	0.040 to 0.070	388		
Soft brass	0.050 to 0.080	388	0.040 to 0.065	388		
Malleable iron, soft cast iron	0.050 to 0.070	309	0.040 to 0.060	309		
Hard brass, bronze	0.045 to 0.065	309	0.035 to 0.050	230		
Low-carbon machinery steel	0.040 to 0.060	230	0.035 to 0.045	183	0.060 to 0.075	183
0.35–0.45 carbon steel	0.035 to 0.050	183	....	...	0.050 to 0.065	157
Case-hardening alloy steel	0.035 to 0.050	183	....	...	0.050 to 0.065	157
0.45–0.60 carbon steel	0.035 to 0.045	157	....	...	0.040 to 0.055	133
High-carbon alloy	0.025 to 0.040	133	....	...	0.030 to 0.045	133

Division of Operating Time.—Operating time of hobbing or any other operation can be divided into several groups, most of which affect the operator. First is the setup time, which varies widely with the type of machine and the work being done. This includes the setting of the hob itself and changing when sharpening becomes necessary.

The importance of cutting time depends to a large extent on its relation to the operator's time in handling work in and out of the machine. The faster the cutting time the fewer machines can be handled by one man. This is affected by the layout of the machines and the amount of travel a man must take between machines. There are but few machine departments which do not show machines idle because the operator is busy on another machine in the group. It is usually better economy to have enough men to keep the machines running as nearly as possible to 100 per cent of the time.

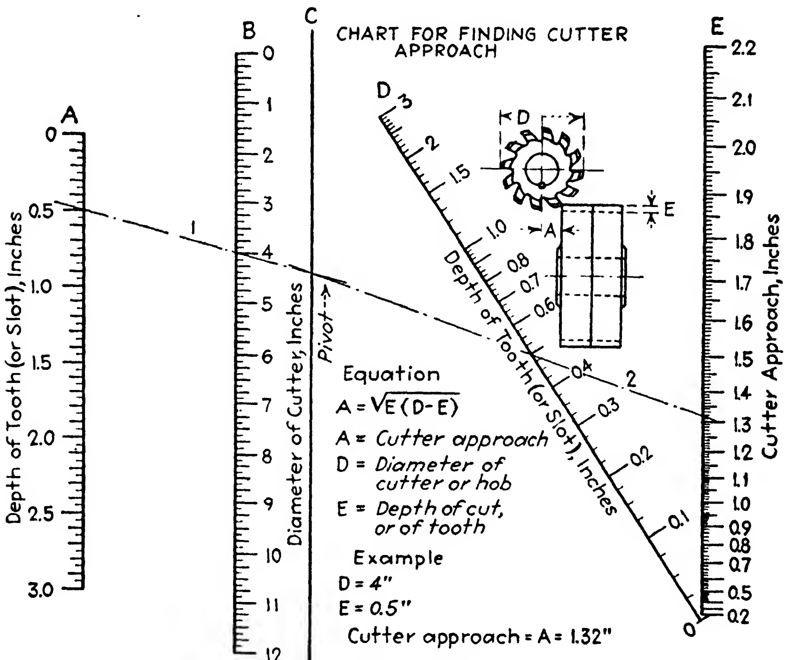


FIG. 35.—Estimating time lost in cutter approach.

Loading and unloading time depends to a great extent on the conveniences provided for handling the work. Quick-change mandrels and other work-holding fixtures must be provided for maximum output. Keeping the work to be handled at a convenient height for the operator will help more than many realize.

Estimating Approach of Cutter.—In estimating the time required for cutting gears allowance must always be made for the lost motion at each end of the travel or stroke, necessary for clearing the work. In the case of the rotary cutter—either the single-formed cutter or the hob—the time of the approach must always be considered. As the approach varies with the diameter of the cutter and the depth of the cut, both these factors must be considered.

The diagram in Fig. 35 makes it easy to solve this problem without any mathematics, although the formula is given above the example. Using this diagram it is necessary only to draw a line from the tooth depth at *A* and through the cutter diameter in column *B*, marking the point of intersection on the pivot line *C*. From this point on *C* draw another line through the angular depth of tooth line, *D*, to the last column *E*. Where this line cuts *E* is the length of the cutter approach in inches. This is equally good for all rotary cutters.

Stabilizer for Hobbing Machines.—The Barber-Colman Co. has developed a stabilizer which adds to the productivity of some types of machine hobbing. The features of this device are described as follows:

In the operation of any hobbing machine there are two critical points, or danger zones when the machine and the hob are both under greater strain than when the machine is operating in the main portion of the cut. The first of these points is when the hob first begins to enter the work and is merely striking with the outer edges of the teeth, the other point being when the hob breaks through the work at the end of the cut.

If the feed and speed should be increased to the degree that the machine might easily stand, the behavior at these two danger points would prevent such feed and speed being used, as the hob would be liable to either dig in when entering the work, or jump forward as the cut ended. This of course would have the effect of producing unacceptable work, limiting production, and shortening the life of both hob and hobbing machine.

To overcome this difficulty, an attachment has been designed and built which is known as a stabilizer. It is really a braking device for creating a "drag" on the hob swivel slide, eliminating backlash in the feed screw and preventing the slide from jumping when the hob chatters on the work.

The stabilizer has proved to be a simple and very effective means of obtaining more satisfactory work with better production, and at the same time of making possible a longer working life for the hob. The first trial made with this device during its development period was a marked success, as the possibility was proved of taking heavier cuts than had ever before been feasible, using the same machine and hob. In fact, it was possible to set the feed and speed of the machine up to the full limit of the belt capacity, until the belt began to slip off the pulleys, without hurting the hob or producing unacceptable work.

Gould and Eberhardt Gear Hobber.—In the Gould and Eberhardt gear hobber the work is supported on the table and the hob is mounted on the vertical slide. Two widely differing jobs are shown in Figs. 36 and 37. One is hobbing a heavy pinion with coarse pitch teeth and the second, the two helical gears on an

Fig. 36.—Hobbing a heavy pinion.

automobile transmission cluster. Figure 38 shows two large helical gears used for power transmission. In getting ready to cut gears on this machine, the spur-gear chart is first consulted to obtain the proper change wheels for the number of teeth to be cut and for the feed. These gears are located and placed in position at the rear of the machine.

The hob cutter is placed upon the cutter arbor and the end support bearing put in position. The cutter is then secured to the arbor by means of the nut, and a space of the cutter set central by the gage furnished. Setting a space central is recommended when cutting gear wheels having a number of teeth less than thirty.

FIG. 37.—Helical gears for automobile transmission.

FIG. 38.—Large helical power-transmission gears.

To set the cutter, the nuts on the spindle bearings are loosened and the cutter-spindle bearings are then adjusted lengthwise by means of the small adjusting screw until a space matches exactly with the gage, when the spindle-bearing bolts are securely fastened and the gage withdrawn from the cutter.

The swivel slide is then adjusted to proper angle, as marked upon the helical cutter, the spur-gear chart being first consulted to get the proper position of the cutter in relation to the perpendicular, which depends on whether the cutter is right or left handed. The swivel slide is then secured to the vertical slide by means of the bolts. Either right- or left-hand hobs may be used. The gears to rotate the cutter in the proper direction are then engaged by means of the handle on the pulley side.

The work mandrel is then inserted in the work table and the work blank securely fastened thereon; care should be taken that the hole and faces of the blank, by which it is clamped, are entirely free from all dirt and burrs, and that the blank is of proper diameter.

The outer support arm is placed in position and the wheel blank is then tested by an indicator to see whether its outer periphery revolves perfectly true with the hole. It is very important to have the blank revolve true to obtain good results in the running of the finished gear.

After the wheel blank has been found correct the machine is allowed to run and the cutter is lowered to a central position in relation to face of blank, care being taken that the blank is clear of the cutter. The blank is then adjusted horizontally by means of a crank handle until the cutter starts to cut the periphery of the blank.

The cutter is then raised above the face of the blank, and the blank is adjusted to its proper depth to be cut. A dial is provided upon the screw to read in thousandths, which is set to zero after the cutter has been raised. The proper depth can then be read therefrom when the blank is adjusted toward the cutter.

When the work blank has been adjusted to the proper depth, the work table is secured to the base by means of the clamping screws. The machine can then be started and the cutter fed to the upper edge of the work by means of the hand wheel and the feed engaged by means of the handle near the pulley.

The dog upon the vertical feed rod is then located so as to stop the machine when the center of the cutter has passed entirely beyond the lower edge of the wheel blank. The machine will automatically generate all of the teeth without further attention upon the part of the operator.

The setting of the gear blank as outlined should not be depended on to give the proper tooth thickness at the pitch line. The teeth should be measured on the pitch line with a good gear-tooth caliper. This should be set to the chordal thickness and corrected addendum, as previously given in Table 8. This measuring should be done when the center of the cutter has just passed the upper edge of the gear blank. If the tooth thickness is not correct, the gear blank should be moved toward, or away from, the cutter, as the case may require.

Spur-gear production can be materially increased by using double-thread hobs. This is particularly true on gears that require two cuts. The roughing cut can be taken with a double-thread hob and the finishing done with a single-thread hob. Where extreme accuracy is not required, such as in automobile flywheel-starter gears, and where there are a large number of teeth, gears are frequently finished with double-threaded hobs.

For cutting helical gears the instructions which accompany each machine should be carefully followed.

All machines of this type should receive great care as to proper lubrication. In addition to the usual oiling of the bearings, they should be thoroughly cleaned at fairly regular periods by washing out all the bearings to be sure that only clean oil gets to the bearing surfaces when the machine is in use.

Cutting lubricant is also important. Cutting compounds which thicken and leave a gummy residue are to be avoided. Special mixtures put out by some of the large oil companies are recommended by the makers of the machine.

Gearing Hobber for Cutting Gears.—The gearing to be used on any make of hobbing machine is shown in tables that accompany the machine. Cutting speeds and feeds are given in Tables 23 and 24. While the principles of hobbing are the same in all machines the instructions of the maker should be carefully followed. This is particularly true where helical gears are to be cut on any machine.

Estimating Production.—The cutting time of gears can easily be estimated after determining the proper feeds and speeds that are to be used on the gears to be cut.

The actual time required to hob a gear can be calculated by using the formula:

$$\text{Cutting time in minutes} = \frac{N \times T}{\text{r.p.m.} \times F}$$

where

N = number of teeth in gear to be cut.

T = total travel of hob slide, which is equivalent to the face of each blank multiplied by the number of blanks cut plus the extra travel of the hob, which is the distance hob travels before it first enters the blank, until it reaches the full depth. If the gears have hubs on one or both sides, the total travel must include the height of hubs in addition to the width or face of the gears.

R.p.m. = revolutions per minute of hob.

F = feed in inches of hob slide, per revolution of blank.

For example, let us assume we are to cut a 24-tooth, 8-pitch 1-in. face gear, without hub, cutting four gears at a time, using a hob speed of 130 r.p.m., 0.050 feed.

The total travel of the hob slide is 1 in. × the number of gears (4) = 4 in. plus 1 in. extra travel making a total travel of 5 in.

Substituting the values in the formula, the following is the solution:

$$\frac{N \times T}{\text{r.p.m.} \times F} = \frac{24 \times 5}{130 \times .05} = \frac{120}{6.5} = 18.46 \text{ min. cutting time.}$$

If 2 min. is added to this for removing the finished gears and reloading the arbor, the total floor-to-floor time would be 20.46 min. for four gears or 5.1 min. each from floor to floor.

The Lees-Bradner Hobbing Machine.—In the Lees-Bradner hobber the hob is carried on a horizontal swiveling table beneath the gear blank or blanks to be hobbed, as in Fig. 39. The work is supported on a mandrel over the hob, the tail-center support sliding on an overarm. The hob swivel head is adjustable through 180 deg. The head is supported on a cylindrical column which permits both swiveling and vertical movement to suit the diameter of the work. The spindle is adjustable horizontally for changing the position of the hob. The work slide moves on square ways which are protected against dirt and chips, and the overarm also slides in a well-supported bearing. There is power quick return and approach, but neither can be engaged until the feed is out and the hob withdrawn from the work. Both the hob and work spindles are worm-driven.

The work spindle has a compound index by which all numbers of teeth from 1 to 200, or more, can be indexed without imposing

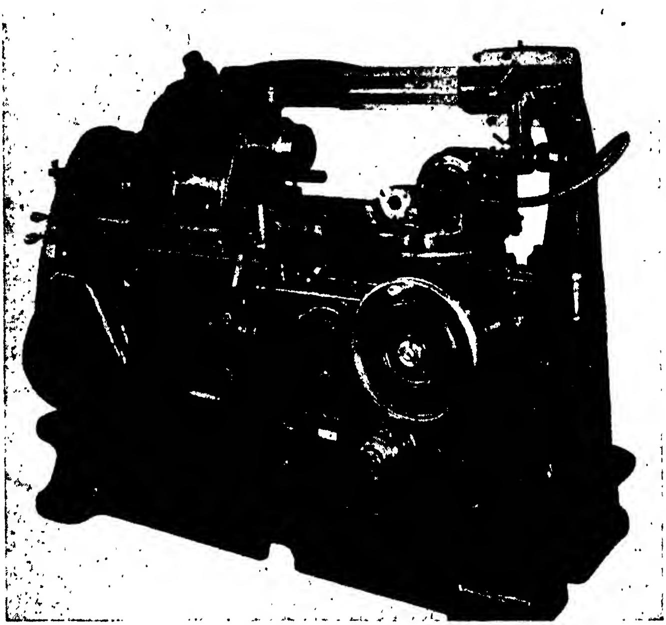


FIG. 39.—Lees-Bradner hobbing machine.

an overload on the indexing mechanism. The calculations are simple. It is necessary only to divide the lead by the feed selected for the job, as charts, which accompany the machine, show the change gears to be used in each case.

The Lees-Bradner Co. also makes a machine in which the hobbing spindle is vertical, as in Fig. 40. The work slide travels past the cutter head on substantial bearing slide, this design having been adopted to secure rigidity and freedom from vibration. The cutter spindle is reversible, which avoids the use of cross belts when driving from the line shaft. This machine has wide range of work, which includes spur and helical gears, spline shafts, serrated shafts, worm wheels, worms, and sprocket wheels.

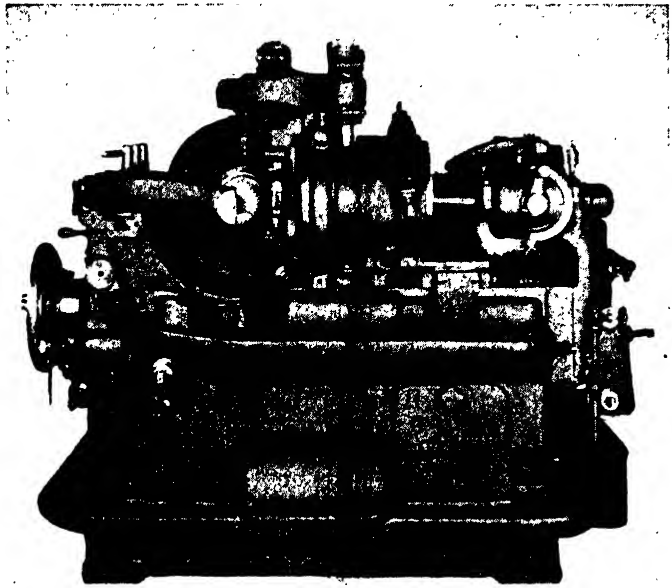


FIG. 40.—A vertical-spindle hobbing machine.

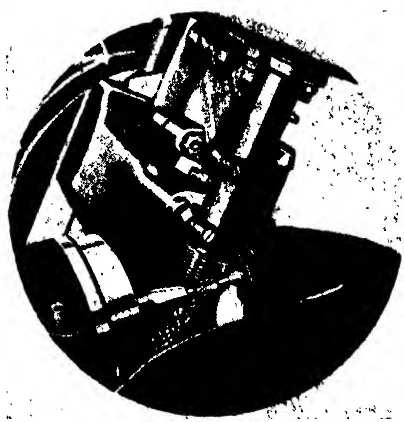


FIG. 41.—Auxiliary high-speed hobbing spindle.

For gears with teeth of 12 pitch or finer, an auxiliary high-speed spindle is provided which bolts to the swivel head, as shown in Fig. 41. It runs at twice the speed of the main spindle, making it especially valuable in cutting small gears, as shown. This auxiliary spindle can also be used for milling the threads of worms as in Fig. 42. The regular spindle can be used in

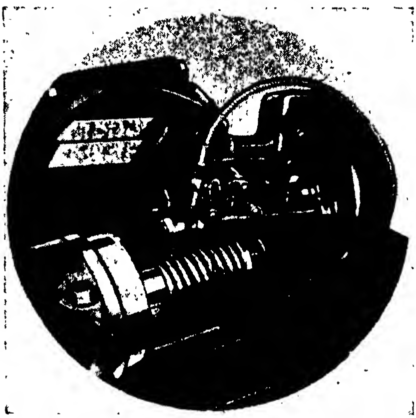


FIG. 42.—Milling worm threads with auxiliary spindle.

cutting worm wheels as in Fig. 43. A special power in-feed is also supplied when needed for feeding the hob into the worm wheel to the proper depth. A micrometer stop prevents the

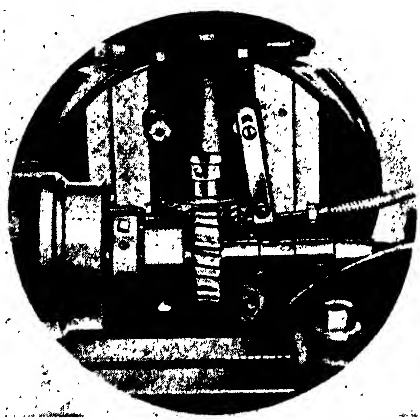


FIG. 43.—Cutting worm wheels with the regular spindle.

hob from feeding below the proper depth. This machine hobs gears up to 19 in. in diameter. For regular use a 3-hp. motor is suggested, while for heavy work a 5-hp. motor is recommended.

Semi-automatic Gear Hobbers.—Semi-automatic gear hobbers are available for mass-production shops, usually within a limited range such as might be used in transmission gears for automotive uses. In machines of this kind the operator merely loads the machine after it has been properly set up by a skilled mechanic. After loading he merely presses a starting button and goes to the

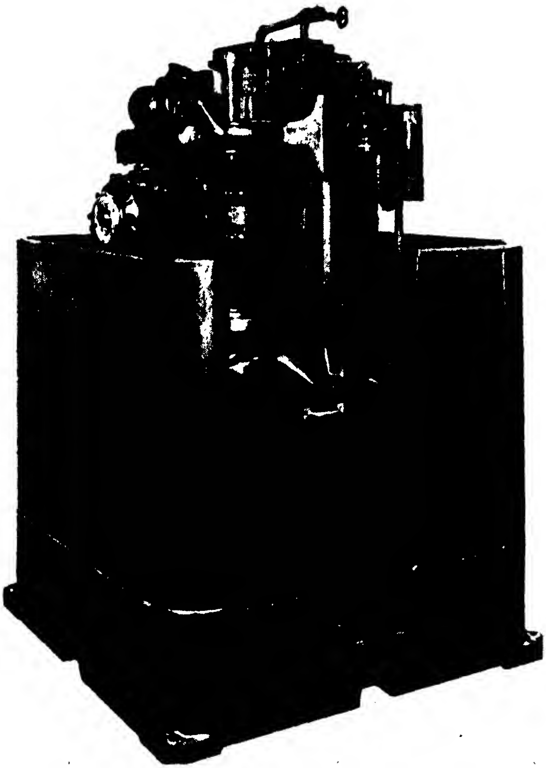


FIG. 44.—A semi-automatic hobbing machine.

next machine. The cutter then feeds rapidly until it reaches the work, then feeds in to depth at the proper rate where it is held until the completion of the gear. The depth stop is automatic and determined by micrometer. When the gear is completed the cutter backs out automatically and the machine stops.

One such machine (Fig. 44) is built by the Lees-Bradner Co. It has capacities up to gears 12 in. in diameter and is driven by a 3-hp. motor. Another automatic hobber is that of the

Cleveland Hobbing Machine Co., shown in Fig. 45. This is a station-type machine with 8 or 10 spindles. Each work head is independent so that work of different lengths can be cut at the same time.

Hobbing Spline Shafts.—One of the common applications of the hobbing machine is the cutting of splines in shafting, largely for the automotive industry. The hobs used depend

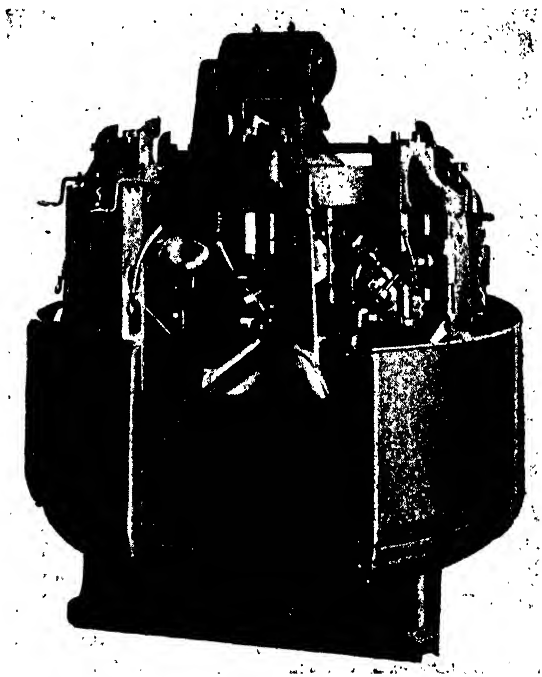


FIG. 45.—A semi-automatic hobber by the Cleveland Hobbing Machine Company.

on the kind of a spline wanted as some bear on the top, some on the bottom, and others on the side. In many cases the plain spline shown at *A* in Fig. 46 is satisfactory, and the hob shown is all that is necessary. This leaves an arc at the base of the tooth, its shape depending somewhat on the cutter and varying from $\frac{1}{4}$ to $\frac{1}{5}$ the depth of the spline.

By putting a clearance lug on the hob, as at *B*, the straight portion of the tooth is increased. This also makes it easier to grind either the sides or the bottom of the spline if desired.

This method works well on splines from four to six keys, but with more than that number the clearance grooves cut out practically all the bearing surface at the root diameter. Other

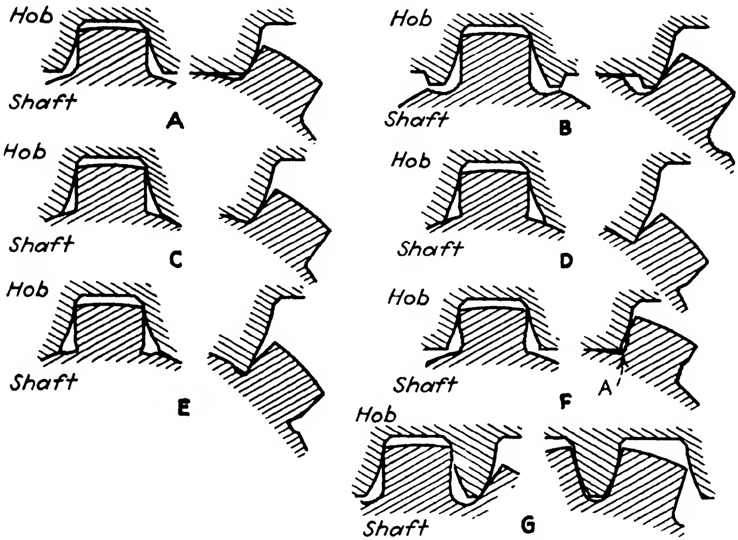


Fig. 46.—Some of the splines that can be hobbed.

forms of hobs and the splines they produce are shown in the other illustrations.

Standard hob sizes for splined shafts are shown in Table 25, while in Table 26 are given the limits that can be secured on both ground and unground hobs.

Table 25.—Barber-Colman Standard Hob Sizes for Spline-shaft Hobs

Number of keys	Depth of spline, in.	Hob size		
		Diameter, in.	Length, in.	Hole, in.
3 and 4	Up to 0.125	3	3	1¼
3 and 4	0.126 to 0.1562	3¼	3¼	1¼
3 and 4	0.157 to 0.1875	3½	3½	1¼
3 and 4	0.188 to 0.250	4	4	1¼
6, 8, 10, etc.	Up to 0.125	2¾	2¾	1¼
6, 8, 10, etc.	0.126 to 0.1562	3	3	1¼
6, 8, 10, etc.	0.157 to 0.1875	3¼	3¼	1¼
6, 8, 10, etc.	0.188 to 0.250	3½	3½	1¼

Table 26.—Spline-shaft Hob Limits

Single-thread unground hobs

Outside diameter of spline shaft, in.	3 and 4 keys		6 keys and more		10 keys and more with round bottom	
	Width, in.	Root diam., in.	Width in.	Root diam., in.	Width, in.	Root diameter
Under 1½	0.002	0.004	0 002	0 002		
1½ to 2½	0 003	0 005	0.002	0.004		
2½ and over	0.003	0.006	0 003	0 005		
2 and less	...	.....	....	...	0 002	Approx. half circle
Over 2	...	.....	...	..	0.003	Approx. half circle

Double-thread unground finishing hobs are not recommended.

Single-thread ground hobs

Outside diameter of spline shaft, in.	3 and 4 keys		6 keys and more		10 keys and more with round bottom	
	Width, in.	Root diam., in.	Width, in.	Root diam., in.	Width, in.	Root diameter
Under 1½	0 001	0 002				
1½ and over	0.002	0 003				
Under 2½	....	....	0 001	0.001		
2½ and over	....	....	0 002	0 002		
2 or under	.....	....	....	....	0.001	Approx. half circle
Over 2	..	....	...	.....	0 0015	Approx. half circle

Double-thread ground hobs

Outside diameter of spline shaft, in.	6 and 10 keys		10 keys and more with round bottom		Variation from key to key regardless of number of keys, in.
	Width, in.	Root diam., in.	Width, in.	Root diameter	
2 and under	0.002	0.002	0.0015	Approx. half circle	0.0005
Over 2	0.002	0.004	0.002	Approx. half circle	0.0007

Double-thread ground hobs are not recommended for four keys or less.

In some instances when the diameter of the shaft to be hobbed is small and the cut is rather light, it is desirable to use a smaller hob on a lighter machine.

Hob sizes as shown in Table 25 are for either ground or unground hobs.

Hobbing Taper Splines.—A new development in splined shafts for all purposes, except where the part slide on the shaft

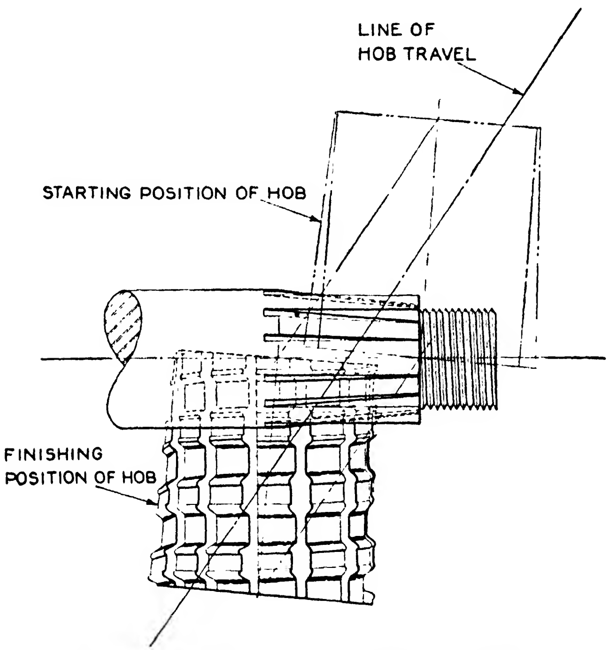


Fig. 47.—Hobbing tapered splines on a shaft.

has been developed by the Barber-Colman Co. and is a taper spline, as seen in the illustrations at the top of Tables 27 and 28. This type of spline does not weaken the shaft or hub of the mating part as much as the straight spline and has the advantage of a taper fit that centers the work and holds it against side movement at the same time. The appearance of the spline is seen in connection with the tables that give dimensions for both four- and six-spline shafts.

These splines are milled with a special hob as shown in Fig. 47. The hob has a peculiar travel as it cuts these splines, starting in the position shown by the dotted lines and moving in the angular path shown as "line of hob travel." It finishes its work in the position shown by the hob itself. This method seems to have advantages over the usual taper fit with its usual key.

An interesting feature of this method is that the mating hub is simply bored to the taper of the bottom of the spline and finished with a straight broach.

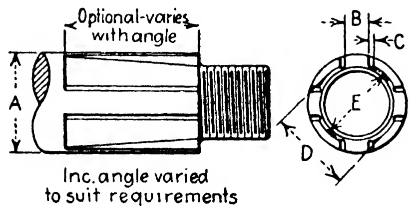


TABLE 27.—FOUR-KEY SPLINE DIMENSIONS

A shaft diam. in.	*B* (nom.) in.	*C* (approx.) in.	*D*, in.	*E* (approx.) in.	Hob No.
½	0.120	0.040	0.414	0.396	
⅝	0.150	0.050	0.539	0.524	
¾	0.180	0.050	0.633	0.610	
⅞	0.210	0.062	0.734	0.696	
1	0.240	0.075	0.828	0.782	

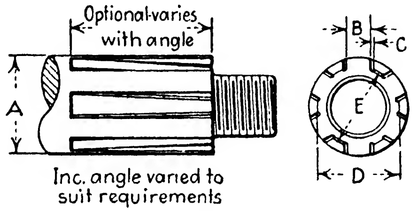
Standardized Hobbing.—Carefully developed instructions for hobbing and inspection are a profitable substitute for the confusion of everyone working to his own ideas, says L. P. Sittig.

A corporation, whose product includes some 120 different kinds of small gears, found that considerable difficulty was experienced in obtaining a uniform product. Volume of work was sufficient to keep a dozen machines busy. Gears of the same kind of material and the same pitch were not alike in running qualities, as judged by the final inspection department. Gears of the same pitch and same diameter showed great variation in the smoothness of teeth; some gears were smooth on one side of the tooth and rough on the other side. A wide variation was also shown in piecework rates.

These factors were investigated and traced to the following causes:

a. Whenever setting up a new job the operator had no instructions in advance, but arbitrarily determined upon a certain feed and then requested the tool-designing department to figure the index gears and feed gears in accordance with that feed. The cutting speed was left to the operator's judgment. If the feed

TABLE 28.—SIX-KEY SPLINE DIMENSIONS



A Shaft diam., in.	*B* (nom.) in.	*C* (approx.) in.	*D*, in.	*E* (approx.) in.	Hob No.
1	1/4	0.075	0.828	0.785	
1 1/8	9/32	0.075	0.953	0.923	
1 1/4	5/16	0.090	1.078	1.041	
1 3/8	11/32	0.090	1.141	1.086	
1 1/2	3/8	0.090	1.266	1.224	
1 5/8	13/32	0.090	1.359	1.320	
1 3/4	7/16	0.100	1.484	1.448	
1 7/8	15/32	0.100	1.578	1.523	
2	1/2	0.100	1.641	1.559	
2 1/4	9/16	0.100	1.891	1.824	
2 1/2	5/8	0.100	2.078	1.996	
2 3/4	11/16	0.110	2.328	2.522	
3	3/4	0.110	2.516	2.425	
3 1/4	13/16	0.125	2.734	2.640	
3 1/2	7/8	0.125	2.953	2.788	
4	1	0.125	3.328	3.196	

seemed too coarse, it was arbitrarily decreased and another combination of gears figured.

b. Because the index gears and feed gears were figured independently for each job, and no comparisons were made with a standard, it was impossible to produce uniformly smooth teeth.

c. The inspection department had no standard by which to judge the smoothness of tooth flanks and the running qualities of the gears, but used an arbitrary judgment in each case. If the chief inspector decided that the cut was too rough, a finer feed had to be chosen and another set of change gears figured for the hobbing machine. Since the gears had to be very accurately cut, it generally took from one-half to three quarters of an hour to figure a set of gears with an error allowance sufficiently small.

To overcome these objections and achieve a degree of standardization that would insure a uniform product, a program such as follows was outlined:

1. The determination of standard feeds based upon the average of past practice.

TABLE 29.—FEEDS FOR HOBBING SCREW STEEL

Pitch[1]	19	20	22	24	26	28	30	32	34	40	48
Spur gears	0.042	0.040	0.039	0.038	0.037	0.036	0.035	0.034	0.033	0.030	0.026
Spiral gears, deg.											
5–10	0.0418	0.0398	0.0388	0.0378	0.0368	0.0358	0.0348	0.0338	0.0328	0.0298	0.0259
10–15	0.0413	0.0394	0.0384	0.0374	0.0364	0.0354	0.0344	0.0334	0.0324	0.0294	0.0256
15–20	0.0405	0.0386	0.0376	0.0367	0.0357	0.0347	0.0338	0.0328	0.0318	0.0289	0.0251
20–25	0.0394	0.0376	0.0366	0.0356	0.0347	0.0338	0.0328	0.0319	0.031	0.0281	0.0244
25–30	0.038	0.0362	0.0353	0.0344	0.0335	0.0327	0.0317	0.0308	0.0299	0.0271	0.0235
30–35	0.0363	0.0346	0.0337	0.0329	0.032	0.0311	0.0303	0.0294	0.0285	0.0259	0.0225
35–40	0.0344	0.0328	0.0319	0.0311	0.0303	0.0294	0.0286	0.0278	0.027	0.0245	0.0212
40–45	0.0321	0.0306	0.0298	0.0291	0.0283	0.0275	0.0268	0.026	0.0252	0.0229	0.0199
45–50	0.0296	0.0283	0.0275	0.0268	0.0261	0.0254	0.0247	0.024	0.0233	0.0212	0.0183
50–55	0.0269	0.0257	0.025	0.0244	0.0237	0.0231	0.0224	0.0218	0.0212	0.0192	0.0167
55–60	0.024	0.0229	0.0223	0.0217	0.0212	0.0206	0.020	0.0195	0.0189	0.0172	0.0149
60–65	0.021	0.020	0.0195	0.019	0.0185	0.018	0.0175	0.017	0.0165	0.015	0.013
65–70	0.0177	0.0169	0.0164	0.016	0.0156	0.0152	0.0147	0.0143	0.0139	0.0126	0.0109
70–75	0.0143	0.0137	0.0133	0.0129	0.0126	0.0123	0.0119	0.0116	0.0112	0.0102	0.0089
75–80	0.0108	0.0103	0.010	0.0098	0.0095	0.0093	0.0089	0.0087	0.0085	0.0077	0.0067
80–85	0.0072	0.0069	0.0067	0.0065	0.0064	0.0062	0.006	0.0059	0.0057	0.0052	0.0045
85–90	0.0036	0.0035	0.0033	0.0033	0.0032	0.0031	0.003	0.0029	0.0028	0.0026	0.0022

Feeds for bronze = 1.5 × above.

2. The figuring of change gears based upon the standard feeds.
3. The establishment of operation instructions to show the hobbing machine setup and the change gears to be used.
4. The issuing of inspection instructions specifying the permissible width of tooth marks on the tooth flanks of gears.

In line with this program of standardization, Tables 29 and 30 were compiled to show the feeds to be used for gears of steel or of bakelite. Very few bronze gears were used and a satisfactory standard for these was established by specifying a feed $1\frac{1}{2}$ times that for steel. The tables show the feed per revolution of work for hobbing steel gears and gears of linen-base bakelite. The feeds for various angles of helix were computed by the formula: "feed per revolution of cutter × sin helix angle = feed per revolution of work."

The use of these tables insured that all gears of the same pitch would have the same spacing of revolution marks on the teeth regardless of their helix angle. Of course, the depth-of-revolution marks were kept to a minimum by checking the eccentricity of the hob on its arbor, and limiting this to 0.0005 in. The hob teeth were machine ground to retain accurate spacing of the cutting edges.

Since all of the hobbing machines in this plant were No. 3 Barber-Colman machines, the change gears had to be figured according to formulas applicable to this type of machine.

For cutting spur gears and ratchets, these formulas are as follows:

N = number of teeth to be cut

for index gears,

$$\frac{12}{N} = \frac{\text{drivers}}{\text{driven}}$$

for feed gears,

$$\frac{\text{feed}}{0.075} = \frac{\text{drivers}}{\text{driven}}$$

For speed gears, Table 31 was used, this being applicable to helical and worm gears also. For cutting worm gears, these formulas were required:

B = number of threads in worm or hob.

C = number of teeth in worm gear.

D = feed per revolution of hob, assumed as follows:

19 pitch —0.011 feed per revolution of hob
20 pitch —0.010 feed per revolution of hob
22 pitch —0.009 feed per revolution of hob
26 pitch —0.008 feed per revolution of hob
28 pitch —0.007 feed per revolution of hob

for index gears,

$$\frac{12 \times B}{N} = \frac{\text{drivers}}{\text{driven}}$$

for vertical feed gears,

$$\frac{D \times C}{0.1875 \times B} = \frac{\text{drivers}}{\text{driven}}$$

Finding gears to correspond with the decimal ratios is best accomplished by the method of continued fractions. To avoid figuring gears that would be too large, the ratios obtained were checked with the following requirements admissible by the gear centers on the machine:

Largest driver = 80.

Largest driven = 76.

Largest sum of driver and driven = 116.

The above data, and those which follow, were assembled in blueprint form and bound in a properly labeled folder so that it would be readily available for whoever was assigned to the duty of writing operation instructions. As will be noticed, the information in this data book is clear enough so that even an inexperienced engineer can figure a set of change gears for a new job.

Index Gears for Helical Gears.—Helical gears require a more extensive calculation than other types, and the following formulas are required: For index gears,

$$\frac{\text{Feed} \times \text{sin helix angle}}{\text{circular pitch}} = \text{fraction } A.$$

Fraction A may be most conveniently chosen from the table of fractional equivalents in the "American Machinists Handbook." Then we have the formula:

$$\frac{12}{\text{No. of teeth} - A} = \frac{\text{drivers}}{\text{driven}}.$$

The gears may be factored from a table which shows the gears available on the machine. If the ratio will not factor, the fraction A must be changed slightly.

It is always advisable to have the direction of helix of the hob the same as that of the gear to be cut. Such a condition produces the smoothest finish and avoids any tendency to harm the cutting edges of the hob. If the direction of angle is not the

TABLE 30.—FEEDS FOR HOBBING BAKELITE

Pitch	19	20	21	22	24	26	28	30
Spur gears	0.084	0.080	0.079	0.078	0.076	0.074	0.072	0.070
Spiral gears, deg.								
5 to 10	0.0836	0.0796	0.0786	0.0777	0.0757	0.0737	0.0717	0.0697
10 to 15	0.0826	0.0788	0.0777	0.0768	0.0748	0.0728	0.0708	0.0689
15 to 20	0.081	0.0772	0.0763	0.0752	0.0734	0.0714	0.0694	0.0676
20 to 25	0.0788	0.0752	0.0742	0.0732	0.0712	0.0694	0.0676	0.0657
25 to 30	0.076	0.0724	0.0715	0.0706	0.0688	0.067	0.0654	0.0634
30 to 35	0.0726	0.0692	0.0684	0.0674	0.0658	0.064	0.0622	0.0606
35 to 40	0.0688	0.0656	0.0647	0.0638	0.0622	0.0606	0.0588	0.0573
40 to 45	0.0642	0.0612	0.0605	0.0596	0.0582	0.0566	0.055	0.0536
45 to 50	0.0592	0.0566	0.0558	0.055	0.0536	0.0522	0.0508	0.0494
50 to 55	0.0538	0.0514	0.0507	0.050	0.0488	0.0474	0.0462	0.0449
55 to 60	0.048	0.0458	0.0453	0.0446	0.0434	0.0424	0.0412	0.0401
60 to 65	0.042	0.040	0.0395	0.039	0.038	0.037	0.036	0.035
65 to 70	0.0354	0.0338	0.0333	0.0328	0.032	0.0312	0.0304	0.0295
70 to 75	0.0286	0.0274	0.027	0.0266	0.0258	0.0252	0.0246	0.0239
75 to 80	0.0216	0.0206	0.0204	0.020	0.0196	0.019	0.0186	0.0181
80 to 85	0.0144	0.0138	0.0137	0.0134	0.013	0.0128	0.0124	0.0121
85 to 90	0.0072	0.0070	0.0068	0.0067	0.0066	0.0064	0.0062	0.0061

TABLE 31.—SPEEDS FOR HOBBING—NO. 3 BARBER-COLMAN MACHINE

Diam. hob	Screw steel	Bakelite and bronze
½	18/36	
¾	18/36	
1	20/34	
1¼	22/32	18/36
1⅞	25/29	20/34
2½	29/25	22/32

Gears shown are as in position on machine, driven above—driver below.

same, and the angle is more than 45 deg., the hob has a tendency to push the work ahead of it, in a manner similar to that of a milling job where the work is fed in the same direction as the cutter teeth. When the stock of hobs on hand calls for the emergency use of a hob whose direction of angle is opposite to that of the work, it is necessary to figure the index gears with the formula:

$$\frac{12}{\text{No. of teeth} + A} = \frac{\text{drivers}}{\text{driven}}.$$

Feed Gears for Helical Gears.—Since the index train and the feed-gear train must work in unison to generate a helical path, it is necessary to use great care in figuring the feed gears. The gears available on the machine will seldom furnish a ratio exactly equal to the theoretical feed. For the product required in the plant here mentioned, an error allowance was worked out for various helix angles, as shown in Table 32. This allowance represents an error of 6 sec. of angle or 0.1 min., which is close enough for the most rigid accuracy requirements in the pitches under consideration. To conform to this standard of accuracy, it is necessary to figure the theoretical feed to six places of decimals, using the formulas:

$$A \times \text{cir. pitch} \times \text{cosec angle} = \text{theoretical feed}.$$

$$\text{feed gears} = \frac{\text{theoretical feed}}{0.075} = \frac{\text{drivers}}{\text{driven}}.$$

Setup Instructions for Hobbing.—Having figured the change gears for a hobbing operation according to standardized conditions, a record of these gears and the setup instructions for the job should be placed on a standard-operation instruction print. The setup diagram would look something like Fig. 48. This allows directions for the use of either hob to be given by crossing out the line not wanted.

If the direction of helix on hob and gear are the same, the angle of swivel on the hobbing machine will be equal to the helix angle minus the thread angle stamped on the hob. When the direction of helix on hob and gear are different, the angle of swivel is equal to the helix angle plus the thread angle.

The standard swivel on the No. 3 Barber-Colman hobbing machine allows of using a 1⅞-in.-diameter hob, which should be specified for all helix angles less than 45 deg. For helical gears of steeper angle, a 2½-in. hob should be used on a special swivel provided with a short arbor and a flywheel on the driving end.

To facilitate setting up, the instructions may include a note based on the data below, and for left-hand hobs, an idler gear must be specified in both the index gear train and the feed gear

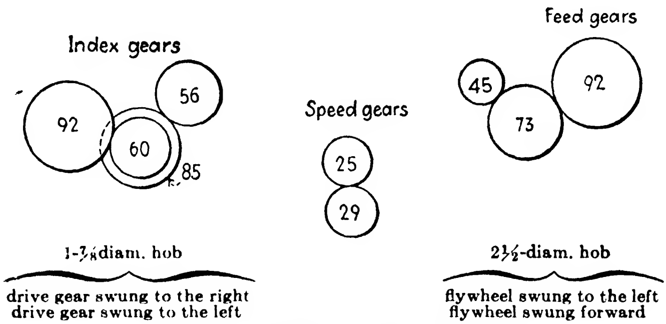


FIG. 48.—Outline of a setup diagram.

train, this to be placed between the first driver and the intermediate gear.

By using operation-instruction prints embodying the data shown, and also specifying a motion study for loading and removing the work, a consistent and equitable piece rate was established for each part.

TABLE 32.—ALLOWABLE ERROR IN HELICAL GEAR ANGLES

Angle, deg.	Error allowed
10	0.0009
20	0.0002
30	0.0001
40	0.00005
50	0.00003
60	0.00002
70	0.00001
80	0.000005

The difficulties met with in inspection were removed by an inspection instruction including such specifications as these:

"Check outside diam. and depth of tooth, 5 per cent.

"Check thickness of tooth at pitch line and helix angle, 20 per cent.

"See that revolution marks on flanks of teeth from hobbing are not longer than 0.034 in. 5 per cent."

The length of revolution mark permissible was specified the same as the feed per revolution for a spur gear of equal pitch.

This experience of eliminating manufacturing difficulties by standardization is indicative of what may be accomplished in many other operations. Briefly, it demands careful analysis and the application of mathematical principles wherever possible.

DIFFERENTIAL AND NONDIFFERENTIAL METHODS OF GEAR HOBBING[1]

Helical gears for parallel axes, helical gears for crossed axes, and worm gears can be hobbed by either the differential or nondifferential methods. Each of these methods has its own peculiar advantages and effects.

Advantages of the Differential Method—Parallel Helical Gears.—The important advantage of the differential method is that disconnecting the feed does not destroy the lead relation between the hob and the blank. In general, a differential is advantageous when it is necessary to take two or more cuts on a helical gear on the same machine. The use of a differential automatically ensures accurate realignment of the hob for the second cut.

The differential method is particularly convenient when a very light recut must be taken to obtain the exact tooth thickness or exact backlash desired. For instance, if a cut has been started and the tooth thickness is found to be a thousandth or two too great, the hob can be returned to its initial position, moved toward the blank the necessary amount, and a recut taken with perfect assurance that it will clean up on both sides of the teeth.

When the differential is used, the feed may be changed at will without disturbing the correlation of the blank and cutter. Therefore, if after starting a roughing cut it is desired to increase or decrease the feed, this may readily be accomplished without losing the lead relation or resorting to gearing calculations. Likewise, when more than one gear is to be cut, the feed may be increased or decreased before cutting the next blank without making gearing calculations. The feed gears are selected from a chart.

In one respect, the original calculations for cutting helical gears of a given number of teeth to an *exact* helix angle, either differentially or nondifferentially, involve about the same amount

[1] Granger Davenport, Research Engineer, Gould & Eberhardt.

of work; the same time is required to calculate the differential lead gears as the nondifferential feed gears. However, the nondifferential method involves additional calculations for the index gears, and under certain conditions involving high *C* constants, the total calculating time may be double that needed for the differential method. See page 102*f* for explanation of *C* constants referred to.

The location of the differential in the gearing train of the machine affects the calculations. It is preferably located in the indexing train in *advance* of the index change gears, to eliminate the number of teeth factor in the lead formula as required when located *after* the index change gears. With the differential in the preferred location, the lead change gears are therefore based upon the axial pitch (sometimes termed the linear pitch or one-tooth lead) rather than upon the lead. In this case, when differentially cutting a series of gears having various numbers of teeth, but of the same pitch and helix angle, the identical lead gears can be used for all. The only requirement is that the index gears be changed for each different number of teeth to be cut; these index gears are selected from the chart, requiring no gearing calculations. Therefore, in any plant where large numbers of gears of the same pitch and helix angle are to be cut, a differential greatly simplifies gearing calculations.

The various effects of placing the differential in other locations will not be discussed at length. It may be located in the cutter drive, after the cutter speed change gears. In some cases two differentials are used, one in the index train and the other in the cutter drive, to permit hobbing prime numbers of teeth without the necessity for corresponding prime change gears.

For some classes of work, all tedious calculations can be dispensed with when using the differential method. For instance, if only one gear and one pinion are to be cut, both on the same machine, the lead calculations can be very *approximate*, for a departure from theoretical helix angle within wide limits is not harmful provided the same departure exists in the mating gear. As an example, if the theoretical helix angle is 30°, the actual angle from gearing could be 30°1′ without impairing the true involute action of the gear in any way, so long as the mating gear has precisely the same angle. The lead ratio need therefore be calculated only to three decimal places. Then by reference to a table of fractional equivalents of decimals, the nearest suitable fraction can be selected directly from the table and readily expanded to figures suitable for change gearing. This eliminates the more extensive calculations necessary for cutting gears nondifferentially or when it is essential that the lead angle be exact. The advantage of a differential for this type of work is obvious, particularly for jobbing shops where a large variety of gears in small lots is produced, and where interchangeability with gears cut on machines having other lead formulas is not a factor.

The differential method greatly simplifies the setup for hobbing double helical gears when the teeth are to intersect exactly at the theoretical apex or are to be staggered. Before commencing the cut, the hob is traversed over the surface of the blank until the hob teeth are aligned with punch marks or scribed lines laid out with proper relation to the apex. Subsequent facing operations to align the pinion and gear faces are thereby avoided.

Likewise, a differential is an advantage when a tooth or space must bear a definite relation to a keyway or some other point on the gear or shaft.

When badly worn gears are to be recut, the differential feature simplifies the positioning of the hob relative to the worn portions of the teeth. By traversing the hob up and down along the worn teeth, it can be readily seen whether the hob is in the correct relation before commencing the cut, thereby minimizing the amount of stock to be removed during the recutting.

A further advantage of the differential method is that the hob can be rotating and sunk into depth before engaging the feed. This is useful where space is limited as, for instance, in the narrow groove at the apex of a double helical gear.

Limitation of the Differential Method—Parallel Helical Gears. The differential method is limited to a specified range of helix angles, pitches, and numbers of teeth, unless additional mechanism is added to cover a greater range. The lead formula and the index formula must be proportioned to obtain a selected average range of helix angles, pitches, and numbers of teeth, according to the range of work to be cut. A differential machine so proportioned for an average range is unsuited for extremities of a total composite range of:

1. **High helix angles, fine pitches, and low numbers of teeth; and the opposite extreme of:**

2. Low helix angles, coarse pitches, and high numbers of teeth.

When hobbing gears near the first of the extreme conditions mentioned, the stresses in certain parts of the machine are very severe. When the driving change gears are large and the driven gears are small, the stresses may be excessive and cause breakage. With the cutter rotating, movement of the cutter slide will be difficult in one direction and unusually easy in the other direction. Stopping the rotation of the cutter will equalize the movement of the cutter slide in either direction.

The excessive stresses will occur in either feeding or returning but not both, depending upon the type of the differential mechanism. A differential can be arranged so that the hob slide feeds "hard" and returns "easy," or vice versa. In the latter case, however, with the first of the extreme conditions mentioned above the differential action might be such as to cause the slide to move down ahead of the feed and cause damage to the hob and work.

For the second set of extreme conditions mentioned above, the difficulty lies in selecting suitable change gears to make the high reduction ratio of gearing required for the lead.

For those comparatively few cases where it is necessary to go beyond the working range selected, the differential may be disconnected and the gear cut nondifferentially.

The range of differential cutting may be widened by arranging the machine with a dual series of gears in either the index or lead trains, or both, with means for connecting or disconnecting either series as required.

It is also possible to cut a gear nondifferentially and return to the initial hob position through the differential, thereby maintaining the lead relation when it is desired to take a recut. To do this the differential must be of the disconnectable type.

Advantages of the Nondifferential Method—Parallel Helical Gears.—A hobbing machine not equipped with a differential will produce helical gears to the same degree of accuracy and at the same actual cutting rate as a machine equipped with a differential.

Moreover, a nondifferential machine has at least 17 fewer gears in its train, resulting in a simpler mechanism. The motion throughout is a direct motion; no other motion is injected by way of a plus or minus increment to affect the rotary movement of either the blank or cutter, as in the differential method. Likewise, when cutting nondifferentially on a differential machine the motion is direct, provided the differential is of the disconnectable type. Regardless of the more complex mechanism of differential machines, it has been demonstrated that on modern hobbing machines no appreciable difference exists in the accuracy of product between the differential and nondifferential methods, although at one time this was not true.

One very important advantage of a nondifferential machine is its substantially lower cost, resulting from the fewer gears and parts.

Another advantage is the fewer change gears required in setting up the machine. In a differential machine three sets of change gears are needed; namely, index, feed, and lead, whereas in a nondifferential machine only two sets are required—index and feed.

For gears cut on a mass-production basis, the nondifferential method is common practice. For this class of work, new setups requiring trial cuts are infrequent, and gearing calculations are a negligible factor. When second cuts are made, ample stock for finishing is allowed. These second cuts are usually taken on another machine, or at least on a different setup, where a differential would be of no benefit. The lower cost of a nondifferential machine then becomes an important consideration, especially if a battery of machines is being installed.

A second or finishing cut can be taken nondifferentially if there is a reasonable amount of stock to be removed. The alignment for the second cut is greatly facilitated by the use of a resetting clutch. The latter is used to disconnect the feed temporarily and cause the rotating cutter to change its relation with respect to the rotating blank. When the cutter is seen to track exactly in the preceding cut, the resetting clutch is reengaged and the cut commences.

When nondifferentially cutting a series of gears having various numbers of teeth, but of same pitch and helix angle, the same C constant can occasionally be used for several of the gears, thereby simplifying calculations somewhat. However, this is a matter of chance, and usually the calculating time exceeds that of the differential method.

A particular advantage of the nondifferential method in some instances is the cutting of prime numbers of teeth without the necessity of having corresponding prime change gears. In the case of cutting prime gears of 100 teeth or fewer, this is no advantage, but when the primes reach very high numbers, as they sometimes do in turbine reduction work, this is a useful expedient. Here again the cut may be made nondifferentially, and the return for recut made through a differential mechanism, as mentioned previously.

Crossed Helical Gears.—In the preceding paragraphs, the term "parallel helical gears" has been used in the commonly restricted sense as applying only to helical gears for connecting parallel shafts.

The term "crossed helical gears" is here used to distinguish the less common application of helical gears for connecting nonparallel nonintersecting shafts.

In hobbing crossed helical gears, both the differential and nondifferential methods are applicable in the same manner as outlined for parallel helical gears. The only difference is that in this case *approximate* calculations for the lead or feed change gears are always satisfactory, because reasonable errors in helix angles of crossed helical gears, either on pinion or gear or both, will not affect the conjugate action when the pair is meshed together.

Worm Gears.—Conventional worm gears are hobbed by either the infeed or tangential feed methods. In one instance a differential is never used, and in the other it is invariably used.

Infeeding is usually employed when the ratio between the worm and gear is high, and the angle of the worm threads does not exceed about 6 deg. In this process the hob is gradually fed in radially toward the center of the blank, and when proper depth has been reached and the work rotated at least one extra revolution, the worm gear is complete. No differential is used, and no calculations are required, as the index and feed gearing are merely selected from a chart. The feed may be changed at will without disturbing the correlation of cutter and work.

Tangential feeding is usually employed when the ratio between worm and gear is low and when the worm is multiple threaded, because there are so few formative cutting teeth within the generating zone that infeeding would not produce a satisfactory finish. Furthermore, when the angle of the worm thread exceeds about 6 deg., there is an interference between a hob being infed and the flanks of the worm gear teeth, requiring that a subsequent finishing cut be taken tangentially with a hob having thicker teeth. In the tangential process the hob or fly tool, as the case may be, is always at the correct center distance and feeds in a tangential direction with respect to the worm-gear blank, preferably in a direction opposed to the rotation of the blank. As the cutter travels across the blank, the rotation of the blank is given a plus or minus increment in order to maintain the proper correlation with the cutter at all times. This is almost invariably accomplished by means of a differential mechanism. The gearing for index and feed are selected from a chart, but the lead gearing must be calculated and the calculations carried out to a high degree of accuracy. The feed may be disconnected or changed at will without interfering with the correlation of cutter and work.

It is possible to hob a worm gear tangentially without the use of a differential, obtaining the plus or minus increment of the blank rotation by suitably correlating the index gearing with the feed gearing in much the same manner as when nondifferentially hobbing helical gears. However, the disadvantages in not being able to disconnect the feed without losing the lead relation, or to change the feed without gearing calculations, are such that this method is used very infrequently in this country, if at all. The method is employed abroad to some extent.

The infeed method is much faster than the tangential, hence is preferable provided the ratio between the worm and gear, and the angle of the worm thread, are such that the resulting accuracy and finish will be satisfactory.

Frequently a combination of infeeding and tangential feeding is employed. The hob removes most of the stock while it is infed to correct centers, using the thinner roughing teeth, and is then tangentially fed a comparatively short distance until its finishing teeth have completed the cut. By this means, the total time as compared with straight tangential hobbing is greatly reduced.

USING THE *C* CONSTANT

To illustrate the use of the *C* constant referred to, a typical set of formulas for hobbing helical gears nondifferentially is as follows:

Let C = feed constant.
F = feed per revolution of blank.
N = number of teeth to be cut.
P^L = axial or linear pitch.
P^{NC} = normal circular pitch.
T = number of threads in hob.
α = helix angle with axis.

Then

$$C = \frac{P^{NC}}{F \sin \alpha}, \quad \text{or} \quad \frac{P^L}{F}$$

The value of C is calculated only to the nearest whole number.

Index change gears:

$$\frac{20\ TC}{NC - 1} = \frac{\text{driver}}{\text{driven}}$$

When the hob and blank are of opposite hand, the denominator in the index formula becomes $NC + 1$.

If the value of C corresponding to the feed selected results in prime numbers for which no change gear is available, the value of C is slightly changed. This has no appreciable effect on the feed, and will not affect the lead so long as the same value of C is used in both index and feed formulas.

Feed change gears:

$$\frac{8P^{NC}}{C \sin \alpha} = \frac{\text{driver}}{\text{driven}}$$

CHAPTER III

THE SHAPING METHOD OF CUTTING GEARS

Fellows Gear Shaper.—The method of forming gear teeth by shaping was originated by E. R. Fellows about 40 years ago and was a radical departure from the use of rotary cutters. This method uses a cutter which is practically a hardened gear with properly relieved teeth. Both the cutter and the gear blank are held on parallel members. The cutter reciprocates with the shaper ram parallel to the work spindle holding the gear blank.

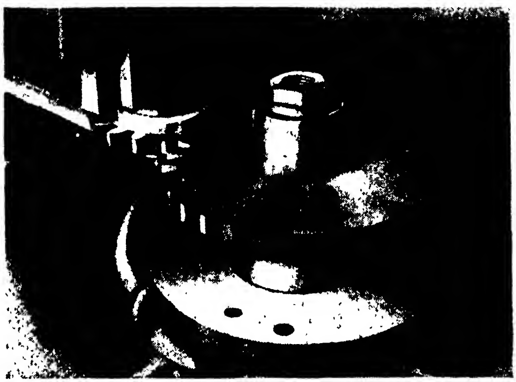


FIG. 49.—The Fellows gear-shaping method.

In starting a gear the cutter is first fed into the blank to the proper depth. Then both cutter and gear blank start to revolve together, as though they were a pair of gears. As they revolve, the cutter generates teeth on the edge of the blank and, when the revolution is completed, all the teeth are cut. The action of the cutter in the work is seen in Fig. 49. The machine itself is shown in Fig. 50.

One advantage of this method over the use of formed cutters is that one cutter of any pitch will cut any number of teeth of that pitch, while with the formed cutter a single cutter is used for only a limited number of teeth.

The cutter ram moves away from the gear blank on the return stroke so as not to drag the cutter on the upstroke. And when we realize the speed of the later machines, which make over 800 strokes per minute, the rapidity of this relieving motion becomes apparent.

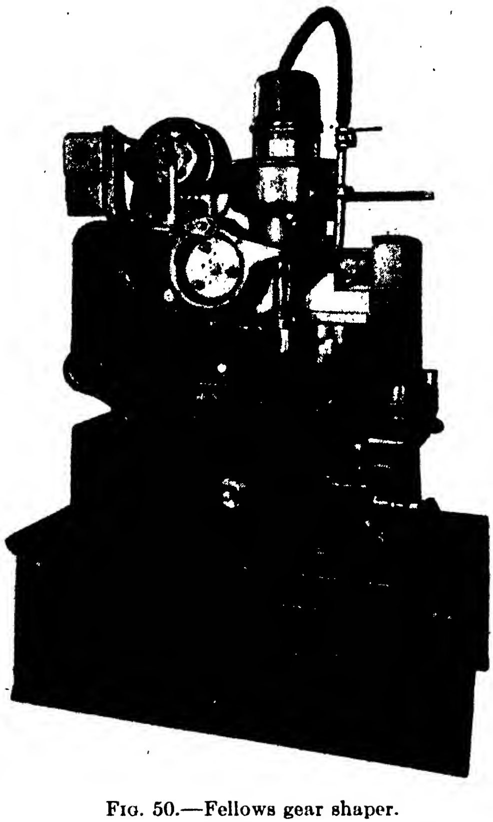


Fig. 50.—Fellows gear shaper.

This type of machine is equally useful in cutting spur or helical gears. In the former case the ram moves past the work in a straight line, while for the helical gears, the ram and blank turn the required amount during the stroke. Suitable guides on the ram give it the proper motion, which is also transmitted to the gear blank. The cutters for machines of this type are easily sharpened and their accuracy is guarded with utmost care.

Suggestions for cutting speeds for different materials are given in the Table 33.

TABLE 33.—CUTTING SPEEDS FOR USE ON FELLOWS GEAR SHAPERS
Based on a maximum cutting stroke of 1 in. and maximum degree of machinability for analysis specified

Material	Max. cutting speed, feet per minute	Remarks
Cast iron	60	Ordinary gray iron castings
Steel (mild)	90	0.15 to 0.20 per cent carbon, no alloys
Steel (high carbon)	65	0.50 per cent carbon, 1.15 per cent manganese, 0.55 per cent chromium
Steel (tool)	70	0.9 to 1 per cent carbon, 25 per cent manganese
Steel (chrome nickel)	50	S.A.E. #3250
Brass (soft)	100	
Bronze (naval)	50	
Aluminum	200	With kerosene as a coolant

NOTE: Reduce cutting speed 10 per cent, for each 1 in. increase in length of stroke. For example: For 5-in length of stroke, cutting speed would be decreased 50 per cent.

Some Details of the Fellows Machine.—The wide use of the Fellows machine makes it advisable to devote space to some of the details of its operation. Full directions for operation details come with the machine but a few suggestions as to mounting the cutter may be advisable. Figure 51 shows the mounting of cutters of several sizes for both the pull and push methods of cutting. Regular cutters are made of 4-in. pitch diameter for gears of six pitch or coarser and 3-in. pitch diameter for seven pitch and finer. For special work, however, in the case of small internal gears, special cutters are necessary as seen at *D* in Fig. 51.

One of the important points is the proper mounting of the cutter on its spindle. The spindle as well as the hole in the cutter must be perfectly clean, the washers must be of the right diameter, and the top washer must be ground to the same angle as the top of the cutter, which is usually 5 deg. The washers should be as large as possible to support the cutter. With the small washer shown at *C* in Fig. 51, an undue strain is imposed on the web of the cutter, tending to crack it even without running the machine. It is very important that only the regular wrench furnished with the machine is used for tightening the

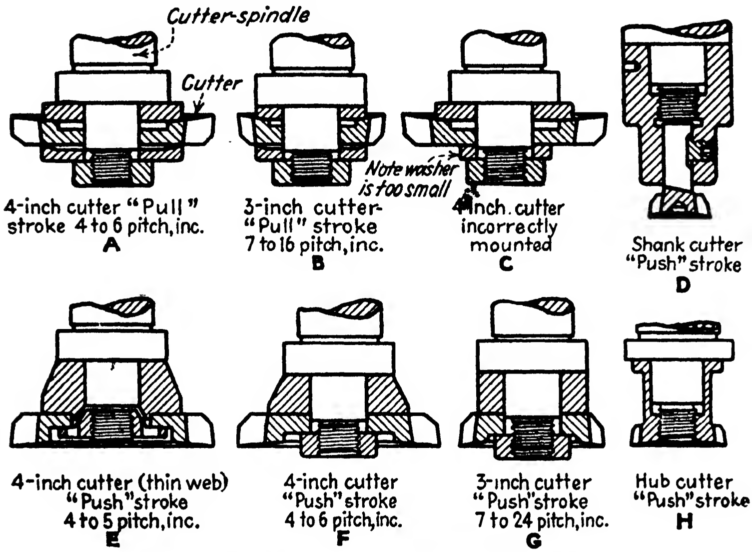


FIG. 51.—Several methods of mounting cutters.

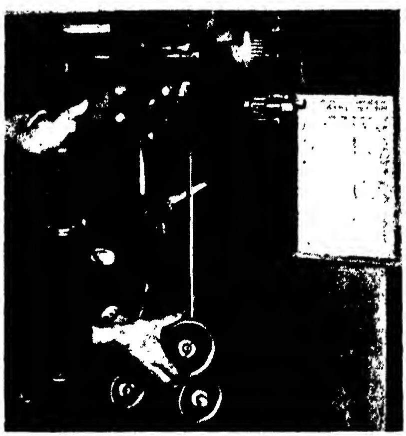


FIG. 52.—Change gears and gear table beside machine.

cutter in place. The nut has a fine thread and no extension wrench should be used.

The selection of the change gears is simple. A complete table is furnished, which should be placed beside the machine as in Fig. 52. This shows the three shafts on which the change gears are placed, in accordance with the corresponding columns in the table shown on the wall. As with all gear-cutting machines it is important that the gear blanks be mounted true on the work

FIG. 53.—Checking concentricity of gear blanks.

FIG. 54.—Centering the cutter with blanks to be cut.

spindle. The concentricity of the blanks can be easily tested with a dial indicator clamped to the work support, as in Fig. 53. The relieving mechanism must be set to suit the cut, whether a pull or a push cut is to be made.

A gage is provided for centering the cutter on the axis or center line of the work spindle. The feed rod can be turned by the crank handle, as in Fig. 54, until the center of one tooth of the cutter is in line with the front edge of the setting gage. When gears of one type are being made in quantity, it is better to set the cutter to a master gear, as in Fig. 55. The master gear is simply placed on the work arbor, but not clamped. The cutter, also unclamped, is then brought into mesh with the master gear and both rotated by hand to insure a close setting. After the cutter is properly adjusted to the master gear, the cutter is tightened on its spindle and the dial set for the pitch to be cut.

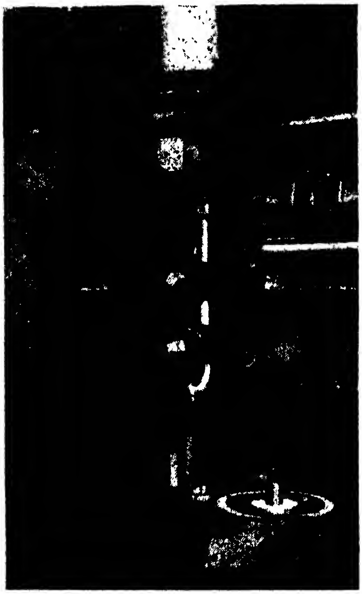


Fig. 55.—Setting cutter to a master gear.

Master gears should be accurately made. The outside diameter and the concentricity should be within 0.001 in. The hole should be square with the face and on face ground. Master gears can be made of cast iron, but for steady use it is better to use steel, hardened and ground. It is not necessary to cut all the teeth; a section of the periphery is all that is necessary.

Graduations are provided for setting the cutter to cut teeth of the proper depth. In cutting stub teeth, such as 6/8 pitch, it must be remembered that, although the pitch of the gear is 6, the depth is that of an 8-pitch gear.

Provision is made so that a gear can be finished in one or two cuts. If two cuts are to be taken, the cutter is set to leave the desired amount for the finish cut, such as 0.010 in. The machine will then take a roughing cut around the gear, move into the proper depth for finishing, and repeat the cycle, stopping when the second cut has been completed.

Holding Gears for Cutting.—A few examples of holding fixtures for gears of different kinds are shown herewith.

Figure 56 is a diagram of the method used in holding a stack of spur gears, while Fig. 57 shows thin clutch rings are held for cutting both external and internal teeth. When gears have long hubs on one side, it is necessary to make holding fixtures

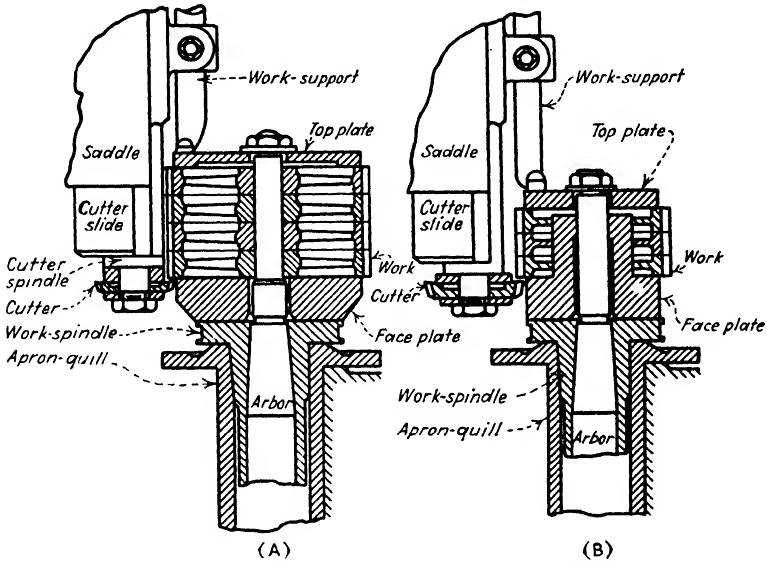


FIG. 56.—Two methods of stacking gears. *A* shows gears held on mandrel while *B* uses a special face plate.

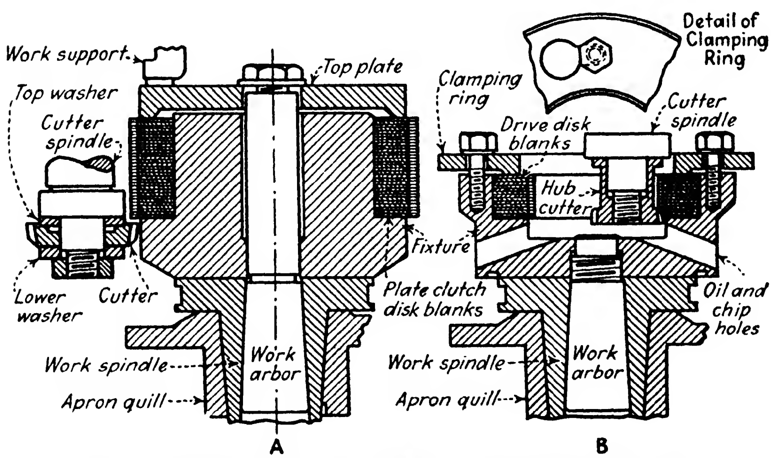


FIG. 57.—Holding outer clutch disks at *A* and inner disks at *B*.

to suit each case, while cluster gears with a hole through them are held on mandrels as in Fig. 58. When cluster gears have end

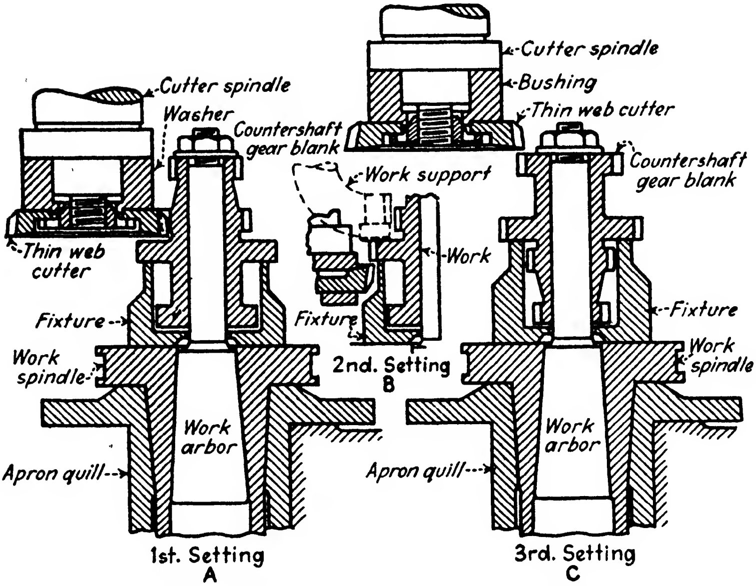


Fig. 58.—Three settings for cutting cluster gears.

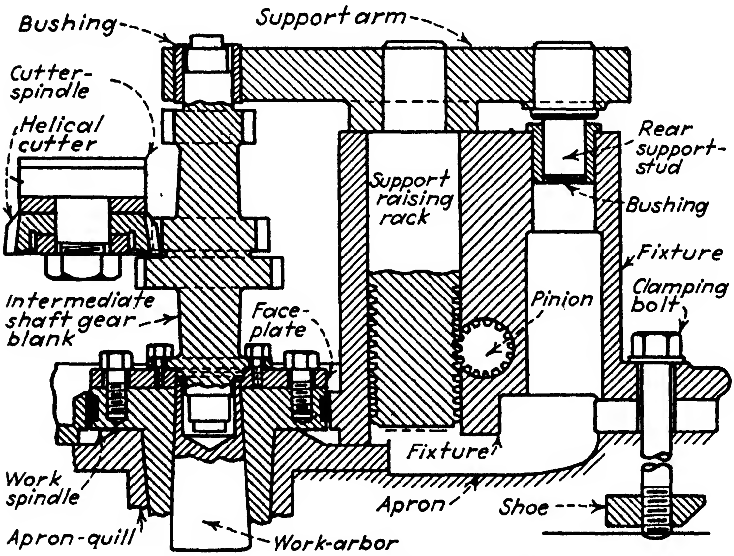


Fig. 59.—Holding cluster gears by end bearings.

bearings, they are held as in Fig. 59, a supporting arm being used for the upper end. This support can take the form of a top center and a bottom center held in the work mandrel beneath,

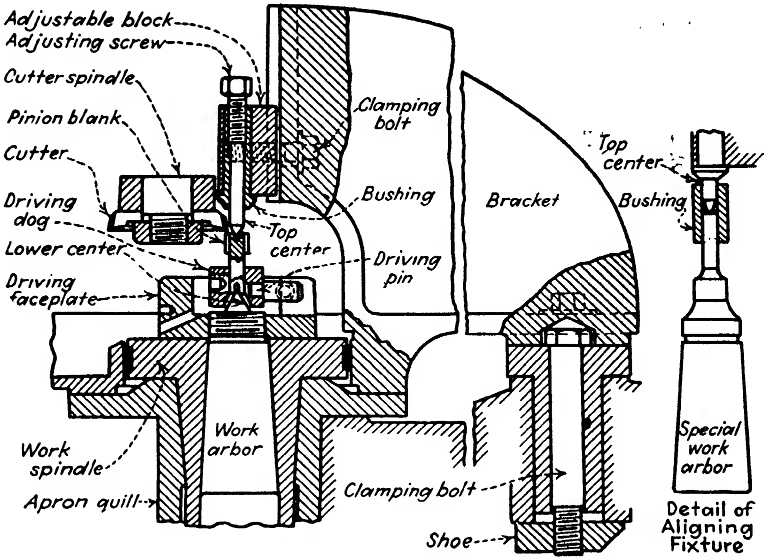


FIG. 60.—Holding work on center.

as in Fig. 60. This cuts the teeth concentric with the centers on which the gear blank was turned.

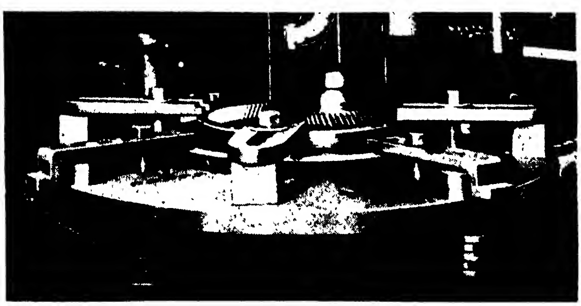


FIG. 61.—A universal faceplate for a variety of work.

There are many other types of fixtures used to suit special work. A universal type of faceplate fixtures for internal work is seen in Fig. 61. This is simply a faceplate attached by screws to the top face of the work spindle. On this faceplate is a centering ring held in place by straps. The holding ring is trued by means of a dial indicator and the work located by means of this ring. Such a fixture covers a large range of work and will be found very convenient in a gear jobbing shop. The

Fig. 62.—Four gears made with the same cutter.

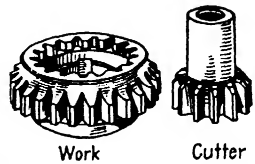


Fig. 63.—An internal clutch and cutter.

holding fixture should be tested every time it is set up to insure accuracy. It is also advisable occasionally to check the truth of the top faces of this plate and to reface it parallel with the lower face, when necessary.

Examples of Gear-shaper Work.—In gear shapers of the Fellows and Sykes types the cutter may be said to resemble the gears being cut except that they are hardened and are ground with a rake angle on the cutting face. In these machines there is a

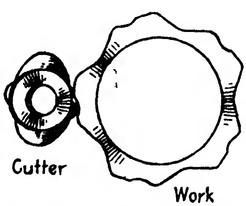


Fig. 64.—Cutter for special engine cam.

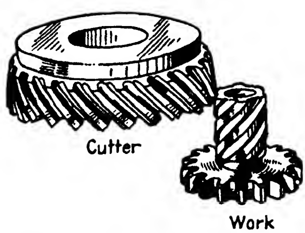


Fig. 65.—Fast helix angle cut on small diameter.

reciprocating motion of the cutter across the face of the gear blank, which gives a planing or shaping action, and at the same time a rotation of both the gear blank and the cutter at a uniform rate, in the same way that two gears roll together. This combined motion cuts a constantly changing path across the gear face and produces a gear tooth of the proper shape. The shape of the gear that is cut is the same as would be produced by rolling a hard gear against a very soft gear blank that would flow up into the teeth of the hard gear as it revolved, and retain its shape. This was what was attempted in the Anderson gear

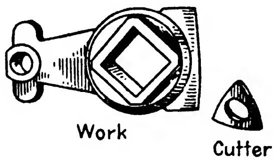


FIG. 66.—Cutting a square hole.

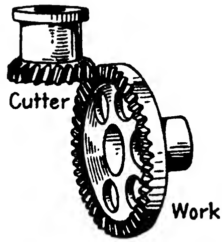


FIG. 67.—Almost a crown gear.

rolling process which gave promise of success but which failed to become commercial. Here the master gears were rolled into a red-hot steel blank and forced the metal in the rim of the blank up into the teeth of the master gears.

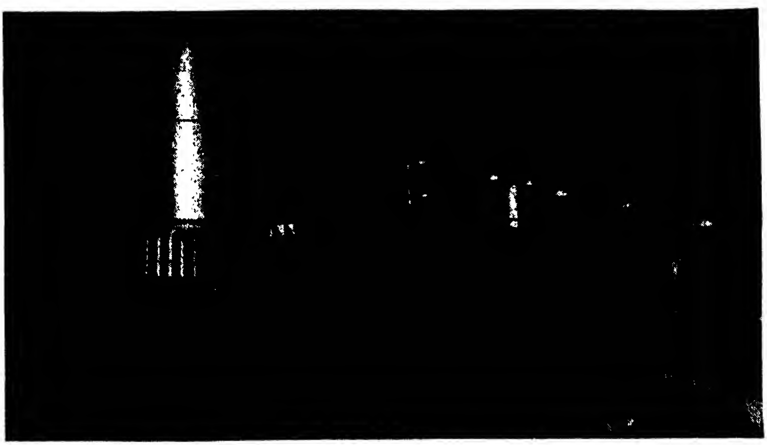


FIG. 68.—Cutting racks on a Fellows machine.

The various figures show a number of applications of the shaping method. In Fig. 62 the same cutter cuts all four gears, each having the properly shaped tooth for its diameter. Where the gears are close together, with but little clearance as in this cluster, there is no other method by which these gears can be cut. An internal job is seen in Fig. 63. Here the inner clutch teeth are cut by the cutter shown, only a shallow clearance groove being necessary for the cutter. In Fig. 64 is a special cam for an airplane-valve gear where the cutter makes two revolu-

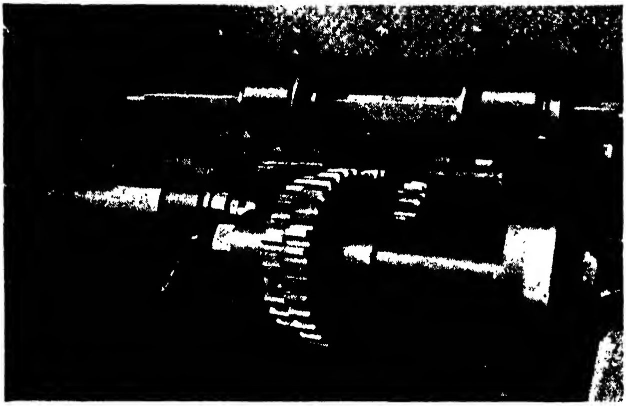


Fig. 69.—Cutting two cluster gears at once.

tions to one of the cam itself. Cutting such cams in stacks of five or six makes this a very economical operation. And with the proper cutter once developed the work is very accurate.

An unusual job in which a fast lead helix is being cut on a small diameter is seen in Fig. 65. Both the cutter and the work

Fig. 70.—Staggered tooth herringbone.

Fig. 71.—Herringbone without clearance groove.

turn at the correct rate to produce the lead required. Another type of work is seen in Fig. 66 where a three-cornered cutter produces a square hole. The work is for an overarm on a milling machine but can, of course, be used in any way desired.

Still another odd job is the skew-bevel gear in Fig. 67. This is

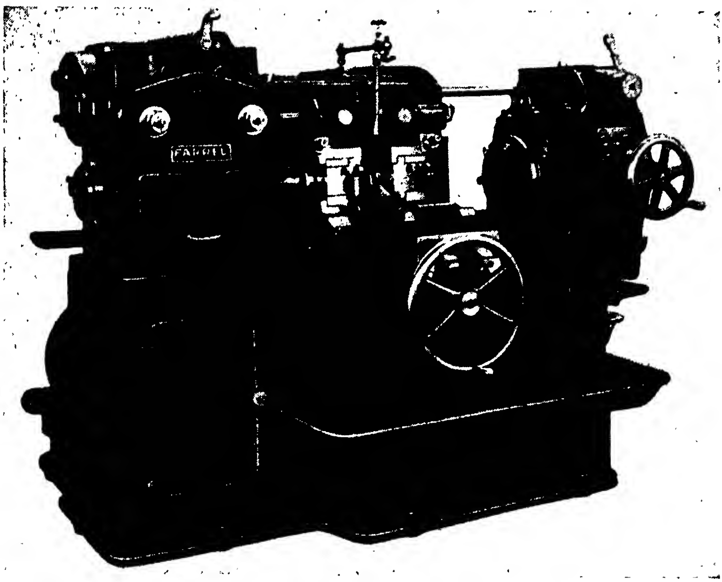


Fig. 72.—A small Sykes gear generator.

almost a crown gear. Here the cutter is set off from the center of the work and a helical cutter is used.

An excellent example of the way in which involute gear teeth are generated on a machine of the Fellows type is seen in Fig. 68. Here a cutter of fairly large diameter is generating the teeth in a number of small racks, held in a special clamp or vise. The action is exactly the same as the rolling of a gear in a rack, for the rack being cut moves at the same speed as the periphery of the cutter as it turns.

Fig. 73.—Tail bracket with clutch and wrench combined.

Sykes Gear Generator.—The Sykes gear generator is of the horizontal reciprocating, or shaping, type, in which the cutter is practically a gear with cutting edges. This machine sometimes has two horizontal cutting spindles, opposed to each other, the cutters feeding toward each other for cutting herringbone or double gears of any kind. Plain spur gears, helical, or herringbone are all cut on this machine. Cluster gears

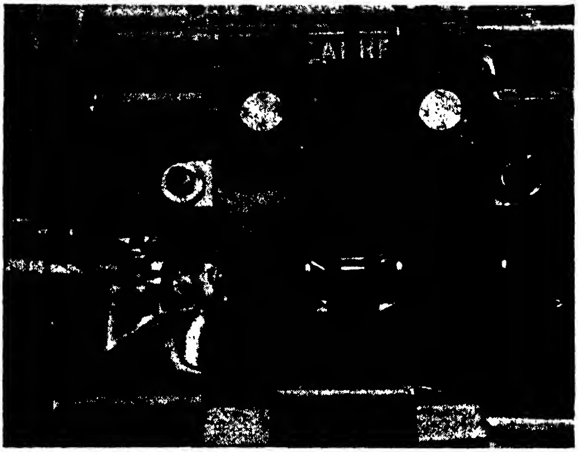


FIG. 74.—Close-up of cutters in cutter head.

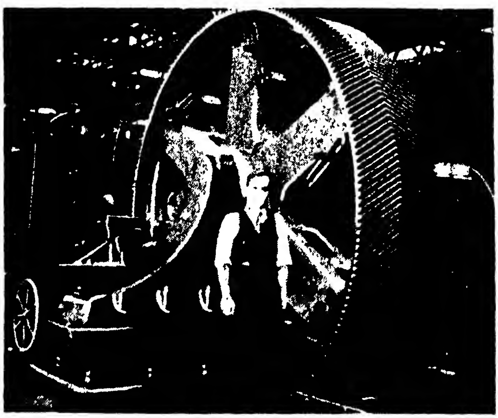


FIG. 75.—Large herringbone gear without clearance groove.

can be cut simultaneously, as in Fig. 69, as well as double helical or herringbone, as shown in Figs. 70 and 71. It will be noted that in both Figs. 69 and 70 the teeth are staggered, and that in Fig. 71 the herringbone is cut without a clearance groove for the cutters. The machines can, of course, be used single-ended when desired.

These machines are built in several sizes, up to gears 22 ft. in diameter, 60 in. width of face, and 6-in. circular patch. This machine, shown in Fig. 75, can handle a gear weighing 50 tons.

The indexing mechanism and the general operation are similar to other machines using gear-shaped cutters and generating by a planing or shaping action.

A small-size machine is shown in Fig. 72 and two of the details of the construction in Figs. 73 and 74. This machine cuts spur, helical, or herringbone gears from $\frac{1}{4}$ to 12 in. in diameter. On spur and helical gears it cuts gear faces up to

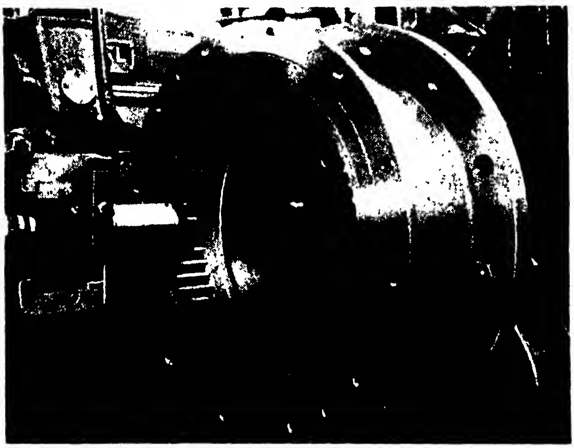


Fig. 76.—Straight-tooth internal gear being cut on a Sykes machine.

$2\frac{1}{2}$ in. wide and on helical gears up to 4 in. wide. The range of pitches is from 24 to 4 diametral pitch with 4-in. pitch-diameter cutters. Figure 73 is the tail bracket in which the wrench is combined with a clutch that permits the operator to loosen the bracket and withdraw it clear of the work mandrel, with the same movement, which reduces the time necessary for setting up. A close-up of the cutter heads is seen in Fig. 74, which also shows how cutter heads can be quickly adjusted by two screws and easily locked in position by the same wrench.

Two examples of large work are shown in Figs. 75 and 76. Fig. 75 shows a large herringbone gear with no clearance groove in the center. One of the cutters can be seen at the rear. The other is a straight-toothed internal gear with the teeth being generated by the cutter shown.

CHAPTER IV

HELICAL AND HERRINGBONE GEARS

HELICAL GEARS

Helical gears are coming into wider use in nearly all kinds of machinery. While their use dates back over half a century, the difficulty of cutting them by the old method with a single-tooth form cutter retarded their use. With the development of the gear-hobbing machine and the reciprocating gear cutter, or gear shaper, helical gears can be machined about as easily as the spur gear. The quietness of the helical gear, due to the simultaneous contact of several teeth, makes them desirable for automobile and many other uses.

Helical gears have been called by several names, "spiral" being the most common, although they have also been called "skew" gears in some sections. There is now a marked effort to call them by their proper name, which is helical and not spiral, as the teeth form a helix around the center just as a fast thread forms a helix on the bolt or shaft. Strictly speaking, there are no spiral gears, the nearest approach to this being a scroll such as is used in the back of a universal chuck. By common consent, however, we retain the name of spiral-bevel as applied to the type of gears developed by the Gleason Works for automobile rear-end drives. While this is not really a spiral, the designation serves to distinguish it from other gears, and it is such an important contribution to the field of gearing that it retains the name given by its originators. The same name is applied to the hypoid gear, which differs from the regular spiral bevel by having the pinion set above or below the center line of the large gear.

While helical gears can be cut on the milling machine in the same way that drills and reamers are fluted, this method is used only in experimental work or where very small quantities are required. They can also be cut with single cutters in gear-cutting machines of the older types, but the great majority are cut on hobbing machines or on gear shapers.

The gearing up of a hobbing machine to cut helical gears requires great care in the calculation and adjustment of the machine to give the required helix angle, at either right or left hand. Some idea of this has been pointed out in the preceding chapter on hobbing machines. In cutting helical or herringbone gears with the gear shaper the angle of the helix to be cut is taken care of by a former or guide, which turns the ram and the cutter at the proper rate. The arrangement of gears for indexing is the same as for spur gears. Makers of hobbing machines, however, provide the operators with a guidebook in which are full instructions and tables that make the setting up of the machine a comparatively easy matter.

Real Pitches of Circular Pitch Helical Gears.—The accompanying table will be found convenient in figuring particulars for spiral gearing, as it eliminates much of the work by shortening the process, thus making it quite an easy and simple matter to find the dimensions for either helical gears with axes parallel to each other or for gears with right-angle drive. See Table 34.

Formulas for use with the table are as follows: Circumference on pitch line = real pitch multiplied by number of teeth.

Lead of spiral = circumference on pitch line ÷ by the tangent.

Pitch diameter = circumference divided by 3.1416.

For whole diameter add the same amount above pitch line as for spur wheels of the same pitch as the normal pitch.

The following is an example of the use of the table: A pair of wheels is required to be: Ratio, 6 to 1; normal pitch, 1 in.; driver, 6 teeth; follower, 36 teeth; angle for driver, 66 deg.; angle for follower, 24 deg.

Referring to the table, we find that the real pitch for the driver is 2.4585.

2.4585 × 6 (*teeth*) = 14.751 (*circumference on pitch line*).

Cir. 14.751 ÷ 2.246 (*tangent*) = 6.567 (*lead of helix*).

Cir. 14.751 ÷ 3.1416 = 4.695 (*pitch diameter*).

For the follower the real pitch is 1.0946.

1.0946 × 36 = 39.4056 (*circumference*).

Cir. 39.4056 ÷ 0.4452 (*tangent*) = 88.512 (*lead of helix*).

Cir. 39.4056 ÷ 3.1416 = 12.543 (*pitch diameter*).

Another method of finding the lead of helix is to multiply the real pitch by the number of teeth, but for this purpose take the real pitch of the mating wheel.

In the above example we should have

Real pitch of follower, $1.0946 \times 6 = 6.5676$ (*lead of helix*).

Real pitch of driver, $2.4585 \times 36 = 88.506$.

It will be noticed that there is a slight difference in the result but this is unimportant, as it is only brought about by the dropping of a few decimal points in the tangent.

Spur-gear Cutters for Helical Gears.—*To find the number of a spur-gear cutter to be used in cutting a given helical gear*, locate

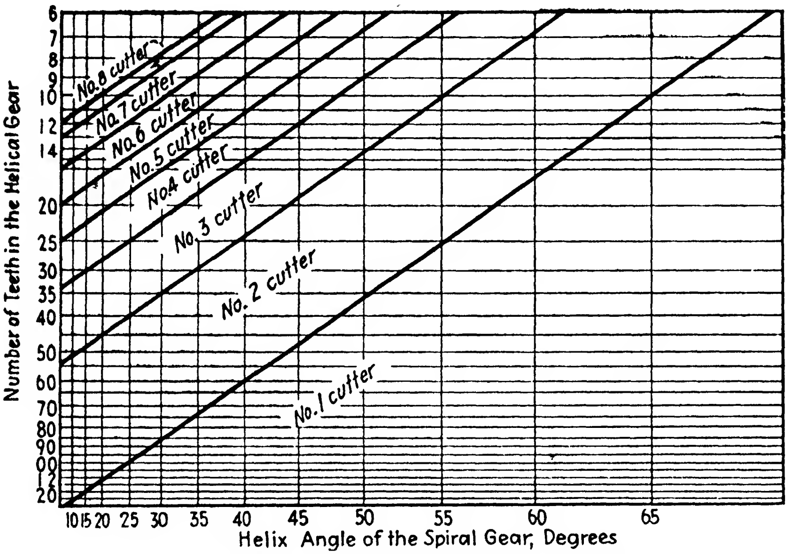


FIG. 77.—Diagram for selecting spur cutters for helical gears.

the intersection of lines traced from the points representing the number of teeth and the helix angle on the two scales. The number in the area on the chart within which the intersection falls is the cutter number of Brown and Sharpe's involute cutter system required. This is shown in Fig. 77.

Helical-gear Table.—While it is better in every case to understand the principles involved before using a table as this tends to prevent errors, it can be used with good results by simply following directions carefully. The subject of helical gears is so much more complicated than other gears that many will prefer to depend entirely on tables such as Table 35.

This table gives the circular pitch and addendum or the diametral pitch and lead of helices for one diametral pitch and with

TABLE 34.—TABLE OF REAL PITCHES FOR CIRCULAR PITCH HELICAL GEARS

Angles from axis, deg.		Normal pitches													
		1/4 in.		5/16 in.		3/8 in.		7/16 in.		1/2 in.		5/8 in.		3/4 in.	
Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower
80	10	1.4396	0.2538	1.7996	0.3173	2.1595	0.3808	2.5194	0.4422	2.8793	0.5077	3.5992	0.6346	4.3190	0.7616
79	11	1.3102	0.2546	1.6377	0.3183	1.9653	0.3820	2.2928	0.4457	2.6304	0.5093	3.2755	0.6366	3.9306	0.7640
78	12	1.2024	0.2556	1.5030	0.3195	1.8036	0.3833	2.1042	0.4473	2.4049	0.5112	3.0060	0.6390	3.6072	0.7666
77	13	1.1113	0.2565	1.3892	0.3207	1.6669	0.3848	1.9449	0.4490	2.2226	0.5131	2.7784	0.6414	3.3338	0.7696
76	14	1.0334	0.2576	1.2917	0.3220	1.5500	0.3865	1.8084	0.4509	2.0668	0.5153	2.5834	0.6440	3.1000	0.7730
75	15	0.9659	0.2591	1.2074	0.3238	1.4489	0.3885	1.6903	0.4529	1.9318	0.5181	2.4148	0.6476	2.8978	0.7772
74	16	0.9069	0.2600	1.1337	0.3250	1.3605	0.3901	1.5872	0.4551	1.8139	0.5201	2.2674	0.6502	2.7210	0.7802
73	17	0.8550	0.2614	1.0688	0.3268	1.2826	0.3931	1.4964	0.4575	1.7101	0.5228	2.1377	0.6536	2.5652	0.7842
72	18	0.8090	0.2628	1.0112	0.3286	1.2135	0.3943	1.4157	0.4600	1.6180	0.5257	2.0225	0.6572	2.4270	0.7886
71	19	0.7678	0.2644	0.9598	0.3305	1.1518	0.3966	1.3438	0.4627	1.5357	0.5288	1.9196	0.6610	2.3036	0.7932
70	20	0.7309	0.2660	0.9137	0.3325	1.0964	0.3990	1.2791	0.4656	1.4619	0.5320	1.8274	0.6651	2.1928	0.7981
69	21	0.6976	0.2678	0.8720	0.3347	1.0464	0.4017	1.2208	0.4686	1.3952	0.5356	1.7440	0.6694	2.0928	0.8034
68	22	0.6673	0.2696	0.8342	0.3370	1.0010	0.4044	1.1679	0.4718	1.3346	0.5392	1.6684	0.6740	2.0020	0.8088
67	23	0.6398	0.2716	0.7998	0.3395	0.9597	0.4074	1.1120	0.4752	1.2796	0.5432	1.5996	0.6790	1.9194	0.8148
66	24	0.6146	0.2736	0.7683	0.3420	0.9220	0.4105	1.0756	0.4789	1.2292	0.5472	1.5366	0.6841	1.8440	0.8210
65	25	0.5915	0.2758	0.7391	0.3448	0.8873	0.4137	1.0352	0.4827	1.1830	0.5516	1.4788	0.6896	1.7746	0.8274
64	26	0.5702	0.2781	0.7129	0.3477	0.8554	0.4172	0.9980	0.4868	1.1406	0.5563	1.4258	0.6954	1.7108	0.8344
63	27	0.5507	0.2806	0.6383	0.3507	0.8260	0.4209	0.9636	0.4910	1.1014	0.5612	1.3766	0.7014	1.6520	0.8418
62	28	0.5325	0.2831	0.6656	0.3539	0.7988	0.4247	0.9319	0.4955	1.0650	0.5662	1.3312	0.7078	1.5976	0.8494
61	29	0.5157	0.2858	0.6446	0.3573	0.7735	0.4287	0.9024	0.5002	1.0314	0.5716	1.2892	0.7146	1.5470	0.8574
60	30	0.5000	0.2886	0.6250	0.3608	0.7500	0.4330	0.8750	0.5051	1.0000	0.5773	1.2500	0.7216	1.5000	0.8660
59	31	0.4853	0.2916	0.6067	0.3646	0.7281	0.4375	0.8494	0.5104	0.9708	0.5833	1.2134	0.7292	1.4562	0.8750
58	32	0.4717	0.2948	0.5897	0.3685	0.7076	0.4422	0.8256	0.5159	0.9435	0.5896	1.1794	0.7370	1.4152	0.8844
57	33	0.4590	0.2981	0.5738	0.3726	0.6885	0.4471	0.8033	0.5216	0.9180	0.5962	1.1476	0.7452	1.3770	0.8942
56	34	0.4470	0.3015	0.5588	0.3769	0.6706	0.4523	0.7824	0.5277	0.8941	0.6031	1.1176	0.7538	1.3412	0.9046
55	35	0.4358	0.3052	0.5448	0.3815	0.6538	0.4578	0.7627	0.5341	0.8717	0.6104	1.0896	0.7630	1.3076	0.9156
54	36	0.4253	0.3090	0.5316	0.3862	0.6380	0.4635	0.7443	0.5408	0.8506	0.6180	1.0632	0.7724	1.2760	0.9270
53	37	0.4154	0.3130	0.5192	0.3913	0.6231	0.4695	0.7269	0.5478	0.8308	0.6260	1.0384	0.7826	1.2462	0.9390
52	38	0.4060	0.3172	0.5076	0.3965	0.6091	0.4759	0.7161	0.5552	0.8121	0.6345	1.0152	0.7930	1.2182	0.9518
51	39	0.3972	0.3217	0.4965	0.4021	0.5959	0.4825	0.6952	0.5628	0.7945	0.6434	0.9930	0.8042	1.1918	0.9650
50	40	0.3889	0.3264	0.4861	0.4079	0.5834	0.4895	0.6806	0.5711	0.7778	0.6527	0.9722	0.8159	1.1668	0.9790
49	41	0.3810	0.3312	0.4763	0.4140	0.5716	0.4969	0.6668	0.5797	0.7621	0.6625	0.9526	0.8280	1.1432	0.9938
48	42	0.3736	0.3364	0.4670	0.4205	0.5604	0.5046	0.6538	0.5887	0.7472	0.6728	0.9340	0.8410	1.1208	1.0092
47	43	0.3665	0.3418	0.4582	0.4272	0.5498	0.5127	0.6415	0.5982	0.7331	0.6836	0.9164	0.8544	1.0996	1.0254
46	44	0.3598	0.3475	0.4498	0.4344	0.5398	0.5213	0.6298	0.6082	0.7197	0.6950	0.8996	0.8688	1.0796	1.0426
45	45	0.3536	0.3536	0.4419	0.4419	0.5303	0.5303	0.6187	0.6187	0.7071	0.7071	0.8839	0.8839	1.0606	1.0606

TABLE 34.—TABLE OF REAL PITCHES FOR CIRCULAR PITCH HELICAL GEARS.—(*Continued*)

Angles from axis, deg.		Normal pitches												Tangent of angle	
		7/8 in.		1 in.		1 1/8 in.		1 1/4 in.		1 3/8 in.		1 1/2 in.			
Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower
80	10	5.0388	0.8884	5.7587	1.0154	6.4785	1.1423	7.1983	1.2692	7.8192	1.3962	8.6380	1.5230	5.6713	0.1763
79	11	4.5856	0.8914	5.2407	1.0187	5.8961	1.1460	6.5510	1.2732	7.2061	1.4007	7.8612	1.5280	5.1446	0.1944
78	12	4.2084	0.8946	4.8098	1.0223	5.4109	1.1501	6.0122	1.2780	6.6134	1.4057	7.2146	1.5335	4.7046	0.2126
77	13	3.8898	0.8980	4.4454	1.0263	5.0001	1.1546	5.5568	1.2828	6.1125	1.4111	6.6681	1.5394	4.3315	0.2309
76	14	3.6168	0.9018	4.1336	1.0306	4.6500	1.1594	5.1669	1.2882	5.6837	1.4171	6.2004	1.5459	4.0108	0.2493
75	15	3.3806	0.9058	3.8637	1.0353	4.3466	1.1658	4.8296	1.2952	5.3126	1.4248	5.7955	1.5543	3.7321	0.2679
74	16	3.1744	0.9102	3.6279	1.0403	4.0814	1.1703	4.5348	1.3004	4.9884	1.4304	5.4419	1.5604	3.4874	0.2867
73	17	2.9928	0.9150	3.4203	1.0457	3.8478	1.1764	4.2754	1.3071	4.7029	1.4378	5.1304	1.5685	3.2709	0.3057
72	18	2.8314	0.9200	3.2360	1.0515	3.6405	1.1829	4.0450	1.3143	4.4495	1.4457	4.8540	1.5771	3.0777	0.3249
71	19	2.6876	0.9254	3.0715	1.0576	3.4554	1.1898	3.8394	1.3220	4.2233	1.4542	4.6072	1.5864	2.9042	0.3443
70	20	2.5582	0.9312	2.9238	1.0641	3.2892	1.1972	3.6548	1.3302	4.0202	1.4632	4.3857	1.5962	2.7475	0.3640
69	21	2.4416	0.9372	2.7904	1.0711	3.1391	1.2050	3.4880	1.3389	3.8368	1.4728	4.1856	1.6067	2.6051	0.3839
68	22	2.3358	0.9436	2.6694	1.0785	3.0030	1.2133	3.3369	1.3480	3.6705	1.4829	4.0041	1.6178	2.4751	0.4040
67	23	2.2240	0.9504	2.5593	1.0863	2.8792	1.2221	3.1991	1.3580	3.5190	1.4937	3.8389	1.6295	2.3559	0.4245
66	24	2.1512	0.9578	2.4585	1.0946	2.7659	1.2314	3.0732	1.3682	3.3805	1.5051	3.6876	1.6418	2.2460	0.4452
65	25	2.0704	0.9654	2.3662	1.1033	2.6619	1.2412	2.9577	1.3792	3.2535	1.5171	3.5493	1.6550	2.1445	0.4663
64	26	1.9960	0.9736	2.2811	1.1127	2.5663	1.2517	2.8515	1.3908	3.1366	1.5298	3.4218	1.6689	2.0503	0.4877
63	27	1.9272	0.9820	2.2026	1.1223	2.4780	1.2626	2.7533	1.4029	3.0287	1.5431	3.3040	1.6835	1.9626	0.5095
62	28	1.8638	0.9910	2.1300	1.1326	2.3963	1.2741	2.6625	1.4157	2.9288	1.5572	3.1950	1.6988	1.8807	0.5317
61	29	1.8048	1.0004	2.0626	1.1433	2.3205	1.2862	2.5783	1.4292	2.8361	1.5721	3.0940	1.7150	1.8040	0.5543
60	30	1.7500	1.0102	2.0000	1.1548	2.2500	1.2990	2.5000	1.4433	2.7500	1.5877	3.0000	1.7320	1.7321	0.5774
59	31	1.6988	1.0208	1.9415	1.1667	2.1843	1.3125	2.4269	1.4583	2.6696	1.6042	2.9134	1.7499	1.6643	0.6009
58	32	1.6512	1.0318	1.8870	1.1790	2.1229	1.3266	2.3588	1.4740	2.5947	1.6213	2.8306	1.7687	1.6003	0.6249
57	33	1.6066	1.0432	1.8360	1.1923	2.0655	1.3414	2.2951	1.4904	2.5246	1.6395	2.7451	1.7885	1.5399	0.6494
56	34	1.5648	1.0554	1.7883	1.2062	2.0118	1.3569	2.2353	1.5077	2.4589	1.6585	2.6824	1.8093	1.4826	0.6745
55	35	1.5254	1.0682	1.7434	1.2207	1.9614	1.3734	2.1793	1.5260	2.3972	1.6786	2.6151	1.8311	1.4281	0.7002
54	36	1.4886	1.0816	1.7012	1.2360	1.9139	1.3905	2.1266	1.5449	2.3392	1.6997	2.5519	1.8541	1.3764	0.7265
53	37	1.4538	1.0956	1.6614	1.2521	1.8693	1.4086	2.0700	1.5652	2.2847	1.7217	2.4924	1.8781	1.3270	0.7536
52	38	1.4322	1.1104	1.6242	1.2690	1.8273	1.4276	2.0303	1.5862	2.2333	1.7449	2.4364	1.9035	1.2799	0.7813
51	39	1.3904	1.1256	1.5890	1.2867	1.7876	1.4476	1.9862	1.6084	2.1847	1.7693	2.3835	1.9301	1.2349	0.8098
50	40	1.3612	1.1422	1.5557	1.3054	1.7502	1.4686	1.9446	1.6317	2.1391	1.7949	2.3336	1.9581	1.1918	0.8391
49	41	1.3336	1.1594	1.5240	1.3250	1.7148	1.4906	1.9054	1.6562	2.0958	1.8219	2.2863	1.9875	1.1504	0.8693
48	42	1.3076	1.1774	1.4944	1.3456	1.6813	1.5138	1.8681	1.6820	2.0549	1.8502	2.2417	2.0184	1.1106	0.9904
47	43	1.2830	1.1964	1.4662	1.3673	1.6495	1.5382	1.8328	1.7089	2.0161	1.8001	2.1994	2.0510	1.0724	0.9325
46	44	1.2596	1.2164	1.4395	1.3901	1.6194	1.5639	1.7994	1.7377	1.9792	1.9114	2.1593	2.0852	1.0355	0.9657
45	45	1.2374	1.2374	1.4142	1.4142	1.5910	1.5910	1.7678	1.7678	1.9445	1.9445	2.1213	2.1213	1.0000	1.0000

TABLE 35.—HELICAL GEAR TABLE

Shaft angles 90 deg. for one diametral pitch

Angle of helix, deg.	To obtain the circular pitch for one tooth, divide by the required diametral pitch	To obtain the pitch diameter, divide by the required diametral pitch and multiply the quotient by the required number of teeth	To obtain the lead of helix, divide by the required diametral pitch and multiply quotient by the required number of teeth		To obtain the pitch diameter, divide by the required diametral pitch and multiply the quotient by the required number of teeth	To obtain the circular pitch for one tooth, divide by the required diametral pitch	Angle of helix, deg.
	Circular pitch	One tooth or addendum	Lead of helix		One tooth or addendum	Circular pitch	
Small wheel	Small wheel	Small wheel	Small wheel	Large wheel	Large wheel	Large wheel	Large wheel
1	3 1419	1.0001	180.05	3 1420	57.298	180 01	89
2	3 1435	1 0006	90.020	3.1435	28.653	90.016	88
3	3.1457	1.0013	60.032	3.1458	19 107	60.026	87
4	3.1491	1.0024	45.038	3.1492	14.335	45.035	86
5	3.1535	1.0038	37.077	3.1527	11 473	36.044	85
6	3.1589	1.0055	30.056	3.1589	9.5667	30.055	84
7	3.1652	1.0075	25.728	3.1651	8.2055	25.778	83
8	3.1724	1 0098	22.573	3.1724	7.1852	22.573	82
9	3.1806	1.0124	20.082	3.1807	6.3924	20.082	81
10	3.1900	1.0154	18.092	3.1901	5 7587	18.092	80
11	3.2003	1.0187	16.464	3.2003	5.2408	16.464	79
12	3.2145	1.0232	15.076	3.2105	4.8097	15.014	78
13	3.2242	1 0263	13.966	3.2294	4 4454	13.988	77
14	3.2377	1.0306	12.986	3.2378	4.1335	12.986	76
15	3.2522	1.0352	12.138	3.2524	3.8637	12.138	75
16	3.2679	1.0402	11.393	3.2678	3.6279	11.397	74
17	3.2848	1.0456	10.417	3.2821	3.4203	10.745	73
18	3.3116	1 0514	10.192	3.3032	3.2360	10 166	72
19	3.3225	1.0576	9.6494	3.3225	3.0715	9.6494	71
20	3.3430	1.0641	9.1848	3.3433	2.9238	9.1854	70
21	3.3650	1.0711	8.7662	3.3652	2.7904	8.7663	69
22	3.3882	1.0785	8.3862	3.3833	2.6694	8.3862	68
23	3 4127	1.0863	8.0399	3.4129	2.5593	8.0403	67
24	3.4451	1.0946	7.7379	3.4391	2.4585	7.7242	66
25	3.4661	1.1033	7.4332	3.4663	2.3662	7.4336	65
26	3.4953	1.1126	7 1664	3.4952	2.2811	7.1663	64
27	3.5258	1.1223	6.9198	3.5257	2.2026	6.9197	63
28	3.5579	1.1325	6.6912	3.5575	2.1300	6.6916	62
29	3.5918	1.1433	6.4799	3.5919	2.0626	6.4799	61
30	3.6276	1.1547	6.2778	3.6277	2.0000	6.2832	60
31	3.6650	1.1666	6.0979	3.6652	1.9416	6.0997	59
32	3.7043	1.1791	5.9282	3.7044	1.8870	5.9282	58
33	3 7457	1.1923	5.7710	3.7459	1.8360	5 7680	57
34	3 7894	1.2062	5.6181	3.7826	1.7882	5.6178	56
35	3.8349	1.2207	5.4754	3.8351	1.7434	5.4770	55
36	3.8830	1.2360	5.3431	3.8834	1.7013	5.3448	54
37	3.9336	1.2521	5.2201	3.9261	1.6616	5.2200	53
38	3.9867	1.2690	5.1028	3.9921	1.6242	5.1026	52
39	4.0482	1.2867	4.9866	4.0416	1 5890	4.9920	51
40	4.1010	1.3054	4.8873	4.1012	1.5557	4.8874	50
41	4.1626	1.3250	4.7885	4.1540	1.5242	4.7884	49
42	4.2273	1.3456	4.6949	4.2272	1.4944	4.6948	48
43	4 2956	1.3673	4.6065	4.2956	1.4662	4.6062	47
44	4.3671	1.3901	4.5223	4.3675	1.4395	4.5225	46
45	4.4428	1.4142	4.4428	4.4428	1.4142	4.4428	45

teeth having angles from 1 to 45 and 45 to 89 deg. For other pitches divide the addendum given and the helix number by the required pitch and multiply the results by the required number of teeth. This will give the pitch diameter and lead of helix for each wheel. For the outside diameter add two diametral pitches as in spur gearing.

Suppose we want a pair of helical gears with 10- and 80-deg. angles, 8 diametral pitch cutter, with 16 teeth in the small gear, having 10-deg. angle and 10 teeth in the larger gear with its 80-deg. angle.

Find the 10-deg. angle of helix and in the third column find 1.0154. Divide by pitch, 8, and get 0.1269. Multiply this by number of teeth $-0.1269 \times 16 = 2.030 =$ pitch diameter. Add 2 pitches $-$ two $\frac{1}{8} = \frac{1}{4}$ and $2.030 + 0.25 = 2.28$ in. outside diameter.

The lead of helix for 10 deg. for small wheel is 18.092. Divide by pitch $= 18.092 \div 8 = 2.2615$. Multiply by number of teeth, $2.2615 \times 16 = 36.18$, the lead of helix, which means that it makes one turn in 36.18 in.

For the other gear with its 80-deg. angle, find the addendum, 5.7587. Divide by pitch, $8 = 0.7198$. Multiply by number of teeth, $10 = 7.198$. Add two pitches, or 0.25, gives 7.448 as outside diameter.

The lead of helix is 3.1901. Dividing by pitch, $8 = 0.3988$. Multiplying by number of teeth $= 3.988$ the lead of helix.

When racks are to mesh with helical gears, divide the number in the circular pitch columns for the given angle by the required diametral pitch to get the corresponding circular pitch.

If we want to make a rack to mesh with a 40-deg. helical gear of 8 pitch: Look for circular pitch opposite 40 and find 4.101. Dividing by 8 gives 0.512 as the circular pitch for this angle. The greater the angle the greater the circular or linear pitch, as can be seen by trying an 80-deg. angle. Here the circular pitch is 2.261 in.

Hobbing-machine-gear Calculation for Cutting Helical Gears on No. 12 Barber-Colman Machine

Given:

PNC = normal circular pitch of single-thread hob.

N = number of teeth in gear to be cut.

K = index constant.

$$= \frac{PNC}{\text{Sine helix angle of gear} \times \text{feed selected}} \quad \text{(Use nearest full number)}$$

$R_1 = 30/N$ for machine with single-thread worm

$R_2 = 15/N$ for machine with double-thread worm

Formula A.—Single-thread hob, single-thread machine.

$$\text{Ratio of index gears} = \frac{30 \times K}{(N \times K) \text{ plus 1 when hob is opposite helix to gear or minus 1 when hob is same helix as gear}}$$

$$\text{Ratio of feed gears} = \frac{PNC}{K \times \text{sine helix angle of gear} \times 0.075}$$

Formula B.—Multiple-thread hob, single-thread machine.

$$\text{Ratio of index gears} = \frac{30 \times K \times \text{No. of threads in hob}}{(N \times K) \text{ plus 1 when hob is opposite helix to gear or minus 1 when hob is same helix as gear}}$$

Ratio of feed gears—Same as in formula *A* above.

Formula C.—Single-thread hob, double-thread machine.

$$\text{Ratio of index gears} = \frac{15 \times K}{(N \times K) \text{ plus 1 when hob is opposite helix to gear or minus 1 when hob is same helix as gear}}$$

$$\text{Ratio of feed gears} = \frac{PNC \times 2}{K \times \text{sine of helix angle of gear} \times 0.075}$$

Formula D.—Multiple-thread hob, double-thread machine.

$$\text{Ratio of index gears} = \frac{15 \times K \times \text{No. of threads in hob}}{(N \times K) \text{ plus 1 when hob is opposite helix angle to gear or minus 1 when hob is same helix angle as gear}}$$

Ratio of feed gears—Same as in formula *C* above.

If the ratios as figured from the above formulas do not result in numbers that can be factored, the value of K must be changed to the nearest whole number that will give a factorable quantity.

Example.—
Right-hand helix gear—right-hand hob, single thread.
$PNC = 0.5236 = 6$ Normal diametral pitch.
$N = 36$ Teeth.
Feed $= 0.060$ per revolution of hob.
$R_1 = 30/N =$ Ratio for machine with single-thread worm.
$a = 20° 50' =$ helix angle of gear to be cut.
$\sin a = 0.35565$
Then

$$K = \frac{0.5236}{0.35565 \times 0.060} = 24.54$$

$$\text{Nearest whole number} = 25 = K$$

$$\text{Ratio of index gears} = \frac{30 \times 25}{(36 \times 25) \text{ minus } 1} = \frac{750}{899}$$

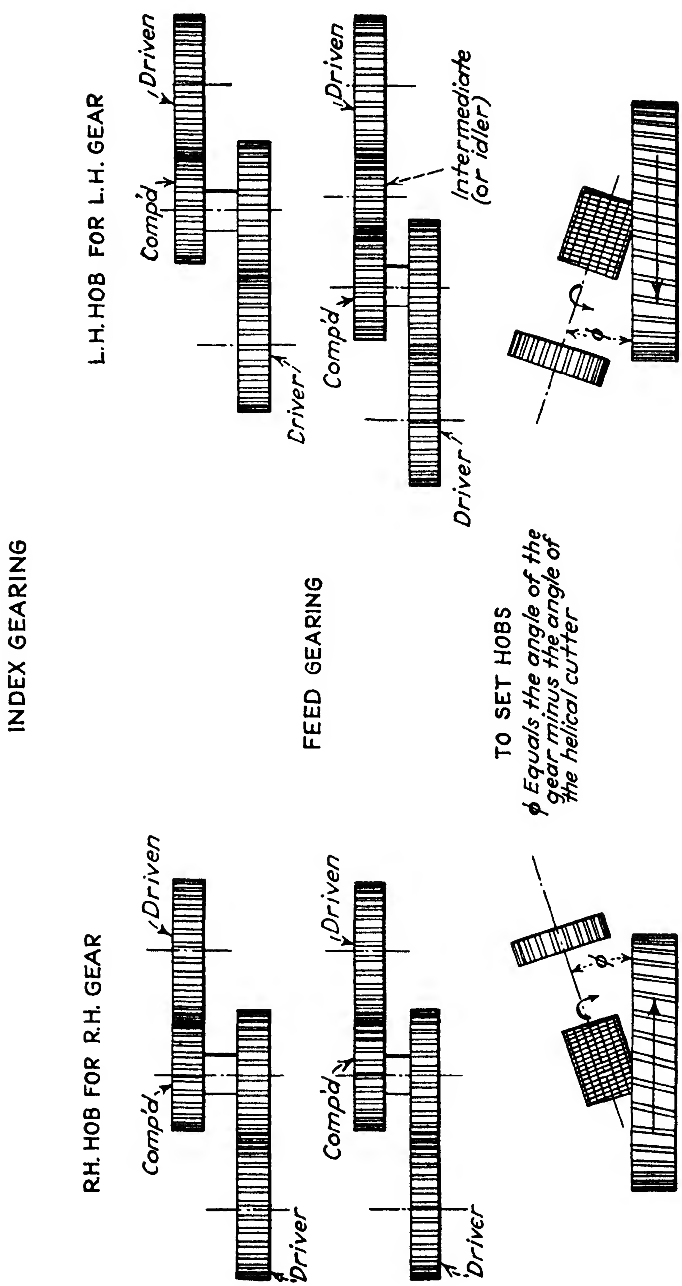
Factoring, we get a gear combination of $75/58 \times 40/62$

$$\text{Ratio of feed gears} = \frac{0.5236}{25 \times 0.35565 \times 0.075} = \frac{0.5236}{0.66684} = 0.785191$$

The gear combination for this ratio is $73/62 \times 60/90$.

Helical-gear Calculations.—Although hobbing machines differ in construction the principles on which they operate are similar and the methods of setting them up follow similar lines. The following suggestions regarding the Gould and Eberhardt machines will give a good general idea of the methods followed.

The formula which follows and the notations, together with the sketch of the gear to be cut, showing the angles and distances to be considered, will serve for other machines as well. Tables 36 to 39 will be found helpful. The example given, with the feed gears and the proof formula, as well as the error shown, should be helpful. There are also tables of cutting feeds and speeds which are convenient as suggestions. In addition there are tables of sines, cosines, and other functions needed in these calculations, together with explanations as to their use. The diagram opposite shows the arrangement of gears on the same machine.



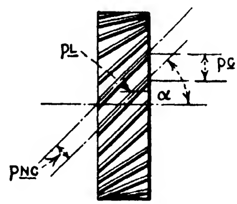
Arrangement of gears on Gould and Eberhardt machine.

Formula to Calculate Helical Gears

$$\text{Pitch diameter} = \frac{N}{P^{ND} \times \cos \alpha} \quad \text{or} \quad \frac{N \times a}{\cos \alpha}$$

$$\text{Outside diameter} = \text{pitch diameter} + 2\,a$$

Notation



N = number of teeth to be cut.
P^{ND} = normal diametral pitch of gear.
P^{NC} = normal circular pitch of gear.
P^{C} = circular pitch of gear.
P^{L} = linear pitch of gear.
a = addendum of tooth.
α = angle of helix with the axis.
β = angle marked on helical cutter.
ϕ = angle to set cutter.
C = feed constant selected from table.
E = difference between 1 and the result of proof formula.
T = number of threads in hob.
V = ratio of feed gearing.

Example: 71 teeth; $6P^{ND}$; $0.5236P^{NC}$; 45 deg.; cast iron; 0.05 feed.

Index gears: See feed constants under 0.05 feed; for $6P^{ND}$; 45 deg.; find $C = 15$; see index change-gear table for nearest $C = 15$; for 71 teeth find

$$\frac{68 \times 50}{67 \times 60} = \frac{\text{driver} \times \text{comp'd}}{\text{comp'd} \times \text{worm}} \quad \text{and} \quad C = 17.$$

$$\text{formula} = \frac{60 \times C}{(N \times C) - 1}.$$

Feed gears:

$$\frac{8 \times P^{NC}}{C \times \text{sine}\ \alpha} = \frac{\text{driver} \times \text{comp'd}}{\text{comp'd} \times \text{driven}} = V = \frac{8 \times 0.5236}{17 \times 0.70711} = 0.34846 \text{ gears} =$$

$$\frac{39 \times 52}{97 \times 60} = 0.348453$$

Proof formula:

$$\frac{8 \times P^{NC}}{V \times C \times \text{sine}\ \alpha} = 1 \leftarrow \text{allowable limits } \frac{0.9999}{1.0001} =$$

$$\frac{8 \times 0.5236}{\frac{39 \times 52}{97 \times 60} \times 17 \times 0.70711} = 1.000021$$

Error:

$$\text{per inch of width} = E \times \tan \alpha = 0.000021 \times 1 = 0.000021$$

$$\begin{array}{r} 1.000021 \\ 1.000000 \\ \hline 0.000021 = E \end{array}$$

TABLE 36.—CUTTING SPEEDS AND REVOLUTIONS PER MINUTE

p_{ND}	Cutter diameter	Cutting speeds, ft. per min.							
		45	50	55	60	70	80	90	100
		Revolution of cutter per minute							
2	$5\frac{3}{4}$	31	34	37	40	47	54	60	68
$2\frac{1}{4}$	$5\frac{1}{4}$	33	37	40	44	51	59	66	73
$2\frac{1}{2}$	5	35	38	42	46	53	61	69	76
$2\frac{3}{4}$	$4\frac{3}{4}$	36	40	45	49	56	65	72	80
3	$4\frac{1}{2}$	38	43	47	51	60	68	77	85
$3\frac{1}{2}$	$4\frac{1}{8}$	42	46	51	56	65	74	83	93
4	$3\frac{7}{8}$	44	49	54	59	69	79	90	99
5	$3\frac{1}{2}$	49	55	60	65	77	87	98	109
6	$3\frac{1}{4}$	53	59	65	71	82	94	106	118
$6\frac{1}{2}$	$2\frac{7}{8}$	60	67	73	80	93	106	119	133
7	3	57	64	70	76	89	102	114	127
8	$2\frac{7}{8}$	60	67	73	80	93	106	119	133
9	$2\frac{3}{4}$	63	70	77	84	97	111	125	139
10	$2\frac{3}{4}$	63	70	77	84	97	111	125	139
12 & 16	$2\frac{1}{2}$	69	77	84	92	107	122	137	153
18	$2\frac{3}{8}$	63	70	77	84	97	111	125	139

NOTE.—These speeds are suggested as the result of long experience but should not be considered as applying to all classes of work.

TABLE 37.—CUTTER FEEDS PER REVOLUTION OF GEAR FOR SPUR GEARS

Feed	Gearing			Material
	Driven	Comp'd	Driver	
0.01	96	24		Tool steel
		75	24	
0.02	80	40		Tough steel
		75	24	
0.03	68	51		Average steel
		75	24	
0.04	60	60		Soft steel
		75	24	
0.05	60	60		Average cast iron
		75	30	
0.06	60	60		Soft cast iron
		75	36	
0.08	60	60		Roughing soft steel
		75	48	
0.10	60	60		Roughing aver. C.I.
		50	40	
0.12	60	60		Roughing aver. C.I.
		50	51	
0.15	60	60		Roughing soft C.I.
		45	54	

$$\text{Formula} = \frac{\text{driver}}{8 \times \text{driven}} = \text{feed}$$

Cutter feeds for helical gears
0–36 deg. same as for spur feed
36–48 deg. = $\frac{4}{5}$ of spur feed
48–60 deg. = $\frac{2}{3}$ of spur feed
60–70 deg. = $\frac{1}{2}$ of spur feed
70 and up = $\frac{1}{3}$ of spur feed

Change gears
24 to 97 teeth inclusive
2–24, 2–30, 3–40, 2–45, 5–60, 2–80, 2–90, 2–96, 1–120 in all 87 gears.

NOTE.—These feeds are suggested as the result of long experience but should not be considered as applying to all classes of work.

TABLE 38.—MULTIPLICATION TABLE USED WITH FEED GEAR FORMULA

P_{ND}	P_{NC}	$8 \times P_{NC}$	P_{ND}	P_{NC}	$8 \times P_{NC}$
1	3.14159	25.1327	6	0.52360	4.1888
1¼	2.51327	20.1061	7	0.4488	3.5904
1½	2.09439	16.7551	8	0.39270	3.1416
1¾	1.79519	14.3615	9	0.34906	2.79248
2	1.5708	12.5664	10	0.31416	2.51328
2¼	1.39626	11.1700	12	0.26180	2.0944
2½	1.25664	10.0531	14	0.22440	1.7952
2¾	1.1424	9.1392	16	0.19635	1.5708
3	1.04720	8.3776	18	0.17453	1.39624
3½	0.89760	7.1808	20	0 15708	1.25664
4	0.78540	6.2832	24	0.13090	1.0472
5	0.62832	5.02656	30	0.10472	0.083776

TABLE 39.—NATURAL FUNCTIONS FOR GEAR CALCULATIONS

Deg.	Min.	Sine	Cosine	Tangent	Cotangent	Secant	Cosecant	Deg.	Min.
0	0	0.00000	1.00000	0.00000	∞	1.00000	∞	90	0
	10	0.00291	1.00000	0.00291	343.77371	1.00001	343.77516		50
	20	0.00582	0.99998	0.00582	171.88540	1.00002	171.88831		40
	30	0.00873	0.99996	0.00873	114.58865	1.00004	114.59301		30
	40	0.01164	0.99993	0.01164	85.93979	1.00007	85.94561		20
	50	0.01454	0.99989	0.01455	68.75009	1.00011	68.75736		10
1	0	0.01745	0.99985	0.01746	57.28996	1.00015	57.29869	89	0
	10	0.02036	0.99979	0.02036	49.10388	1.00021	49.11406		50
	20	0.02327	0.99973	0.02328	48.96408	1.00027	42.97571		40
	30	0.02618	0.99966	0.02619	38.18846	1.00034	38.20155		30
	40	0.02908	0.99958	0.02910	34.36777	1.00042	34.38232		20
	50	0.03199	0.99949	0.03201	31.24158	1.00051	31.25758		10
2	0	0.03490	0.99939	0.03492	28.63625	1.00061	28.65371	88	0
	10	0.03781	0.99929	0.03783	26.43160	1.00072	26.45051		50
	20	0.04071	0.99917	0.04075	24.54176	1.00083	24.56212		40
	30	0.04362	0.99905	0.04366	22.90377	1.00095	22.92559		30
	40	0.04653	0.99892	0.04658	21.47040	1.00108	21.49368		20
	50	0.04943	0.99878	0.04949	20.20555	1.00122	20.23028		10
3	0	0.05234	0.99863	0.05241	19.08114	1.00137	19.10732	87	0
	10	0.05524	0.99847	0.05533	18.07498	1.00153	18.10262		50
	20	0.05814	0.99831	0.05824	17.16934	1.00169	17.19843		40
	30	0.06105	0.99813	0.06116	16.34986	1.00187	16.38041		30
	40	0.06395	0.99795	0.06408	15.60478	1.00205	15.63679		20
	50	0.06685	0.99776	0.06700	14.92442	1.00224	14.95788		10
4	0	0.06976	0.99756	0.06993	14.30067	1.00244	14.33559	86	0
	10	0.07266	0.99736	0.07285	13.72674	1.00265	13.76312		50
	20	0.07556	0.99714	0.07578	13.19688	1.00287	13.23472		40
	30	0.07846	0.99692	0.07870	12.70621	1.00309	12.74550		30
	40	0.08136	0.99668	0.08163	12.25051	1.00333	12.29125		20
	50	0.08426	0.99644	0.08456	11.82617	1.00357	11.86837		10
5	0	0.08716	0.99619	0.08749	11.43005	1.00382	11.47371	85	0
	10	0.09005	0.99594	0.09042	11.05943	1.00408	11.10455		50
	20	0.09295	0.99567	0.09335	10.71191	1.00435	10.75849		40
	30	0.09585	0.99540	0.09629	10.38540	1.00463	10.43343		30
	40	0.09874	0.99511	0.09923	10.07803	1.00491	10.12752		20
	50	0.10164	0.99482	0.10216	9.78817	1.00521	9.83912		10
6	0	0.10453	0.99452	0.10510	9.51436	1.00551	9.56677	84	0
	10	0.10742	0.99421	0.10805	9.25530	1.00582	9.30917		50
	20	0.11031	0.99390	0.11099	9.00983	1.00614	9.06515		40
	30	0.11320	0.99357	0.11394	8.77689	1.00647	8.83367		30
	40	0.11609	0.99324	0.11688	8.55555	1.00681	8.61379		20
	50	0.11898	0.99290	0.11983	8.34496	1.00715	8.46466		10
7	0	0.12187	0.99255	0.12278	8.14435	1.00751	8.20551	83	0
	10	0.12476	0.99219	0.12574	7.95302	1.00787	8.01565		50
	20	0.12764	0.99182	0.12869	7.77035	1.00825	7.83443		40
	30	0.13053	0.99144	0.13165	7.59575	1.00863	7.66130		30
	40	0.13341	0.99106	0.13461	7.42871	1.00902	7.49571		20
	50	0.13629	0.99067	0.13758	7.26873	1.00942	7.33719		10
8	0	0.13917	0.99027	0.14054	7.11537	1.00983	7.18530	82	0
	10	0.14205	0.98986	0.14351	6.96823	1.01024	7.03962		50
	20	0.14493	0.98944	0.14648	6.82694	1.01067	6.89979		40
	30	0.14781	0.98902	0.14945	6.69116	1.01111	6.76547		30
	40	0.15069	0.98858	0.15243	6.56055	1.01155	6.63633		20
	50	0.15356	0.98814	0.15540	6.43484	1.01200	6.51208		10
9	0	0.15643	0.98769	0.15838	6.31375	1.01247	6.39245	81	0
	10	0.15931	0.98723	0.16137	6.19703	1.01294	6.27719		50
	20	0.16218	0.98676	0.16435	6.08444	1.01342	6.16607		40
	30	0.16505	0.98629	0.16734	5.97576	1.01391	6.05886		30
	40	0.16792	0.98580	0.17033	5.87080	1.01440	5.95536		20
	50	0.17078	0.98531	0.17333	5.76937	1.01491	5.85539		10
10	0	0.17365	0.98481	0.17633	5.67128	1.01543	5.75877	80	0
	10	0.17651	0.98430	0.17933	5.57638	1.01595	5.66533		50
	20	0.17937	0.98378	0.18233	5.48451	1.01649	5.57493		40
	30	0.18224	0.98325	0.18534	5.39552	1.01703	5.48740		30
	40	0.18509	0.98272	0.18835	5.30928	1.01758	5.40263		20
	50	0.18795	0.98218	0.19136	5.22566	1.01815	5.32049		10
11	0	0.19081	0.98163	0.19438	5.14455	1.01872	5.24084	79	0
	10	0.19366	0.98107	0.19740	5.06584	1.01930	5.16359		50
	20	0.19652	0.98050	0.20042	4.98940	1.01989	5.08863		40
Deg.	Min.	Cos	Sine	Cotan	Tan	Cosec	Sec	Deg.	Min.

Table 39.—Natural Functions for Gear Calculations.—(*Continued*)

Deg.	Min.	Sine	Cosine	Tangent	Cotangent	Secant	Cosecant	Deg.	Min.
11	30	0.19937	0.97992	0.20345	4.91516	1.02049	5.01585		30
	40	0.20222	0.97934	0.20648	4.84300	1.02110	4.94517		20
	50	0.20507	0.97875	0.20952	4.77286	1.02171	4.87649		10
12	0	0.20791	0.97815	0.21256	4.70463	1.02234	4.80973	78	0
	10	0.21076	0.97754	0.21560	4.63825	1.02298	4.74482		50
	20	0.21360	0.97692	0.21864	4.57363	1.02362	4.68167		40
	30	0.21644	0.97630	0.22169	4.51071	1.02428	4.62023		30
	40	0.21928	0.97566	0.22475	4.44942	1.02494	4.56041		20
	50	0.22212	0.97502	0.22781	4.38969	1.02562	4.50216		10
13	0	0.22495	0.97437	0.23087	4.33148	1.02630	4.44541	77	0
	10	0.22778	0.97371	0.23393	4.27471	1.02700	4.39012		50
	20	0.23062	0.97304	0.23700	4.21933	1.02770	4.33622		40
	30	0.23345	0.97237	0.24008	4.16530	1.02842	4.28366		30
	40	0.23627	0.97169	0.24316	4.11256	1.02914	4.23239		20
	50	0.23910	0.97100	0.24624	4.06107	1.02987	4.18238		10
14	0	0.24192	0.97030	0.24933	4.01078	1.03061	4.13357	76	0
	10	0.24474	0.96959	0.25242	3.96165	1.03137	4.08591		50
	20	0.24756	0.96887	0.25552	3.91364	1.03213	4.03938		40
	30	0.25038	0.96815	0.25862	3.86671	1.03290	3.99393		30
	40	0.25320	0.96742	0.26172	3.82083	1.03368	3.94952		20
	50	0.25601	0.96667	0.26483	3.77595	1.03447	3.90613		10
15	0	0.25882	0.96593	0.26795	3.73205	1.03528	3.86370	75	0
	10	0.26163	0.96517	0.27107	3.68909	1.03609	3.82223		50
	20	0.26443	0.96440	0.27419	3.64705	1.03691	3.78166		40
	30	0.26724	0.96363	0.27732	3.60588	1.03774	3.74198		30
	40	0.27004	0.96285	0.28046	3.56557	1.03858	3.70315		20
	50	0.27284	0.96206	0.28360	3.52609	1.03944	3.66515		10
16	0	0.27564	0.96126	0.28675	3.48741	1.04030	3.62796	74	0
	10	0.27843	0.96046	0.28990	3.44951	1.04117	3.59154		50
	20	0.28123	0.95964	0.29305	3.41236	1.04206	3.55587		40
	30	0.28402	0.95882	0.29621	3.37594	1.04295	3.52094		30
	40	0.28680	0.95799	0.29938	3.34023	1.04385	3.48671		20
	50	0.28959	0.95715	0.30255	3.30521	1.04477	3.45317		10
17	0	0.29237	0.95630	0.30573	3.27085	1.04569	3.42030	73	0
	10	0.29515	0.95545	0.30891	3.23714	1.04663	3.38808		50
	20	0.29793	0.95459	0.31210	3.20406	1.04757	3.35649		40
	30	0.30071	0.95372	0.31530	3.17159	1.04853	3.32551		30
	40	0.30348	0.95284	0.31850	3.13972	1.04950	3.29512		20
	50	0.30625	0.95195	0.32171	3.10842	1.05047	3.26531		10
18	0	0.30902	0.95106	0.32492	3.07768	1.05146	3.23607	72	0
	10	0.31178	0.95015	0.32814	3.04749	1.05246	3.20737		50
	20	0.31454	0.94924	0.33136	3.01783	1.05347	3.17920		40
	30	0.31730	0.94832	0.33460	2.98869	1.05449	3.15155		30
	40	0.32006	0.94740	0.33783	2.96004	1.05552	3.12440		20
	50	0.32282	0.94646	0.34108	2.93189	1.05657	3.09774		10
19	0	0.32557	0.94552	0.34433	2.90421	1.05762	3.07155	71	0
	10	0.32832	0.94457	0.34758	2.87700	1.05869	3.04584		50
	20	0.33106	0.94361	0.35085	2.85023	1.05976	3.02057		40
	30	0.33381	0.94264	0.35412	2.82391	1.06085	2.99574		30
	40	0.33655	0.94167	0.35740	2.79802	1.06195	2.97135		20
	50	0.33929	0.94068	0.36068	2.77254	1.06306	2.94737		10
20	0	0.34202	0.93969	0.36397	2.74748	1.06418	2.92380	70	0
	10	0.34475	0.93869	0.36727	2.72281	1.06531	2.90063		50
	20	0.34748	0.93769	0.37057	2.69853	1.06645	2.87785		40
	30	0.35021	0.93667	0.37388	2.67462	1.06761	2.85545		30
	40	0.35293	0.93565	0.37720	2.65109	1.06878	2.83342		20
	50	0.35565	0.93462	0.38053	2.62791	1.06995	2.81175		10
21	0	0.35837	0.93358	0.38386	2.60509	1.07115	2.79043	69	0
	10	0.36108	0.93253	0.38721	2.58261	1.07235	2.76945		50
	20	0.36379	0.93148	0.39055	2.56046	1.07356	2.74881		40
	30	0.36650	0.93042	0.39391	2.53865	1.07479	2.72850		30
	40	0.36921	0.92935	0.39727	2.51715	1.07602	2.70851		20
	50	0.37191	0.92827	0.40065	2.49597	1.07727	2.68884		10
22	0	0.37461	0.92718	0.40403	2.47509	1.07853	2.66947	68	0
	10	0.37730	0.92609	0.40741	2.45451	1.07981	2.65040		50
	20	0.37999	0.92499	0.41081	2.43422	1.08109	2.63162		40
	30	0.38268	0.92388	0.41421	2.41421	1.08239	2.61313		30
	40	0.38537	0.92276	0.41763	2.39449	1.08370	2.59491		20
	50	0.38805	0.92164	0.42105	2.37504	1.08503	2.57698		10
Deg.	Min.	Cos	Sine	Cotan	Tan	Cosec	Sec	Deg.	Min.

Table 39.—Natural Functions for Gear Calculations.—(*Continued*)

Deg.	Min.	Sine	Cosine	Tangent	Cotangent	Secant	Cosecant	Deg.	Min.
34	30	0.56641	0.82413	0.68728	1.45501	1.21341	1.76552		30
	40	0.56880	0.82248	0.69157	1.44598	1.21584	1.75808		20
	50	0.57119	0.82082	0.69588	1.43703	1.21830	1.75073		10
35	0	0.57358	0.81915	0.70021	1.42815	1.22077	1.74345	55	0
	10	0.57596	0.81748	0.70455	1.41934	1.22327	1.73624		50
	20	0.57833	0.81580	0.70891	1.41061	1.22579	1.72911		40
	30	0.58070	0.81412	0.71329	1.40195	1.22833	1.72205		30
	40	0.58307	0.81242	0.71769	1.39336	1.23089	1.71506		20
	50	0.58543	0.81072	0.72211	1.38484	1.23347	1.70815		10
36	0	0.58779	0.80902	0.72654	1.37638	1.23607	1.70130	54	0
	10	0.59014	0.80730	0.73100	1.36800	1.23869	1.69452		50
	20	0.59248	0.80558	0.73547	1.35968	1.24134	1.68782		40
	30	0.59482	0.80386	0.73996	1.35142	1.24400	1.68117		30
	40	0.59716	0.80212	0.74447	1.34323	1.24669	1.67460		20
	50	0.59949	0.80038	0.94900	1.33511	1.24940	1.66809		10
37	0	0.60182	0.79864	0.75355	1.32704	1.25214	1.66164	53	0
	10	0.60414	0.79688	0.75812	1.31904	1.25489	1.65526		50
	20	0.60645	0.79512	0.76272	1.31110	1.25767	1.64894		40
	30	0.60876	0.79335	0.76733	1.30323	1.26047	1.64268		30
	40	0.61107	0.79158	0.77196	1.29541	1.26330	1.63648		20
	50	0.61337	0.78980	0.77661	1.28764	1.26615	1.63035		10
38	0	0.61566	0.78801	0.78129	1.27994	1.26902	1.62427	52	0
	10	0.61795	0.78622	0.78598	1.27230	1.27191	1.61825		50
	20	0.62024	0.78442	0.79070	1.26471	1.27483	1.61229		40
	30	0.62251	0.78261	0.79544	1.25717	1.27778	1.60639		30
	40	0.62479	0.78079	0.80020	1.24969	1.28075	1.60054		20
	50	0.62706	0.77897	0.80498	1.24227	1.28374	1.59475		10
39	0	0.62932	0.77715	0.80978	1.23490	1.28676	1.58902	51	0
	10	0.63158	0.77531	0.81461	1.22758	1.28980	1.58333		50
	20	0.63383	0.77347	0.81946	1.22031	1.29287	1.57771		40
	30	0.63608	0.77162	0.82434	1.21310	1.29597	1.57213		30
	40	0.63832	0.76977	0.82923	1.20593	1.29909	1.56661		20
	50	0.64056	0.76791	0.83415	1.19882	1.30223	1.56114		10
40	0	0.64279	0.76604	0.83910	1.19175	1.30541	1.55572	50	0
	10	0.64501	0.76417	0.84407	1.18474	1.30861	1.55036		50
	20	0.64723	0.76229	0.84906	1.17777	1.31183	1.54504		40
	30	0.64945	0.76041	0.85408	1.17085	1.31509	1.53977		30
	40	0.65166	0.75851	0.85912	1.16398	1.31837	1.53455		20
	50	0.65386	0.75661	0.86419	1.15715	1.32168	1.52938		10
41	0	0.65606	0.75471	0.86929	1.15037	1.32501	1.52425	49	0
	10	0.65825	0.75280	0.87441	1.14363	1.32838	1.51918		50
	20	0.66044	0.75088	0.87955	1.13694	1.33177	1.51415		40
	30	0.66262	0.74896	0.88473	1.13029	1.33519	1.50916		30
	40	0.66480	0.74703	0.88992	1.12369	1.33864	1.50422		20
	50	0.66697	0.74509	0.89515	1.11713	1.34212	1.49933		10
42	0	0.66913	0.74314	0.90040	1.11061	1.34563	1.49448	48	0
	10	0.67129	0.74120	0.90569	1.10414	1.34917	1.48967		50
	20	0.67344	0.73924	0.91099	1.09770	1.35274	1.48491		40
	30	0.67559	0.73728	0.91633	1.09131	1.35634	1.48019		30
	40	0.67773	0.73531	0.92170	1.08496	1.35997	1.47551		20
	50	0.67987	0.73333	0.92709	1.07864	1.36363	1.47087		10
43	0	0.68200	0.73135	0.93252	1.07237	1.36733	1.46628	47	0
	10	0.68412	0.72937	0.93797	1.06613	1.37105	1.46173		50
	20	0.68624	0.72737	0.94345	1.05994	1.37481	1.45721		40
	30	0.68835	0.72537	0.94896	1.05378	1.37860	1.45274		30
	40	0.69046	0.72337	0.95451	1.04766	1.38242	1.44831		20
	50	0.69256	0.72136	0.96008	1.04158	1.38628	1.44391		10
44	0	0.69466	0.71934	0.96569	1.03553	1.39016	1.43956	46	0
	10	0.69675	0.71732	0.97133	1.02952	1.39409	1.43524		50
	20	0.69883	0.71529	0.97700	1.02355	1.39804	1.43096		40
	30	0.70091	0.71325	0.98270	1.01761	1.40203	1.42672		30
	40	0.70298	0.71121	0.98843	1.01170	1.40606	1.42251		20
	50	0.70505	0.70916	0.99420	1.00583	1.41012	1.41835		10
45	0	0.70711	0.70711	1.00000	1.00000	1.41421	1.41421	45	0
Deg.	Min.	Cos	Sine	Cotan	Tan	Cosec	Sec	Deg.	Min.

Circumference is assumed to be divided into 360 equal parts, termed degrees, written °. Each degree is divided into 60 parts, termed minutes, written ′. Each minute is divided into 60 parts, termed seconds, written ″.

Degrees° Minutes ′ Seconds ″

Example: 45° 20′ 11″ reads: Forty-five degrees, twenty minutes and eleven seconds.

For gearing it is seldom necessary to figure closer than degrees and minutes.

For convenience in arranging these tables, they contain sines, cosines, etc., for every 10 min. Sines, etc., for every minute may be easily obtained as follows:

Required sine of 26° 34′.

Refer to tables, select sine of nearest angle greater than angle given, also the nearest angle less than angle given.

Subtract sine of lesser angle from sine of greater; difference will be sine for 10′. Divide this by ten, which equals sine of 1′. Multiply this by number of minutes over nearest lesser angle; and add the result to the nearest lesser angle.

Example.—Sine 26° 40′.. .... 0.4488
Sine 26° 30′... .... 0.4462
0 0026 ÷ 10 = 0 00026 = sine 1′.

0 00026 × 4 = 0 00104 = sine of 4′.
26° 30′.... 0 4462
0 4′... 0.00104

0.44724 = sine 26° 34′.

Accuracy of Helical Gears.—In order to determine some of the causes for inaccuracies in the cutting of helical gears the Barber-Colman Co. made an extensive series of tests by cutting a series of gears and then checking all the inaccuracies. A summary of the conclusions reached as a result of these tests is of interest to all having to do with gears of this type. The conclusions are:

1. That "centering" of the hob tooth is a difficult operation and of no proven value.
2. That index worm and worm gear looseness shows up in the work by a slicing off of the tooth on the point or outer end.
3. That hob runout of 0.001 in. is highly detrimental to accuracy of tooth form if parallel with axis of hob spindle and even

more so if runout is in the form of a wobble, both conditions being commonly met with.

4. That a poorly finished or inaccurate hob cannot produce good work.

5. That a good hob poorly sharpened is also spoiled for best work.

6. That whether the gear material is steel or cast iron has little effect on the accuracy.

7. That there is not much difference between accuracy of one-cut or two-cut jobs under proper conditions.

8. That change-gear errors exert a very minor and almost negligible effect on the final results.

HERRINGBONE GEARS

Herringbone gears are becoming increasingly popular where it is necessary to transmit heavy loads with a minimum of noise and side or end thrust. The thrust is balanced by the opposite angles. They are double helical gears in which the helix angles are opposite on the two side of the gear. They are sometimes made of two opposite-angle helical gears fastened together. When solid herringbone gears are milled or hobbed with a standard cutter, it is necessary to leave a central groove of sufficient width to allow the cutter to run out in the center. This width can sometimes be reduced by staggering the teeth on the two sides so that the cutter can run out into the space of the other side.

Herringbone gears are sometimes cut with end mills of the proper contour. In such a case the central clearance groove can be made much narrower, and some makers pass the cutter from one side to the other leaving slight curve where teeth of the two sides join.

With the Sykes machine, shown in some detail in Figs. 73 and 74, the teeth on both sides of the gear are cut at the same time, without the necessity of a clearance groove. This machine is a gear shaper or planer with horizontal tool rams that are so timed as not to interfere in the center of the teeth.

The American Gear Manufacturers Association has adopted the following practice for use with herringbone gears:

Diametral pitch and tooth proportion shall be known, referred to, and calculated on the basis of diametral pitch in plane of rotation. For instance, if the teeth are cut with a standard spur-gear hob, the resulting fractional diametral pitch and pressure angle in plane of rotation shall be referred to and quoted without any reference to tooth proportions on the normal section.

In the following:

A = Addendum.
AF = Active face.
B = Backlash.
C = Clearance.
CDi = Center distance.
D = Dedendum.
DP = Diametral pitch in plane of rotation.
GD = Groove depth.
GW = Groove width.
N = Number of teeth.
OD = Outside diameter.
PD = Pitch diameter.
FW = Face width.
TL = Tooth load.
VP = Pressure angle in plane of rotation.
VH = Helix angle.
WD = Whole depth.

PD (pitch diameter) = N/DP

VP (pressure angle) in plane of rotation = maximum 25 deg.
= minimum 15 deg. 23 min.

NOTE: The maximum VP in combination with the maximum VH will result in a pressure angle of approximately 18 deg. 15 min. on the normal. The minimum VP given is the approximate resultant from 14½-deg. pressure angle on the normal with the minimum VH.

VH (helical angle) = maximum 45 deg.
= minimum 20 deg.
A (addendum) = maximum $1/DP$
= minimum $0.7/DP$
C (clearance) = maximum $0.3/DP$
= minimum $0.157/DP$

NOTE: If the space bottom (a line joining two fillets of adjacent tooth profiles in the same plane) is a curved-concave recess tangent to the fillets of the adjacent tooth profiles, the clearance C shall be increased by an amount equal to the depth of the recess.

D (dedendum) = $A + C$
WD (whole depth) = $2A + C$

$$OD \text{ (outside diameter)} = \frac{N}{DP} + 2A$$

Note: If the pressure angle (VP) and helical angle (VH) are known and the pressure angle of the hob or of tooth profile normal to helical angle is desired, then the tangent of that angle is equal to the tangent of VP multiplied by the cosine of VH. Likewise, if the pressure angle of the hob and helical angle (VH) are known and the pressure angle (VP) in plane of rotation is desired, then the tangent of that angle is equal to the tangent of the hob pressure angle divided by the cosine of VH.

If diametral pitch (DP) and helical angle (VH) are known, the diametral pitch of the tooth normal to the helical angle or that of the hob is equal to DP divided by the cosine of VH. Likewise, if the diametral pitch of the hob and VH are known, then DP equals the diametral pitch of the hob multiplied by the cosine of VH.

If DP and the diametral pitch of the hob are fixed, then the cosine of VH equals DP divided by the diametral pitch of the hob.

AF (active face) shall be equal to the sum of the right- and left-hand helices measured parallel to the axis of the gear.

$$\text{Minimum } AF \text{ (active face)} = \frac{1.15 \times 2\pi}{DP \times \tan VH}$$

FW (face width) shall be equal to AF plus space between helices.

GW (groove width) if any, between helices, shall be the minimum permissible on type of cutting machine and diameter of hob used.

GD (groove depth) if any, between helices shall as a maximum equal WD plus 1/32, except in case of middle bearing.

If it is desired to eliminate the undercutting of the hob, the outside diameter of the pinion should be increased (no increase in WD) an amount equal to

$$2D - \frac{(\text{sine } VP)^2 \times N}{DP}$$

If it is desired to simply obtain full involute action over the working depth of teeth between pinion and its mating gear, the outside diameter of the pinion should be increased (no increase in WD) an amount equal to

$$2A - \frac{(\text{sine } VP)^2 \times N}{DP}$$

If the outside diameter of the pinion is increased and the theoretical center distance (CDi) is maintained, then the outside diameter of the mating gear must be reduced (no decrease in WD) a similar amount, but if the center distance can be increased, then increase it by one-half the increase of the pinion and make no reduction in the gear diameters.

Backlash (B) is the shortest distance between nondriving tooth surfaces of adjacent teeth in mating gears.

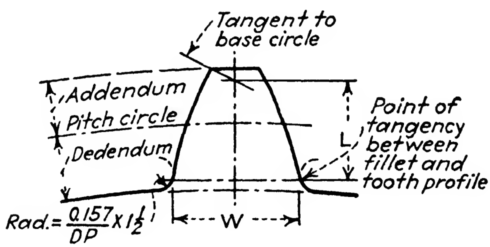
TABLE 40.—BACKLASH OF HELICAL GEARS

Minimum industrial gears		Minimum high-speed gears
DP	*B*, in.	*B*, in.
24	0.002	0.003
16	0.002	0.003
12	0.003	0.004
10	0.003	0.004
8	0.004	0.005
6	0.005	0.007
5	0.006	0.008
4	0.008	0.010
3	0.010	0.013
2½	0.012	
2	0.015	
1½	0.020	
1	0.030	

The ends of the teeth of each helix shall not be beveled except to remove cutting burrs.

Maximum *TL* (tooth load) per inch *AF* (active face) =

$$\frac{Y \times S \times K}{DP \times P}.$$



Section in plane of rotation.

$$Y = \text{tooth proportion factor for 1 } DP = \frac{W^2}{6L}$$

$$S = \text{allowable static stress of material}$$

Material.

High-carbon or alloy steels heat-treated to an elastic limit of approx. 60,000 lb. per square inch.........	15,000
0.40 to 0.50 carbon steel heat-treated to an elastic limit of approx. 50,000 lb. per square inch.............	12,500

0.40 to 0.50 carbon steel untreated with an elastic limit of approx. 40,000 lb. per square inch	10,000
Cast steel A. S. T. M. Class B, elastic limit approx. 36,000 lb. per square inch	7,500
Cast iron, tensile strength approx. 24,000 lb. per square inch	4,000
Bronze 88–10–2, tensile strength approx. 27,000 lb. per square inch	4,000

NOTE: It should be noted that the teeth of herringbone gears are cut after the material is treated. Higher physical characteristics than those quoted above for steels may result in material too hard to cut.

$$K = \text{velocity factor in feet per minute} = \frac{78}{78 + \sqrt{\text{pitch-line velocity}}}$$

P = wear and lubrication factor.

For enclosed gears in which the viscosity and character of the lubricant are correctly chosen for the type of gear and service a P factor of 1.15 is recommended.

NOTE: The above formula gives allowable tooth loads for uniform and smooth operation but if operating conditions are such that either the driving or driven elements cause uneven or intermittent loads, then the tooth load should be reduced by a utility factor which can be determined only by actual experience in the particular class of service.

HOBBING HELICAL GEARS USING DIFFERENTIAL[1]

Lead Gears.—The lead gears are calculated from the formula shown on page 128. Substitute the proper *values in the formula and reduce* to a decimal. From a table of Decimal Equivalents of Fractions select the fraction whose decimal equivalent is nearest the desired value.

Assume that the formula reduces to the decimal 0.244 (slide rule accuracy is satisfactory). The table shows that 21/86 = 0.2441860.

Four gears are required properly to mesh the lead gears and 21/86 must be expanded to four numbers suitable for use as change gears. Multiply both numerator and denominator by 40,

$$\frac{21}{86} = \frac{21 \times 40}{86 \times 40}$$

and then multiply the original numerator and the new denominator by 2, which gives

$$\frac{42 \times 40}{86 \times 80} = \frac{\text{drivers}}{\text{driven}}$$

[1] Granger Davenport, Research Engineer, Gould & Eberhardt.

Gears are placed on the machine thus:

Lead Driver	Compound	Driven
A—42 meshes with	*B*—86	
	C—40 meshes with	*D*—80

The gears can be variously arranged as shown under Gearing Calculations that follow.

When both the right- and left-hand gears for parallel shafts are hobbed on the same machine or with the same lead gears, the foregoing method of figuring the lead gears is satisfactory.

However, if the gears are to be mounted on nonparallel shafts, or if the mating gears are to be cut on another machine with different lead gearing, or if for any reason it is desired to hold the lead exact, then the formula should be calculated to five decimal places and a fraction calculated whose decimal equivalent does not vary more than 2 in the fifth decimal place. In this case, the same formula is used, but the proper gearing for the fraction is calculated as given in Gearing Calculations, page 141*c*.

Before cutting gears differentially, check index and lead ratios so as not to exceed the capacity of the machine and to avoid damage to differential parts. Be sure that the differential mechanism is engaged before starting to hob. A lever located in the change-gear compartment directly below the index change gears must be pushed in to be engaged. Use the safety latch to lock this lever in position. If a second or finishing cut is to be taken, withdraw the hob from the work at the end of the first cut, engage the rapid traverse upward until hob clears the work, then adjust stanchion to full depth. It will not be necessary to realign the hob with the work when cutting helical gears as the lead is maintained by the differential mechanism. The feed can then be engaged for the second cut.

HOBBING HELICAL GEARS WITHOUT USING DIFFERENTIAL

When hobbing helical gears without using the differential, lock the feed lever with the thumbscrew so that the feed will not become disengaged during the cut and thus lose the lead. If the machine is equipped with a differential mechanism, be sure that it is disengaged before starting to hob. Use the safety latch to lock the lever in position.

Index Gears.—Refer to the chart on page 205 and, using the formula, calculate the value of the feed constant C to the nearest whole number. Substitute this value of C in the index formula for helical gears and reduce to a fraction. Note that when using a hob having the same hand as the gear, the 1 in the denominator is subtracted. If the hands of the gear and hob are opposite, the 1 is added. Opposite hand hobs can be used but when the helix angle is high, most satisfactory results are obtained using hobs of the same hand as the gear to be cut.

The ratio of the index gears must be exact. Under no circumstances should this be reduced to a decimal as no approximation can be used. If by using this value of C, either the numerator or denominator results in a prime number greater than 97, change the value of C slightly and recalculate, because change gears having prime numbers of teeth greater than 97 are not furnished.

Changing the value of C has no effect on the accuracy of the lead. It merely changes the feed slightly. A higher value of C reduces the feed and a lower value increases it. A difference of 1 or 2 per cent in the value of C makes only a slight difference in the amount of feed from that selected. Use fractional values of C if necessary.

Assuming the ratio obtained is 120/575, factor as follows:

$$\frac{120}{575} = \frac{5 \times 24}{25 \times 23}$$

To obtain factors for which suitable change gears can be selected, multiply both numerator and denominator by 6. This does not change the value of the fraction.

$$\frac{5 \times 24}{25 \times 23} \times \frac{6}{6} \quad \text{or} \quad \frac{5 \times 24}{25 \times 23} \times \frac{6}{2 \times 3} = \frac{5 \times 6 \times 24}{25 \times 3 \times 23 \times 2}$$

$$= \frac{30 \times 24}{75 \times 46}$$

Because of the low number of teeth, the 24-tooth gear is not large enough to mesh with the 46-tooth gear and therefore, to make centers on the machine, multiply both numerator and denominator of second fraction by 2.

$$\frac{30 \times 24}{75 \times 46} \times \frac{2}{2} = \frac{30 \times 48}{75 \times 92}$$

Index gears are placed on the machine thus:

Index Driver	Compound	Driven
A—30 meshes with *B*—75		
	C—48 meshes with *D*—92	

These gears, of course, can be variously arranged as shown in the example under Gearing Calculations.

Feed Gears. —To obtain the proper feed gears, substitute in the feed gear formula for helical gears the same value of C used in the index formula and reduce the equation to a decimal. Gearing Calculations show how to obtain the gearing from this decimal.

GEARING CALCULATIONS

Feed change gears for hobbing helical gears nondifferentially, and *lead* change gears for hobbing helical gears differentially under certain conditions, are accurately calculated as follows:

Substitute the proper values in the feed or lead formula, whichever the case may be, and using a calculating machine reduce to a decimal carried out to at least the fifth decimal place. Values used in the formula must also be accurate to the fifth decimal place.

Assume that the equation reduces to the decimal 0.797576. The nearest fraction to this number is

$$\frac{67}{84} = 0.797619$$

Since this decimal differs from the formula decimal by 4 in the fifth place, obtain a more accurate fraction by dropping the first figure (*i.e.* 7) of 0.797576 and look up a fraction equivalent to 0.97576, which is

$$\frac{40}{41} = 0.975609$$

With the 7 originally dropped, consider the mixed number $0.7\frac{40}{41}$ and reduce to a proper fraction.

$$0.7\frac{40}{41} = \frac{7\frac{40}{41}}{10} = \frac{\frac{(7 \times 41 + 40)}{41}}{10} = \frac{(7 \times 41) + 40}{41 \times 10} = \frac{327}{410}$$

$$= 0.797560$$

For even greater accuracy proceed as follows:

Note that the decimal whose fraction is required (*i.e.* 0.797576) lies between 63/79 and 67/84. Add these two fractions numerator to numerator and denominator to denominator thus:

$$\frac{63 + 67}{79 + 84} = \frac{130}{163} = 0.797546$$

Decimal required now lies between 130/163 and 67/84, and adding as before:

$$\frac{130 + 67}{163 + 84} = \frac{197}{247} = 0.797570$$

But 197 is prime and cannot be factored. A 197 tooth change gear would not be practical to use nor would it fit on the machine; hence this fraction cannot be used.

Therefore, continue adding fractions between which the required decimal lies.

$$\frac{197 + 67}{247 + 84} = \frac{264}{331} = 0.797583, \text{ but 331 is prime.}$$

$$\frac{264 + 197}{331 + 247} = \frac{461}{578} = 0.7975778, \text{ but 461 is prime.}$$

$$\frac{461 + 197}{578 + 247} = \frac{658}{825} = 0.797575$$

The last decimal is correct to the fifth decimal place and the fraction is factorable. Factor as follows:

$$\frac{658}{825} = \frac{2 \times 329}{25 \times 33} = \frac{2 \times 7 \times 47}{25 \times 33} = \frac{14 \times 47}{25 \times 33}$$

Factors for 329 were taken from a book of composite numbers and factors.

To obtain factors for which suitable change gears can be selected, multiply both numerator and denominator of the first fraction by 2. This does not change the value of the fraction.

$$\frac{14 \times 47}{25 \times 33} \times \frac{2}{2} = \frac{28 \times 47}{50 \times 33}$$

To make centers on the machine the gears should be larger, so multiply the first numerator and the second denominator by 2 again.

$$\frac{28 \times 47}{50 \times 33} \times \frac{2}{2} = \frac{56 \times 47}{50 \times 66}, \text{ which is satisfactory.}$$

Gears are placed on machine thus:

Driver		Compound		Driven
56	meshes with	50		
		47	meshes with	66

The gears can be variously arranged for convenient mounting as follows:

Driver		Compound		Driven
56	meshes with	66		
		47	meshes with	50
		or		
47	meshes with	66		
		56	meshes with	50
		or		
47	meshes with	50		
		56	meshes with	66

but note that in all cases, the numerator numbers 56 and 47 must be the drivers and the denominator numbers 66 and 50 must be the driven.

If the gears cannot be made to mesh even by rearranging, multiply or divide the numerator and denominator by the same number until a suitable set of gears is obtained. Multiplying or dividing both numerator and denominator by the same number will not change the gearing ratio. For instance, in this case note that 56/50 = 28/25 and, if these gears are too small, multiply both numerator and denominator by 3,

$$\frac{28}{25} \times \frac{3}{3} = \frac{84}{75}$$

and use the gearing

$$\frac{84 \times 47}{75 \times 66}$$

In making the foregoing computations, a calculating machine is quicker and less susceptible to error than longhand or logarithmic methods.

CHAPTER V

HOBS AND CUTTERS

Terms Used by Makers.—It is important to know the names of machine parts if confusion is to be avoided, and this seems particularly true of hobs. A single, right-hand-thread ground hob and the terms used in describing the various parts are shown

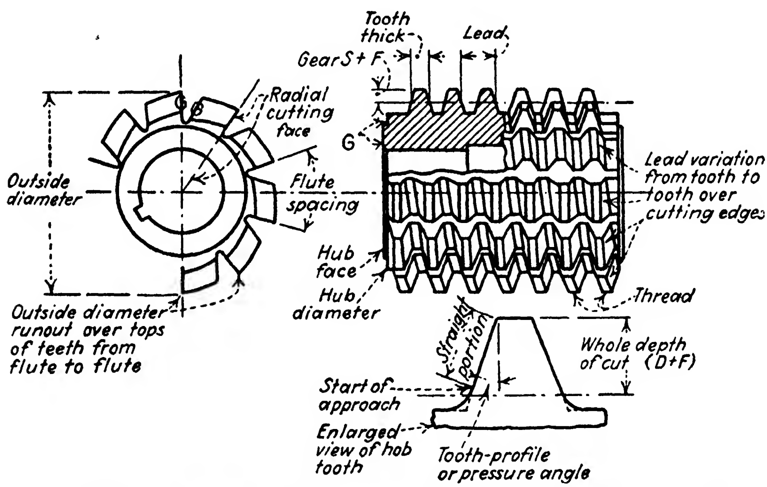


FIG. 78.—Terms used for parts of single, right-hand-thread ground hob.

in detail in Fig. 78. It will pay to study these names carefully, and use them in ordering hobs.

Standard $14\frac{1}{2}$-deg. pressure-angle hobs can be used for both spur and helical gears. Right-hand hobs can cut both right- and left-hand gears except when the helix angle is greater than 25 or 30 deg. In such cases it is best to use hobs of the same "hand" as the gear to be cut. For gears with a greater helix angle than 35 deg. the hob should be tapered on one end to reduce the strain when entering the cut. A right-hand hob will be tapered on the left end, a left-hand hob on the right end, as viewed with the teeth coming from the top of the hob. Standard spur- and helical-gear hobs have straight gashes where the thread angle is less than 4 deg. With greater thread angles the hobs are made with helical gashes.

Hobs for Gear Cutting.—Hobs form one of the large items of cost in gear cutting, as do the tools in other gear-generating methods. They can perhaps be compared to broaches in this respect, and both are possible of effecting great economies when properly used. Greatest economy involves careful consideration of the diameter, length, ground or unground teeth, number of leads best suited to work and machine, number of gashes in hob, best clearance angles for top, sides, and faces of teeth, and the best material and heat treatment for the hob. As many of these are points about which the user can hardly know as much as the maker of hobs, it is best to select a maker with long experience and good reputation, and consult with him before deciding on all hob details.

Hobs should be kept in the best of condition, both as to care in the tool crib and sharpening. It is advisable to sharpen often rather than to see how many gears can be cut between grinds. A hob with teeth burned from heat in grinding or with the teeth "dubbed off," or with gashes improperly spaced, cannot give good service.

Economical speed of the hob is important. It should generally be kept as high as possible considering the other factors which affect economy. Among these are the feed per revolution, the material being cut, the size of the tooth and the coolant used. Both quality and volume of the coolant are important. Likewise the rigidity and power of the machine itself have their effect.

Good production depends largely on a balancing of speeds and feeds. Too much speed will burn and abrade the corners of the teeth of the hob and too low a speed with heavy feed may break off the teeth, particularly if they have been ground back by sharpening or where there are too many gashes for good proportion and strength.

Large users of hobs frequently divide them into two classes: those which are new and those which are about half worn out. The half-worn hobs will give much more service if they are run with a smaller feed than a new hob, but they can sometimes stand a higher speed if the feed is lessened.

It is interesting to note that hobbing cuts metal in a way not found in other machining operations. There are three simultaneous motions, two being rotary and the other linear. This is because the hob and the work rotate together and the hob advances through the work at the same time. This combination gives a cutting action not found in other cutting tools.

One advantage of hobbing is the uniform heating of the work rather than its concentration where gears are cut by formed cutters. To avoid this it is customary to jump from one tooth to another some distance away, when formed cutters are used. This is sometimes known as block indexing.

Straight- or Taper-hole Hobs.—One of the problems in hobbing is to secure a true-running hob. Straight-hole hobs are usually made with a tolerance of $-0.0000 + 0.0025$ in., this tolerance being reversed in the spindle. With a minimum spindle and a maximum hob there can be a total runout of 0.005 in., unless great care is taken by the operator to true the hob on the spindle. And even if trued when the work is started, the hob can shift to an eccentric position, owing to vibration and stresses to which it is subjected. Unless the operator frequently rechecks the hob, this can easily pass unnoticed.

On this account the Barber-Colman Co. advises the use of a taper hole in the hob, their own practice being a taper of 2 in. per foot. With such a taper and ordinary care in seeing that the hole and spindle are clean, the total maximum runout should not exceed 0.0003 in., with the average very much below this figure.

Taking One or Two Cuts.—In cutting coarse-pitch gears, or where extreme accuracy is required, it is customary to take a roughing and a finishing cut. In either case the finishing cut is taken with the machine under little or no strain. Where only small quantities are being cut, some follow the practice of taking one cut with the hob set to slightly less than the full depth and then going around again at maximum depth.

Hobbing machines permit the hob to be shifted endwise to bring different teeth into action after it has become dull in one position. Hobs are usually made long enough so that several such settings can be made before it is necessary to remove the hob for sharpening.

The accuracy to which ground hobs are now made can be seen in Tables 41 and 42. These are given by the National Twist Drill & Tool Company.

TABLE 41.—MANUFACTURING TOLERANCES OF GEAR HOBS WITH GROUND FORM

Diam. pitch	Indexing of gashes		Runout		Cutting faces	Start of approach	
	Accumulated error	Error from tooth to tooth	Outside diam.	Hub diam. and face	Radial within	Located within + or −	Symmetrical within + or −
3	0.006	0.002	0.0015	0.0003	0.001	0.015	0.007
4	0.005	0.002	0.0015	0.0003	0.0007	0.015	0.007
5	0.004	0.0015	0.0015	0.00025	0.0007	0.015	0.007
6	0.003	0.001	0.001	0.00025	0.0007	0.010	0.005
7	0.003	0.001	0.001	0.00025	0.0007	0.010	0.005
8 and up	0.003	0.001	0.001	0.00025	0.0005	0.010	0.005

Hole diameter: $^{+0.00025}_{-0.0000}$, 75 per cent of each bearing.

TABLE 42.—MANUFACTURING TOLERANCES OF GEAR HOBS WITH GROUND FORM

Tooth thickness			Class of tolerance	Form variation		Lead variation		
Diam. pitch	Topping hobs	Non-topping hobs		Straight portion	Symmetrical within	Any one turn	Tooth to tooth	Any five turns
3	+0.000 −0.001	+0.000 −0.002	*A* *B*	0.0004 0.0005	0.0005 0.0006	0.0010 0.0015	0.0005 0.0007	0.0015 0.0022
4	+0.000 −0.001	+0.000 −0.002	*A* *B*	0.0003 0.0004	0.0004 0.0005	0.0008 0.0013	0.0004 0.0006	0.0012 0.0020
5	+0.000 −0.0005	+0.000 −0.002	*A* *B*	0.0002 0.0004	0.0003 0.0004	0.0007 0.0011	0.0003 0.0005	0.0009 0.0015
6	+0.000 −0.0005	+0.000 −0.002	*A* *B*	0.0002 0.0003	0.0002 0.0003	0.0006 0.0010	0.0003 0.0004	0.0008 0.0012
7 to 12	+0.000 −0.0005	+0.000 −0.001	*A* *B*	0.0002 0.0002	0.0002 0.0003	0.0005 0.0009	0.0002 0.0004	0.0007 0.0012
13 & up	+0.000 −0.0005	+0.000 −0.001	*A* *B*	0.0002 0.0002	0.0002 0.0003	0.0005 0.0008	0.0002 0.0003	0.0007 0.0010

NOTE: All hobs for finishing automobile passenger car transmission gears, timing gears, pump gears, and camshaft gears are *A* tolerance unless otherwise specified.

Positioning of Hobs.—When hobbing work that is relatively small in diameter, such as spline shafts, with a hob of standard length, only a small portion of the hob is in contact with the work. This is the only part of the hob that is subject to dulling. After this portion has been dulled the hob can be shifted to a new position and the teeth in that position dulled. This process can be repeated until all the teeth in the hob are dulled. Only then is it necessary to resharpen the hob. In this way the maximum hob life and tool economy can be realized. This is illustrated in Fig. 79.

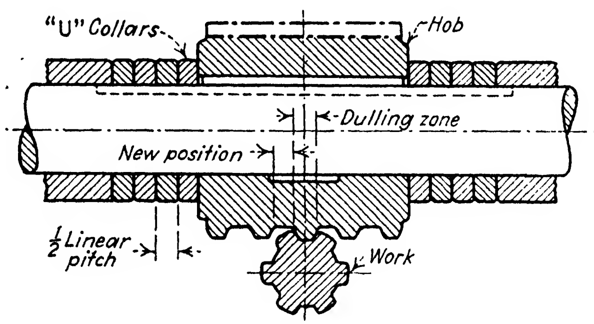


FIG. 79.—Positioning hobs to secure longest life.

In order to get the maximum number of positions possible in the length of a hob, a series of spacing collars should be used. The width of each collar should be equal to one-half the linear pitch of the hob. When multiple-thread hobs are used, the width of each spacing collar should be equal to the number of threads times one-half the linear pitch. The reason for these widths of spacing collars is that, when in operation, approximately one-half the teeth in a convolution are dulled in any one position. One convolution of a hob is equal to the linear pitch, and hence the widths of collars specified. One pitch collars are also used.

To facilitate the shifting of the hob from one position to the next, the spacing collars may be made U-shaped to fit snugly over the hob arbor. When changing positions, it is only necessary to loosen the arbor nut, to move one of the U collars from one end of the hob to the other and to retighten the nut. The hob is then ready for operation again. Gould & Eberhardt recommend solid collars instead of U-shaped.

Sharpening and Maintenance.—To produce accurate and otherwise satisfactory work by the hobbing process there are several factors that must be carefully considered. It is not only necessary to have hobs of the correct design and contour; but it is equally important that the hobs are maintained to the proper accuracy throughout their useful life.

For instance, if a hob is permitted to be run until it is excessively dulled, the contour of the work will be affected, because dulling does not occur uniformly over the cutting edges. The tendency is for the sharp corners or thin sections to become dull first, while the heavier portions of the teeth may be comparatively sharp. Further, it is poor economy to overdull a hob, because of the larger amount that must be ground away before it is sharp again.

If the helix angle of the teeth, or the rake angle, is altered during resharpening, the shape of the work is quickly affected. Too much attention cannot be paid to these angles. Most manufacturers make a practice of marking the angle of the teeth, as well as the rake angle (if any), on the hob itself, because they realize the importance of maintaining them when the hob is resharpened by the user.

The use of the correct grade and grain of grinding wheels is another important point. Even a slight amount of overheating of the cutting edges, when grinding, will quickly cause them to soften. The result is premature dulling and loss of accuracy. A soft, free-cutting, and not too coarse wheel should be used. Sufficient time must be allowed for the sharpening operation so that there is no danger of burning the cutting edges. The grinding face of the wheel should be dressed often, so as to maintain straight cutting faces on the hob teeth. Concave or convex cutting faces will produce inaccurate work.

Always keep in mind that a hob is one of the most accurate, as well as most expensive, cutting tools in the shop. Sharpening abuses not only are wasteful of the life of the hob, but lead to the production of unsatisfactory or rejected work.

Where hobs are to be fluted helically, both the lead and the angle of the flutes can be determined from the accompanying chart, Fig. 80, by K. Takahashi, Tokyo, Japan. No calculations are necessary, but, of course, the pitch diameter of the hob and the lead of its thread must be known. The data are almost always known, but a laborious calculation is often required before the lead and the angle of the flutes can be obtained. Increased

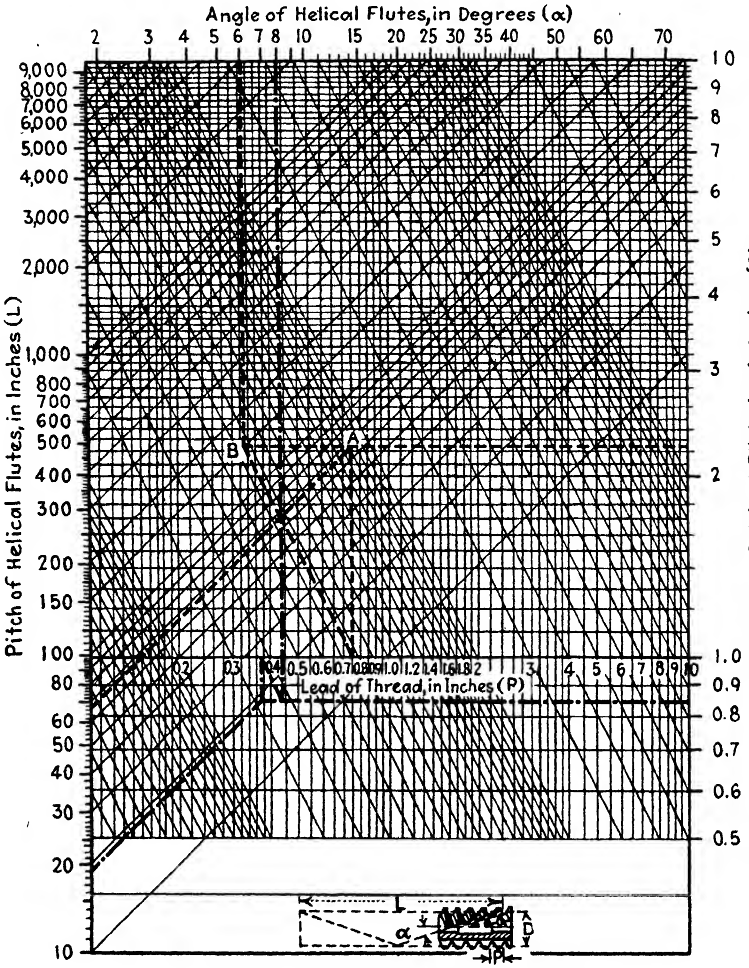


FIG. 80.—Lead and angle of hob flutes.

accuracy can, of course, be attained by increasing the size of the chart.

Example.—A hob having a lead of 0.75 in. and a pitch diameter of 2.25 in. is to be fluted helically so that the faces of its teeth will be at an angle of 90 deg. to the flutes at the pitch line. Find the pitch diameter of the hob at the right of the chart, and the lead of its thread on the horizontal line near the bottom. Draw lines from these points intersecting at *A*. From the intersection *A* draw a diagonal line downward, to the left, parallel with the lines of the chart inclining in that direction, to the left-hand edge of the chart. There it will be found that the lead of the helical flutes is 67 in. From 0.75, near the bottom, draw a diagonal line upward, to the left, parallel with the lines of the chart inclining in that direction, and continue the horizontal line from *A* to the left until these two lines intersect at *B*. From the intersection *B* draw a vertical line to the top of the chart. There it will be found that the axial angle of the flutes is 6 deg. This procedure is indicated by the dotted lines.

For a hob having a lead of 0.375 in. and a pitch diameter of 0.85 in., it will be found, by following the dash-dot lines, that the lead of the helical flutes is 19 in. and their axial angle is found to be 8 deg.

Worm-gear Hobs.[1]—If we go back even a comparatively few years, we find that the tooth shape of a worm thread was usually defined as 29-deg. included angle on the axial section. The use of a straight-sided tooth shape on this section was a natural result of the fact that chasing in a lathe was the almost universal method of production. The selection of the 29-deg. angle naturally followed from the fact that this angle was the established standard for nearly all types of gearing.

Because of changing methods of worm-thread cutting and increasing use of multiple-threaded worms, many of which introduce manufacturing difficulties both in the worm and in the corresponding hob if the 29-deg. angle is retained, this tooth shape has been largely superseded.

The American Gear Manufacturers Associations recommended practice indicates the cutting of single- and double-threaded worms with a 29-deg. thread-milling cutter and three- and four-thread worms with a cutter of 40-deg. included angle. This leaves worms having a larger number of threads without specific recommendations, and consequently, the rule for the selection of the cutter angle, which has been in use for some years and which covers all possible cases, may be of interest:

When the angle of thread with axis is 78 deg. or more, use 29-deg. cutters.

When the angle of thread with axis is 70 to 78 deg., use 40-deg. cutters.

[1] Presented by L. R. Mayo, Brown and Sharpe Manufacturing Company, to the American Gear Manufacturers Association, 1932.

When angle of thread with axis is 65 to 70 deg., use 45-deg. cutters.

When angle of thread with axis is 65 deg. or less use 50-deg. cutters.

Worms mating with wheels of 24 teeth or less should be finished with a cutter of 40 deg. or more included angle to avoid undercutting the wheel.

If the angle of worm thread with the axis is 72 deg. or more, make tooth depth equal 0.6866 × axial pitch. If less than 72 deg., make tooth depth equal 0.6866 × normal circular pitch.

It is well known that when cutting a worm thread, a rotary cutter does not reproduce its own shape. The use of an included-angle cutter having straight cutting edges results in the worm threads being formed by a convex curve. The amount of this departure varies greatly, being practically negligible on the ordinary single-thread worm but becoming a serious consideration on the high multiples.

The diameter of the cutter has an influence on the resulting thread shape, the difference between the cutter shape and the resulting space increasing with the cutter diameter. Consequently, the hob maker should know the approximate diameter of the cutter to be used in the manufacture of the worm.

In this connection, it might be observed that a worm milled with a straight-sided included-angle cutter is an approximation of an involute shape, this approximation being decreased as the cutter diameter increases. If the cutter were of infinite size, a true involute shape would result.

The statement is frequently heard that the difference between the cutter shape and the shape of the resulting worm "varies with the helix angle." That this is not strictly correct is evident when we consider that a worm of infinite diameter becomes a helical rack, in the cutting of which no cutter distortion can take place, even though the helix angle be large. It is more accurate to say that the worm's departure from the cutter shape varies with the difference between the helix angles found at the worm-root diameter and outside diameter.

It has been demonstrated in practice that many hobs can be satisfactorily made with straight-tooth shapes; in fact, the nature of this approximation is fortunately such as to ease off the tooth contact near the top and bottom of the worm-gear tooth. Up to a certain point, this is perhaps an advantage as it avoids interference and tends toward easy engagement. If carried too far, the amount of contact would be seriously reduced. In such cases, the hob tool is fitted to a dummy or master worm. The rules used to determine when the tool shall be fitted to a dummy are:

Fit hob tool to dummy worm when (*a*) angle of thread with axis is less than 78 deg., or (*b*) when the difference between the helix angles at the worm outside diameter and root diameter exceeds 5 deg.

In the design of worm gears, the ratio existing between the worm and gear is, at times, fixed. Frequently, however, an exact ratio is not required, and in such a case, the question arises as to which is more desirable—an even or a prime ratio. It is contended by some that even ratios are best because any given wheel tooth is then acted upon by only one thread of the worm and that, consequently, it soon conforms to the individual tooth shape and index peculiarities of that thread.

Using a modern worm having hardened and ground threads, it is possible that the even ratio might not show measurable advantages. From the standpoint of the gear manufacturer, however, the prime or "hunting" ratio is preferable as it automatically produces uniform gear teeth, regardless of the hob's design or condition.

In the case of an even ratio, a repeating error is difficult to avoid. This error is conveniently measured by means of a ball-point indicator applied to the gear spaces and contacting at approximately the pitch line. What is usually found is that the readings will repeat, the number of readings in each cycle being equal to the number of threads in the hob or worm.

Excluding tangential feed hobs for the present, it can be readily shown that these repeating errors are to be expected:

(*a*) If the hob contains any errors of either manufacture, sharpening, or mounting.

(*b*) If the number of gashes are not exactly divisible by the number of threads.

The reason for this last statement is illustrated by Fig. 81 at *A*, which shows in diagrammatic form a development of a hob having three threads and seven gashes. Straight gashes have been used to simplify the illustration.

It will be noted that the tooth C is the only one in the whole hob which is on center with respect to the gear being cut. Also, that in threads Nos. 2 and 3, the teeth marked E and F are nearest to this center line. By calculation, their distance is found to be 0.143 for the case in question, and further calculations show that when tooth C reaches the root diameter of the gear, teeth E and F will be 0.0041 away.

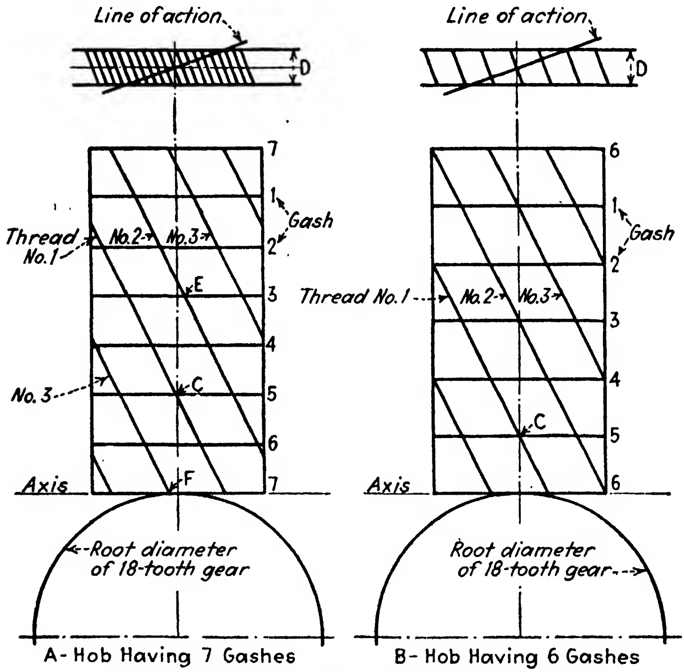


FIG. 81.—Effect of number of gashes in hob teeth.

In Fig. 81 B is like A in every way except that the number of gashes has been reduced to six. Under this condition, we find that the disposition of the teeth in each thread is identical; in other words, if a tooth in one thread is centered with the gear, a tooth in each of the other threads will also be centered. Theoretically, the use of such a hob made without errors and perfectly mounted will produce uniform spaces in the gear being cut even though the ratio between hob and gear is even.

At the top of both of these diagrams, all of the cutting teeth have been projected and are shown in relation to an assumed line of action. It will be noted that in diagram A there are 14 teeth within the line of action but only four in diagram B. This, of course, is because the hob at B contains three sets of duplicate teeth.

From the foregoing, it appears that, while making the number of gashes exactly divisible by the number of threads has resulted in uniformity of wheel teeth, the number of generating positions is small. This produces flats on the teeth which are frequently objectionable, particularly on coarse pitches and low numbers of teeth.

A method of overcoming this is sometimes referred to as "stepping"; that is, causing each thread of the hob to pass through each space in the wheel by successively feeding to depth, withdrawing, and indexing the wheel one tooth and again going to depth. The hob in this case should be made as illustrated at A—the number of threads and the number of gashes having no common factor. "Stepping," of course, takes some additional time and may require a better type of machine operator. It is, therefore, not suggested as a regular manufacturing procedure, but rather, as a means of improving the quality of the occasional job which may be giving trouble.

In our original example, a ratio of 3 to 18 was assumed. If, instead of 18 teeth, the gear had 17 or 19 teeth, then a hob made like A would be decidedly preferable because of the larger number of effective generating teeth. From these considerations the following rules, governing the selection of the number of gashes in multiple-thread worm-gear hobs, have been derived:

1. If the number of threads in the worm and the number of teeth in the wheel have no common factor, then the same condition should prevail between the number of threads in the hob and the number of gashes.

2. When the worm and gear have a common factor, or when the hob is to be used for general-purpose work, the number of gashes should be exactly divisible by the number of threads. An exception would be made if it were known that a hob user was willing to "step" his even ratios.

It has at times been found that a gear manufacturer, cutting an even ratio with a hob similar to that illustrated at A, can greatly reduce his difficulties by adding or subtracting one tooth from the gear. This, of course, is not always permissible, but when it is, the resulting improvement is frequently great. To do this, no changes in center distance or worm-gear-blank dimensions are required. Whether a tooth should be added or subtracted will depend upon the individual requirements of the job. It should be remembered, however, that if an additional tooth is put in a gear having a low number of teeth, or one whose

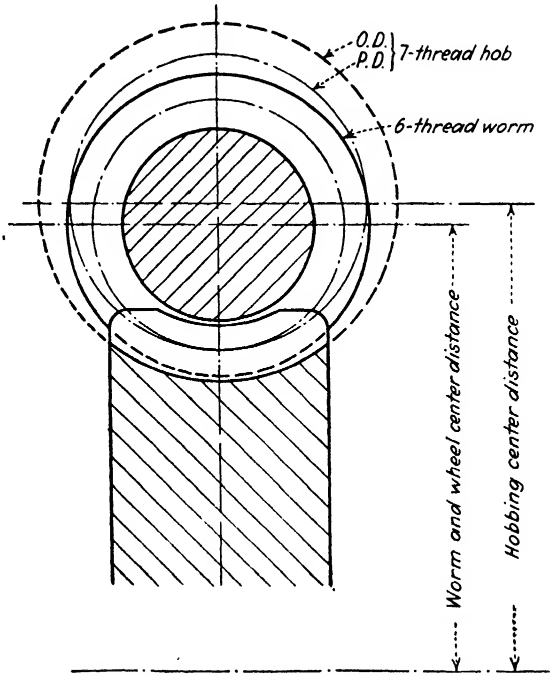


FIG. 82.—Contact of worm in worm wheel.

pressure angle is low, an objectionable amount of undercut may result.

An effective method of getting uniformity in the teeth of even-ratio worm gears is to add a thread to the hob. For example: If the worm had six threads and the gear 120 teeth, the hob would be made with seven threads. The pitch diameter and lead of the hob would be proportionately increased, thus retaining the original helix angle.

It is true that this results in a somewhat decreased initial contact between the worm and gear, but when the worm has five or more threads, the addition of one more in the hob is frequently not objectionable. Figure 82 shows such a case, the proportions being those of a six-threaded worm and a seven-threaded hob. Any suitable number of gashes could be used in this hob except seven or a multiple of seven.

If a regular- or radial-feed worm-gear hob could have an infinite number of cutting teeth, it would be theoretically possible to secure perfect initial contact between the worm and gear. Obviously, this is not possible, but this condition is approximated by the use of a taper or tangential-feed hob. While such hobs have about the same number of gashes as the regular type, the relatively slow tangential-feed motion causes each cutting edge to assume many positions as compared with the one which it would maintain in a radial-feed hob.

The use of the tangential-feed method is particularly beneficial when cutting even ratios as it does away with having to choose between "stepping" on the one hand and coarse generating flats on the other. To secure the full advantage of tangential hobs, they should be made with ground teeth. This is particularly desirable when cutting even ratios.

The number of gashes in a tangential hob are not important if an even ratio is being cut. If, however, the number of gashes is not exactly divisible by the number of threads, the various threads of the hob will begin cutting progressively, instead of simultaneously, the appearance of the gear being somewhat as illustrated at *A* in Fig. 83.

If the ratio is not even, the number of gashes should be exactly divisible by the number of threads; otherwise, one hob thread will remove more than its share of the stock from the gear and thus greatly decrease the rate of feed which could be used successfully. It is, therefore, the general practice to make the number of gashes of all tangential hobs exactly divisible by the number of threads. An exception would be made to this in a case where the hob was to cut an even ratio only if this were necessary to produce a well-proportioned length of hob tooth.

At *B* in Fig. 83, the method ordinarily employed in establishing the angle of taper is illustrated. If various-sized gears were to be cut with the same hob, the largest of these would be considered in establishing this angle. It will be noted that the amount of taper equals three-quarters of the tooth depth and that the taper is established tangent to the throat circle.

At C and D, two methods of tapering are illustrated, the hobs being identical except that the taper is applied at opposite ends. It is believed that the type of taper illustrated at C is used more frequently. No attempt is made to say which type is preferable, although an analysis of the thrust and backlash conditions shows that there is a tendency for hob D to jump forward to the extent of the backlash between the hob slide screw and nut.

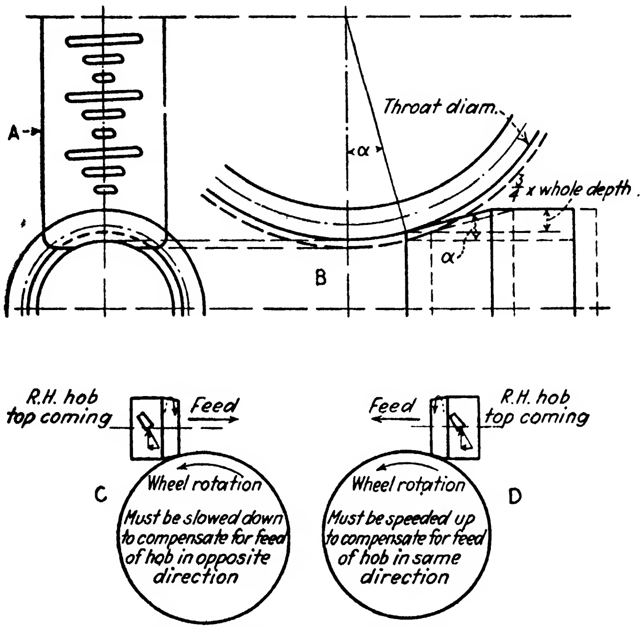


FIG. 83.—For tangential feeding the hob taper should be normal to the throat diameter of the blank and three-quarters of the tooth depth.

On the other hand, the action of the differential when using hob C is required to slow down the index rotation of the gear being cut. This condition appears to introduce the possibility of backlash. It is felt that the experience of users who may have used both types of tapers may determine conclusively which is to be preferred.

The length of the parallel section of a tangential hob seems to be debatable. Theoretically, a single finishing tooth in each thread is sufficient. From a standpoint of actual practice, it is felt to be safer to increase this number. A minimum of three teeth in each thread has been established as regular practice.

Instances have been encountered, however, where a user insisted on a much longer parallel section for one or both of two reasons: First, so that he may use the hob with either radial or tangential feed, and second, because of the contention that when using tangential feed there should be sufficient full teeth to insure complete generating action after the machine is relieved of the strains set up by the roughing teeth. Such procedure may result in an improved product, but the cutting time is naturally increased. It is felt that a hob user wishing a hob of this description should clearly indicate his requirements when ordering the hob.

Most worm gears are manufactured on an interchangeable basis; that is, a standard center distance must be maintained, and in addition, any worm must run satisfactorily with any gear. Another usual requirement is that an angle of 90 deg. between the axes of the gears be adhered to. As the teeth of a hob must be relieved or backed off, it follows that a new hob must be made oversize in diameter in order that it may give reasonable service before being discarded because of a reduced diameter.

Finishing hobs for unusually important work, such as index wheels, are usually made with a very small allowance for sharpening, the amount which the diameter is made oversize on the new hob frequently being as little as 0.005 to 0.020, depending upon the pitch. For regular commercial work, such small allowances would decrease the hob life to a degree that would be prohibitive; consequently, larger sharpening allowances are generally used.

For many years this allowance has been based on one factor only; namely, the axial pitch of the hob. Experience has shown that under this condition, single-thread hobs would have an unnecessary short life while, at the same time, a multiple-threaded hob would give poor tooth contact because of the presence of the sharpening allowance. Consequently, a formula has been written which takes into consideration the factors of normal circular pitch and also the helix angles found at the hob outside diameter and pitch diameter. This formula is:

$$\text{Sharpening allowance} = 0.075 \times \text{normal circular pitch} \times \left[\frac{16 - (A\text{-}B)}{16}\right] + 0.010$$

A = angle of thread with axis at hob outside diameter, degrees.
B = angle of thread with axis at hob pitch diameter, degrees.

The characteristics of the sharpening allowance calculated by means of this formula may be summarized as follows:

For a given axial pitch, the influence which "normal circular pitch" in the formula has on the 0.075 constant will vary from approximately 1 to 0.7.

Within the range of practical hobs, the value of $\left[\frac{16 - (A\text{-}B)}{16}\right]$ will run from 1 to 0.5.

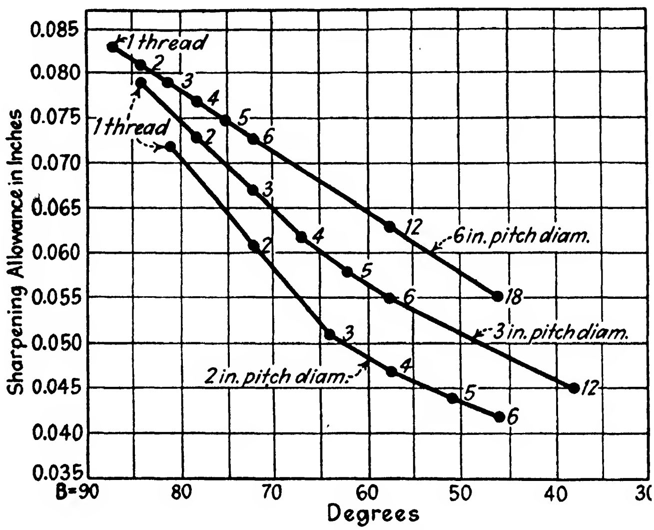


FIG. 84.—Sharpening allowances on various hobs of 1-in. axial pitch are compared graphically: tooth depth = 0.6866 × axial pitch when B = 72 deg. or more; tooth depth = 0.6866 × normal circular pitch when B is less than 72 deg.

Thus, two hobs of the same axial pitch may, in an extreme case, have sharpening allowances varying as much as 3 to 1. The one receiving the smaller allowance will be of much smaller normal circular pitch and will be of small diameter relative to its pitch, resulting in a hob which would not give satisfactory tooth contact if made with a large sharpening allowance.

The last item of the formula (+0.010) is designed to prevent the allowance on fine pitch hobs from becoming unreasonably small. For example: Without this factor, a hob of 0.100 axial pitch would, neglecting the helix angle, receive an allowance of only 0.0075, while with it, this becomes 0.0175. The chart in Fig. 84 shows graphically the sharpening allowances resulting from the use of the formula as applied to a considerable number of hobs, all of which are of 1-in. axial pitch.

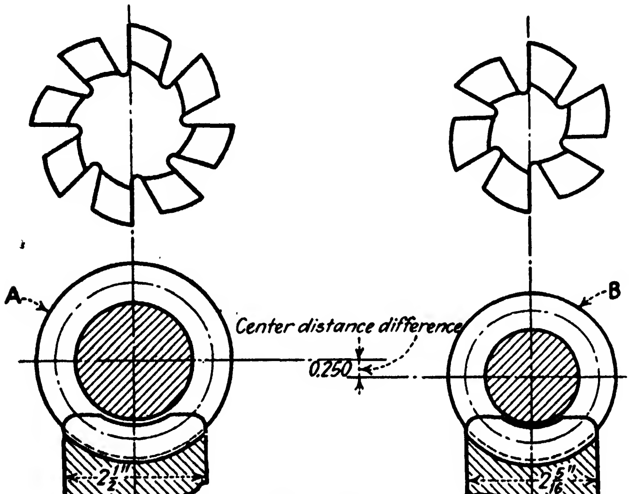
If a gear is cut with a hob that is oversize in diameter, the resulting contact on the gear teeth will be on diagonally opposite corners. To avoid this, it is common practice to adjust the hob axis slightly from the 90-deg. relation with the axis of the work spindle. The amount of this adjustment depends upon the difference between the angle which the worm thread makes with its axis as compared with the similar angle of the oversize hob. In each case, this angle is calculated at the pitch line, the pitch line of the hob being considered to have been expanded as much as its outside diameter is oversize.

In measuring the diameter of a hob to determine the amount which it is oversize, it should be remembered that the hob tooth is made to provide a clearance between the outside diameter of the worm and the bottom of the gear tooth. To take a specific example: If a 1-in. axial-pitch worm had an outside diameter of 3 in., the corresponding hob would (using standard tooth depth based on the axial rather than on the normal circular pitch) have an outside diameter of 3.100 if no sharpening allowance existed. The amount by which the hob outside diameter might exceed this figure would be the amount to add to the worm pitch diameter when calculating the angle of the hob teeth with their axis as referred to above.

It should perhaps be stated that setting the hob spindle in accordance with the calculation referred to cannot be depended upon to give the best possible contact between the gear teeth. The calculated angle, however, gives a good approximation, but for the best results, a further slight adjustment as dictated by an inspection of the first gear cut is frequently desirable. The need of this second adjustment usually increases with the number of threads in the hob.

A method of avoiding the complications resulting from the use of a sharpening allowance, which is sometimes practiced, is to make the worm diameter to correspond with the existing hob diameter. This, of course, results in noninterchangeable worms and gears, although it is possible to retain a standard center distance. The advantages of this practice are that the hob life is materially increased and that the hob spindle can remain set at an angle of 90 deg. with the work spindle at all times.

A recommended standard for the diameter of worms, made with a bore in keeping with the pitch, has been adopted by the American Gear Manufacturers Association. It is believed that a similar standard could profitably be devised for those worms which are made as a part of the shaft. It is felt that frequently gear manufacturers have, when all factors are considered, adopted



Worm Data

	A	B
Axial Pitch	1.000	1.000
No. of Threads	4	4
Pitch Diam.	2.750	2.250
Outside Diam.	3.487	2.987
Angle of Thread with Axis	65°9'	60°30'
Theor. Eff. (Coef. of Friction assumed as 0.05)	88.17%	89.28%

FIG. 85.—Small worm diameters should be avoided as they impair tooth contact and increase cost of cutting.

undesirably small worm diameters. This tendency has presumably been caused by a desire to obtain high efficiency. Figure 85 shows two worm and gear sections with their corresponding hobs which illustrate the point in question. The diameter of worm *A* was determined by the formula which appears in the current edition of the Brown and Sharpe "Formulas in Gearing" which reads:

$$\text{Pitch diameter} = (2.35 \times \text{axial pitch}) + 0.4.$$

Worm *B* is the same as *A* except that its pitch diameter has been reduced ½ in. The disadvantages of design *B* as compared with *A* include:

1. Hob is much weaker and has fewer generating teeth.
2. Hob cost per gear will probably be much higher.
3. Hobbing time per gear will be greater because of hob weakness.
4. Good initial tooth contact will be more difficult to attain.
5. The maximum width of face of the mating gear is less, thus reducing the maximum load carrying capacity.

It is, of course, true that the center distance when using worm *A* is increased ¼ in.; also, that the efficiency calculated on the

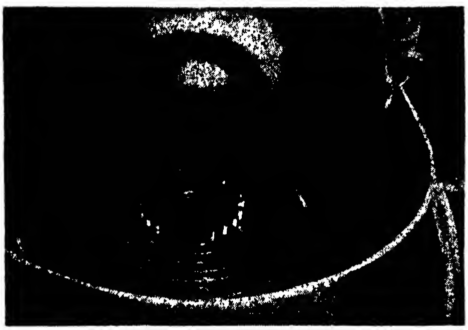


Fig. 86.—Sharpening cutter on rotary grinder.

basis of a coefficient of friction of 0.05 is reduced approximately 1 per cent. It is felt, however, that these points are relatively unimportant when compared with the unavoidable disadvantages of extreme small worm diameters, and that though general adoption of a formula, such as that suggested, would work to the mutual advantage of both the gear manufacturer and the maker of the necessary cutting tools.

Sharpening Gear-shaper Cutters.—It is difficult to overemphasize the necessity of keeping gear cutters sharp. Dull cutters not only take more power but delay the work and do not produce so smooth a finish. The ease with which gear-shaper cutters can be sharpened leaves no excuse for using dull ones.

Sharpening spur-gear cutters is a simple job of face-grinding. The easiest method is to use a rotary surface grinder where one is available, as in Fig. 86, or they can also be easily ground on a universal grinder as in Fig. 87. The cutter is centered on an expanding pin and ground to a 5-deg. angle on the cutting face.

Spur-gear cutters are made from high-speed steel and care must be taken to prevent checking of the cutter face, caused by unequal heating and cooling. Wet grinding is advised with a copious supply of cutting compound or plain water. This is more easily secured on the rotary grinder in Fig. 86 than on the universal grinder. The Fellows Co. uses an alundum wheel, 10 in. in diameter with ¾-in. face, running at 4,500 surface feet per minute. The grade is 46 J and the cutter runs at 120 ft. per minute surface speed. The depth of cut should not exceed 0.0015 in. per traverse of the wheel nor the traverse feed be over

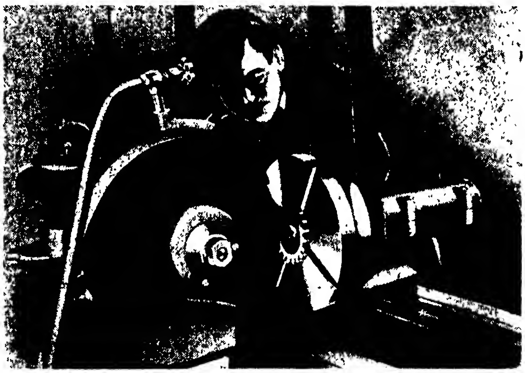


FIG. 87.—Using a universal grinder for sharpening.

20 in. per minute. The last three or four passes should be without feed, or what is known as sparking out. The cutter should be completely submerged in water during the grinding.

Opinions differ as to the best cooling compound. Plain water cools well but has the disadvantage of causing rust. One pint of sal soda to 10 quarts of water, mixed cold, gives good results.

Helical gear cutters require more attention in sharpening, and as it is much more difficult to secure a good supply of cooling compound they are usually ground dry, and at a slower rate. Two angular settings are required, one for the helix angle and the other for the 5-deg. top rake, the same as the spur-gear cutter. The method of grinding is shown in outline in Fig. 88. The cutter is held on an extended arm of the attachment which can be swiveled to the desired angle to correspond with the helix angle of the cutter teeth, to bring the face of the tooth at right angles to the helix angle. In addition, the arm is located permanently at an angle of 5 deg. with the top face of the grinder table.

No indexing mechanism is necessary except a spring stop which fits between the teeth as shown. The cutter is first centered by the gage provided for that purpose and the tooth used for centering is the first tooth to be ground. The index plunger plate is then adjusted so that the indexing plunger stop will locate properly in the space between the cutter teeth. The cutter is

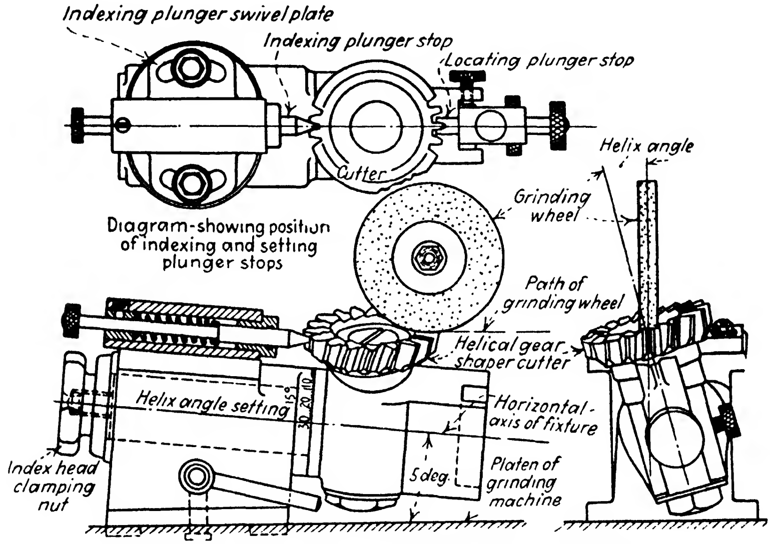


FIG. 88.—How helical cutters are held for grinding.

then traversed back and forth under the grinding wheel as shown. The face of the wheel should be kept true with a diamond. Light cuts should be taken to avoid overheating the cutter which is ground dry.

The same type of wheel is used as for the spur-gear cutters, but it is 4 in. in diameter and ½-in. face. The surface speed is somewhat slower, 4,300 ft. per minute. The traversing is done by hand and the feed per traverse should not exceed 0.001 in. A large number of light cuts is advised to avoid heating. For finishing, the cutter should be indexed around twice without any feed. This gives a smoother finish to the cutter teeth.

CHAPTER VI

BEVEL GEARS

Bevel gearing is one of the oldest methods of transmitting power at an angle. Cast-iron bevel gears and gears with wooden teeth set into iron bodies and the teeth held in place by wooden wedges were common in the early days of mills driven by water power. When the shafts were at right angles, these were frequently called miter gears. Some also reserve this term to bevel gears having the same number of teeth.

Bevel gears have teeth cut on an angular face for transmitting motion and power to shafts that are set at an angle to each other. The face angles of the gears vary widely both on account of varying shaft angles and because of different diameters of the gears which run together. Where the sizes vary materially, the small gear is usually called a pinion, as with spur gears. The various parts of bevel gears and of the teeth are shown in Fig. 89, these designations being sponsored by the Gleason Works, long known as specialists in bevel-gear work.

The simplest form of bevel gear has plain, or straight, teeth which are radial with the shaft or gear blank. The desire for noise elimination led to cutting teeth at an angle on the beveled face of the gear. These were known as "skew" gears, but they have given way to the spiral-bevel gear, developed by the Gleason Works, of Rochester, N. Y., especially for the rear-axle drive of automobiles. This form of gear tooth is cut with special face milling cutters, as will be shown later. The curved teeth give a very smooth and silent action and this type of bevel gear is now being largely used in different mechanisms.

Straight-tooth bevel gears are cut both with rotary cutters and by the generating process, where the teeth are planed, or shaped, by a tool traveling across the face of the gear blank which rolls so as to produce the proper form of tooth. Two types of these machines are built in this country, one by Bilgram of Philadelphia, and the other by Gleason. Each method will be briefly described, as will the use of rotary cutters. For, though rotary cutters cannot produce so accurate a bevel gear as the planing method, there are many places where the milling machine can be used to advantage in this way, especially in small shops and on repair work. The Bilgram machine is shown in Fig. 90.

Bevel gears involve several problems not found in spur gears and it is well to understand these differences. The tapered teeth,

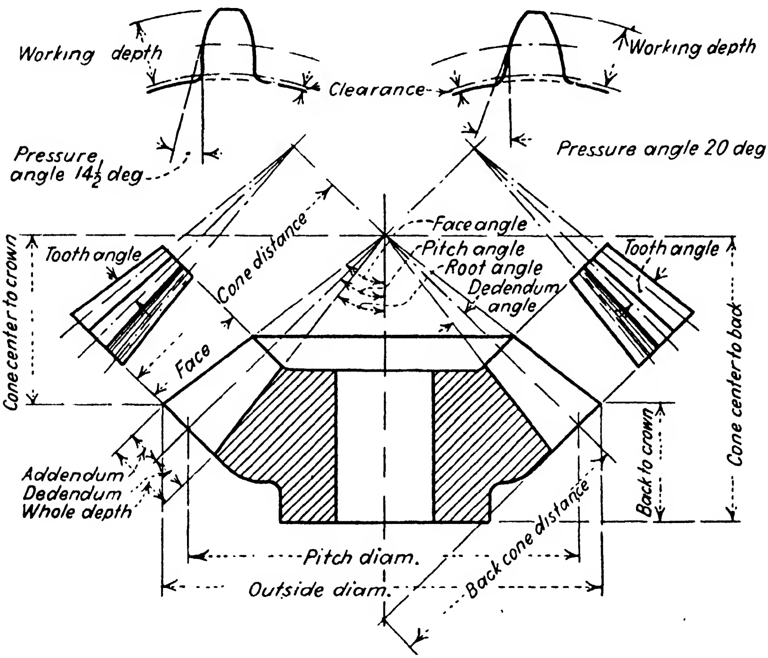


FIG. 89.—Names of parts of bevel gears and teeth.

which vary in width from one end to the other, are perhaps the greatest single difference. A study of Fig. 89, which gives the names of the different parts of the teeth and shows the different angles and definitions that must be considered, will give some idea of the problems that must be met in bevel-gear work. With these terms fixed in mind there will be less difficulty in understanding those who speak the bevel-gear language.

Cutting Bevel Gears in the Milling Machine.—As the teeth of bevel gears change constantly in pitch from one end to the other, the only method of cutting them correctly is by planing radially from the center of the cone from which the pitch lines are drawn. But though correct teeth cannot be cut with a rotary cutter, a close approximation can be made by a little manipulation. The cutter used must be correct for the outer end of the teeth and this will leave the curve too straight at the small end. Where better than ordinary gears are wanted, the teeth are cut as nearly correct

FIG. 90.—Bilgram bevel-gear planer.

as possible and the small ends of the teeth filed to the proper shape.

The method suggested by the Brown and Sharpe Manufacturing Company is as follows:

Pitch of Bevel Gear.—The pitch of a bevel gear is always considered as that at the largest end of the teeth.

Data Required to Cut Bevel Gears with Rotary Cutter.—Pitch and number of teeth in each gear.

The whole depth of tooth spaces at both large and small ends of teeth.

The thickness of teeth at both ends.

The height of teeth above the pitch line at both ends.

The cutting angle; the angle to set index head on milling machine, and the proper cutter or cutters.

Scratch Depth Line on Blank.—Before placing the blank on machine, measure the length of face, angles, and outside diameter of blank, and, if all dimensions are correct, place the blank on the arbor and fasten it securely in place; then scratch the whole depth of space at large end with a depth of gear tooth gage similar to that shown in Fig. 91.

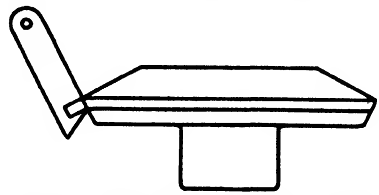


FIG. 91.—Using gage for depth of gear tooth.

Selection of Cutter for Bevel Gears.—The length of teeth or face on bevel gears is not ordinarily more than one-third the apex distance, *Ab*, Fig. 92, and cutters usually carried in stock are suitable for this face. If the face is longer than one-third the apex distance, special thin cutters must be made.

Rule for Selecting Cutter.—Measure the back cone radius *ab* for the gear, or *bc* for the pinion. This is equal to the radius of a spur gear, the number of teeth in which would determine the cutter to use. Hence twice *ab* times the diametral pitch equals the number of teeth for which the cutter should be selected for the gear. Looking in the list given on page 57, the proper number for the cutter can be found.

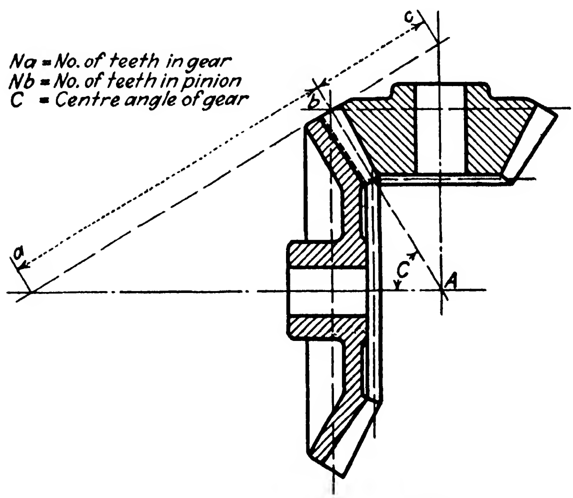


FIG. 92.—Layout of bevel-gear angles.

Thus, let the back cone radius *ab* be 4 in. and the diametral pitch be 8. Twice four is 8, and 8 × 8 is 64, from which it can be seen that the cutter must be of shape No. 2, as 64 is between 55 and 134, the range covered by a No. 2 cutter.

The number of teeth for which the cutter should be selected can also be found by the following formula:

$\tan C = Na/Nb$

No. of teeth to select cutter for gear $= \dfrac{Ng}{\cos C}$

No. of teeth to select cutter for pinion $= \dfrac{Np}{\sin C}$

If the gears are miters or are alike, only one cutter is needed; if one gear is larger than the other, two may be needed.

Setting Cutter out of Center.—As the cutter cannot be any thicker than the width of space at small end of teeth, it is necessary to set it out of center and rotate the blank to make the spaces of the right width at the end of the teeth.

The amount to set cutter out of center can be calculated from Table 43, and the following formula:

$$\text{Set-over} = \frac{Tc}{2} - \frac{\text{factor from table}}{P}$$

P = diametral pitch of gear to be cut.

Tc = thickness of cutter used, measured at pitch line.

Given as a rule, this would read: Find the factor in the table corresponding to the number of the cutter used and to the ratio of apex distance to width of face; divide this factor by the diametral pitch and subtract the quotient from half of the thickness of the cutter at the pitch line.

As an illustration of the use of this table in obtaining the set-over, take the following example: A bevel gear of 24 teeth, 6 pitch, 30-deg.-pitch cone angle and 1¼-in. face. These dimensions call for a No. 4 cutter and an apex distance of 4 in.

In order to get the factor from the table, the ratio of apex distance with length of face must be known. This ratio is $4/1.25 = 3.2/1$, or about $3\frac{1}{4}/1$. The factor in the table for this ratio with a No. 4 cutter is 0.280. Next, measure the cutter at the pitch line. To do this, refer to the regular Table of Tooth Parts on pages 21 to 24, and get the depth of space below pitch line $s + f$. This depth of space below pitch line can also be found by dividing 1.157 by the diametral pitch. In the case of 6 pitch $s + f = 0.1928$ in. The thickness of the cutter at the pitch line is then found to be 0.1745 in. This dimension will vary with different cutters, and will vary in the same cutter as it is ground away, since formed bevel-gear cutters are commonly provided with side relief. Substituting these values in the formula, the following result is obtained:

$\text{Set-over} = \dfrac{0.1745}{2} - \dfrac{0.280}{6} = 0.0406$ in., which is the required dimension.

After selecting a cutter and determining how much to set it out of center, proceed as follows:

Set the cutter central with the universal index-head spindle, as the machine may be equipped.

Set the head to the proper cutting angle.

Set the index head for the number of teeth to be cut, placing the sector on the straight row of holes that are numbered to start with.

Set the dial on the cross-feed screw to the zero line.

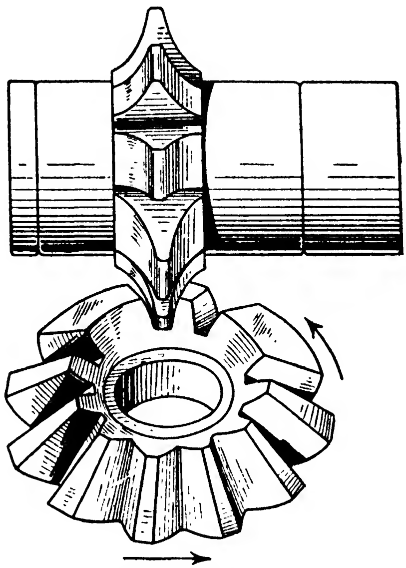


Fig. 93.—Setting work off-center to widen tooth space at outer end.

Scratch the depth of both the large and small end of the tooth to be cut in the blank.

Index and cut two or three grooves or center cuts to conform to the lines in depth.

Set the cutter out of center the trial distance, according to the formula on the previous page, by moving the saddle and noting adjustment on the cross-feed-screw dial.

Rotate the gear in the opposite direction from that in which the table is moved off center (Fig. 93) until the side of the cutter nearest the center line of the gear will cut the entire surfaces of the approaching sides of the teeth.

After making one or more cuts in accordance with this setting, move the table the same distance on the opposite side of the center and rotate the gear in the opposite direction from that in which the table is moved until the cutter just touches the side of a tooth at the small end and cuts the entire surface of this side the same as the other.

Cut one or more spaces and measure the teeth at both large and small ends, either with a gear tooth vernier or with gages made from thin pieces of metal and having a slot cut to give the correct depth and width at the pitch line.

If the teeth at the large end are too thick when the small end is correct, the amount to set the table out of center must be increased. On the other hand, if the small end is too thick when the large end is correct, the amount the table is set out of center is too great. In either case, the settings must be changed, and the operations of cutting repeated, remembering that the blank must be rotated and the table moved the same amount each side of center, otherwise the teeth will not be central. It is well to bear in mind that too much out of center leaves the small end proportionately too thick, and too little out of center leaves the small end too thin.

The adjustment of the cutter and the rotating of the blank are shown in Fig. 93, which shows the setting, so that the right side of cutter will trim the left side of tooth and widen the large end of the space. The table has been moved to the right and the blank brought to the position shown, by rotating it in the direction of the arrow; the first out of center cut was taken when the cutter was set on the other side of the center.

After determining the proper amount to set cutter out of center, the teeth can be finished, without making a central cut, by cutting around the blank with the cutter set out of center, first on one side and then on the other.

To prevent the teeth being too thin at either end, it is important, after cutting once around the blank with cutter out of center, to give careful attention to the rotative adjustment of the gear blank, when setting the cutter for trimming the opposite sides of the teeth. If by measurement, both ends are a little too thick, but proportionately right, rotate the gear blank and make trial cuts until one tooth is of the correct thickness at both ends. The cutting can then be continued until the gear is finished. Teeth of incorrect thickness may be more objectionable than a slight variation in depth.

The finished spaces, or teeth, as already mentioned, are of the correct form at the larger ends, and the teeth are of the correct thickness their entire length, but the tops of the teeth at the small ends are not rounded over enough. It is, therefore, generally necessary to file the faces of the teeth slightly above the pitch line at the small ends, as indicated by the dotted lines *FF*, Fig. 94. In filing the teeth, they should not be reduced any in thickness at or below the pitch line.

When cutting cast-iron gears coarser than five diametral pitch, it is best to make one central cut entirely around the blank before attempting to find the correct setting of the cutter or rotation of the blank for correct thickness of teeth; and it is generally advantageous to take a central cut on nearly all bevel gears of steel.

TABLE 43.—OBTAINING SET-OVER FOR CUTTING BEVEL GEARS

Ratio of apex distance to width of face = apex/face

No. of cutter	3/1	$3\frac{1}{4}$/1	$3\frac{1}{2}$/1	$3\frac{3}{4}$/1	4/1	$4\frac{1}{4}$/1	$4\frac{1}{2}$/1	$4\frac{3}{4}$/1	5/1	$5\frac{1}{2}$/1	6/1	7/1	8/1
1	0.254	0.254	0.255	0.256	0.257	0.257	0.257	0.258	0.258	0.259	0.260	0.262	0.264
2	0.266	0.268	0.271	0.272	0.273	0.274	0.274	0.275	0.277	0.279	0.280	0.283	0.284
3	0.266	0.268	0.271	0.273	0.275	0.278	0.280	0.282	0.283	0.286	0.287	0.290	0.292
4	0.275	0.280	0.285	0.287	0.291	0.293	0.296	0.298	0.298	0.302	0.305	0.308	0.311
5	0.280	0.285	0.290	0.293	0.295	0.296	0.298	0.300	0.302	0.307	0.309	0.313	0.315
6	0.311	0.318	0.323	0.328	0.330	0.334	0.337	0.340	0.343	0.348	0.352	0.356	0.362
7	0.289	0.298	0.308	0.316	0.324	0.329	0.334	0.338	0.343	0.350	0.360	0.370	0.376
8	0.275	0.286	0.296	0.309	0.319	0.331	0.338	0.344	0.352	0.361	0.368	0.380	0.386

Cutters for Right-angle Gears and Pinions.—In Table 44 will be found the cutters that can be used in milling right-angle bevel gears and their mating pinions. These have been calculated by the National Twist Drill & Tool Company and will be found very convenient. Pinions are given at the top and the mating gears at the side of the table. The first number of the pair gives the cutter for the gear and the second number the cutter for the pinion. This shows that with a 20-tooth pinion and a 50-tooth gear, the pinion should have a No. 5 cutter and the gear a No. 2 cutter. These tables cover a large range of combinations and will be found useful when cutting bevels in this way.

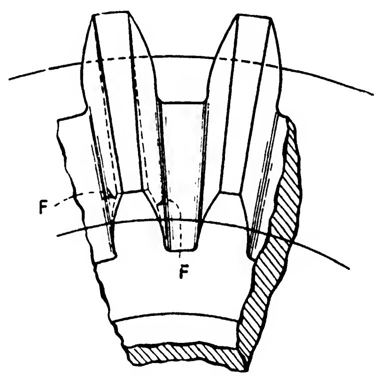


FIG. 94.—Teeth must be rounded at the small end.

Roughing Out Bevel Gears.—In many shops where straight-tooth bevel gears are made in large quantities they are roughed out on milling machines so as to leave comparatively little metal to be removed by the planing or generating machines. Figure 95

Table 44.—Cutters for Use in Cutting Bevel Gears

Gear	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
12	7-7																		
13	6-7	6-6																	
14	5-7	6-6	6-6																
15	5-7	5-6	5-6	5-5															
16	4-7	5-7	5-6	5-6	5-5														
17	4-7	4-7	4-6	5-6	5-5	5-5													
18	4-7	4-7	4-6	4-6	4-5	4-5	5-5												
19	3-7	4-7	4-6	4-6	4-6	4-5	4-5	4-4											
20	3-7	3-7	4-6	4-6	4-6	4-5	4-5	4-4	4-4										
21	3-8	3-7	3-7	3-6	4-6	4-5	4-5	4-5	4-4	4-4									
22	3-8	3-7	3-7	3-6	3-6	3-5	4-5	4-5	4-4	4-4	4-4								
23	3-8	3-7	3-7	3-6	3-6	3-5	3-5	3-5	3-4	4-4	4-4	4-4							
24	3-8	3-7	3-7	3-6	3-6	3-6	3-5	3-5	3-4	3-4	3-4	4-4	4-4						
25	2-8	2-7	3-7	3-6	3-6	3-6	3-5	3-5	3-5	3-4	3-4	3-4	4-4	3-3					
26	2-8	2-7	3-7	3-6	3-6	3-6	3-5	3-5	3-5	3-4	3-4	3-4	3-4	3-3	3-3				
27	2-8	2-7	2-7	2-6	3-6	3-6	3-5	3-5	3-5	3-4	3-4	3-4	3-4	3-4	3-3	3-3			
28	2-8	2-7	2-7	2-6	2-6	3-6	3-5	3-5	3-5	3-4	3-4	3-4	3-4	3-4	3-3	3-3	3-3		
29	2-8	2-7	2-7	2-7	2-6	2-6	3-5	3-5	3-5	3-4	3-4	3-4	3-4	3-4	3-3	3-3	3-3	3-3	
30	2-8	2-7	2-7	2-7	2-6	2-6	2-5	2-5	3-5	3-5	3-4	3-4	3-4	3-4	3-4	3-3	3-3	3-3	3-3
31	2-8	2-7	2-7	2-7	2-6	2-6	2-6	2-5	2-5	2-5	3-4	3-4	3-4	3-4	3-4	3-3	3-3	3-3	3-3
32	2-8	2-7	2-7	2-7	2-6	2-6	2-6	2-5	2-5	2-5	2-4	2-4	3-4	3-4	3-4	3-3	3-3	3-3	3-3
33	2-8	2-8	2-7	2-7	2-6	2-6	2-6	2-5	2-5	2-5	2-4	2-4	2-4	3-4	3-4	3-4	3-3	3-3	3-3
34	2-8	2-8	2-7	2-7	2-6	2-6	2-6	2-5	2-5	2-5	2-4	2-4	2-4	2-4	2-4	3-4	3-3	3-3	3-3
35	2-8	2-8	2-7	2-7	2-6	2-6	2-6	2-5	2-5	2-5	2-4	2-4	2-4	2-4	2-4	2-4	2-3	3-3	3-3
36	2-8	2-8	2-7	2-7	2-6	2-6	2-6	2-5	2-5	2-5	2-5	2-4	2-4	2-4	2-4	2-4	2-3	2-3	2-3
37	2-8	2-8	2-7	2-7	2-6	2-6	2-6	2-5	2-5	2-5	2-5	2-4	2-4	2-4	2-4	2-4	2-3	2-3	2-3
38	2-8	2-8	2-7	2-7	2-6	2-6	2-6	2-5	2-5	2-5	2-5	2-4	2-4	2-4	2-4	2-4	2-4	2-3	2-3
39	2-8	2-8	2-7	2-7	2-6	2-6	2-6	2-5	2-5	2-5	2-5	2-4	2-4	2-4	2-4	2-4	2-4	2-3	2-3
40	1-8	2-8	2-7	2-7	2-6	2-6	2-6	2-5	2-5	2-5	2-5	2-4	2-4	2-4	2-4	2-4	2-4	2-3	2-3
41	1-8	1-8	2-7	2-7	2-6	2-6	2-6	2-6	2-5	2-5	2-5	2-4	2-4	2-4	2-4	2-4	2-4	2-3	2-3
42	1-8	1-8	2-7	2-7	2-6	2-6	2-6	2-6	2-5	2-5	2-5	2-5	2-4	2-4	2-4	2-4	2-4	2-3	2-3
43	1-8	1-8	1-7	2-7	2-6	2-6	2-6	2-6	2-5	2-5	2-5	2-5	2-4	2-4	2-4	2-4	2-4	2-4	2-3
44	1-8	1-8	1-7	1-7	2-6	2-6	2-6	2-6	2-5	2-5	2-5	2-5	2-4	2-4	2-4	2-4	2-4	2-4	2-3
45	1-8	1-8	1-7	1-7	1-7	2-6	2-6	2-6	2-5	2-5	2-5	2-5	2-4	2-4	2-4	2-4	2-4	2-4	2-3
46	1-8	1-8	1-7	1-7	1-7	2-6	2-6	2-6	2-5	2-5	2-5	2-5	2-4	2-4	2-4	2-4	2-4	2-4	2-3
47	1-8	1-8	1-7	1-7	1-7	1-6	2-6	2-6	2-5	2-5	2-5	2-5	2-4	2-4	2-4	2-4	2-4	2-4	2-3
48	1-8	1-8	1-7	1-7	1-7	1-6	1-6	2-6	2-5	2-5	2-5	2-5	2-4	2-4	2-4	2-4	2-4	2-4	2-3
49	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	2-5	2-5	2-5	2-5	2-4	2-4	2-4	2-4	2-4	2-4	2-3
50	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	2-5	2-5	2-5	2-5	2-4	2-4	2-4	2-4	2-4	2-4	2-3
51	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-5	2-5	2-5	2-5	2-5	2-4	2-4	2-4	2-4	2-4	2-4	2-4
52	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-5	1-5	2-5	2-5	2-4	2-4	2-4	2-4	2-4	2-4	2-4
53	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-5	1-5	1-5	2-5	2-4	2-4	2-4	2-4	2-4	2-4	2-4
54	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-5	1-5	1-5	1-5	2-4	2-4	2-4	2-4	2-4	2-4	2-4
55	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	2-4	2-4	2-4	2-4	2-4	2-4

These tables apply only to bevel gears with axis at right angles. Pinions at top.

shows a Milwaukee milling machine with a special fixture mounted on the table for holding the gear blank at the proper angle and also for indexing it from tooth to tooth. Double fixtures are also made for this purpose and, where the gears are

TABLE 44.—CUTTERS FOR USE IN CUTTING BEVEL GEARS.—(*Continued*)

Gear	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50
31	3-3																			
32	3-3	3-3																		
33	3-3	3-3	3-3																	
34	3-3	3-3	3-3	3-3																
35	3-3	3-3	3-3	3-3	3-3															
36	2-3	3-3	3-3	3-3	3-3	3-3														
37	2-3	2-3	2-3	2-3	3-3	3-3	3-3													
38	2-3	2-3	2-3	2-3	2-3	3-3	3-3	3-3												
39	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2											
40	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2										
41	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2									
42	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2								
43	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2							
44	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2						
45	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2					
46	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2				
47	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2	2-2			
48	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2	2-2	2-2		
49	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2	2-2	2-2	2-2	
50	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2	2-2	2-2	2-2	2-2
51	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2	2-2	2-2	2-2
52	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2	2-2	2-2	2-2
53	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2	2-2	2-2	2-2
54	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2	2-2	2-2	2-2
55	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2	2-2	2-2	2-2	2-2

These tables apply only to bevel gears with axis at right angles. Pinions at top.

small enough, three or four gears are mounted on each side of the fixture, as in Fig. 96. Here one side can be loaded while the other is in the cut and so make nearly continuous milling. All the gears in each bank are indexed at the same time.

TABLE 44.—CUTTERS FOR USE IN CUTTING BEVEL GEARS.—(*Continued*)

Gear	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
56	1-8	1-8	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	2-4	2-4	2-4	2-4	2-4
57	1-8	1-8	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	2-4	2-4	2-4	2-4
58	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	2-4	2-4	2-4
59	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	2-4	2-4
60	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	2-4
61	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
62	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
63	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
64	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
65	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
66	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
67	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
68	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
69	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
70	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
71	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
72	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
73	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
74	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1 4	1-4	1-4
75	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
76	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
77	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
78	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
79	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
80	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
81	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
82	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
83	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
84	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
85	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
86	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
87	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
88	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
89	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
90	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
91	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
92	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
93	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
94	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
95	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
96	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
97	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
98	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
99	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4
100	1-8	1-8	1-7	1-7	1-7	1-6	1-6	1-6	1-6	1-5	1-5	1-5	1-5	1-4	1-4	1-4	1-4	1-4	1-4

These tables apply only to bevel gears with axis at right angles. Pinions at top.

The Gleason Works also make a three-spindle machine for roughing out straight-tooth bevel gears. They use one milling cutter approximately 21 in. in diameter, of the inserted blade type. They also make what is called a completing generator

TABLE 44.—CUTTERS FOR USE IN CUTTING BEVEL GEARS.—(*Continued*)

Gear	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50
56	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2	2-2	2-2
57	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2	2-2	2-2
58	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2	2-2	2-2
59	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2	2-2	2-2
60	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2	2-2
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62	1-3	1-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2	2-2
63	1-3	1-3	1-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2	2-2
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65	1-4	1-3	1-3	1-3	1-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2	2-2	2-2
66	1-4	1-3	1-3	1-3	1-3	1-3	1-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2
67	1-4	1-3	1-3	1-3	1-3	1-3	1-3	1-3	2-3	2-3	2-3	2-3	2-3	2-3	2-3	2-2	2-2	2-2	2-2	2-2
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75	1-4	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-2	1-2	1-2	1-2
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85	1-4	1-4	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-2	1-2	1-2
86	1-4	1-4	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-2	1-2	1-2
87	1-4	1-4	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-2	1-2	1-2
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93	1-4	1-4	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-2	1-2
94	1-4	1-4	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-2	1-2
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98	1-4	1-4	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-2	1-2
99	1-4	1-4	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-2	1-2
100	1-4	1-4	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-3	1-2	1-2

These tables apply only to bevel gears with axis at right angles. Pinions at top.

designed especially for both roughing and finishing the teeth in straight-tooth bevel gears and pinions. This machine has three work spindles and a 10¾-in. rotary cutter for roughing.

The teeth are finished by a pair of reciprocating tools which represent the adjacent sides of a tooth in an imaginary crown gear. Both the tools and the work contribute to the generating motion, the work spindle moving in the turret head only when in the finishing position.

FIG. 95.—Roughing out bevel-gear teeth before planing.

As now made all the cutter blades are of full height but are spaced in stepped positions. In this way each blade takes a light cut which varies with the width of the tooth space until the final blades accurately finish the tooth space to the full width.

GLEASON MACHINES

Gleason Bevel-gear Generator.—The Gleason bevel-gear generator, or planer, consists of a tool head carrying two tool slides and a work head with means for holding the gear blanks at the correct angle and for rotating them so as to form the proper tooth form. As each machine is accompanied by a very complete book of instructions, no attempt will be made to show details of setting up and operating the machine. The tool-cutting

FIG. 96.—Fixture for roughing three gears at once.

speeds on the 12-in. machine vary from 85 to 442 strokes per minute, which gives an idea of the rapidity with which these machines operate. The feed-gear chart shows that the cutting time per tooth can vary from 7.6 sec. to 1 min. and 26.5 sec., making a wide range. The tools and tool slide are shown in Fig. 97.

In some Gleason machines the generating motion is not used in roughing cuts, the cradle being locked during this operation. A simple feed motion carries the tools to the required depth. In finishing, the tools and blanks are rolled together in one direction to take a first finishing cut, and on the return roll the blank is fed slightly toward the tools, making a light and final finishing cut. When a tooth is completed, the blank is withdrawn and indexed, then returned to position for cutting the next tooth in the same

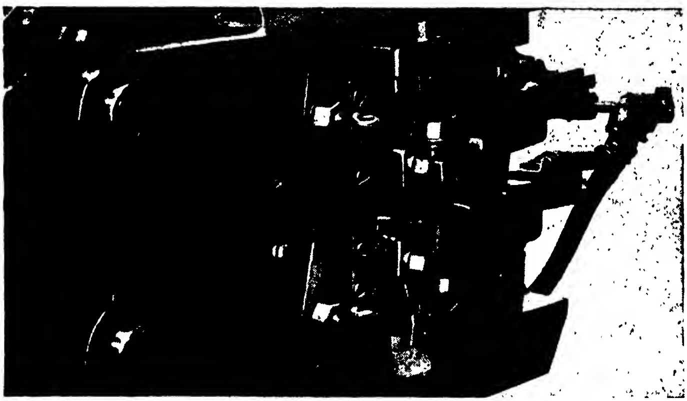


Fig. 97.—Tools and tool slides of Gleason bevel-gear planer.

manner. The machine stops automatically when the last tooth has been cut. Figure 98 shows the tools in three positions and makes clear the way in which the teeth are generated.

Provision is made so that two tooth spaces can be roughed out at the same time, the index being set for half the number of teeth in the gear. This method leaves a little more stock for removal by the finishing cut, but it is particularly useful when large quantities of gears are to be cut, as it speeds up the roughing operation approximately 80 per cent.

Cutting tools of high-speed steel are recommended. Standard and stub-tooth tools can be used for cutting teeth with equal addenda, or with long or short addenda, as desired. A coolant supply of 5 gal. per minute is recommended.

Cutting Spiral Bevel Gears.—As stated before, spiral bevel gears are cut with special face-milling cutters in machines built for the purpose by the Gleason Works of Rochester, N. Y. One of their late-type machines for roughing out the teeth of such gears is shown in Fig. 99. This is a plain roughing machine and no generating motion is employed in forming the teeth. The tooth slots are cut by a simple depth feed of the cutter into the

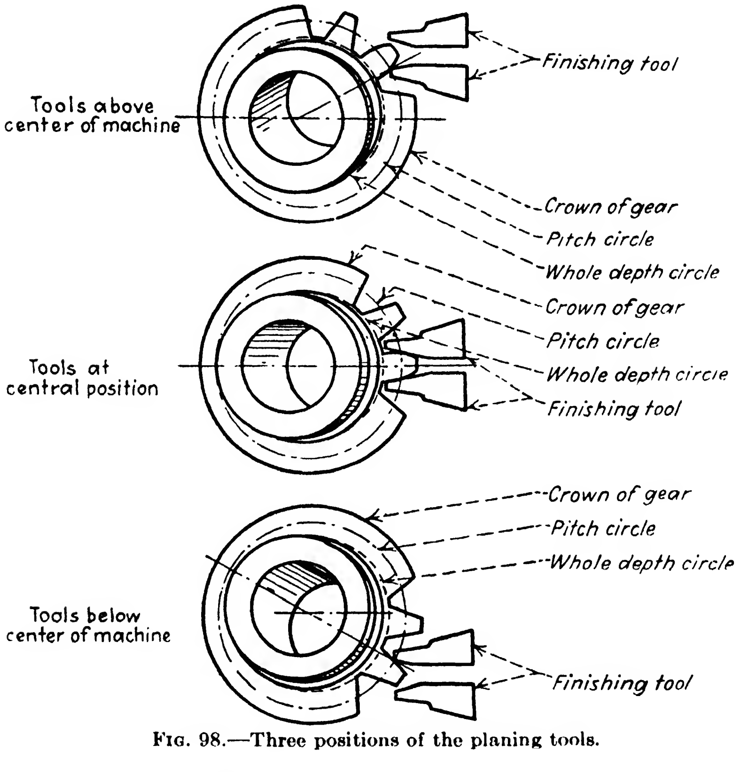


FIG. 98.—Three positions of the planing tools.

work. The finishing machine generates the tooth shape and produces teeth that are correct in shape for this type of gear.

The gear blanks are chucked hydraulically after they are put in place and a clamp plate placed over them. Pushing the starting button moves the head into position and starts the cutter motor. The cutter spindle is mounted in a saddle in the upright at the left of the machine, the entire saddle being adjustable both horizontally and vertically for setting the cutter for the proper spiral angle. The cutter is driven through a pinion and internal gear bolted to the cutter spindle faceplate. Change gears provide convenient means of securing various cutter speeds while a heavy flywheel on the spindle faceplate insures a smooth cutting action.

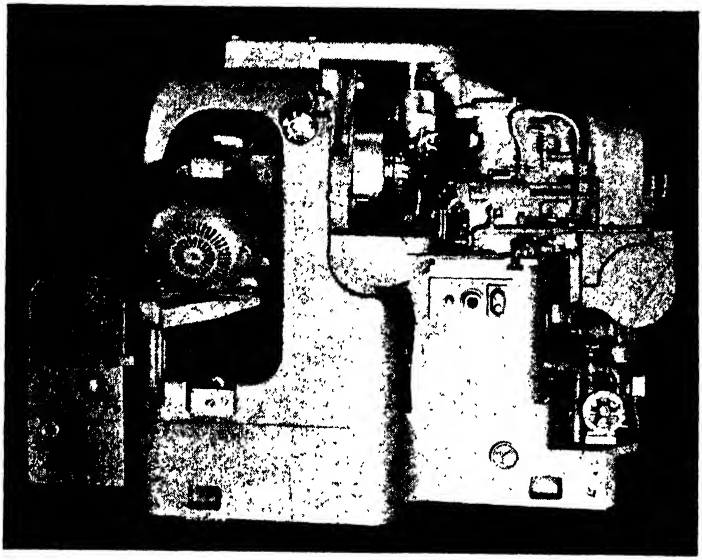


Fig. 99.—Gleason machine for rough-cutting spiral-bevel gears.

The work head is mounted directly on the frame of the machine and has all adjustments necessary to secure various pitch angles and cone distances. In this roughing machine the work spindle remains stationary except when indexing. When the last tooth has been roughed out, the work automatically moves away from the cutter far enough to permit easy removal of the gear blank. The indexing mechanism is of the notched type and is automatic. Index plates are changed to suit the gears being cut. A close-up view of the action of the cutter on the gear being cut is seen in Fig. 100.

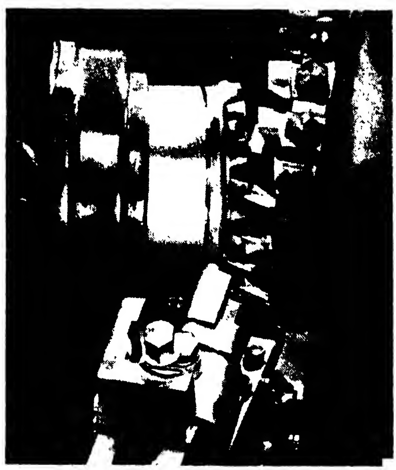


Fig. 100.—Close-up of cutter action.

Spiral Bevel-gear Generator.—The latest machine of this type is the Gleason Hypoid Generator and consists of three major parts: the frame, the work head, and the cutter cradle. The frame is cast in one piece, the cutter cradle is held in a circular housing bolted to the frame, and the work head is bolted to a large sliding carriage which moves at right angles to the root line of the gear being cut. Such a machine is shown in Fig. 101.

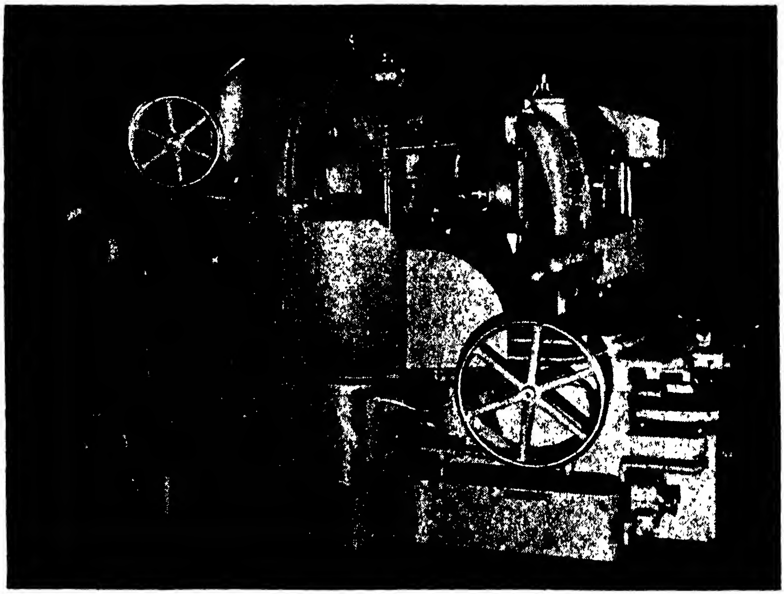


FIG. 101.—Gleason hypoid pinion generator.

The work is alternately fed into the cutter, then withdrawn and indexed for the next tooth space. As the cutter rotates through the blank to cut the lengthwise tooth curve, a relative rolling generating movement is produced at the same time between the cutter and the gear blank to generate the correct tooth profile. This rolling motion consists of a slow rotation of the cradle which carries the cutter in timed relation to a corresponding rotation of the work spindle. It corresponds to the motion of a gear rolling on a crown gear of which the cutter represents a tooth. This sequence of operations continues until the last tooth is cut when the automatic stop acts to stop the machine and prevent recutting the next tooth. This makes the action of the machine automatic after the work has been put in place and the machine started.

Although designed as a finishing machine, gears can be both rough cut and finished on this machine where the output is not sufficient to warrant having both machines. When used for roughing, this machine requires a double-track feed cam. The change from rough to finish operation is made by turning a crank

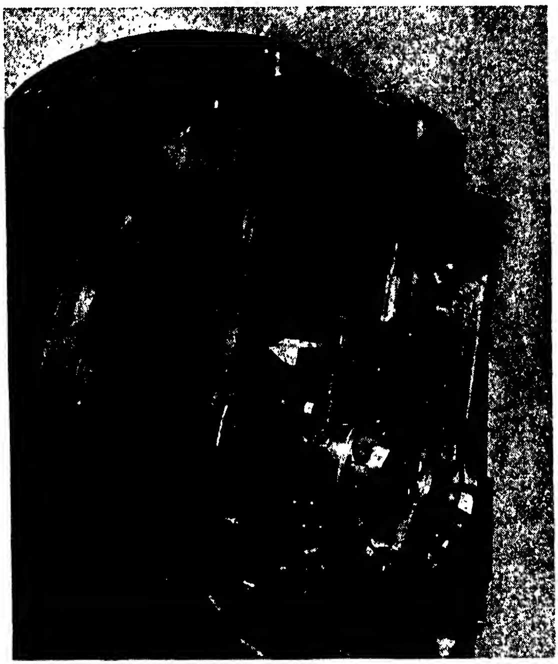


FIG. 102.—Close-up of cutter and spindle.

which shifts the feed-cam roller. A close-up of the cutter spindle and cutter is shown in Fig. 102 which gives a good idea of its construction and the angular and radial adjustments. Another close view showing the head in position to cut a hypoid pinion, with the cutter below center, is seen in Fig. 103. Cutter speeds vary from 64 to 236 ft. per minute and the cutting time per tooth from 12 to 73 sec.

Finishing Spiral-bevel and Hypoid Gears.—The Gleason Works have developed a special machine and cutter for finishing spiral-bevel and hypoid gears. This is a second operation that follows rough cutting on standard Gleason roughing machines. The finishing cutters may be likened to circular broaches having two sets of teeth, or inserted blades. The first set of teeth each take cuts that increase the depth of the teeth and shape them correctly, as with a broach. The second set of teeth each cut the full depth and take light chips until the teeth are of correct size and shape and have the desired finish.

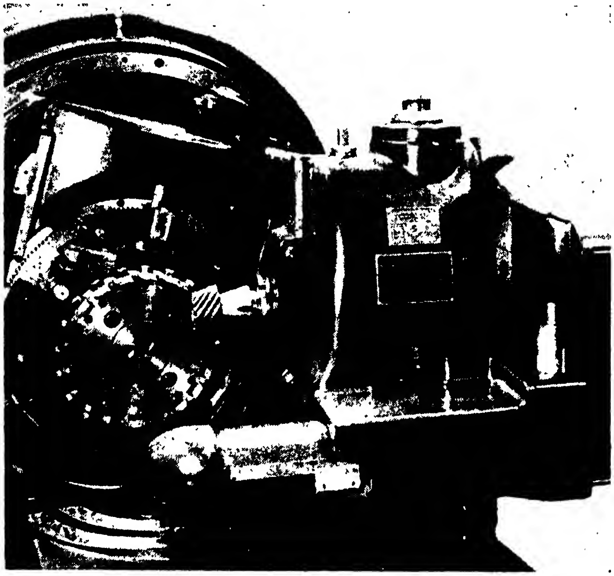


FIG. 103.—How hypoid pinions are cut.

There is a space or gap between the last finishing blade and the first blade of the first set. The gear is indexed while this gap is passing the gear so that the first set of teeth begin work on the next tooth as soon as they come into action. The cutter runs at a uniform speed and remains at the normal tooth depth at all times. It makes one revolution for each tooth of the gear. All feeding mechanism is hydraulic working at 600 lb. pressure. The machine stops after the last tooth has been cut. A detail of the cutter is shown in Fig. 104.

Three Methods of Finish Cutting.—Three methods of finish-cutting spiral-bevel gear teeth are used on these gear generators. They are known as the single-side method, the fixed-setting method, and the spread-blade method.

With the *single-side method* all the teeth are cut on one side only, then both the work and the cutter are readjusted and the opposite side is completely cut. This method is advised for small quantity lots.

The *fixed-setting method* is used only for high productions of pinions. Two machines are used, one for each side of the pinion teeth. Two cutters are used, one having all inside cutting blades and the other all outside blades. This method cuts pinions more rapidly and with greater accuracy than the single-side method.

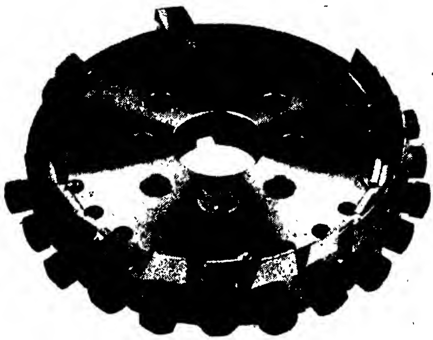


Fig. 104.—New type of cutter for Gleason machine.

The *spread-blade method* is named from the type of cutter used. The cutter blades are spread sufficiently so that the inside and outside cutting edges generate both sides of the tooth space at the same time, although the standard average radius is maintained. In this way the gear is completed in one revolution. The pinion teeth for gears cut in this way may be generated by either of the first two methods. The spread-blade method is suitable only for quantity production work and offers a considerable saving in time when the quantity warrants its use.

As an indication of the accuracy which it is necessary to maintain in order to secure the high quality of work demanded we show the method used in checking the spindles of these machines. Figure 105 shows the indicators used and the way in which they are attached. The radial runout should not exceed 0.0002 in. and the face runout should not be greater than 0.0003 in. When it is necessary to adjust these bearings, it should be done only by one thoroughly familiar with all the conditions.

Spiral-bevel-gear cutting machines are also built by the Cleveland Hobbing Machine Company, the cutting portion of one being shown in Fig. 106. These machines use a scroll-shaped,

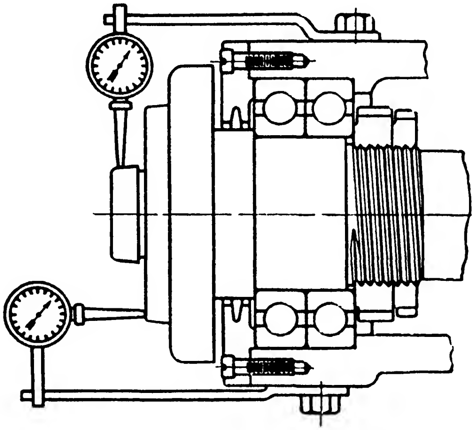


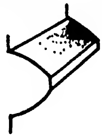
Fig. 105.—Method of checking spindle runout.

inserted-tooth cutter or hob, that cuts continuously as the work turns.

Bearing of Bevel-gear Teeth.—In order to secure the most satisfactory gearing it is necessary to have the bearing points or surfaces of the teeth come at the right places. It has been the experience of the Gleason Works that for straight bevel-gear teeth it is best to have the heaviest bearing at the small end, about as shown in Fig. 107. The illustrations show 10 examples of different tooth bearings and suggestions for correcting them to give better results.

Fig. 106.—Cutter and work in Cleveland hobbing machine.

Similar suggestions for adjusting the assembly and tooth bearings on spiral-bevel gears are also given. These are recommended by the American Gear Manufacturers Association as well as by the Gleason Works. These suggestions are quoted directly from the recommendations mentioned, Typical desirable bearing



VARIOUS KINDS OF INCORRECT BEARING

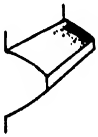
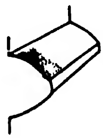
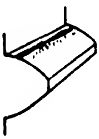
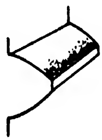
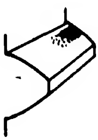
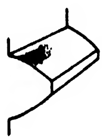
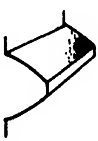
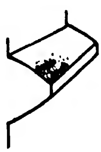
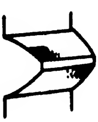
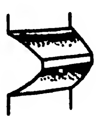
1		Heavy at small end. Change lengthwise bearing only.	2		Heavy at large end. Change lengthwise bearing only.
3		Low on tooth. Change profile bearing only.	4		High on tooth. Change profile bearing only.
5		Heavy at small end and low on tooth. Change both lengthwise and profile bearing.	6		Heavy at large end and low on tooth. Change both lengthwise and profile bearing.
7		Heavy at small end and high on tooth. Change both lengthwise and profile bearing.	8		Heavy at large ena and high on tooth. Change both lengthwise and profile bearing.
9		Cross bearing:-Heavy at small end on one side and heavy at large end on other side. Change lengthwise bearing. In connection with cross bearing, profile bearing may be high or low on one or both sides in which case change in profile bearing is necessary.	10		Lame bearing:-High on one side and low on other side. Change profile bearing. In connection with lame bearing, lengthwise bearing may be at the large or small end on one or both sides, in which case change in lengthwise bearing is necessary.

FIG. 107.—Examples of different tooth bearings.

Adjustment of Bevel Gears in Assembly.—The proper adjustment of bevel gears in assembly is a vital factor in obtaining quiet and durable gears.

There are two distinct considerations in obtaining the proper tooth contact; one is the bearing along the tooth, lengthwise bearing, and the other is the bearing up and down the tooth or profile bearing. It is essential that the two be considered separately to obtain the proper results in combination.

In connection with Fig. 107 will be found graphic definitions of the terms used in describing the proper procedure to mount a pair of bevel gears. There is no difference in the method of adjusting spiral or straight bevels, or hypoids and while the following statements are particularly applicable for spiral bevels, they are also true for straight bevels and hypoids.

Bevel gears are cut with a predetermined amount of backlash to suit the pitch and operating conditions. This backlash should not be altered by any great amount to obtain the proper tooth contact, as the necessity of such a step indicates a fault either in the cutting or in the alignment of the supporting bearings. The usual amount of backlash is from 0.004 in. on eight pitch gears to 0.008 in. on three pitch. In the case of hypoids it is especially important that the backlash adjustment be carefully made.

Bevel gears mounted in a rigid testing machine should show a bearing toward the small end of the tooth. The amount the bearing favors the small end is determined experimentally by the stiffness of the mounting in which the gears are to be finally assembled. Any spring in the mounting of the gears under load will cause the bearing to move toward the large end. In no case should the bearing be heavy at the large end of the tooth under the operating load. Any extra load, such as induced by suddenly applying the full load, will cause the bearing to become concentrated on the top corner of the large end of the tooth and breakage will ensue. Figure 108 shows typical tooth bearings as obtained in a testing machine and when mounted in an automobile axle; also the effect of spring in the mounting on the apexes of the gear and pinion. It will be observed that the apexes do not coincide when the gear mounting has sprung under the working load. Therefore, when cutting the gears, it is necessary to make provision for this lift or spring.

Bevel gears are cut either to run flush at the large end or to the correct distances from the center. Mounting distances at which the gears run best on the testing machine, may be used

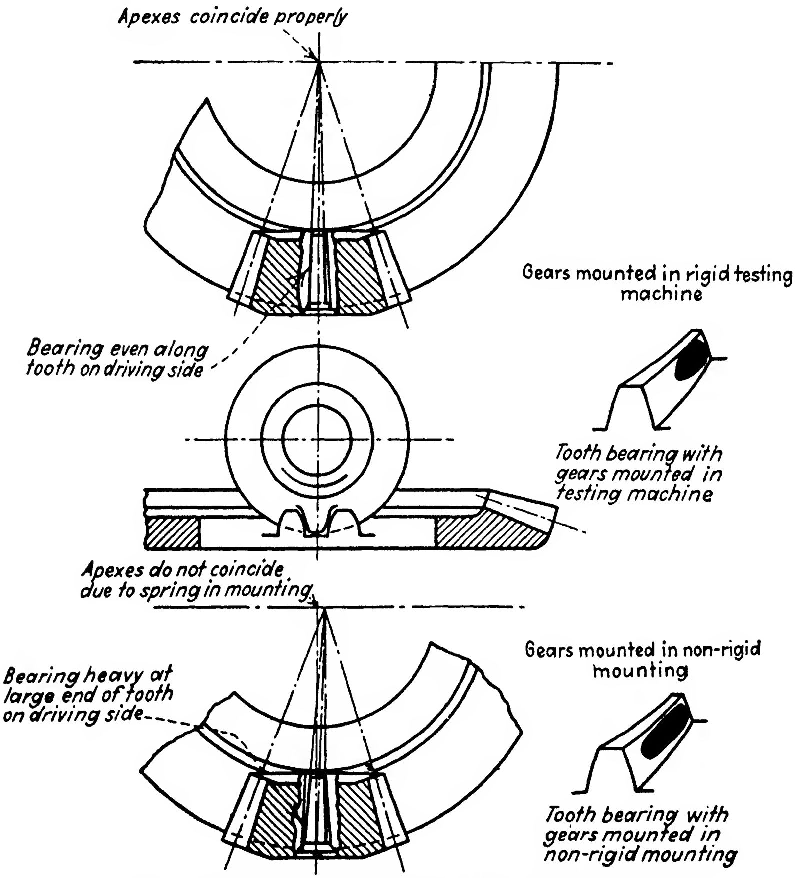


FIG. 108.—Definitions of terms used in bevel-gear tooth bearings.

in the final assembly to locate the gears for the first trial. Without the mounting distances it will be necessary to assemble the gears to run flush at the back angle, after which readjustments are made until the gears run satisfactorily. Powdered red lead and any light machine oil should be mixed and spread over the working surfaces of the teeth with a brush to show clearly the tooth contact obtained.

After mounting the gears with the proper amount of backlash, they should be operated under load in each direction for a minute. In the case of the automobile rear axle driving gears, the rear axle should first be raised to have the wheels clear the floor. Next start the motor and drive the wheels in both directions with the brakes applied to obtain the necessary load.

The data and illustrations that follow are of special interest in the automobile field. Beginning with the diagram for correct

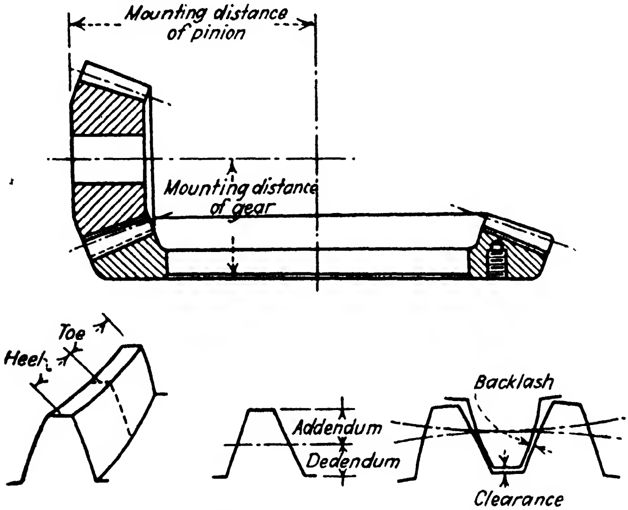


Fig. 109.—Adjustment of bevel-gear assembly.

assembly of gear and pinion, Figs. 109 and 110, the explanations make all points clear.

All figures show the bearing on the gear tooth. With a right-hand spiral-bevel gear, (mating with a left-hand spiral pinion) mounted in an automobile, the driving side is on the convex side of the tooth and the concave side of the tooth is used when in reverse.

The tooth bearing, both lengthwise and profile, should appear as shown in *A* and *B*, but a condition of tooth contact may be obtained as indicated in Figs. *C* to *N*. The lengthwise bearing adjustments will be considered first.

Lengthwise Bearing Adjustments.—Figures *C* and *D* show what is called a cross bearing; it is caused by either a misalignment of the mounting or an error in the cutting. The mounting should be tested and if found faulty, should be corrected. If the drive side has a toe bearing and the reverse a heel bearing, the gears are serviceable provided the bearing is about 5/8 of the tooth length, but if the heel bearing occurs on the drive side, it should not be used and the cutting conditions should be altered if the mounting is found correct.

Figures *E* and *F* show a toe bearing on each side of the tooth. Increase the mounting distance of the gear to increase the length-

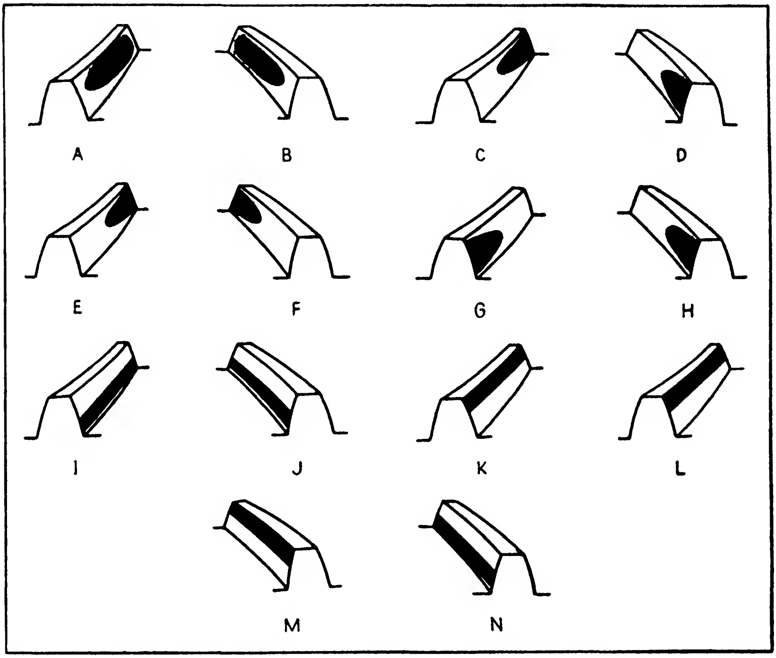


Fig. 110.—Tooth bearings formed in spiral-bevel gears.

wise bearing, which on ratios 1 to 1 to approximately 4 to 1, will change the profile bearing to some extent. An adjustment of the pinion may be required as described under Profile Bearing. This movement of the gear will introduce more backlash and the gear cutting should be changed properly to locate the bearing if this increase in backlash becomes excessive.

Figures *G* and *H* show a heel bearing on both sides and mounting distance of the gear should be decreased to increase the lengthwise bearing, which on ratios 1 to 1 to approximately 4 to 1, will change the profile bearing to some extent and an adjustment of the pinion may be required as described under Profile Bearing. This movement will decrease the backlash and the gear cutting should be changed properly to locate the bearing if there is insufficient clearance.

Profile Bearing.—Figures *I* and *J* show a low bearing on gear tooth which may appear at any position along the tooth. The mounting distance of the pinion should be increased and on ratios 1 to 1 approximately 4 to 1, the mounting distance of the gear should be decreased to maintain the proper backlash. This movement of the gear will alter the lengthwise bearing and several adjustments, for both lengthwise and profile bearing, may be required to obtain the proper tooth bearing.

Figures *K* and *L* show a high bearing on gear tooth which may appear at any position along the tooth. The pinion mounting distance should be decreased, and on ratios 1 to 1 to approximately 4 to 1 the gear mounting distance should be increased to maintain the backlash. This movement of the gear will alter the lengthwise bearing and several adjustments for both lengthwise and profile bearing may be required to obtain the proper tooth bearing.

Figures *M* and *N* show a lame bearing. It is possible to adjust the gears and obtain a fair driving condition as indicated in Fig. *A*, but a poor coast or reverse, and the only method of completely eliminating the trouble is to cut the gears properly.

It must be borne in mind that the adjustments cited should be moderate and if great amounts of adjustments are needed the mounting and gear cutting must be carefully checked, and the necessary steps taken to correct the trouble in the manufacture of the gears or mounting.

Chucks and Mandrels for Bevel Gears.—In making chucks and mandrels for holding gear blanks in Gleason machines, the makers recommend that they be provided with shoulders as shown in Table 45. The shoulder should be drawn firmly against the spindle nose when the mandrel is tightened. This shoulder acts as an additional support when the gears are being cut and prevents the tapered shank from being drawn so far into the tapered bore of the spindle as to swell the spindle and damage the machine.

In the table given with the suggestion for chuck design, the amount *A* is the distance the shoulder should stand away from the spindle nose when the mandrel is wrung into the spindle by hand. This is the distance the mandrel should draw into the taper when it is tightened in place.

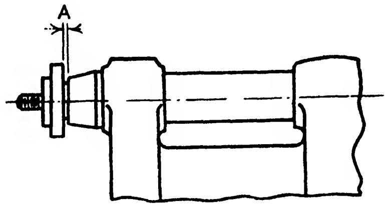


TABLE 45.—MANDRELS FOR GLEASON MACHINES

Taper per foot, in.	Diameter at large end, in.	Amount of draw *A*, in.
½	1.066	0.006 to 0.008
½	1.796	0.012
½	2.292	0 012
½	2.615	0.012
½	3.427	0 012
0.606	3.906	0.010
⅝	3.4375	0.010
⅝	5.047	0.010
⅞	2.432	0.007
⅞	2.937	0.007

CHAPTER VII

WORM AND WORM GEARING

Worm gearing also plays an important part in modern mechanisms. It involves problems not found in gearing of the other types. Worm gearing combines the use of a screw and a worm wheel, which is a helical gear in which the face is concave to match the outside of the screw, or worm. This type of gearing does not always have a concave face nor the teeth at the angle of the worm, the Sellers drive for planers being an excellent example. Here the gear teeth in the rack are straight and the worm is set at the angle of the lead of the thread, which brings the shaft that drives the worm out at the side of the planer bed so that it can be driven without interfering with the table travel. The usual worm wheel is concaved to fit the worm so as to increase the bearing surface, but the worm and rack in the Sellers planer are noted for their long life.

Another type of worm gearing, known by various names, has the worm in the hourglass shape in order to contact more of the surface of the worm than can be done with the straight worm. This type is known as the Hindley, the hourglass, and as the Cone worm, the last referring to the name of the designer and not to its shape. Much discussion has been waged as to the advantages of the different types. Each type has stanch advocates and the various types are in use with complete satisfaction in numerous cases.

Worm wheels are usually cut with hobs having teeth that will produce the required shape on the rim of the worm wheel as it revolves under the rotating hob. It is customary first to gash, or rough-cut, the wheels instead of trying to finish the teeth at one cut. The difference between hobbing a worm wheel and hobbing a spur or helical gear is that in the latter the hob moves across the face of the gear as it cuts. In the worm wheel neither work nor hob has any side movement. Worm wheels are sometimes cut with a fly cutter in place of a hob.

Cutting Worm Wheels.—There are two ways of producing worm wheels on a milling machine. The first and more commonly used consists briefly of two operations: gashing the teeth, and finishing with a hob, as in Figs. 111 and 112.

Finishing a worm wheel on a milling machine by the "gash and hob" method requires two separate operations. First, the

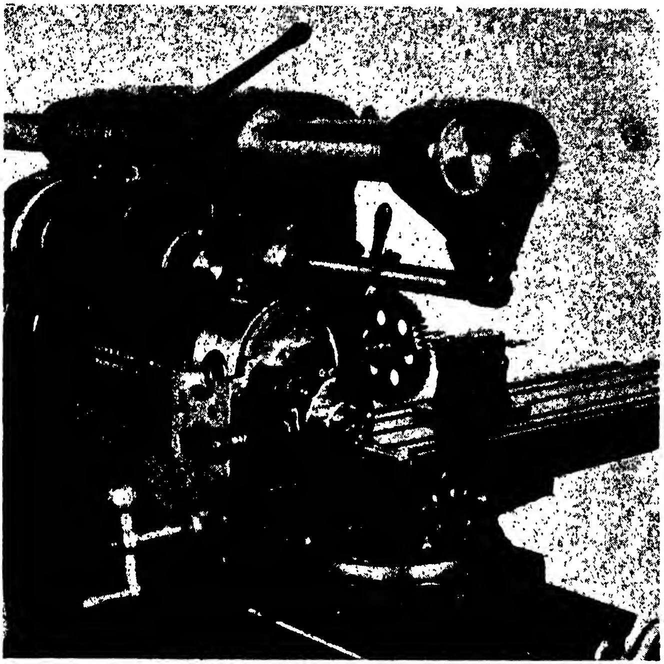


FIG. 111.—Gashing worm-wheel teeth in milling machine.

operation of gashing the teeth, shown above, is performed; and then the teeth are hobbed, as shown in the illustration on the page following.

In gashing the teeth, the blank is dogged to the headstock spindle, and the swivel table is swung to the required angle. The vertical feed is used and the teeth are indexed the same as in cutting a spur gear. Most of the stock is removed in gashing, only enough being left to allow the hob to take a light finishing cut.

The work is then set up practically the same as in the operation of gashing the teeth, only the dog on the arbor is removed and the swivel table is set at zero. The worm wheel revolves freely on the centers, being rotated by the hob.

The wheel can be hobbed to the right depth by using a steel rule at the back of the knee to measure a distance equal to the center distance of the worm and wheel from a line marked "center," on the vertical slide to the top of the knee. This line on

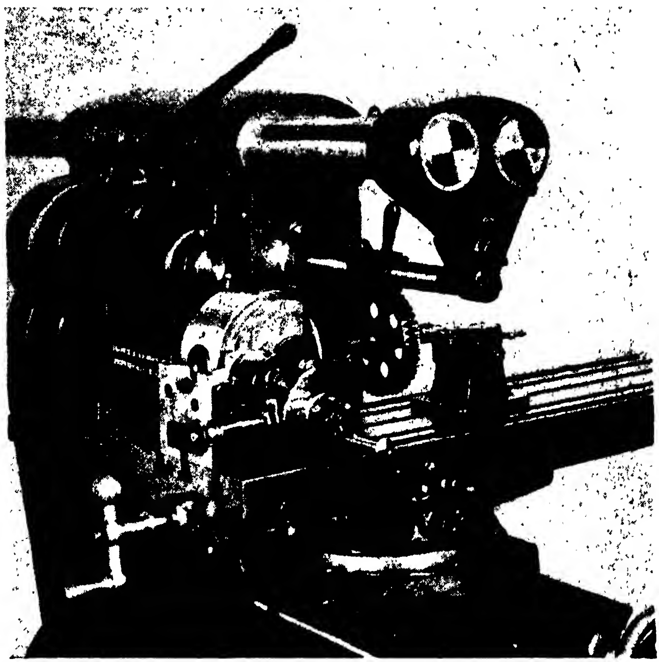


FIG. 112.—Miller set for hobbing worm wheel.

the vertical slide indicates the position of the top of the knee when the index centers are at the same height as the center of the machine spindle.

The other method, used by some for doing this type of work, is described by the Brown and Sharpe Manufacturing Company for use with the milling machine. It should not be assumed that the milling machine is suited to the production of worm wheels on a large scale, for the milling-machine method is comparatively slow; or, that a high degree of accuracy may be obtained on every wheel, because, in many cases, the best result obtainable on a milling machine is an approximation. On the other hand, some worm wheels may be hobbed to very close limits by this method. Although the occasions when a milling machine may be used profitably for such work are undoubtedly few, its ability, should the necessity arise, to substitute for a hobbing machine, even in a small and limited measure, may prove a decidedly valuable feature.

The difficult part about hobbing a worm wheel successfully in a milling machine is in synchronizing the rotation of the spindle and the rotation of the work. The mechanical skill involved in setting up the machine is insignificant to any experienced operator. The worm wheel to be cut is held on a mandrel and mounted between the headstock and footstock of the universal index centers. The mandrel is driven by a dog in the same manner used to cut helices. The work is brought into the proper position in relation to the hob, which is mounted on the cutter arbor, and the table feed screw is disengaged. To insure against longitudinal movement of the table, stops should be set so that the table is locked securely in position. Change gears are selected that will give the required rotative movement to the work, the knee is raised vertically, by hand, and the worm wheel is fed upward into the hob until the elevating screw dial, or the vertical scale and vernier, depending upon which method of measurement is used, indicates that the proper depth has been reached.

The difficulty encountered in hobbing worm wheels on a milling machine is in obtaining the right relationship between the rotation of the machine spindle and the rotation of the work. It is apparent that a 40-tooth worm wheel, being cut by a single-threaded hob, must make *exactly* one revolution to *exactly* 40 revolutions of the hob; and this ratio must be maintained when the arbor is revolving at the available spindle speed that most nearly provides a correct surface speed for the hob. If the ratio, of 40 to 1 in this particular case, changes ever so slightly, it can be seen that, if the worm wheel is allowed to rotate for an indefinite length of time after the hob has reached the proper depth, the teeth will be thinned a little at each revolution, and, eventually, all the teeth in the worm wheel will be gone. The number of revolutions it would take to do this, or to thin the teeth to a point where the wheel is too inaccurate for use, depends upon the amount of variation from the proper ratio of hob and worm wheel. In most cases the ratio may be held closely enough so that the work will be commercially satisfactory. In some instances an exact ratio may be found, in which cases a number of extra revolutions made by the worm wheel is unimportant, because the teeth of the hob will track in the worm wheel at exactly the same places every time.

To obtain the desired relationship between the speed of the hob and the speed of the work, two things must be known, both of which must be found by figuring from the speed of the main driving shaft through the machine gearing; first, the exact number of revolutions per minute of the machine spindle for each available combination of spindle gears, and second, the exact feed of the machine table longitudinally, with each combination of gears available in the feed mechanism. This latter requirement is necessary because, with the table screw disconnected, the feed in inches per minute is translated into revolutions per minute of the driving gear in the train on the universal index centers. Assuming that the speed of the hob is known, and that the speed of the shaft from which the headstock is driven is also known, it follows that variations in the speed of the work, to obtain the right ratio between the speeds of hob and work, may be made by the selection of change gears for the headstock. And it is in the selection of these change gears that most of the figuring to get the required ratio between hob and worm wheel is done. We say "most of the figuring" advisedly, because, although a spindle speed is usually selected arbitrarily to provide a reasonably correct cutting speed for the hob, sometimes it is necessary, after unsuccessful attempts to obtain a set of change gears to use with that spindle speed, to select another, and start over again. Likewise, it may be necessary to try more than one table feed before a satisfactory ratio is found.

In regard to speeds and feeds, it must be understood that the plates listing the ranges of spindle speed and table feed furnished with the machines, while accurate enough for any milling purpose, are much too inaccurate to permit hobbing. The speeds and feeds for this purpose must be computed from the charts of gearing of the machine to be used, and the results should be carried out to five or six decimal places. On the Brown and Sharpe No. 2A standard universal milling machine, for example, the fastest spindle speed listed on the dial of the machine is 496 r.p.m. Computed from the gearing in the spindle mechanism, and carried out to five places, this speed is 496.36356. Similarly the fastest feed listed is 19⅞ in. per minute; carried out to six places it is 19.873270.

Returning to the selection of the change gears for the headstock, two formulas can be used to aid in this selection.

1. When driving directly to spindle of indexing head.

$$\frac{\text{Drivers}}{\text{Driven}} = \frac{\text{r.p.m. of machine spindle} \times \text{ratio required between headstock spindle and machine spindle}}{4 \times \text{feed in inches per minute}}$$

2. When driving through worm and wheel of headstock.

$$\frac{\text{Drivers}}{\text{Driven}} = \frac{40 \times \text{r.p.m. of machine spindle} \times \text{ratio required between headstock spindle and machine spindle}}{4 \times \text{feed in inches per minute}}$$

Both of these formulas may be written in another form which may be more readily understood by students of gearing.

1. $$\frac{\text{Drivers}}{\text{Driven}} = \frac{\text{r.p.m. of machine spindle} \times \text{No. threads in hob}}{4 \times \text{feed in inches per minute} \times \text{No. teeth in worm wheel}}$$

2. $$\frac{\text{Drivers}}{\text{Driven}} = \frac{40 \times \text{r.p.m. of machine spindle} \times \text{No. threads in hob}}{4 \times \text{feed in inches per minute} \times \text{No. teeth in worm wheel}}$$

In each formula, 4 × feed in inches per minute is used to translate the longitudinal movement of the table into rotative movement in revolutions per minute. The table screw has four threads to the inch. Consequently, four times the feed in inches per minute equals the speed of the table driving shaft, from which the headstock is driven. The screw, of course, is disconnected for hobbing, so that there is no longitudinal movement.

The usual method employed in solving these equations for the proper change gears is to resolve the term to the right of the equality sign into its decimal equivalent. From this point a table of logarithms and of logarithmic gear ratios are used, and the change gears are chosen as demonstrated below.

Compound Ratios.—In this example we have assumed that only four change gears are required. Six gears may be found in the same manner by selecting two arbitrary ratios as a starting point, instead of the single ratio selected for a four-gear train.

Let the ratios be represented by:

$$\frac{A}{B} \times \frac{C}{D} = R \qquad \text{or} \qquad \frac{A}{B} = R \div \frac{C}{D}$$

in which $A/B \times C/D$ are the required change gears, and R represents the decimal equivalent ratio obtained from solving the right-hand side of our first equation, as outlined in a preceding paragraph.

The ratio C/D is chosen arbitrarily, and if the ratio A/B, as found, results in too great an error in the final result, another ratio for C/D is tried. It is advisable, when it is necessary to choose another ratio, to take the one next to the first one that was chosen, and to continue until one is found that will yield a result sufficiently accurate.

Supposing that R in the equation above equals 0.7348, and that four gears must be found whose ratios will equal that number. Working with logarithms, and remembering that to perform division, the logarithm of the denominator is subtracted from the logarithm of the numerator, it is apparent that, having assumed an arbitrary ratio of 34/79 for C/D,

$$\begin{aligned} \log R &= -1.866169 \\ \log \frac{34}{79} &= -1.633852 \\ \hline \text{Difference} &= 0.232317 \end{aligned}$$

Nearest log from table of logarithmic gear ratios = 0.232314 = 70/41, the value of A/B.

If, in this example, the first ratio tried had been 24/56 = log − 1.632023, the nearest corresponding ratio would then have been log 0.234083 = 60/35. To check for the error

$$\frac{24}{56} \times \frac{60}{35} = 0.73469$$

$$\begin{aligned} \text{Number required} &= 0.7348 \\ \text{Number found} &= 0.73469 \\ \hline \text{Error} &= 0.00011 \end{aligned}$$

which is too great. Similarly, the next consecutive ratio 89/52 and its corresponding ratio 24/56 are also decided to be not close enough, so the process is continued until the ratio $34/79 \times 70/41 = 0.734795$ is found, which has an error of only 0.000005. This figure, it must be remembered, is only the ratio error, and, to get a true picture of whether or not the gears selected are satisfactory, this must be translated into the amount of circumferential error that will result on the pitch line of the worm wheel, per revolution. The error per revolution equals:

> The difference between the ratio wanted and the ratio obtained $\times$ pitch circumference of the worm wheel.

While the method of selecting change gears described above is a choose-and-try method, and may necessitate several calculations, it is possible in most cases to find gears that will give very close results. To do this sort of work, it is, of course, necessary to have or make change gears in addition to those furnished with the universal index centers.

Example.—Required to hob a worm wheel with 40 teeth, ½-in. circular pitch, with a single-threaded hob 3 in. in diameter. Referring to the preceding explanation, we first select a spindle speed that will give a reasonably correct cutting speed for the hob. From our list of spindle speeds, figured from the gearing in the speed train, we take 116.61211 as the most acceptable. Next we choose a feed, also figured from the machine gearing: in this case 19.873270. Then, according to our formula,

$$\frac{\text{Drivers}}{\text{Driven}} = \frac{40 \times 116.61211 \times 1}{4 \times 19.873270 \times 40}$$
$$= \frac{116.61211}{79.493080}$$
$$= 1.4669466$$

To find the change gears that will most nearly give this ratio, we use the compound logarithmic formula noted above, in which

$$\frac{A}{B} \times \frac{C}{D} = R$$

and

$$\frac{A}{B} = R \div \frac{C}{D}$$

Using the value 1.4669466 for R

$$\log \frac{A}{B} = \log 1.4669466 - \log \frac{C}{D}$$

If, for $\frac{C}{D}$, we take an arbitrary ratio of $\frac{100}{69}$, then

$$\log \frac{A}{B} = 0.1664123 - 0.161151 = 0.005261$$

The nearest logarithm to this figure found in the table of logarithmic gear ratios is 0.005264, or a ratio of $\frac{83}{82}$.

Checking this result,

$$\frac{100}{69} \times \frac{83}{82} = \frac{8300}{5658} = 1.4669494$$

$$\begin{array}{ll} \text{Ratio found} & = 1.4669494 \\ \text{Ratio required} & = 1.4669466 \\ \hline \text{Difference} & = 0.0000028 \end{array}$$

This error is the error in the ratio. To see the real effect on the work, per revolution,

difference $\times$ pitch circumference of worm wheel $= 0.0000028 \times 20 =$ 0.0000560

Cutting Small Worm Wheels.—A makeshift method of cutting small worm wheels is credited to Nils Peter Johnson by T. B. Wilde. The illustrations, in Fig. 113 make the plan clear. The hob is held between centers and driven by a lathe dog. The hob is made from a threaded piece similar to the worm to be used with the flutes cut with a hack saw, the teeth backed off very little, and the screw hardened enough to cut the stock used for the worm.

The gear blank is held on a vertical mandrel that allows it to revolve freely as the hob revolves. It is fed against the hob by the cross-feed screw of the lathe. A pin or bolt on which the gear blank can revolve without binding or shake is all that is needed. This shows the worm-gear blank held between two square pieces drilled for the bolt, and clamped in the tool post. The worm-gear blank can be turned to the proper radius or it can be left square across the edge. On small gears this is of little consequence.

Cone Worm Gears.—Among the advantages claimed for the Cone type of worm are much greater working contact, less unit pressure per tooth because of more teeth being in contact, less bending of the worm shaft, and lower cost. In a six-threaded worm, for example, a line contact on four of the teeth is claimed for this type of worm. A Cone worm-gear set, with the hob and cutter used, is shown in Fig. 114. A "Cone" hob is shown at work in Fig. 115.

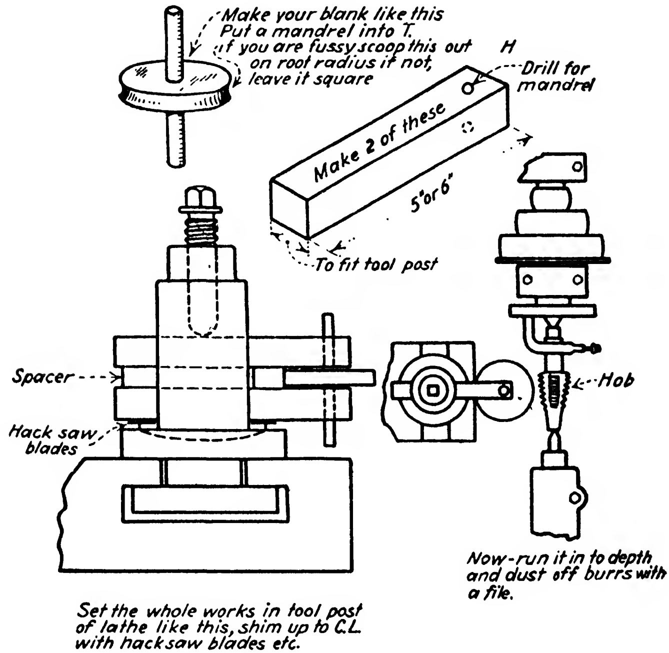


FIG. 113.—Cutting small worm wheels in a lathe.

Since the development of Cone hourglass worm gearing the Michigan Tool Company has gone into volume production,

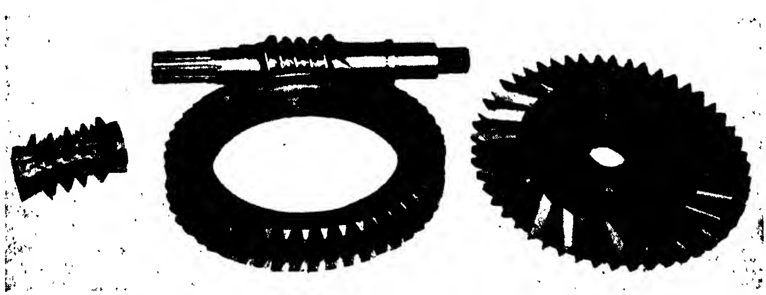


FIG. 114.—A Cone worm gear with hob and cutter used.

necessitating the building of special machinery and testing equipment. For large scale manufacture Gould & Eberhardt hobbing machines employ two hobs per workpiece, fed into the work from opposite directions. When the correct depth is reached, one hob is rotated in one direction to cut the tooth form on the front of the tooth and the other is rotated in the opposite direction to form the flank on the back of the tooth. Production rates as high as 270 pieces per hour have been attained. For small production a special fly cutter is used.

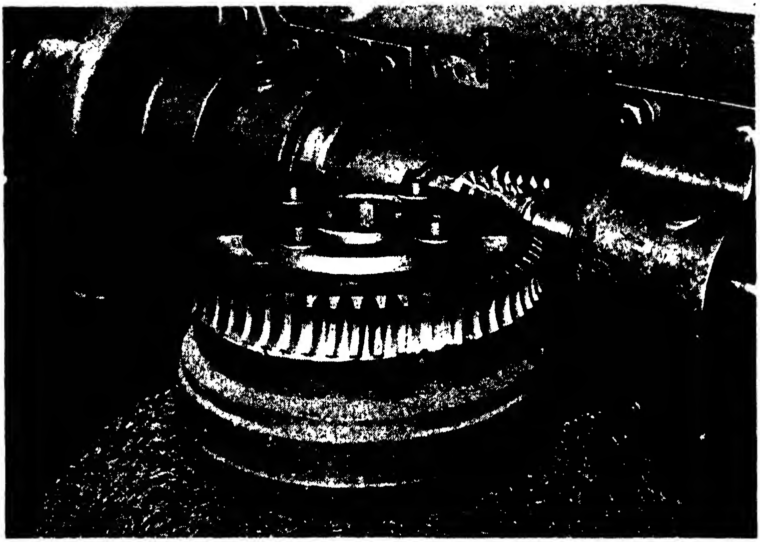


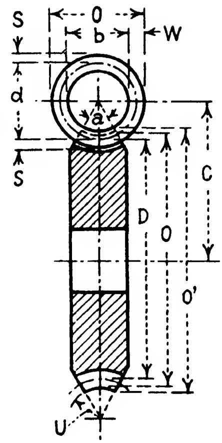
FIG. 115.—Cone hob cutting a worm wheel.

Because of the large contact between the worm and wheel in Cone gearing, unit pressures are low and lubrication is easily maintained. The gears are said to grow more quiet with use, but where exceptional quietness is specified at the start lapping on a special machine is done. In one automotive application a speed of 30,000 r.p.m. is encountered.

Rules and Formulas for Worm Gears.—To compute the necessary dimensions for a worm-gear drive the following formulas should be used with figure on next page and Table 46.

P = circular pitch of wheel and linear pitch of worm.
l = lead of worm.
n = number of threads in worm.
S = addendum.

d = pitch diameter of worm.
D = pitch diameter of worm wheel.
o = outside diameter of worm.
O = throat diameter of worm wheel.
O' = diameter of worm wheel over sharp corners.
b = bottom diameter of worm.
N = number of teeth in worm wheel.
W = whole depth of worm tooth.
T = width of thread tool at end.
B = helix angle of worm.
$-B$ = gashing angle of worm wheel.
U = radius of curvature of worm-wheel throat.
C = center distance.



To Cut Worm Wheels on Hobbing Machine. *For Single-thread Worm Wheels.*—The chart and Table 47 show a typical example of the way in which change gears are used. This is a Gould & Eberhardt chart.

It is always well before starting to cut into any gear just to allow the hob to scrape the surface and count the number of marks to make sure that the machine is geared up correctly.

The diagrams of gearing with the hobber can be followed when gearing the machine for spur-gear or worm wheels. The infeed gears are then selected and put in place. The hob cutter is secured to the arbor after the end bearing is secured in position.

The cutter slide is adjusted to obtain the relation that the center line of the worm is to bear to the axis of the work wheel to be cut. The gears to rotate the cutter in the proper direction are then engaged by means of the handle on the pulley side of the machine. The infeed gears must be arranged with or without intermediate gear, depending upon whether the worm gear is right- or left-hand, so that the work will feed toward the cutter. The work is secured to the mandrel and tested as in spur gears.

Select a suitable hob that will correspond to the worm which will run in the worm wheel.

The set screw on offside of cutter carriage should be clamped to prevent vertical movement of carriage. The blank is adjusted until it just touches the hob cutter. The infeed is then engaged on the front end of the machine. Care should be taken that the down feed is disengaged.

TABLE 46.—RULES AND FORMULAS FOR WORM GEARS

To find	Rule	Formula
Linear pitch	Divide the lead by the number of threads. It is understood that by the number of threads is meant, not number of threads per inch, but the number of threads in the whole worm—one, if it is single-threaded; four, if it is quadruple-threaded, etc...	$P = 1/n$
Addendum of worm tooth	Multiply the linear pitch by 0.3183.	$S = 0.3183P$
Pitch diameter of worm	Subtract twice the addendum from the outside diameter	$d = o - 2S$
Pitch diameter of worm wheel	Multiply the number of teeth in the wheel by the linear pitch of the worm, and divide the product by 3.1416	$D = NP/3.1416$
Center distance between worm and gear	Add together the pitch diameter of the worm and the pitch diameter of the worm wheel, and divide the sum by 2	$C = \frac{D + d}{2}$
Whole depth of worm tooth	Multiply the linear pitch by 0.6866	$W = 0.6866P$
Bottom diameter of worm	Subtract twice the whole depth of tooth from the outside diameter	$b = o - 2W$
Helix angle of worm	Multiply the pitch diameter of the worm by 3.1416, and divide the product by the lead; the quotient is the tangent of the helix angle of the worm.	$\text{Tan } B = \frac{3.1416d}{l}$
Width of thread tool at end	Multiply the linear pitch by 0.31	$T = 0.31P$
Throat diameter of worm wheel	Add twice the addendum of the worm tooth to the pitch diameter of the worm wheel	$O = D + 2S$
Radius of worm-wheel throat	Subtract twice the addendum of the worm tooth from half the outside diameter of the worm...	$U = \frac{o}{2} - 2S$
Outside diameter of worm	Add together the pitch diameter and twice the addendum	$o = d + 2S$
Pitch diameter of worm	Subtract the pitch diameter of the worm wheel from twice the center distance	$d = 2C - D$
Diameter of worm wheel to sharp corners	Multiply the radius of curvature of the worm-wheel throat by the cosine of half the face angle, subtract this quantity from the radius of curvature, multiply the remainder by 2, and add the product to the throat diameter of the worm wheel	$O' = 2(U - \cos\frac{\infty}{2}U) + O$
Gashing angle of gear	Divide the lead of the worm by the circumference of the pitch circle. The result will be the tangent of the gashing angle	$\text{Tan } (90° - B) = l/\pi d$

The proper speed for the cutter must be provided for, and then the machine is started. The dog on the horizontal rod can be set to disengage the infeed when the blank is cut to the proper depth. A vernier scale is provided to give the exact distance between the center of the hob and the wheel blank.

TABLE 47.—SPUR, HELICAL, AND WORM GEAR CHART—GOULD AND EBERHARDT

N	A	B	C	D	N	A	B	C	D	N	A	B	C	D
6	75	30	40	60	26	40	78	60	80	46	40	60	30	92
7	90	30	40	84	27	25	90	60	45	47	40	60	30	94
8	75	30	40	80	28	45	90	60	84	48	40	60	30	96
9	75	45	40	60	29	40	87	60	80	49	40	70	30	84
10	80	40	30	60	30	40	60	45	90	50	30	60	30	75
11	60	30	40	88	31	40	93	60	80	51	40	60	25	85
12	75	45	40	80	32	45	60	40	96	52	60	78	24	96
13	45	78	80	60	33	60	88	40	90	53	40	53	24	96
14	45	84	80	60	34	60	85	40	96	54	40	60	25	90
15	40	80	60	45	35	30	84	60	75	55	40	60	24	88
16	45	96	80	60	36	60	96	40	90	56	30	80	40	84
17	75	45	30	85	37	30	60	40	74	57	40	60	25	95
18	75	45	30	90	38	75	95	30	90	58	60	87	24	96
19	60	30	25	95	39	40	60	30	78	59	30	59	30	90
20	60	40	30	90	40	40	60	30	80	60	40	60	24	96
21	30	84	60	45	41	40	60	30	82	61	40	61	24	96
22	30	88	80	60	42	40	60	30	84	62	30	80	40	93
23	30	92	60	45	43	40	60	30	86	63	25	75	40	84
24	30	90	75	60	44	40	60	30	88	64	25	80	45	90
25	45	90	60	75	45	40	60	30	90	65	30	75	30	78

N	A	B	C	D	N	A	B	C	D	N	A	B	C	D
66	30	88	40	90	85	30	85	30	90	106	24	120	25	53
67	30	67	30	90	86	30	86	30	90	108	24	81	30	96
68	25	85	45	90	87	30	87	30	90	110	24	75	25	88
69	30	90	40	92	88	30	88	30	90	111	24	74	25	90
70	24	84	45	90	89	30	89	30	90	112	24	84	30	96
71	40	71	24	96	90	24	90	40	96	114	25	90	30	95
72	30	60	25	90	91	24	78	30	84	115	24	90	30	92
73	40	73	24	96	92	30	90	30	92	116	24	80	25	87
74	40	74	24	96	93	30	90	30	93	117	24	78	25	90
75	30	75	30	90	94	30	90	30	94	118	25	59	24	120
76	25	80	40	95	95	30	90	30	95	119	24	84	25	85
77	24	84	40	88	96	30	90	30	96	120	24	90	30	96
78	30	78	30	90	97	30	90	30	97	121	25	88	32	110
79	40	79	24	96	98	25	70	24	84	122	25	61	24	120
80	30	60	24	96	99	40	90	25	110	123	24	82	25	90
81	25	75	30	81	100	24	80	30	90	124	24	93	30	96
82	30	82	30	90	102	25	85	30	90	125	30	75	24	120
83	40	83	24	96	104	24	78	30	96	126	24	84	25	90
84	30	84	30	90	105	24	84	30	90	128	24	80	25	96

N	A	B	C	D	N	A	B	C	D	N	A	B	C	D
129	24	86	25	90	153	24	85	25	108	176	24	88	25	120
130	30	78	24	120	154	24	84	25	110	177	24	90	25	118
132	24	88	25	90	155	30	93	24	120	178	24	89	25	120
133	24	84	25	95	156	24	78	25	120	180	24	90	25	120
134	24	67	25	120	158	24	79	25	120	182	24	91	25	120
135	30	81	24	120	159	24	90	25	106	183	24	90	25	122
136	24	85	25	96	160	24	80	25	120	184	24	92	25	120
138	24	90	25	92	161	24	92	25	105	185	30	111	24	120
140	30	84	24	120	162	24	81	25	120	186	24	93	25	120
141	24	90	25	94	164	24	82	25	120	187	24	102	25	110
142	24	71	25	120	165	24	90	25	110	188	24	94	25	120
143	24	78	25	110	166	24	83	25	120	189	24	108	25	105
144	24	90	25	96	168	24	84	25	120	190	24	95	25	120
145	30	87	24	120	169	30	104	24	117	192	24	96	25	120
146	24	73	25	120	170	24	85	25	120	194	24	97	25	120
147	24	90	25	98	171	24	95	25	108	195	30	117	24	120
148	24	74	25	120	172	24	86	25	120	196	24	98	25	120
150	24	75	25	120	174	24	87	25	120	198	25	110	24	108
152	24	96	25	95	175	24	84	24	120	200	24	96	24	120

Table 47.—Spur, Helical, and Worm Gear Chart.—(*Continued*)

Gear

N = No. teeth to be cut
α = Helix angle with axis

Index gears

$$\text{Spur gears} = \frac{10T}{N} = \frac{A \times C}{B \times D} = \frac{\text{driver}}{\text{driven}}$$

$$\text{Helical gears} = \frac{10TC}{NC - 1} = \frac{A \times C}{B \times D} = \frac{\text{driver}}{\text{driven}}$$

Notation and formulae

P^{NC} = Normal circular pitch
P^{L} = linear pitch
F = feed per rev. of gear
C = feed constant

$$C = \frac{P^{NC}}{F \times \sin \alpha} = \frac{P^{L}}{F}$$

Hob setting angle

$$\phi = \alpha - \beta$$

Hob

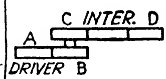
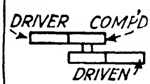
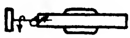
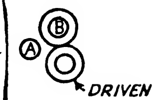
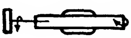
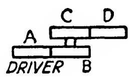
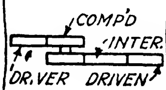
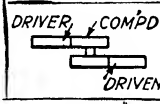
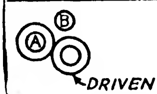
T = No. threads in hob
β = Angle marked on hob

Feed gears

$$\text{Spur gears} = \frac{8F}{1} = \frac{\text{driver}}{\text{driven}}$$

$$\text{Helical gears} = \frac{8\,P^{NC}}{C \sin \alpha} = \frac{\text{driver}}{\text{driven}}$$

Hob and Change Gear Arrangement

Gear	Hob	Rotations	Index	Feed	Hob speeds
Spur	R.H.				
Worm gear	R.H.				
	L.H.				
L.H. Helical	L.H.				
R.H. Helical	R.H.				

Feeds

Feed per rev.	Dr.	Comp'd		Drn.
0.03	30	75	48	80
0.04	30	60	48	75
0.05	48	80	60	90
0.06	48	75	60	80
0.07	48	75	70	80
0.08	48	60	60	75
0 09	48	80	90	75
0.100	72	60	60	90
0.115	60	80	90	72
0.135	72	80	80	90
0.155	60	60	90	72
0.175	70	60	72	60
0.195	72	60	90	70
0.220	70	48	90	75
0.250	72	48	80	60

$$\frac{8F}{1} = \frac{\text{driver}}{\text{driven}}$$

Change gears Limited equipment

Index, feed, and speed one combination for each.

Full equipment

Index and feed

24 to 87 including 2-24, 2-25, 2-30, 2-40, 2-45, 2-60, 2-90, 2-96, and 108, 110, 2-120.
Total 87 gears.

Speed

37, 41, 45, 49, 53, 57.

Speeds special

21, 25, 29, 33, 61, 65, 69, 73.

Infeeds

Feed per rev.	Dr.	Comp'd		Drn.
0 003	30	80	32	89
0 004	30	82	48	97
0 005	30	88	53	80
0.006	30	89	60	75
0.007	30	75	59	75
0.008	30	75	61	67
0.010	45	80	60	75
0.013	60	60	48	82
0 016	45	75	72	60
0 020	45	60	72	60

$$\frac{45F}{1} = \frac{\text{driver}}{\text{driven}}$$

Automatic trip, stops machine at centers set—disengage feed—machine clutch engaged for several revolutions of blank.

Speeds

R.p.m.	Dr.	Drn.
38	21	73
48	25	69
60	29	65
72	33	61
87	37	57
105	41	53
125	45	49
145	49	45
175	53	41
205	57	37

$$\frac{\text{R.p.m.}}{133} = \frac{\text{driver}}{\text{driven}}$$

The work binding nuts must be adjusted so that the table is a nice sliding fit. They must not be adjusted too tightly.

Multiple-thread Worm Wheels.—To cut multiple-thread worm wheels use the formula:

$$\frac{60 \times \text{No. of threads in worm}}{\text{No. of teeth}} = \frac{\text{driver} \times \text{compound}}{\text{compound} \times \text{worm}}$$

and gear up the machine accordingly.

Example.—To cut a worm wheel having 67 teeth for a triple-thread worm,

$$\frac{60 \times 3}{67 \times 1} = \frac{60 \times 90}{67 \times 30}$$

When worm-wheel hobs are sharpened, the reduction in diameter will change the helix angle. This is especially apparent when using multiple-thread hobs and is of a sufficient amount to necessitate a correction in order to maintain the proper bearing between the worm and worm wheel.

There are two ways to correct this difference, namely:

1. Swivel the hob spindle slightly so that the hob is set at the proper angle to generate the correct helix angle on the wheel to correspond with the worm.
2. Reduce the diameter of the worm equivalent to the amount the hob is reduced due to sharpening.

Standard Design of Worm Gearing.—The design of worm gears, using existing hobs which do not correspond to the sizes recommended as standard, is as follows:

Face of Wheel.—For wheels engaging with single- or double-thread worms the width of face should not be greater than the chord of the worm outside circle which is tangent to the worm-pitch circle plus one-half of the linear pitch.

For wheels engaging with triple- and quadruple-thread worms, the width of face should not be greater than the chord of the worm outside circle which is tangent to the worm-pitch circle plus one-quarter of the linear pitch.

Outside Diameters.—For wheels meshing with single- or double-thread worms:

$$OD = PD + (3.5 \times A)$$

For wheels meshing with triple- or quadruple-thread worms, where the pressure angle is 20 deg. or greater:

$$OD = PD + (3 \times A)$$

For wheels meshing with triple- or quadruple-thread worms, where the pressure angle is less than 20 deg.:

$$OD = PD + (2.75 \times A)$$

Top Round.—The radius which joins the outside and the sides of the rim of the wheel has been called the top round and it is recommended that this be made equivalent to one-quarter the linear pitch.

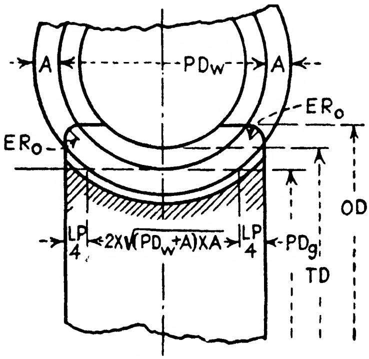


Fig. 116.—Diagram worm and gear with 29-deg. thread.

In Fig. 116 a diagrammatic illustration of the foregoing recommendations is shown, together with formulas for calculating them.

Worm-pitch Diameter.—The worms, of course, must be made to correspond to the pitch diameter of the hob, in order to secure proper contact.

Length of Worm Face.—The length of the worm may be made in accordance with the formula for standard worms:

$$FL = LP \times \left(4.5 + \frac{N}{50}\right)$$

Hub Extensions.—The hub extensions may be made in accordance with the formula:

$$HE = LP$$

Top Round of Worm Thread.—The top edge of the worm thread should in all cases be rounded to avoid cutting action in case the gears are not mounted on exact theoretical centers. The amount of the rounding may be made equivalent to 0.05 of the linear pitch.

For explanation of the symbols used in Figs. 116 and 117 see gearing nomenclature at end of Chap. I.

Worm Threads.—Worm threads having a 29-deg. included angle are made in the proportions shown in Fig. 118. These threads either are cut with a single-point tool or are milled, as with other screw threads. The proportions of the threads are given in Table 49. As with other screw threads, worms are made with either single or multiple threads, according to the service for which they are intended. A single-lead worm turns the worm wheel one tooth for each revolution of the worm. A double-thread, or start, worm moves the worm wheel two

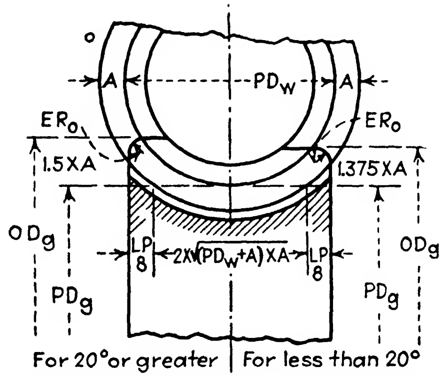
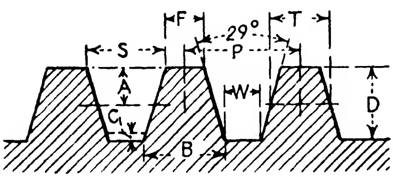


FIG. 117.—Diagram of threads of different angles.

teeth per revolution, and so on. Table 48 gives worm threads for diametral pitch and the gears necessary to cut them.

As the number of starts increase, with the same diameter of worm, the helix angle often becomes very sharp. In some mechanisms where high speeds are necessary, as in cream separators, the worm wheel drives the worm. This necessitates a



N = No. of threads per inch
P = $1/N$ = linear pitch
$D = 0.6866P$ $A = 0.3183P$
$F = 0.335P$ $C = T/10$
$W = 0.31P$ $S = 0.665P$
$T = 0.5P$ $B = 0.69P$

$$\text{Pitch} = \frac{1}{\text{No. of threads per inch}}$$

$$\text{Depth of thread} = \frac{0.6866}{\text{No. of threads per inch}}$$

FIG. 118.—Proportions of standard 29-deg. worm thread.

very sharp helix angle on the thread of the worm, often exceeding 45 deg. As with the cutting of screw threads, increased helix angle adds to the difficulty of cutting the thread on account of tool clearance. Threads on worms with a large helix angle are usually milled. When cutting worms with a multiple-lead thread it is necessary to remember that the thread depth and other dimensions must be divided by the number of leads. That is, a triple-lead screw of 1-in. pitch will be only one-third as deep and one-third as wide as if it was a single lead.

Worm-wheel Hobs.—In order to provide clearance between the top of the worm thread and the bottom of the worm-wheel tooth, it is customary to provide the hob twice the normal clearance for the teeth themselves. A single clearance is one one-tenth of the thickness of the tooth at the pitch line. As this thickness is one-half the pitch, the normal clearance is one-fifth of the pitch, and the hob clearance is twice this amount. All this

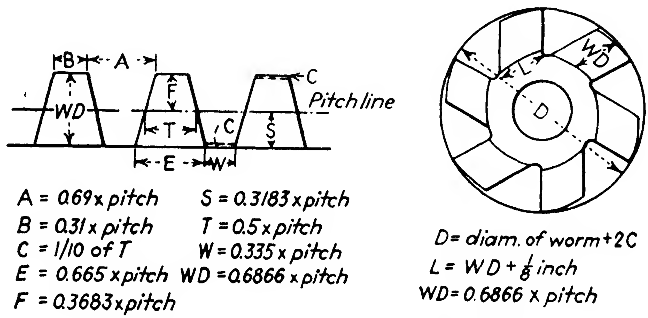


FIG. 119.—Standards for worm wheel hobs.

is shown clearly in Fig. 119, which gives the proportions of the teeth in a standard hob.

Lead Gears for Diametral Pitch Worms.—Practically all spur, helical, and bevel gears are now made to the diametral pitch system because of the simplicity of its calculations. Worm gearing has been made to the circular pitch system because of the ease of gearing a machine to produce worms with leads which are simple fractions. This practice is now being gradually superseded by the diametral pitch system. The one disadvantage, that of calculating change gears, is obviated by giving the four gears necessary to produce the correct lead on a worm of any diametral pitch from one to sixty and any number of threads from one to eight.

The following suggestions for selecting the proper gears for cutting worms for gears with diametral pitch are given by C. A. Johnson of the Perkins Gear & Machine Company. These will be found very useful in many cases.

The diametral pitch is found at the left-hand side of the table, and in the column under the desired number of threads are found the change gears which will produce the correct lead. The table is divided horizontally into three sections. Gearing for the coarse pitches (1 to 4) is calculated for a machine with a 3-in. constant, that is, a machine which will produce a lead of 3 in. with equal gears. The medium pitches (5 to 20) are calculated for a machine with a 1-in. constant, and the fine pitches (22 to 60) are calculated for a machine with a ½-in. constant. In every case the largest gears to produce the desired ratio are given. For some ratios these may be factored if smaller gears are desired. All of the gear ratios in the table are the closest to the desired ratio which may be obtained using gears with tooth numbers from 24 to 100. The required gears are as follows:

3-in. machine—35 gears in all

1 each—24, 25, 28, 30, 34, 35, 36, 42, 43, 44, 47, 48, 51, 55, 60, 61, 62, 68, 70, 72, 75, 77, 84, 85, 86, 88, 89, 95, 98, 99.

2 each—25, 50, 100.

1-in. machine—59 gears in all

1 each—25, 27, 28, 29, 31, 32, 33, 34, 35, 40, 41, 42, 43, 44, 45, 47, 48, 51, 52, 55, 56, 58, 59, 61, 62, 63, 64, 65, 66, 68, 70, 72, 73, 76, 77, 78, 80, 81, 82, 84, 85, 86, 87, 88, 89, 90, 91, 93, 94, 95, 96, 98, 99.

2 each—50, 75, 100.

½-in. machine—62 gears in all

1 each—26, 27, 28, 29, 31, 32, 33, 34, 35, 41, 42, 43, 44, 45, 46, 47, 48, 51, 52, 55, 56, 57, 58, 59, 61, 62, 63, 64, 65, 68, 69, 70, 73, 76, 77, 78, 79, 80, 81, 82, 84, 85, 86, 87, 88, 89, 90, 91, 92, 94, 95, 96, 98, 99.

2 each—50, 66, 75, 100.

It may be necessary to cut a worm on a machine of different constant than that for which the change gears are tabulated. The procedure for finding the correct set of gears is as follows:

Suppose that a 3-diametral pitch gear is to be cut on a 2-constant machine. The gears for a 3-constant machine are of 34, 75, 77, and 100 teeth. To maintain the same lead on the worm, however, it will be necessary to increase the ratio of the change gears in, the ratio of 3 to 2. Thus, $\frac{3 \times 34 \times 77}{2 \times 75 \times 100}$, which by factoring becomes $\frac{68 \times 77}{100 \times 100}$.

Table 48.—Lead Gears for Diametral-pitch Worms

Diametral pitch	Number of threads 1				2				3				4				5				6				7				8				Machine constant
	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	
1	68	50	77	100	68	50	77	50	51	25	77	50	68	25	77	50	85	25	77	50	51	25	77	25	85	24	89	43	68	25	77	25	
1¼	68	84	89	86	68	84	89	43	85	70	89	43	68	42	89	43	68	25	77	50	85	35	89	43	85	30	89	43	100	95	77	43	
1½	68	75	77	100	68	75	77	50	68	50	77	50	68	25	77	75	85	25	77	75	68	25	77	50	85	36	89	43	98	47	75	28	
1¾	68	100	88	100	68	100	88	50	68	50	99	75	68	50	88	50	85	50	88	50	68	25	99	75	68	25	77	50	68	25	88	50	
2	68	100	77	100	68	100	77	50	51	50	77	50	68	50	77	50	85	50	77	50	51	25	77	50	85	48	89	43	68	25	77	50	3
2½	34	84	89	86	68	84	89	86	62	55	68	61	68	84	89	43	68	50	77	50	85	70	89	43	85	60	89	43	68	42	89	43	
3	34	75	77	100	68	75	77	100	68	50	77	100	68	75	77	50	85	75	77	50	68	50	77	50	85	72	89	43	68	75	77	25	
3½	68	100	44	100	68	100	88	100	68	100	99	75	68	100	88	50	85	100	88	50	51	50	88	50	68	50	77	50	68	50	88	50	
4	34	100	77	100	68	100	77	100	51	50	77	100	68	100	77	50	85	100	77	50	51	50	77	50	85	48	89	86	68	50	77	50	
5	62	55	34	61	62	55	68	61	93	55	68	61	68	56	89	43	51	25	77	50	51	28	89	43	85	40	89	43	85	35	89	43	
6	68	100	77	100	68	100	77	50	51	50	77	50	68	50	77	50	85	50	77	50	51	25	77	50	85	48	89	43	68	25	77	50	
7	68	100	66	100	68	100	99	75	51	50	99	75	68	50	99	75	85	50	99	75	68	50	99	50	51	25	77	50	68	25	66	50	
8	51	100	77	100	51	100	77	50	66	58	88	85	51	50	77	50	51	40	77	50	66	29	88	85	77	29	88	85	51	25	77	50	
9	34	75	77	100	68	75	77	100	68	50	77	100	68	75	77	50	85	75	77	50	68	50	77	50	85	72	89	43	68	25	77	75	
10	31	55	34	61	62	55	34	61	62	55	51	61	62	55	68	61	51	50	77	50	93	55	68	61	85	80	89	43	85	70	89	43	
11	51	100	56	100	68	100	84	100	68	100	63	50	68	100	84	50	51	50	98	70	68	50	63	50	51	50	98	50	68	50	84	50	1
12	34	100	77	100	68	100	77	100	51	50	77	100	68	100	77	50	85	50	77	100	51	50	77	50	85	48	89	86	68	50	77	50	
13	32	73	43	78	41	65	59	77	48	73	86	78	85	91	89	86	51	50	77	65	96	73	86	78	85	52	89	86	85	91	89	43	
14	51	100	44	100	68	100	66	100	51	50	66	100	68	100	99	75	85	100	99	75	51	50	99	75	51	50	77	50	66	50	68	50	
15	42	94	45	96	34	84	89	86	62	55	34	61	68	84	89	86	68	50	77	100	62	55	68	81	68	96	89	43	68	42	89	86	
16	33	85	44	87	51	100	77	100	66	87	66	85	51	50	77	100	51	50	77	80	99	87	88	85	77	58	88	85	51	50	77	50	

TABLE 48.—LEAD GEARS FOR DIAMETRAL-PITCH WORMS.—(*Continued*)

Diametral pitch	Number of threads																																Machine constant
	1				2				3				4				5				6				7				8				
	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	Driver	Follower	
17	42	100	44	100	56	100	66	100	84	100	66	100	84	100	88	100	84	100	99	90	84	100	99	75	98	100	99	75	84	75	99	75	
18	31	99	34	61	34	75	77	100	68	100	77	100	68	75	77	100	51	50	77	90	68	50	77	100	85	72	89	86	68	75	77	50	1
19	32	70	34	94	41	77	59	95	43	73	80	95	63	76	75	94	82	77	59	76	86	73	80	95	85	76	89	86	63	76	75	47	
20	27	64	35	94	31	55	34	61	55	95	70	86	62	55	34	61	51	50	77	100	62	55	51	61	85	80	89	86	62	55	68	61	
22	51	100	56	100	68	100	84	100	68	100	63	50	68	100	84	50	51	50	98	70	68	50	63	50	51	50	98	50	68	50	84	50	
24	34	100	77	100	68	100	77	100	51	50	77	100	68	100	77	50	85	50	77	100	51	50	77	50	85	48	89	86	68	50	77	50	
26	32	73	43	78	41	65	59	77	48	73	86	78	85	91	89	86	51	50	77	65	96	73	86	78	85	52	89	86	85	91	89	43	
28	51	100	44	100	68	100	66	100	51	50	66	100	68	100	99	75	85	100	99	75	51	50	99	75	51	50	77	50	66	50	68	50	
30	42	94	45	96	34	84	89	86	62	55	34	61	68	84	89	86	68	50	77	100	62	55	68	81	68	96	89	43	68	42	89	86	
32	33	85	44	87	51	100	77	100	66	87	66	85	51	50	77	100	51	50	77	80	99	87	88	85	77	58	88	85	51	50	77	50	
34	42	100	44	100	56	100	66	100	84	100	66	100	84	100	88	100	84	100	99	90	84	100	99	75	98	100	99	75	84	75	99	75	
36	31	99	34	61	34	75	77	100	68	100	77	100	68	75	77	100	51	50	77	90	68	50	77	100	85	72	89	86	68	75	77	50	
38	32	70	34	94	41	77	59	95	43	73	80	95	63	76	75	94	82	77	59	76	86	73	80	95	85	76	89	86	63	76	75	47	½
40	27	64	35	94	31	55	34	61	55	95	70	86	62	55	34	61	51	50	77	100	62	55	51	61	85	80	89	86	62	55	68	61	
42	34	100	44	100	68	100	44	100	68	100	66	100	68	100	88	100	85	100	88	100	68	100	99	75	68	100	77	50	68	100	88	50	
44	34	100	42	100	51	100	56	100	68	100	63	100	68	100	84	100	85	100	84	100	68	100	63	50	51	50	98	100	68	100	84	50	
46	26	69	29	80	32	69	43	73	64	92	43	73	63	92	75	94	82	92	59	77	64	92	86	73	85	92	89	86	63	46	75	94	
48	26	79	35	88	34	100	77	100	51	100	77	100	68	100	77	100	85	100	77	100	51	50	77	100	85	96	89	86	68	100	77	50	
50	27	94	42	96	28	57	44	86	56	95	55	86	56	57	44	86	62	55	34	61	56	57	66	86	85	100	89	86	56	57	88	86	
56	34	100	33	100	51	100	44	100	51	100	66	100	68	100	66	100	85	100	66	100	51	50	66	100	51	50	77	100	68	100	99	75	
60	27	94	35	96	42	94	45	96	31	55	34	61	34	84	89	86	68	100	77	100	62	55	34	61	70	82	79	92	68	84	89	86	

TABLE 49.—TABLE OF PROPORTIONS OF WORM THREADS TO RUN IN WORM WHEELS

CP Circular pitch	*π* Threads per inch	*DP* Diametrical pitch	*H* Tooth above pitch line	*D* Working depth of tooth	*C* Clearance	*S* Depth of space below pitch line	*WD* Whole depth of tooth	*T* Thickness of tooth on pitch line	*W* Width of threads tool at end	*B* Width of thread at top
CP in.	$\pi = 1/CP$	$DP = \pi/CP$	$H = 1/DP$	$D = 2 \times 1/DP$	$C = T/10$	$S = H + C$	$WD = D + C$	$T = CP/2$	$W = 0.31 \times CP$	$B = 0.335 \times CP$
2	1/2	1.5708	0.6366	1.2732	0.1000	0.7366	1.3732	1.0000	0.6200	0.6708
1 3/4	4/7	1.7952	0.5570	1.1141	0.0875	0.6445	1.2016	0.8750	0.5425	0.5862
1 1/2	2/3	2.0944	0.4775	0.9549	0.0750	0.5525	1.0299	0.7500	0.4650	0.5025
1 1/4	4/5	2.5133	0.3979	0.7958	0.0625	0.4604	0.8583	0.6250	0.3875	0.4187
1	1	3.1416	0.3183	0.6366	0.0500	0.3683	0.6866	0.5000	0.3100	0.3350
3/4	1 1/3	4.1888	0.2387	0.4775	0.0375	0.2762	0.5150	0.3750	0.2325	0.2512
2/3	1 1/2	4.7124	0.2122	0.4244	0.0333	0.2455	0.4577	0.3333	0.2066	0.2233
1/2	2	6.2832	0.1592	0.3183	0.0250	0.1842	0.3433	0.2500	0.1550	0.1675
2/5	2 1/2	7.8540	0.1273	0.2546	0.0200	0.1473	0.2746	0.2000	0.1240	0.1340
1/3	3	9.4248	0.1061	0.2122	0.0166	0.1227	0.2288	0.1666	0.1033	0.1117
2/7	3 1/2	10.9956	0.0909	0.1819	0.0143	0.1052	0.1962	0.1429	0.0886	0.0957
1/4	4	12.5664	0.0796	0.1591	0.0125	0.0921	0.1716	0.1250	0.0775	0.0838
2/9	4 1/2	14.1372	0.0707	0.1415	0.0111	0.0818	0.1526	0.1111	0.0689	0.0744
1/5	5	15.7080	0.0637	0.1273	0.0100	0.0737	0.1373	0.1000	0.0620	0.0670
1/6	6	18.8496	0.0531	0.1061	0.0083	0.0614	0.1144	0.0833	0.0517	0.0558
1/7	7	21.9911	0.0455	0.0910	0.0071	0.0526	0.0981	0.0714	0.0443	0.0479
1/8	8	25.1327	0.0398	0.0796	0.0062	0.0460	0.0858	0.0625	0.0388	0.0419
1/9	9	28.2743	0.0354	0.0707	0.0055	0.0409	0.0763	0.0555	0.0344	0.0372
1/10	10	31.4159	0.0318	0.0637	0.0050	0.0368	0.0687	0.0500	0.0310	0.0335
1/12	12	37.6992	0.0265	0.0530	0.0042	0.0307	0.0572	0.0416	0.0258	0.0279
1/14	14	43.9824	0.0227	0.0454	0.0036	0.0263	0.0490	0.0357	0.0221	0.0239
1/16	16	50.2655	0.0199	0.0398	0.0031	0.0230	0.0429	0.0312	0.0194	0.0209
1/18	18	56.5488	0.0176	0.0352	0.0025	0.0201	0.0377	0.0255	0.0172	0.0186

NOTE.—The above table refers to single threads only. For multiple threads, divide the sizes given in the table for the same pitch, by 2 for double. 3 for triple, 4 for quadruple threads, etc.

To find the pitch diameter (given pitch and number of teeth).

Rule.—Multiply the number of teeth by the circular pitch and divide by 3.1416.

Example.—Number of teeth, 96; circular pitch, 1/4. Then 96 × 1/4 = 24. 24 ÷ 3.1416 = 7.639 = pitch diameter. NOTE.—Diameter of Hob equals diameter of Worm plus twice the clearance *C*.

Proportions of Circular-pitch Worms.—For those who still use circular pitch in calculating worms, the data to be found in Table 49 will be found useful.

Worm-thread Measurement.—Worm-thread measurement is important if best results are to be secured. The following method was prepared for the *American Machinist* by T. R. Rideout, of the Nuttall Works Division of the Westinghouse Electric and Manufacturing Company. It may be well to remember that a radian, as mentioned in the formulas, is equal to 180 divided by 3.1416, or 57.2958.

Gear-tooth vernier calipers measure straight-line distances only, but the normal circular thickness of a worm thread lies on a helix normal or complementary to the helix of the thread at the pitch line. This is a warped line; that is, one whose projection on any plane will be curved. The axial thickness is, however, a straight line.

In order to measure the normal circular thickness of a worm thread, it is necessary to determine the shortest straight-line distance between the ends of the warped-line length of the normal circular thickness. This straight line will be the normal chordal thickness of the worm thread.

The following formulas are for obtaining caliper settings for measuring threads. These formulas have been checked several times for extreme cases by accurate laboratory measurements and have been found to be exact. In view of the ever-increasing use of high lead angles and the demand for greater accuracy, the formulas and their derivations are given for use in cases where conventional approximations are not sufficiently accurate.

Formulas for Normal Chordal Tooth Dimensions of Worm Threads

See diagram (Fig. 120):

T_A = axial thickness of thread.
T_N = normal circular thickness of thread.
R = pitch radius of worm.
λ = lead angle of worm.

$$T_N = T_A \cos \lambda$$

T_N = distance AB on warped line of complementary helix.

$$m = \text{arc} \quad CD = T_N \sin \lambda$$

$$\delta \text{ (radians)} = \frac{m}{R} \; (\delta^\circ = \delta \text{ radians} \times 57.29578)$$

$$\text{Chord } CD = 2(R \sin \tfrac{1}{2}\delta^\circ)$$

$$n = T_N \cos \lambda$$

$$\text{Normal chordal thickness} = \sqrt{\text{chord } CD^2 + n^2}$$

$$h = R(1 - \cos \tfrac{1}{2}\delta^\circ)$$

$$\text{Corrected addendum} = \text{addendum} + h$$

Cutting Hourglass Worms on the Gear Shaper.—A new development of the Fellows gear shaper enables the machine to cut

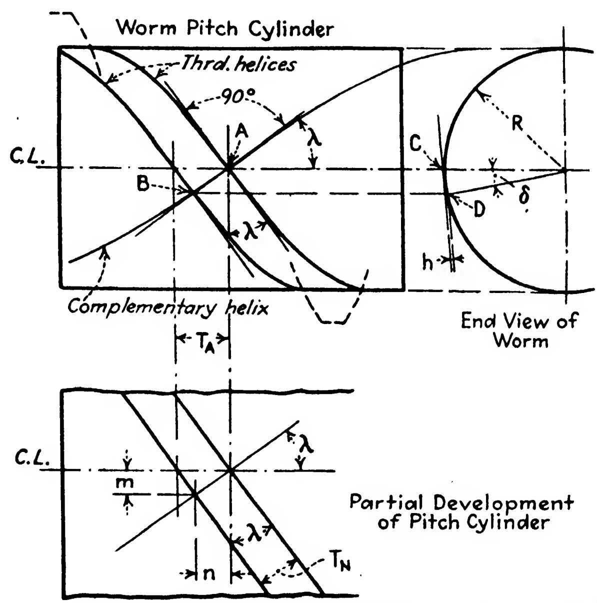


FIG. 120.—Normal chordal tooth dimensions of worm threads.

hourglass worms rapidly, accurately, and with a high finish. These are now being used in connection with helical gears, which reduces the difficulties of mounting to a minimum, and both worm and gear are cut with the same tool.

The enveloping nature of the hourglass worm brings a greater number of teeth into contact with the mating helical gear than is possible with the straight type of worm. It also increases the bearing area, assuring longer life and greater load-carrying capacity. When considerable latitude in endwise positioning of the worm is necessary, the worm is cut with a cutter having a greater number of teeth than the mating helical gear.

Hourglass worms are used extensively in automobile steering gears. To meet this demand the Fellows company has developed a method of cutting these hourglass worms on a gear shaper. The worms mate with a helical gear instead of a worm wheel, and usually cut with a cutter having the same number of teeth as the gear with which it mates. Sometimes considerable latitude is necessary in endwise positioning. This is secured by using a cutter having more teeth than the mating helical gears.

FIG. 121.—Cutting hourglass worm on gear shaper.

Principle of Operation.—In operation, the gear-shaper cutter is reciprocated as it rotates in harmony with the work, and produces a worm thread, which, in the axial plane, is the complement of the shape of the mating helical gear tooth. In other words, by reciprocating the cutter as it is rotating in harmony with the work, a "full bearing" on the worm thread is obtained. This is not possible when the cutter is fed radially into the work and rotated without the accompanying reciprocating action.

This machine is provided with a relieving-type saddle, which withdraws the cutter on the "return" stroke and then advances the cutter into cutting position on the "down" stroke. The bed carrying this relieving-type saddle is advanced toward the work by means of a feed cam, the speed of rotation of which, with relation to the work, is timed by change gears. The cutter is fed into depth at the same time that it is rotating in harmony with the work. When the cutter reaches full depth, the cam "dwells" until the work has completed one revolution and a sufficient portion of a second revolution to overlap that portion of the thread left uncompleted when the cutter was feeding into depth. This arrangement produces an unusually fine finish on the work. A close view of the operation is seen in Fig. 121.

CHAPTER VIII

INTERNAL GEARS

Teeth of internal gears point inward, or toward the center, instead of being on the outside as in the spur gear. The tooth action is also different, the contacts being longer for the same ratio of numbers of teeth between the gear and the pinion. The teeth also go into action and leave it with less slippage.

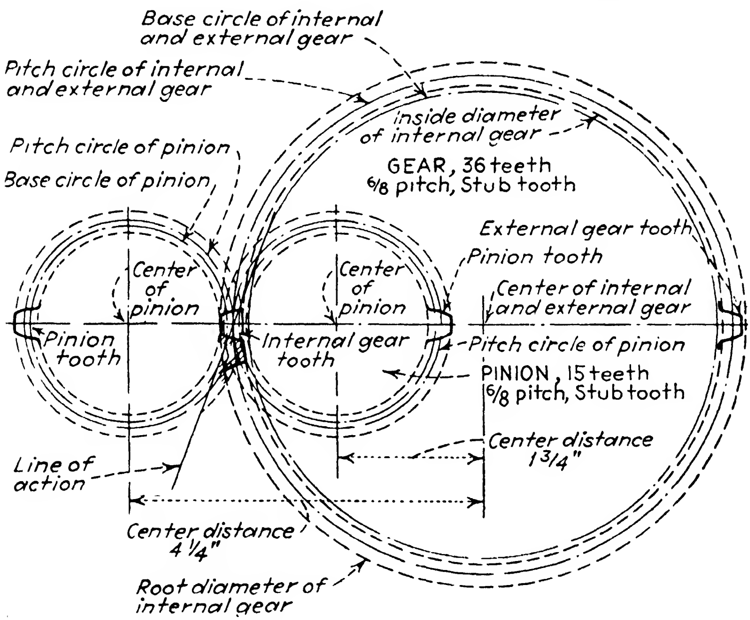


FIG. 122.—Comparison of center distance between internal and spur gears.

The internal gear has several advantages when properly used. For rear-axle drives in heavy-duty vehicles, and for use in speed reducers, or in "step-up" units, it has many uses. Where compact design is essential, the use of internal gears allows a much shorter center distance between the gear and its mating pinion than is possible with spur gears, as can be seen in Fig. 122. The internal-gear tooth is also stronger and has the advantage of having the gear form its own guard. Figure 123 shows the relative length of contact between an external gear and pinion, an external gear and a rack, and an internal gear and pinion of the same pitch and pressure angle.

Until the development of the method of cutting gears by the shaping process internal gears were a real problem in the shop and were consequently avoided in design as much as possible. While they can be cut in a shaper, using a formed tool, the problem of indexing makes this difficult in the average shop. Large

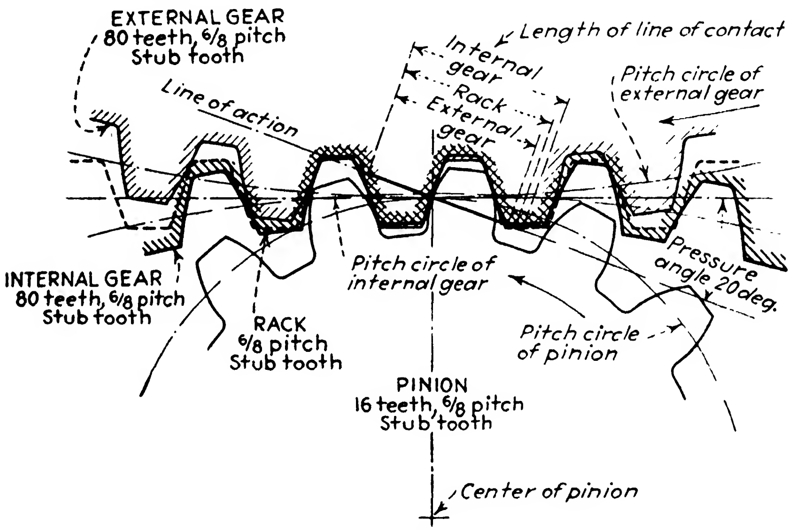


FIG. 123.—Length of contact of pinion and external gear, of rack and of internal gear.

internal gears have been cut by formed milling cutters fed across the inner face by special milling heads. Indexing remains the same problem as in the shaper. The shaping process, as exemplified first by the Fellows machine and later by the Sykes, makes the cutting of internal gears practically as simple as that of spur gears. This process is particularly valuable in the cutting of internal gears where the teeth are shrouded by inner flange, making it a blind hole, so that the cutter cannot pass through.

The various elements of an internal gear and its pinion are shown in Fig. 124. This gives the various parts of the gear and of the gear teeth and shows clearly the difference in the action of internal and spur gears.

In some cases it is desirable to have the number of teeth in the pinion as nearly equal to those in the gear as possible without interfering with satisfactory running. This, of course, is where but a slight difference in the speed of the parts is desired. The

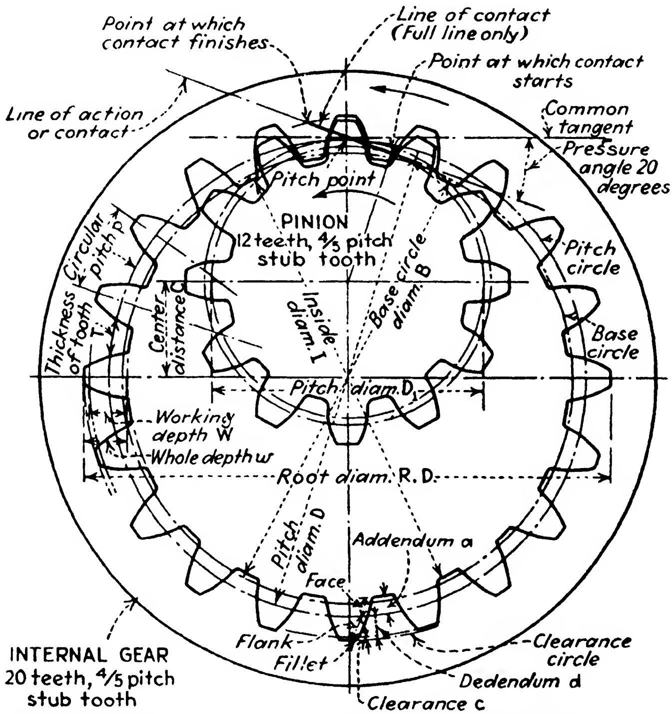


FIG. 124. —Showing the various elements of internal gear and pinion.

Fellows Gear Shaper Company, which has had a wide experience in this line of work, finds that as a general rule the smallest permissible difference between the number of teeth in the pinion and in the internal gear is seven teeth where the 20-deg. stub-tooth form is used, and 12 teeth for the full-length 14½-deg. involute form. Gears with smaller tooth difference than this require considerable modification in the shape of the tooth. Smaller difference than this in the number of teeth results in interference as the teeth enter and leave the mating gear.

For the same reason it is necessary to use care in selecting the cutter for internal gears, as the gear-shaper cutter is virtually a hardened gear. If too large in diameter, it will foul, or cut away portions of the teeth as it enters and leaves the cut. To

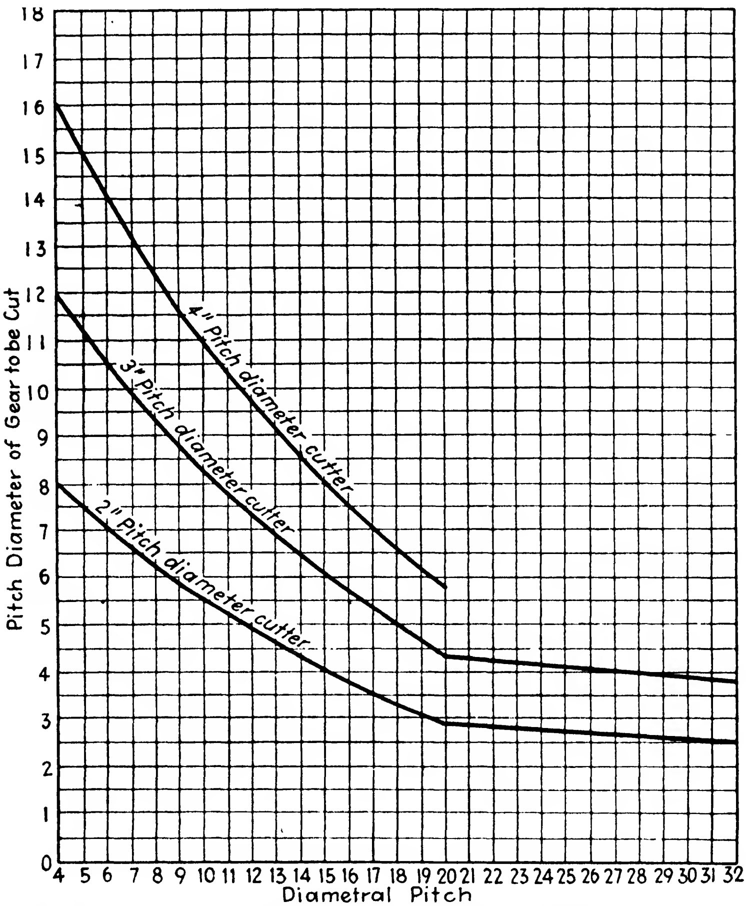


FIG. 125.—Smallest number of 14½ deg. full-depth teeth, that can be cut with 2-, 3-, or 4-inch gear-shaper cutter.

make it easy to select the proper cutter, the Fellows Company has prepared two charts, Figs. 125 and 126, which have been laid out for cutters having pitch diameters of 2, 3, and 4 in., respectively. This chart shows that a 3-in. cutter can be used on a 12-in. gear of 4 diametral pitch, on a 9¼-in. gear of 8 pitch, on a

$4\frac{3}{8}$-in. gear of 20 pitch, a $3\frac{3}{4}$-in. gear of 32 pitch, and all the other combinations on the 3-in. cutter line. This is for a full-length tooth of $14\frac{1}{2}$-deg. pressure angle. With 20-deg. stub teeth the same cutter will handle a larger range of gears, as shown in Fig. 126.

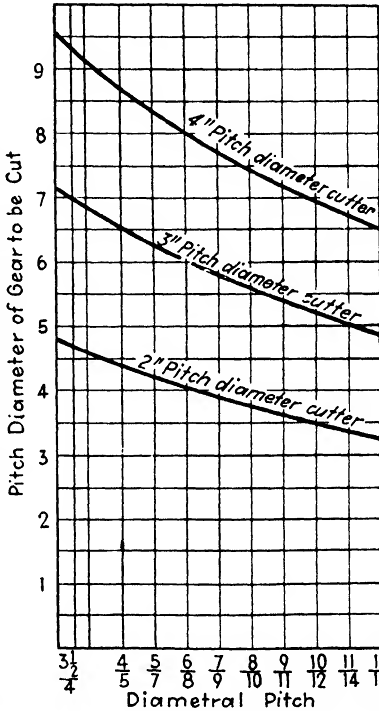


FIG. 126.—This gives similar data for 20-deg. stub-tooth gears.

Clearance Grooves for Internal Gears.—In cutting internal teeth where the gear is not a plain ring in which the cutter can pass straight through, it is necessary to provide a clearance groove, space, or recess into which the cutter can pass after completing the cut. This recess must be of sufficient width to allow an excess cutter stroke of $\frac{3}{32}$ in. and also to provide space for the chips. Otherwise they will pack into the space and interfere with the action of the cutter.

Table 50 gives the space necessary to provide proper clearance for spur gears and to helical gears with helix angles of both 15 and 23 deg. It also gives the excess of ram travel for both of these helix angles. The greater the angle the more clearance is necessary, owing to the necessity of having the cutting edges of the teeth ground at right angles to the angle of the helix.

The illustrations at the top of the table show the difference between the plain spur-gear cutters and the cutter for helical teeth.

Calculating Elements of Internal Gears.—The rules for finding the dimensions of an internal spur gear are from the practice of the Fellows Gear Shaper Company. They are similar in most cases to those given for external spur gears, except for the modifications made necessary by the fact that the center distance of an internal gear is equal to the difference between the pitch radii of the gear and pinion instead of their sum. Figure 124 shows a typical layout with all the parts named. In calculating the addendum and dedendum, the inside diameter of an internal

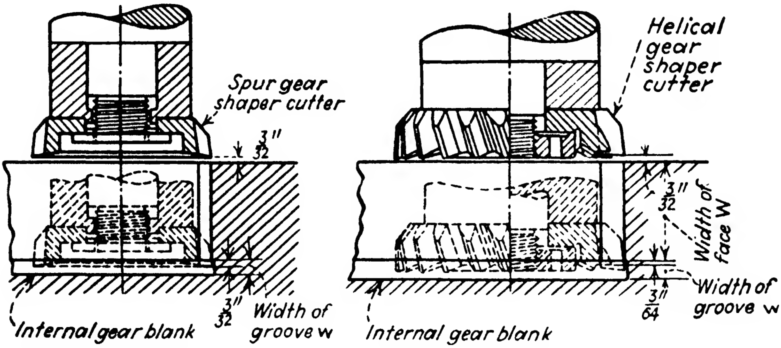


TABLE 50.—MINIMUM WIDTH OF CLEARANCE GROOVE AND EXCESS TRAVEL OF CUTTER SLIDE FOR CUTTING SPUR AND HELICAL INTERNAL GEARS

Diametral pitch of gear to cut	Width of groove w, spur cutter, in.	Helical gear-shaper cutter			
		Width of groove w 15-deg. helix angle, in.	Width of groove w 23-deg. helix angle, in.	Excess travel of ram 15-deg. helix angle, in.	Excess travel of ram 23-deg. helix angle, in.
4	9/32	11/32	25/64	13/64	17/64
5	17/64	5/16	23/64	9/64	13/64
6	1/4	19/64	21/64	7/64	11/64
7	1/4	9/32	5/16	5/64	11/64
8	15/64	17/64	19/64	5/64	7/64
10	7/32	17/64	9/32	3/64	7/64
12	7/32	1/4	17/64	3/64	5/64
14	13/64	1/4	17/64	1/32	5/64
18	13/64	15/64	1/4	....	3/64
24	3/16	15/64	1/4	....	1/32

gear takes the place of the outside diameter of an external gear. The internal diameter is, of course, the diameter of the hole in the blank. To use a crude expression, an internal gear is simply an external gear turned inside out.

Interference in Internal Gears.—Owing to the interference that results when an internal gear and mating pinion are almost of the same size, certain limits have been laid down for the size of the pinion. When teeth of standard 14½-deg. pressure-angle involute form are used, the difference between the numbers of teeth in the pinion and gear should in no case be less than 12, and for the stub form of tooth, 7. It might be stated, however, that there are certain exceptions to this rule when modified shapes of teeth are used, but the rolling action is never so good when the difference between the number of teeth in the pinion and gear is less than that stated.

One exception to the rule just stated is in automobile-clutch gears. Here the number of teeth in the pinion is the same as the number of teeth in the internal gear. In this case, however, as will be more fully explained later, the pinion and gear do not rotate in mesh with each other, but simply slide back and forth.

The rules for calculating six elements of an internal gear that differ from an external gear are as follows: (Table 51 gives the formulas.)

Pitch diameter.—When the *addendum* and *inside diameter* of an internal gear are known, the *pitch diameter* can be found by adding twice the standard *addendum* to the standard *inside diameter*.

Example.—Find the pitch diameter of an internal gear having an inside diameter 4 in. and an addendum of 0.125 in.

$$4 + (2 \times 0.125) = 4.250 \text{ in., pitch diameter}$$

Center Distance.—When the *number of teeth* in the gear and pinion and the *diametral pitch* are known, the *center distance* can be found by subtracting the *number of teeth* in the *pinion* from the *number of teeth* in the *gear* and dividing the remainder by twice the *diametral pitch*.

Example.—Find the center distance of an internal gear and pinion having 60 and 20 teeth, respectively, the diametral pitch being eight.

$$\frac{60 - 20}{2 \times 8} = \frac{40}{16} = 2.5 \text{ in., center distance}$$

When the *number of teeth* in the *gear* and *pinion* and *circular pitch* are known, the *center distance* can be found by multiplying the difference between the *numbers of teeth*, by the *circular pitch*, and dividing the product by 6.2832.

Example.—Find the center distance of an internal gear and pinion having 60 and 20 teeth, respectively, the circular pitch being 0.3927 in.

$$\frac{(60 - 20) \times 0.3927}{6.2832} = 2.5 \text{ in., center distance}$$

Inside Diameter.—When the *number of teeth* and *diametral pitch* are known, the *inside diameter* can be found by subtracting 2 from the *number of teeth* and dividing the remainder by the *diametral pitch.*

Example.—Find the inside diameter of an internal gear having 60 teeth of 8-diametral pitch.

$$\frac{60 - 2}{8} = 7.25 \text{ in., inside diameter}$$

When the *number of teeth* in the *gear* and *circular pitch* are known, the *inside diameter* can be found by subtracting 2 from

TABLE 51.—RULES AND FORMULAS FOR CALCULATING DIMENSIONS OF INTERNAL SPUR GEARS

Dimension wanted	Rule	Formula
Pitch diameter	Add twice the addendum to the inside diameter	$D = I + 2a$
Center distance	Subtract the number of teeth in the pinion from the number of teeth in the gear and divide remainder by two times diametral pitch.......	$C = \frac{N - n}{2P}$
Center distance	Multiply difference between number of teeth in gear and pinion by circular pitch and divide product by 6.2832.......................	$C = \frac{(N - n)p}{6.2832}$
Inside diameter	Subtract 2 from the number of teeth and divide remainder by diametral pitch..............	$I = \frac{N - 2}{P}$
Inside diameter	Subtract 2 from number of teeth, multiply remainder by circular pitch and divide product by 3.1416...........................	$I = \frac{(N - 2)p}{3.1416}$
Inside diameter	Subtract twice the addendum from pitch diameter.................................	$I = D - 2a$
Involute-base-circle diameter......	Multiply cosine of pressure angle by pitch diameter................................	$B = D \times \cos a$

the *number of teeth*, multiplying the remainder by the *circular pitch*, and dividing the product by 3.1416.

Example.—Find the inside diameter of an internal gear having 60 teeth of 0.3927-in. circular pitch.

$$\frac{(60 - 2) \times 0.3927}{3.1416} = 7.25 \text{ in., inside diameter}$$

When the *addendum* and the *pitch diameter* are known, the *inside diameter* of an *internal gear* can be found by subtracting twice the *addendum* from the *pitch diameter.*

Example.—Find the inside diameter of an internal gear having a pitch diameter of 7.5 in. and an addendum of 0.125 in.

$$7.5 - (2 \times 0.125) = 7.25 \text{ in., inside diameter}$$

Involute-base-circle Diameter.—When the *pressure angle* and the *pitch diameter* of an *internal gear* are known, the *involute base circle* can be found by multiplying the *pitch diameter* by the *cosine* of the *pressure angle.*

Example.—What is the involute-base-circle diameter of an internal gear of standard 14½-deg. involute-tooth form having a pitch diameter of 7.5 in.?

$$7.5 \times 0.9682 = 7.261 \text{ in., involute-base-circle diameter}$$

For 20-deg. pressure angle, multiply the cosine of 20 deg. or 0.9397 by the pitch diameter of the gear.

Internal Clutch Gears.—Internal clutch gears used in automobile transmissions generally are of drum form inside of which is located a series of disks. The internal diameter of the drum (which could be called the driving gear), and the external diameter of the inner clutch member (which could be called the driven gear), are provided with gear teeth. Located between these two gears is a series of disks. These are so arranged with gear teeth that each alternate disk meshes with the internal gear and external gear, respectively. Two forms of teeth are quite commonly used, *viz.*, the 14½-deg. involute and the 20-deg. stub tooth.

Limits for Size of Internal Clutch Gears.—In an internal clutch gear, of course, the internal gear has the same number of teeth as those on the disks that fit into it, and theoretically speaking, the pitch diameter of the teeth in the internal gear and disk should coincide. From a practical standpoint, however, this is impossible; and in actual practice, the pitch diameter of the teeth on the disk is made slightly smaller than that of the internal gear. The allowance between these two dimensions to provide for sliding without excessive backlash or shake varies all the way from 0.002 to 0.020 in., depending on the number of teeth and their form.

In cutting internal clutch gears of small diameter, there are certain limitations to the number of teeth that can be cut on the gear shaper without cutting away the interference points on the teeth of the internal gear. Of course, as shown in Fig. 127, if there is no objection to the trimming of the points of three or

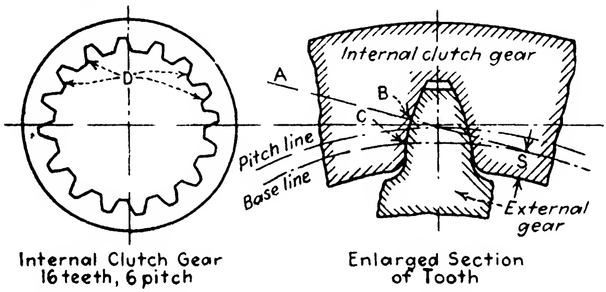


Fig. 127.—How teeth of internal gears are trimmed and suggested method of increasing internal diameter to avoid interference.

four of the teeth of an internal gear, which takes place when the cutter is being fed into depth, then the number of teeth can be much smaller than when cutting the points would be objectionable. As a general rule, 20 teeth for a stub tooth and 24 teeth for a 14½ involute tooth is the limitation of this practice.

Cutting Internal Clutch Gears.—When cutting an internal clutch gear having a small number of teeth, the inside diameter of the clutch should be enlarged to the base diameter of the gear. This in no way affects the positive action or strength of the clutch, because the involute stops at the base line and none of the tooth curve below this point is in contact with the mating external gear. Furthermore, the clutch has such an excess of strength over the other gears of the transmission that the clutch teeth could, if necessary, be cut clear down to the pitch line, or even beyond without any danger of seriously affecting its strength or positive action.

This can be clearly seen by referring to the enlarged diagram, Fig. 127. In the diagram, line *A* represents a cord being wound around a cylinder marked "base line." Point *B* on this line traces the involute curve of the tooth. It is evident that the tracing point cannot travel below the cylinder upon which it is wound without giving a reverse curve and causing interference. The involute curve then ends at point C, and there is no tooth contact below this point. Metal S is, therefore, useless. This useless metal makes tooth cutting more difficult, and in order to eliminate the trouble that is experienced in cutting internal teeth, it is advisable to bore the internal gear to the base-line diameter.

Smallest Number of Teeth That Can Be Cut with the Gear-shaper Cutter.—In cutting an internal gear having a small number of teeth, the cutter has a tendency to rub on the tooth of the gear when the apron is withdrawn. In order to eliminate this, the eccentric roll is adjusted out of contact with the backing-off plunger, so that the apron is not relieved on the return stroke of the cutter. In other words, the apron is locked continuously during the cutting and return strokes.

As has been previously stated, the smallest number of teeth of $14\frac{1}{2}$-deg. form and of standard inside diameter, that can be cut without interference is 24, and the relieving roll should be brought out of contact with the plunger when cutting 30 teeth or less. When the inside diameter is enlarged to the base diameter of the gear, the minimum number of teeth of $14\frac{1}{2}$-deg. form that can be cut without interference is 20, and the relieving roll should be backed away from the plunger for 25 teeth or less.

When cutting teeth of 20-deg. stub-tooth form, the minimum number of teeth that can be cut without interference when the gear is of standard inside diameter, is 23 for 6/8 pitch, 20 for 5/7 pitch; and the eccentric roll should be adjusted out of contact with the plunger when cutting 27 teeth or less of 6/8 pitch, or 25 teeth or less of 5/7 pitch. When the teeth are of 20-deg. stub form with the *inside diameter* enlarged to the *base diameter*, the minimum number of teeth that can be cut without interference when of 6/8 pitch, is 20, and the roll should be adjusted for 25 teeth or less. For gears of 5/7 pitch, the minimum number of teeth is 17, and the roll should be adjusted for 21 teeth or less. Below 17 teeth, the corners of three or four of the teeth are trimmed as indicated in Fig. 127. In cutting an internal gear, the cutter is first fed into full depth and then the rotary feed is engaged.

CHAPTER IX

BURNISHING, SHAVING, LAPPING AND GRINDING GEAR TEETH AND BORES

The demand for quieter gears led to many experiments in the endeavor to secure smoother contact surfaces even before the days of the automobile. Automobile transmissions have, however, been the greatest stimulant to research and to the trial of different methods to secure gears with less noise. This quest for quiet gears has led to the development of several different methods which may be divided into two general classes, those which correct the tooth shape before hardening and those which correct errors in the gear after it has been hardened. Burnishing and shaving come in the first class and lapping and grinding in the second. All have their advocates and many good gears have been made by the various methods.

One of the questions to be carefully considered is that of cost, and for this reason the methods which correct errors in the gear before hardening find favor in many places. The accuracy of the finished gear in these cases depends very largely on the methods of heat treatment, and presupposes that this department can turn out gears in which the distortion due to heating and cooling is negligible.

Gear-tooth Burnishing.—One of the methods of improving gears after they have been cut and before they are hardened is to burnish them by rolling them between hardened gears to iron out minute inequalities in the surface of the teeth. Advocates of other methods of finishing contend that this distorts the surface of the metal in the tooth gear and sets up stresses that are relieved in the heat treatment, and nullify at least part of the correction of form that may have been made.

There are, nevertheless, those who still believe that burnishing has its place in some gear production. An example of this is seen in Fig. 128 where a small but accurate pinion is being burnished between hardened and ground master gears in a massive machine for work of this size.

Gear-tooth Shaving.—One of the methods of finishing green, or unhardened, gears is by shaving. One of the shaving methods

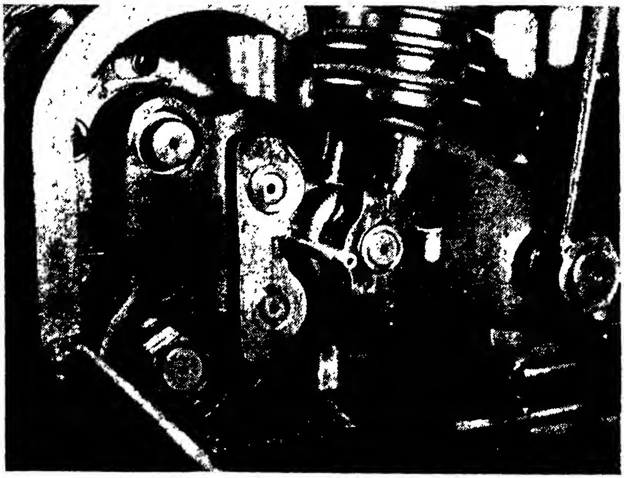


Fig. 128.—Burnishing a small pinion between master gears.

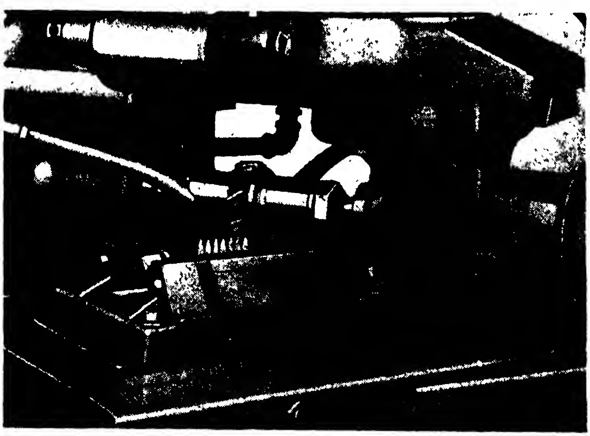


Fig. 129.—Shaving a helical gear with a straight rack tool.

is shown in Fig. 129, which illustrates a machine by the Michigan Tool Company. In this machine the shaving tool takes the form of a rack with small cutting edges on the teeth, as can be seen by close examination. This rack is fastened to the bed of what is virtually a small planer, the gear being mounted on a mandrel on the cross rail. A straight rack is used for gears with helical teeth and a helical, or angular, rack for spur gears. This combination gives the cutting teeth a shearing action across the face of the teeth as the gear rolls along the rack.

These racks are made up of individual serrated segments and held to very close limits by photoelectric inspection, with a maximum tolerance of 0.000025 in. The rack blades can be resharpened by removing from 0.004 to 0.008 in. With 28-tooth, 10-pitch gears of ¾-in. face a rack cuts 30,000 gears between grinds, and the life of a rack is from 20 to 25 grinds.

In operation the gear travels over the rack a predetermined number of strokes, from 1 to 50, depending on the gear. The head feeds with a definite pressure on the work. When the proper number of strokes have been made, the head rises automatically, the table stops and the work can be changed. By using two mandrels production is practically continuous.

From 0.005 to 0.010 in. of stock should be left for shaving. For estimating production time a 28-tooth, 10-pitch gear of ¾-in. face may be used. With the average stock removal these gears can be shaved at the rate of one per minute.

Rotary Crossed-axis Shaving.—Another method of shaving gear teeth after they have been cut has been developed by R. S. Drummond of the National Broach & Machine Company. The basic idea behind this method is the crossing of the axes of the gear being shaved and that of the shaving cutter. The gear is rotated under only slight pressure in contact with the special cutter developed for this purpose. This cutter is in the form of a spur or a helical gear, with the teeth gashed to present a series of cutting edges to the faces of the gear teeth as both revolve together, as in Fig. 130.

One advantage of crossing the axes is the limiting of the contact pressure required in shaving and the accurate guiding of the cutting edges of the tool. Cutting can be increased by increasing the angle between the axes but this tends to sacrifice the guiding action and to reduce the width of the contact zone. With an angle of from 10 to 15 deg. between the work and the cutter, a pressure of approximately 40 lb. is all that is necessary for good results. These cutters produce very fine, hairlike chips that curl, rather than coil, off the teeth. They should be strained out of the lubricant or coolant, before it is again pumped to the work.

Figure 131 shows three types of cutters used and Fig. 132 the shaving of an internal gear by this method. As will be seen, the coolant is directed close to the point of contact.

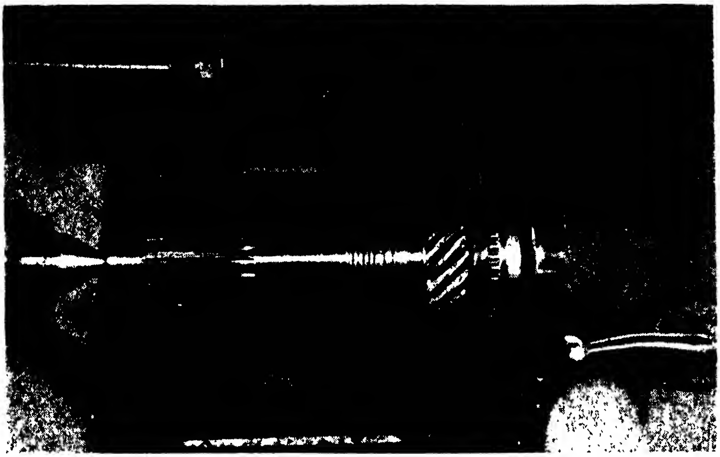


FIG. 130.—Shaving a helical gear by the crossed-axis method.

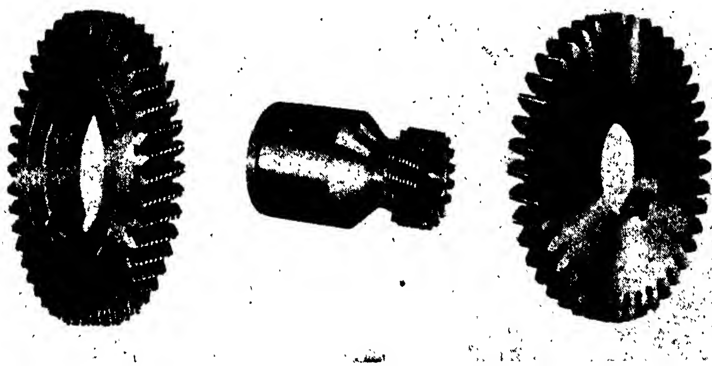


FIG. 131.—Three types of cutters used.

With the machine on cutting stroke, the surface produced has minute lines up and down the tooth. These indicate the spacing of the cross travel of the work per revolution. But because the cutter contacts the work at a different point on each stroke, these lines can be completely removed by one or more idling strokes after the shaving is completed. The final finish is smooth and mirror-like with no effect of compression as with burnishing.

Cutter speeds are frequently 400 surface feet per minute. It is now found possible to produce these cutters with an accumulative error not to exceed 0.0003 in. The cutter life is high, as from 6,000 to 12,000 work pieces can be shaved before regrinding is necessary. In normal pitch sizes these cutters can be reground from seven to ten times. . On 1 five-pitch cutter, for example, a total of from 50,000 to 100,000 pieces can be shaved by a single cutter.

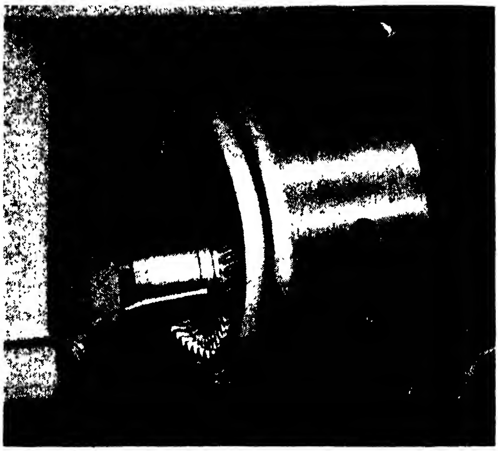


FIG. 132.—Shaving an internal gear.

Fellows Gear-finishing Process.—A recent development of a gear-finishing process by the Fellows Gear Shaper Company is of interest. This includes both a machine having special characteristics and cutters with new features. The principles involved are best shown by line as in Fig. 133. The angle to which the tool head is set depends on the helix angle of the gear or cutter used. A straight-toothed cutter is used for finishing a helical gear and a helical cutter for a spur gear. It should be noted that the tool slide reciprocates, but not in line with the axis of the tool spindle. Instead, it slides parallel to the axis of the gear being finished.

The cutting edges of the tool intersect the faces of the teeth as seen in Figs. 134 and 135. With the cutter and work in contact and rotated under pressure, the diverging path through which the teeth travel moves the cutting edges lengthwise along the teeth and finishes them with a shearing action. Behind the cutting edges the pressure gives a burnishing action that produces

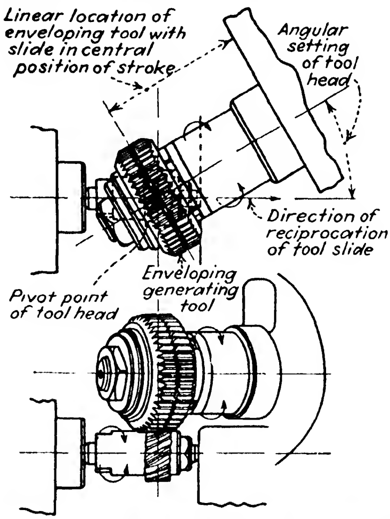


Fig. 133.—Principle of operation of Fellows enveloping gear generator.

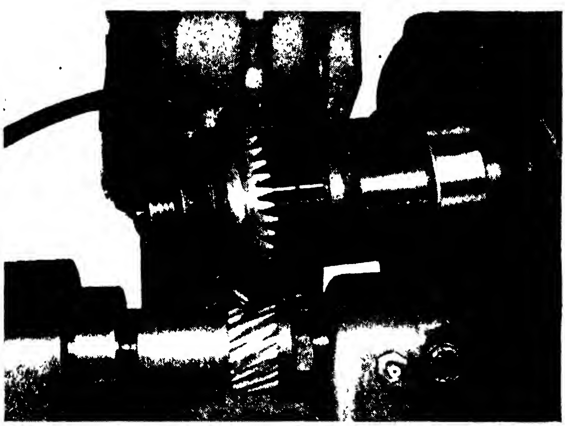


Fig. 134.—Fellows gear-finishing process.

a smooth finish on the mating surfaces of the teeth. Cutting action depends on the angularity of the tool and work axes. The greater this angle the more pronounced the "skid" and the more efficient the cutting action.

This generating tool is so made that the teeth control the teeth in the gear almost the entire length, which gives a positive control, free cutting, and fine finish. Although the tool resembles the regular gear-shaper cutter, it differs in many respects. There is no side clearance or outside angle and the sides of the teeth are bowed inward to match the sides of the teeth on the gear being cut. The sides of the teeth on the tool used for cutting an external gear are made concave, whereas for an internal gear they are bowed outward, or made convex.

FIG. 135.—Finishing an internal helical gear.

The width of the space and the degree of curvature, or amount of enveloping contact of the teeth, govern the amount of material removed at each stroke, as well as the finish of the work. This tool operates at low pressure and a "hunting" tooth is provided to secure greater accuracy. Where advisable, as when shafts are slightly out of line, a crowning effect can be produced on the teeth. While Fig. 135 resembles Fig. 132 the cutting action is quite different.

There is also a combined shaving and burnishing method of finishing gear teeth known as the Burni-shave, by the National Tool Company of Cleveland, O. The tool consists of a hardened helical gear with its face divided into comparatively narrow sections by grooves cut to the bottom of the teeth. It resembles a series of thin helical gears with spaces between them. The edges of the cutter teeth shave the gear while the faces burnish. The faces act as burnishing tools. These tools replace the hardened master gears on any burnishing machine or fixture.

Lapping Spur and Helical Gears.—Probably the first attempt to lap gears was to run them together with a fine abrasive compound. Usually a load was applied and they were also run in

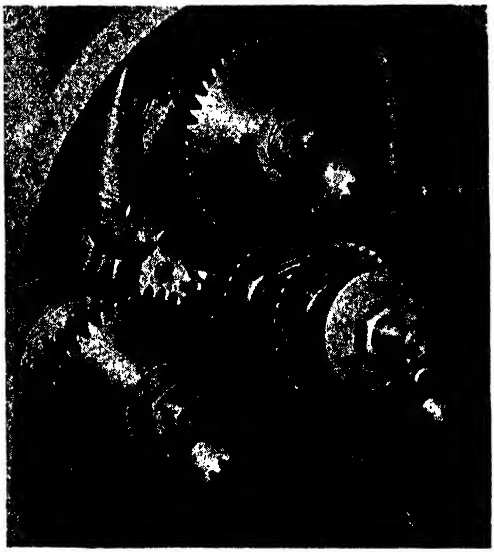


FIG. 136.—Lapping a transmission gear between three master gears.

both forward and reverse directions. In manufacturing plants it is customary to run the gears with master or lapping gears instead of the gears with which they are to run. These lapping gears are usually of cast iron.

In one type of machine the gear to be lapped is run between three master gears, as in Fig. 136. In another type, as the Fellows planetary machine shown in Fig. 137, the lapping gear is a wide-faced internal gear which permits several spur or helical gears to be lapped at the same time. In addition to the rotary motion of the ring gear it has a reciprocating movement as well. In the illustration shown there are six small gears on each of the three spindles. These gears come directly from the heat-treating department without removing them from the mandrels on which they are hardened. The 18 gears are lapped in about ½ min. Both spur and helical gears are lapped by the same method, different laps being used for gears of varying diameter.

Another method which had some advocates is to cast a soft metal form around a gear of the size to be lapped, but with a wider face. This internal gear formed the lap and the gear was reciprocated through this soft gear. After a given number of strokes the gear came out of the lap and was indexed one tooth or more, when the lapping was repeated. This method gave fairly satisfactory results but has been replaced by those requiring much less time.

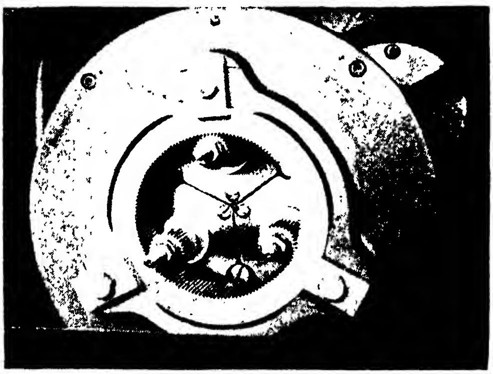


FIG. 137.—Using an internal master gear for lapping.

Gear-lapping Practice.—There are differences of opinion as to the merits of lapping gears, largely due to wrong practices in the past. Some advocate axial reciprocations, others that the center distances be changed slightly during the lapping, while still others contend that lapping at crossed axis corrects the unlapped pitch-line condition if the amount of crossed axis and speed of reciprocation and the brake pressure are properly selected to suit the gear being lapped.

An example of the three-lap method is seen in the Michigan Tool Company's machine in Fig. 138. This shows the lapping of a herringbone gear by the use of three helical laps, each side of the herringbone gear being lapped separately. This machine permits the three laps to be set with their axis at an angle, measuring pins being provided for use with gage blocks for setting the angle of each lap. After the machine is set up, the two lower laps are held in a fixed position, the upper lap being raised and lowered as the work is handled in and out of the machine.

The makers estimate a production of 20 gears per hour of 3 in. diameter and 10 pitch. Errors of from 0.001 to 0.002 in lead can be corrected to from two to three "tenths" on most helical gears.

Modern Gear-lapping Methods.—According to R. S. Drummond, an engineer of wide experience in gearing, recent advances

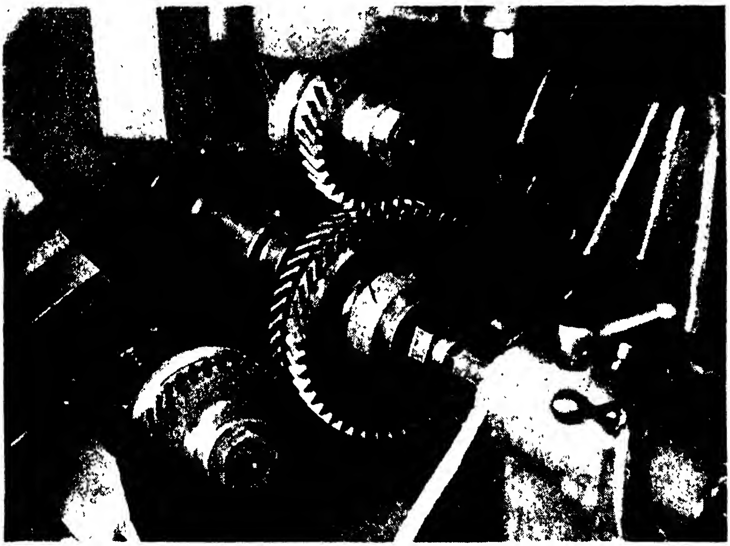


FIG. 138.—Lapping each side of a herringbone separately.

in gear-correction practice have made it possible to remove an amount of material hitherto considered impracticable, if not impossible. In fact, it is now economically practicable to remove all the distortion resulting from normal heat treatment. Lap life has also been materially extended, and in certain instances, from 500 to 1,000 gears are corrected without effecting an error of more than 0.001 in. in the profile of the lap, even though the chordal thickness of the lap teeth may be reduced as much as 0.030 to 0.060 in.

Insistent demands for higher quality have resulted in the development of two new lapping practices within the last three years. Both are based on the use of a separate lap and a crossing of the axes of gear and lap. The first of these is known as cramp lapping, in which the lap tooth is given an abnormal thickness and in which the feed reduces the center distance between gear and lap. The lap teeth contact both sides of the gear tooth; cutting is accomplished by a lapping compound fed into the gears during operation.

Cramp lapping is rapid and is an ideal means of correcting excessive eccentricity or index. It has become standard practice in a number of progressive shops manufacturing automotive gears. Any material change effected in the center distance between gear and lap is compensated by a change in the angular setting of the axes, inasmuch as the lap and work gears mate in the same way as a pair of skew gears. This angular relation of the two axes is controlled by a vernier fixture on the machine column. The setting is controlled by the use of an angle-checking fixture which indicates the tooth error by the use of a rack tooth and integral sine bar. The radius of the sine bar is the same as a division on the vernier column scale of the lapping machine.

Although the worm lapping process was the first to use crossed axes, the angle between the axes in that process is approximately 83 deg. The practice of crossed-axis lapping introduced in 1930 reduces this angle to less than 30 deg., and it frequently ranges below 10 deg. The angle used in any given case is the difference between the helical angles of the gear and the lap.

The results attained by those newer methods have been so significant that practically 75 per cent of the lapping tools purchased since their introduction incorporate the new principles. One outstanding case comes to mind of a set of gears having the following errors: involute, 0.004; eccentricity, 0.005; cumulative index, 0.008; helical angle, 0.005 on a 3-in. radius. These gears were corrected to a final involute error of 0.0005; eccentricity, 0.001; cumulative index, 0.001; helical angle, 0.001 on a 3-in. radius. Although considerable material had to be removed to attain this result and backlash was considerably increased, it is an extreme example of what can be done by means of the newer lapping methods.

In the second of these methods, known as power tailstock lapping, the center distance between gear and lap remains constant, and a power tailstock is used to effect a braking action on the work gear, which is driven by the lap. After the work gear has been driven a given number of revolutions in one direction, rotation is reversed and progfesses in the opposite direction a number of revolutions to complete the cycle. The brake operates throughout the continuous cycle, first in one direction and then in the other. Thus both sides of each work-gear tooth are given the same amount of processing. As in cramp lapping, already described, the axes are crossed. In both methods, an automatic electric reversing and timing mechanism controls the number of revolutions.

Lapping Materials.—The cutting or lapping compound has also been given a great deal of study and has shown remarkable improvement in recent years. By the improvement of lapping compound, lap life has been raised from approximately 100 gears to 500 and as high as 1,000 in special cases. A lap which was used for automotive transmission gears shows a total wear of 0.079 in. in the chordal thickness but a variation in profile of only 0.001 in.

Too much attention cannot be devoted to the proper choice and inspection of grain size and quality in a lapping compound. There is a vast difference in the character of individual grains in any given-size material. For instance, grain specified for abrasive paper is quite unsatisfactory for gear lapping because it contains a high percentage of needles and slivers. Grain for lapping compound should be of a rounded, irregular shape but never needle-like. Tests have proved the assertion that efficiency of cutting action and lap life can be greatly increased by a proper choice of grain and lubricant. Some use two lubricants, one an oil-soluble solution for washing in kerosene and the other a water-soluble solution for washing in oil or soda water. The same size and quality of grain are used in each and the same amount of grain. Nevertheless, by carefully conducted tests, the water-soluble compound is much more efficient than the one having the oil base. The average shop man does not appreciate the potential saving available in the proper choice of a cutting medium.

Standard lapping time today for reasonably well cut gears is about 2 min. per unit having a 1-in. face and 4-in. diameter. This assumes reasonable control of gear cutting and heat treatment.

One very good example was a lot of 30,000 gears. The rejection for gear noise after lapping was only 1 per cent, and the lapping time averaged 1 min. per gear. The lap life on this job was approximately 900 gears per cutting, and the lap was recut twice, which gave a total life of 2,700 gears. These were standard automobile transmission gears of several well-known designs which were accurately cut and heat-treated.

When excessive distortion occurs because of high carbonizing heats on large gears, it has been found advantageous to take a single roughing cut with a grinding machine so as to remove this excess before final lapping. In one case in which the grinding time on 12-in. diameter gears was 67 min., it was possible to use a single grinding operation of 15 min. and lapping time of 6 min., a total of 21 min.

Oil-treated Gears.—In the lapping of oil-treated gears, the standard lapping time varies from 1 to 4 min. on gears of 1-in. face and 4-in. diameter, and the proportional size of the gears indicates the increased lapping time required for the same cutting and heat-treatment conditions.

The use of accurate laps in the finishing of heat-treated gears gives the shop operators a clear indication of the errors in the cutting, as the high parts are easily recognized. On production runs, the cutting operations are frequently subject to prompt investigation because of the appearance of the gears after normal lapping, which shows the profile or angle of the gears to be in error.

Lapping Spiral-bevel Gears.—In order to remove slight inequalities due to distortion in hardening and also to secure a highly polished surface, the Gleason Works have developed a combination testing and lapping machine especially for spiral-bevel gears, as shown in Fig. 139. This machine imparts an oscillating motion to the gears being lapped, this consisting of an in-and-out motion along the pinion axis and an up-and-down movement of the gear. These motions are secured by cams.

When lapping spiral-bevel or hypoid gears, it is often found desirable to locate the heads of the machines so that the distance from the center of the gear spindle to the face of the pinion mandrel varies a few thousandths of an inch from the correct cone center to the nose of spindle dimension. Several factors must be considered in determining the best setting at which to lap the gears. First, the tendency of the pinion when under load in the carrier is (for the usual left-hand spiral combination) to move away from the cone center on the drive side and toward the cone center on the reverse side. An allowance for this would call for lapping the drive ("bottom") side with an increased and the reverse ("top") side with a decreased mounting distance, the amount of increase or decrease depending upon the amount of pinion movement. Next, account must be taken of the lapping action which is different for spiral bevels and hypoids. Spiral

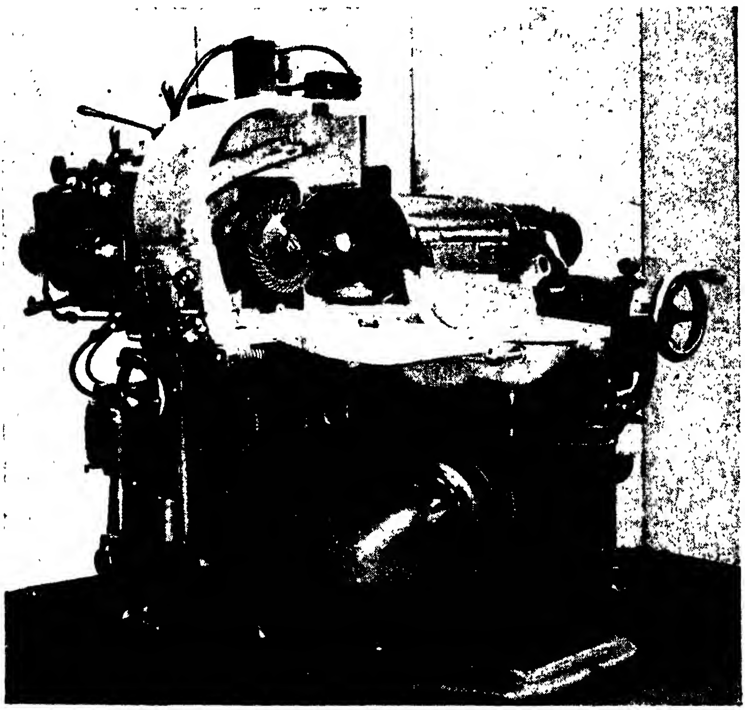


FIG. 139.—Lapping spiral-bevel gears.

bevels, which have only rolling motion at the pitch line, must be moved out of position to lap the pitch line, but for hypoids the sliding action which is inherent in this type of gear laps the pitch line in any position. Finally, it has been found that spiral bevels usually run "in" (meaning that they will show a good bearing at a position which is closer to, but not farther from, the cone center than the lapping position) while hypoids run "out."

From the foregoing it is evident that the best setting for lapping the gears must be found by experiment. Usually there will be one setting for the "top" side and another for the "bottom" side.

In setting up the machine it is customary first to locate gear and pinion at their correct mounting distances and set dials at zero. Then after the gears have been run and the bearing observed, the dials can be used for reference if the position of the heads is to be changed for lapping.

Sometimes when lapping gears which show a heavy heel or toe bearing, a change in vertical setting is made in order to concentrate the lapping action at the large or small end of the tooth.

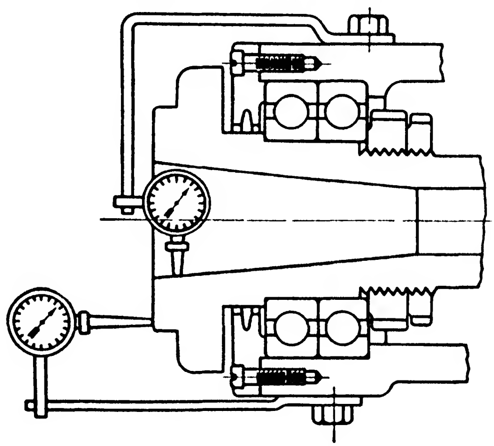


FIG. 140.—Checking accuracy of lapping spindles.

If this change should be more than 0.010 in. the lapping motion should not be used because a cross bearing will be introduced which might harm the tooth profile of the side which is not being lapped.

Never knock the mandrels out of the spindles. The shock may chip the ball bearings or break the oil seal allowing the oil to escape.

A spanner nut should be provided with each mandrel for removing it from the spindle. This is done by screwing the spanner nut against the face of the spindle.

It is necessary that the spindles of the lapping machines be checked frequently for accuracy as this is important if the best results are to be secured. Figure 140 shows how the indicators should be placed for checking. Neither the radial nor the face runout should exceed 0.0002 in.

Lapping Compound.—The Gleason Works recommend a lapping compound of a nonfluid oil and abrasive. For the finest work they suggest Turkey oilstone powder and No. 000000 nonfluid oil that will hold the abrasive in suspension, in the proportions of 7 lb. of powder to 10 lb. of oil. While coarser abrasives are sometimes used, they do not produce so satisfactory results. To make up for the loss of abrasive qualities with use new abrasive should be added whenever needed, and the compound should be cleaned out of the machine once a week and be replaced with a fresh mixture.

When starting up a new machine or entirely renewing abrasive mixture it is best to put in all the oil first and, with pump running, gradually add the abrasive, taking care that the mixture does not become too thick to be handled by pump.

Gear lapping, to be successful, must be a polishing operation to produce a smoothness that would normally be attained (for automobile gears) after 500 miles of car operation. It is apparent, therefore, that the lapping compound used should be a very mild abrasive, such as Turkey oilstone powder. When salvage work must be performed and a material change made in the tooth form to correct errors in cutting or heat treatment, a coarser abrasive may be used as a time saver with a natural lessening in the quality of smooth operation.

The accepted standard method of lapping provides for smoothing the entire working surface of the tooth. This is important since a fixed tooth-bearing position cannot be maintained and some latitude must be allowed for the usual machining tolerances. Gears mounted and lapped in one fixed position will be satisfactory, however, if the carrier in which they are held is extra rigid in construction and is selected for center height so that the gears, after being mounted in it, will show the same bearing as on the testing machine.

In connection with production lapping there are two essential points—control of cutting and control of heat treatment—which are very important when producing closely uniform gears and pinions. If variations in cutting or heat treatment occur only to a small extent, the tooth bearing of gears operated at correct mounting distances will not extend to either end of the tooth, although it may shift slightly along the tooth. When the variations are large, the tooth bearing may become short and heavy at one end of the tooth, which defeats the idea of production because special settings and additional operations are required to transfer the tooth bearing to a suitable operating position for the final mounting.

There must always be sufficient length of bearing to insure smooth, even operation without depending entirely on the profile contact for tooth overlap.

Grinding Helical Gears with Generating Wheel.—The way in which gear teeth are ground in a Pratt & Whitney machine is seen in Fig. 141, where two gears are being finished at the same

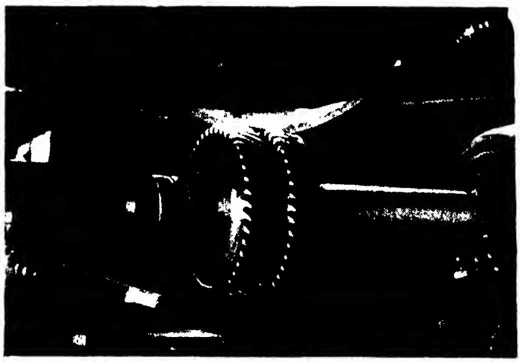


FIG. 141.—Grinding helical gears with a generated tooth form.

time. The grinding wheel is mounted in a head resembling a shaper and travels back and forth across the work, which is set at the proper angle and rotates as the wheel traverses the cut. The grinding wheels are so dressed as to cut on the side on a flat face, as the teeth are generated, instead of being ground with a wheel formed to the contour of the tooth.

The principal field for this method is where gears of great accuracy but used in comparatively small quantities are needed, such as in high-grade machine tools and similar work. Where great quantities are needed, as in automotive work, gears are usually shaved or burnished or both and finally lapped after heat treatment.

Stock Control for Grinding Spiral-bevel Gears.—The problem of dividing the stock on hardened spiral-gear teeth depends on the warpage of the teeth during the quenching operation. The quenching press is fitted with a plate clamp which holds the ring gear to a fairly even shape during quenching, but the shape

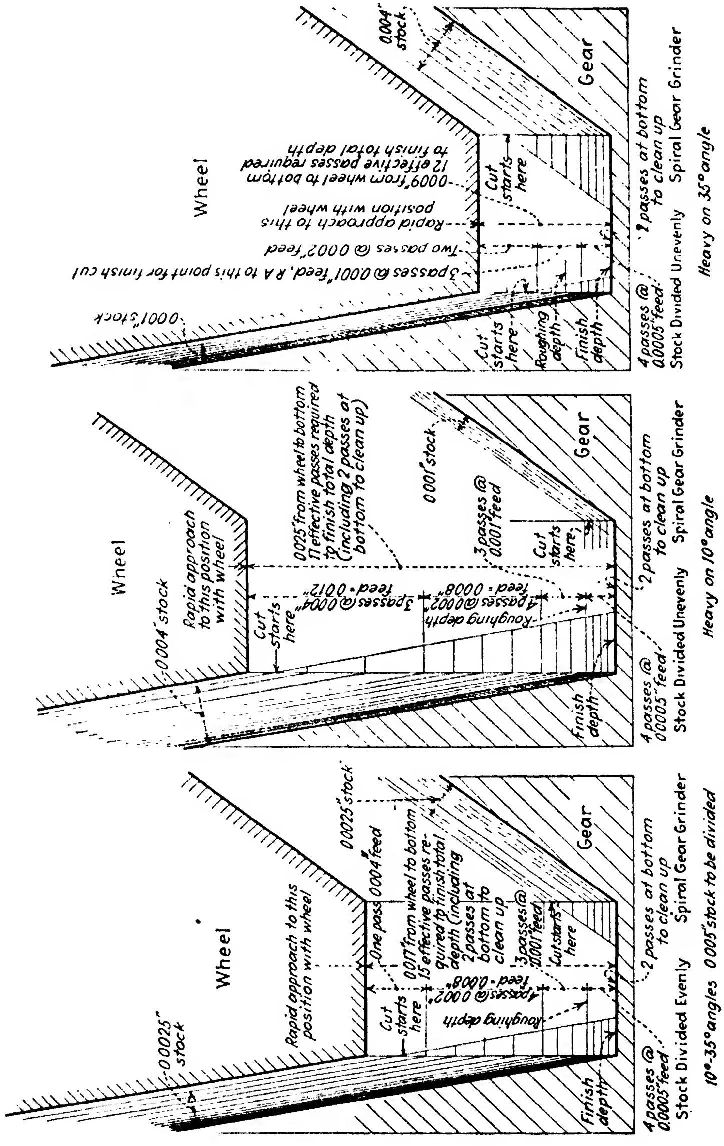


FIG. 142.—Stock control, showing metal removed from each side of tooth.

of the teeth after quenching may bring one of three different conditions in dividing the stock to be removed. If the gear teeth remain uniform, they will come out of the quenching operation with the stock divided evenly. If there is warpage, the stock may be heavy on the acute angle side or vice versa as the case may be.

As illustrated in the production studies and in Fig. 142, with the stock divided evenly, 15 passes will be required to finish. In case of the stock being heavy on the steep side of the tooth, 17 passes will be required to finish. In the case of the stock being heavy on the flat side of the tooth, only 12 passes will be required to finish. While this may not seem like a great difference, yet it must be noted that in a gear of 100 teeth the difference between extreme conditions would be a matter of 500 passes of the grinding wheel which would add appreciably to the production time for the piece.

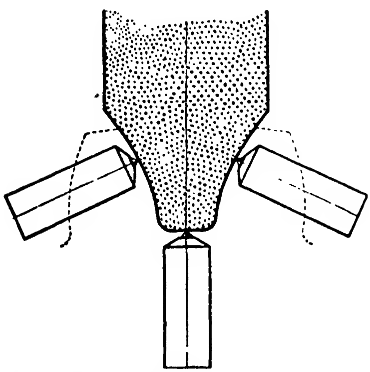


FIG. 143.—Dressing abrasive wheel for form-tooth grinding.

As will be seen from the production studies illustrated, the best possible condition for the balance of the grinding-wheel pressure in the cut and for production time is with the stock heavy on the flat-angle side of the tooth.

Tooth-profile-comparator projections will reveal the division of the stock; control may be established depending on the metal warping characteristic, to hold the division most advantageous for production.

Formed Wheels for Gear-tooth Grinding.—The formed-tooth method of grinding gear teeth resembles the use of the formed cutter in cutting the teeth. In both cases the cutting tool is made to conform to the desired shape of the tooth. With the grinding machine the tooth contour is secured by the use of three diamonds, as shown in Fig. 143. These diamonds are mounted in holders on the machine and their movement is controlled by form plates or templates which are six times the size of the tooth itself. These templates, in turn, are made from master forms that are eighteen times the size of the tooth. In this way any error is greatly reduced so that a practically perfect tooth form is secured in the wheel and in the work. Both the indexing and the grinding may be compared to the cutting of gear teeth with formed cutters.

Internal gears can be ground when they are of sufficient size and there is space for the grinding wheel to pass through, as in Fig. 144. Both these examples are from the Gear Grinding Company, Ltd., of Birmingham, England, which is an outgrowth

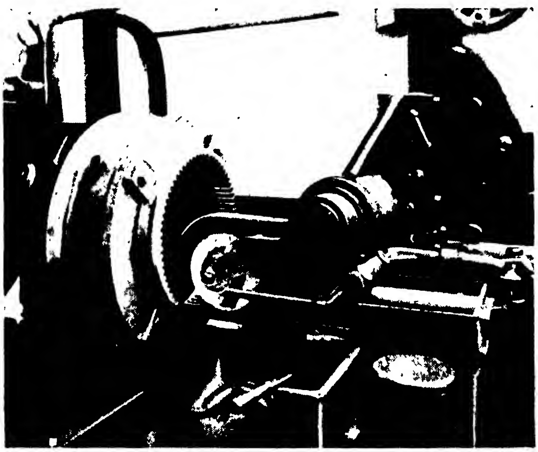


Fig. 144.—Grinding large internal-gear teeth.

of the similar company in Detroit. A train of ground gears drives the grinding wheel and a single diamond dresses both sides of the wheel.

Gears for Automobile Transmissions.—The demand for quiet gears began to grow with the increasing use of closed cars, as noise of any kind is more apparent here than in open cars. Gears are frequently blamed for noises that originate in bearings or elsewhere. According to the Gear Grinding Company of Birmingham, England, who have long specialized on precision gearing, much of the fault lies in the design of the gear box itself and not in the gears. While, they say, the accuracy of gears can be guaranteed, the running of these gears varies according to the design of the gear box in which they are run. Gears without extreme accuracy run comparatively quiet in well-designed boxes while really high-class gears may be noisy in other boxes.

As an indication of the character of the work done by this company, which makes their judgment regarding gear-box design of real importance, we give the limits of accuracy to which they guarantee their gears.

Small gears as for automobiles, aeroreduction and supercharger gears, motorcycle timing gears, machine tool gears, and all gears below 12 in. outside diameter have the following limits:

Eccentricity.............	0.0005 in.	0.0127 mm.
Tooth contour...........	0.0002 in.	0.0051 mm.
Index error..............	0.0001 in.	0.0025 mm.
Parallelism..............	0.0001 in.	0.0025 mm. (in 3 in)

Large gears as for electric railways, and large reducing gears, as well as all gears above 12 in. outside diameter, have the following limits.

Eccentricity.............	0.001 in.	0.0254 mm.
Tooth contour...........	0.0005 in.	0 0127 mm.
Index error..............	0.001 in.	0.0254 mm.
Parallelism.............	0 001 in.	0.0254 mm. (in 6 in.)

Grinding the Bores of Gears.—In grinding the holes in gear hubs after hardening it is necessary to hold the gear accurately by the pitch line of the teeth. There have been many methods used for this work, but the latest seems to be to use a cage which carries floating rollers that contact the pitch line of the gear to be ground. A section view of such a chuck of the Heald Machine Company is shown in Fig. 145. The cage is separate from the chuck. It is slid over the gear with the rollers between the teeth and the outside of the rolls are clamped by the chuck jaws. The rolls must be of the proper diameter to bear approximately on the pitch line of the gear and project above the gear so as to be clamped by the jaws.

A much more intricate design is seen in Fig. 146 where a spool with a gear cut on each end is being held by both gears. The rolls in the rear jaws, which contact the smaller gear, are fixed, but the outer jaws have a separate cage as in the other design. This can be used on a plain machine or with the Gage-Matic of the Heald Company.

Another example, the grinding of the bore of a small gear, is seen in Fig. 147. The chuck consists of two essential parts, the holder for the gear, shown on the spindle, and the centering chuck itself, seen on the cross rest of the lathe. The holder is a disk with eight pins that fits between every third tooth of the 24-tooth gear, on the pitch line. The disk of the holder is ground true both on the faces and with the outside diameter of the pins.

The body of the chuck is bored and ground to a close sliding fit for the outside of the pins in the disk, and square with the face to afford a seat for the disk when in place. With the gear in place in its holder as shown, the knurled cap is unscrewed from the chuck, the gear holder put into the chuck body, and the cap screwed in place in this way the gear is held true with the pitch line and the bore is accurately ground.

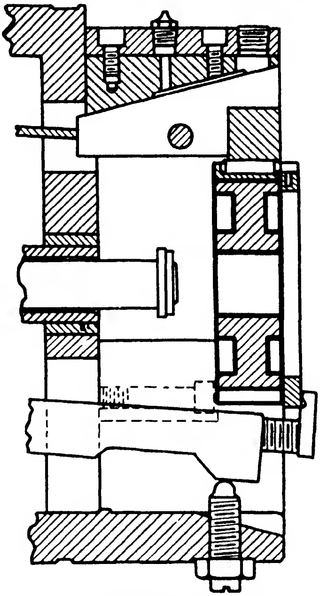


Fig. 145.—Holding a gear by rollers between the teeth.

Chucks for bevel gears locate the gear on pitch-line rollers, while clamps hold the gear in contact with the rolls.

Gear-holding Chucks.—The Garrison gear chuck, shown in

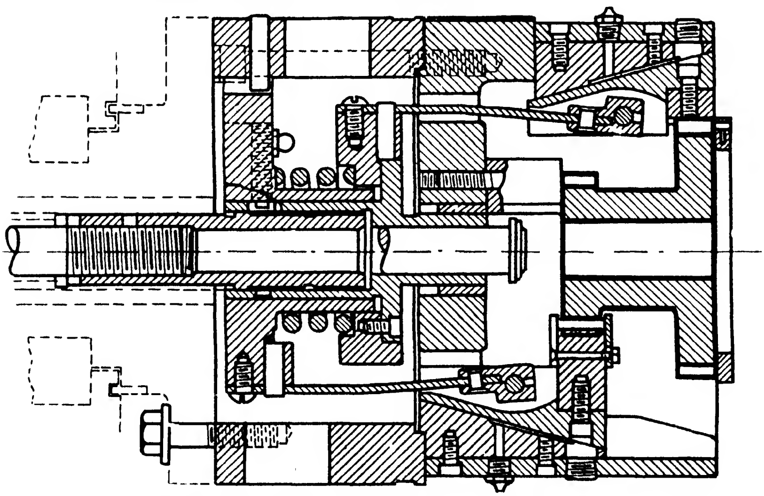


Fig. 146.—Double chuck which centers both gears by rollers on pitch line.

Fig. 148, holds the gear to be ground, or diamond-bored, by pinions that bear on the pitch line of the gear, and center it by the pitch circle. Either helical or plain spur gears are held in

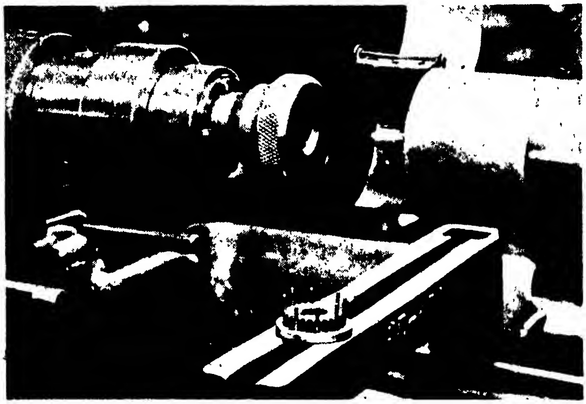


Fig. 147.—Chuck for grinding bore of small gears.

the same way. The pinions are slightly eccentric and a partial turn of the gear itself both centers and locks it in place. In some cases a wrench is used to turn the holding pinions. By

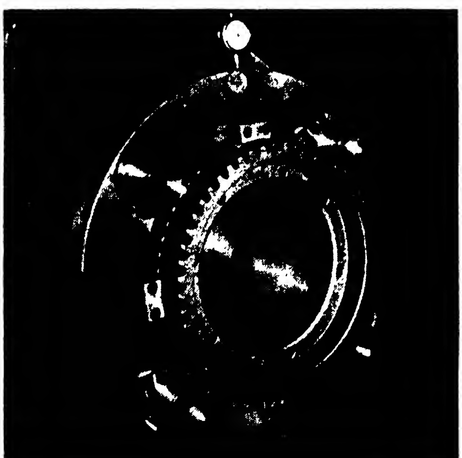


Fig. 148.—Garrison chuck for holding gear by pitch line.

reversing the pinions they will hold internal gears as well. The dial indicator checks any runout.

Bevel pinions and gears are held by conical pins that contact the teeth on the pitch line, as in Fig. 149. Here the pressure of the center, on the end of the pinion shaft, keeps the work in place during the grinding operation.

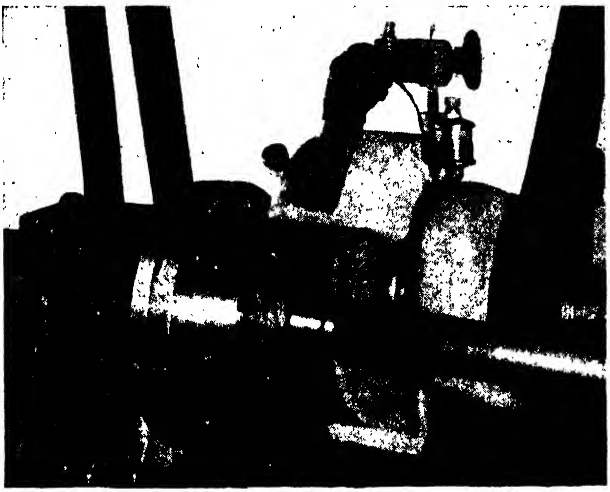


FIG. 149.—Centering bevel gear on conical pins between teeth.

Ring gears are held in a similar manner but are clamped by a ring on the back face, as in Fig. 150. The bevel of the chucking plate centers the gears while the pins, bearing on the pitch line

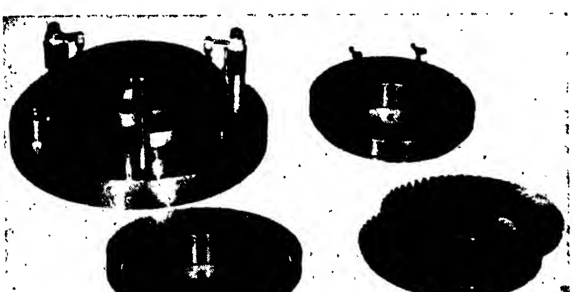


FIG. 150.—Two methods of clamping ring gears.

of the teeth, hold the gear square with the tooth outlines. Figure 150 shows two methods of clamping, one from inside the ring gear and the other by means of clamping posts on the outside.

A convenient cabinet for storing a number of these chucks is seen in Fig. 151. It is in use in the shop of the Monarch Machine Tool Company. When the door is raised concealed lights illumi-

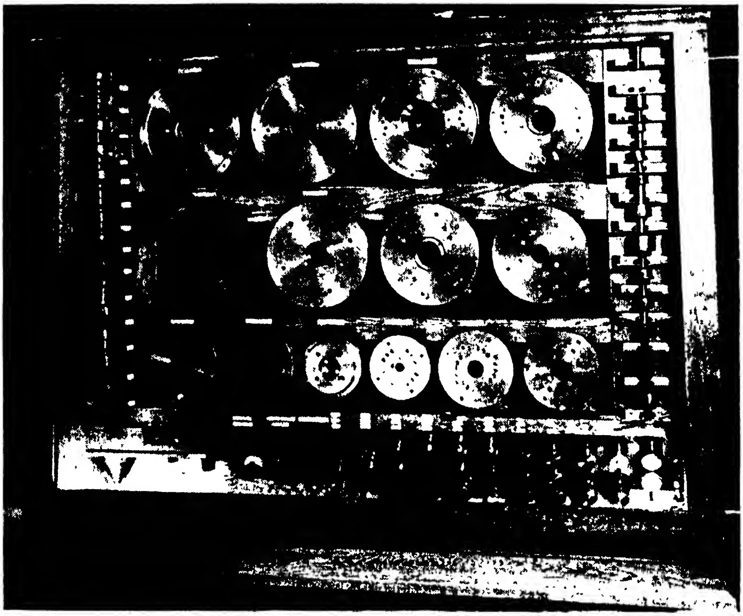


FIG. 151.—Cabinet containing a set of gear holding chucks.

nate the interior and allow any chuck to be set up for the machine before its removal. Beneath the top shelves are setup charts which can be pulled out, in plain sight of the setup man. A roll cover protects the chucks as well as conserving space in the shop.

CHAPTER X

HEAT TREATMENT OF GEARS

According to E. F. Davis, metallurgist of the Warner Gear Company, heat treatment of gears (and other machine parts) begins when steel is made and extends through all the heating processes, including the forging. Gear teeth are subjected to great strains and the directing of the fibers plays an important part in the strength of the gear. In one case where there was trouble from breakage of a driving pinion in an automobile, the method of making the pinion was changed from a drop forging to upsetting, and the increased strength secured by this method stopped the breakage.

Forgings should be sufficiently heated and worked under hammers of sufficient size. The finishing temperature is more important than the forging temperature. After forging they should be normalized by heating up to 1700 to 1850°F., holding for 1 or 2 hr., and then cooling at a fairly fast rate to prevent grain growth. Annealing is not the same as normalizing. Gears annealed but not normalized change more in final hardening than gears which are both annealed and normalized. The steel must be given the best structure for the operation of cutting the teeth, as the cutters used in the other operations can be modified if necessary.

Most gears are carburized, with a case from 1/64 to 1/8 in. deep, the depth depending on the temperature used and the length of time in the furnace. The temperature varies from 1650 to 1750°F. With a fairly uniform temperature there is seldom any material change in the gear during carburizing. When carburized gears are hardened the movement is toward the hole; the periphery and the pitch line change very little. With oil-hardening steel, gears almost invariably expand permanently in hardening, increasing the pitch diameter. This increase usually approximates 0.001 in. per inch of diameter. The rate of cooling is one of the most important factors in gear distortion. Quenching conditions should be adjusted to the mass to be hardened and the cooling rate kept as low as will give the desired hardness. In selecting gears it should be remembered that carburized gears are used in the higher class automobiles.

The lead pot has been a favorite method for heat-treating of gears and is still in use in several plants. The chief objection to this method of hardening is the adherence of the lead to the parts and the inability to control the temperature because of its high heat conduction.

The cyanide method of hardening is at present the most popular, not only on account of the method itself but because of the useful properties imparted to the surface of the gear. Probably 75 per cent of the gears produced are now being hardened in a cyanide bath.

There are several ways of performing this operation. One is by preheating the gears in a rotary or Hump furnace and then giving the gears an immersion for about 1½ min. in cyanide; another is by direct immersion in a stationary furnace; and the third is by the continuous cyanide furnace. The cyanide method is of sufficient importance that some of its metallurgical features will be discussed.

Many gear makers still hold a prejudice against dipping a cold gear into a molten bath believing that this act produces distortion. There is a fostered belief that, unless steel is heated slowly and carefully, it will warp, and to take a cold gear and plunge it into a molten mass of salt is violating all the principles of good heat-treating practice. They prefer to preheat the gear and merely to dip it into the cyanide after it has had the slow and careful treatment. This is the so-called cyanide dip method. When such a gear is first placed in the cyanide bath a more or less violent reaction occurs caused by the cyanide attacking the oxide of iron on the gear surface, for cyanide is a solvent for oxides. In about a minute, this reaction subsides and the gear may then be removed and quenched. Quite a bit of cyanide adheres to the surface; hence the take-out loss is high. For that reason the cheaper grades of cyanide such as the 45 per cent should be employed. The cyanide-dipped job is not file-hard, and the case is comparatively low in nitrogen.

The direct-cyanided gear is immersed in the cyanide bath without any previous preheat except drying, and is brought up to temperature in the cyanide itself. After a definite period, which seldom exceeds an hour, it is removed from the bath and quenched.

Continuous Cyaniding.—In the continuous type, the work is carried through the bath by means of a conveying mechanism exterior to the bath itself. This latter type of furnace may be automatically controlled within 20°F. because at no time is any heavy mass of metal added to the bath. The pots used are of necessity large. Therefore the screw-type conveyor should be employed. The Buick furnace has a rotary mechanism and is circular and of a doughnut shape. These furnaces are practical only for firms having considerable production. Another advantage of the continuous type is that the time in the cyanide furnace is regulated mechanically by the conveying system, and the human element is removed.

Anyone contemplating cyaniding should not forget to use a lower carbon steel than would be necessary for the furnace-treated oil-hardening grades. Thus a 0.50 chrome steel would be too brittle when cyanided. A 0.40 carbon-chrome steel when cyanided would have the same impact values as a 0.50 carbon-chrome steel. The cyanide process should be regarded as a seminitriding process for nitrides as well as carbides constitute the case.

The latest modification of the cyanide process is the energized bath of which the aero case method is representative. This produces a case similar to cyanide but is somewhat higher in carbides and lower in nitrides. In this type of bath, a neutral salt melting at about 950°F. is used. To this are added various activating agents of which calcium cyanide is typical. Some objection to the use of calcium cyanide is its decomposition into lime which is difficult to wash from the surface.

Wearing Qualities of the Case.—The advantage of the cyanide case is its excellent wearing qualities. Dynamometer tests have shown the cyanide case to be the best wearing of any kind of tooth surface. Taking the furnace-hardened gear steel of 0.50 carbon as a unit, dynamometer results indicate that the cyanide-dipped case will give four times the wearing life of the furnace-hardened, and the carburized surface eight times the oil-hardening furnace-treated steel. The cyanided surface gears show 11 times the wear of the furnace-hardened gear. It will be noted that the cyanided gear wears even better than the carburized gear. This is caused by the extremely hard nitrides of the cyanided case. The extreme outer layers for a depth of 0.0005 in. will show a nitrogen content up to 14 per cent. Although the preceding results are based on the time required for overloaded gear teeth to show pitting, yet in actual field results a worn or pitted gear is rare.

Another myth is that the process of cyaniding makes noisy gears. Bad tooth form is the chief cause for noisy gears.

Since adopting the cyanide process, Mr. Davis states that the wearing qualities of gears have improved and the breakage has decreased. Worn gears do not last long after the tooth form is destroyed by wear and pitting, and such gears fail quickly from fatigue. If the involute is maintained, better gear life can be expected.

How Buick Handles Gears Steels.—In a recent paper before the Society of Automotive Engineers Mr. R. B. Schenck, of the Buick Motor Car Company, outlined the practice of that company in the heat treatment of gears. Liberal extracts from his paper, beginning with the annealing of the steels, follow:

Annealing.—Much depends upon the operation of annealing, the purpose of which is to develop the most suitable structures for machining and hardening. Fortunately, the best structure for machining is usually the best for hardening.

Annealing temperatures and time cycles vary with the different steels, depending upon their individual characteristics and variations in plant practice. The double treatment, consisting of normalizing and annealing in two separate operations which was extensively used at one time, is now practically obsolete, and has been replaced by the single high-temperature treatment.

Continuous annealing furnaces with automatic temperature regulation, as shown in Fig. 152, are now in general use and permit much better control of the annealing cycle than was possible with the old-type batch furnace formerly employed.

The low-carbon steels are either furnace-cooled or air-cooled through the lower critical point from temperatures in the neighborhood of 1750°F. or higher. Steel 2515, with its high alloy content, has a tendency to air-harden and requires slow cooling. Practice varies with 4615 and 4620, some plants using an air cool and others a slow cool. In air-cooling, the gears should be spread out so as to obtain as nearly a uniform cooling rate as possible. Sometimes the charge is allowed to drop 100° or more in the furnace before discharging for air-cooling.

The high-carbon steels all require slow cooling through the lower critical range. Annealing temperatures vary, but usually range from 1550 to 1750°F. These steels are extremely sensitive to the rate of cooling through the critical point and the greatest possible uniformity is necessary in order to maintain a desired structure. They are also sensitive to variations in annealing temperature, an increase in temperature requiring a slower rate of cooling.

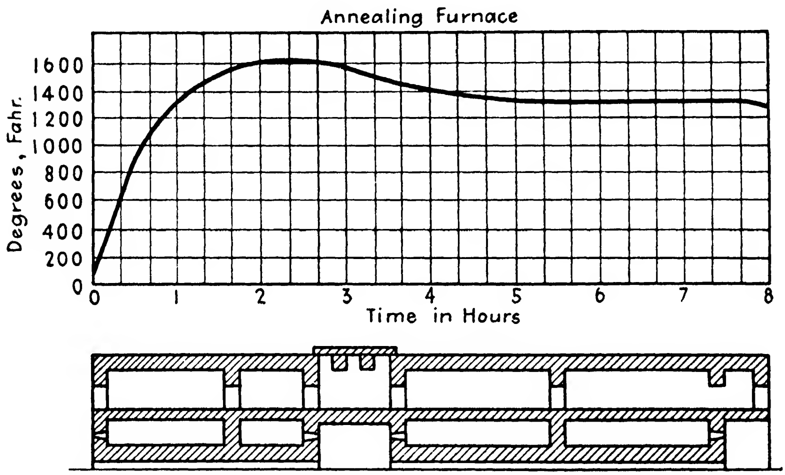


FIG. 152.—Automatic regulation of continuous annealing furnaces.

A cooling rate which produces lamellar pearlite or its equivalent seems to give the best all-round results. The 4600 steels do not readily form lamellar pearlite, but develop a corresponding optimum structure of their own which is peculiar to steels containing molybdenum. Structures containing more than small amounts of either sorbite or spheroidal cementite are liable to cause trouble in machining. Excessive spheroidization in the high-carbon steels may cause serious difficulties in hardening where short-time cycles are employed.

Differences in grain size have an effect on pearlite formation and tend to produce irregularities in structure. A fine grain must be cooled through the critical point more rapidly than a coarse grain to develop the same amount of pearlite. The so-called duplex or mixed structures are, for this reason, more difficult to anneal properly than either a uniformly coarse or fine grain.

Since, in duplexed steel, the various-sized grains start rapid growth at different temperatures, the best results should be obtained by using either a very high or a comparatively low temperature for annealing. A sufficiently high temperature tends to produce a uniformly coarse grain, while a temperature below the coarsening point of all the grains will develop a uniformly fine grain. Since coarse-grained steel generally machines more freely, the high-temperature anneal should be preferable. High temperatures are also more effective in correcting banded structures which sometimes occur in an aggravated form, and tend to cause poor machining and excessive distortion in hardening. However, temperatures in excess of 1750°F., which is probably below the minimum for uniform coarsening, produce excessive scaling and high maintenance cost of equipment and are not extensively used.

Equipment for cyanide hardening comprises a multitude of different designs and arrangements. The simplest form consists of a manually operated single-pot furnace with the gears placed on the brickwork

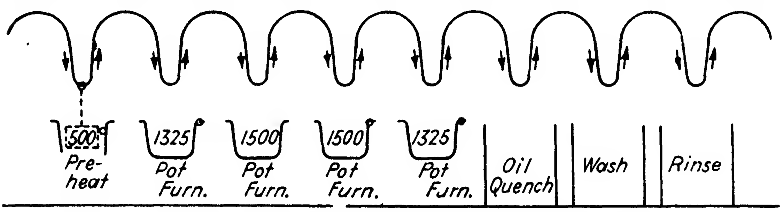


FIG. 153.—Cycle of automatic passage of work through the furnace.

around the flange of the pot for preheating. A more advanced design is shown in Fig. 153, where the stock is carried automatically through the preheat, the four cyanide pots, the quench, the wash, and the rinse.

The annealing operation is performed in underfired continuous furnaces with automatic temperature control, as shown in Fig. 152. These furnaces have four separately controlled zones and are extremely flexible. The central cooling zone with the thin removable roof permits accelerated cooling from the second high-temperature zone. This saves annealing time and tends to decrease banding. Figure 152 also shows the time-temperature cycle. The annealing capacity of each furnace is 1,250 lb. per hour.

Hardening.—In modern practice, nearly all transmission gears are case-hardened. Low-carbon gears are pack-carburized and high-carbon gears are hardened from cyanide or some similar activated bath. The only exception to this practice for high-carbon gears is a new development in controlled-atmosphere hardening which will be described later. While cyanide hardening of gears has been employed for many years, it is only recently that it has come into such general use. Formerly, large quantities of high-carbon gears were hardened from lead pots and atmospheric furnaces, but recent demands for a more wear-resisting surface have made these older methods obsolete.

The usual practice with low-carbon gears is to harden directly from the carburizing box. With the fine-grained steels now available, this treatment develops very high physical properties and greatly simplifies the hardening procedure. The carburizing temperature is usually 1700°F. The gears are quenched from this temperature or the box may be allowed to cool 100° or more before quenching. In cases where distortion requires closer control and maximum core properties are not essential, the gears are cooled in the carburizing box and reheated in cyanide to 1400 to 1500°F. for hardening.

The carburizing furnaces are usually of the continuous type with automatic temperature control and may be either fuel-fired or electric. Boxes are generally made of heat-resisting alloy and are either cast or fabricated from rolled alloy by welding. Both cast and rolled alloy are often used in the same box. An arrangement sometimes used to insure uniform quenching conditions consists of a frame in the quenching tank, similar to the partitions in an egg crate, which provides a separate compartment for each gear.

Two different methods are used in the cyanide hardening of high-carbon gears. The one consists of a complete heating cycle in cyanide, and the other, a cycle which starts in an atmospheric furnace and finishes in cyanide. The first method is the one in most general use.

In the first method, the gears are usually preheated to a low temperature before they are placed in the cyanide bath. This removes moisture and prevents explosions which might otherwise occur. The preheat also lessens the time required to reach full temperature in the cyanide. The preheat furnace may be independently fired or it may be heated by the products of combustion from the cyanide pot.

In the second method, the stock is brought to temperature in an electric or fuel-fired atmospheric furnace and transferred to the cyanide bath. Before placing in the cyanide, the gears are usually given a rapid wire brushing to remove the scale which forms in the first furnace. The time in the cyanide varies from a few seconds, sometimes called a "dip," to an exposure of several minutes. The temperature of the cyanide may be the same as that of the atmospheric furnace or it may be lower. It is generally believed that a drop in temperature to just above the upper critical point before quenching tends to reduce distortion.

The correct time-temperature cycle for hardening can best be determined by experiment. Not only the composition of the steel, but the grain size, the annealed structure, the size and shape of the gears, and the equipment available, all have a bearing on the time-temperature combination required to produce the best results.

Cyanide Hardening.—The cyanide hardening equipment is shown in Fig. 153. A conveyor system of the "jack-rabbit" type carries the gears through the complete cycle, consisting of preheat, four cyanide pots, quench, wash, and rinse. The five furnaces are independent units standing on legs and can be moved by means of an electric truck. When a pot burns out or other repairs are needed, the fuel and air lines are disconnected and a spare unit, which is always held in readiness, replaces the one to be repaired.

The four cyanide furnaces are gas-fired with automatic temperature control. The preheat furnace is heated by the products of combustion

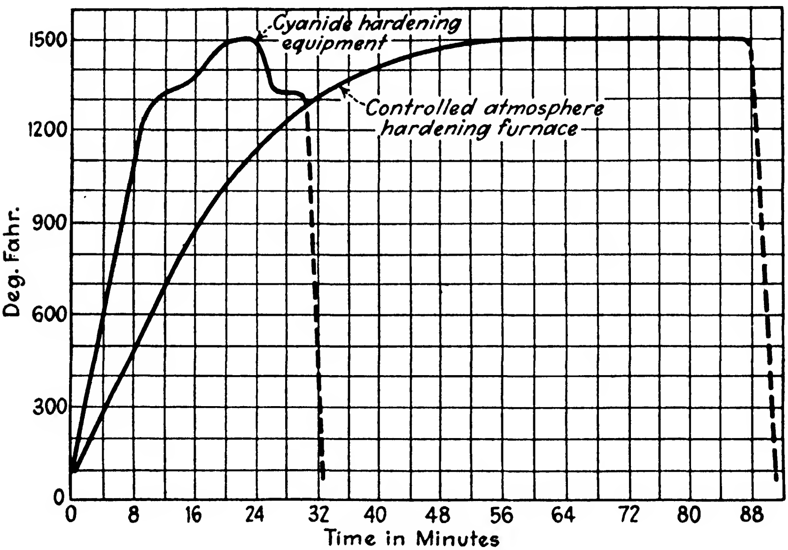


FIG. 154.—Time-temperature cycle for average gear.

from the adjacent furnace by means of a brick flue which is easily broken and repaired when either unit requires moving. The pots are either cast steel or heat-resisting alloy, both materials giving satisfactory service. The best combination is to use alloy for the first two pots, as these receive the hardest firing, and steel for the last two. The pot dimensions are 40 in. long by 25 in. wide by 18 in. deep. An old pot, with the bottom cut out, is used in the preheat furnace, which is the same construction as the other units.

The time cycle is 6 min., which makes a total of 24 min. in the four pots. The temperatures of each unit are shown in Fig. 153 and the probable time-temperature cycle for the average gear is given in Fig. 154. The capacity of this equipment is 800 lb. per hour. The gears are suspended from the conveyor on suitable fixtures spaced so as to provide uniform quenching conditions. The oil flow is automatically shut off for 90 sec. when the gears go into the quench tank, as it was found that distortion could be more effectively controlled with still oil.

The cyanide case obtained averages about 0.002 in. It is purposely held to this low figure owing to the belief that the least thickness of cyanide case necessary to provide adequate wear resistance is the best. While no test data are available either to prove or to disprove this point, it is believed that a heavier cyanide case is not beneficial and may be detrimental to the fatigue resistance of gear teeth with a core hardness of C-50 or above. There is ample evidence that a heavy-cyanide case decreases shock resistance.

The sodium cyanide content of the bath in all four pots is held to from 3 to 6 per cent. The baths are replenished with a mixture containing

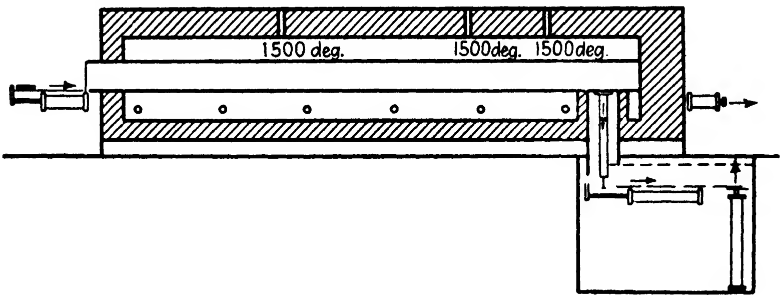


Fig. 155.—General outline of hardening furnace.

25 per cent sodium cyanide with the balance equal parts of soda ash and salt. Small quantities of 96 to 98 sodium cyanide are added as required to maintain the desired composition.

During the past two years, several millions of gears have been hardened by this method. The automatic features, together with the flexibility of the temperature cycle, make possible a degree of control not attainable with the more common designs of cyanide equipment.

Controlled-atmosphere Hardening Furnace.—An improved method for hardening high-carbon gears, which offers certain advantages not attainable with the cyanide process, was developed at the Buick plant. The equipment consists of a continuous gas-fired muffle furnace with automatic temperature regulation and controlled atmosphere. Auxiliary equipment is provided for gas conditioning.

A line drawing of the furnace is shown in Fig. 155. The gears are placed on skeleton trays and pass through the muffle and quenching tank as indicated by the arrows. Conditioned gas is passed through the muffle and quenching tank as indicated by the arrows. Conditioned gas is passed through the muffle to produce the desired atmosphere. The muffle is 22 ft. long from the outer door to the chute, 3 ft. 2 in. wide and 17 in. high. The capacity is 1,200 lb. per hour with a total time of

88 min. The temperature setting for the three zones is 1500°F. With the proper atmosphere, a case averaging 0.006 in. is obtained and the case is of excellent quality.

Gears hardened from this furnace are drawn at the usual temperature of 450°F. and show an increase in Rockwell hardness of about 2 points over the cyanided gears. The toughness is distinctly better than that of the cyanided gears and this toughness is maintained to a remarkable degree as the case depth is increased above the usual 0.006 in. The new method shows less distortion than the cyanide. It is believed that this is due to the slower heating rate and longer total time in the furnace, which are natural characteristics of this type of furnace. In Fig. 154 the time-temperature cycles for both methods of hardening are shown on the same chart.

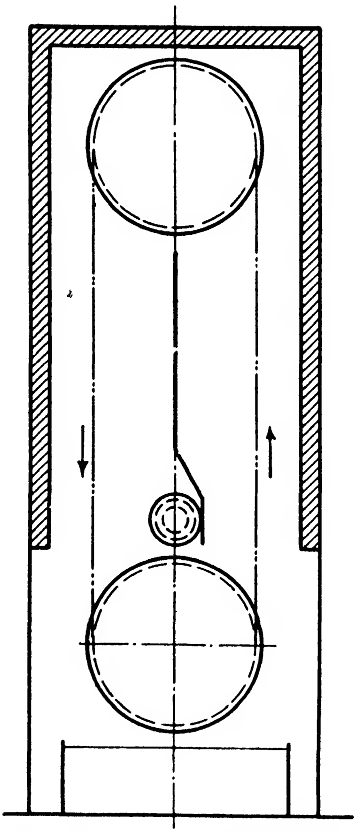


Fig. 156.—Vertical gas-fired drawing furnace.

Drawing.—Transmission gears are usually drawn at temperatures ranging from 300° to 500°F. Although drawing at these low temperatures is usually considered a comparatively simple operation, the importance of accurate time-temperature cycles cannot be disregarded. At one time the oil bath was extensively used for this class of work, but recently the trend has been toward atmospheric furnaces. Electric furnaces with forced draft for circulation of the atmosphere have overcome many of the difficulties previously encountered and are now in general use. Gas-fired continuous furnaces with automatic temperature control have also gone through a period of development and are giving very satisfactory results.

The drawing operation is performed in a vertical gas-fired furnace with automatic temperature control, as shown in Fig. 156. The gears are placed on projecting pins fastened to the conveyor and after passing through the drawing cycle, are cooled in a tank under the furnace, containing soluble oil compound, and returned to the operator for removal. The capacity of this furnace is 1,200 lb. per hour. The drawing temperature is 450°F. and the time in the heating chamber is 2 hr.

Ring-gear Quenching Machine.—Ring gears, of the type used to drive automobile axles, are liable to be considerably distorted in hardening when dipped in an open tank after heating. The Gleason quenching press is designed to hold the gear firmly between dies before it is quenched and to control the amount of cooling oil forced to the different surfaces while the gear is submerged in the tank itself. A pilot enters a segmental bushing placed in the gear as it comes from the furnace and centers it in the machine. These segments round out the bore of the gear before the upper die comes down on the top of the gear to hold it flat.

Both the upper and lower die holders are controlled by air pressure. While in the loading position, the lower holder is held above the loading line so that the hot gear can be readily put in place. Then the upper die comes down and clamps the gear firmly. The pressure on the upper die increases and forces the lower die and the gear, under the oil. As it descends fresh oil is forced through the teeth of the gear at a high rate and cools them quickly. The gear is free either to shrink or to expand but is always held round and flat.

A sectional view of the machine is seen in Fig. 157 and of the die, or gear holder, in Fig. 158. The oil distribution can be changed to secure different rates of quenching for various parts of the gear. When used properly, the gears come out flat and require very little lapping, even for the closest tolerances. The 15-in. machine holds 85 gal. of oil and the 25-in. machine 150 gal. From 15 to 25 cu. ft. of air are used per minute, depending on the machine used. Inlet oil temperature may be varied from 70° to 100°F., but the outlet temperature remains constant at 115°F.

One of the main causes of distortion in hardened parts is the uneven cooling of the different sections during the quenching operation. By holding the work between dies which direct the flow of oil away from thin sections and provide ample circulation about the heavier sections, distortion from this cause may be reduced to a minimum.

The upper and lower dies are made to fit the die holders in accordance with the dimensions shown in Fig. 158. This shows the arrangement for clamping the work and for controlling the flow of oil. For bevel ring gears of the web type the equalizer upper die is recommended. This arrangement takes care of machining variations in web thickness, backing, and face angles

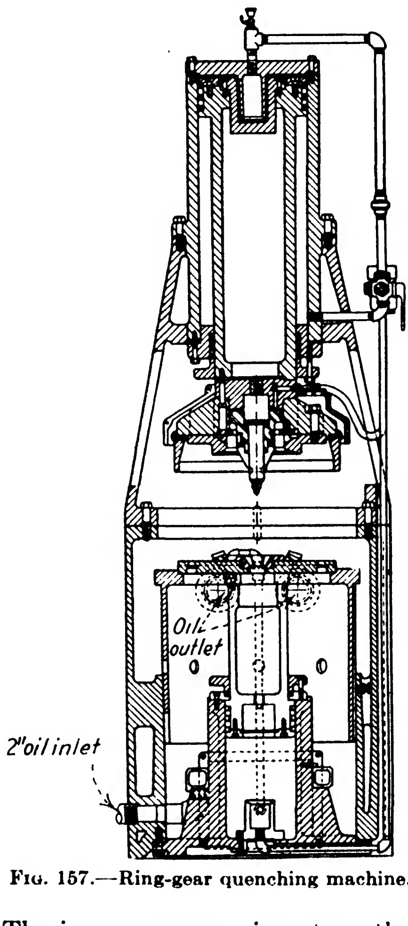


Fig. 157.—Ring-gear quenching machine.

of the gears. The inner pressure ring stops the oil from circulating across the web of the gear. Ample oil circulation is provided through the teeth giving a drastic quench to the parts desired hardest.

The upper die is made of cast iron and is designed to suit the particular part to be quenched, to hold it true and to provide the proper oil circulation. The pressure rings of the equalizer die are made of case-hardened steel. It is important that the die spacing dimension, $3\frac{7}{16}$ in. (+0.250 in. − 0.000) in Fig. 158 be

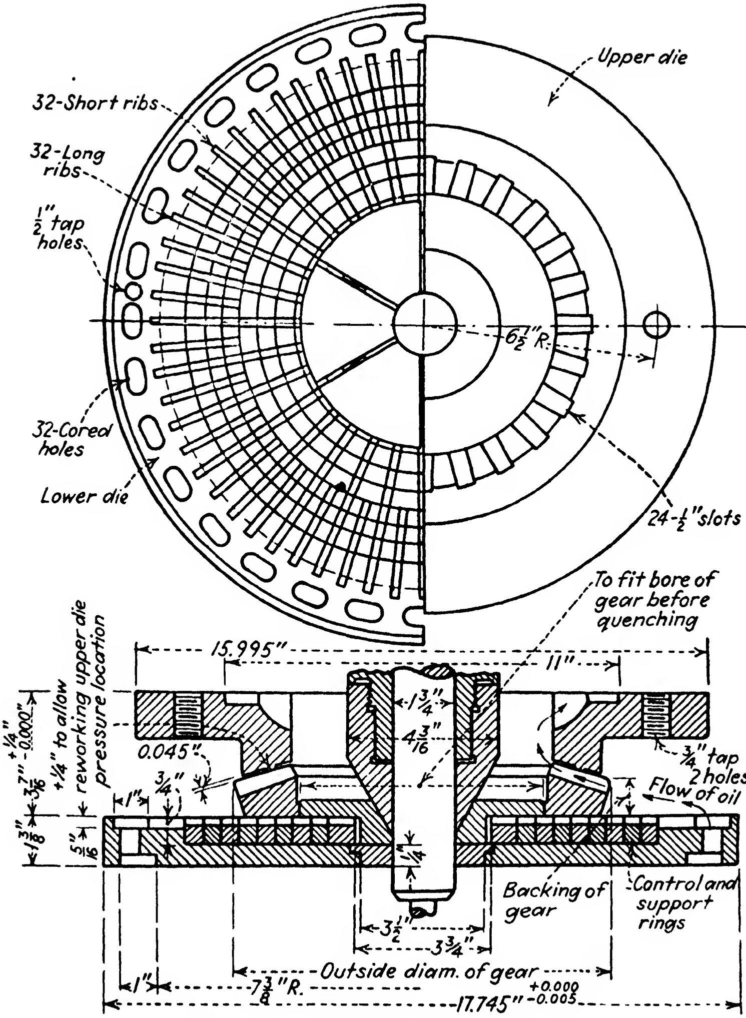


FIG. 158.—How the gear holder is made.

maintained to permit the proper latitude of movement of the tapered expander.

While it is often an advantage to make a special lower die to suit the particular work to be quenched, the universal lower die.

shown in Fig. 158, will satisfactorily take a wide range of work. The slotted control rings provide means of regulating the flow of oil to the under side of the gear. When other than flat back gears are to be quenched, support rings of the correct height are made to replace the corresponding control rings in the universal lower die. The lower die, the control and support rings are made of cast iron.

The split expander bushings are usually made of case-hardened steel. When the quantities are small, cast iron is used. The dimensions for fitting are shown in Fig. 158.

Operation of Quenching Machine.—This machine is designed for oil quenching only. The air-control valve should be operated carefully to prevent the upper die striking too sharply against the gear. This will bruise the face of the teeth owing to the semi-plastic state of a hot gear.

If a gear is found to "dish" in hardening, change the oil circulation by varying the position of control or support rings which can be rotated in the ring slots or inverted to shut off completely the oil. If necessary raise one of the rings by means of shims until the "dish" is eliminated.

If the gear can be "rocked" on a surface plate, either the oil is not being distributed properly or the dies and gear are not seating together correctly. Uneven distribution of oil may be caused by improper alignment of the milled ring slots in the various rings on which the work rests. Or the slots in the rings may not be aligned with the slots in the die. The failure of dies and gear to seat properly is due either to the face angle of the gear not being true with the back or the support ring face not being parallel with lower die.

"Out-of-round" bores may be caused by insufficient air pressure on the expander or too great an air pressure on the upper and lower dies, either of which can be adjusted by means of pressure reducing valves. "Out-of-round" bores are also caused by the gear not bearing evenly on dies owing to faulty machine work or dirt or burrs.

When gears properly carbonized come from the machine soft, one of two things is indicated: either a faulty circulation of oil through dies or too hot oil. The temperature of the oil should not exceed 110°F. for best results and 125°F. should be considered the maximum.

If gear bores open up in hardening, it is generally due to excessive air pressure on expander or striking gear too sharply by improper manipulation of the air-control valve.

If found necessary to reface the lower die and rings, the depth of the counter bore for expanding the bushing hub should be also deepened to maintain the $\frac{3}{4}$-in. dimension.

When refacing the upper die, a shim should be used on top of the die to retain the $3\frac{7}{16}$-in. (+0.250 in. − 0.000) dimension.

Steels vary in their characteristics. Consequently no one fixed procedure will meet all conditions, but a little of experimentation along the lines indicated in these instructions will usually produce the desired results.

When the machine is operating correctly the following points will be observed:

The pressure on the upper gage should not drop more than 10 to 15 lb. during the quenching process.

The lower gage should not register more than 5 lb. over the amount for which it was set when the small cylinder is forced to the top of its stroke.

A slight escape of air, from the relief valve, should be felt as the taper expander comes in contact with the expander bushing.

Heat Treatment for S. A. E. Gear Steels.—Long experience has developed special heat treatments for the various kinds of steels recommended by automotive engineers for use in making gears for the various parts of automobiles. The qualities of these steels are given in Chap. 1, on page 39.

No. 2115. Carburize at from 1650° to 1675°F. Cool in the pots. Reheat to 1600° to 1650°F. Quench in oil. Reheat again to 1350° to 1400°F. and quench in oil.

No. 2315. Carburize at from 1625° to 1675°F. Cool in pots or quench in oil. Reheat to 1550° to 1575°F. Quench in oil. Reheat to 1300° to 1350°F. and quench in oil or water.

No. 2520. Carburize at from 1625° to 1675°F. Cool in pots or quench in oil. Reheat to 1550° to 1575°F. Quench in oil. Reheat 1300° to 1350°F. and quench in oil. Draw to 350°F.

No. 3145. Heat to 1440° to 1460°F. and cool slowly. This should Brinell from 255° to 364°F. After machining heat to 1440° to 1460°F. and quench in oil. Draw as required.

No. 3250. Heat to 1475° to 1500°F. and hold for 2 hr., then cool slowly. Reheat to 1425° to 1450°F. and quench in oil. Draw as required.

No. 5150. Heat to 1575° to 1600°F. and hold it at this temperature for 2 hr. Cool slowly in the furnace. This should Brinell at from 321° to 418°F. After machining heat to 1450° to 1475°F. and quench in oil. Draw as required.

CHAPTER XI

GEAR-INSPECTION METHODS AND EQUIPMENT

With the demand for better and quieter gears has come the development of new equipment for gear inspection. Among the leaders in this line are the Illinois Tool Works from whose instructions we quote quite liberally. These machines include the measuring of the involute profile, of normal pitch and spacing, of the helical lead and the making of charts that show tooth profile, tooth spacing, tooth interference, and eccentricity.

Involute-profile Measuring Machine.—In the production of gearing, it is necessary to be able correctly to inspect the various gear elements. One of the most important of these elements in the involute system of gearing is the involute curve of the gear tooth.

Even a casual study of how gear teeth transmit motion will indicate the obvious necessity of true involute curves. The distortion of involute curves occurs in ordinary production from many causes but commonly from improperly sharpened gear cutting tools and torsional strains set up during the cut.

To determine the nature and amount of error in the involute curve, so that proper corrections can be made, is the function of an involute-profile measuring machine. In checking an involute curve to 1/10,000 in., it is necessary to know that the method of checking, or the equipment used, does not introduce any errors of itself that will confuse the readings.

Furthermore, the instrument must be so arranged that the inspection operation is simply and easily made.

The instrument shown in Fig. 159 was designed by the Illinois Tool Works to meet these requirements. It provides rapid set-up and quick readings for the general run of automobile, truck, and tractor gears up to 10 in. in diameter, and those having shafts up to 13 in. between centers.

Cluster gears and gears integral on shaft can also be reached readily by moving the indicator slide to the desired position. Shaper cutters may also be held on shoulder arbors and readily checked on either side of the tooth by merely swinging the dial indicator from right to left position.

Helical gears, too, may be readily checked by simply turning the contact finger to the approximate slant of the spiral in the gear. In addition a sector graduated in degrees is provided at the side of the instrument in easy reading position. This permits of the involute curve being checked in increments which may be plotted into a graph for record purposes.

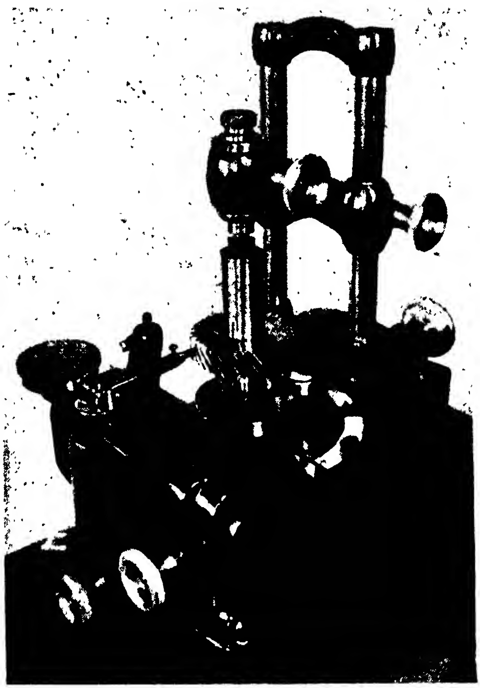


FIG. 159.—Involute-profile measuring machine.

Base circles must be made up for each different gear to be checked. Base circles must have 1.500 in. (+0.0005 – 0.0000) hole and $\frac{7}{8}$-in. face. These base circles are easily calculated by the simple rule which follows. The relationship of the base circle to the pitch circle and pressure angle or line of pressure can be seen in Fig. 160.

The base circles for use in this machine are ground on the outside diameter to plus or minus 0.0005 and surface must be true and parallel with axis. Base circles can be made of cast iron but preferably steel, and need not be hardened.

Calculating the Base Circles.—To find the base circles for spur gears multiply the pitch diameter by the cosine of the pressure angle. The cosines of the pressure angles in common use are given in the small table herewith.

COSINES OF COMMON PRESSURE ANGLES

Degrees	
10	= 0.98481
14½	= 0.96815
17½	= 0.95372
20	= 0.93969
22½	= 0.92388
25	= 0.90631
27½	= 0.88701
30	= 0.86603

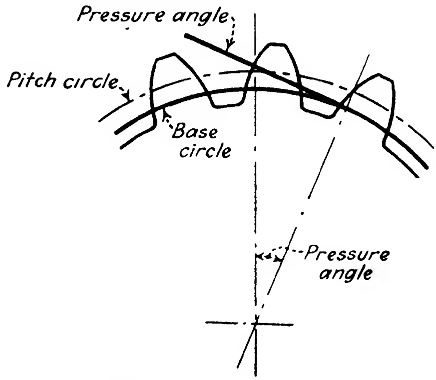


FIG. 160.—Relation of base circle, pitch circle and pressure angle.

Example.—A gear of 10 pitch, 14½-deg. pressure angle with 30 teeth. Dividing the number of teeth by the pitch gives 30 divided by 10, making the pitch diameter 3 in. Multiplying this by the cosine of 14½ deg., which is 0.96815, gives 2.9044 in. as the base circle.

On helical gears, or gears with shallow tooth form, the base circle occasionally is smaller than the root diameter of the gear, as in Fig. 161. In that event care must be taken to avoid bumping the contact finger against the root diameter of the gear.

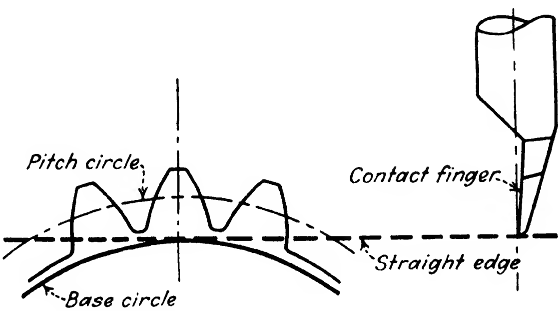


FIG. 161.—Where base circle is smaller than root diameter.

Inasmuch as the contact finger must be set exactly even with the straight edge of the instrument and as this straight edge rolls on the base circle, there will be interference with the contact finger, if the root diameter is larger. Yet, the involute curve can be checked from the bottom fillet to the top of the tooth without interference.

Whenever the contact finger needs to be ground to present a new unworn surface, the grinding should be done on the end of the finger, The finger is so shaped that it will thus present the same kind of contact to the gear tooth at all times.

To reset the finger even with the straight edge, place a parallel on straight edge and swing slowly by the contact finger until end of finger

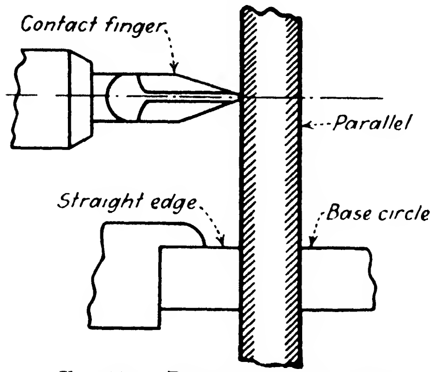


FIG. 162.—Resetting contact finger.

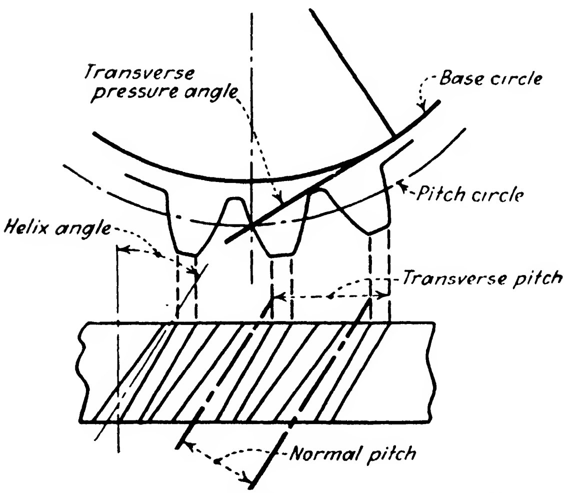


FIG. 163.—Finding base circle of helical gears.

just touches, as in Fig. 162. The finger has an adjustment screw at the back end to facilitate adjustment.

In checking helical gears, it is necessary to find the transverse (plane of rotation) pressure angle of the gear to obtain the correct base circle. All involutes are checked in the plane of rotation or in the transverse plane as it is commonly called.

To Find the Base Circle of Helical Gears.—Tangent of normal pressure angle times secant of helix angle equals tangent of transverse pressure angle.

Cosine of transverse pressure angle times pitch diameter equals base-circle diameter (see Fig. 163).

To Find Transverse Pitch.—Normal circular pitch times secant of helix angle equals transverse circular pitch.

Number of teeth times transverse pitch equals pitch circumference.

Pitch circumference divided by 3.1416 equals pitch diameter.

Example.—5 pitch 20-deg. normal pressure angle, 22 teeth, 45-deg. helix angle

Tangent 20-deg. pressure angle 0.36397 times secant, 45-deg. helix angle 1.4142 equals 0.51473 which is the tangent of 27 deg. 14 min. transverse pressure angle.

Cosine 27 deg. 14 min. pressure angle 0.88915 times pitch diameter 6.2225 equals 5.5327 base-circle diameter.

Usually information on helical gears is given on the normal plane. The above simple rules will permit transposition of normal dimensions to transverse dimensions or plane of rotation which is the plane in which all involute curves unwind.

Imperfect machining and heat-treated distortions in gear teeth can readily be discerned with this involute check and steps taken to correct the errors.

Normal-pitch- and Space-measuring Machine.—The most vital of all gear elements is normal pitch and yet this is one gear element that has received almost no recognition in the gear cutting shop. In gear engineering and design it is, of course, recognized and employed, but in the production of gears the normal pitch is seldom if ever inspected or measured.

It can be stated definitely that normal pitch is the index to gear performance and the yardstick of quietness in gears.

The reason for this is plain when it is understood that in the involute system the normal pitch is the distance between adjacent gear teeth on the line of pressure upon which the mating action occurs. Therefore, gears that measure alike on the normal pitch regardless of their diameters or center distance will mate quietly.

It is not enough to know that gears are alike in circular pitch, but they must be alike in normal pitch to mate quietly and efficiently.

The Illinois normal-pitch and space-measuring machine (Fig. 164) is designed to measure rapidly normal pitch in the rotational plane of gears and to check spacing in various types of gears in planes other than rotational. The operation is semi-automatic and the average automobile, truck, or tractor gears of 30 teeth may easily be checked on all teeth in 1 min.

The machine may be used on its own bed or it may be clamped to any machine that has a pair of centers. It may also be attached to

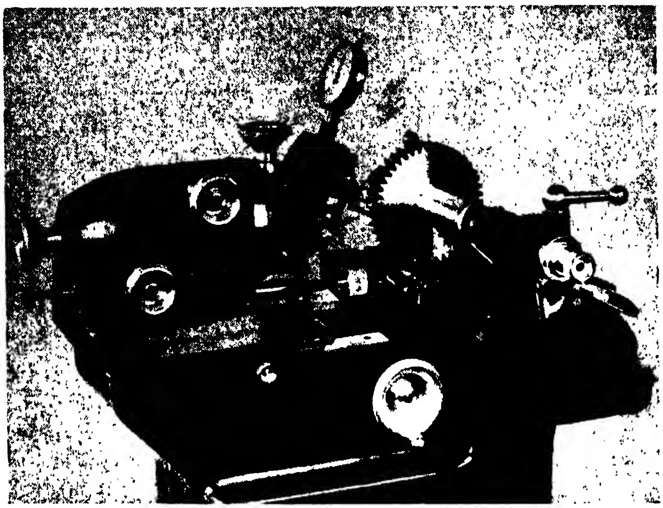


FIG. 164.—Normal-pitch and space-measuring machine.

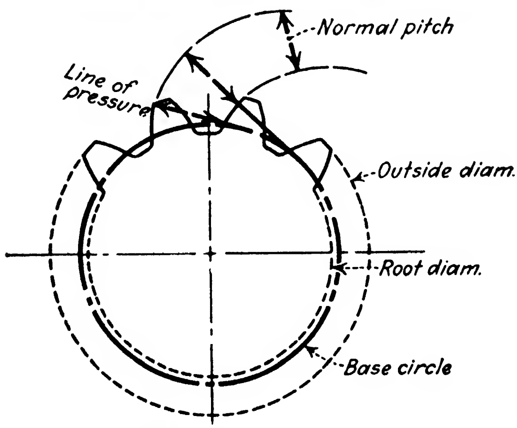


FIG. 165.—Normal-pitch diagram.

worm- and bevel-gear cutting machines and the gears checked on their own work arbors.

In Fig. 165 is shown a diagram of normal pitch which equals the base circle times 3.1416 ÷ number of teeth.

One of the features of involute curves is that they run parallel to each other as they extend out from the base circle. Therefore, any line tangent to the base circle that crosses the involutes will be the same length between the involutes at any point of crossing. This distance is called the normal pitch.

The other feature that makes this normal-pitch distance so important is that the mating gear teeth have their contact on this tangent line or line of pressure.

Therefore, mating gear teeth that do not have alike normal pitches will introduce violent disturbances in the rotating mass.

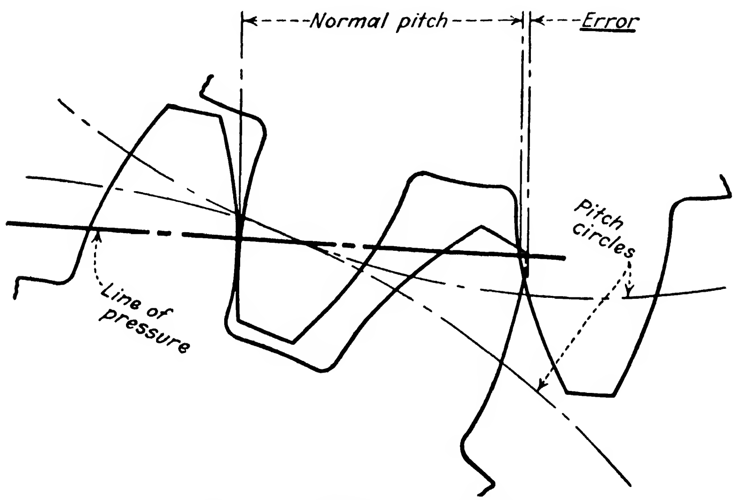


Fig. 166.—Diagram of mating teeth.

Figure 166 indicates the manner in which gear teeth mate. If one 10-pitch 30-tooth, 14½-deg. pressure-angle gear had a normal pitch of 0.3041 and the mating gear a normal pitch of 0.3045 there would be considerable clash and noise.

It can also be noted from this diagram that the gear teeth cannot contact simultaneously at the pitch lines or on the circular pitch.

Gears are often checked for circular-pitch spacing, which may be alike in both mating gears. In gears that are hobbed the circular pitch is almost always correct because the same hob tooth that cuts at the pitch line on one tooth also cuts on the pitch line on all the teeth in the gear and the indexing of the hobbing machine is continuous.

In machines using intermittent indexing, or gear cutting tools which do not have the same cutting tooth repeat in the same spot on successive gear teeth, the circular pitch is more apt to be off.

Thus, in purely mathematical gears normal pitch and circular pitch are complementary to each other. In actual production it is common to get gears with accurate circular pitch or tooth spacing and still have an error in normal pitch where the gear teeth mate.

Helical Gears.—Helical gears have two normal pitches.

1. The normal pitch of the involute curve lies in the rotational plane of the gear or as commonly stated the transverse plane. This normal pitch is commonly known as base pitch.

2. The normal circular pitch of the helical teeth lies in the plane that is normal or right angle to the helix angle of the gear. This normal

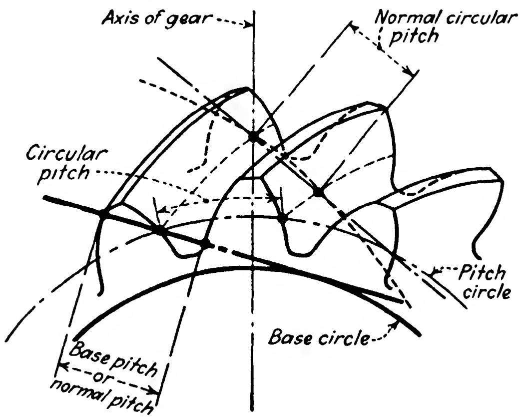


Fig. 167.—Showing the three pitches common to helical gears.

circular pitch is the one that is always used in reference to helical gears, worms, etc. These two normal pitches must not be confused.

A study of Fig. 167 will show the three pitches common to helical gears. To check the normal pitch of the involute of a helical gear it is necessary to have the contact fingers of the measuring machine stand in the rotational or transverse plane which is perpendicular to the axis of the gear. To find the pressure angle at which the head is set it is necessary to transpose from the nominal pressure angle of the gear which is usually given normal to the pitch. Thus,

Tangent of normal pressure angle $\times$ secant of helix angle $=$ tangent of transverse pressure angle.

In order that the contact fingers may be set tangent to the base circle so that they contact on the line of pressure, a base circle is used. The base circle is placed between centers and the outer surface of the contact fingers are brought up until they touch or tangent the base circle. Then the inward stop of the head is adjusted to keep them in this position. The fingers may then be moved upon their own slide until they are in position to contact two adjacent teeth. Any sliding of the fingers up or down takes place in the tangent line.

To find the base circle to use for a helical gear:

Cosine of transverse pressure angle × pitch diameter = base-circle diameter.

After the base circle is obtained, the contact fingers may be set up as described in the foregoing.

It is not necesssary, however, to set the head to the pressure angle of the transverse plane. The head may be set at any pressure angle within its scope and the contact fingers would still measure on the line of

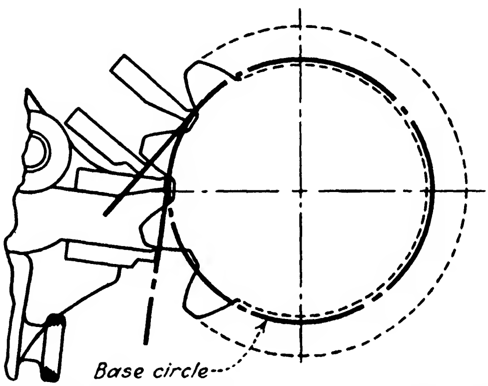


FIG. 168.—How contact fingers may be placed on the tangent line.

pressure as long as contact fingers are first set tangent to the base circle. Figure 168 indicates how contact fingers may be placed on the tangent line.

A micrometer may be used for measuring the distance between the contact fingers. After the fingers have been set in position and the dial indicator registers zero, the fingers are backed away from the gear teeth and a micrometer reading is made of the overall distance of the two fingers. Then the lower finger is measured and this amount is subtracted from the overall measurement, thus giving the measurement of the normal pitch.

To check the normal circular pitch of the helical teeth on the gear, the head is turned to the helix angle of the gear. The contact fingers are distributed equally on either side of the axis and in this position will check the chordal pitch of the helical teeth. When the head is set in this position, the check being made is really a spacing check and compares the spacing of successive teeth.

Inspecting Worms.—The spacing of worm threads, particularly on multiple-threaded worms, is often in error and is the source of much trouble when worms run at high speed; even on slow-moving worms poor spacing gives an irregular tooth load. Some worm threads are made to have straight profiles in the axial plane such as would be produced with a lathe tool ground flat on top. Other worms are made to have straight profiles in the plane normal to the thread helix such as would be produced with a thread-milling cutter.

The worm thread when produced with a thread-milling cutter does not actually have a straight profile normal to the helix angle because while the profile of the cutter is straight, the cutter operates correctly only on the pitch-line helix. The helix angles at the root diameter and

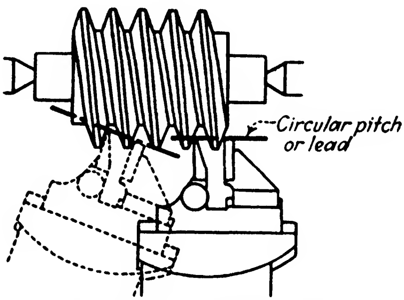


Fig. 169.—Method of checking threads on worms.

outside diameter of the worm are different and so contact the cutter profile in planes normal to themselves. This condition in milling actually produces a very slight convex surface on the thread which is usually acceptable.

Another type of worm thread is made to have straight-line elements tangent to its base cylinder. There are several other variations of thread profiles used, but in all cases where multiple threads are employed the problem of correct spacing is present.

The Illinois normal-pitch and space-measuring machine can be adapted for this sort of checking. Place the worm between the centers and bring the contact fingers into the plane parallel with the axis of the worm. Also have the ends of the fingers parallel to axis. Bring the contact fingers into contact near the pitch line of the worm thread and set the dial indicator to zero. Then rotate the worm to bring another pair of threads into contact. Variations in indicator readings on successive pairs of threads will show the amount of error in axial pitch of the threads. It is in the axial plane that the worm threads contact with the worm-gear teeth and therefore, must be correct. Figure 169 shows the conventional method of checking threads on worms.

Bevel Gears.—While it has not been common practice to check bevel gears for tooth spacing, there are nevertheless many bevel-gear jobs which would benefit from an inspection of tooth spacing and this same machine can be used for this work.

Bevel-gear noise is usually attributed to the improper position of pitch cones and to incorrect axis of gears. However, if both these conditions are correct, it is still obvious that incorrect spacing will cause trouble.

In checking bevel gears the head may be set at right angles to any pitch cone angle and the contact fingers adjusted to read either nominal-normal-pitch or circular-pitch tooth spacing. When nominal normal pitch is measured, the contact fingers should be at the large end of the pitch cone, as in Fig. 170.

Checks may be made at the small end of the pitch cone or any place in between, but these checks will only register tooth spacing and not nominal normal pitch. This is true for either straight- or spiral-bevel gears.

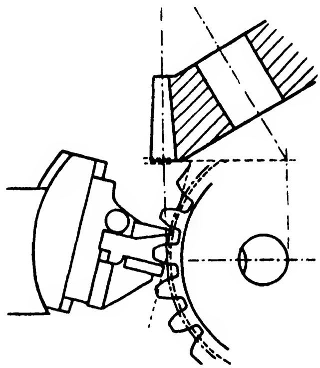


FIG. 170.—Setting contact fingers at large end of bevel teeth.

A protractor may be used against the lower flange on the head and in this way the setting angle in relation to the gear may be obtained.

Inspecting Helical Gears.—Because the helix angle of a gear tooth changes with the distance from the root diameter to the outside, engineers have decided that it is better to check the lead of the helix rather than its angle. For this reason inspection of this factor of helical gears is now done by lead checking machines, as shown in Fig. 171.

Helical-lead Checking Machine.—It is generally accepted that helical gears produce less noise than spur gears and that this is almost entirely due to the pitch helix; also that other gear elements have only a remote influence.

It may be stated that a spur gear transmits motion uniformly from one shaft to the other by means of its tooth curve while the helical gear transmits the motion principally through the pitch helix, the other parts of the tooth curve being of secondary importance. This means, in the first case, that uniformity of motion depends on the accuracy of the tooth curve, while in the helical gear it is the accuracy of the pitch helix that counts.

However, it must be understood that the tooth curve in helical gears must be accurate enough to avoid interference. It is also generally assumed that it is much easier to produce a helix accurately than an involute curve according to present machining practice. A helix is simply the product of rotation and translation and it is thought that almost any machine tool will produce it accurately.

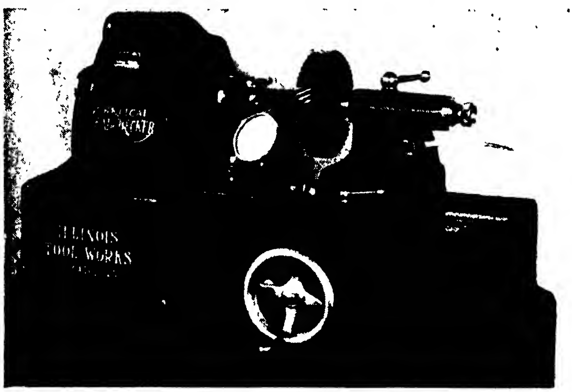


FIG. 171.—Helical lead checking machine.

However, experience has shown that this last assumption is not always correct and that, while production machinery may be set up theoretically correct, the helices on the gear are often imperfect. Difference in helices between mating gears on parallel shafts is to be avoided if full efficiency and quiet operation are to be obtained.

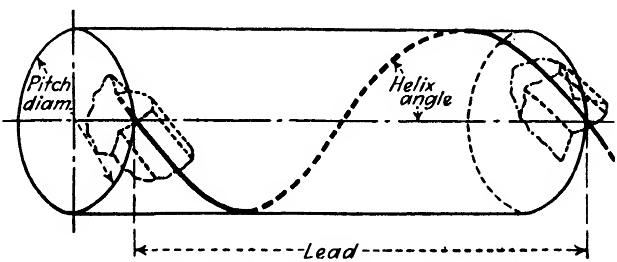


FIG. 172.—Method of obtaining lead of helix.

The Illinois Tool Works make a machine for checking the helical lead, which will take gears up to 12 in. diameter and 15 in. between centers. It will check helix angles from zero to 90 deg. All work is held between centers and when thin gears, such as master gears or burnishing gears, are checked, they should be held on shoulder arbors to avoid side wobble. It is shown in Fig. 171.

The diagram, Fig. 172, illustrates the method of obtaining lead of helical work. The formula is

Lead-pitch diameter × 3.1416 × cotangent of helix angle.

The machine has one horizontal slide and one vertical slide. These slides are connected by means of a sine bar which may be set to any desired angle. Movement of one slide immediately transmits the motion to the other slide. Both the horizontal and vertical slides are run on V-grooved ball tracks and these are adjusted tight enough to prevent any backlash and yet permit a floating movement of the slides. Full directions for use accompany the machine.

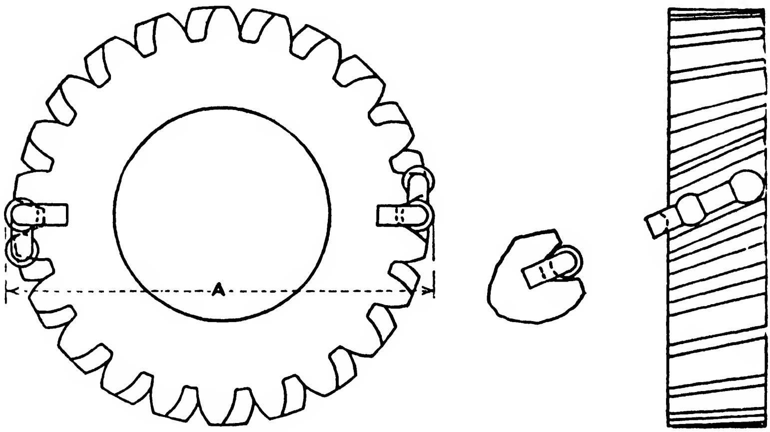


FIG. 173.—Using ball pins in measuring helical gears.

Helical-gear-tooth Measurement.[1]—Sizing helical-gear teeth by measuring across ball pins (*A*, Fig. 173) has many advantages, being comparable to sizing spur gears by measuring across pins. In many cases, the measurement can be determined with comparative ease. Some advantages of the method are: It does not require the use of special measuring instruments other than the ball pins; the gears can be sized without removing from the cutting machine; the measurement is independent of the outside diameter; the tooth thickness of a given gear can be identified without reference to another gear. This last is an important consideration where the gears are of opposite hand. Here the ball-pin method has proved invaluable for gears where the helix angle does not exceed 45 deg.

[1] Sam Trimbath, gear consultant.

Measuring the thickness of helical-gear teeth by means of balls has been restricted because the calculations are somewhat complex. However, it will be found that the problem can be reduced nearly to the simplicity of determining spur-gear measurements

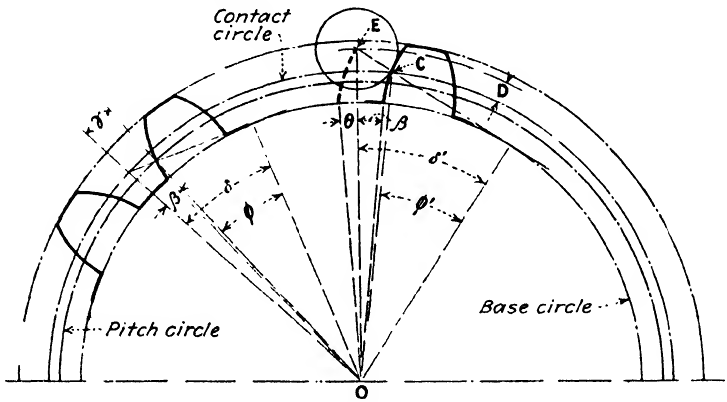


FIG. 174.—Diagram of gear with ball measurement.

across pins through the use of the formula for cutter selection when helical gears are cut with rotary cutters. This formula is:

$$\frac{N}{\cos^3 \beta} = T$$

(Brown and Sharpe, "Formulas in Gearing")
or

$$\frac{\text{Number of teeth}}{\text{Cosine of helix angle cubed}} = \text{number of teeth in a spur gear to}$$
which the helical tooth shape conforms normal to the pitch helix.

The distance from pitch line to ball center (D, Fig. 174) is then computed by assuming a spur gear having this number of teeth.

For even teeth, add to the pitch diameter of the helical gear twice the distance from the pitch line to the center of the ball, plus the ball diameter for the measurement across balls. For odd teeth, add to the pitch radius the distance from pitch line to ball center. Multiply this by the cosine of 90 deg. divided by the number of teeth, then by 2 and add the ball diameter for the measurement across balls.

Work of this kind is much simplified through the use of forms, a sample of which is shown in Fig. 175. Their use tends to avoid mistakes and the necessity for assimilating a lot of technical data, a matter that may consume considerable time when work of this kind is done at long intervals. When any considerable number of calculations are to be made, their use is almost indispensable. The necessary formulas for making the computation

WORK SHEET

HELICAL GEAR MEASUREMENT ACROSS BALLS

Data required, No. of teeth (N) 15 Pitch radius (R) 1.0714

Normal pitch (P) 8 Helix $\angle$(b) 28°57′ Ball diam. 0.220

Normal pressure $\angle$ (ϕ) 14°30′

$$\frac{N}{\cos\angle b^3}\quad \frac{15}{0.875^3} = \frac{15}{0.6699} = 22.391 = N'$$

$$\frac{N'/2}{P}\quad \frac{11.1955}{8} = 1.3994 = \text{Pitch radius of } N'$$

$$\underset{1.3994}{\text{Pitch radius}} \times \underset{0.96815}{\cos\phi} = 1.3548 = \text{Base radius of } N'$$

$$\frac{90°}{N'\ 22.391} = 4.020° = 4°\ 1' = \gamma$$

$$\underset{4°\ 1'}{\gamma} + \underset{14°\ 30'}{\phi} = 18°\ 31' = \delta = 0.32318 \text{ in radians}$$

$$\underset{0.32318}{\delta} - \underset{0.25862}{\operatorname{Tan}\phi} = 0.06456 = \beta \text{ in radians}$$

$$\frac{\text{Ball radius } 0.110}{\text{Base radius } 1.3548} = 0.08119 = \theta + \beta$$

$$\underset{0.08119}{\theta+\beta} - \underset{0.06456}{\beta} = 0.01663 = \theta = \operatorname{Tan}\delta' - \delta' \quad 20°43' = \delta'$$

$$\frac{\text{Base radius } 1.3548}{\cos\delta'\ 0.93534} = 1.4465 = OE, \text{ center } N' \text{ to ball center}$$

$$\underset{1.4485}{OE} - \underset{1.3994}{\text{Pitch radius }(N')} = 0.0491 = D, \text{ pitch line to ball center}$$

EVEN TEETH	ODD TEETH
= 2D	0.0491 = D
= Ball diameter	1.0714 = Pitch radius (R)
= Pitch diameter (2R)	1.1205 = R + D
= Distance across balls	$\underset{1.1205}{R+D} \times \underset{0.99452}{\cos\frac{90°}{N}} = 1.1144$
	2
	2.229
	Ball diameter = 0.220
	Distance across balls = 2.449

FIG. 175.—Form for work sheet of gear problems.

are included in the work sheet. With reference to Fig. 174, they will be readily understood. Angle θ is the difference between the tangent of δ' and δ' expressed in radians; angle δ' is found by reference to a table of involute functions (Buckingham's book, "Spur Gears").

An example has been worked out for a gear of standard tooth thickness or $3.1416/2P$ and filled in the form shown in Fig. 175. For gears under standard thickness, β increases. To determine the measurement divide the amount under standard tooth thickness by the pitch diameter of N', and add this to the value of β in the calculation. For example, to reduce the tooth thickness 0.005, $0.005/2.7988 = 0.00179$ = increase in β. However, any necessary reduction in tooth thickness from standard must be gained from either experience or a consideration of the other factors involved and can be expressed just as readily in terms of a reduction in measurement across balls as in terms of tooth thickness. For this reason, computation of the standard measurement is usually sufficient.

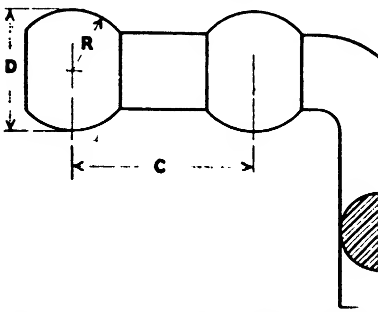


Fig. 176.—Ball pin with angle extension.

From the above it can be seen that the computation is similar to determining the measurement of spur gears across pins. In fact, the same form can be used for both spur and helical gears by using the actual number of teeth N instead of N' when computing spur-gear measurements.

It will be noted that the pitch radius of the assumed gear N' is larger than the actual pitch radius R. For this reason, great care must be taken in determining angle δ' for helical gears, especially when the pitch is coarse and the helix angle large, as the radius of N' may be such that one minute's difference in the angle would affect the computation appreciably. In other words, the use of the pin method in spur gearing is more or less restricted to gears of such size as can be conveniently measured with micrometers, and the calculations are direct. In helicals, the tooth shape normal to the helix may conform to spur teeth with many times the number of teeth in the work, and the diameter of assumed gear N' may be much larger than the work itself.

Another restriction to the use of balls for measuring helical gears is the problem of holding and locating the balls or in obtaining balls of a given diameter. One solution that has worked out well in practice has been the substitution of ball pins as shown in Fig 176. Diameter D should be held close to the ball size used in computing the measurement, while the diameter in the opposite direction can vary 0.001 to 0.002 minus without affecting the measurement appreciably. The distance between the ball sections of the pin C should be no greater than necessary for convenient handling. The object of having two ball sections is to provide a means for retaining the pins in the teeth while adjusting the micrometers to the work. Still another method would be an arrangement as shown in Fig. 177, where the ball pin is attached to a small plate, allowing the pin to accommodate itself to various angles.

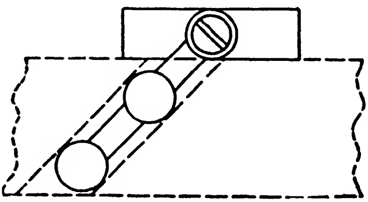


FIG. 177.—Using the ball pin with a small plate.

As the helix angle increases and the number of teeth decreases, the problem of accurate location of the balls becomes more acute, and it is probably advisable to restrict the method to gears of 45 deg. or less and to numbers of teeth 10 or greater.

The sizes in Table 52 are standard; no backlash is allowed; tooth and tooth space are taken as equal. For pitches finer than 1 diametral pitch, divide the figure opposite the number of teeth, in the column corresponding to the pressure angle, by the pitch. To obtain the pin diameter, divide 1.750 in. by the pitch.

Example.—20 teeth, 10 pitch, 25-deg. pressure angle:

$$\frac{1.750}{10} = 0.175 = \text{size of pins}$$

$$\frac{22.468}{10} = 2.2468 = \text{size over pins}$$

The size for gears having numbers of teeth not shown, such as between 42 and 52, can be found readily by interpolation. When interpolating, care should be taken to use the upper half of the table if the teeth are even in number, and the lower half if the teeth are odd in number. Thus, the size of a 47-tooth gear would be found between 43 and 53, not between 42 and 52.

For numbers of teeth greater than 142 and 143, a close approximation can be found readily by adding two for every two teeth greater, to the figures for 142 if even, and 143 if odd. For pressure angles falling between those given in the table, a close

Table 52.—Pin Measurements for Gear-tooth Inspection
1 Diametral pitch—Size of gear teeth over pins 1.750 in. in diameter

Teeth	14½ deg.	17½ deg.	20 deg.	22½ deg.	25 deg.	27½ deg.	30 deg.
	Even teeth						
8	10.389	10.389	10.394	10.403	10.416	10.432	10.451
10	12.415	12.411	12.413	12.420	12.431	12.445	12.462
12	14.436	14.429	14.428	14.433	14.442	14.455	14.471
14	16.454	16.442	16.440	16.443	16.450	16.462	16.478
16	18.468	18.454	18.450	18.451	18.457	18.468	18.483
18	20.482	20.464	20.458	20.457	20.463	20.473	20.487
20	22.493	22.473	22.465	22.463	22.468	22.478	22.491
22	24.502	24.479	24.470	24.468	24.472	24.481	24.494
24	26.512	26.487	26.475	26.472	26.475	26.484	26.497
26	28.520	28.492	28.481	28.476	28.479	28.487	28.499
28	30.526	30.497	30.484	30.479	30.482	30.489	30.501
30	32.532	32.500	32.487	32.482	32.484	32.491	32.502
32	34.536	34.505	34.491	34.484	34.486	34.492	34.504
34	36.542	36.509	36.494	36.487	36.488	36.493	36.506
36	38.547	38.512	38.496	38.489	38.489	38.495	38.506
38	40.553	40.516	40.499	40.490	40.491	40.496	40.507
40	42.557	42.519	42.502	42.492	42.492	42.498	42.508
42	44.561	44.523	44.503	44.495	44.493	44.499	44.509
52	54.576	54.533	54.510	54.500	54.498	54.502	54.513
72	74.597	74.546	74.521	74.507	74.504	74.510	74.518
102	104.616	104.557	104.528	104.513	104.511	104.513	104.521
142	144.625	144.562	144.535	144.514	144.514	144.514	144.523
	Odd teeth						
9	11.256	11.254	11.258	11.265	11.277	11.291	11.309
11	13.307	13.301	13.302	13.308	13.318	13.331	13.348
13	15.346	15.336	15.335	15.338	15.347	15.359	15.375
15	17.376	17.362	17.359	17.361	17.368	17.379	17.395
17	19.400	19.384	19.378	19.379	19.384	19.395	19.409
19	21.420	21.402	21.394	21.393	21.398	21.408	21.422
21	23.437	23.415	23.406	23.405	23.409	23.418	23.432
23	25.452	25.428	25.417	25.415	25.419	25.427	25.440
25	27.464	27.438	27.427	27.423	27.427	27.434	27.447
27	29.476	20.448	29.435	29.431	29.433	29.441	29.453
29	31.485	31.456	31.442	31.437	31.439	31.446	31.458
31	33.495	33.463	33.449	33.443	33.444	33.451	33.462
33	35.503	35.469	35.454	35.448	35.448	35.455	35.466
35	37.511	37.476	37.460	37.453	37.453	37.459	37.470
37	39.516	39.480	39.463	39.455	39.455	39.461	39.472
39	41.523	41.485	41.467	41.459	41.458	41.465	41.475
41	43.529	43.489	43.472	43.463	43.463	43.467	43.478
43	45.534	45.493	45.475	45.465	45.465	45.470	45.481
53	55.554	55.508	55.486	55.477	55.474	55.479	55.490
73	75.580	75.530	75.505	75.490	75.487	75.492	75.500
103	105.600	105.544	105.513	105.499	105.496	105.500	105.507
143	145.619	145.555	145.526	145.506	145.506	145.507	145.518

The data in Table 52 are by Sam Trimbath, an authority on gears and gearing.

approximation can be obtained by interpolation. The greatest difference is between $27\frac{1}{2}$ and 30 deg. for an 8-tooth gear, and this is only 0.019 in. If the figure for $27\frac{1}{2}$ deg. had been used on a 30-deg. pressure angle for a 10-pitch gear, the error would be 0.0019. Assuming a pressure angle of 29 deg. and adding $\frac{3}{5}$ of 0.019 to the figure for $27\frac{1}{2}$ deg., the result is 10.442. Actual calculation gives 10.443, an error of only 0.001 for a 1-diametral-pitch gear or 0.0001 for a 10-diametral-pitch gear. When it is considered that these errors are in diameter and that the errors in tooth thickness are approximately half again as large, these figures will be seen to have sufficient accuracy.

The Red Liner Inspection Machine.—The Red Liner inspection machine derives its name from the form in which the errors of the gears being inspected are recorded. It is built by the Fellows Gear Shaper Company and is based on the principle that errors in gear-tooth elements cause either changes in velocity or in the center distance of the gears. Such changes are recorded automatically on this machine, on a charting paper which multiplies the errors 200 times. All errors such as eccentricity, tooth spacing, and tooth shape are so recorded as to be easily distinguishable and the amount of each error can be determined, each type of error causing a characteristic deviation from a straight line. In addition both sides of the teeth are inspected at the same setting and in one revolution. Figure 178 shows the machine set for inspecting internal gears.

Automatic Gear Inspection.—This machine is the result of years of careful study and analysis of gear cutting and gear inspection, with the aim of developing a method of inspection which would not only detect the individual errors in tooth dimensions, but also indicate them in combination.

A close study of the action of gear teeth will reveal the fact that there is always more than one tooth affecting the operation of a mating pair of gears when they are in mesh, so that a careful checking of each individual tooth is inadequate—it does not tell what effect a certain error in another tooth will have on the one being measured.

A combined inspection of all errors in tooth elements more nearly meets the demands of manufacturing requirements, because it approaches more closely the actual conditions obtained when the gears are in operation.

When gears are operating together, any error which brings two teeth into action before they reach the correct position for initial contact results in an excessive load on the teeth. This excessive load, which tends to accelerate the driven member, has a radial component that separates the axes of the two gears. If, therefore, the gears to be inspected are brought into positive contact without backlash, the tendency to change the angular velocity is restricted and the tendency to lengthen the center distance is

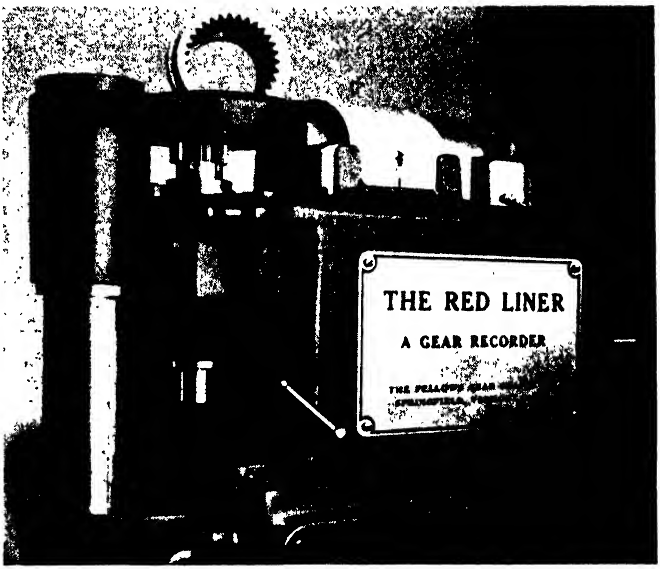


FIG. 178.—Machine set for inspecting internal gears.

increased. The advantages of this method of inspection have been recognized for some time, but obtaining the "story" in a permanent form has been a matter which was not easily accomplished.

Gears Tested under Operating Conditions.—Summarizing the operation of the Red Liner, it should be emphasized that this machine tests a gear under conditions comparable to those of actual operation by bringing the gear to be tested into actual contact with a master gear of known accuracy.

It should be explained that two cut gears should not be tested together, unless they can be operated at standard center distance. It is therefore necessary that the teeth be cut to standard tooth parts—without provision for backlash—so that they can be brought into positive contact without decreasing the center distance. A decrease in the center distance may cause interference which would not exist under normal operating conditions. The master gear method of testing is preferable, and the thickness of the teeth of the master gear should be made greater than the

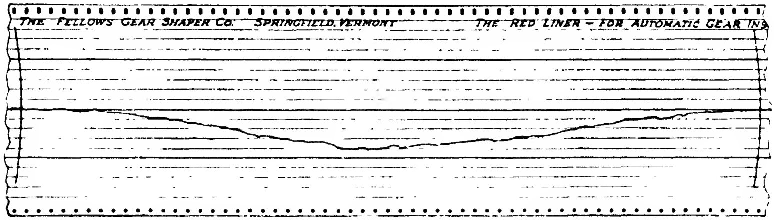


FIG. 179.—Gear chart showing eccentricity of 0.004 in.

"standard" dimension. This increase in thickness should be equal to the amount of reduction in thickness of the teeth of the gear being tested.

Reading the Red Liner Charts.—Two typical Red Liner charts are shown in Figs. 179 and 180. The first shows a gear with an eccentricity of 0.004 in., each line on the chart being 0.001 in. Three charts are combined in Fig. 180 to show the effect of heat treatment and lapping. The finishing cut left the gear correct within 0.001 in. but heat treatment practically doubled the

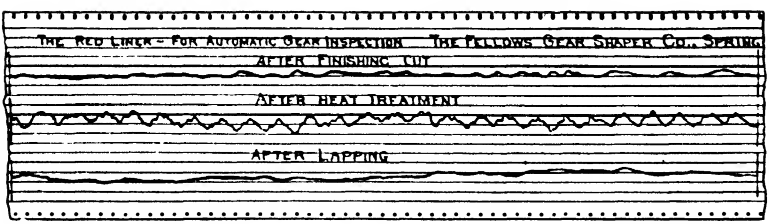


FIG. 180.—Combined chart showing distortion due to heat treating and result of lapping in removing errors.

errors. Lapping, however, restored the gear to its original accuracy and also smoothed out some of the most pronounced "humps" left by the cutting tool.

Details of three applications of the Red Liner to different gears are shown in Figs. 181, 182, and 183. Figure 181 shows how the internal gears seen in Fig. 178 are held for inspection. This gives a good idea of the close center distances that can be secured with internal gears when necessary. Both cluster- and shank-type gears are shown in Fig. 182, while in Fig. 183 are the details for holding a face, or crown, gear for inspection. The machine is easily operated and there are many cases where the permanent record is of considerable value.

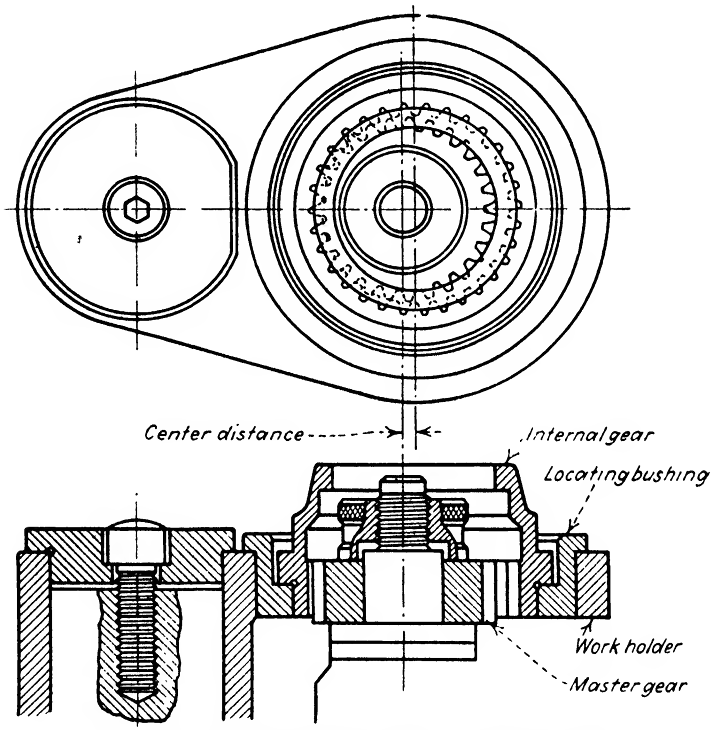


FIG. 181.—Holding an internal gear on the Red Liner.

The Illinois Tool Works also make a gear charting machine which gives a record of the characteristics of the gears being tested. The charts enable the inspector to detect the cause of error, whether it be uneven tooth spacing, eccentricity, interference, or other irregularities. Such charts make it possible to study gear action intelligently and also to keep records for future use.

Inspecting Gears with Theoretically Correct Rack.—The Gear Grinding Company, Ltd., of Birmingham, England, has devel-

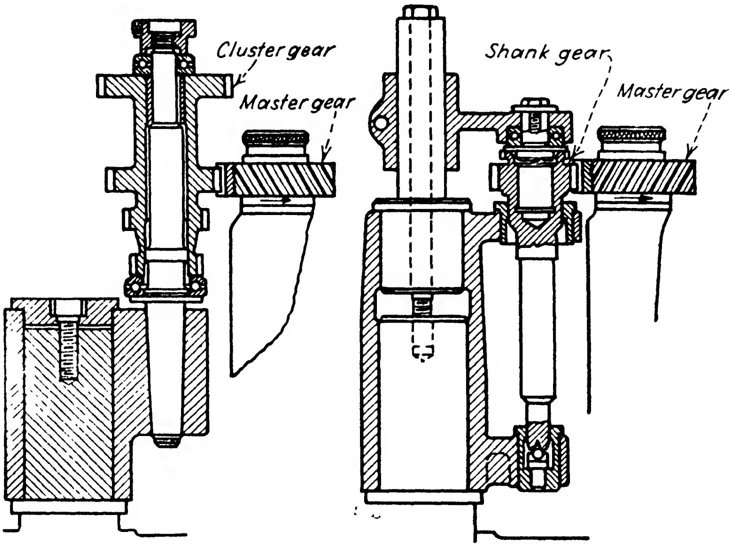


FIG. 182.—How cluster gears are held.

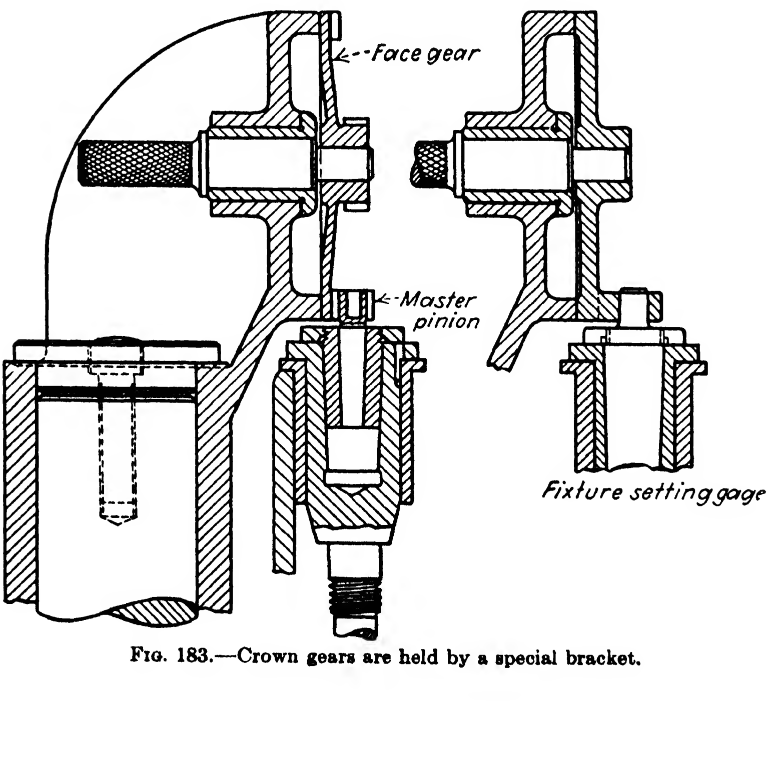


FIG. 183.—Crown gears are held by a special bracket.

oped an interesting method of inspecting gears by rolling them with a rack that is theoretically correct, as in Fig. 184. The same base is used as for rolling two gears together in the usual manner. But the rack used for inspection is mounted across the bed at right angles and can be moved crosswise so as to rotate the gear.

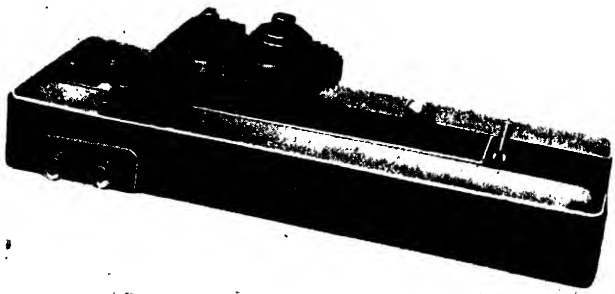


Fig. 184.—Using a rack to inspect gears.

The rack itself is shown in Fig. 185 and, as can be seen, is built up with individual teeth screwed to the small cross slide. The tooth in the center is held by a screw with an extended head that forms the handle by which the slide is moved. This special rack is very carefully made, each rack tooth being ground and lapped, and compared with gages corresponding to the circular

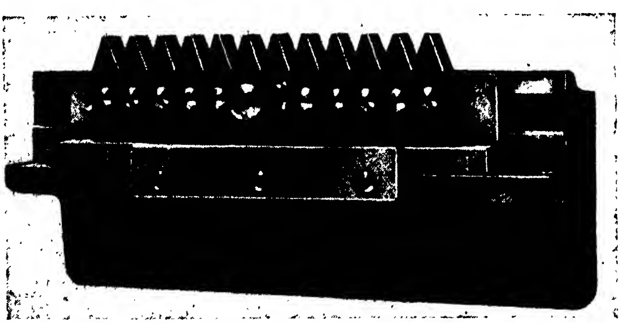


Fig. 185.—How the rack is made.

pitch. The tooth angles are all ground in the same fixture to insure duplication. After they have been assembled an overall check is made and again compared with gages. This is to check the accumulated error which is held within 0.0002 in. in a 12-tooth rack.

Accuracy in Gear Gaging.—A review of measuring devices by J. A. Potter, chief draftsman, Singer Manufacturing Company, now in use, with several new suggestions, is of unusual interest to all gear men.

There are many formulas for the design of gearing and a number of gages for setting the cutters to depth when machining, but too little attention is sometimes paid to the measurement of gears for accuracy.

Consider a pair of spur gears located on two studs on a block. The studs are spaced the same distance apart as the shafts in the machine on which the gears are to be used. This arrangement, which is familiarly known as a run-around gage, is shown in Fig. 186. It can be elaborated in design by using interchangeable studs of different diameters, mounted on adjustable slides, so that a range of center distances may be obtained between the studs, permitting the use of this device for gears varying in pitch, diameter, and bore. This gage will tell just one thing—whether the gears, when assembled at their proper center distance, will mesh and revolve. It may be found that they run hard at one or more points, and the teeth at these points may be chalked and fitted; yet the gage does not show how much fitting will be required. Nor does it show what caused the variation, nor what the effect would be if the gears were run under a load at a high speed.

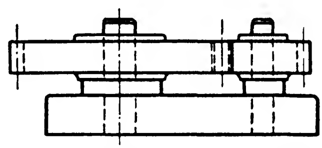


FIG. 186.—A plain "run-around" gage.

It will be conceded by gear manufacturers that the tooth form and its position in relation to the shaft are the principal working considerations in a gear. So why not measure for both? In 1916, the *American Machinist* published one of his articles giving a formula for calculating the diameter of a plug to be held at the pitch line and measured from the bore of a spur gear by a vernier caliper. This process was, of course, tedious, but it showed that a difference of 0.001 in. caused considerable noise at high speed. With the spur gears, the plug lies parallel with the shaft; but bevel and helical gears present different conditions.

With a speed of 5,000 r.p.m., it is desirable that the contact at the pressure angle be equidistant on all teeth in relation to the bore for both spur and helical gears, and in relation to the bore and the seat of the hub for bevel gears. A simple gage for bevel gears, employing a dial indicator, is shown in Fig. 187. The gear to be tested is located on the shaft and is held in position by a set-screw. If the gear has no set-screw, plungers or gibs can be used to hold it against the shoulder on the shaft. The shaft is put into the gage, and when the end *B* contacts with the adjustable screw, *C*, readings may be taken on the dial indicator. Note that the indicator is attached to a bracket having plugs 5 in. apart, the same as in a sine bar, so it may be set angularly

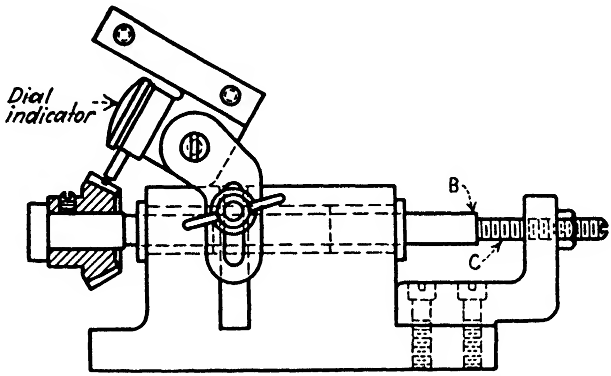


FIG. 187.—A simple gage for bevel gears.

with accuracy. The bracket is adjustable vertically on a tongue to accommodate larger or smaller gears. The gage can be further elaborated by providing interchangeable bushings for various sizes of bores. A ball point may be used on the indicator spindle, but a cone point is preferable. A cone having its sides perpendicular to the line of contact can be worked out by using the formula given in Fig. 190. From this formula, it will be noted that the cone-contact angle varies with the number of teeth in the gear only, regardless of the pitch diameter or the diametral pitch.

The manufacturers of cutters for gears keep to a high standard regarding tooth form, and evidently have their own methods of master gaging to maintain that quality. However, cutters will wear, but their form at the points of contact may easily be tested by optical measuring machines of either the direct-observation, or the prismatic type. The image of the form is greatly enlarged and may be projected on a sensitive plate or traced directly. Reversing the tracing and superimposing it on the original will show whether both sides are uniform. Gears may also be inspected by this method. A bevel gear will, of course, have to

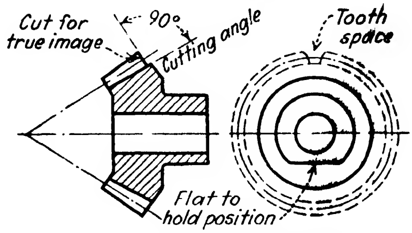


Fig. 188.—Checking bevel gears.

be held at an angle of cut and have the back slabbed off, as in Fig. 188, so that the correct image can be projected.

It is obvious that while the gage in Fig. 187 will show whether the contact points are in the same relative position to the shaft, it will not show this distance in relation to the pitch line. A gage for obtaining positive measurements using a micrometer

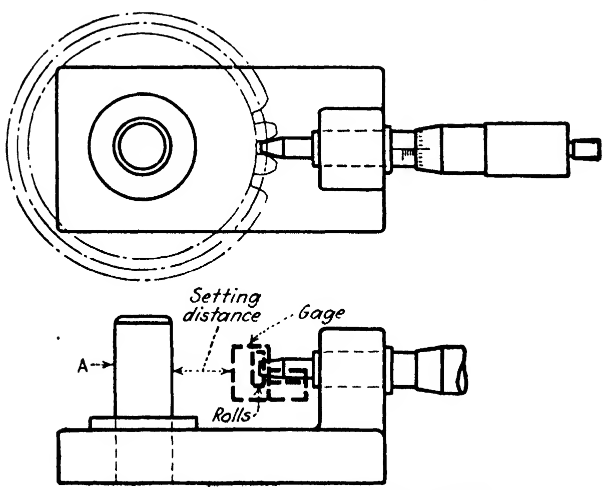


Fig. 189.—Gage using micrometer head.

head is shown in Fig. 189. Note that the end of the spindle is cone-shaped and that it engages the gear as a tangent on each tooth profile at a predetermined depth, thus providing for measurement in relation to the shaft regardless of the outside diameter of the gear. It would appear that there might be considerable difficulty in setting the micrometer for this gage, but a small auxiliary gage can be made for this purpose.

This gage consists of a block containing two steel rolls of proper diameter and center distance to contact the spindle cone at the correct theoretical distance from the gear center. After the cone angle has been determined as in Fig. 190, a reasonable

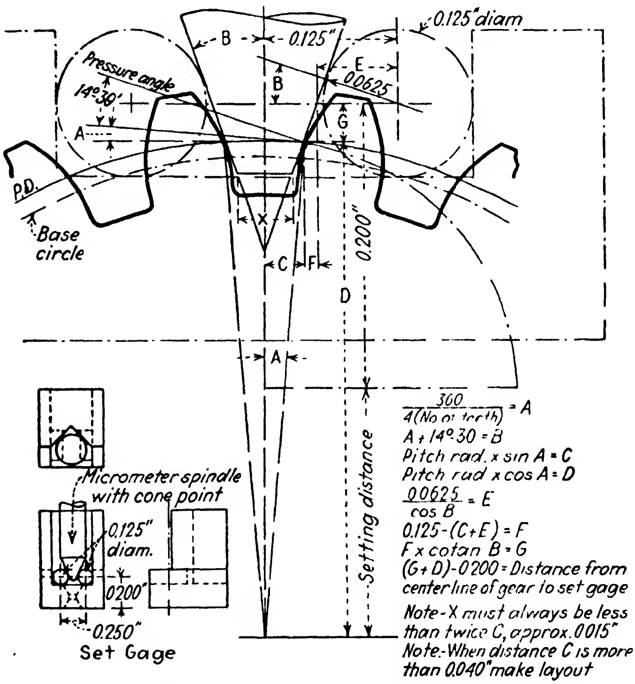
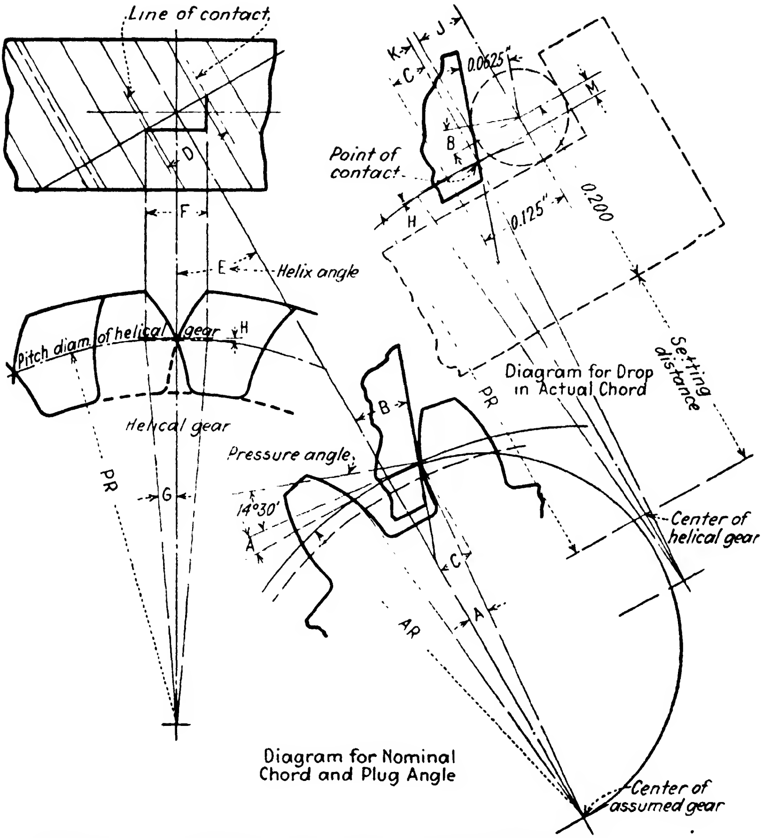


FIG. 190.—Use of cone for checking accuracy of the teeth.

roll diameter and block thickness are arbitrarily assumed. From these dimensions, the setting distance from the back of the gage to the gear center may be computed as shown. In setting the gage, half the arbor diameter is deducted from this distance and size blocks equal to the remainder are placed between the arbor and auxiliary gage. The micrometer spindle is brought into contact with the rolls and adjusted so that it reads an even figure such as zero, 0.250, or 0.500. This setting then represents the calculated micrometer reading when in contact with a theo-



Divide the number of teeth in helical gear by the cube of the cosine of the helical angle

$\frac{n}{(\cos E)^3}$ = *Number of teeth in assumed gear* = N

$\frac{N}{\text{Normal pitch}}$ 0.5 = *Assumed radius* = AR

$\frac{360}{4N}$ = *Assumed angle* = A $\qquad A + 14°30' = B$

$AR \times \sin A = C \qquad 2C = D$

E = *Helix angle* $\qquad D \times \cos E = F$

$\frac{0.5F}{PR} = \sin G \qquad PR - (PR \cos G) = H$

$\frac{0.0625}{\cos B} = J \qquad 0.125 - (C+J) = K \qquad K \times \text{cotan}\, B = M$

$(PR-H) - (0.200'' - M)$ = *Setting distance*

Note.- When distance C is more than 0.040″ make layout of taper plug and setting gage

Note - Bottom of taper plug must be less than twice C, approx. 0.015″

FIG. 191.—**Measuring helical teeth.**

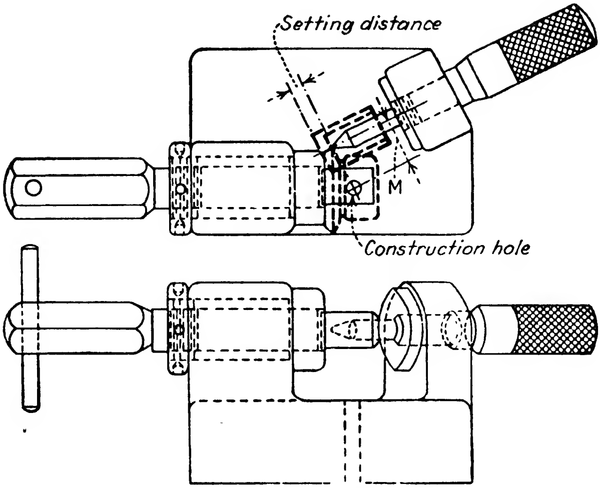


Fig. 192.—Gage for bevel gears.

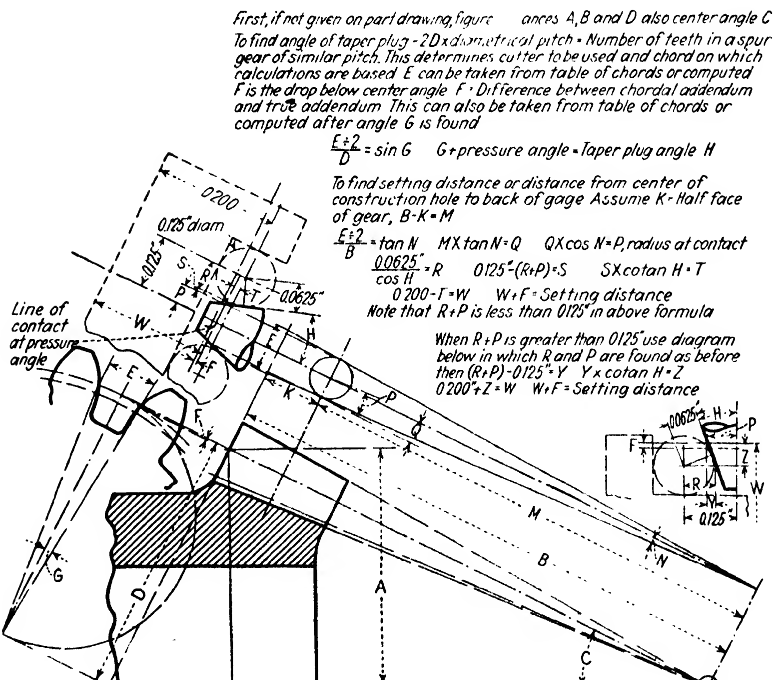


Fig. 193.—Locating micrometer point for a theoretically correct gear.

retically correct gear. Variations from this reading are noted with the gear in place.

Helical gears may be measured in exactly the same way, though the computations for determining the initial gage setting are naturally somewhat more complicated. These are clearly shown in Fig. 191.

For bevel gears the method must be modified. A gage fixture shown in Fig. 192 is employed which uses a construction hole to set the micrometer. Figure 193 shows the mathematics required to obtain the location of the micrometer point for a theoretically correct gear. The micrometer axis is set at right angles to the center angle of the gear at a distance *M* from the cone apex. An auxiliary gage, similar to that used for spur and helical gears, is set at the computed distance from the construction hole. The micrometer spindle set to this gage is at the correct zero reading.

The micrometer method of gaging, as described for spur, helical, and bevel gears, differs from most other methods in that it does more than compare tooth profiles for a given gear. It is based on the true gear form and definitely measures variations from this standard.

GEARING NOMENCLATURE

Confusion is frequently caused by using different terms for the same thing, either in different parts of the country or in different shops in the same locality. This is especially true of the many terms used in connection with gears and gear teeth. To avoid delays and expense caused by such misunderstandings the American Gear Manufacturers Association has adopted names and definitions covering practically all phases of gear work. While these were first adopted in October 1926, they have been revised twice, the list printed herewith being dated May 1931. They will be found most convenient for reference.

This recommended practice includes a complete set of symbols for all elements having a dimensional value. Angles are designated by Greek letters to conform with scientific and engineering textbook practice.

The symbols designating gear-tooth elements and angles have been allotted according to a definite plan, the latter comprising a section of this practice. Only those formulas which are liable to be used in engineering or sales correspondence have been retained, and in order to apply these to the typewriter, American Gear Manufacturers Association standard abbreviations have been used instead of the symbols.

The subject of *gearing nomenclature* is divided into four general classifications:

a. That dealing with a description or definition of the different types of gearing.

b. That dealing with the establishment of abbreviations for use on shop drawings, also for use in formulas when presented in engineering or sales correspondence.

c. That dealing with the establishment of symbols for various gear elements which have a dimensional value for use in scientific and engineering calculations, in textbooks, etc.

d. That dealing with a description or definition of gear elements, including a plan of pictorial presentation.

In this practice classification (*a*) is not covered. It is the opinion of the committee that a description of the different types of gears should be included as an appendix.

A. G. M. A. Adopted Plan for Allotment of Abbreviations. - Abbreviations used to represent gear elements have been developed in accordance with the following rules:

Single-word Terms, such as addendum, clearance, etc.:

RULE 1: Use the first letter of the word capitalized. *Example:* Addendum—*A*; Clearance—*C*.

RULE 2: If there are two or more terms having the same initial letter, use the first letter in combination with the second letter in the word, the first letter to be capital and the additional letter lower case. *Example:* Backlash—*B*; Backing—*Ba*.

Multiple-word Terms, such as circular pitch, cone distance, center distance, etc.:

RULE 3: Use the first letter of each word in capitals, and in the order they occur. *Example:* Circular pitch—*CP*.

RULE 4: If the combination of words results in the same combination of letters for two or more of the terms, the last word in the term should be abbreviated in accordance with Rule 2, under Single-word Terms. *Example:* Cone distance—*CD*; Center distance—*CDi*.

Angles:[1]

RULE 5: The first letter of the abbreviation is always capital *V*. The remainder of the abbreviation is developed in accord-

[1] For convenience all angles have been grouped under angle, and the different angles arranged in alphabetical order.

ance with rules under Single-word Terms and Multiple-word Terms. *Example:* Shaft angle—VS; Spiral angle—VSp.

Special Applications: (To distinguish between gear and pinion abbreviation).

Rule 6: Use the symbol for the term followed by a subletter in lower case for gear and pinion, and subnumbers for intermediates. *Example:* Number of teeth in gear—N_g: in pinion—N_p; in first intermediate—N_1; in second intermediate—N_2; in third intermediate—N_3; etc.

Rule 7: For worm and worm wheel, use capitals for the worm wheel, and lower-case letters for the worm.

Plan of Symbols for Angles—Using Greek Alphabet

Greek letter	Used as a symbol for	Explanation
α	Addendum angle	(alpha—a—addendum)
δ	Dedendum angle	(delta—d—dedendum)
γ_o	Face angle	(gamma—subscript o—outside pitch angle)
ψ	Helix angle	(psi—used as standard mathematical symbol)
λ	Lead angle	(lambda—l—lead)
γ	Pitch angle	(gamma—g—gear)
ϕ	Pressure angle	(phi—p—pressure)
γ_r	Root angle	(gamma—subscript r—root pitch angle)
Σ	Shaft angle	(sigma—s—shaft)

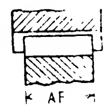
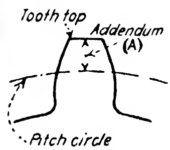
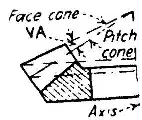
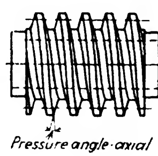
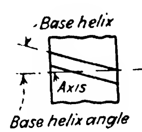
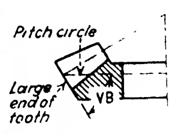
Plan of Symbols for Gear Elements

Letter	Used as a symbol for	Used as a subscript for	Letter	Used as a symbol for
a	Addendum	Active Approach	*A*	Cone distance
b	Dedendum	Base Backing Back cone	*B*	Backlash
c	Clearance	Chordal	*C*	Center distance
d		Diameter	*D*	Diameter
e		Equivalent	*E*	Offset
f	Feed	Face	*F*	Face
g	Number of gashes	Gear	*G*	Lead of gash
h	Tooth depth	Height	*H*	
i		Internal	*I*	Increment
j			*J*	
k		Working	*K*	
l			*L*	Lead
m	Ratio		*M*	Module
n		Normal Normal base Normal chord	*N*	Number of teeth
o		Outside	*O*	
p	Circular pitch	Pitch Pinion	*P*	Diametral pitch
q			*Q*	Arc
r	Radius	Recession Root	*R*	Radius
s			*S*	Space width
t		Throat Total Whole	*T*	Thickness
u			*U*	
v		Virtual	*V*	Velocity
w		Worm	*W*	
x		Axial Axial base Linear	*X*	
y			*Y*	
z			*Z*	

GEARING NOMENCLATURE

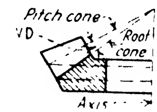
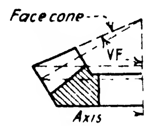
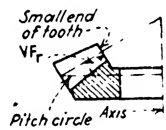
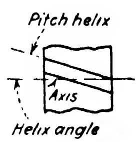
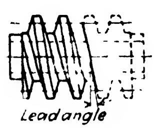
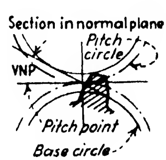
Spur, Helical, Herringbone, Bevel, Spiral Bevel and Worm Gearing

NOTE: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
1	Active face	F_a	*AF*	Is the width of face which actually comes into contact with a mating gear. On herringbone gears it includes both right and left hand helices, minus the groove width if any		
2	Addendum	a	*A*	Is the radial or perpendicular distance between the pitch circle and the top of the tooth Applies to all types of gearing	$A = 1/DP \times$ factor	
3	Angle, addendum	α	*VA*	Is the angle between elements of the pitch and face cones in a plane containing the axis of the gear. Applies to bevel gearing	Tan $VA = A/CD$	
4	Angle, axial pressure	ϕ_r	*VAP*	Is the angle in an axial plane between the side of the tooth or thread and a line perpendicular to the axis. Applies to helical and worm gearing	Tan $VAP = \dfrac{\tan VNP}{\cos VL}$	
5	Angle, base helix	ψ_b	*VBH*	Is the helix angle on the base cylinder. Applies to helical and worm gearing	Tan $VBH = \dfrac{BD \times 3.1416}{L}$	
6	Angle, back		*VB*	Is the angle between the plane of the pitch circle and a plane tangent to the large end of the tooth. Applies to bevel gearing	$VB = VPi$	

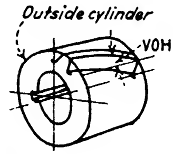
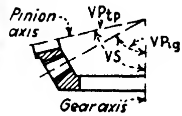
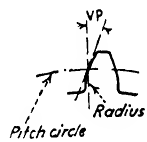
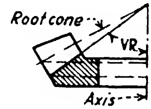
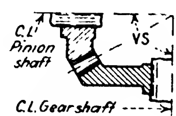
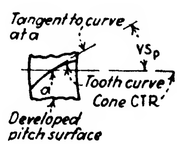
GEARING NOMENCLATURE.—(*Continued*)

NOTE: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
7	Angle, dedendum	δ	*VD*	Is the angle between elements of the pitch and root cones in a plane containing the axis of the gear. Applies to bevel gearing	$\tan VD = D/CD$	
8	Angle, face	γ_o	*VF*	Is the angle between an element of the face cone and its axis. Applies to bevel gearing	$VF = VPi + VA$	
9	Angle, front		VF_r	Is the angle between the plane of the pitch circle and a plane tangent to the small end of the tooth. Applies to bevel gearing	$VF_r = VPi$	
10	Angle, helix	ψ	*VH*	Is the angle between a tangent to a helix and an element of the cylinder. Unless otherwise specified, the pitch helix is referred to. Applies to helical and worm gearing	$\tan VH = \dfrac{PD \times 3.1416}{L}$	
11	Angle, lead	λ	*VL*	Is the angle between a tangent to the pitch helix and a plane of rotation. Applies to worm and helical gearing	$\tan VL = \dfrac{L}{PD \times 3.1416}$	
12	Angle, normal pressure	ϕ_n	*VNP*	Is the pressure angle in a plane normal to the pitch line element. Applies to helical, worm and spiral-bevel gearing	$\tan VNP = \tan VP \times \cos VH$	

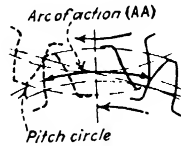
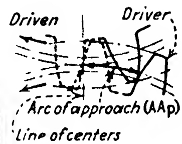
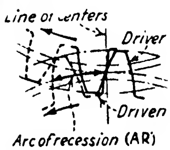
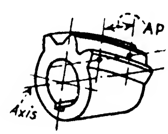
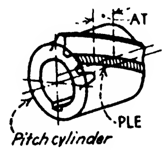
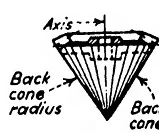
GEARING NOMENCLATURE.—(*Continued*)

NOTE: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
13	Angle, outside helix	ψ_o	*VOH*	Is the helix angle on the outside cylinder. It is sometimes used for trial settings. Applies to helical and worm gearing	Tan $VOH = \dfrac{OD \times 3.1416}{L}$	
14	Angle, pitch	γ	*VPi*	Is the angle between an element of the pitch cone and its axis. Applies to bevel gearing	When $VS = 90°$ Tan $VPig = Ng/Np$	
15	Angle, pressure	ϕ	*VP*	Is the angle between a tangent to the tooth profile, and a line perpendicular to the pitch surface. Applies to all types of gears. In spiral-bevel gears, pressure angle means normal pressure angle at the middle of the face unless otherwise specified		
16	Angle, root	γ_r	*VR*	Is the angle between an element of the root cone and its axis. Applies to bevel gearing	$VR = VPi - VD$	
17	Angle, shaft	Σ	*VS*	Is the included angle between the shafts upon which a pair of mating gears are to operate. Applies to helical bevel and worm gearing	$VS = VPig + VPip$	
18	Angle, spiral	ψ	*VSp*	Is the angle at which a tooth intersects an element of the pitch cone. Unless otherwise specified, this angle is understood to be at the middle of the face. Applies to spiral-bevel gearing		

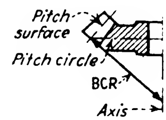
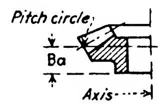
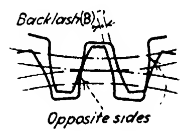
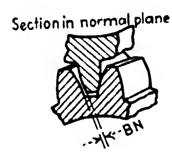
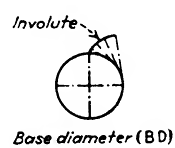
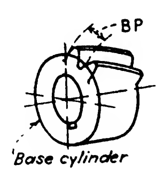
GEARING NOMENCLATURE.—(*Continued*)

NOTE: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
19	Arc of action	Q	AA	Is the arc of the pitch circle through which a tooth travels from the time it first makes contact with a mating tooth until contact with mating tooth ceases. Applies to all types. In bevel gearing, the virtual pitch circles are used		
20	Arc of approach	Q_a	AAp	Is the arc of the pitch circle through which a tooth travels from the time it first makes contact with a mating tooth until it is in contact at the pitch point. Applies to all types		
21	Arc of recession	Q_r	AR	Is the arc of the pitch circle through which a tooth travels from the time it is in contact with a mating tooth at the pitch point until contact ceases. Applies to all types		
22	Axial pitch	p_x	AP	Is the distance parallel to the axis of the gear between corresponding sides of adjacent teeth. Applies to helical and worm gearing	$AP = NCP/\sin VH$	
23	Axial thickness	T_x	AT	Is the distance parallel to the axis between the two pitch line elements of the same tooth. Applies to helical and worm gearing	$AT = NCT/\sin VH$	
24	Back cone		BC	Is the cone generated by revolving the back cone radius about the axis of the gear. Applies to bevel gearing		

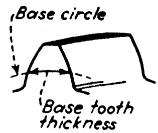
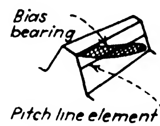
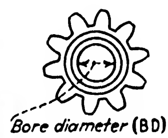
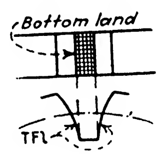
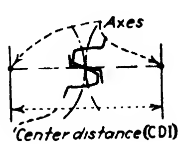
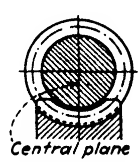
GEARING NOMENCLATURE.—(*Continued*)

NOTE: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
25	Back cone radius	R_b	*BCR*	Is the distance perpendicular to the pitch surface from the pitch circle to the axis. Applied to bevel gearing	$BCR = CD \times \tan V$	
26	Backing		*Ba*	Is the distance parallel to the axis from the pitch circle to a shoulder or hub extension. Applies to bevel gearing		
27	Backlash	B	*B*	Is the shortest distance between nondriving tooth surfaces of adjacent teeth in mating gears. Applies to all types		
28	Backlash, normal	B_n	*BN*	Is the backlash taken in a plane normal to the pitch line element. Applies to helical, worm and spiral-bevel gearing		
29	Base diameter	D_b	*BD*	Is the diameter of the circle from which the involute is generated. Applies to all involute type gearing	$BD = PD \times \cos VP$	
30	Base pitch	p_b	*BP*	Is the circular pitch taken on the circumference of the base circles, and is also the distance along the line of action between two successive and corresponding involute tooth profiles. Applies to all forms of involute gearing	$BP = CP \times \cos VP$	

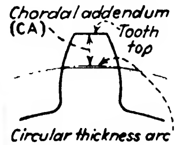
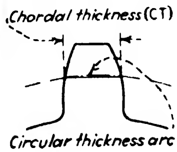
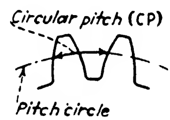
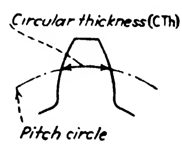
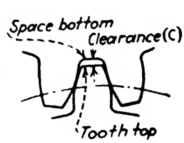
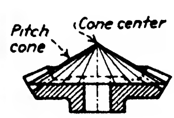
GEARING NOMENCLATURE.—(*Continued*)

NOTE: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
31	Base-tooth thickness	T_b	*BTT*	Is the distance on the base circle between the involutes of the same tooth. Applies to all forms of involute gearing		
32	Bias bearing		*BB*	Is a tooth bearing which is oblique to the pitch line element. Applies particularly to bevel gearing		
33	Bore diameter		*BDi*	Is the diameter of the hole in the gear. Applies to all types		
34	Bottom land		*BL*	Is the surface of the gear between the flanks of adjacent teeth. Applies to all types		
35	Center distance	C	*CDi*	Is the shortest distance between non-intersecting axes of mating gears	$CDi = \dfrac{N_g + N_p}{2DP}$	
36	Central plane		*CPl*	Is the plane of rotation of worm gear passing through worm axis. Applies to worm gearing		

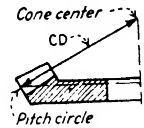
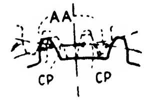
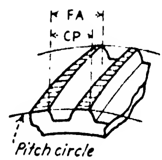
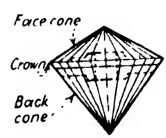
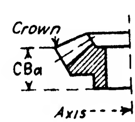
GEARING NOMENCLATURE.—(*Continued*)

NOTE: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
37	Chordal addendum	a_c	CA	Is the radial distance from the circular thickness chord to the top of the tooth. Applies to all types		
38	Chordal thickness	T_c	CT	Is the length of the chord subtended by the circular thickness arc. Applies to all types		
39	Circular pitch	p	CP	Is the distance on the circumference of the pitch circle between corresponding points of adjacent teeth. Applies to all types	$CP = 3.1416/DP$	
40	Circular thickness	T	CTh	Is the thickness of the tooth on the pitch circle. Applies to all types		
41	Clearance	c	C	Is the radial distance between the top of a tooth and the bottom of the mating tooth space. Applies to all types	$C = 1/DP \times$ factor	
42	Cone center		CC	Is the apex of the pitch cone. Applies to bevel gearing		

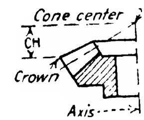
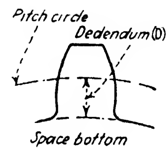
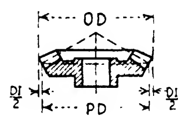
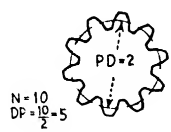
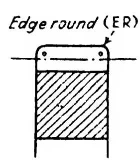
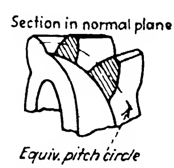
GEARING NOMENCLATURE.—(*Continued*)

NOTE: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
43	Cone distance	A	CD	Is the distance from the cone center to any point on the pitch circle. Applies to bevel gearing	$CD = \frac{\frac{1}{2}PD}{\sin VPi}$	
44	Contact ratio	m_p	CR	Is the ratio of the arc of action to the circular pitch. Applies to all types. See number of teeth in contact	$CR = AA/CP$	
45	Contact ratio, face	m_f	CRf	Is the ratio of the face advance to the circular pitch. Applies to helical and spiral-bevel gearing	$CRf = FA/CP$	
46	Contact ratio, total	m_t	CRt	Is the ratio of the sum of the arc of action and the face advance to the circular pitch. Applies to helical and spiral-bevel gearing	$CRt = \frac{FA + AA}{CP}$	
47	Crown		Cr	Is the circle formed by the intersection of the face cone and the back cone extended. Applies to bevel gearing		
48	Crown backing		CBa	Is the distance parallel to the axis from the crown to a shoulder or hub extension. Applies to bevel gearing	$CBa = MD - CH$	

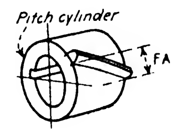
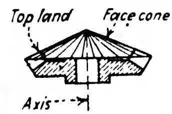
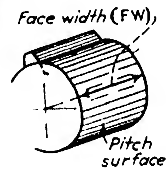
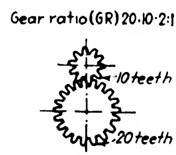
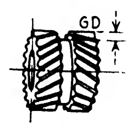
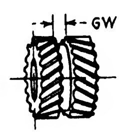
GEARING NOMENCLATURE.—(*Continued*)

NOTE: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
49	Crown height		*CH*	Is the distance parallel to the axis from the cone center to the crown of the gear. Applies to bevel gearing	$CH = (CD \times \cos VPi) - (A \times \sin VPi)$	
50	Dedendum	b	*D*	Is the radial or perpendicular distance between the pitch circle and the bottom of the tooth space. Applies to all types	$D = 1/DP \times$ factor	
51	Diameter increment	I_d	*DI*	Is the amount added to the pitch diameter to obtain the outside diameter. Applies to bevel gearing	$DI = 2A \times \cos VPi$	
52	Diametral pitch	P	*DP*	Is the ratio of the number of teeth to the number of inches in the pitch diameter. It indicates the number of teeth in the gear for each inch of pitch diameter. Applies to all types	$DP = N/PD$	
53	Edge round		*ER*	Is the radius on the circumferential edge of a gear tooth—to break the sharp corner. Applies to all types		
54	Equivalent pitch radius	R_e	*EPR*	Is the radius of curvature of the pitch surface in a plane normal to the pitch line element. Applies to helical and spiral-bevel gearing		

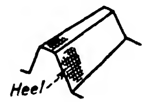
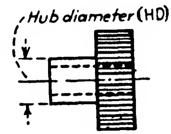
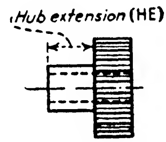
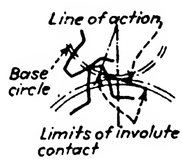
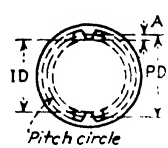
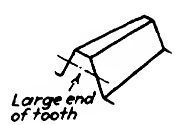
GEARING NOMENCLATURE.—(*Continued*)

NOTE: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
55	Face advance	Q_f	*FA*	Is the distance on the pitch circle that a gear tooth travels from the time pitch point contact is made at one end of the tooth until pitch point contact is made at the other end. Applies to helical and spiral-bevel gearing		
56	Face cone		*FC*	Is the right circular cone whose elements contain the top lands of the gear. Applies to bevel gearing		
57	Face width	*F*	*FW*	Is the width of the pitch surface. For herringbone gears this applies to the width of the surface containing both of the helices plus the groove width		
58	Gear ratio	*m*	*GR*	Is the ratio of the numbers of teeth in mating members	$GR = Ng:Np$	
59	Groove depth		*GD*	Is the depth of the clearance groove between helices. Applies to herringbone gearing		
60	Groove width		*GW*	Is the width of the clearance groove between helices. Applies to herringbone gearing		

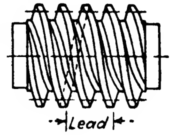
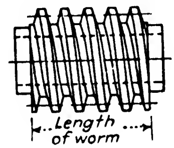
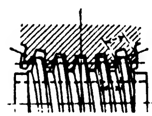
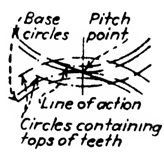
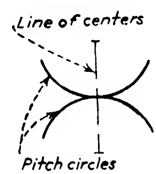
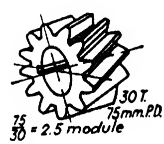
GEARING NOMENCLATURE.—(*Continued*)

NOTE: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
61	Heel		H	Is a portion of the tooth at the large end. Applies to bevel gearing		
62	Hub diameter		HD	Is the diameter of the central part of the gear body surrounding the bore and extending beyond the web, spokes or body		
63	Hub extension		HE	Is the distance that the hub extends beyond the face of the gear body		
64	Interference		I	Is contact between mating teeth at some other point than along the line of action		
65	Internal diameter	D_i	ID	Is the diameter of that circle which contains the tops of the teeth of an internal gear	$ID = PD - 2A$	
66	Large end of tooth		LET	Is that end of a bevel gear tooth most remote from the cone center		

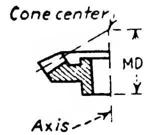
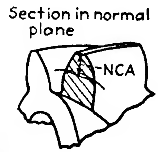
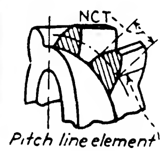
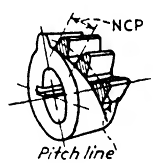
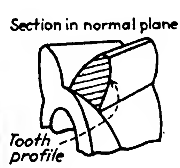
GEARING NOMENCLATURE.—(*Continued*)

NOTE: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
67	Lead	L	L	Is the axial advance of the helix in one complete turn. Applies to helical and worm gearing	$L = N \times AP$	
68	Length of worm		WL	Is the length of full-threaded portion of worm		
69	Linear pitch	p_x	LP	Is the distance parallel to the axis between corresponding sides of adjacent teeth or thread on a worm. Same as axial pitch	$LP = \dfrac{3.1416}{DP \text{ (gear)}}$	
70	Line of action		LA	Is that portion of the common tangent to the base circles along which contact between mating involutes occurs. Applies to all types		
71	Line of centers		LC	Is the straight line through the center of tangent pitch circles. Applies to all types. In bevel gearing, the circles are virtual pitch circles		
72	Module (metric)	M	M	Is the ratio of the pitch diameter in millimeters to the number of teeth	$M = PD/N$	

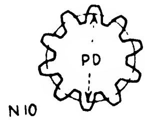
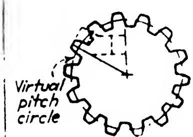
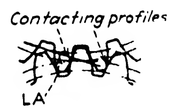
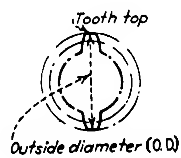
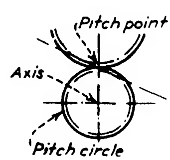
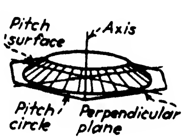
GEARING NOMENCLATURE.—(*Continued*)

NOTE: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
73	Mounting distance		*MD*	Is the distance parallel to the axis from the cone center to the shoulder or hub end against which the gear is mounted. Applies to bevel gearing		
74	Normal chordal addendum	a_n	*NCA*	Is the perpendicular distance from the normal thickness chord to the top of the tooth. Applies to helical and spiral-bevel gearing		
75	Normal chordal thickness	T_n	*NCT*	Is the length of the normal thickness chord between the pitch line elements of a tooth. Applies to helical and spiral-bevel gearing		
76	Normal circular pitch	p_n	*NCP*	Is the shortest distance on the pitch surface between corresponding pitch line elements of adjacent teeth. Applies to helical and spiral-bevel gearing	$NCP = \frac{3.1416}{NDP}$	
77	Normal diametral pitch	P_n	*NDP*	Is the diametral pitch corresponding to the normal circular pitch. Applies to helical, and herringbone gearing	$NDP = \frac{DP}{\cos VH}$	
78	Normal tooth profile		*NTP*	Is the outline formed by the intersection of a tooth surface and a plane perpendicular to its pitch line element. Applies to helical and spiral-bevel gearing		

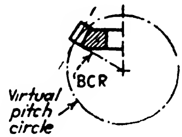
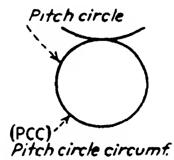
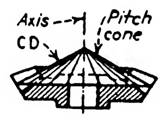
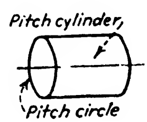
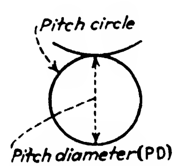
GEARING NOMENCLATURE.—(*Continued*)

NOTE: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
79	Number of teeth or threads	N	N	Is the number of teeth or threads contained in a gear or worm	$N = PD \times DP$ $N_v = PD_t \times DP$ $N_p = PD_p \times DP$	
80	Number of teeth, virtual	N_v	NV	Is the number of teeth of a given pitch which would be contained in the virtual pitch circle. Applies particularly to bevel gearing	$NV = \frac{N}{\cos VPi}$	
81	Number of teeth in contact	Z	NTC	Is the number of profile contacts in a pair of mating gears at any given instant. For average number of teeth in contact, see Contact Ratio. Applies to all types		
82	Outside diameter	D_o	OD	Is the diameter of the circle which contains the tops of the teeth. Applies to all types	$OD = 2A + PD$	
83	Pitch circle		PC	Is the circle through the pitch point having its center at the axis of the gear. Applies to spur, helical and worm gearing		
				Is the circle formed by the intersection of the pitch cone and a plane perpendicular to the axis. Applies to bevel gearing		

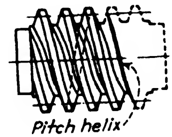
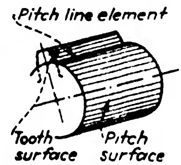
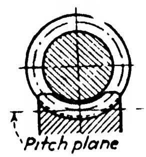
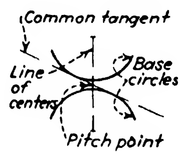
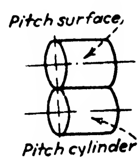
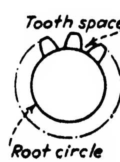
GEARING NOMENCLATURE.—(*Continued*)

NOTE: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
84	Pitch circle, virtual		PC_v	Is the circle whose radius is the back cone radius. Applies to bevel gearing		
85	Pitch circle circumference		PCC	Is the circumference of the pitch circle. Applies to all types	$PCC = PD \times 3.1416$	
86	Pitch cone		PCo	Is the cone generated by revolving the cone distance line about the axis of the gear. Applies to bevel gearing		
87	Pitch cylinder		PCy	Is the cylinder corresponding to the pitch circle		
88	Pitch diameter	D	PD	Is the diameter of the pitch circle. Applies to all types	$PD = N/DP$; $PD_g = N_g/DP$, $PD_p = N_p/DP$	
89	Pitch radius virtual	R_v	PR_v	Is the radius of the virtual pitch circle. Applies to bevel gearing	$PR_v = BCR$	

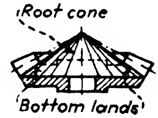
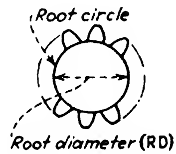
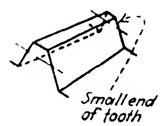
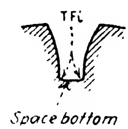
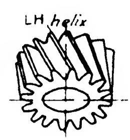
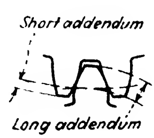
Gearing Nomenclature.—(*Continued*)

Note: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
90	Pitch helix		*PH*	Is the helix formed by the intersection of the surface of a helical tooth or thread with the pitch cylinder		
91	Pitch line element		*PLE*	Is a line curved or straight, formed by the intersection of the pitch surface and the tooth surface. Applies to all types		
92	Pitch plane		*PPl*	Is the plane which is tangent to both the pitch cylinder of the worm and the pitch cylinder of the gear. Applies to worm gearing		
93	Pitch point		*PP*	Is the intersection, between the axes, of the line of centers and the common tangent to the base circles. Applies to all types of involute gearing		
94	Pitch surface		*PS*	Is the surface of the pitch cylinder or pitch cone, which rolls with the surface of the mating member. Applies to all types of gearing		
95	Root circle		*RC*	Is the circle containing the bottoms of the tooth spaces. Applies to all types		

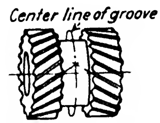
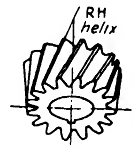
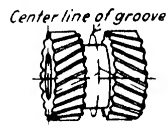
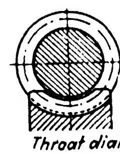
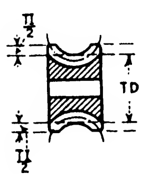
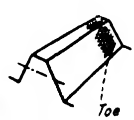
GEARING NOMENCLATURE.—(*Continued*)

NOTE: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
96	Root cone		*RCo*	Is the right circular cone whose elements contain the bottom lands of the gear. Applies to bevel gearing		
97	Root diameter	D_r	*RD*	Is the diameter of the root circle. Applies to all types	$RD = PD - 2D$	
98	Small end of tooth		*SET*	Is the end of a bevel gear tooth nearest the cone center		
99	Space bottom		*SB*	Is a line joining two fillets of adjacent tooth profiles in the same plane. Applies to all types		
100	Teeth, left-hand		*Tlh*	Are teeth which twist to the left, or in a counter-clockwise direction, as they recede from the observer. Applies to helical and spiral-bevel gearing		
101	Teeth, long and short addendum		*Tlsd*	Have a longer addendum on the teeth of one member than on those of the mating gear. Applies to all types of gearing		

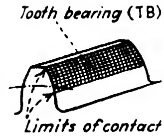
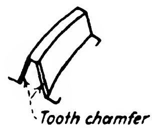
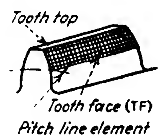
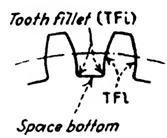
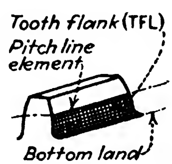
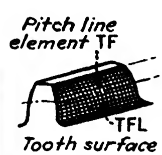
GENERAL NOMENCLATURE.—(*Continued*)

NOTE: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
102	Teeth, matched		*Tm*	Are herringbone gear teeth, the pitch line elements of which intersect or would intersect at the center of the groove if prolonged. Applies to herringbone gearing		
103	Teeth, right-hand		*Trh*	Are teeth which twist to the right, or in a clockwise direction, as they recede from the observer. Applies to helical and spiral-bevel gearing		
104	Teeth, staggered		*Ts*	Are herringbone gear teeth, the pitch line elements of which do not intersect, or would not intersect at the center of the groove if prolonged. Applies to herringbone gearing		
105	Throat diameter	D_t	*TD*	Is the outside diameter of worm gear in central plane. Applies to worm gearing		
106	Throat increment	I_t	*TI*	Is the amount added to the throat diameter to obtain the maximum diameter of the worm wheel blank. Applies to worm gearing	$TI = OD - TD$	
107	Toe		*T*	Is a portion of the tooth at the small end. Applies to bevel gearing		

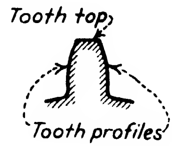
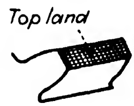
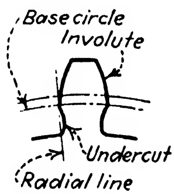
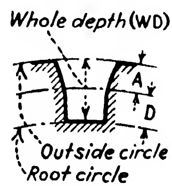
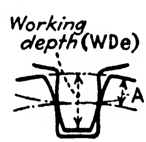
GEARING NOMENCLATURE.—(*Continued*)

NOTE: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
108	Tooth bearing		*TB*	Is that portion of the tooth surface which actually comes into contact. Applies to all types		
109	Tooth chamfer		*TC*	Is the bevel between the end of a tooth and the tooth surface —to break the sharp edge. Applies to all types		
110	Tooth face		*TF*	Is the surface between the pitch line element and the top of the tooth. Applies to all types		
111	Tooth fillet		*TFi*	Is the curved line joining the tooth flank and the bottom of the tooth space. Applies to all types		
112	Tooth flank		*TFl*	Is the surface between the pitch line element and the bottom land —it includes the fillet. Applies to all types		
113	Tooth surface		*TS*	Is the total area including the tooth face and tooth flank. Applies to all types		

GEARING NOMENCLATURE.—(*Continued*)

NOTE: Only those terms having a dimensional value are symbolized

No.	Term	Symbol	Abb.	Definition	Formula	Illustration
114	Tooth top		*TT*	Is a line joining the outer ends of two adjacent tooth profiles in the same plane. Applies to all types. In internal gearing it is the inner ends of the teeth		
115	Top land		*TL*	Is the surface of the top of the tooth. Applies to all types		
116	Undercut		*U*	Is that portion of the tooth surface, adjacent to the involute, lying inside a radial line passing through the imaginary intersection of the involute and the base circle. Applies to all types		
117	Whole depth	h_t	*WD*	Is the radial distance between the outside circle and the root circle. Applies to all types of gearing	$WD = A + D$	
118	Working depth	h_k	*WDe*	Is the greatest depth to which a tooth of one gear extends into the tooth space of a mating gear. Applies to all types of gearing	$WD_e = A_p + A_g$	

TABLES FOR INDEXING

Number of divisions	Index circle	Number of turns of index	Gear on worm	No. 1 hole First gear on stud	No. 1 hole Second gear on stud	Gear on spindle	Idlers No. 1 hole	Idlers No. 2 hole
2	Any	20						
3	39	$13\frac{13}{39}$						
4	Any	10						
5	Any	8						
6	39	$6\frac{26}{39}$						
7	49	$5\frac{35}{49}$						
8	Any	5						
9	27	$4\frac{12}{27}$						
10	Any	4						
11	33	$3\frac{21}{33}$						
12	39	$3\frac{13}{39}$						
13	39	$3\frac{3}{39}$						
14	49	$2\frac{42}{49}$						
15	39	$2\frac{26}{39}$						
16	20	$2\frac{10}{20}$						
17	17	$2\frac{6}{17}$						
18	27	$2\frac{6}{27}$						
19	19	$2\frac{2}{19}$						
20	Any	2						
21	21	$1\frac{19}{21}$						
22	33	$1\frac{27}{33}$						
23	23	$1\frac{17}{23}$						
24	39	$1\frac{26}{39}$						
25	20	$1\frac{12}{20}$						
26	39	$1\frac{21}{39}$						
27	27	$1\frac{13}{27}$						
28	49	$1\frac{21}{49}$						
29	29	$1\frac{11}{29}$						
30	39	$1\frac{13}{39}$						
31	31	$1\frac{9}{31}$						
32	20	$1\frac{5}{20}$						
33	33	$1\frac{7}{33}$						
34	17	$1\frac{3}{17}$						
35	49	$1\frac{7}{49}$						
36	27	$1\frac{3}{27}$						
37	37	$1\frac{3}{37}$						
38	19	$1\frac{1}{19}$						
39	39	$1\frac{1}{39}$						

TABLES FOR INDEXING.—(*Continued*)

Number of divisions	Index circle	Number of turns of index	Gear on worm	No. 1 hole		Gear on spindle	Idlers	
				First gear on stud	Second gear on stud		No. 1 hole	No. 2 hole
40	Any	1						
41	41	40/41						
42	21	20/21						
43	43	40/43						
44	33	30/33						
45	27	24/27						
46	23	20/23						
47	47	40/47						
48	18	15/18						
49	49	40/49						
50	20	16/20						
51	17	14/17	24			48	24	44
52	39	30/39						
53	49	35/49	56	40	24	72		
54	27	20/27						
55	33	24/33						
56	49	35/49						
57	21	15/21	56			40	24	44
58	29	20/29						
59	39	26/39	48			32	44	
60	39	26/39						
61	39	26/39	48			32	24	44
62	31	20/31						
63	39	26/39	24			48	24	44
64	16	10/16						
65	39	24/39						
66	33	20/33						
67	21	12/21	28			48	44	
68	17	10/17						
69	20	12/20	40			56	24	44
70	49	28/49						
71	18	10/18	72			40	24	
72	27	15/27						
73	21	12/21	28			48	24	44
74	37	20/37						
75	15	8/15						
76	19	10/19						
77	20	10/20	32			48	44	

TABLES FOR INDEXING.—*(Continued)*

Number of divisions	Index circle	Number of turns of index	Gear on worm	No. 1 hole		Gear on spindle	Idlers	
				First gear on stud	Second gear on stud		No. 1 hole	No. 2 hole
78	39	20/39						
79	20	10/20	48			24	44	
80	20	10/20						
81	20	10/20	48			24	24	44
82	41	20/41						
83	20	10/20	32			48	24	44
84	21	10/21						
85	17	8/17						
86	43	20/43						
87	15	7/15	40			24	24	44
88	33	15/33						
89	18	8/18	72			32	44	
90	27	12/27						
91	39	18/39	24			48	24	44
92	23	10/23						
93	18	8/18	24			32	24	44
94	47	20/47						
95	19	8/19						
96	21	9/21	28			32	24	44
97	20	8/20	40			48	44	
98	49	20/49						
99	20	8/20	56	28	40	32		
100	20	8/20						
101	20	8/20	72	24	40	48		24
102	20	8/20	40			32	24	44
103	20	8/20	40			48	24	44
104	39	15/39						
105	21	8/21						
106	43	16/43	86	24	24	48		
107	20	8/20	40	56	32	64		24
108	27	10/27						
109	16	6/16	32			28	24	44
110	33	12/33						
111	39	13/39	24			72	32	
112	39	13/39	24			64	44	
113	39	13/39	24			56	44	
114	39	13/39	24			48	44	
115	23	8/23						

TABLES FOR INDEXING.—(*Continued*)

Number of divisions	Index circle	Number of turns of index	Gear on worm	No. 1 hole		Gear on spindle	Idlers	
				First gear on stud	Second gear on stud		No. 1 hole	No. 2 hole
116	29	$\frac{10}{29}$						
117	39	$\frac{13}{39}$	24			24	56	
118	39	$\frac{13}{39}$	48			32	44	
119	39	$\frac{13}{39}$	72			24	44	
120	39	$\frac{13}{39}$						
121	39	$\frac{13}{39}$	72			24	24	44
122	39	$\frac{13}{39}$	48			32	24	44
123	39	$\frac{13}{39}$	24			24	24	44
124	31	$\frac{10}{31}$						
125	39	$\frac{13}{39}$	24			40	24	44
126	39	$\frac{13}{39}$	24			48	24	44
127	39	$\frac{13}{39}$	24			56	24	44
128	16	$\frac{5}{16}$						
129	39	$\frac{13}{39}$	24			72	24	44
130	39	$\frac{12}{39}$						
131	20	$\frac{6}{20}$	40			28	44	
132	33	$\frac{10}{33}$						
133	21	$\frac{6}{21}$	24			48	44	
134	21	$\frac{6}{21}$	28			48	44	
135	27	$\frac{8}{27}$						
136	17	$\frac{5}{17}$						
137	21	$\frac{6}{21}$	28			24	56	
138	21	$\frac{6}{21}$	56			32	44	
139	21	$\frac{6}{21}$	56	32	48	24		
140	49	$\frac{14}{49}$						
141	18	$\frac{5}{18}$	48			40	44	
142	21	$\frac{6}{21}$	56			32	24	44
143	21	$\frac{6}{21}$	28			24	24	44
144	18	$\frac{5}{18}$						
145	29	$\frac{8}{29}$						
146	21	$\frac{6}{21}$	28			48	24	44
147	21	$\frac{6}{21}$	24			48	24	44
148	37	$\frac{10}{37}$						
149	21	$\frac{6}{21}$	28			72	24	44
150	15	$\frac{4}{15}$						
151	20	$\frac{5}{20}$	32			72	44	
152	19	$\frac{5}{19}$						
153	20	$\frac{5}{20}$	32			56	44	

TABLES FOR INDEXING.—(*Continued*)

Number of divisions	Index circle	Number of turns of index	Gear on worm	No. 1 hole: First gear on stud	No. 1 hole: Second gear on stud	Gear on spindle	Idlers: No. 1 hole	Idlers: No. 2 hole
154	20	5/20	32			48	44	
155	31	8/31						
156	39	10/39						
157	20	5/20	32			24	56	
158	20	5/20	48			24	44	
159	20	5/20	64	32	56	28		
160	20	5/20						
161	20	5/20	64	32	56	28		24
162	20	5/20	48			24	24	44
163	20	5/20	32			24	24	44
164	41	10/41						
165	33	8/33						
166	20	5/20	32			48	24	44
167	20	5/20	32			56	24	44
168	21	5/21						
169	20	5/20	32			72	24	44
170	17	4/17						
171	21	5/21	56			40	24	44
172	43	10/43						
173	18	4/18	72	56	32	64		
174	18	4/18	24			32	56	
175	18	4/18	72	40	32	64		
176	18	4/18	72	24	24	64		
177	18	4/18	72			48	24	
178	18	4/18	72			32	44	
179	18	4/18	72	24	48	32		
180	18	4/18						
181	18	4/18	72	24	48	32		24
182	18	4/18	72			32	24	44
183	18	4/18	48			32	24	44
184	23	5/23						
185	37	8/37						
186	18	4/18	48			64	24	44
187	18	4/18	72	48	24	56		24
188	47	10/47						
189	18	4/18	32			64	24	44
190	19	4/19						
191	20	4/20	40			72	24	

TABLES FOR INDEXING.—(*Continued*)

Number of divisions	Index circle	Number of turns of index	Gear on worm	No. 1 hole: First gear on stud	No. 1 hole: Second gear on stud	Gear on spindle	Idlers: No. 1 hole	Idlers: No. 2 hole
192	20	4/20	40			64	44	
193	20	4/20	40			56	44	
194	20	4/20	40			48	44	
195	39	8/39						
196	49	10/49						
197	20	4/20	40			24	56	
198	20	4/20	56	28	40	32		
199	20	4/20	100	40	64	32		
200	20	4/20						
201	20	4/20	72	24	40	24		24
202	20	4/20	72	24	40	48		24
203	20	4/20	40			24	24	44
204	20	4/20	40			32	24	44
205	41	8/41						
206	20	4/20	40			48	24	44
207	20	4/20	40			56	24	44
208	20	4/20	40			64	24	44
209	20	4/20	40			72	24	44
210	21	4/21						
211	16	3/16	64			28	44	
212	43	8/43	86	24	24	48		
213	27	5/27	72			40	44	
214	20	4/20	40	56	32	64		24
215	43	8/43						
216	27	5/27						
217	21	4/21	48			64	24	44
218	16	3/16	64			56	24	44
219	21	4/21	28			48	24	44
220	33	6/33						
221	17	3/17	24			24	56	
222	18	3/18	24			72	44	
223	43	8/43	86	48	24	64		24
224	18	3/18	24			64	44	
225	27	5/27	24			40	24	44
226	18	3/18	24			56	44	
227	49	8/49	56	64	28	72		
228	18	3/18	24			48	44	
229	18	3/18	24			44	48	

TABLES FOR INDEXING.—*(Continued)*

Number of divisions	Index circle	Number of turns of index	Gear on worm	No. 1 hole: First gear on stud	No. 1 hole: Second gear on stud	Gear on spindle	Idlers: No. 1 hole	Idlers: No. 2 hole
230	23	4/23						
231	18	3/18	32			48	44	
232	29	5/29						
233	18	3/18	48			56	44	
234	18	3/18	24			24	56	
235	47	8/47						
236	18	3/18	48			32	44	
237	18	3/18	48			24	44	
238	18	3/18	72			24	44	
239	18	3/18	72	24	64	32		
240	18	3/18						
241	18	3/18	72	24	64	32		24
242	18	3/18	72			24	24	44
243	18	3/18	64			32	24	44
244	18	3/18	48			32	24	44
245	49	8/49						
246	18	3/18	24			24	24	44
247	18	3/18	48			56	24	44
248	31	5/31						
249	18	3/18	32			48	24	44
250	18	3/18	24			40	24	44
251	18	3/18	48	44	32	64		24
252	18	3/18	24			48	24	44
253	33	5/33	24			40	56	
254	18	3/18	24			56	24	44
255	18	3/18	48	40	24	72		24
256	18	3/18	24			64	24	44
257	49	8/49	56	48	28	64		24
258	43	7/43	32			64	24	44
259	21	3/21	24			72	44	
260	39	6/39						
261	29	4/29	48	64	24	72		
262	20	3/20	40			28	44	
263	49	3/49	56	64	28	72		24
264	33	5/33						
265	21	3/21	56	40	24	72		
266	21	3/21	32			64	44	
267	27	4/27	72			32	44	

TABLES FOR INDEXING.—(*Continued*)

Number of divisions	Index circle	Number of turns of index	Gear on worm	No. 1 hole		Gear on spindle	Idlers	
				First gear on stud	Second gear on stud		No. 1 hole	No. 2 hole
268	21	$3/21$	28			48	44	
269	20	$3/20$	64	32	40	28		24
270	27	$4/27$						
271	21	$3/21$	56			72	24	
272	21	$3/21$	56			64	24	
273	21	$3/21$	24			24	56	
274	21	$3/21$	56			48	44	
275	21	$3/21$	56			40	44	
276	21	$3/21$	56			32	44	
277	21	$3/21$	56			24	44	
278	21	$3/21$	56	32	48	24		
279	27	$4/27$	24			32	24	44
280	49	$7/49$						
281	21	$3/21$	72	24	56	24		24
282	43	$6/43$	86	24	24	56		
283	21	$3/21$	56			24	24	44
284	21	$3/21$	56			32	24	44
285	21	$3/21$	56			40	24	44
286	21	$3/21$	56			48	24	44
287	21	$3/21$	24			24	24	44
288	21	$3/21$	28			32	24	44
289	21	$3/21$	56			72	24	44
290	29	$4/29$						
291	15	$2/15$	40			48	44	
292	21	$3/21$	28			48	24	44
293	15	$2/15$	48	32	40	56		
294	21	$3/21$	24			48	24	44
295	15	$2/15$	48			32	44	
296	37	$5/37$						
297	33	$4/33$	28	48	24	56		
298	21	$3/21$	28			72	24	44
299	23	$3/23$	24			24	56	
300	15	$2/15$						
301	43	$6/43$	24			48	24	44
302	16	$2/16$	32			72	24	
303	15	$2/15$	72	24	40	48		24
304	16	$2/16$	24			48	44	
305	15	$2/15$	48			32	24	44

TABLES FOR INDEXING.—(*Continued*)

Number of divisions	Index circle	Number of turns of index	Gear on worm	No. 1 hole		Gear on spindle	Idlers	
				First gear on stud	Second gear on stud		No. 1 hole	No. 2 hole
306	15	2/15	40			32	24	44
307	15	2/15	72	48	40	56		24
308	16	2/16	32			48	44	
309	15	2/15	40			48	24	44
310	31	4/31						
311	16	2/16	64	24	24	72		
312	39	5/39						
313	16	2/16	32			28	56	
314	16	2/16	32			24	56	
315	16	2/16	64			40	24	
316	16	2/16	64			32	44	
317	16	2/16	64			24	44	
318	16	2/16	56	28	48	24		
319	29	4/29	48	64	24	72		24
320	16	2/16						
321	16	2/16	72	24	64	24		24
322	23	3/23	32			64	24	44
323	16	2/16	64			24	24	44
324	16	2/16	64			32	24	44
325	16	2/16	64			40	24	44
326	16	2/16	32			24	24	44
327	16	2/16	32			28	24	44
328	41	5/41						
329	16	2/16	64	24	24	72		24
330	33	4/33						
331	16	2/16	64	44	24	48		24
332	16	2/16	32			48	24	44
333	18	2/18	24			72	44	
334	16	2/16	32			56	24	44
335	33	4/33	72	48	44	40		24
336	16	2/16	32			64	24	44
337	43	5/43	86	40	32	56		
338	16	2/16	32			72	24	44
339	18	2/18	24			56	44	
340	17	2/17						
341	43	5/43	86	24	32	40		
342	18	2/18	32			64	44	
343	15	2/15	40	64	24	86		24

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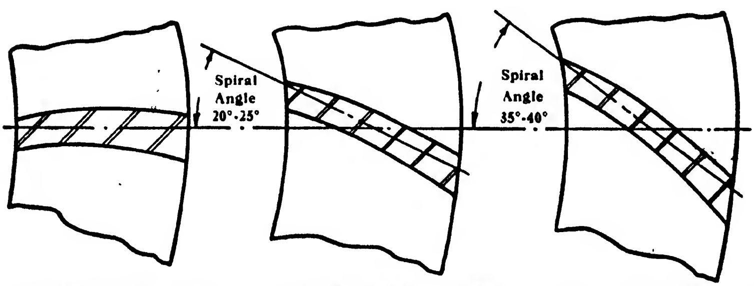
WARTIME DATA SUPPLEMENT

Bevel-gear Developments.—Recent developments in bevel-gear practice by the Gleason Works have added much to the applications and value of bevel gears. Among these the Zerol bevel gear, the Formate method of cutting spiral bevels, and the machines for grinding spiral bevel-gear teeth after hardening are outstanding.



Zerol bevel gears combine low axial thrust of straight bevel gears with localized tooth contact of curved-tooth spiral bevel gears.

Spiral bevel gears preferred for many industrial drives where provision can be made for moderate axial thrusts.

Spiral bevel gears for quietest operation at high speeds. Bearings must be designed for increased axial thrusts.

FIG. 1*A*.—Zerol bevel-gear tooth, at left, compared with spiral bevel-gear teeth of different angles.

The Zerol Gear.—The name *Zerol* is due to the fact that the curved teeth used have a zero or no spiral angle. This is shown in Fig. 1*A*, which gives, in outline, the Zerol or no-angle tooth and spiral bevels of two angles that are commonly used. They combine the localized tooth contact of spiral bevel gears and the low thrust loads of straight bevel teeth. They can be substituted for straight bevels without changing the thrust bearings.

One advantage of localized tooth contact is that loads are not concentrated on the ends of the teeth if slight misalignment should occur. And since no gear mounting is absolutely rigid, this is an important point in actual practice.

This form of gear also lends itself to grinding after hardening, as illustrated in Fig. 2*A*. Using a specially shaped cup wheel as shown, it is an easy matter to grind these teeth in the new Gleason gear-grinding machines. Figure 3A shows how this form of tooth is cut on a Gleason machine.

The Zerol gears can also be made with teeth having long or short addenda and with a large fillet radius for maximum strength of tooth.

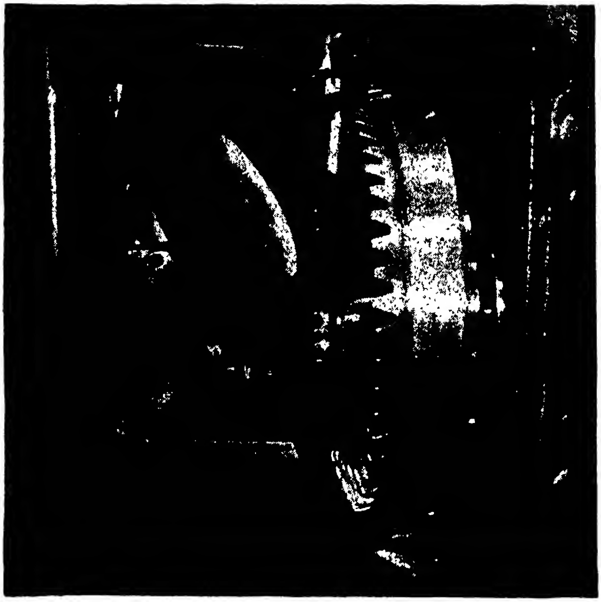


Fig. 2A.—Grinding a Zerol-tooth gear with a cap wheel.

The gears illustrated in Fig. 4A show how a spiral bevel tooth effect can be obtained by making a straight-toothed bevel gear in an infinite number of sections and arranging them in continuously advancing steps. This gives a gear with curved oblique teeth; if the sections are thin enough, this makes curved teeth as in the spiral bevel gear.

The Formate Gear Tooth.—This is a development of the original hypoid where a relative rolling motion of both the work and the cutter generates the correct tooth shape. The motion may be compared to that of a gear rolling with a crown gear, the cutter representing a tooth of the gear.

With the Formate gear tooth, however, only the cutter is in motion during the cutting of the tooth. Each succeeding blade of the cutter projects a little more than the one before so that each blade takes a light cut. A single revolution of the cutter finishes a tooth space. The teeth are not continuous all around the cutter; there is a gap which permits indexing, as in the cutter shown in Fig. 104 on page 183.

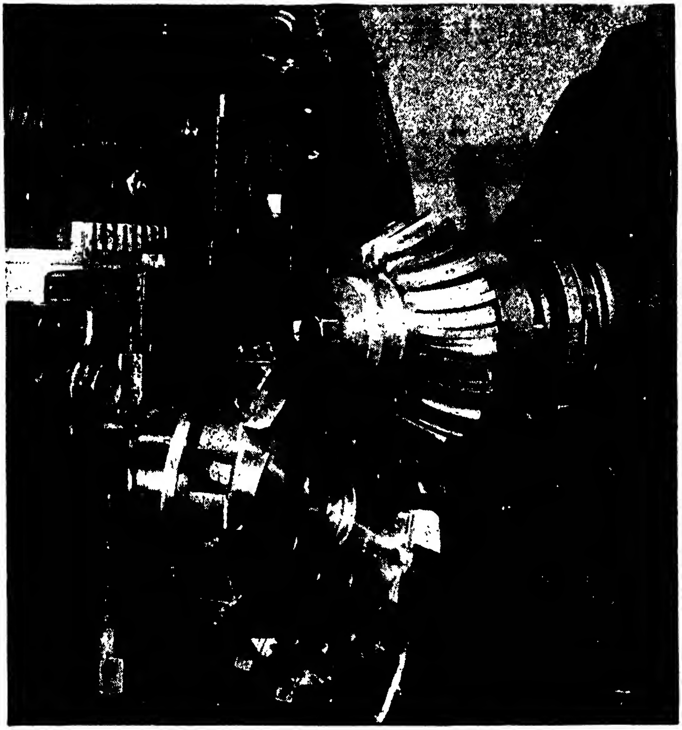


FIG. 3A.—Cutting a Zerol-tooth gear on a Gleason machine.

Gears cut with the blank stationary are called Formate because the teeth are formed by the cutters, without being generated, as when the gear blank rolls during the cut. These gears are perfectly satisfactory when used with a pinion whose teeth are generated to match the gear.

Grinding Hypoid Gear Teeth.—Teeth of hypoid gears are now being ground with the Hypoid Gear Grinder, which is one of the latest Gleason developments. It will grind generated pairs of gears within its range, or will grind generated pinions that have been cut to mate with gears ground by the Formate method.

The generating method differs from any formerly used, combining continuous rotation of the work spindle with a reciprocating motion of the cradle that carries the grinding wheel. This

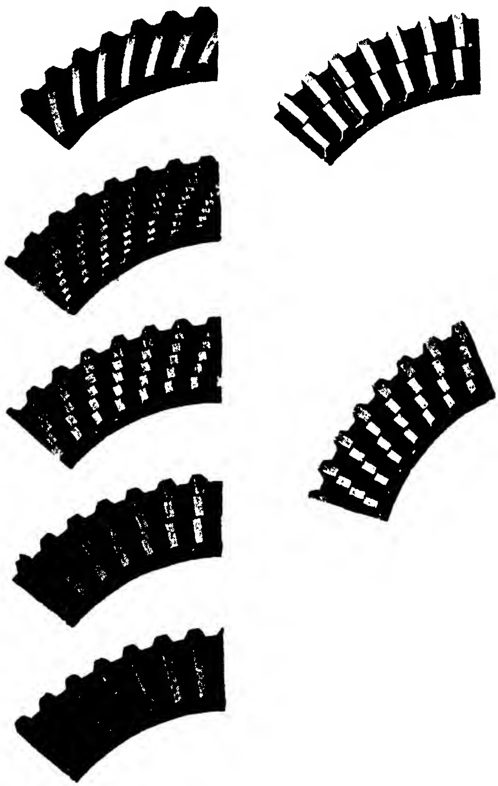


FIG. 4A.—How a number of sections of a straight-tooth bevel gear can be made to secure the action of a spiral bevel gear.

combination makes it possible to secure grinding speeds greater than those previously secured in a generating grinder.

When a gear is being ground, the wheel is fed into the work during the up-roll of the cradle and withdrawn during the down-roll. The work, however, rotates continuously, so that the grinding wheel starts work on the next tooth when the cradle starts on the up-roll.

Wheel-dressing diamonds are hydraulically controlled. They operate automatically and take into account the advance of the wheel and of the shaping of the side, the end, and the radius.

The diamonds are completely adjustable for feed, pressure angle, radius, and wheel diameter.

Tooth bearing is controlled by dressing the wheel to a modified shape and by modifying the generating roll. Change gears control the indexing as well as the speed. The feed and the movement of the generating roll are controlled by cams. The grinding wheel is of a special cup shape comparable with the face-mill type of cutter used on the hypoid generators.

Angle of Worm Threads.—There always seems to have been more or less confusion regarding the angle of a worm thread. This has caused some readers to question the formula on page 204 of this book. This formula is correct, however, and checks exactly with the A.G.M.A. formula on page 306.

The trouble or misunderstanding evidently arises from different interpretations of the term "thread angle." In the past this was usually considered as the angle between the thread and a line at a *right angle* to the axis of the worm. The present practice is to use the term "helix angle," which is the angle between the thread and the axis of the worm. In the old terminology a worm might be said to have a *thread angle of* 3 *deg.*, while present practice is to speak of a thread having an 87-*deg. helix angle.* It is the same thread in both cases.

Confusion is also caused by the use of both "helical" and "spiral" in referring to gears with teeth not parallel with the axis. In the United States we have adopted the term "helical gear" pretty generally, although the old term persists in some places. Engineers now agree on the terms "helical" and "helix" because they correctly describe the type of gear. If one remembers that a screw thread is a helix and that about the only spiral in use is a clock spring or its equivalent, the difference is readily understood.

Using the term "helical," we naturally, and correctly, refer to the "helix angle" of the gear tooth. The British, however, still use the term "spiral gear" and consequently refer to the angle as a "spiral angle." This difference still causes some confusion in correspondence between the two countries.

In helical gears the helix angle is usually less than 45 deg. In a single-thread worm, however, the helix angle may be as much as 85 deg., or even a little more. Calculations are frequently more conveniently based on the *complement* of the angle instead of using the larger value. The complement of 85 is 5, the difference between 85 and 90 deg., and the smaller figure is easier to handle.

In the example just mentioned, 5 deg. is the "lead angle." The *lead angle* is the angle between the thread and a plane of rotation perpendicular (or at right angles) to the axis. The British also use the term "lead angle" in the same way. The

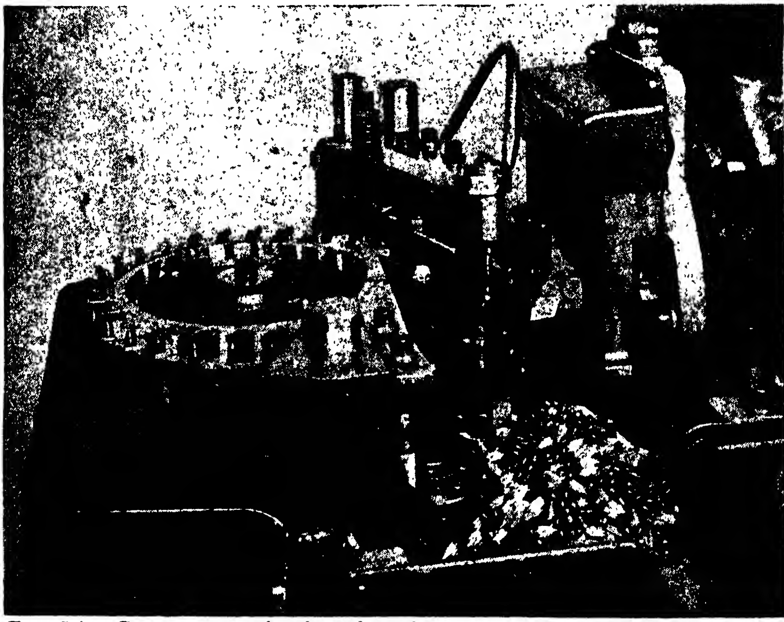


FIG. 5A.—Cutter, cutter head, and work of a Fellows gear shaper for small pinions. This shows the dial feed and the loading arm.

term "lead angle" is used more generally in connection with worm-thread specifications and practice.

In screw-thread practice, however, the helix angle is the angle between the thread and a plane perpendicular to the axis. Thread angle in this case is the included angle between the sides of the thread.

It is unfortunate that the same terms are not used for the same geometric values in both gear and thread practice.

Gear Shaper for Small Pinions.—The Fellows Gear Shaper Company has developed a new type of cutter and a new machine especially for small gears and pinions, such as are used by the clock, watch, and other fine-instrument works. It has a capacity up to $1\frac{1}{2}$ in. pitch diameter for spur gears and 1 in. for helical gears. Instead of using a circular cutter as in the other machines, this has a rack which moves past the gear blank as it turns, the rack being long enough to complete the gear. For small pinions

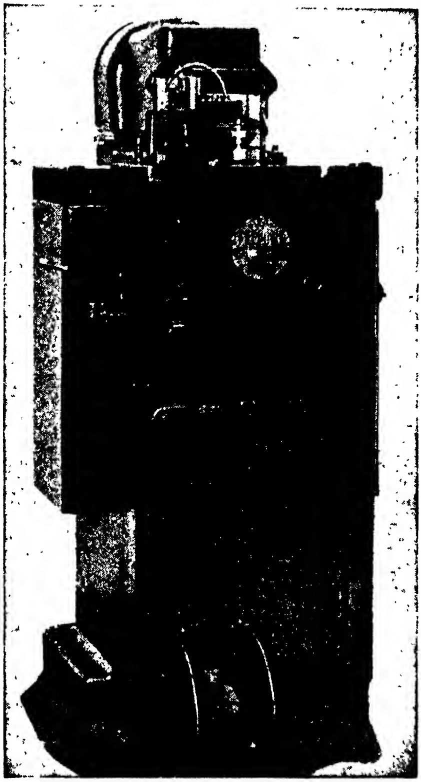


FIG. 6A.—The complete machine showing the treadle that raises the clamping support.

the cutter has several sets of teeth, each section being long enough to cover the number of teeth in the gear to be cut. This allows the use of one set of teeth after the other until all are dull, before the cutter needs to be removed for sharpening.

This rack-type cutter is held in an adjustable holder mounted on the reciprocating and relieving cutter slide. A swivel head behind the cutter slide permits angular adjustment so that the machine can be used for both spur and helical gears. Owing to the lightness of the reciprocating parts, speeds up to 1,500 strokes per minute can be secured in regular work. Figure 5*A* shows a close-up of the cutter, the cutter head, the work, and the way in which the gear blank is carried to the loading arm, which swings it into position as soon as the gear in position is finished and drops out of the way.

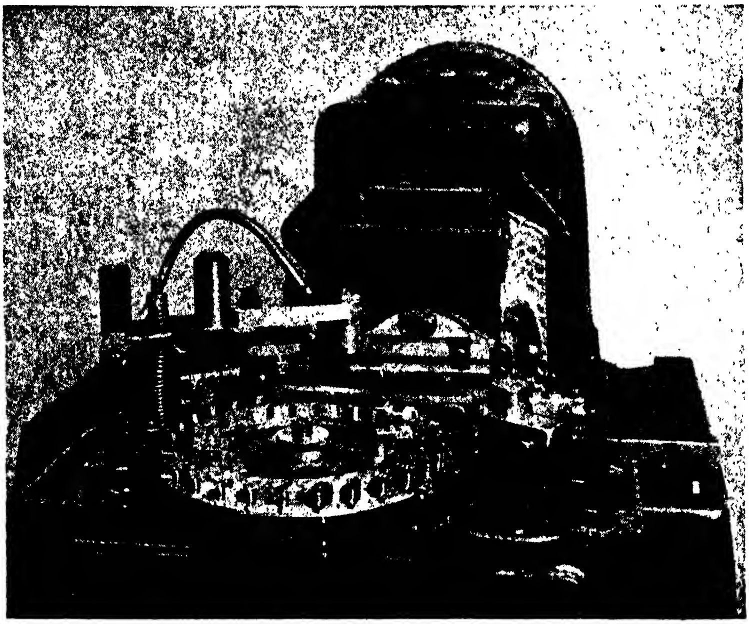


Fig. 7*A*.—Another view of the head with the loading arm.

This machine is designed, as previously mentioned, especially for small gears and pinions with fine teeth. As shown in Fig. 6*A*, it is compact and easily operated by the few controls shown. The clamping support is raised by the foot treadle, while the handle ejects the work after the gear teeth have been cut. The machine stops automatically but is easily started after reloading by a push-button control. An oil plug gives access to the reservoir, which should be filled with SAE 30 oil until it comes up to the center of the gage. The oil cup under the handle should also have the same treatment. A close-up of a front view of the work holder is given in Fig. 7*A*. Figure 8*A* shows two kinds of cutter used.

The following cutting speeds are recommended for various materials:

TABLE 1A.—APPROXIMATE CUTTING SPEEDS

Material	Surface Speed—Ft. per Min.
Steel, mild, 15 to 20 per cent carbon, no hardening alloys	65
Steel, alloy, 50 per cent carbon, 1.15 per cent manganese, 55 per cent chrome	50
Steel, SAE 3250	40
Steel, free cutting stainless	35
Brass, soft	100
Bronze, naval	50
Aluminum	200

Feeds vary widely with the number of cutting strokes per minute. These are listed in Table 2A, as based on a pitch diameter of 1 in. It should be remembered, however, that the feed

FIG. 8A.—Two of the rack type of cutter used.

is governed by the material, pitch, shape of tooth, and character of work. Fine-pitch gears cannot be cut accurately with a coarse feed on account of the delicate teeth in the cutter.

Neither speed nor feed can be arbitrarily stated; correct values for both can be determined only by trial. It is advisable to start with fairly low cutting speeds and feeds. The speed range is 600, 900, 1200, and 1500 strokes per minute, the highest being for such metals as aluminum and soft brass. Slow speeds should be used when the width of the gear face approaches the maximum stroke of the machine. Fine-pitch gears, such as 60-diametral pitch, should be started at slow speeds and feeds. The automatic loading fixture shown in Fig. 5*A* is advisable only on large runs. For lots of ordinary size, hand feeding is more economical.

TABLE 2*A*.—FELLOWS SMALL GEAR CUTTER; FEED PER STROKE OF CUTTER, BASED ON 1-IN. PITCH DIAMETER OF WORK*

Strokes of cutter per minute			
600	900	1200	1500
Feed per stroke in inches, based on 1-in. pitch diameter			
0.0315	0.0210	0.0157	0.0126
0.0229	0.0152	0.0114	0.0091
0.0182	0.0120	0.0091	0.0073
0.0135	0.0090	0.0068	0.0054
0.0108	0.0072	0.0054	0.0043
0.0079	0.0052	0.0039	0.0031
0.0055	0.0037	0.0028	0.0022
0.0036	0.0024	0.0018	0.0014

* To obtain the feed per stroke of cutter for other pitch diameters of work, multiply the feed per stroke given in the above table by the pitch diameter of the work.

Helical gears can also be cut on this machine by using the proper cutter, which is in reality a helical rack with teeth that cut.

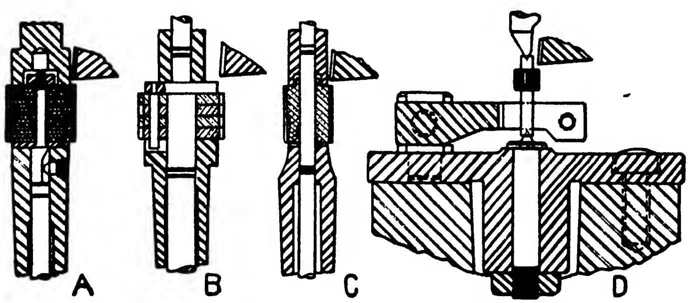


FIG. 9*A*.—Methods of holding pinion blanks.

Successful use of such a machine as this requires careful consideration of several factors such as: how can the blank be located and held while being cut; can more than one piece be cut at once; the quantity to be cut and the accuracy required.

Two methods of holding thin blanks are shown in Fig. 9*A*. At *A* the work mandrel has a V-slot in its shank which is held and driven by the cone-pointed screw. The blanks are held by a nut on top. With very thin blanks having small holes, necessitating a very slender mandrel, a clamp of some kind should be used rather than depend on a nut.

The blanks in *B* have a small hole beside the center hole. This permits the use of a pin in the small hole for driving the blanks. In *C* is a small pinion rather than a series of thin blanks. The blank is centered by a close-fitting pin and is

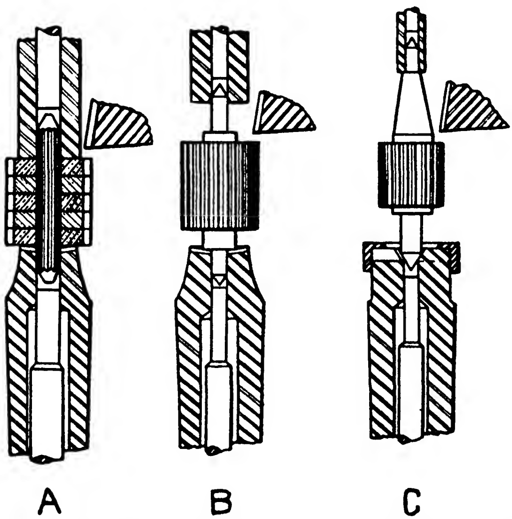


FIG. 10*A*. —Mounting five blanks in a mandrel for use with magazine feed.

clamped from above. The last example *D* is a pinion on a shaft which must be held between centers. Here a driving dog is clamped to the shank below the pinion. By using two dogs, one can be in use while the other is being loaded. In each case the cutting rack is shown just above the work.

Where the amount of work justifies the use of a magazine feed, the blanks are mounted on mandrels, as shown in Fig. 10*A*. At *A* five blanks are forced on a splined mandrel. At *B* the blank is driven by teeth at the lower end of the cylindrical part of the mandrel. But as no such surface is available at *C*, there are three tapered knife edges for driving from the lower cone. The coned ends assist in centering the work in the machine.

Fine-tooth Gears.—As very little information is available on gears with very fine teeth, Table 3*A*, by courtesy of the Fellows

TABLE 3A.—TOOTH PROPORTIONS FOR FINE-PITCH GEARS*

Diametral pitch	Dimension, in.				
	Circular thickness	Addendum	Dedendum	Whole depth	Double depth
30	0.0524	0.0333	0.0396	0.0729	0.1458
31	0.0507	0.0323	0.0383	0.0706	0.1412
32	0.0491	0.0313	0.0371	0.0684	0.1368
33	0.0476	0.0303	0.0364	0.0667	0.1334
34	0.0462	0.0294	0.0353	0.0647	0.1294
35	0.0449	0.0286	0.0347	0.0633	0.1266
36	0.0436	0.0278	0.0337	0.0615	0.1230
37	0.0425	0.0270	0.0331	0.0601	0.1202
38	0.0413	0.0263	0.0323	0.0586	0.1172
39	0.0403	0.0256	0.0314	0.0570	0.1140
40	0.0393	0.0250	0.0311	0.0561	0.1122
41	0.0383	0.0244	0.0303	0.0547	0.1094
42	0.0374	0.0238	0.0296	0.0534	0.1068
43	0.0365	0.0233	0.0293	0.0526	0.1052
44	0.0357	0.0227	0.0287	0.0514	0.1028
45	0.0349	0.0222	0.0280	0.0502	0.1004
46	0.0342	0.0217	0.0274	0.0491	0.0982
47	0.0334	0.0213	0.0273	0.0486	0.0972
48	0.0327	0.0208	0.0267	0.0475	0.0950
49	0.0321	0.0204	0.0262	0.0466	0.0932
50	0.0314	0.0200	0.0256	0.0456	0.0912
51	0.0308	0.0196	0.0255	0.0451	0.0902
52	0.0302	0.0192	0.0250	0.0442	0.0884
53	0.0296	0.0189	0.0245	0.0434	0.0868
54	0.0291	0.0185	0.0241	0.0426	0.0852
55	0.0286	0.0182	0.0240	0.0422	0.0844
56	0.0281	0.0179	0.0235	0.0414	0.0828
57	0.0276	0.0175	0.0232	0.0407	0.0814
58	0.0271	0.0172	0.0228	0.0400	0.0800
59	0.0266	0.0169	0.0227	0.0396	0.0792
60	0.0262	0.0167	0.0222	0.0389	0.0778
61	0.0258	0.0164	0.0219	0.0383	0.0766
62	0.0253	0.0161	0.0219	0.0380	0.0760
63	0.0249	0.0159	0.0215	0.0374	0.0748
64	0.0245	0.0156	0.0212	0.0368	0.0736

* Applies only to gears generated by Fellows rack-type cutters.

TABLE 3A.—TOOTH PROPORTIONS FOR FINE-PITCH GEARS.*—(Continued)

Diametral pitch	Dimension, in.				
	Circular thickness	Addendum	Dedendum	Whole depth	Double depth
65	0.0242	0.0154	0.0209	0.0363	0.0726
66	0.0238	0.0152	0.0205	0.0357	0.0714
67	0.0234	0.0149	0.0203	0.0352	0.0704
68	0.0231	0.0147	0.0203	0.0350	0.0700
69	0.0228	0.0145	0.0200	0.0345	0.0690
70	0.0224	0.0143	0.0197	0.0340	0.0680
71	0.0221	0.0141	0.0194	0.0335	0.0670
72	0.0218	0.0139	0.0194	0.0333	0.0666
73	0.0215	0.0137	0.0192	0.0329	0.0658
74	0.0212	0.0135	0.0189	0.0324	0.0648
75	0.0209	0.0133	0.0187	0.0320	0.0640
76	0.0207	0.0132	0.0184	0.0316	0.0632
77	0.0204	0.0130	0.0182	0.0312	0.0624
78	0.0201	0.0128	0.0182	0.0310	0.0620
79	0.0199	0.0127	0.0179	0.0306	0.0612
80	0.0196	0.0125	0.0177	0.0302	0.0604
81	0.0194	0.0123	0.0175	0.0298	0.0596
82	0.0192	0.0122	0.0173	0.0295	0.0590
83	0.0189	0.0120	0.0173	0.0293	0.0586
84	0.0187	0.0119	0.0171	0.0290	0.0580
85	0.0185	0.0118	0.0168	0.0286	0.0572
86	0.0183	0.0116	0.0167	0.0283	0.0566
87	0.0181	0.0115	0.0165	0.0280	0.0560
88	0.0179	0.0114	0.0165	0.0279	0.0558
89	0.0177	0.0112	0.0164	0.0276	0.0552
90	0.0175	0.0111	0.0162	0.0273	0.0546
91	0.0173	0.0110	0.0160	0.0270	0.0540
92	0.0171	0.0109	0.0158	0.0267	0.0534
93	0.0169	0.0108	0.0157	0.0265	0.0530
94	0.0167	0.0106	0.0157	0.0263	0.0526
95	0.0165	0.0105	0.0156	0.0261	0.0522
96	0.0164	0.0104	0.0154	0.0258	0.0516
97	0.0162	0.0103	0.0152	0.0255	0.0510
98	0.0160	0.0102	0.0151	0.0253	0.0506
99	0.0159	0.0101	0.0149	0.0250	0.0500

* Applies only to gears generated by Fellows rack-type cutters.

TABLE 3A.—TOOTH PROPORTIONS FOR FINE-PITCH GEARS.*—(*Continued*)

Diametral pitch	Dimension, in.				
	Circular thickness	Addendum	Dedendum	Whole depth	Double depth
100	0.0157	0.0100	0.0148	0.0248	0.0496
101	0.0156	0.0099	0.0148	0.0247	0.0494
102	0.0154	0.0098	0.0147	0.0245	0.0490
103	0.0153	0.0097	0.0145	0.0242	0.0484
104	0.0151	0.0096	0.0144	0.0240	0.0480
105	0.0149	0.0095	0.0143	0.0238	0.0476
106	0.0148	0.0094	0.0142	0.0236	0.0472
107	0.0147	0.0093	0.0140	0.0233	0.0466
108	0.0145	0.0093	0.0139	0.0231	0.0462
109	0.0144	0.0092	0.0139	0.0231	0.0462
110	0.0143	0.0091	0.0138	0.0229	0.0458
111	0.0142	0.0090	0.0137	0.0227	0.0454
112	0.0140	0.0089	0.0136	0.0225	0.0450
113	0.0139	0.0088	0.0135	0.0223	0.0446
114	0.0138	0.0088	0.0133	0.0221	0.0442
115	0.0137	0.0087	0.0132	0.0219	0.0438
116	0.0135	0.0086	0.0132	0.0218	0.0436
117	0.0134	0.0085	0.0132	0.0217	0.0434
118	0.0133	0.0085	0.0130	0.0215	0.0430
119	0.0132	0.0084	0.0129	0.0213	0.0426
120	0.0131	0.0083	0.0129	0.0212	0.0424
121	0.0130	0.0083	0.0127	0.0210	0.0420
122	0.0129	0.0082	0.0126	0.0208	0.0416
123	0.0128	0.0081	0.0126	0.0207	0.0414
124	0.0127	0.0081	0.0124	0.0205	0.0410
125	0.0126	0.0080	0.0124	0.0204	0.0408
126	0.0125	0.0079	0.0124	0.0203	0.0406
127	0.0124	0.0079	0.0123	0.0202	0.0404
128	0.0123	0.0078	0.0122	0.0200	0.0400
129	0.0122	0.0078	0.0120	0.0198	0.0396
130	0.0121	0.0077	0.0120	0.0197	0.0394
131	0.0120	0.0076	0.0119	0.0195	0.0390
132	0.0119	0.0076	0.0118	0.0194	0.0388
133	0.0118	0.0075	0.0117	0.0192	0.0384
134	0.0117	0.0075	0.0116	0.0191	0.0382

* Applies only to gears generated by Fellows rack-type cutters.

TABLE 3A.—TOOTH PROPORTIONS FOR FINE-PITCH GEARS.*—(*Continued*)

Diametral pitch	Dimension, in.				
	Circular thickness	Addendum	Dedendum	Whole depth	Double depth
135	0.0116	0.0074	0.0116	0.0190	0.0380
136	0.0116	0.0074	0.0115	0.0189	0.0378
137	0.0115	0.0073	0.0115	0.0188	0.0376
138	0.0114	0.0073	0.0114	0.0187	0.0374
139	0.0113	0.0072	0.0113	0.0185	0.0370
140	0.0112	0.0071	0.0113	0.0184	0.0368
141	0.0111	0.0071	0.0112	0.0183	0.0366
142	0.0111	0.0070	0.0111	0.0181	0.0362
143	0.0110	0.0070	0.0110	0.0180	0.0360
144	0.0109	0.0069	0.0110	0.0179	0.0358
145	0.0108	0.0069	0.0109	0.0178	0.0356
146	0.0108	0.0068	0.0109	0.0177	0.0354
147	0.0107	0.0068	0.0108	0.0176	0.0352
148	0.0106	0.0068	0.0107	0.0175	0.0350
149	0.0105	0.0067	0.0107	0.0174	0.0348
150	0.0105	0.0067	0.0106	0.0173	0.0346
151	0.0104	0.0066	0.0105	0.0171	0.0342
152	0.0103	0.0066	0.0104	0.0170	0.0340
153	0.0103	0.0065	0.0104	0.0169	0.0338
154	0.0102	0.0065	0.0103	0.0168	0.0336
155	0.0101	0.0064	0.0103	0.0167	0.0334
156	0.0101	0.0064	0.0103	0.0167	0.0334
157	0.0100	0.0064	0.0102	0.0166	0.0332
158	0.0099	0.0063	0.0102	0.0165	0.0330
159	0.0099	0.0063	0.0101	0.0164	0.0328
160	0.0098	0.0063	0.0100	0.0163	0.0326
161	0.0098	0.0062	0.0099	0.0161	0.0322
162	0.0097	0.0062	0.0098	0.0160	0.0320
163	0.0096	0.0061	0.0098	0.0159	0.0318
164	0.0096	0.0061	0.0098	0.0159	0.0318
165	0.0095	0.0061	0.0097	0.0158	0.0316
166	0.0095	0.0060	0.0097	0.0157	0.0314
167	0.0094	0.0060	0.0096	0.0156	0.0312
168	0.0094	0.0060	0.0095	0.0155	0.0310
169	0.0093	0.0059	0.0095	0.0154	0.0308

* Applies only to gears generated by Fellows rack-type cutters.

TABLE 3A.—TOOTH PROPORTIONS FOR FINE-PITCH GEARS.*—(Continued)

Diametral pitch	Dimension, in.				
	Circular thickness	Addendum	Dedendum	Whole depth	Double depth
170	0.0092	0.0059	0.0094	0.0153	0.0306
171	0.0092	0.0058	0.0094	0.0152	0.0304
172	0.0091	0.0058	0.0093	0.0151	0.0302
173	0.0091	0.0058	0.0093	0.0151	0.0302
174	0.0090	0.0057	0.0093	0.0150	0.0300
175	0.0090	0.0057	0.0092	0.0149	0.0298
176	0.0089	0.0057	0.0091	0.0148	0.0296
177	0.0089	0.0056	0.0091	0.0147	0.0294
178	0.0088	0.0056	0.0090	0.0146	0.0292
179	0.0088	0.0056	0.0090	0.0146	0.0292
180	0.0087	0.0056	0.0089	0.0145	0.0290
181	0.0087	0.0055	0.0089	0.0144	0.0288
182	0.0086	0.0055	0.0088	0.0143	0.0286
183	0.0086	0.0055	0.0087	0.0142	0.0284
184	0.0085	0.0054	0.0087	0.0141	0.0282
185	0.0085	0.0054	0.0087	0.0141	0.0282
186	0.0085	0.0054	0.0086	0.0140	0.0280
187	0.0084	0.0053	0.0086	0.0139	0.0278
188	0.0084	0.0053	0.0086	0.0139	0.0278
189	0.0083	0.0053	0.0085	0.0138	0.0276
190	0.0083	0.0053	0.0084	0.0137	0.0274
191	0.0082	0.0052	0.0084	0.0136	0.0272
192	0.0082	0.0052	0.0084	0.0136	0.0272
193	0.0081	0.0052	0.0083	0.0135	0.0270
194	0.0081	0.0052	0.0082	0.0134	0.0268
195	0.0081	0.0051	0.0082	0.0133	0.0266
196	0.0080	0.0051	0.0082	0.0133	0.0266
197	0.0080	0.0051	0.0081	0.0132	0.0264
198	0.0079	0.0051	0.0081	0.0132	0.0264
199	0.0079	0.0050	0.0081	0.0131	0.0262
200	0.0079	0.0050	0.0080	0.0130	0.0260

* Applies only to gears generated by Fellows rack-type cutters.

Gear Shaper Company, will be found useful in shops making instruments of various kinds. This table includes gears of from 30 to 200 diametral pitch, which covers all sizes likely to be needed. It must be noted that these values apply only to gears generated by Fellows rack-type cutters.

Flame Hardening of Gear Teeth.—While the hardening of gear teeth by the use of oxyacetylene flame is not new, its

FIG. 11*A*.—Fellows flame-hardening machine.

development in recent years now makes it an important factor where gear life is concerned. There are several gear-hardening machines now available, among them being the one by the Fellows Gear Shaper Company, and the one by the Gleason Works.

The Fellows machine spins the gears while numerous flame jets heat the teeth. The speed of rotation is approximately 100 surface feet per minute for average work. Figure 11*A* shows a cluster gear in position with four torches around it. The machine has six jets but only four are being used in this case. The machine ejects and quenches the work automatically at any set time. The flame equipment is made by the Linde Air

Products Company. A diagram of the arrangement of the jets is shown in Fig. 12*A*. Table 4*A* gives data as to the number of torches, the flame tips per torch, the total number of flame tips, and the approximate heating time for gear of various pitches.

Table 5*A* gives hardening temperatures for various steels and the hardness produced by quenching in both oil and water.

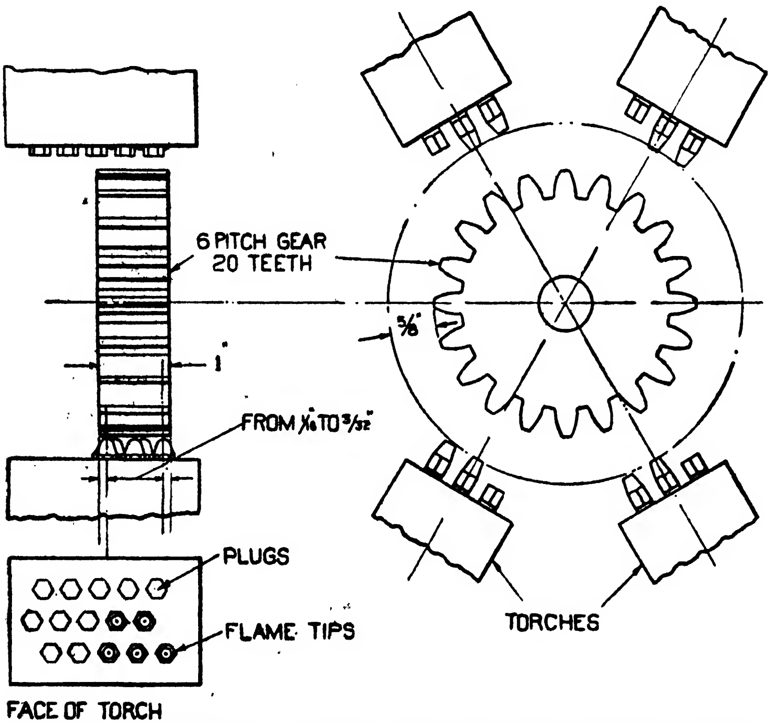


FIG. 12*A*.—Arrangement of torches on the Fellows flame-hardening machine.

Any steel that can be hardened in a furnace can be flame-hardened. This means that the carbon content should not be below 0.30 and preferably should be between 0.35 and 0.70 per cent carbon. Alloy steels, such as the SAE steels, can be flame-hardened. It is also possible to harden malleable and cast irons of certain compositions. Data as to the grades that can be hardened are given by the Fellows Company. Flame-hardening gives a high resistance to wear, the same as that of chilled cast iron.

TABLE 4A.—NUMBER OF TORCHES AND FLAME TIPS USED AND APPROXIMATE HEATING TIME IN SECONDS

Number of teeth	Number of torches	Flame tips per torch	Total number of flame tips	Approximate heating time, sec.
6 pitch; 1-in. face width				
10	2	5	10	10
14	3	5	15	14
20	4	5	20	20
25	4	7	28	25
30	4	7	28	30
35	5	7	35	35
40	6	7	42	40
45	6	7	42	51
50	6	7	42	63
55	6	7	42	72
60	6	7	42	90
8 pitch; 1-in. face width				
10	2	5	10	8.6
14	2	5	10	12.0
22	3	5	15	15.0
25	3	5	15	17.0
30	4	5	20	25.7
35	4	5	20	30.0
40	4	5	20	34.5
46	6	7	42	38.0
50	6	7	42	40.0
55	6	7	42	41.0
61	6	7.2	43	42.0
65	6	7	42	45.0
70	6	7	42	50.0
75	6	7	42	60.0
78	6	7.3	44	68.0
80	6	7.5	45	70.0

TABLE 4A.—NUMBER OF TORCHES AND FLAME TIPS USED AND APPROXIMATE HEATING TIME IN SECONDS.—(*Continued*)

Number of teeth	Number of torches	Flame tips per torch	Total number of flame tips	Approximate heating time, sec.
10 pitch; 1-in. face width				
10	2	5	10	5.0
14	2	5	10	7.0
20	3	5	15	10.0
25	3	5	15	12.5
30	4	5	20	15.0
35	4	5	20	17.5
40	4	7	28	20.0
45	4	7	28	22.5
50	5	7	35	25.0
55	5	7	35	27.5
60	6	7	42	30.0
65	6	7	42	34 5
70	6	7	42	41.5
75	6	7	42	47.5
83	6	7	42	58.0
85	6	7	42	60.0
90	6	7	42	68.0
95	6	7	42	78.0
100	6	7	42	85.0
12 pitch; 1-in. face width				
10	2	5	10	5.0
15	2	5	10	7.0
20	2	5	10	8.0
25	2	5	10	10.0
30	3	5	15	12.0
35	3	5	15	14.0
40	4	5	20	16.0
45	4	5	20	18.0
50	5	5	25	20.0
55	6	5	30	22.0

TABLE 4A.—NUMBER OF TORCHES AND FLAME TIPS USED AND APPROXIMATE HEATING TIME IN SECONDS.—(*Continued*)

Number of teeth	Number of torches	Flame tips per torch	Total number of flame tips	Approximate heating time, sec.
12 pitch; 1-in. face width—(*Continued*)				
60	6	5	30	24.0
65	6	7	42	20.0
70	6	7	42	22 0
75	6	7	42	24.0
80	6	7	42	26.0
85	6	7	42	28.0
90	6	7	42	30.0
95	6	7	42	32.0
100	6	7	42	34.0
120	6	7	42	36.0
14 pitch; 1-in. face width				
20	2	5	10	5.0
25	2	5	10	6.0
30	2	5	10	7.5
35	3	5	15	9.0
40	3	6	18	10.0
45	4	5	20	11.0
50	4	5	20	12.5
55	4	5	20	13 7
60	4	5	20	15.0
70	4	5	20	17.5
80	4	5	20	20.0
90	4	5	20	22.5
100	6	7	42	25.0
110	6	7	42	27.5
120	6	7	42	30.0
130	6	7	42	32.5
140	6	7	42	35.0

TABLE 5A.—HARDENING TEMPERATURES, QUENCHING MEDIA, AND RESULTANT HARDNESS NUMBERS FOR FERROUS MATERIALS

Material	Hardening range, deg. F.	Hardness numbers					
		Oil quenched			Water quenched		
		Scleroscope	Rockwell C	Brinell	Scleroscope	Rockwell C	Brinell
SAE 1035	1525–1600	.....	.....	....	50–70	35–50	350–500
SAE 1040	1450–1550	.....	.....	....	60–75	45–55	450–550
SAEX1040	1450–1550	.....	.....	....	60–75	45–55	450–550
SAE 1045	1450–1550	.....	.....	....	70–90	50–65	500–700
SAEX1045	1450–1550	.....	....	....	70–90	50–65	500–700
SAE 1050	1450–1550	.....	.....	....	70–90	50–65	500–700
SAEX1050	1450–1550	.....	.....	....	70–90	50–65	500–700
SAE 1055	1450–1550	.....	.....	....	70–90	50–65	500–700
SAEX1055	1450–1550	.....	.....	....	70–90	50–65	500–700
SAE 1060	1450–1550	55–70	40–50	400–500	75–90	55–65	500–700
SAE 1095	1400–1525	55–70	40–50	400–500	75–90	55–65	500–700
SAET1330	1500–1550	.....	.....	....	75–90	55–65	500–700
SAET1335	1500–1550	.....	.....	....	75–90	55–65	550–700
SAET1340	1425–1475	55–75	45–55	450–550	75–90	55–65	550–700
SAET1345	1425–1475	55–75	45–55	450–550	75–90	55–65	550–700
SAE 2330	1425–1475	55–70	40–50	400–500			
SAE 2335	1425–1475	55–70	40–50	400–500			
SAE 2340	1425–1475	55–70	40–50	400–500			
SAE 2345	1425–1475	55–70	40–50	400–500			
SAE 2350	1425–1475	55–70	40–50	400–500			
SAE 3140	1500–1550	55–70	40–50	400–500	70–80	50–60	500–600
SAE 3145	1500–1550	55–70	40–50	400–500	70–80	50–60	500–600
SAE 3230	1500–1550	55–70	40–50	400–500	70–80	50–60	500–600
SAE 3240	1500–1550	55–70	40–50	400–500	70–80	50–60	500–600
SAE 3335	1425–1500	60–70	45–55	450–550	70–80	50–60	500–600
SAE 3340	1425–1500	60–70	45–55	450–550	70–80	50–60	500–600
SAE 3435	1425–1500	60–70	45–55	450–550	70–80	50–60	500–600
SAE 5140	1450–1600	.....	.....	....	70–85	55–60	550–650
SAE 52100	1450–1550	70–80	50–60	500–600	80–90	60–65	650–750
SAE 6135	1550–1650	.....	.....	....	75–85	55–60	550–650
SAE 6140	1550–1650	.....	.....	....	75–85	55–60	550–650
SAE 4130	1500–1600	55–75	40–55	400–550	70–85	50–60	500–650
SAEX4130	1500–1600	55–75	40–55	400–550	70–85	50–60	500–650
SAE 4135	1500–1600	55–75	40–55	400–550	70–85	50–60	500–650
SAE 4140	1500–1600	55–75	40–55	400–550	70–85	50–60	500–650
SAE 4640	1450–1550	60–70	42–50	400–500	70–80	50–60	500–600

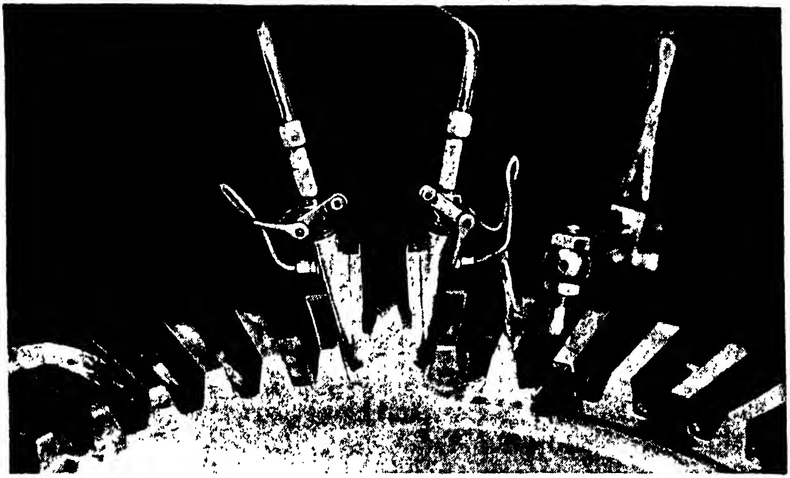


FIG. 13A.—Gleason hardening jets and indexing pin for large bevel gears.

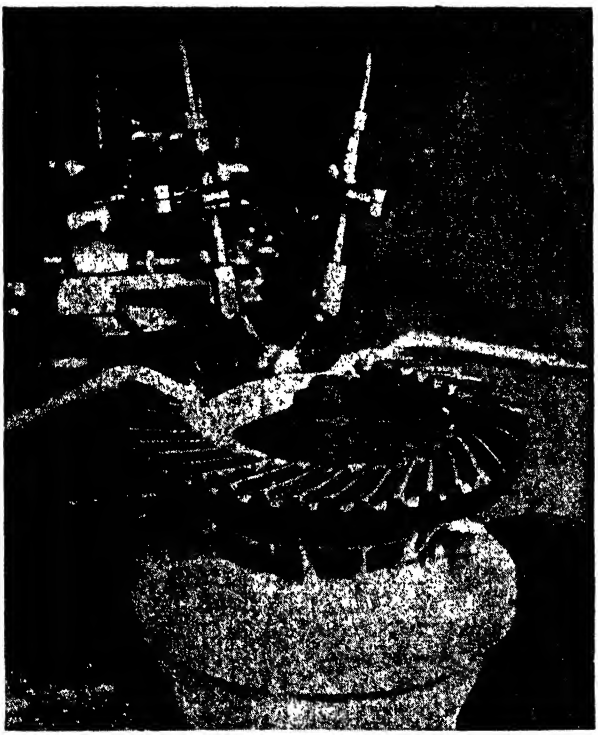


FIG. 14A.—Smaller gear-hardening machine, also by Gleason.

The surface to be flame-hardened must be free from scale, blowholes, pits, seams, laps, and other defects, since these tend to develop cracks during the quenching. Flame-hardening produces a clean surface. The tips should not be too close to the work on account of the intense heat of the jet, which is about 6300°F. $\frac{1}{16}$ in. from the jet. The hardening range rarely exceeds 1600°F.

The core can be given any heat-treatment desired before the teeth are flame-hardened, and this treatment is not affected by the flame-hardening of the teeth.

The Gleason Methods.—The Gleason Works has developed two machines for the surface hardening of gear teeth. They handle straight bevel gears, spiral bevel gears, and gears with Zerol teeth and hypoid teeth. They can also harden spur, helical, and internal gears up to 120 in. in diameter. Gears as small as 2 in. pitch diameter and 16 diametral pitch have also been hardened on these machines.

A close-up of the hardening jets and the indexing stop is shown in Fig. 13*A*, and part of a machine at work on a smaller spiral bevel gear is seen in Fig. 14*A*. These machines have mechanical control features which make possible the surface-hardening of gear teeth with great uniformity. This includes mechanical speed-up over the ends of the teeth to maintain the proper heat balance and reversal of the travel of the burner slide at a fast rate of speed to preheat the next tooth. Each tooth is hardened to the same depth, and both sides are hardened at the same time.

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