

BACK TO BASICS...

Generating and Checking Involute Gear Teeth

Fellows Corp.
Springfield, VT

It has previously been demonstrated that one gear of an interchangeable series will rotate with another gear of the same series with proper tooth action. It is, therefore, evident that a tooth curve driven in unison with a mating blank, will "generate" in the latter the proper tooth curve to mesh with itself. Similarly, a gear, which is made up of a series of tooth curves, is capable of "generating" in a blank a corresponding series of tooth curves, suitable for meshing with itself. This method of tooth "generation" is known as the "molding-generating" process, which will be more fully explained in the following pages.

The "Molding-Generating" Process

It is, therefore, evident that if one gear is provided with suitable cutting clearances on the teeth and is hardened, it can be used as a generating tool. If this tool is rotated in the correct ratio with a gear blank, and at the same time is reciprocated, it will generate teeth in the blank suitable for meshing with itself, or with another gear of a corresponding series. This, basically, is the principle upon which the Gear Shaper operates, as is indicated diagrammatically in Fig. 1.

The Gear Shaper Cutter

Fig. 2 shows a Gear Shaper cutter designed for cutting spur gear teeth. It will be noted that cutting clearances are provided both on the ends and sides of the teeth. It has the appearance of, but is not a bevel gear; the reason being that the sides of the teeth are developed, or ground from a com-

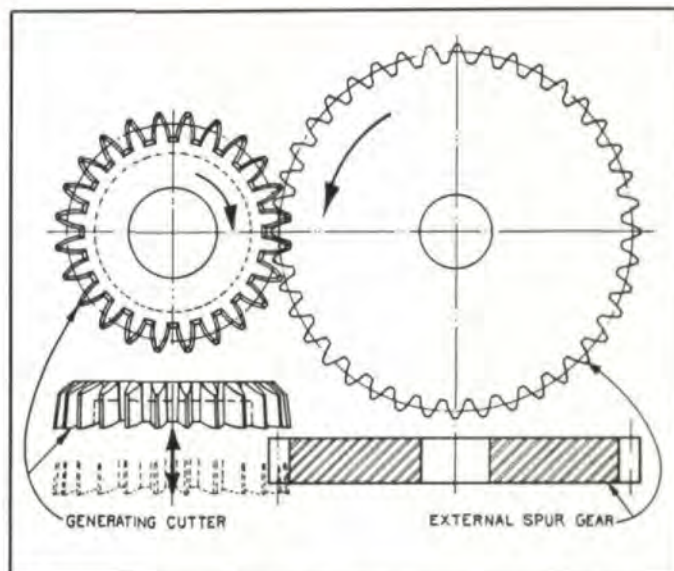
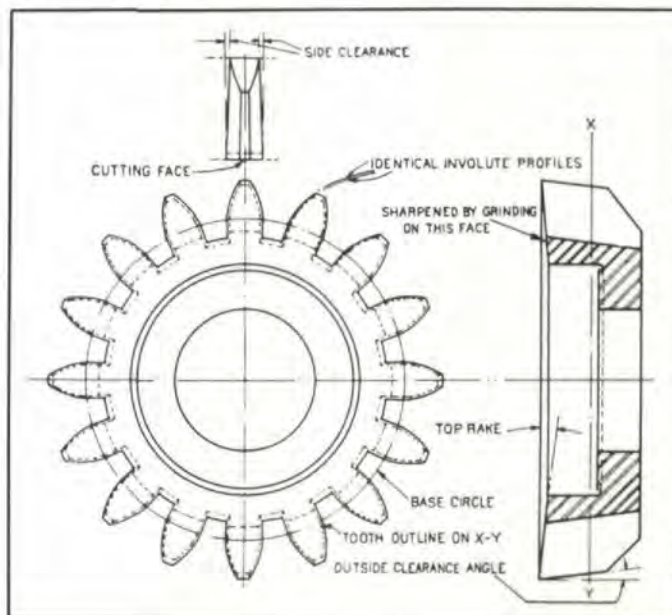


Fig. 1—Diagram Illustrating That the Gear Shaper Employs the "Molding-Generating" Process for Cutting Gears.

mon base circle, so that the involute profile extends along the whole length of the tooth from the front face to the back of the cutter. The teeth are simply thinner at the back than at the front or cutting face. They are shortened to correspond with the reduction in thickness, by beveling the outside diameter of the cutter. This enables the cutter to cut the required width of tooth space as the cutter teeth are reduced in thickness by repeated resharpening. As the cutter is ground

Fig. 2—The Gear Shaper Cutter for Cutting Spur Gears, Showing That It Has Cutting Clearances, but Does Not Change in Shape as It Is Ground Back by Re-sharpening.



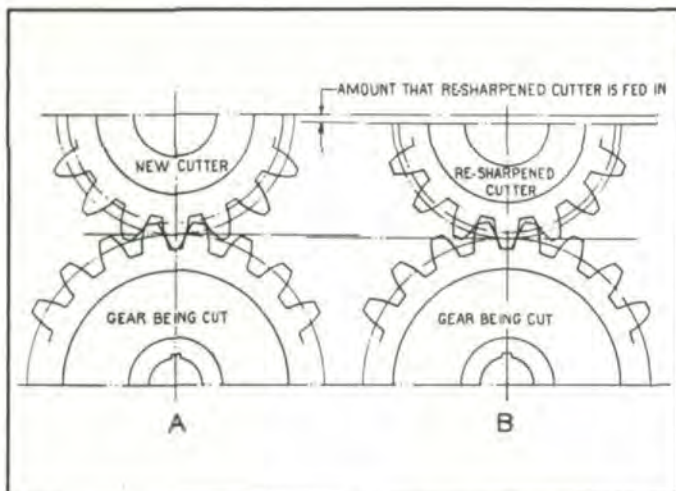


Fig. 3—Diagram Showing a New Gear Shaper Cutter in Mesh with a Gear, and the Same Cutter after Re-sharpening in Correct Mesh with the Same Gear.

back, thus thinning the teeth, the center distance between cutter and work is decreased.

This condition is illustrated in Fig. 3. Here at A a "new" cutter is shown in mesh with a gear, the teeth of which have been generated by the cutter. At B, the same cutter, is shown in mesh with the same gear, but the teeth on the cutter have been reduced in thickness by repeated resharpening. The only change in these two illustrations is in the center distance of cutter and gear. Since the base circle of the cutter has not been changed by reducing the thickness of the teeth, it will, of course, produce teeth of the same shape as before.

Generating Flank and Fillet

That portion of a gear tooth lying inside the base circle from which the involute is developed is of non-involute shape. How this shape is produced by the Gear Shaper cutter is shown in Fig. 4. In the tooth space to the right of the illustration are shown the successive positions taken by the cutter tooth as it "rolls" into the tooth space. The cutter teeth, of course, are longer than the gear teeth, so as to provide the necessary clearance at the bottom of the tooth spaces for the

Fig. 4—Diagram Illustrating the Generating Action of the Fellows Gear Shaper Cutter, Showing How the Involute Portion, Flank and Fillet of the Teeth Are Produced.

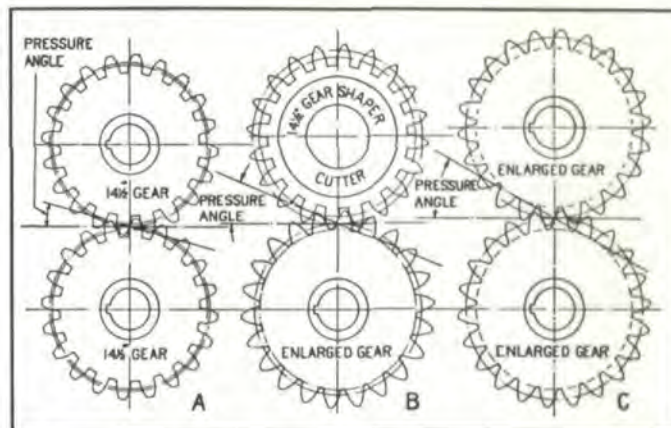
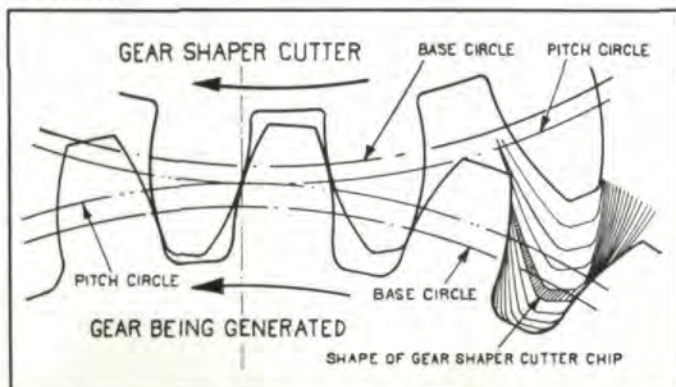


Fig. 5—Diagram Illustrating Two "Standard" Gears in Mesh. A Cutter in Mesh with an Oversize Gear, and Two Oversize Gears in Mesh Which Have Been Cut with the Same Cutter.

mating gear. This illustration also indicates the nature of the chip taken by the Gear Shaper cutter, the heaviest portion of the chip, after reaching full depth, being in the flank and fillet, and the lightest chip on the involute portion of the tooth, thus assuring a fine finish.

Cutting Different Pressure Angles With the Same Cutter

The pitch circles and pressure angle are variable quantities depending on variations in the center distance. This is illustrated diagrammatically in Fig. 5. At A are shown two 20-tooth "standard" $14\frac{1}{2}^\circ$ pressure angle gears, in mesh with each other at "standard" center distance. Both gears have been cut with the same Gear Shaper cutter, and, therefore, have the same tooth profiles.

At B is shown a $14\frac{1}{2}^\circ$ pressure angle Gear Shaper cutter in mesh with a 20-tooth gear. The size of the blank, however, has been enlarged and is suitable for a 22, instead of a 20-tooth gear. It will be noted that there is no change in the base circle diameter of this gear, but the center distance between cutter and gear has been increased over that shown at A. Therefore, the generating pressure angle between gear and cutter has been increased, and a new pitch point has located the pitch circle of the gear farther out towards the ends of the cutter teeth.

At C, two of these enlarged gears are shown in mesh with each other. The base circles are the same as before, but the center distance has been increased, thus establishing a new and greater operating pressure angle. Now these two gears were cut with the same cutter and, hence, have the same tooth profiles. All of the other elements—pitch circles, pressure angle, etc., have changed due to the increase in the center distance.

Note, however, that the base circle once established never changes. The diameter of the base circle of the cutter is in the same ratio to the diameter of the base circle of the gear cut, as the number of teeth in the cutter is to the number of teeth in the gear. Hence, the base circle will remain fixed, no matter how the pitch diameter, pressure angle, outside diameter, etc., vary.

The Gear Shaper Cutter and Interference

Interference in involute gearing can be corrected, in cut-

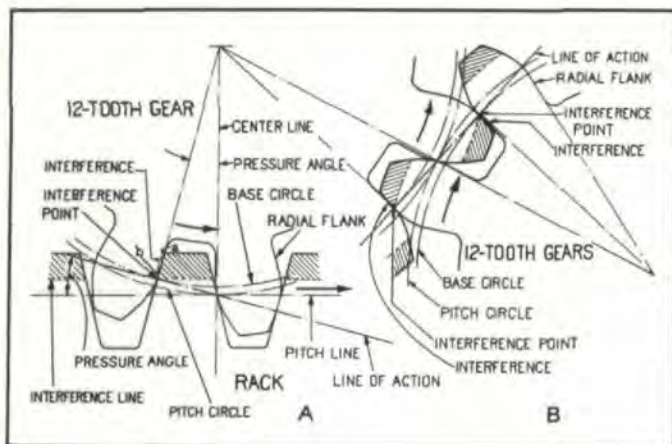


Fig. 6—Diagram Showing How an Unmodified Rack Tooth Interferes with a 12-Tooth Pinion; Also Interference between Two Unmodified 12-Tooth Pinions.

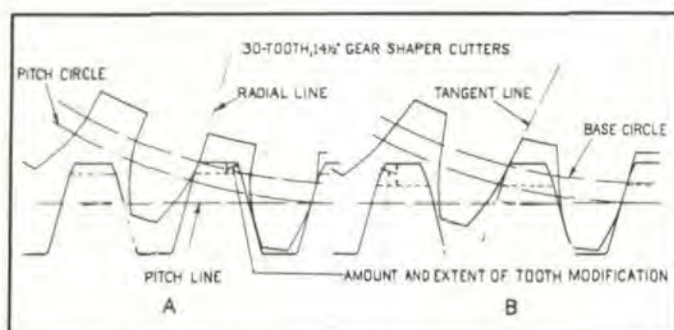


Fig. 7—Diagram Illustrating How the Gear Shaper Cutter Can Be Made to Modify a Rack Tooth so as to Avoid Interference with a Mating Gear.

ting, by the Gear Shaper cutter. At A in Fig. 6 a 12-tooth $14\frac{1}{2}^\circ$ degree pressure angle pinion is shown in mesh with a straight-sided rack tooth. It will be noted that the side of the rack tooth cuts into, or overlaps, the radial flank of the pinion tooth, at the point marked "interference." The "path" of the point of the rack tooth is indicated by the line *a-b*. In this particular case, the interference is so great that it extends almost to the pitch circle of the pinion. This interference condition can be avoided by modifying the rack tooth, or by undercutting the flank of the pinion tooth.

This interference is due to the fact that the teeth of a $14\frac{1}{2}^\circ$ full-depth rack tooth are too long to permit proper conjugate action, without tooth modification, with a 12-tooth pinion of the same system. The line of contact on the pinion tooth terminates, of course, at the point where it is tangent to the base circle. This tangent point is the "natural" interference point, and no portion of the rack tooth, which extends beyond this point, unless modified, can have proper action with a radial flank 12-tooth pinion.

At B, in Fig. 6 two 12-tooth, $14\frac{1}{2}^\circ$ full-depth radial-flank pinions are shown in mesh. Here it will be noted that the tooth profiles on both pinions would have to be modified, or the flanks of the pinion teeth undercut, in order to eliminate interference and provide proper tooth action. The extent of the interference in the illustrations at A and B is indicated by the cross-hatched portions of the teeth.

How the Gear Shaper Cutter Removes Interference

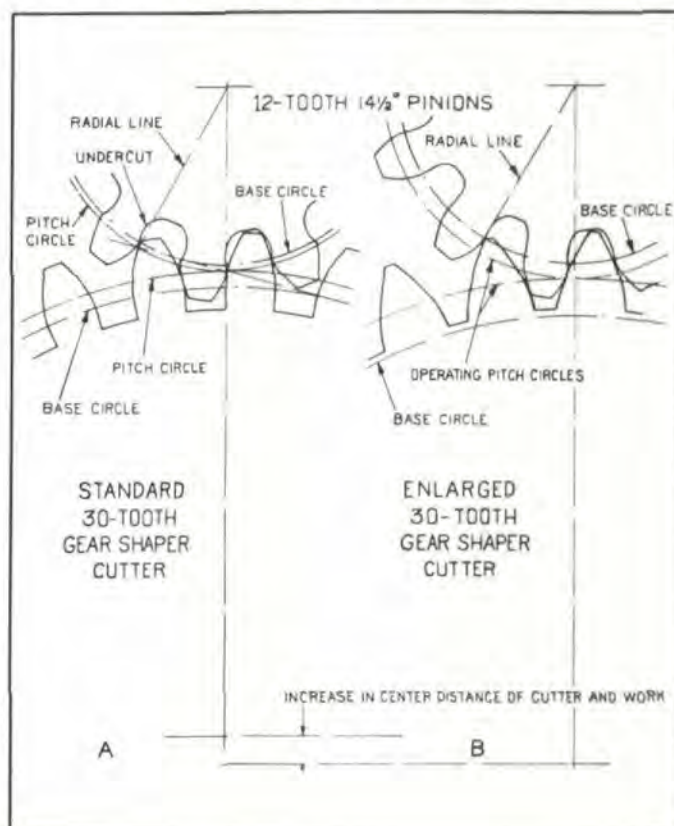
The Gear Shaper cutter, being a cutting tool in the form of a gear, can be so made that it will remove those portions of the gear or rack teeth that interfere with each other. At A in Fig. 7 is shown a 30-tooth $14\frac{1}{2}^\circ$ Gear Shaper cutter in contact with a rack tooth. The flank of this cutter is made radial, and as indicated modifies a portion of the rack tooth above the pitch line. At B the flank of the Gear Shaper cutter is "filled in" still more than at A and consequently the amount and extent of modification is greater. Gear teeth can be modified in a similar manner.

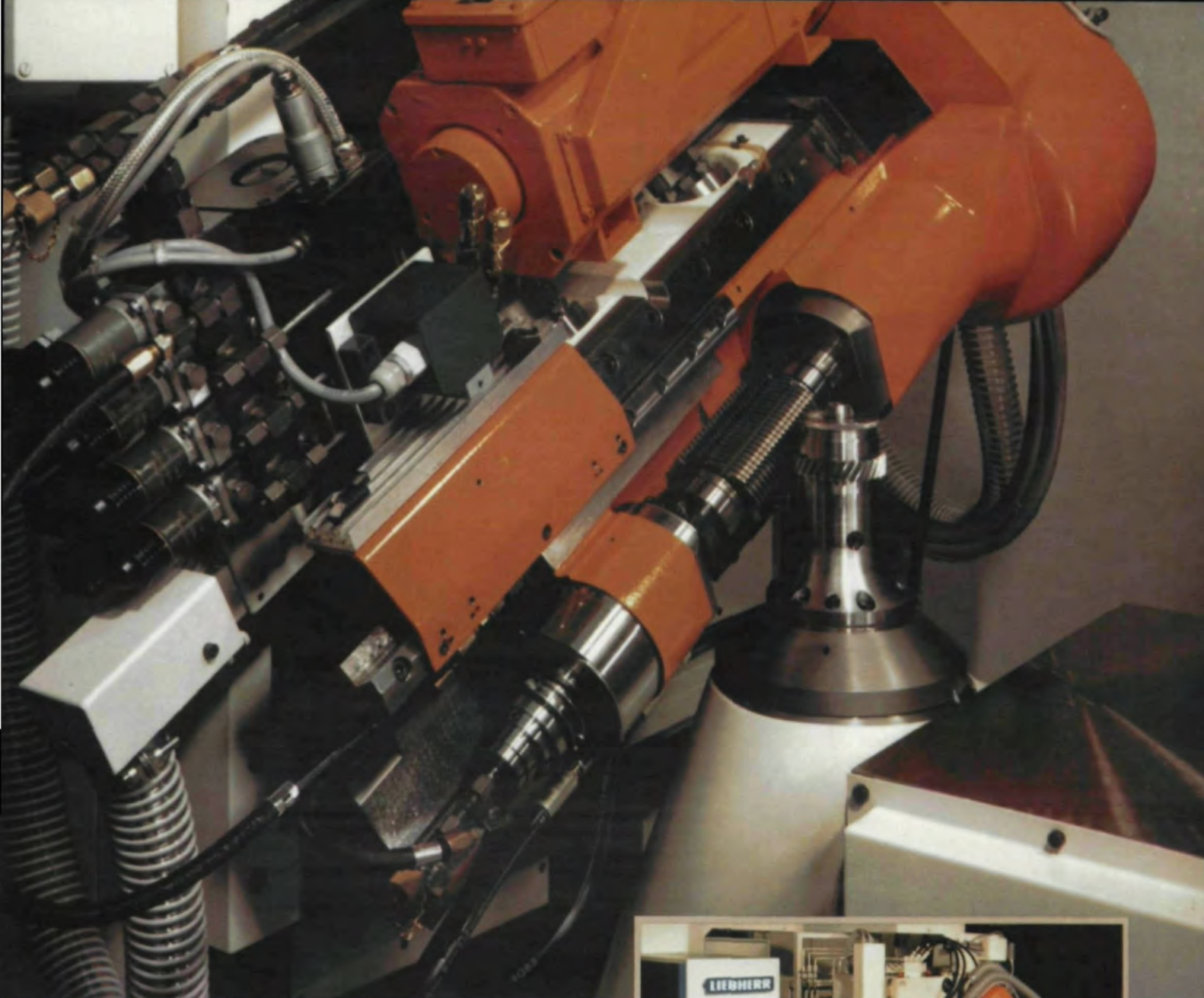
At A in Fig. 8A a 30-tooth $14\frac{1}{2}^\circ$ Gear Shaper cutter is shown in mesh with a 12-tooth pinion. Here it will be noticed in generating the flank and fillet that the ends of the cutter teeth "sweep" inside the radial line, and, hence, undercut the flanks of the pinion. As the gear or the rack is generally stronger than a pinion having a small number of teeth, the gear or rack teeth are generally modified where a severe interference condition exists. This is also the practice where the teeth have to be modified to provide for tooth deflection under heavy loading conditions.

When undercutting of the flank of a pinion having a small number of teeth is to be avoided, several methods can be employed:

1. The blank diameter of the pinion can be enlarged, as shown at B in Fig. 5 and the outside diameter of the mating gear reduced a similar amount. This results in long and short addendum teeth.

Fig. 8A—Diagram Illustrating How the Gear Shaper Cutter Can Be Made so as to Avoid Undercutting the Flank of a 12-Tooth Pinion.





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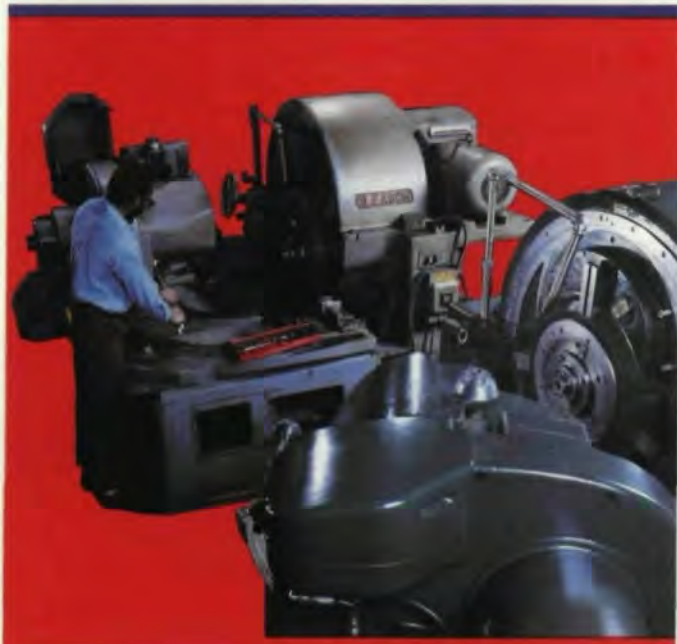
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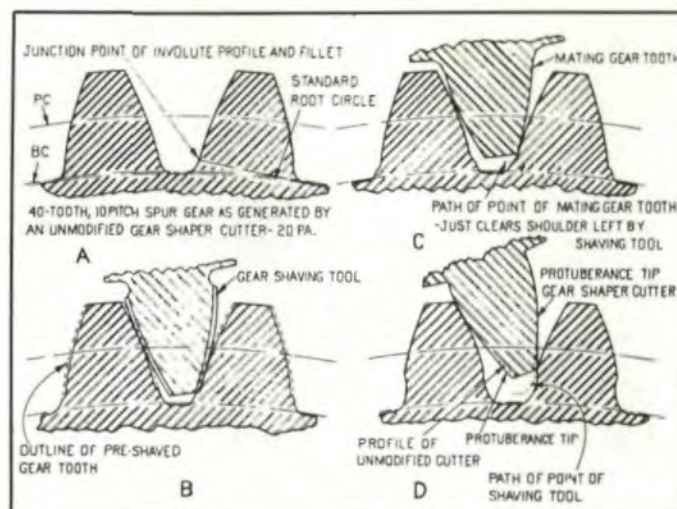


Fig. 8B - Diagram Illustrating Type of Gear Shaper Cutter Used When Gears Are to Be Finished by a Shaving Tool, and Where Flank of Gear Tooth Must Be Undercut.

2. An increased pressure angle can be used.
3. An increased pressure angle combined with shorter addendums.
4. An enlarged Gear Shaper cutter can be used, as shown in B in Fig. 8A.

Here, it will be noticed that the ends of the cutter teeth do not "sweep" inside the radial line. Enlarging the cutter has the effect of increasing the generating pressure angle, and, hence, changes the pitch circle or "rolling circle," and in effect withdraws the cutter from the gear, thus preventing the cutter teeth from undercutting the flanks of the teeth. This, of course, does not avoid interference with the mating member, which would have to be modified, as indicated in Fig. 7. It does, however, provide a stronger tooth shape in the pinion, as undercutting the flank naturally weakens the tooth, all other factors remaining the same.

Pre-Shaving Gear Shaper Cutters

When finishing gears by means of a shaving tool, it is sometimes necessary to undercut the flank of the gear tooth to prevent contact of the shaving tool inside the base circle. Contact with the flank of the gear tooth by the shaving tool can produce two undesirable results. One is that it causes deflections that result in modifications near the top of the gear tooth; the other is that a ridge may be formed in the fillet of the gear tooth which might interfere with the proper operation of the mating gear.

Pre-shaving cutters are, therefore, made with a protuberance, or what might be called a plus involute on the tips of the teeth. This protuberance tip "sweeps" out the flank of the gear tooth and prevents the shaving tool from contacting the non-involute portion of the gear tooth. It also avoids the formation of a ridge in the fillet of the gear tooth.

These conditions are illustrated diagrammatically in Fig. 8A. At A is shown a 40-tooth, 20° full-length gear tooth as cut with an unmodified Gear Shaper cutter. At B this pre-

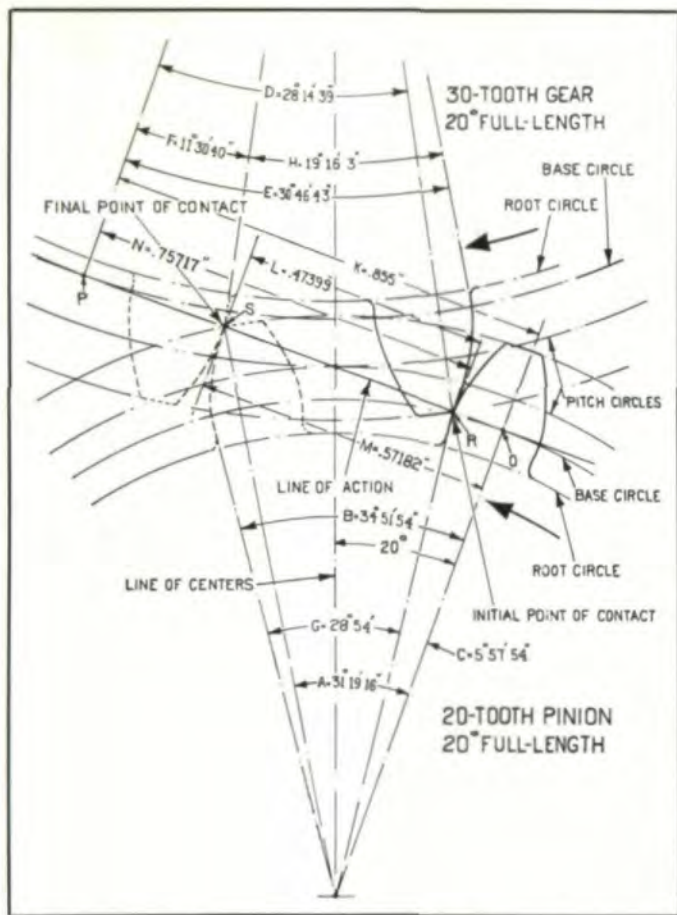


Fig. 9—Diagram Illustrating Gear Tooth Action, Active Length of Involute Profile, and Length of Contact.

shaved tooth is shown as finished with a shaving tool. Note that the shaving tool leaves a ridge in the fillet of the tooth, which is liable to interfere with the proper operation of the mating gear, as shown at C. If this gear had been cut with a cutter having a protuberance tip, as shown at D, contact of the shaving tool with the flank of the gear tooth would have been avoided, and no ridge left to interfere with the proper operation of the mating gear.

This type of cutter is usually recommended for gears 16 pitch and coarser. For finer pitches where an undercutting of the flank is necessary prior to shaving, the pressure angle on the cutter is made less than the specified pressure angle, and by "feeding" this cutter in farther than "standard" depth, it "sweeps" out the non-involute portion of the tooth to provide "clearance" for the shaving tool.

Analyzing and Checking Involute Tooth Profiles

With the exception of the circle, the involute is one of the easiest curves to reproduce and check accurately. It is possible to chart and measure the shape of involute tooth profiles, and to accurately determine the angular location and amount of any deviation of the tooth profile from the "true" involute shape. The angular location of the "initial" and "final" points of contact between two mating gears can be calculated, but it is also possible to determine these points diagrammatically, and to correlate the diagram with charts made on Involute Measuring Instruments, as will be subsequently explained.

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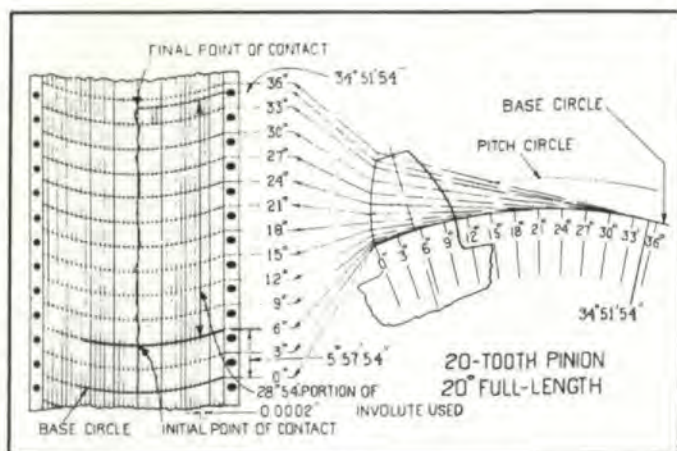


Fig. 10 - Chart of 20-Tooth Pinion in Fig. 9 as Produced on Fellows Involute Measuring Instrument.

Checking Involute Tooth Profiles

An analysis of gear tooth action, as represented in Fig. 9, will help to explain the application of the diagrammatic and charting method for determining the actual length of contact of mating gears, and the angle of involute used.

In general, the involute profile of a gear tooth is checked from its origin, or base circle, to the outside circle, or top of the tooth. Fig. 9 shows a pair of involute gears in mesh for the purpose of illustrating the "initial" and "final" points of tooth contact, and the active portions of the tooth profiles. Points *O* and *P* are the "natural" interference points, and indicate the maximum permissible length of the line of action, provided that the outside circles of gear and pinion, respectively, were extended to the interference points.

Assuming that the pinion is the driver, the point *R* where the outside circle of the gear cuts the line of action is the "initial" point of contact; and point *S* where the outside circle of the pinion cuts the line of action is the "final" point of contact. On the 20-tooth pinion, angle *B* represents the total angle of involute; and, on the 30-tooth gear, *E* represents the total angle of involute. The active angles of involute are *G* on the pinion and *H* on the gear.

Deviations in the profiles within the spaces confined by angles *C* and *F* can be neglected, as far as any involute action is concerned for this particular gear ratio, tooth length, etc., as shown in Fig. 9. Angles *G* and *H*, and the length of the line of contact can be calculated, as will be explained later. They also can be determined approximately from this diagram.

It is also possible to approximately determine the location of the "initial" and "final" points of contact by placing two mating gears on pins spaced at the correct center distance, "rolling" the gears, and marking the points on the tooth profiles where contact starts at *R* and finishes at *S*, as shown in Fig. 9.

It is evident from a study of this diagram that only a portion of the involute on both the pinion and gear is used. Hence, the angles of involute of both members charted in Figs. 10 and 11, as produced on the Involute Measuring Instrument, are greater than those portions of the involutes that are used. It will be noticed in Fig. 10 that the used portion

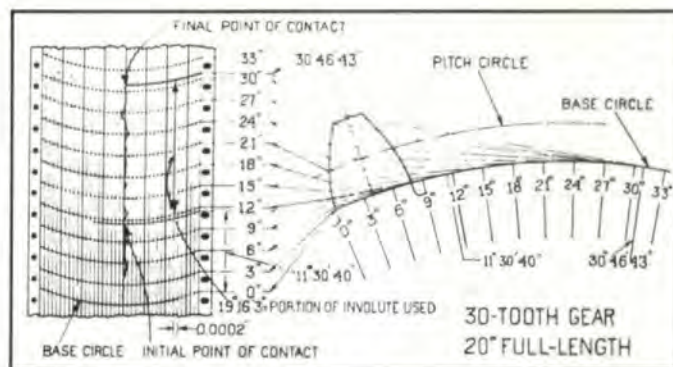


Fig. 11 - Chart of 30-Tooth Gear in Fig. 9 as Produced on Fellows Involute Measuring Instrument.

of the involute on the chart is the same as the angle of involute *G* in Fig. 9. The same applies to the chart in Fig. 11 for the gear, where *H* is the angle of involute used.

The "initial" point of contact on the pinion is $5^{\circ} 57' 54''$ from the origin or base circle, which is equal to angle *C* in Fig. 9, and on the gear, the final point of contact is $11^{\circ} 30' 40''$ from the origin or base circle and is equal to the angle *F*.

On the charts in Figs. 10 and 11, the involute "reading" is recorded with reference to a straight line, and any departure from this straight line represents the amount of deviation from the "true" involute shape. The angular location of deviation in the tooth profile is indicated by the position of the charted line relative to the accurately spaced curved lines. The space between the vertical lines on the chart represents a deviation of 0.0002 inch. The measuring instrument is provided with a ratio mechanism, so that the space between the curved lines can represent either 3 degrees or $\frac{1}{2}$ degree of involute "roll."

For the benefit of those who are interested in determining these angles, etc. mathematically, the formulas and methods of procedure are presented in the following pages.

Calculating Involute Angles

The angular position of any point on an involute with respect to its origin on the base circle can be determined in degrees of rotation of the base circle necessary to develop the involute to that particular point. This value is known as the angle of involute. On a gear, the total angle of involute is the number of degrees of rotation of the base circle from the origin at the base circle to the top of the tooth, or outside circle.

Fig. 12 is a schematic diagram of an involute profile developed by rolling a base cylinder along a base line in which a tracing point is located. The full lines show this cylinder tangent to the base line at the tracing point, which is the origin of the involute. As the base cylinder is rolled counter-clockwise along the base line, the involute is developed by the tracing point, as shown in the illustration.

When the base cylinder has rolled along the base line to the position indicated by the dotted outline, *B* is the angle of involute "roll," *M* is the linear distance traveled, and *OR* is the radial distance from the center of the cylinder to the tracing point, or end of the involute. Angle *B* and distance

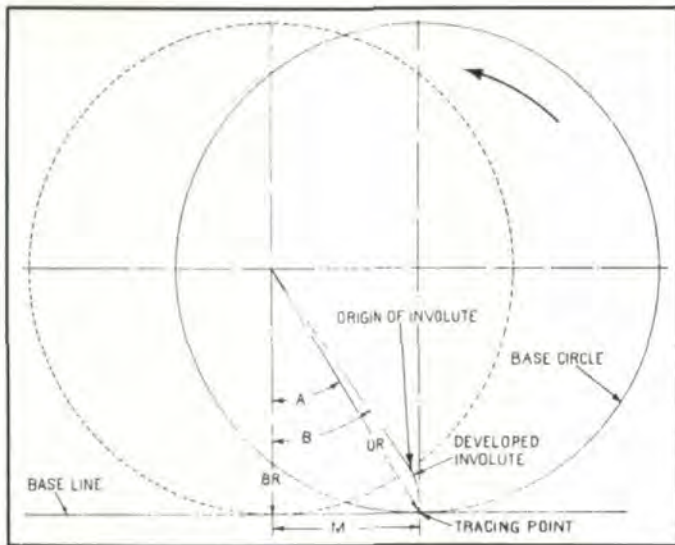


Fig. 12 - Schematic Diagram Showing Development of an Involute Profile.

M can be determined from the following formulas: in which

OR = Radius to end of involute profile

BR = Radius of base cylinder

B = Total angle of involute

Then:

$$\cos A = \frac{BR}{OR}$$

$$B = \frac{\tan A}{.0174533}$$

$$M = \tan A \times BR$$

Determining Angular Location of any Point on Involute Profile; also Length of Contact

As a practical example, we will take the two gears represented in the diagram Fig. 9, the data on which is:

Data	Pinion	Gear
Number of teeth	20	30
Diametral pitch	10	10
Circular pitch	.3142"	.3142"
Pressure angle	20°	20°
Outside radius	1.100"	1.600"
Pitch radius	1.000"	1.500"
Base radius	.9397"	1.4095"
Center distance	2.500"	

Fig. 13 presents a diagram of the gear and pinion shown in Fig. 9, which will be used in calculating the total angles of involute, active and inactive angles of involute, total length of line of action and active length of line of action.

In this diagram, Fig. 13, the involute profiles of pinion and gear are shown in dotted outline at the "initial" and "final" points of contact. The total length of the line of action is the distance between points O and P. Assuming that the pinion is the driver, contact starts at point R and ends at point S.

The total angle of involute, the active and inactive angles of involute of gear and pinion; the total length of the line of action, and the active length of the line of action, as well

as the overlap of action can be determined by the simple formulas which follow. Notation for the 20-tooth pinion follows:

A = The included angle between the point of origin and the point where the outside circle of the pinion cuts the line of action

B = Total angle of involute

C = Inactive angle of involute

G = Active angle of involute

OR₂₀ = Outside radius

BR₂₀ = Base radius

For the 30-tooth gear:

D = The included angle between the point of origin to the point where the outside circle of the gear cuts the line of action

E = Total angle of involute

F = Inactive angle of involute

H = Active angle of involute

OR₃₀ = Outside radius

BR₃₀ = Base radius

For 20-tooth pinion and 30-tooth gear:

PA = Pressure angle

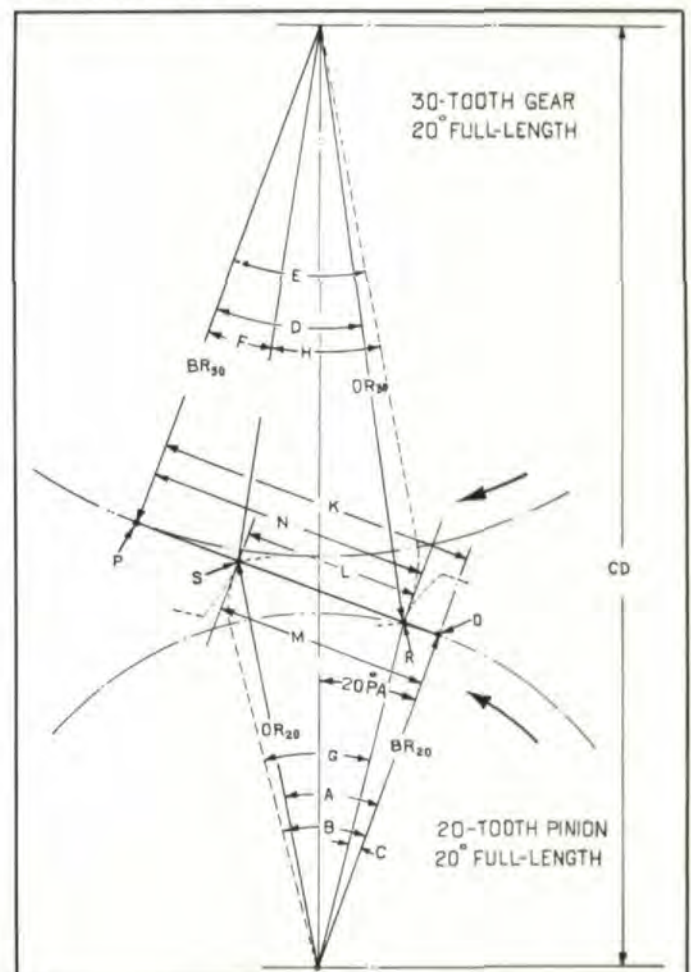
CD = Center distance

K = Total length of line of action

L = Active length of action

M = Distance on line of action corresponding to angle B

Fig. 13 - Diagram Giving Notation for Determining Involute Angles, Length of Line of Action, etc.



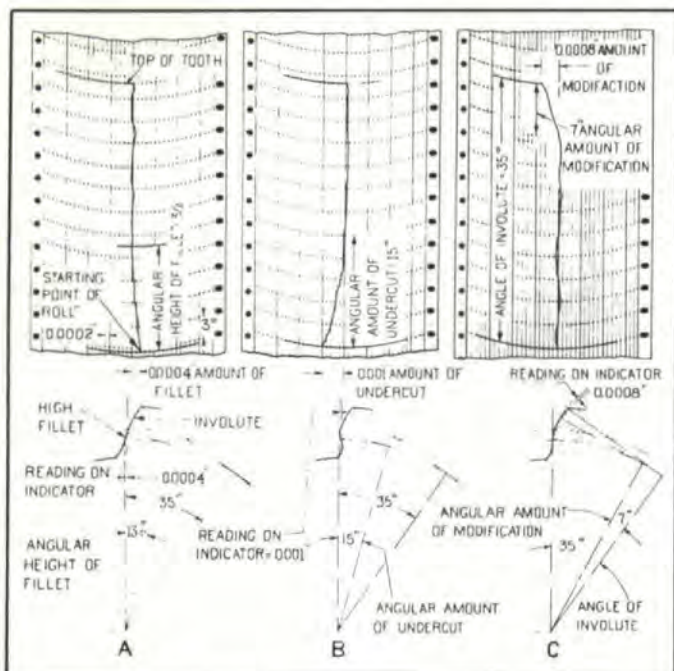


Fig. 14—Diagram of Charts Illustrating High Fillet, Undercut and Tip Modification, Produced on Involute Measuring Instrument.

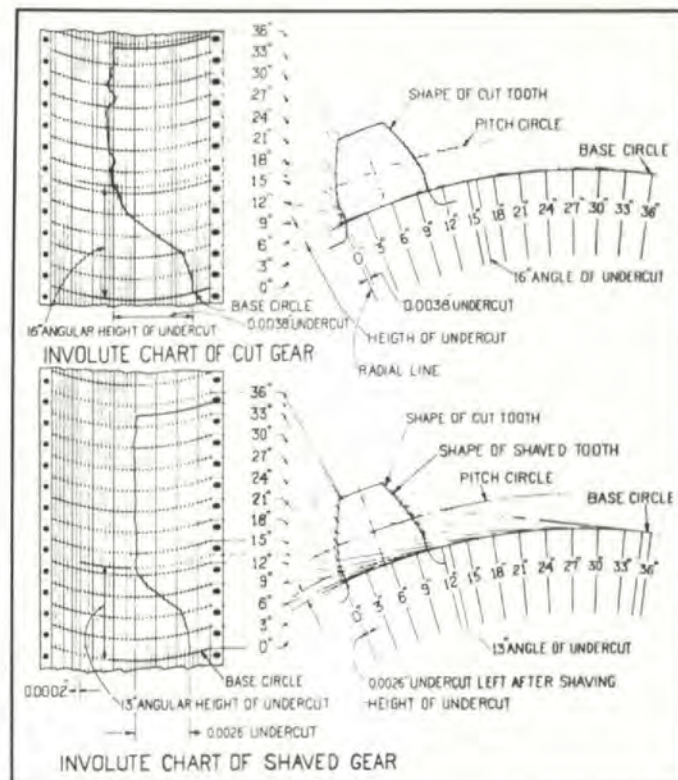


Fig. 15—Charts Made on Involute Measuring Instrument of a Gear after Cutting and after Shaving.

N = Distance on line of action corresponding to angle E

CP = Circular pitch

BP = Base pitch

Then:

$$\cos A = \frac{BR_{20}}{OR_{20}}$$

$$B = \frac{\tan A}{.0174533}$$

$$M = \tan A \times BR_{20}$$

$$K = CD \times \sin PA$$

$$\cos D = \frac{BR_{30}}{OR_{30}}$$

$$E = \frac{\tan D}{.0174533}$$

$$N = \tan D \times BR_{30}$$

$$L = (M + N) - K$$

$$G = \frac{L}{.0174533 \times BR_{20}}$$

$$C = B - G$$

$$H = \frac{L}{.0174533 \times BR_{30}}$$

$$F = E - H$$

$$BP = CP \times \cos PA$$

$$\text{Percentage of overlap of contact} = \frac{L}{BP} - 1$$

Determining the various values for pinion and gear, using the foregoing formulas, we find that:

$$\cos A = \frac{.9397}{1.100} = .85427$$

$$A = 31^\circ 19' 16'' \text{ and } \tan A = .60852$$

$$B = \frac{.60852}{.0174533} = 34^\circ 51' 54''$$

$$M = .60852 \times .9397 = .57182 \text{ inch}$$

$$K = 2.5 \times .34202 = .855 \text{ inch}$$

$$\cos D = \frac{1.4095}{1.600} = .88094$$

$$D = 28^\circ 14' 39'' \text{ and } \tan D = .53719$$

$$E = \frac{.53719}{.0174533} = 30^\circ 46' 43''$$

$$N = .53719 \times 1.4095 = .75717 \text{ inch}$$

$$L = (.57182 + .75717) - .855 = .47399 \text{ inch}$$

$$G = \frac{.47399}{.0174533 \times .9397} = 28^\circ 54'$$

$$C = 34^\circ 51' 54'' - 28^\circ 54' = 5^\circ 57' 54''$$

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$$H = \frac{.47399}{.0174533 \times 1.4095} = 19^\circ 16' 3''$$

$$F = 30^\circ 46' 43'' - 19^\circ 16' 3'' = 11^\circ 39' 40''$$

$$BP = .3142 \times .93969 = .2952 \text{ inch}$$

$$\text{Percentage of overlap of contact} = \frac{.47399}{.2952} - 1 = .60 \text{ or } 60\%$$

It will be seen with this particular ratio, pressure angle and tooth length, that there is no involute or fillet interference, and that there is sufficient overlap of contact to provide continuous action. These conditions are verified by the charts presented in Figs. 10 and 11.

Charting Involute Gear Teeth

The gear teeth and the charts shown in Figs. 10, 11 and 12 presented a gear combination without any tooth modifications. Fig. 14 presents three charts illustrating high fillet, undercut, and involute modification at the tip of the tooth. Referring to the chart at A, Fig. 14, the tooth has a fillet which extends beyond the base circle, and if the angular height of this fillet is greater than the angle C in Fig. 9, the pinion tooth will interfere with the mating gear, and will prevent free rotation.

The diagram at B, Fig. 14, shows an undercut condition in which a portion of the involute profile above the base circle is removed. If the angular amount of undercut is greater than the angle C, Fig. 9, it will shorten the length of the line of contact, and may result in lack of continuous action with the mating gear.

The diagram at C, Fig. 14, shows a gear having tip relief or involute modification, necessary in some cases, and undesirable in others. It is important to know definitely the angular location and amount of this modification in order to determine if continuous action will be obtained when the gears are in mesh. When the tip of the tooth is modified the angle F, Fig. 9, is increased, because the "final" point of contact does not advance as far along the line of action, and thus shortens the length of contact. When this information is determined graphically or mathematically, the chart provides a means for accurately determining if such modifications exist and their angular location and amount.

Fig. 15 presents charts of an 8/10 pitch helical gear having 23 teeth, 20° pressure angle, and 23° helix angle. The involute profile of a helical gear is checked in the plane of rotation, the same as a spur gear. The chart at the top of the illustration is of the gear as cut prior to shaving. It will be noted that the flank of the tooth is undercut 0.0038 inch covering 16 degrees of involute, which extends to a point halfway between the base circle and pitch circle. When this gear is shaved the undercut is reduced to 0.0026 inch and covers only 13, instead of 16 degrees. This chart, as previously ex-

plained, when compared with a similar chart of the mating gear can be used to determine the usable portions of the profiles on both gears, and the actual length of the line of contact can be determined from the preceding formulas.

A WHEEL SELECTION TECHNIQUE . . . (continued from page 14)

$$= 36.2$$

$$\begin{aligned} \text{Total Machining Time} \\ = 22.1 + 14.2 + 36.2 = 72.6 \text{ min.} \end{aligned}$$

CBN Wheel

$$\begin{aligned} \text{Machining Time} \\ = 125 \text{ teeth} \times 2 \text{ in. stroke/5 ipm} + 125 \text{ indexes} \times \frac{1}{60} \\ = 52.1 \text{ min.} \end{aligned}$$

Appendix 3 — Machine/Operator Rate

$$M = \frac{W_o}{N_m M_T} (1 + \text{worker overhead}) + (1 + \text{machine overhead})$$

$$M_T = \frac{\text{cost of machine}}{(\text{hours/year}) \times (\text{depreciation years})}$$

$$\frac{\$750,000}{(4000)(10)} = \$18.75 \text{ hr.}$$

$$\text{assume: } W_o = \$14/\text{hr.}, N_m = 1$$

$$M = \frac{14}{1} (1 + 100\%) + 18.75 (1 + 100\%) = \$65.5/\text{hr.}$$

This paper was presented at the AGMA Fall Technical Conference Oct. 1985.

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