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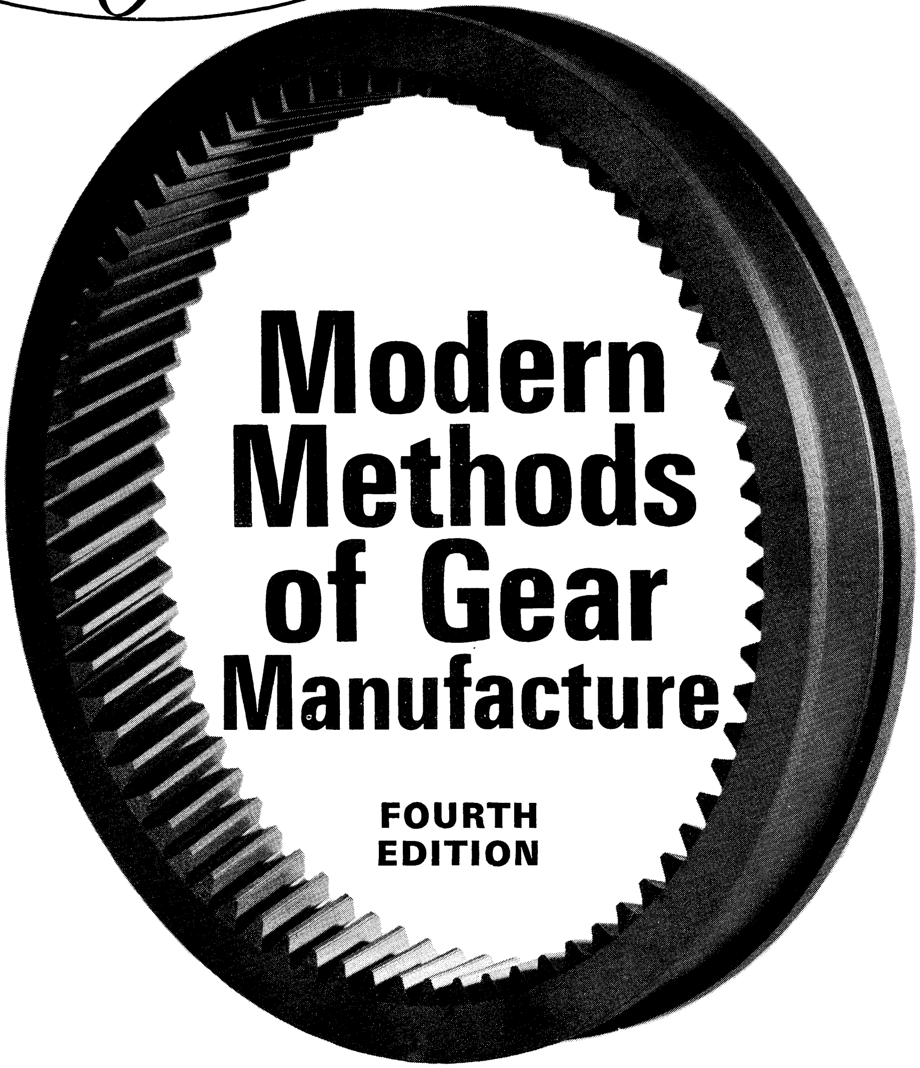

Modern Methods of Gear Manufacture

FOURTH EDITION

Published by

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INTRODUCTION

Today, as it has always been in the past, the design and manufacture of gears is an area of extremely specialized knowledge. The gear specialist continues to occupy a key role in the operation of the modern metalworking plant. Whether the individual be a manager, a product designer, a manufacturing engineer, a production supervisor, a consultant, or even a supplier, the connotation "Gear Specialist" indicates respect for one whose advice and knowledge is actively solicited.

Gear manufacturing practice is in a constant state of change. This results from the continuing search for new and lower-cost methods of producing key components that are among the most difficult and expensive to produce because of stringent requirements for uniform high accuracy, strength and service life capabilities coupled with low operating noise levels.

This fourth edition of Modern Methods of Gear Manufacture, like the first one produced in 1937, has been compiled by National Broach & Machine Div., Lear Siegler, Inc. to provide gear specialists with important reference information not normally found in gear handbooks. This includes design and manufacturing data, the latest production process developments, case history reference material, and machine tool and cutting tool specifications.

As in the past three editions, the material included in this new 160-page reference manual reflects copious records and notes we have compiled from over 40 years of experience in working on every conceivable type of gear problem. It has been organized to provide the kind of information gear specialists need to carry out their everyday work of making better geared power transmission equipment.

There are many new subjects covered in this fourth edition. More design data and heat treatment information are included as they relate to gear material selection. New advances in blank forming, tooth forming, tooth finishing and gear inspection are covered along with many examples of modern gear designs and special gear development work in a wide range of industrial applications.

Of course it is impossible to compress into these pages all of the information needed by specialists in the gear production field. It is, however, the hope of the editors that this treatise will provide a concise reference work for those who are, or aspire to be "Gear Specialists".

Ben F. Bregi
Richard F. Erxleben

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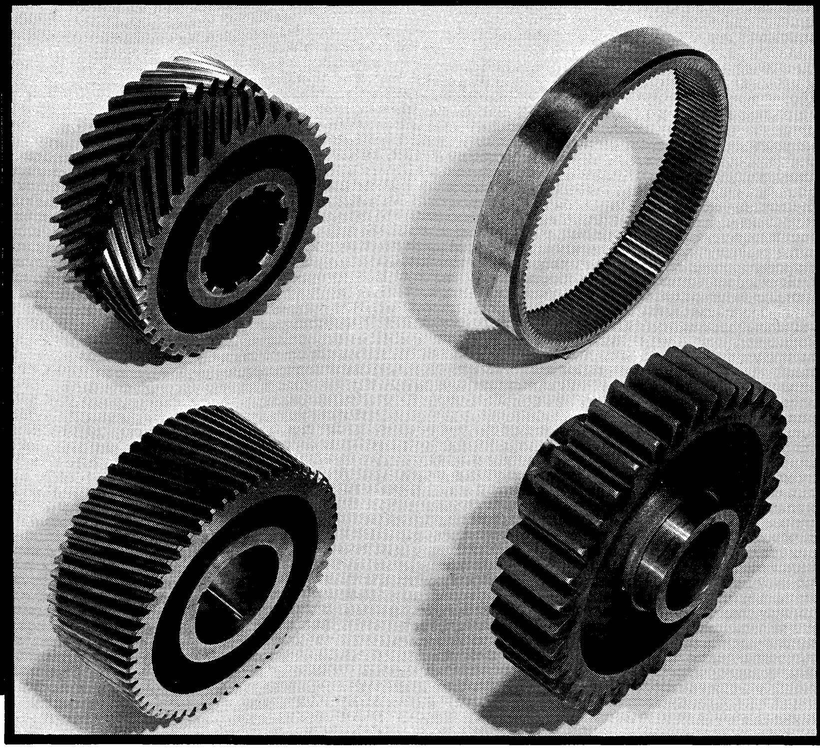


Chapter ONE

Gear Design

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

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

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

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

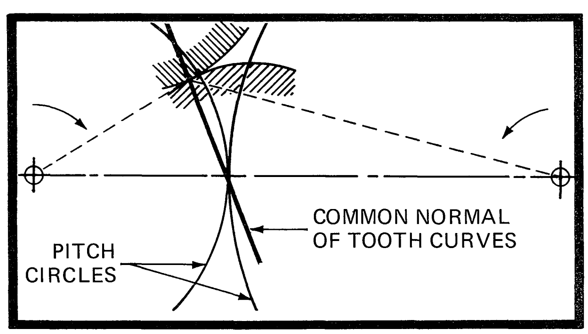


Fig. 1-1—For constant instantaneous relative motion between two engaging gears, the common normal of the curves of the two teeth in mesh must pass through the common point, or point of contact, of the pitch circles of the two gears.

generated or form cut, as with gear cutters on standard milling machines.

Because of inherent good properties and easy reproducibility, the family of cycloid curves was adopted early (1674) and used extensively for gear tooth profiles. The common normal of cycloidal gears is a curve, Fig. 1-2, which is not of a fixed direction, but varies from a maximum inclination with respect to the common tangent at the pitch point to coincidence with the direction of this tangent. Cycloidal gears roll with the direction of this tangent. Cycloidal gears roll with

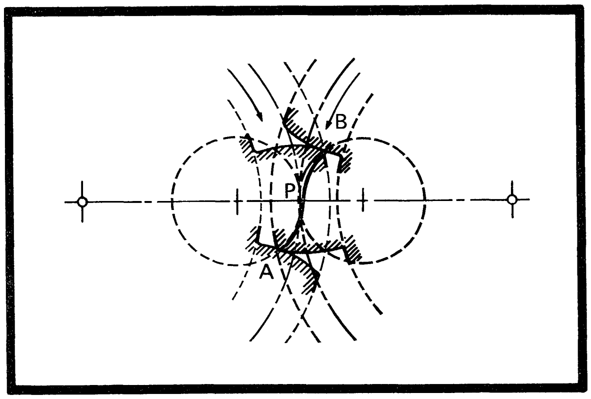


Fig. 1-2—The common normal of cycloidal gears is a curve which varies from a maximum inclination with respect to the common tangent at the pitch point to coincidence with the direction of this tangent. For cycloidal gears rotating as shown here, the arc *BP* is the *Arc of Approach* and the arc *PA*,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

conjugate tooth action providing constant power with uninterrupted rotary motion. One disadvantage of this type of gear is that the center distance between mates must be held to fairly close tolerances, otherwise mating gears will not perform satisfactorily.

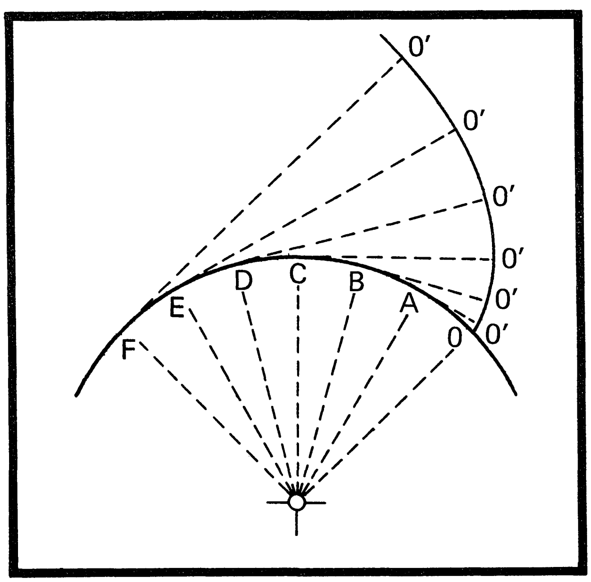


Fig. 1-3—The involute tooth form used for virtually all gearing today is generated by the end of a taut line as it is unwound from the circumference of a circle. The circle from which the line is unwound is the *Base Circle.*

The involute curve was first recommended for gear tooth profiles in the year 1694 but was not commonly used until 150 years later. The curve is generated by the end of a taut line as it is unwound from the circumference of a circle, Fig. 1-3. The circle from which the line is unwound is commonly known as the "base circle". The common normal of involute gear teeth is a straight line (AB in Fig. 1-4). Gears of this type satisfy all the requirements for smooth, accurate and continuous motion. Gears with involute tooth profile are very flexible in both geometric modification and center distance variation.

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

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

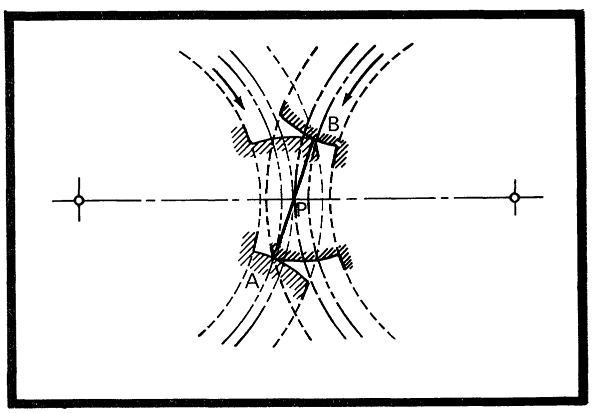


Fig. 1-4—The common normal of involute gear teeth is a straight line *AB.*

Ratio

The primary purpose of gears is to transmit motion and at the same time, multiply either torque or speed. Torque is a function of the horsepower and speed of the power source. It is an indication of the power transmitted through a driving shaft and from it the gear tooth loads are calculated. The loads applied to gear trains can vary from practically nothing to several tons or more. Gears, properly designed and meshed together in mating pairs, can multiply the torque and reduce the higher rotational speed of a power producing source to the slower speeds needed to enable the existing power to move the load. Where application requires speed rather than torque, the process is reversed to increase the speed of the power source.

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

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

Where N_p and N_G = Number of teeth in pinion and gear.

n_p and n_G = Revolutions per minute (rpm) of pinion and gear respectively.

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

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

Velocity

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

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

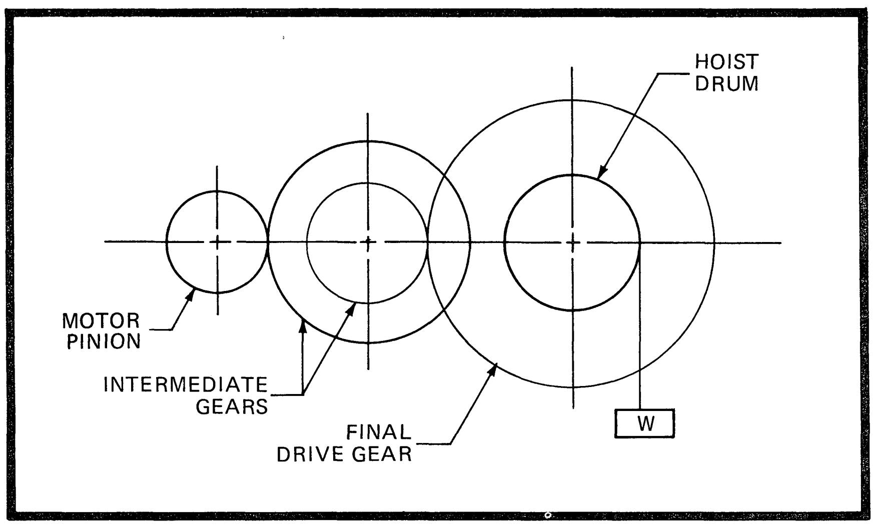


Fig. 1-5—Ratios of the gear teeth in this hypothetical hoist drive would depend upon weight (*W*) to be lifted and torque (*T*) available from the motor.

Elements of Gear Teeth

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

Pitch

Pitch is generally defined as the distance between equally spaced points or surfaces along a given line or curve. On a cylindrical gear it is the arc length between similar points on successive teeth and is known as **circular pitch** (*p*). See Fig. 1-6. Therefore, by definition, circular pitch of gear teeth is a function of circumference and numbers of teeth, varying with diameter and evolving into straight line elements as shown in Figs. 1-7 and 1-8. In Fig. 1-7 the teeth are shown as **helical,** or at an angle to the axis of the gear cylinder. If the teeth were parallel to the axes they would be straight or **spur** teeth as they are more commonly called. With **spur** teeth, Fig. 1-7, the **normal circular pitch** and the **transverse circular pitch** would be equal and the **axial pitch** (a straight line element) would be infinite.

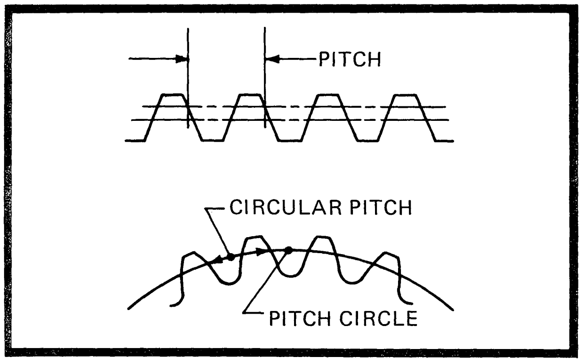


Fig. 1-6—*Circular Pitch* of gear teeth is the arc length along the pitch circle between identical points on successive teeth.

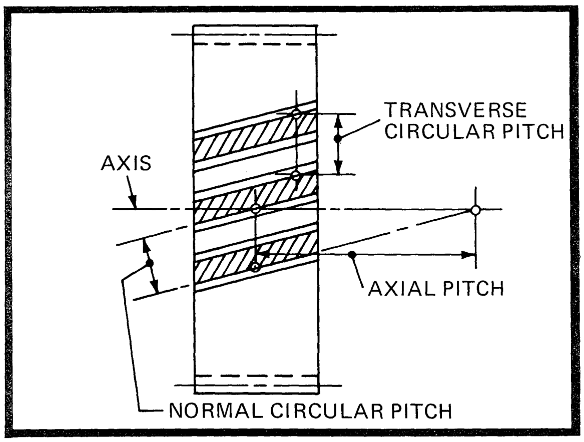


Fig. 1-7—For helical gear teeth, pitch may be measured along a a line normal to the gear teeth (*Normal Circular Pitch*), in a direction perpendicular to the axis of rotation (*Transverse Circular Pitch*), and in a direction parallel to the axis of rotation (*Axial Pitch*).

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

Originally gears were classified and calculated beginning with circular pitch. With the number of teeth (N) and the circular pitch (p) given, the circumference of the circle and consequently the **pitch diameter** (D) can be calculated from

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

For simplification, developers of gear design techniques created a separate term for the value of π divided by circular pitch (π/p). This is **diametral pitch** (P) Fig. 1-10 which is the ratio of

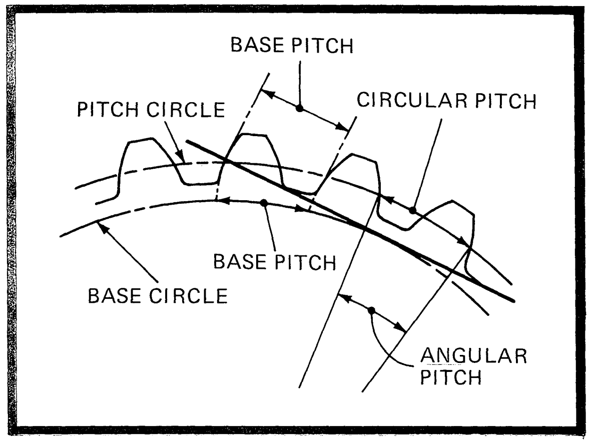


Fig. 1-8—*Base Pitch* and *Angular Pitch* as defined by this drawing are important gear terms. In order to roll together properly, involute gears must have the same *Normal Base Pitch.*

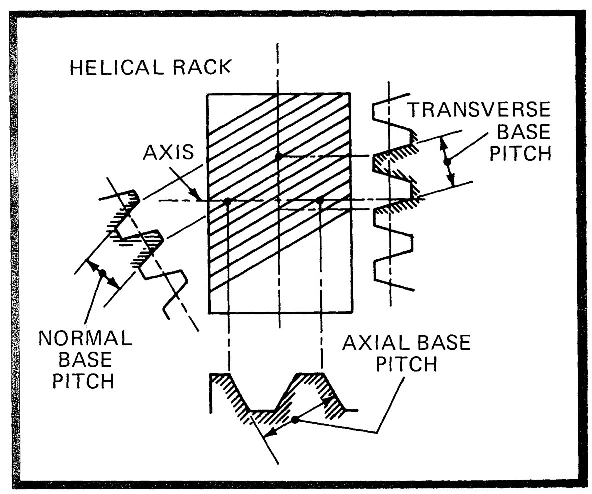


Fig. 1-9—This drawing defines *Transverse Base Pitch, Normal Base Pitch* and *Axial Base Pitch* for a helical rack.

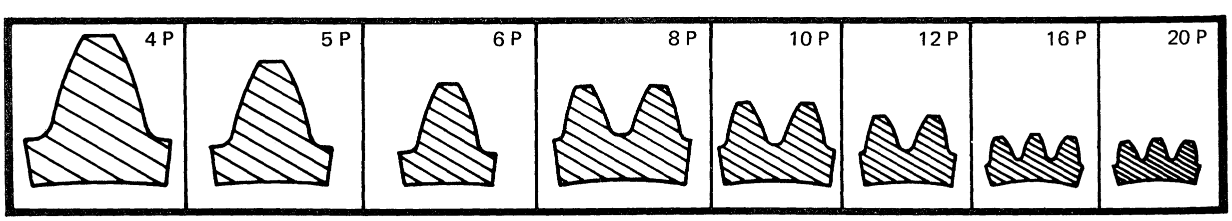


Fig. 1-10—Gear teeth of different diametral pitch, full size, 20-Deg. pressure angle.

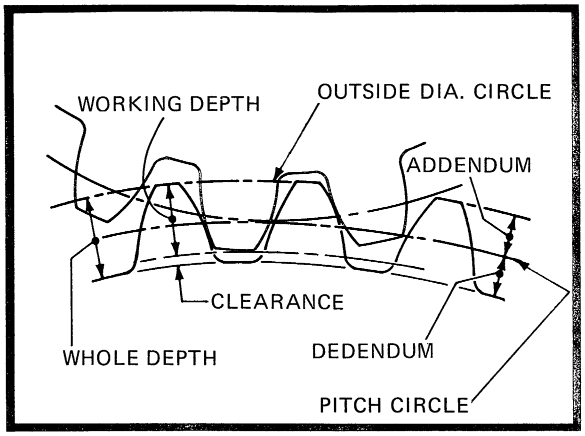


Fig. 1-11—The portion of a gear tooth above the pitch circle is called the *Addendum*; the portion of the tooth below the pitch circle is called the *Dedendum.*

teeth to the pitch diameter in inches. It is a number, it cannot be seen or measured. However, the system developed since the inception of diametral pitch is used almost exclusively wherever the decimal system of measuring is used.

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

Pressure angles

Pressure angles in involute gears are generally designated by the greek letter phi (ϕ), with sub-

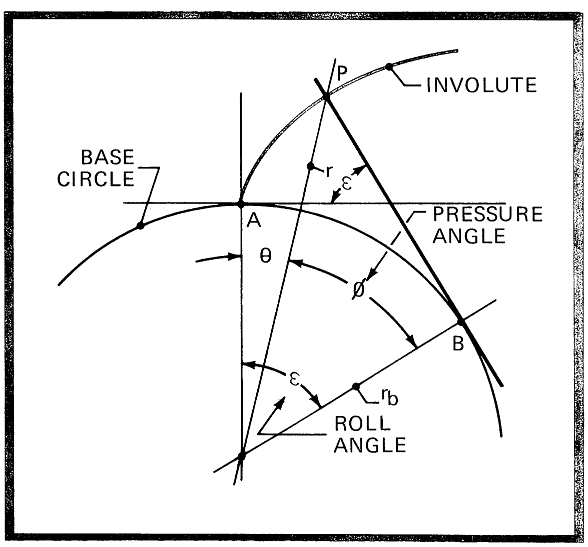


Fig. 1-12—This drawing defines *Roll Angle* (ϵ), *Pressure Angle* (ϕ) and *Polar Angle* (θ).

scripts to denote the various sections and diameters of the gear tooth, Fig. 1-12.

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

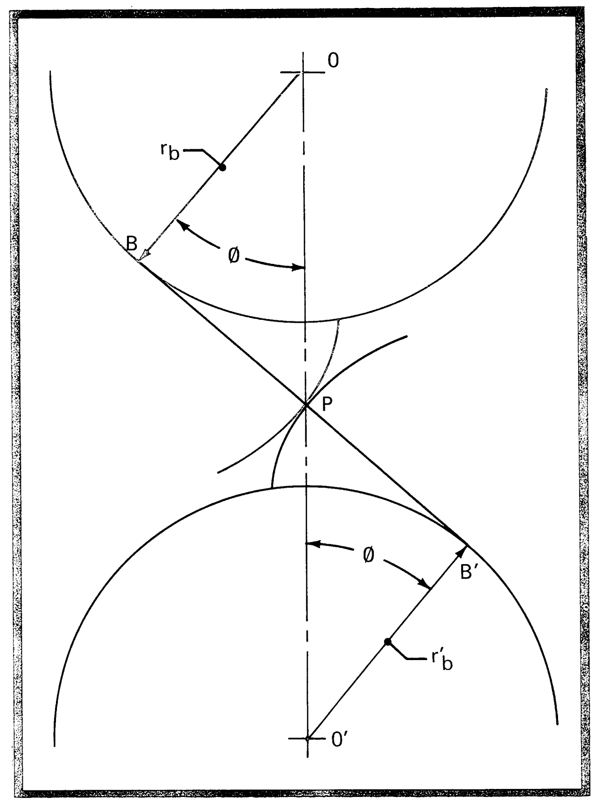


Fig. 1-13—When two involute gear teeth are brought into contact and made tangent at a point *P*, pressure angle ϕ is equal on both.

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

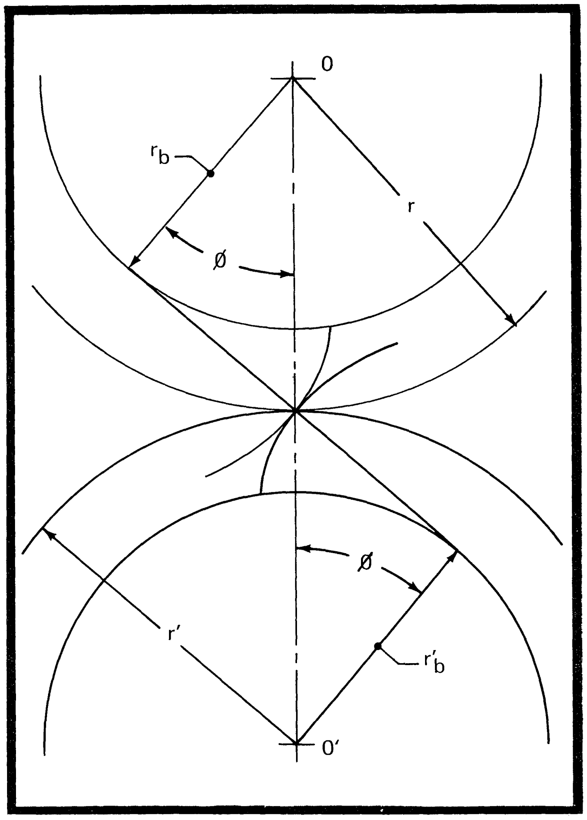


Fig. 1-14—*Base Diameter* (D_b) is twice the base radius (r_b) or the product of the pitch diameter (D) and the cosine of the pressure angle (ϕ).

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

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

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

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

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

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

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

Diametral Pitch, Numbers of Teeth and Pitch Circles

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

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

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

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

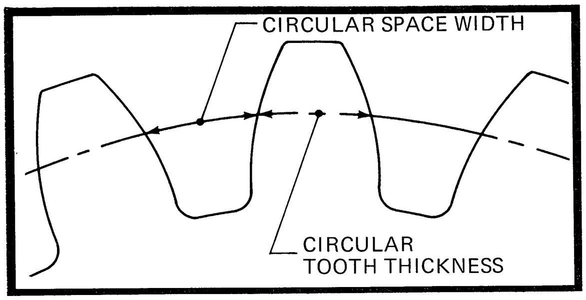


Fig. 1-15—Circular space width and circular tooth thickness defined. When dealing with these measurements, the diameter at which they are measured must be specified.

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

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

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

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

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

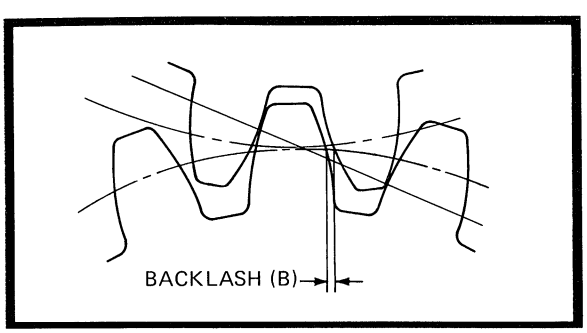


Fig. 1-16—Practical gear teeth usually are thinned to provide a certain amount of *Backlash* (B) to prevent tooth interference under the worst conditions of manufacturing tolerances and expansion due to temperature increase.

$$N = PD$$

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

The Teeth

The elements which describe and regulate the gear tooth proportions are more or less standardized under the diametral pitch system. Most of the circular dimensions are shown in Fig. 1-8 with the radial dimensions shown in Fig. 1-11. In Fig. 1-15, the tooth and space thicknesses are illustrated. These dimensions may be given or calculated at any defined diameter. Sections 1 through 4 in Chapter II, list the tooth elements in dimensional form with reference to diametral pitch. Chapter II also contains many equations for calculating these and other dimensions useful in defining gear tooth design elements.

Gear teeth having involute profiles are very versatile and adaptable to the many variations that may be required. A large percentage of involute gears have **standard** teeth. This class of gear is designed using the general equations given in Chapter II.

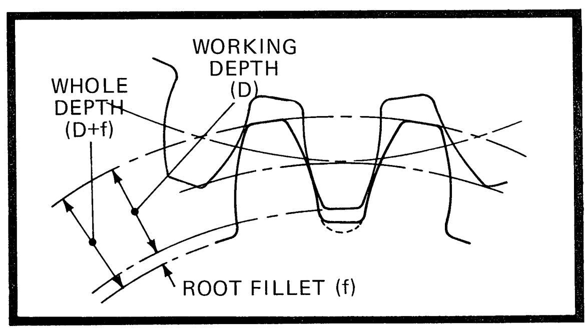


Fig. 1-17—*Whole Depth* ($D + f$) is the sum of working depth and root fillet depth.

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

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

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

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

The expression $D + f$ is sometimes used when referring to the whole depth of gear teeth. The term has an early beginning and for a considerable length of time was the expression for whole depth of tooth. It was originally derived from the sum of working depth, D, and the desired root fillet, f, Fig. 1-17. Since the mating teeth make contact somewhat above the working depth circle, this method of obtaining the whole depth assured clearance between the mating tooth tip and the root fillet. The working depth is flexible and changeable within certain limits to be discussed later. With these two determinants it was a simple matter to obtain a practical whole depth of tooth which allowed mating gears to rotate freely. The term $D + f$ is specified on most gear cutting tools to define the depth of tooth which will be produced by the tool when the tooth thickness of the gear is one-half the circular pitch.

Despite use of the term, $D + f$, whole depth remains addendum plus dedendum because the

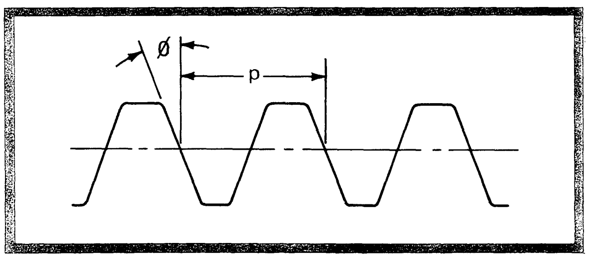


Fig. 1-18—An involute tooth form gear of infinite pitch diameter is called a *Rack.* The teeth have straight sides whose angle equals the chosen pressure angle.

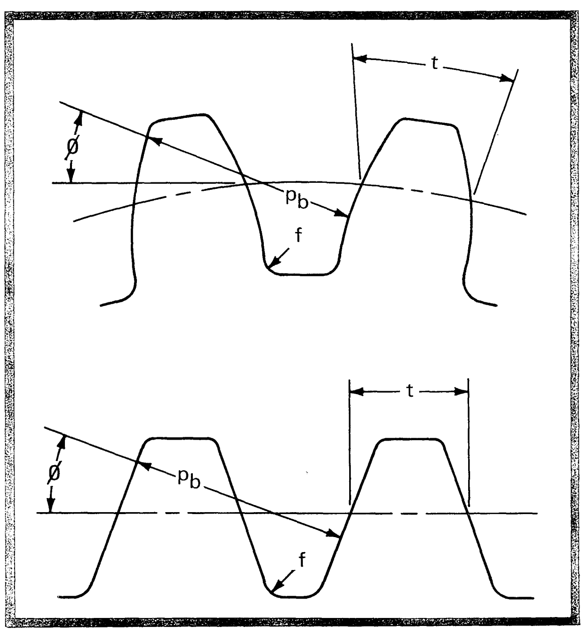


Fig. 1-19—Since a circular involute gear is designed to the same parameters as the basic rack, it will roll freely with the basic rack and with any other circular gear using the same design system.

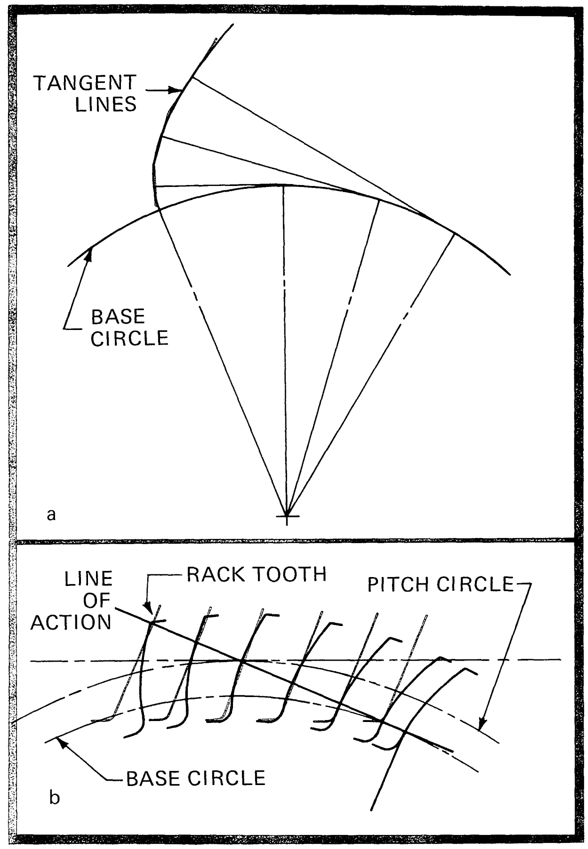


Fig. 1-20—An involute curve may be generated by a series of tangents, *a*. Therefore, if the profile of the basic rack tooth is considered to be tangent to an involute curve from a base circle, the rack becomes the generator of an involute gear with a given number of teeth, *b*.

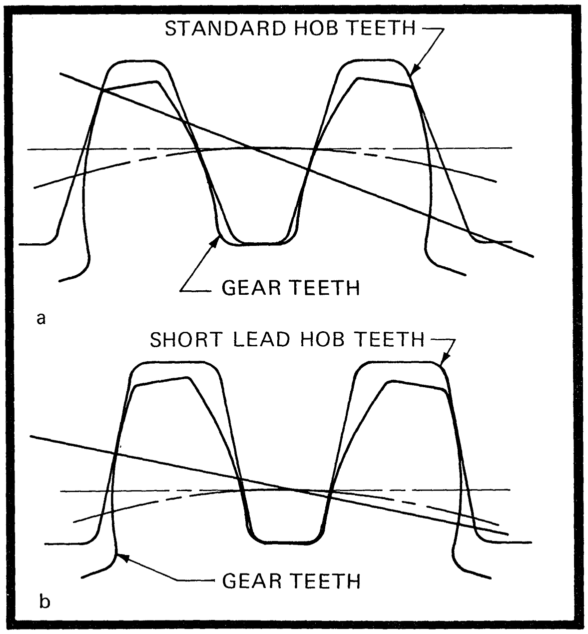


Fig. 1-22—The difference in rolling diameters for *Standard Generating, a,* and *Short Lead* hobs, *b.*

the root fillet produced will be the very minimum and second, the corner will wear away—producing an inconsistent and unknown root fillet. In many cases the root fillet produced by a standard tip corner radius of approximately 1/20 of normal circular pitch would suffice. Very often the load to be carried by the gear teeth will dictate the type and size of root fillet. Therefore, the gear designer should be aware of his control over the profile and root conditions to be produced in gears of his design.

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

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

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

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

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

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

Root fillets should never extend above the form diameter. The shaving cutter must finish the gear

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

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

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

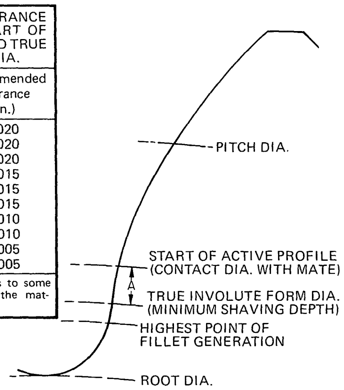


Fig. 1-23—When gear teeth are to be finished by shaving, teeth must be cut to allow for this as shown.

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

profile for the desired length without contacting the gear root fillet, Fig. 1-23. Therefore, gear designers must regulate form diameter, root fillet shape, and root diameter of the tooth for ease of manufacture and satisfactory performance while allowing for the required load.

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

In determining the actual whole depth and root fillet shape, the designer must consider the load which the tooth is to bear. If there is no capacity problem, the whole depths shown in Sections 1 thru 4 in Chapter II will allow ample clearance for the shaving cutter under ordinary conditions. These whole depths are based on the use of preshaving hobs and shaper cutters with a standard tip radius of 1/20 of the normal circular pitch.

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

During the shaving operation a small amount of material is removed from the gear tooth thickness. This material is termed **shaving stock,** and

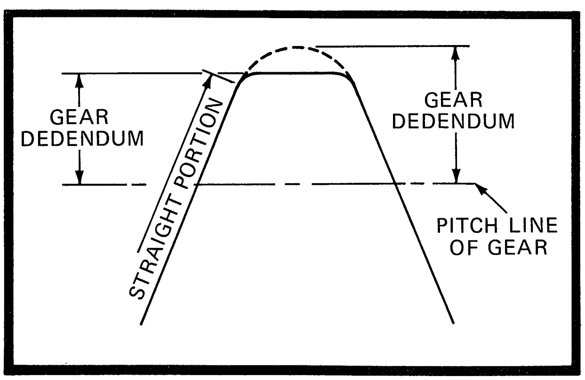


Fig. 1-24—Tooth form of preshaving hob. If full rounded root fillets are desired, additional whole depth becomes necessary.

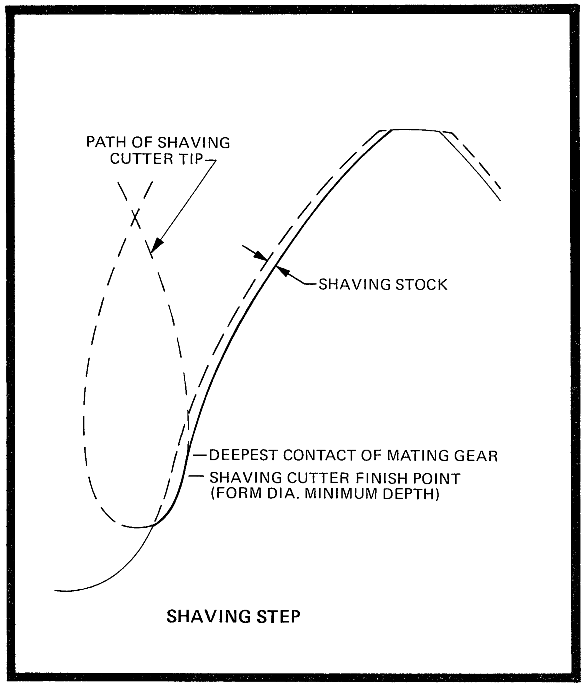


Fig. 1-25—A step in the gear tooth flank will result if the profile in the vicinity of the fillet diameter is not relieved in any way.

recommended amounts for each diametral pitch are listed in Sections 1 through 4 in Chapter II. If the profile in the vicinity of the fillet diameter is not relieved in any manner, a step in the tooth flank will result, Fig. 1-25. This step, caused by the shaving cutter digging in, is detrimental to shaving action. It not only causes excessive wear of the shaving cutter teeth, but also affects the accuracy of the shaved profile. Thus some amount of undercut must be provided to minimize the shaving cutter contact with the gear tooth flank.

The basic problem is to move the fillet and a short portion of tooth profile out of the path of the shaving cutter tip. In some cases with small numbers of teeth, a natural undercutting of the tooth flank occurs. Sometimes this will provide sufficient clearance. However, since natural undercut only occurs in specific cases, the necessary clearance must usually be provided by other means.

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

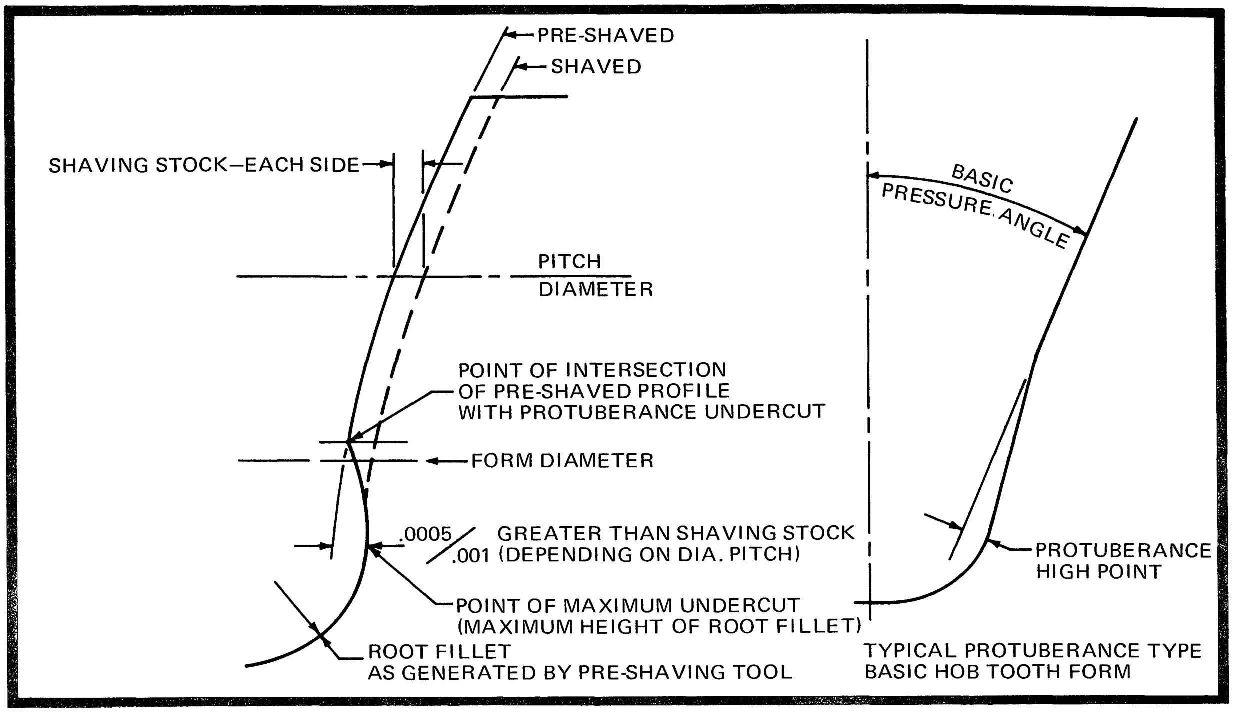


Fig. 1-26—Undercut produced by a protuberance hob and the basic hob tooth form.

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

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

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

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

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

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

Generating Gear Root Fillets

First step is to make a master profile chart on some transparent material. The chart has a master involute profile curve which is laid out

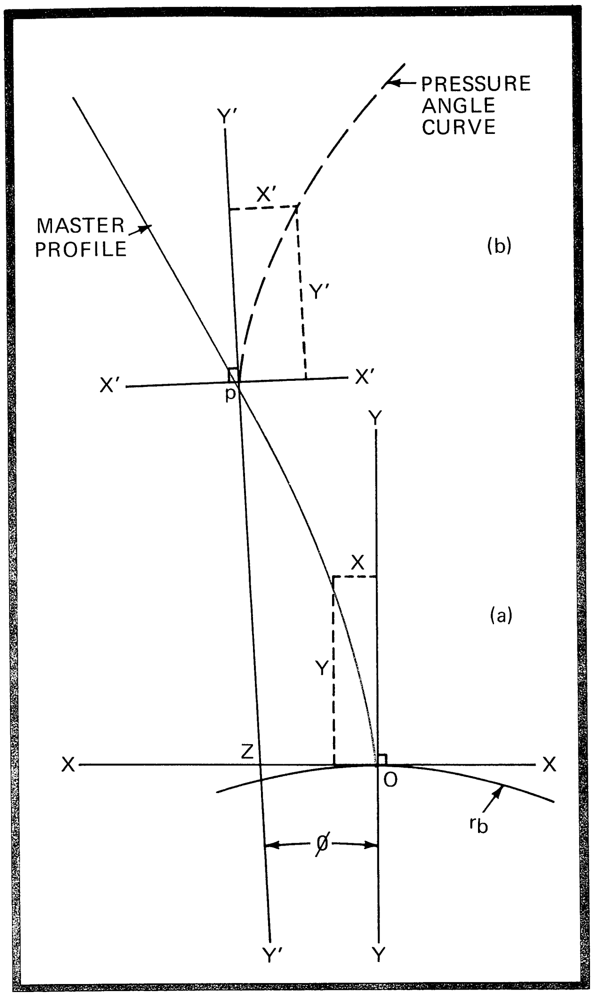


Fig. 1-27—Construction of, *a*, the master involute profile, and, *b*, the pressure angle curve.

by means of rectangular co-ordinates. The base circle diameter must be large enough to produce desirable scales for the sizes of gears being used. The scale of the layout is determined by the ratio of the base circle diameters of the master profile and the work gear. Depending upon the scale desired, a single master involute profile could be used for a number of rolling layouts.

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

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

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

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

Master Profile Chart Layout

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

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

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

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

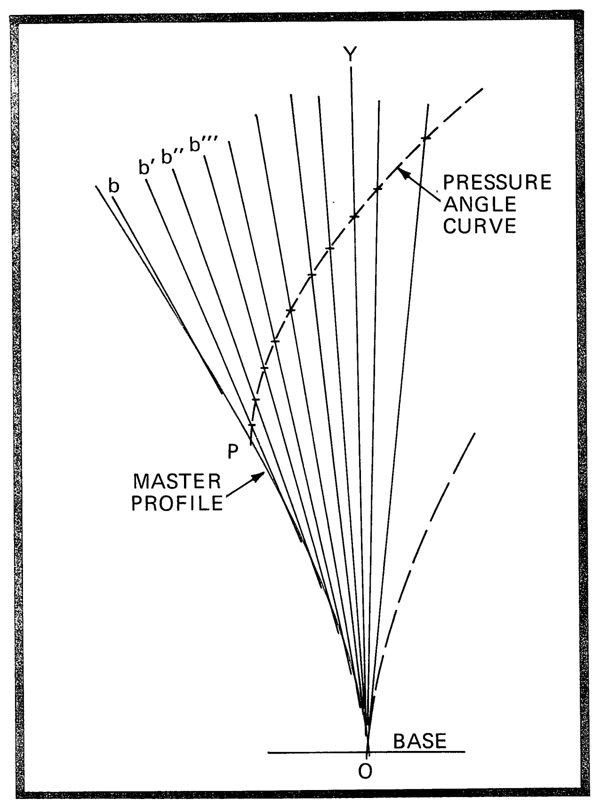


Fig. 1-28—Control lines representing positions of straight side of hob in generation of gear tooth form.

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

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

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

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

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

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

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

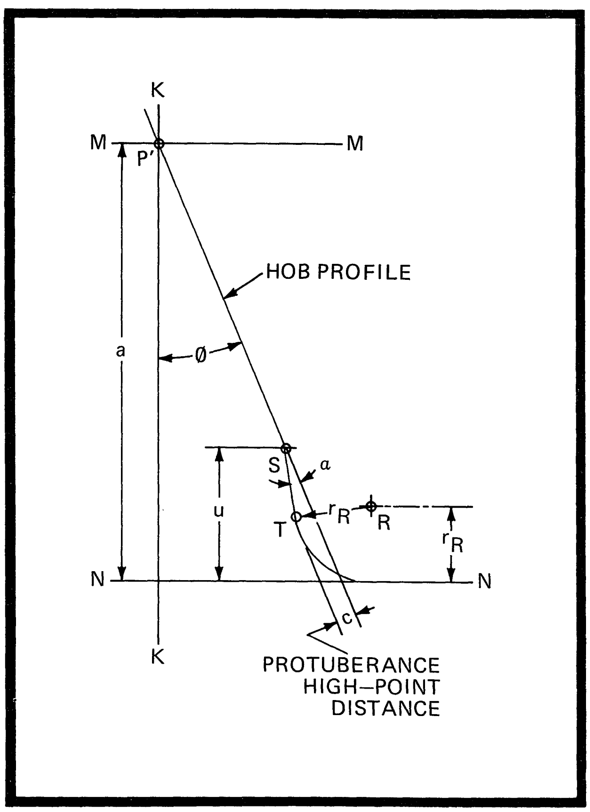


Fig. 1-29—Hob tooth profile.

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

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

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

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

Hob Tooth Layout

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

$$a = r - r_r$$

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

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

Using the Master Profile

For most efficient use, prints or duplications of the master profile chart should be of the type which show the lines of the layout on a white background. Then as shown in Fig. 1-30, the layout of the hob is laid over the master profile with point *P′* on the hob profile directly over *P* on the master profile and with the main profile of the hob coincident with the initial control line *b* which is tangent to the profile at point *P*. Using the point of a pricker, point *P′* is aligned with *P* as perfectly as possible and then points *S*, *T* and *R* on the hob form are punched through to the print below. A compass set to the scaled tip radius of the hob is centered at the transferred point *R* and the radius is inscribed through the transferred point *T*. Points *S* and *T* are joined by a straight line. In all cases point *S* should fall on a control line coinciding with the main profile of the hob. Subsequent positions of the hob tip are determined by the same procedure, using the intersection points between the control lines and the pressure angle curve for locating point *P′*. As *P′* proceeds along the pressure angle curve, the fillet and the undercut are formed to the root of the gear tooth space by the hob tip radius, protuberance and approach.

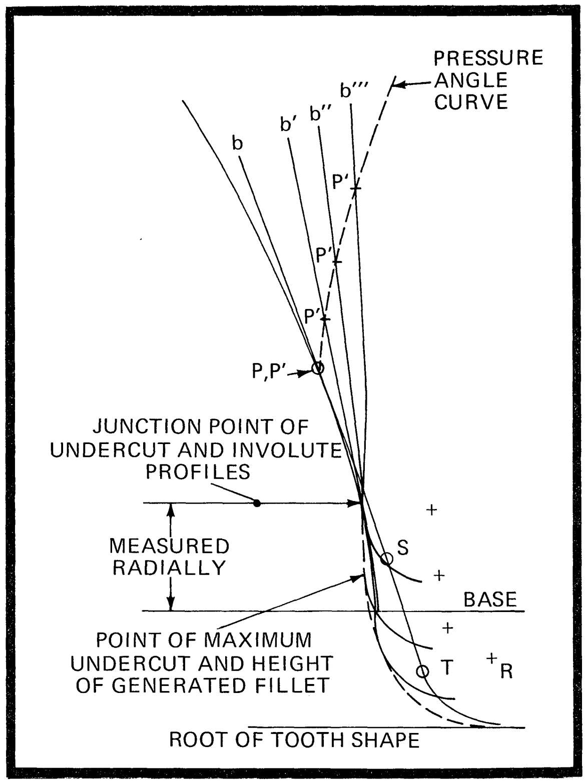


Fig. 1-30—Generation of root fillet by hob protuberance.

Modifications of Standard Gear Tooth Forms

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

Stub Tooth Gears

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

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

A special case of stub tooth design is in its use as non-running involute spline teeth. Involute splines have maximum strength at the base; they can be accurately spaced and are self-centering

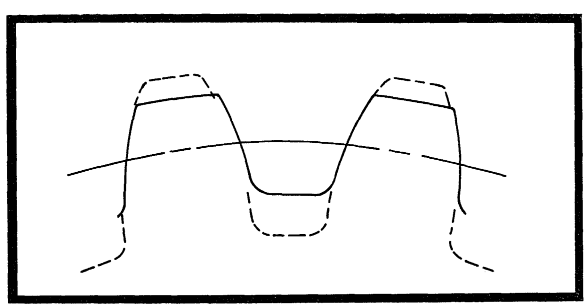


Fig. 1-31—*Stub Teeth* (solid line) are shorter than *Standard Teeth* (dashed line). They are stronger and minimize undercut in small pinions.

which equalizes the bearing and stresses. The teeth can be measured and fitted accurately. Normally the tooth height is standardized at 50% of that based on the diametral pitch. For example, a 5 diametral pitch spline tooth would have a 10 pitch addendum and whole depth. Usually the pressure angle is 30 deg. However, this is not mandatory and quite often is changed to suit design conveniences. Involute spline teeth may be either helical or spur.

Extended Addendum Gears

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

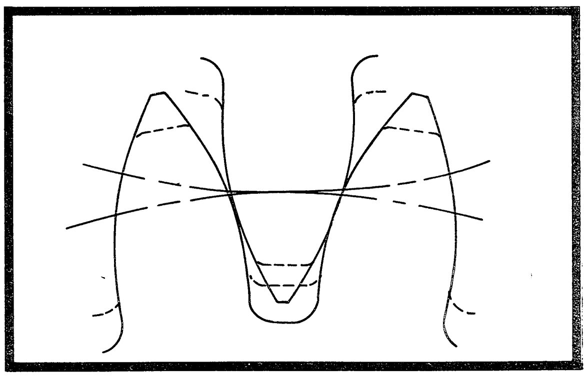


Fig. 1-32—*Extended Addendum Gears* (solid line) have longer period of contact to provide a smoother roll between adjacent mating teeth.

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

Long and Short Addendum Gears

Occasionally a design will require a gear set wherein one member is considerably smaller than the other. If the tooth proportions are made standard, root fillet conditions produced in the small pinion and mating gear contacts may result in a poorly operating set of gears. As an example, Fig. 1-33 shows a pair of mating gears with the standard outside diameter of the larger member extended beyond the limit of the involute curved profile of the pinion. Since all profile action stops at the base circle, the mating gear addendum

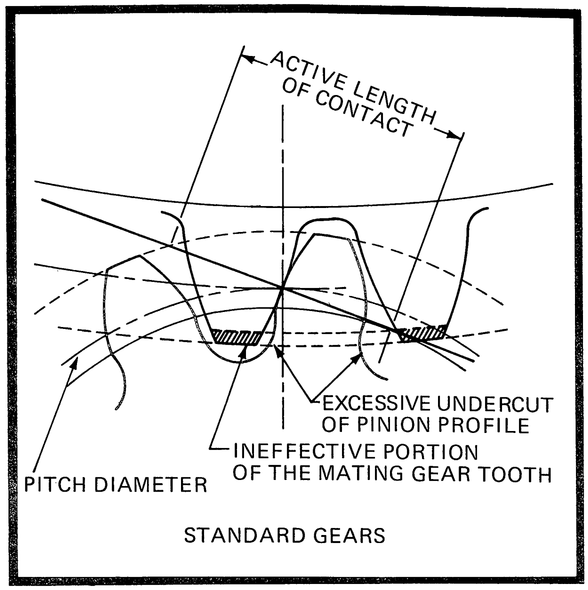


Fig. 1-33—A pair of mating gears with the standard outside diameter of the larger member extending below the limit of the involute profile of the pinion.

extending beyond the line of action represents a loss of contact between mates. Further, if the root of the pinion flank had not been excessively undercut, thus weakening the tooth, the mating gear tip would have found fillet metal as interference to its trochoidal sweep. The **Long and Short Addendum Method** was devised to alleviate this type of situation.

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

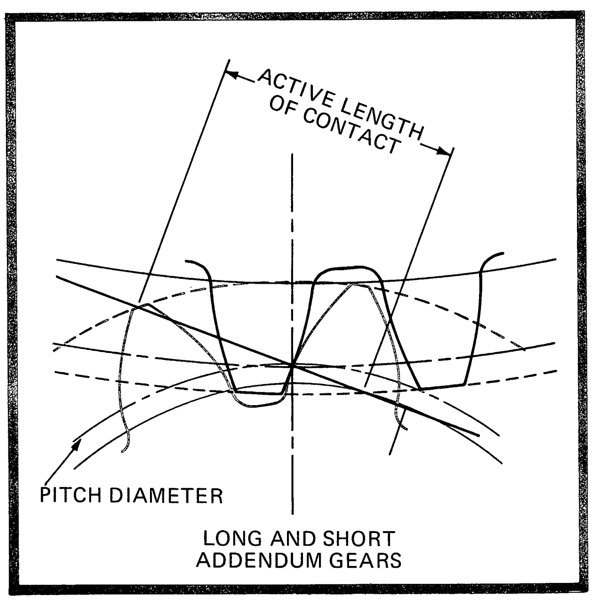


Fig. 1-34—*Long and Short Addendum Gears* minimize the problems where the pinion is very small compared to the mating gear.

working depth and center distance remain unchanged, see Fig. 1-34.

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

The actual amount of *profile shift*, as it is known, can be determined in various ways, including an outright guess. However, there are more exact methods. For instance, Section 15 of Chapter II illustrates and explains the calculation of the maximum outside diameter of the gear which will not extend beyond the limit of involute profile on the pinion. The difference between this new outside diameter and the original standard design divided by two is the *profile shift*. An approximate maximum shift would be one-half of standard addendum for the given diametral pitch. In any event, changing the diameters by two times the profile shift results in an oversize (long addendum) pinion and an undersize (short addendum) gear compared with the original standard units. Consequently, it becomes necessary to revise the tooth thicknesses.

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

Therefore,

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

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

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

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

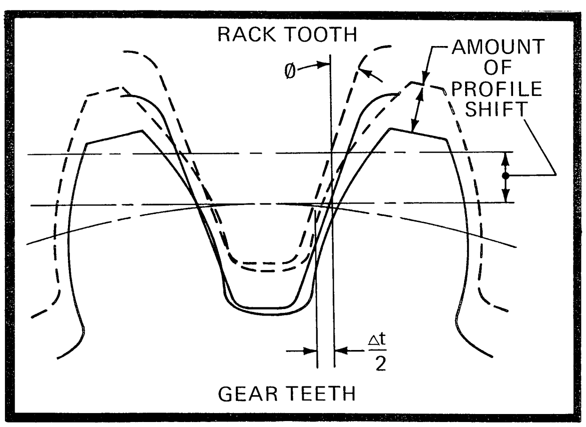


Fig. 1-35—Long addendum pinion showing profile shift.

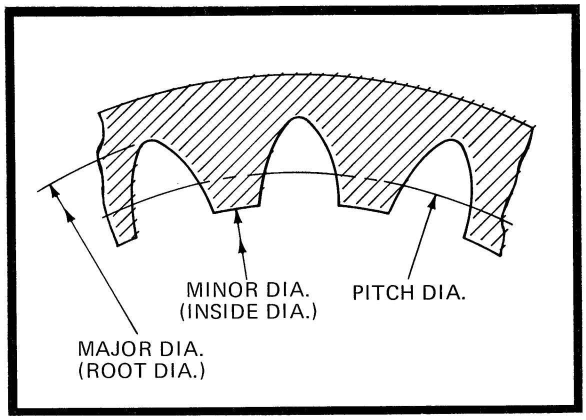


Fig. 1-36—Internal gear teeth protrude inwardly from the inside diameter of a ring.

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

Non-Standard Center Distance Gears

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

Using the given center distance and ratio of numbers of teeth, the operating pitch diameters, circular pitch and tooth thicknesses can be calculated through the use of the equations in Chapter II. Equations are also available for obtaining the outside diameters, size control and other dimensions necessary to complete the gear set design.

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

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

Internal Gear Teeth

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

Involute internal gear teeth have concave curved profiles. Theoretically, an external and internal gear with the same number of teeth and tooth proportions would conform exactly, except for root clearances. Internal gears with involute curve profiles have the same basic fundamentals as

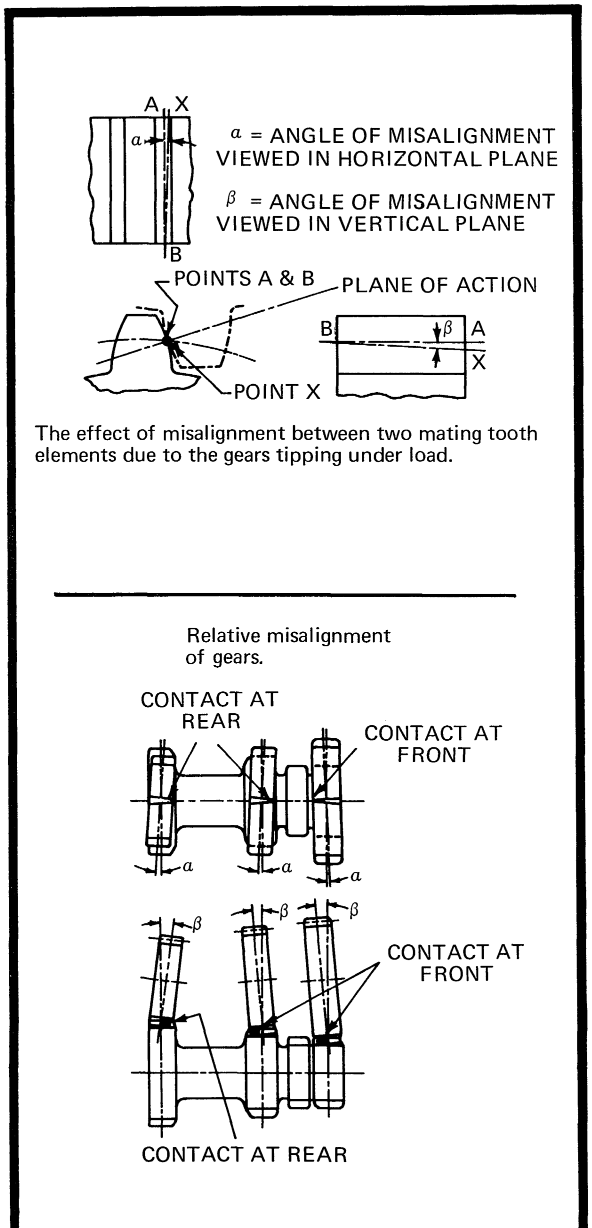


Fig. 1-37—Certain undesirable meshing conditions lead to premature failure of gear teeth. Some of these are shown here.

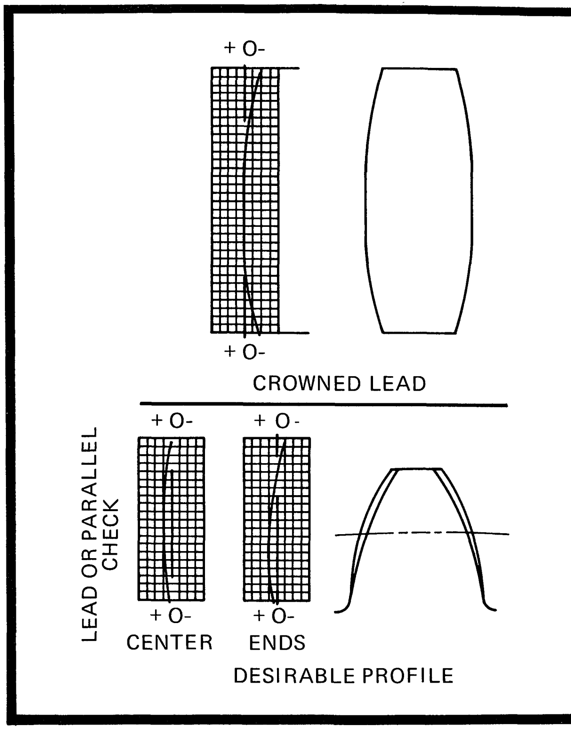


Fig. 1-38—Crowned gear teeth minimize the undesirable effects of departures from theoretical tooth form.

the externals. In some of the equations it will be necessary to subtract instead of adding. However, the same forms and modifications used with external gears may be used with internal gears.

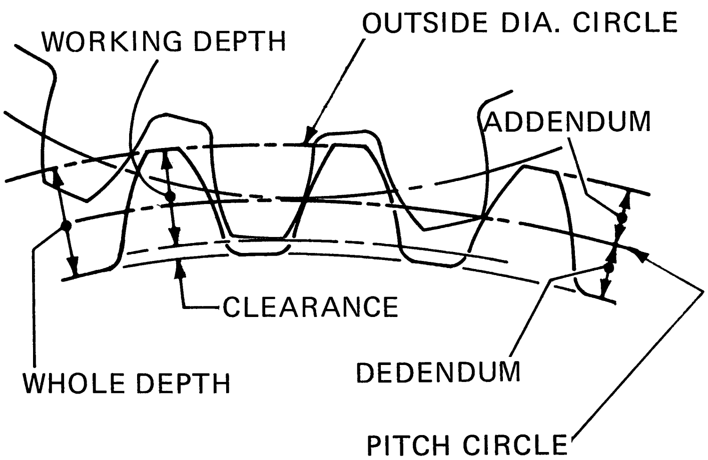
Modification of Tooth Profile and Lead

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

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

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

The subject of gear profile and lead modifications will be covered in a later chapter.



Chapter TWO

Gear Calculations

The number of equations used in designing and analyzing gears of all types is almost countless. Since the cylindrical involute-form gear is the most widely used in industry today, this chapter is devoted to the equations deemed most basic and necessary to calculations for this type of gearing. While many of the equations are found in other handbooks available to gear engineers, there are many which are not.

Wherever possible, all nomenclature, letter symbols and letter subscripts used in this chapter conform to those published in the AGMA Standard 112.04 "Gear Nomenclature—Terms, Definitions, Symbols and Abbreviations."

Letter Symbols for Gear Dimensions and Calculations

a	Addendum
b	Dedendum
B	Backlash
B_n	Normal Circular Backlash
c	Clearance
C	Center Distance
d	Diameter of Measuring Pin or Ball
D	Pitch Diameter
D_b	Base Diameter
D_c	Contact Diameter
D_f	Form Diameter
D_F	Fillet Diameter
D_i	Minor Diameter (Internal Gears)
D_m	Diameter of Circle Through Center of Measuring Pins or Balls
D_{Me}	Measurement Over Pins or Balls (External)
D_{Mi}	Measurement Between Pins or Balls (Internal)
D_o	Outside Diameter
D_r	Rolling or Operating Pitch Diameter
D_R	Root Diameter
D_s	Shaved Diameter
F	Face Width
h_k	Working Depth

Letter Symbols for Gear Dimensions and Calculations *(Cont'd)*

h_t	Whole Depth
L	Lead
LA	Total Length of Line of Action
m	Module (Metric System)
m_F	Face Contact Ratio (Helical Overlap)
m_n	Normal Module
m_p	Involute Contact Ratio (Involute Overlap)
m_t	Transverse Module
N	Number of Teeth in Gear (N_G) or Pinion (N_p)
p	Circular Pitch
p_b	Base Pitch
p_n	Normal Circular Pitch
p_t	Transverse Circular Pitch
p_x	Axial Pitch
P	Diametral Pitch
P_n	Normal Diametral Pitch
P_t	Transverse Diametral Pitch
SAP	Start of Active Profile
S	Circular Tooth Space
SRP	Start of Radius Profile
S_n	Normal Circular Tooth Space
S_t	Transverse Circular Tooth Space
t	Circular Tooth Thickness (t_n, t_t, etc.)
t_c	Chordal Tooth Thickness (t_{nc}, t_{tc}, etc.)
Z	Length of Contact

Angles

ϕ	Pressure Angle
ϕ_m	Pressure Angle to Center of Measuring Pin or Ball
ϕ_n	Normal Pressure Angle
ϕ_r	Operating Pressure Angle (ϕ_{nr}, ϕ_{tr}, etc.)
ϕ_t	Transverse Pressure Angle
ϕ_{to}	Transverse Pressure Angle at OD
ϕ_x	Axial Pressure Angle
ψ	Helix Angle
ψ_b	Base Helix Angle
ψ_r	Operating Helix Angle
ψ_o	Helix Angle at OD
X	Crossed Axes Angle

Notes

1. The addition of an arc (⌒) over the symbol for an angle indicates that the angle is in radians, rather than degrees.
2. Subscripts are used with symbols to differentiate between various diameters and angles and to indicate whether pinion or gear characteristics are involved.

Terminology

Transverse Characteristics (subscript $_t$) are taken in the plane of rotation, parallel to the gear face and perpendicular to the axis.

Normal Characteristics (subscript $_n$) are taken from a section of the gear teeth which is normal to the helix at a given diameter.

Axial Plane Characteristics (subscript $_x$) are in a plane through the teeth and axis of the gear, perpendicular to the gear face.

Start of Active Profile (*SAP*) is the lowest point of mating gear contact as measured along the line of action in inches or degrees of roll from zero (base diameter).

Contact Diameter (D_c) is the diameter through the lowest point of mating gear contact.

Form Diameter (D_f) is the diameter through the lowest point on the gear profile where the desired involute tooth form is to start.

Start of Radius Profile (*SRP*) is the height of the generated root fillet on a gear as measured along the line of action from the base diameter.

Fillet Diameter (D_F) is the diameter through the start of radius profile.

Shaved Diameter (D_s) is the diameter through the lowest point of contact of the shaving cutter; i.e., the start of the shaved profile of a gear.

Crossed Axes Angle (X) is the sum or difference of the gear and shaving cutter helix angles, dependent upon the hand of the helix and center distance. If the hands are opposite, the crossed axes angle will be equal to the difference between the helix angles; if the hands are the same, it will be equal to the sum.

SECTION 1

standard gear tooth elements (inches)

NORMAL DIAMETRAL PITCH	NORMAL CIRCULAR PITCH	STANDARD CIRCULAR TOOTH THICKNESS	STANDARD ADDENDUM	STANDARD WORK DEPTH	*MINIMUM CLEARANCE	*MINIMUM WHOLE DEPTH	TOTAL SHAVING STOCK ON TOOTH THICKNESS
2.5	1.256637	.6283	.4000	.8000	.090	.890	.0040/.0050
3	1.047198	.5236	.3333	.6666	.083	.750	.0040/.0050
3.5	.897598	.4488	.2857	.5714	.074	.645	.0040/.0050
4	.785398	.3927	.2500	.5000	.068	.568	.0035/.0045
4.5	.698132	.3491	.2222	.4444	.060	.504	.0035/.0045
5	.628319	.3142	.2000	.4000	.056	.456	.0035/.0045
6	.523599	.2618	.1667	.3334	.050	.383	.0030/.0040
7	.448799	.2244	.1429	.2858	.044	.330	.0030/.0040
8	.392699	.1963	.1250	.2500	.040	.290	.0030/.0040
9	.349066	.1745	.1111	.2222	.038	.260	.0025/.0035
10	.314159	.1571	.1000	.2000	.035	.235	.0025/.0035
11	.285599	.1428	.0909	.1818	.032	.214	.0020/.0030
12	.261799	.1309	.0833	.1666	.030	.197	.0020/.0030
14	.224399	.1122	.0714	.1428	.026	.169	.0020/.0030
16	.196350	.0982	.0625	.1250	.023	.148	.0015/.0025
18	.174533	.0873	.0556	.1112	.021	.132	.0015/.0025

***Based on standard tip radius on preshaving hobs and shaper cutters.**
Increase whole depth by approximately .005" to .010" for crown shaving.

SECTION 2

fine pitch gear tooth elements (inches)

NORMAL DIAMETRAL PITCH	NORMAL CIRCULAR PITCH	STANDARD CIRCULAR TOOTH THICKNESS	STANDARD ADDENDUM	STANDARD WORK DEPTH	*MINIMUM CLEARANCE	*MINIMUM WHOLE DEPTH	TOTAL SHAVING STOCK ON TOOTH THICKNESS
20	.157080	.0785	.0500	.1000	.020	.120	.0005 to .0015
22	.142800	.0714	.0454	.0908	.020	.111	
24	.130900	.0654	.0417	.0834	.020	.103	
26	.120830	.0604	.0385	.0770	.019	.096	
28	.112200	.0561	.0357	.0714	.019	.090	
30	.104720	.0524	.0333	.0666	.018	.085	
32	.098175	.0491	.0312	.0624	.018	.080	
36	.087266	.0436	.0278	.0556	.016	.072	
40	.078540	.0393	.0250	.0500	.015	.065	
44	.071400	.0357	.0227	.0454	.014	.059	
48	.065450	.0327	.0208	.0416	.013	.055	
52	.060415	.0302	.0192	.0384	.012	.050	.0003 to .0007
56	.056100	.0281	.0178	.0356	.011	.047	
60	.052360	.0262	.0167	.0334	.010	.043	
64	.049087	.0245	.0156	.0312	.010	.041	
72	.043633	.0218	.0139	.0278	.010	.038	

***Based on standard tip radius on preshaving hobs and shaper cutters. Increase whole depth by approximately .005" to .010" for crown shaving.**

SECTION 3

standard stub gear tooth elements (inches)

NORMAL DIAMETRAL PITCH	NORMAL CIRCULAR PITCH	STANDARD CIRCULAR TOOTH THICKNESS	STANDARD ADDENDUM	STANDARD WORK DEPTH	*MINIMUM CLEARANCE	*MINIMUM WHOLE DEPTH	TOTAL SHAVING STOCK ON TOOTH THICKNESS
3/4	1.047198	.5236	.2500	.5000	.083	.583	.0040/.0050
4/5	.785398	.3927	.2000	.4000	.068	.468	.0035/.0045
5/7	.628319	.3142	.1429	.2858	.056	.342	.0035/.0045
6/8	.523599	.2618	.1250	.2500	.050	.300	.0030/.0040
7/9	.448799	.2244	.1111	.2222	.044	.266	.0030/.0040
8/10	.392699	.1963	.1000	.2000	.040	.240	.0030/.0040
9/11	.349066	.1745	.0909	.1818	.038	.220	.0025/.0035
10/12	.314159	.1571	.0833	.1666	.035	.202	.0025/.0035
12/14	.261799	.1309	.0714	.1428	.030	.173	.0020/.0030
14/18	.224399	.1122	.0556	.1112	.026	.137	.0020/.0030
16/21	.196350	.0982	.0476	.0952	.023	.118	.0015/.0025
18/24	.174533	.0873	.0417	.0834	.021	.104	.0015/.0025
20/26	.157080	.0785	.0385	.0770	.020	.097	.0010/.0015
22/29	.142800	.0714	.0345	.0690	.020	.089	.0010/.0015
24/32	.130900	.0654	.0313	.0626	.020	.083	.0010/.0015
26/35	.120830	.0604	.0286	.0572	.019	.076	.0010/.0015
28/37	.112200	.0561	.0270	.0540	.019	.073	.0010/.0015
30/40	.104720	.0524	.0250	.0500	.018	.068	.0010/.0015
32/42	.098175	.0491	.0238	.0476	.018	.066	.0010/.0015
34/45	.092400	.0462	.0222	.0444	.017	.061	.0010/.0015
36/48	.087266	.0436	.0208	.0416	.016	.058	.0010/.0015
38/50	.082673	.0413	.0200	.0400	.016	.056	.0010/.0015
40/54	.078540	.0393	.0185	.0370	.015	.052	.0010/.0015

*Based on standard tip radius on preshaving hobs and shaper cutters.
Increase whole depth by approximately .005″ to .010″ for crown shaving.

SECTION 4

standard gear tooth elements (metric system)

NORMAL MODULE	NORMAL CIRCULAR PITCH (MM)	STANDARD CIRCULAR TOOTH THICKNESS (MM)	*MINIMUM ROOT CLEARANCE (MM)	*MINIMUM WHOLE DEPTH (MM)	TOTAL SHAVING STOCK ON TOOTH THICKNESS (MM)	NORMAL DIAMETRAL PITCH	NORMAL CIRCULAR PITCH (INCHES)	STANDARD CIRCULAR TOOTH THICKNESS (INCHES)	STANDARD ADDENDUM (INCHES)	*MINIMUM ROOT CLEARANCE (INCHES)	*MINIMUM WHOLE DEPTH (INCHES)
1.0	3.142	1.571	.390	2.390	.015/.040	25.400	.1237	.0618	.0394	.0154	.0941
1.25	3.927	1.963	.480	2.980	.020/.045	20.320	.1546	.0773	.0492	.0189	.1173
1.50	4.712	2.356	.560	3.560	.025/.050	16.933	.1855	.0928	.0591	.0220	.1402
1.75	5.498	2.749	.640	4.140	.030/.055	14.514	.2164	.1082	.0689	.0252	.1630
2.0	6.283	3.142	.720	4.720	.035/.060	12.700	.2474	.1237	.0787	.0283	.1858
2.25	7.069	3.534	.790	5.290	.040/.065	11.289	.2783	.1391	.0886	.0311	.2083
2.50	7.854	3.927	.860	5.860	.045/.070	10.160	.3092	.1546	.0984	.0339	.2307
2.75	8.639	4.320	.920	6.420	.045/.070	9.2364	.3401	.1701	.1083	.0362	.2528
3.0	9.425	4.712	.990	6.990	.050/.075	8.4667	.3711	.1855	.1181	.0390	.2752
3.25	10.210	5.105	1.050	7.550	.050/.075	7.8154	.4020	.2010	.1280	.0413	.2972
3.50	10.996	5.498	1.110	8.110	.055/.080	7.2571	.4329	.2165	.1378	.0437	.3193
3.75	11.781	5.890	1.160	8.660	.055/.080	6.7733	.4638	.2319	.1476	.0457	.3409
4.0	12.566	6.283	1.220	9.220	.060/.085	6.3500	.4947	.2474	.1575	.0480	.3630
4.25	13.352	6.676	1.270	9.770	.060/.085	5.9765	.5256	.2628	.1673	.0500	.3846
4.50	14.137	7.069	1.320	10.320	.060/.085	5.6444	.5566	.2783	.1772	.0520	.4063
4.75	14.923	7.461	1.370	10.870	.065/.090	5.3474	.5875	.2938	.1870	.0539	.4280
5.0	15.708	7.854	1.420	11.420	.065/.090	5.0800	.6184	.3092	.1969	.0559	.4496
5.25	16.493	8.247	1.470	11.970	.065/.090	4.8381	.6493	.3247	.2067	.0579	.4713
5.50	17.278	8.639	1.520	12.520	.070/.095	4.6182	.6803	.3401	.2165	.0598	.4929
5.75	18.064	9.032	1.560	13.060	.070/.095	4.4174	.7112	.3556	.2264	.0614	.5142
6.0	18.850	9.425	1.610	13.610	.070/.095	4.2333	.7421	.3711	.2362	.0634	.5358
6.5	20.420	10.210	1.710	14.710	.070/.095	3.9077	.8040	.4020	.2559	.0673	.5791
7.0	21.991	10.996	1.820	15.820	.075/.100	3.6286	.8658	.4239	.2756	.0717	.6228
7.5	23.562	11.781	1.920	16.920	.075/.100	3.3867	.9276	.4638	.2953	.0756	.6661
8.0	25.133	12.566	2.020	18.020	.075/.100	3.1750	.9895	.4947	.3150	.0795	.7094
9.0	28.274	14.137	2.240	20.240	.080/.105	2.8222	1.1132	.5566	.3543	.0882	.7968
10.0	31.416	15.708	2.440	22.440	.080/.105	2.5400	1.2368	.6184	.3937	.0960	.8835

***Based on standard tip radius on preshaving hobs and shaper cutters.**
Increase whole depth by approximately .130 mm to .250 mm (.005" to .010") for crown shaving.

SECTION 5

for spur gears

TO FIND:	ENGLISH (inches)	METRIC (Millimeters)
1. Pitch Diameter (D)	$\frac{N}{P}$	mN
2. Addendum (a)	$\frac{1}{P}$	Module (m) (In Millimeters and Parts of Millimeters)
3. Standard Outside Diameter (D_O)	$D + (2a)$	$D + (2m)$
4. Base Diameter (D_b)	$D \cos \phi$	
5. Circular Pitch (p)	$\frac{\pi}{P}$	πm
6. Standard Circular Tooth Thickness (t)	$\frac{p}{2}$	
7. Average Backlash Per Pair (B)	$\frac{.040}{P}$	$.040m$
8. Tooth Depth (h_t)	See Sections 1 thru 4	
9. Root Diameter (D_R)	$D_o - (2h_t)$	

Conversions

Diametral Pitch (P) $= \frac{25.4}{m}$

Module (m) $= \frac{25.4}{P}$

Millimeters (mm) $= \frac{\text{Inches}}{.03937} = 25.4$ Inches

Inches $= .03937$ mm $= \frac{\text{mm}}{25.4}$

SECTION 6

helical gear equations

1. Transverse Diametral Pitch (P_t) $= P_n \cos\psi$

2. Pitch Diameter (D) $= \dfrac{N}{P_t}$

3. Addendum (a), Standard $= \dfrac{1}{P_n}$

4. Outside Diameter (D_o) $= D + 2a$

5. Transverse Pressure Angle (ϕ_t):

$$\tan\phi_t = \frac{\tan\phi_n}{\cos\psi}$$

6. Base Diameter (D_b) $= D\cos\phi_t$

7. Lead (L) $= \pi D \cot\psi = \dfrac{\pi D}{\tan\psi}$

8. Normal Circular Pitch (p_n) $= \dfrac{\pi}{P_n}$

9. Standard Normal Circular Tooth Thickness (t_n) $= \dfrac{p_n}{2}$

10. Axial Pitch (p_x) $= \dfrac{\pi}{P_n \sin\psi} = \dfrac{p_n}{\sin\psi} = \dfrac{L}{N}$

11. Transverse Circular Pitch (p_t) $= \dfrac{\pi}{P_t} = \dfrac{p_n}{\cos\psi}$

See Sections 1, 2, 3 and 4 for whole tooth depth (h_t)

SECTION 7

miscellaneous gear equations

1. Helix Angle (ψ), when given center distance is standard:

$$\cos\psi = \frac{N_p + N_G}{2p_nC}$$

2. Operating Pitch Diameter (Pinion); with non-standard center distance (C):

$$D_{rp} = \frac{2CN_p}{N_p + N_G}$$

3. Operating Pressure Angle (ϕ_{rt}); with non-standard center distance (C):

$$\cos\phi_{rt} = \frac{D_{bp} + D_{bG}}{2C}$$

4. Normal Diametral Pitch (P_n) $= P_t \sec\psi$

5. Helix Angle (ψ):

$$\cos\psi = \frac{N}{D\,P_n}$$

$$\sin\psi = \frac{\pi N}{P_n L}$$

At any diameter, D_2, $\tan\psi_2 = \dfrac{D_2 \tan\psi_1}{D_1}$

6. Transverse Circular Pitch (p_{t2}) at any Dia. (D_2)

$$p_{t2} = \frac{\pi D_2}{N}$$

7. Involute Function of Pressure Angle:

$$\text{inv}\,\phi = \tan\phi - \widehat{\phi}$$

Note: Tables of involute functions are available.

8. Normal Pressure Angle (ϕ_n):

$$\tan\phi_n = \tan\phi_t \cos\psi$$

9. Transverse Pressure Angle (ϕ_t) at Any Diameter (D_2).

$$\cos\phi_{t2} = \frac{D_b}{D_2}$$

10. Base Helix Angle (ψ_b):

$$\cos\psi_b = \frac{\cos\psi\cos\phi_n}{\cos\phi_t} = \frac{\sin\phi_n}{\sin\phi_t}$$

$$\sin\psi_b = \sin\psi\cos\phi_n$$

$$\tan\psi_b = \tan\psi\cos\phi_t$$

11. Base Pitch (p_b) $= \dfrac{\pi D_b}{N} = p\cos\phi$

SECTION 8

circular tooth thickness when dimension over two or more teeth is given (Fig. 2-1)

FOR HELICAL GEARS

1. Find helix angle cosine at base circle ($\cos\psi_b$)

$$\cos\psi_b = \frac{\sin\phi_n}{\sin\phi_t}$$

2. Find dimension over teeth in plane of rotation (M_t)

$$M_t = \frac{M_n}{\cos\psi_b}$$

3. Find transverse circular tooth thickness (t_t) at pitch diameter

$$t_t = D\left(\frac{M_t}{D_b} - \frac{\pi N_s}{N_G} - \text{inv}\,\phi_t\right)$$

where N_s = Number of tooth spaces

4. Find normal circular tooth thickness (t_n)

$$t_n = t_t \cos\psi$$

SECTION 9

circular tooth thickness at theoretical pitch diameter of oversize or undersize gear when running with a standard mate

FOR HELICAL GEARS

All circular dimensions must be calculated in trans. plane

1. Find operating pitch diameters (D_{rG}, D_{rP})

$$D_{rG} = 2C \frac{N_G}{N_G + N_P},\ D_{rP} = 2C \frac{N_P}{N_G + N_P}$$

2. Find pressure angle (ϕ_{tr}) and circular pitch (p_{tr}) at operating pitch diameter (D_r). Same for gear and pinion.

$$\cos \phi_r = \frac{D_{bG}}{D_{rG}} = \frac{D_{bP}}{D_{rP}}$$

$$p_r = \frac{\pi D_{rG}}{N_G} = \frac{\pi D_{rP}}{N_P}$$

3. Find tooth thickness (t_{rP}) of standard mate at operating pitch diameter (D_{rP}). Tooth thickness of standard mate (t_t) at some diameter (D) must be known.

$$t_{trP} = D_{rP}\left(\frac{t_t}{D} + \text{inv } \phi_t - \text{inv } \phi_{tr}\right)$$

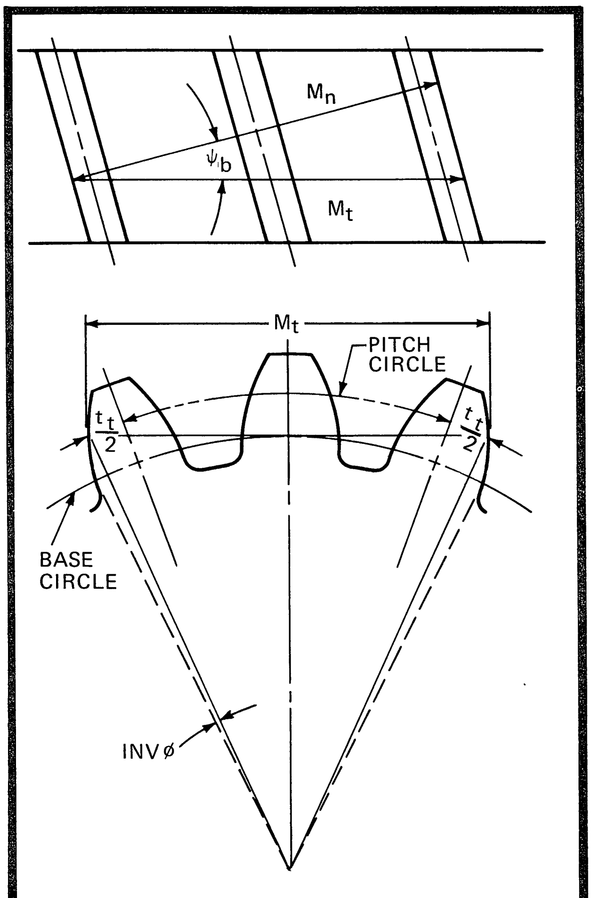


Fig. 2-1—Circular tooth thickness when dimension over two or more teeth is given.

4. Find tooth thickness (t_{trP}) of oversize or undersize gear at operating pitch diameter (D_{rG})

$$t_{trG} = p_{tr} - (t_{trP} + B_t)$$

5. Find tooth thickness (t_{tG}) of oversize or undersize gear at theoretical pitch diameter (D_G)

$$t_{tG} = D_G\left(\frac{t_{trG}}{D_{rG}} + \text{inv } \phi_{tr} - \text{inv } \phi_t\right)$$

6. Find normal tooth thickness of gear

$$t_{nG} = t_{tG} \cos \psi_G$$

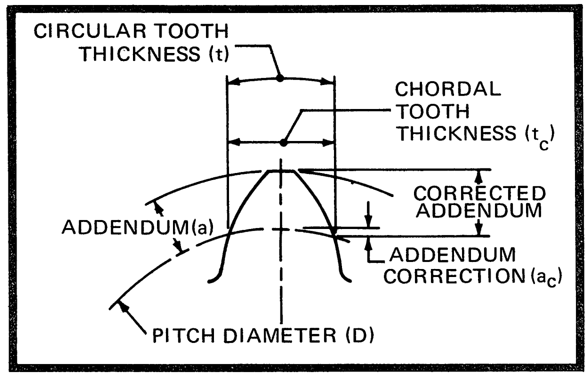


Fig. 2-2—Corrections for gear tooth calipers.

SECTION 10

corrections for gear tooth calipers (Fig. 2-2)

FOR SPUR GEARS

1. Corrected Addendum $(a_c) = a + \frac{t^2}{4D}$

2. Chordal Tooth Thickness $(t_c) = t - \frac{t^2}{4DN}$

FOR HELICAL GEARS

1. Corrected Addendum $(a_c) = a + \frac{(t_n \cos\psi)^2}{4D}$

2. Normal Chordal Tooth Thickness

$$(t_{nc}) = t_n - \frac{(t_n \cos\psi)^2}{4DN}$$

Note: These formulas are approximations, but the errors involved are too small to be measured with ordinary gear tooth calipers on most gears.

SECTION 11

dimension over two pins or balls

FOR SPUR GEARS

1. Tooth Space $(S) = p - t$

2. Pin Diameter $(d) = \frac{1.728}{P}$ (Approx.)

3. Involute ϕ_m = Involute Function for Pressure Angle to Center of Pin

$$\text{inv } \phi_m = \text{inv } \phi + \frac{\frac{d}{\cos \phi} - S}{D}$$

or

$$\text{inv } \phi_m = \text{inv } \phi + \frac{d}{D_b} - \frac{S}{D}$$

From Involute Function Tables find ϕ_m in degrees

From Trigonometric Function Tables find $\cos \phi_m$

4. For Gears With Even Number of Teeth:

Dimension over two pins $(D_{Me}) = \frac{D_b}{\cos \phi_m} + d$

5. For Gears With Odd Number of Teeth:

Dimension over two pins (D_{Me}):

$$D_{Me} = \frac{D_b \cos \frac{90°}{N}}{\cos \phi_m} + d$$

SECTION 12

dimension over two balls

FOR HELICAL GEARS

1. Normal Circular Tooth Space $(S_n) = p_n - t_n$

2. Ball Diameter $(d) = \frac{1.728}{P_n}$ (Approx.)

3. Involute ϕ_m = Involute Function for Transverse Pressure Angle to Center of Ball

$$\text{inv } \phi_m = \text{inv } \phi_t + \frac{\frac{d}{\cos \phi_n} - S_n}{\frac{N}{P_n}}$$

Note: $\frac{N}{P_n}$ is not the Pitch Diameter

From Involute Function Tables find ϕ_m in degrees

From Trigonometric Function Tables find $\cos \phi_m$

4. For Gears With Even Number of Teeth:

Dimension over two balls $(D_{Me}) = \frac{D_b}{\cos \phi_m} + d$

5. For Gears With Odd Number of Teeth:

Dimension over two balls (D_{Me}):

$$D_{Me} = \frac{D_b \cos \frac{90°}{N}}{\cos \phi_m} + d$$

SECTION 13

dimension between two pins or balls

FOR INTERNAL SPUR GEARS

1. Circular Space $(S) = p - t$

2. Pin Diameter $(d) = \frac{1.44}{P}$ (Approx.)

3. Involute ϕ_m = Involute Function of Pressure Angle to Center of Pin

$$\text{inv } \phi_m = \text{inv } \phi - \frac{\frac{d}{\cos \phi} - S}{D}$$

or

$$\text{inv } \phi_m = \text{inv } \phi - \frac{d}{D_b} + \frac{S}{D}$$

From Involute Function Tables find ϕ_m in degrees

From Trigonometric Function Tables find $\cos \phi_m$

4. For Gears With Even Number of Teeth:

Dimension between two Pins (D_{Mi}):

$$D_{Mi} = \frac{D_b}{\cos \phi_m} - d$$

5. For Gears With Odd Number of Teeth:

Dimension between two Pins (D_{Mi}):

$$D_{Mi} = \frac{D_b \cos \frac{90°}{N}}{\cos \phi_m} - d$$

SECTION 14

dimension between two balls

FOR INTERNAL HELICAL GEARS

1. Normal Circular Space $(S_n) = p_n - t_n$

2. Ball Diameter $(d) = \frac{1.44}{P_n}$ (Approx.)

3. Involute ϕ_m = Involute Function of Pressure Angle to Center of Ball.

$$\text{inv } \phi_m = \text{inv } \phi_t - \frac{\frac{d}{\cos \phi_n} - S_n}{\frac{N}{P_n}}$$

Note: $\frac{N}{P_n}$ is not the Pitch Diameter

From Involute Function Tables find ϕ_m in degrees.

From Trigonometric Function Tables find $\cos \phi_m$.

4. For Gears with Even Number of Teeth:
Dimension between two Balls (D_{Mi}):

$$D_{Mi} = \frac{D_b}{\cos \phi_m} - d$$

5. For Gears with Odd Number of Teeth:
Dimension between two Balls (D_{Mi}):

$$D_{Mi} = \frac{D_b \cos \frac{90^\circ}{N}}{\cos \phi_m} - d$$

SECTION 15

maximum outside diameter without involute interference (fig. 2-3)

FOR EXTERNAL GEARS

Max. Theoretical Outside Diameter Without involute Interference $= \sqrt{D_b^2 + (2C \sin \phi_{tr})^2}$

In practice it is advisable to use an outside diameter at least .010″ to .020″ smaller than the theoretical value to eliminate dangerous contacts in the vicinity of the base diameter of the mating gear.

SECTION 16

length of contact and contact ratio

FOR EXTERNAL GEARS

1. Length of Contact (Z):

$$Z = \frac{\sqrt{D_{oG}^2 - D_{bG}^2} + \sqrt{D_{oP}^2 - D_{bP}^2}}{2} - C \sin \phi_{tr}$$

2. Involute Contact Ratio $(m_p) = \frac{Z}{p_{bt}}$

3. Face Contact Ratio (m_F):

$$m_F = \frac{F \tan \psi}{p_t} = \frac{F \sin \psi}{p_n}$$

SECTION 17

start of active profile and contact diameter (fig. 2-4)

FOR EXTERNAL GEARS

1. Total Length of Line of Action (LA):

$$LA = \frac{\sqrt{(2C)^2 - (D_{bG} + D_{bP})^2}}{2}$$

or

$$LA = C \sin \phi_{tr}$$

2. Start of Active Profile (SAP_P):

$$SAP_P = LA - \frac{\sqrt{D_{oG}^2 - D_{bG}^2}}{2}$$

3. Start of Active Profile (SAP_G):

$$SAP_G = LA - \frac{\sqrt{D_{oP}^2 - D_{bP}^2}}{2}$$

4. Contact Diameter $(D_c) = \sqrt{(2\ SAP)^2 + D_b^2}$

The start of Active Profile is measured in inches along the Line of Action from zero (Base Diameter). For conversion to degrees of roll use equations in Section 22.

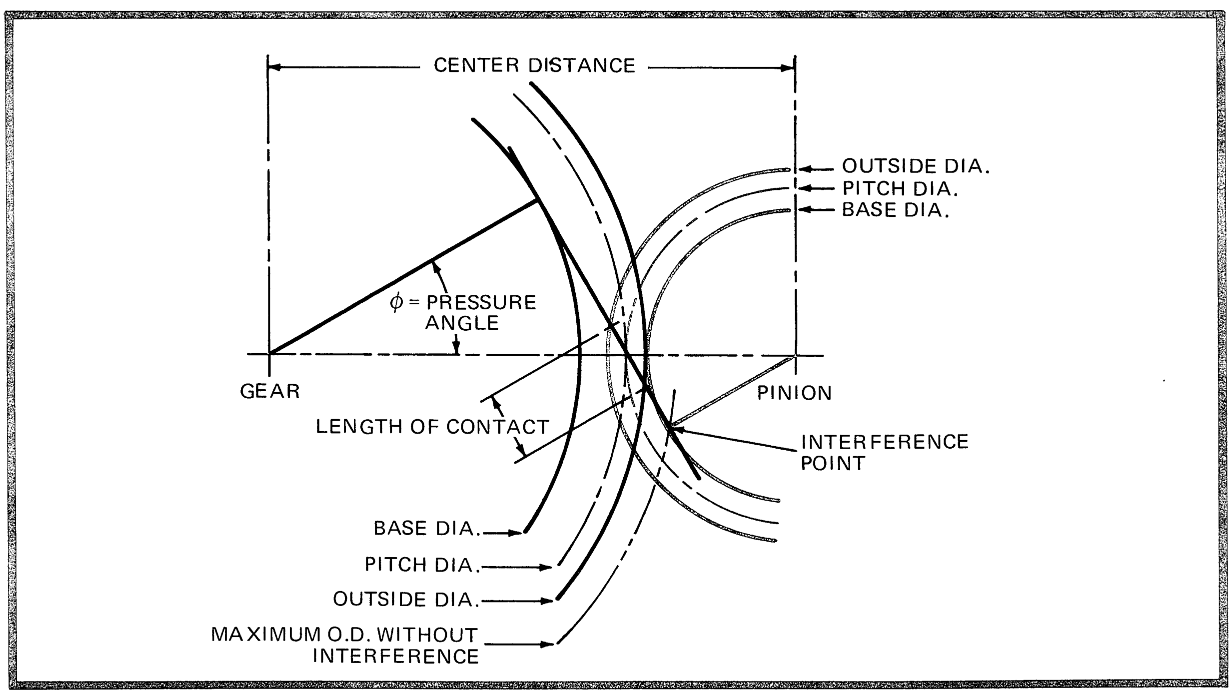


Fig. 2-3—Maximum outside diameter without interference.

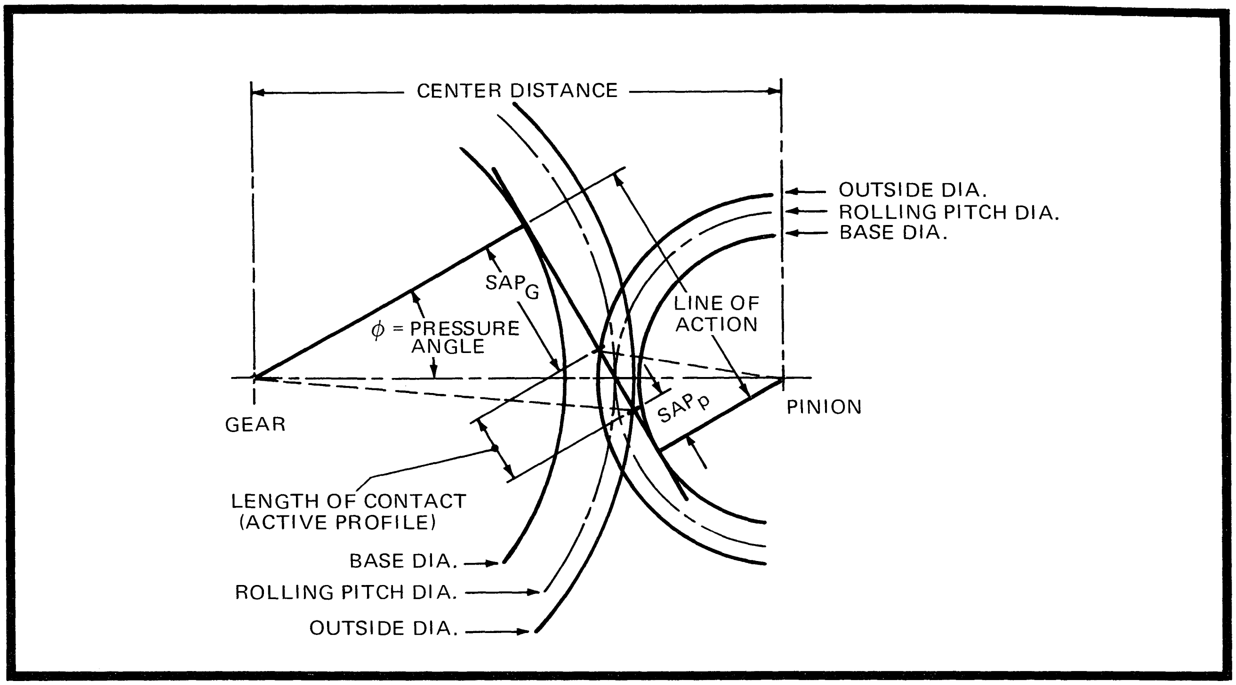


Fig. 2-4—Start of active profile and contact diameter.

SECTION 18

minimum minor diameter without involute interference (fig. 2-5)

FOR INTERNAL SPUR AND HELICAL GEARS

Involute interference must not be confused with tip interference or trimming.

Min. Theoretical Minor Diameter (D_i) without Interference $= \sqrt{D_{bi}^2 + (2C \sin \phi_{tr})^2}$

In practice it is advisable to use a minor diameter at least .010″ to .020″ larger than the theoretical value to eliminate dangerous contacts in the vicinity of the base diameter of the mating gear.

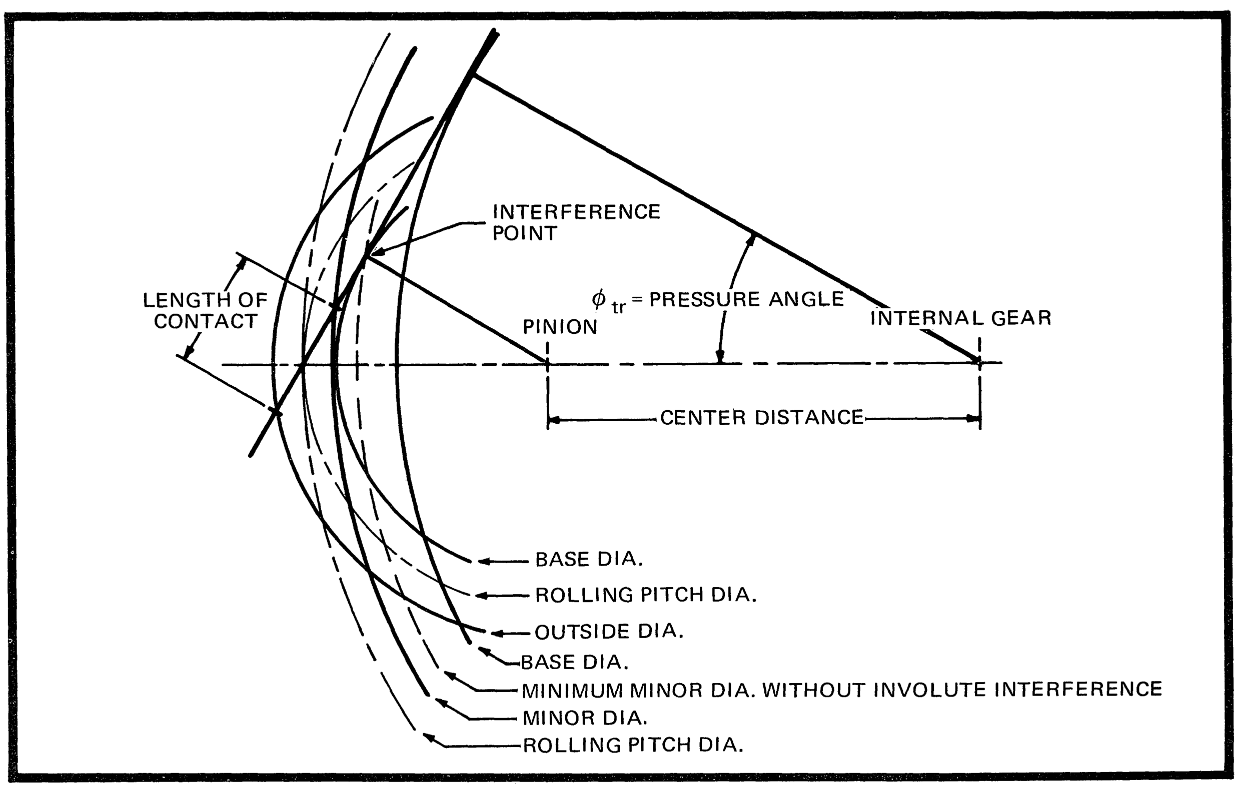


Fig. 2-5—Minimum minor diameter without involute interference.

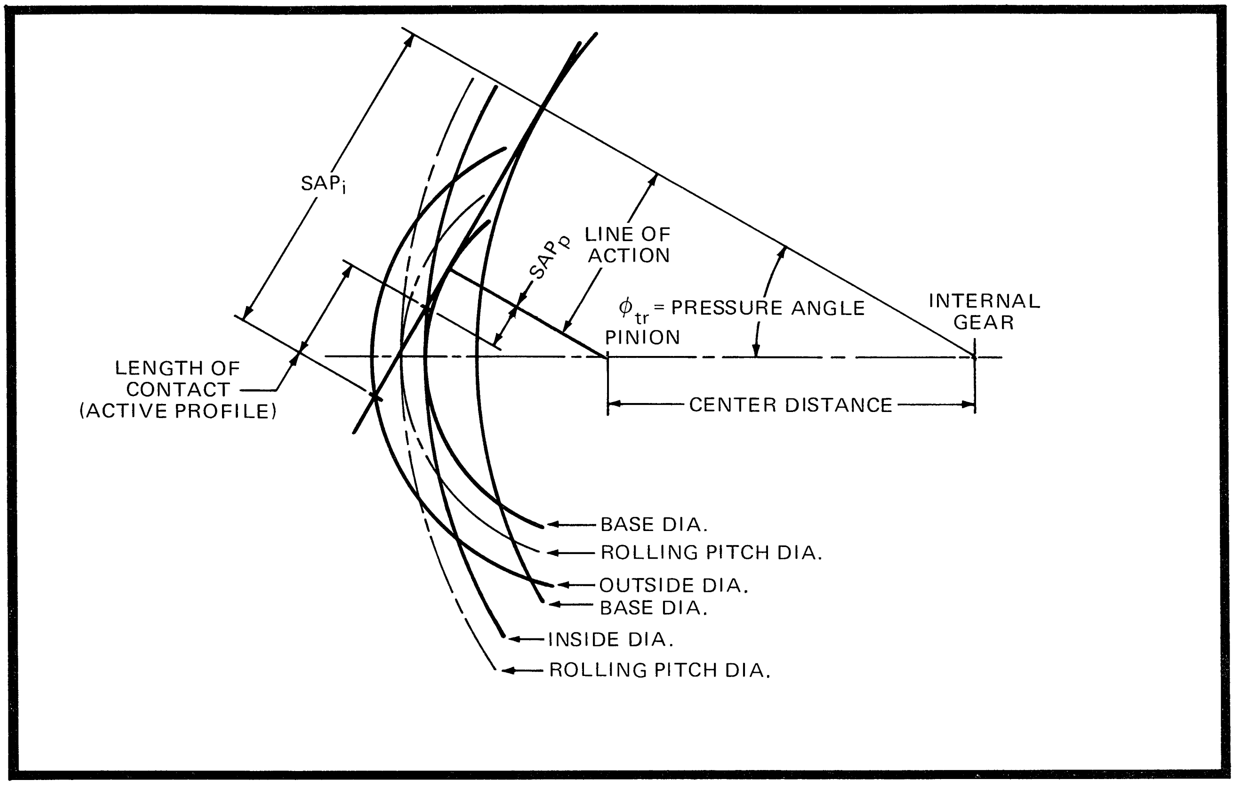


Fig. 2-6—Start of active profile and contact diameter for internal gears.

SECTION 19

length of contact and contact ratio

FOR INTERNAL GEARS

1. Length of Contact (Z):

$$Z = \frac{\sqrt{D_{op}^2 - D_{bP}^2}}{2} + C \sin \phi_{tr} - \frac{\sqrt{D_i^2 - D_{bi}^2}}{2}$$

2. Involute Contact Ratio (m_p) $= \dfrac{Z}{p_{bt}}$

SECTION 20

start of active profile and contact diameter (fig. 2-6)

FOR INTERNAL GEARS

1. Total Length of Line of Action (LA):

$$LA = \frac{\sqrt{(2C)^2 - (D_{bi} - D_{bP})^2}}{2}$$

or

$$LA = C \sin \phi_{tr}$$

2. Start of Active Profile (SAP_P):

$$SAP_P = \frac{\sqrt{D_i^2 - D_{bi}^2}}{2} - LA$$

3. Start of Active Profile (SAP_i):

$$SAP_i = \frac{\sqrt{D_{oP}^2 - D_{bP}^2}}{2} + LA$$

4. Contact Diameter (D_c) $= \sqrt{(2\ SAP)^2 + D_b^2}$

The Start of Active Profile is measured in inches along the Line of Action from zero (Base Diameter). For conversion to degrees of roll use equations in Section 22.

SECTION 21

start of active profile and contact diameter

FOR RIGHT-ANGLE DRIVE GEARS

1. Start of Pinion Active Profile (SAP_P):

$$SAP_P = \frac{U_1 - \dfrac{(U_4 - U_3) \cos \psi_{bP}}{\cos \psi_{bG}}}{2}$$

2. Start of Gear Active Profile (SAP_G):

$$SAP_G = \frac{U_3 - \dfrac{(U_2 - U_1) \cos \psi_{bG}}{\cos \psi_{bP}}}{2}$$

where U_1, U_2, U_3 and U_4 are calculated as follows to simplify equations 1 and 2.

$$U_1 = \sqrt{D_{rP}^2 - D_{bP}^2} \qquad U_2 = \sqrt{D_{oP}^2 - D_{bP}^2}$$

$$U_3 = \sqrt{D_{rG}^2 - D_{bG}^2} \qquad U_4 = \sqrt{D_{oG}^2 - D_{bG}^2}$$

3. Pinion Contact Diameter (D_{cP}):

$$D_{cP} = \sqrt{(2\ SAP_P)^2 + D_{bP}^2}$$

4. Gear Contact Diameter (D_{cG}):

$$D_{cG} = \sqrt{(2\ SAP_G)^2 + D_{bG}^2}$$

The Start of Active Profile is measured in inches along the Line of Action from zero (Base Diameter). For conversion to degrees of roll use equations in Section 22.

SECTION 22

conversion of dimensions on the line of action

From inches to Degrees:

$$\text{Degrees Roll} = \frac{360^\circ\ (\text{Length in Inches})}{\pi D_b}$$

From Degrees to Inches:

$$\text{Length in Inches} = \frac{\pi D_b\ (\text{Degrees Roll})}{360^\circ}$$

SECTION 23

operating helix angles, operating normal diametral pitch and basic helix angles

FOR SPIRAL GEARS WITH AXES CROSSED AT 90 DEG.

With numbers of teeth, N_1 and N_2, fixed center distance, C, normal diametral pitch, $\check{P}_n$, normal pressure angle, ϕ_n, and helix angle, ψ_1 known

1. Operating Helix Angles (ψ_{r1}, ψ_{r2}):

$$\tan \psi_{r1} = \frac{(2CP_n \sin \psi_1) - N_2}{N_1}$$

$$\psi_{r2} = 90 - \psi_{r1}$$

2. Normal Operating Diametral Pitch (P_{nr}) for Both Gears:

$$P_{nr} = \frac{P_n \sin \psi_1}{\sin \psi_{r1}}$$

3. Normal Operating Pressure Angle (ϕ_{nr}) for Both Gears:

$$\cos \phi_{nr} = \frac{\sin \psi_1 \cos \phi_n}{\sin \psi_{r1}}$$

4. Operating or Rolling Pitch Diameters (D_{r1}, D_{r2}):

$$D_{r1} = \frac{N_1}{P_{nr} \cos \psi_{r1}}$$

$$D_{r2} = \frac{N_2}{P_{nr} \cos \psi_{r2}}$$

Backlash in any form is calculated at these diameters.

5. Actual Center Distance (C) for Checking Purposes:

$$C = \frac{D_{r1} + D_{r2}}{2}$$

6. Basic Helix Angle ψ_2 of Gear N_2:

$$\sin \psi_2 = \frac{P_{nr} \sin \psi_{r2}}{P_n}$$

Note: The basic helix angles (ψ) are *not* the *base* helix angles (ψ_b) of these spiral gears. They are the helix angles at the normal diametral pitch (P_n) and the normal pressure angle (ϕ_n) of the basic rack (hob) which generates the teeth.

After ψ_{r1}, ψ_{r2}, D_{r1}, D_{r2} and ψ_2 have been determined, both gears may be treated as separate spiral gears. The normal characteristics, P_n and ϕ_n (at ψ_1 and ψ_2), and P_{nr}, ϕ_{nr} and p_{nr} (at ψ_{r1} and ψ_{r2}) will be identical. The transverse plane characteristics, P_t, ϕ_t, P_{tr}, ϕ_{tr} and p_{tr}, will be different on each gear.

If backlash is considered, it must be taken at the operating pitch diameters, D_{r1} and D_{r2}.

There is only one pair of operating helix angles, ψ_{r1} and ψ_{r2}, for a given ratio and center distance which will allow the gears to contact on the centerline of crossed axes. Any other combination will force contact to one side or the other of the centerline.

Two important limiting factors in this type of spiral gear design are:

1. Pointed teeth on extremely oversize pinions
2. Excessive undercut of undersize pinions

SECTION 24

shaving cutter analysis to determine the compatibility of a gear and a shaving cutter design

Rotary shaving cutters for finishing gear teeth are normally designed for a specific gear or range of gears. However, when a plant accumulates a variety of these cutters, the investment prompts study of their potential use on new gear designs.

There is a definite procedure for such analysis. Calculations made with the equations given will show whether or not the proportions of the shaving cutter tooth are such that the gear will be shaved to the correct depth and size.

It is not possible to determine whether the shaving cutter analyzed will produce a satisfactory profile on the gear. After all, a shaving cutter performs best and provides the most accurate lead and profile forms when it shaves the gear or gears for which it was designed.

1. Required Information—The data listed below must be known for each gear and shaving cutter under consideration. Subscripts $_G$ and $_c$ denote gear and shaving cutter.

Symbol	*Description*
P_n	Generating normal diametral pitch
ϕ_n	Generating normal pressure angle
N_G, N_c	Number of teeth
ψ_G, ψ_c	Helix angle with direction (hand) at the generating normal diametral pitch
D_G, D_c	Generating pitch diameter in the transverse plane
ϕ_{tG}, ϕ_{tc}	Transverse pressure angle at the generating pitch diameter
D_{bG}, D_{bc}	Base diameter
t_{nG}, t_{nc}	Normal circular tooth thickness at the generating pitch diameter
D_{oG}, D_{oc}	Outside or limiting diameters
SAP_G	Start of active profile on the gear measured along the line of action from the base diameter
D_{RG}	Maximum root diameter of gear
SRP_G	Height of generated root fillet on gear measured along the line of action from the base diameter.

Because only existing shaving cutters will be considered, it will be necessary to obtain the number of cutter teeth (N_c), helix angle (ψ_c), normal circular tooth thickness (t_{nc}) and outside diameter (D_{oc}) from the cutter. The N_c and ψ_c values are usually stamped on rotary shaving cutters or they can be obtained from tool sheets supplied. The D_{oc} and t_{nc} dimensions should be taken directly from the shaving cutter by measurement. Circular-tooth thickness of cutter (t_{nc}) is best determined by a check over two pins.

It is also necessary to check the depth of the serrations on the profile of the shaving cutter. This can be a visual check to determine if the remaining tool life is sufficient for possible redevelopment and extensive use.

2. Classify Gears to Be Shaved—Arrange the gears in ascending or descending order of normal circular-tooth thickness at the generating pitch diameter.

3. Determine Over and Undersize Gears—After making the above classification, use equation (1) to determine the percentage of over- and undersize of the gears based on the basic-rack principle.

$$\text{Percentage Factor} = \frac{\triangle t_{nG} \times P_n}{2 \times \tan \phi_n} \qquad (1)$$

Where: $\triangle t_{nG}$ is the difference between the actual normal circular-tooth thickness at the generating-pitch diameter and one-half the normal circular pitch less one-half the required normal backlash between mating parts. If t_{nG} is the greater, the gear is oversize.

$$\pm \triangle t_{nG} = t_{nG} - \left[\frac{\pi}{2P_n} - \frac{\text{Nor. Backlash}}{2}\right] \qquad (2)$$

The percentages obtained from equation (1) are reference values used for determining the magnitude of the oversize or undersize conditions prevalent in the gears. These values can be charted, if desired. However, the usual procedure is to first group the gears with ranges approximately as shown in the chart and then in ascending order of the percentage factor computed from equation (1).

Table 2-1—Tooth Ranges of Shaving Cutters for Jobbing Work

Diametral Pitch	RANGE NUMBER 1	2	3	4	5
	Teeth in Gears				
4 and 5	15 to 17	17 to 25	24 to 37	35 to 60	55 and up
6 and 7	15 to 17	17 to 26	23 to 40	35 to 60	55 and up
8 to 10	14 to 18	17 to 29	27 to 60	60 and up	
12 to 18	14 to 18	17 to 34	25 to 60	60 and up	
20 to 30	13 to 22	20 and up			
32 to 64	12 to 24	20 and up			

Note: Numbers of teeth smaller than those shown will require special consideration.

4. Select the Shaving Cutter:

(*a*) For the gears under consideration, P_{nc} and ϕ_{nc} must be identical to P_{nG} and ϕ_{nG}.

(*b*) Consider the crossed-axes angle. The approximate angle is obtained by taking the algebraic sum of the gear and cutter helix angles. For unlike hands the crossed-axes angle is equal to the difference of the helix angles; for like hands the angle is equal to the sum of the helix angles. On open gears with no crossed-axes restrictions, best results are obtained with crossed axes from 10° to 15°. However, the range can be extended from 3° to 18° in some cases, if knowledge of the results obtained from such practice is known. Gears with restricting shoulders must be treated separately, wherein the maximum crossed-axes angle allowable is obtained by layout or computation.

(*c*) If possible determine the t_{nG} and percentage factor (1) of the gear for which the cutter was originally designed. This knowledge is helpful when deciding upon the cutter design, which is most likely to be successful on the gears under consideration. This is true especially if the percentage factor is close and the number of teeth falls within the proposed group range. Usually gears of one group falling within a 50% bracket can be shaved with the same cutter (0 to +50%, 0 to −50%, etc.). Best results are obtained by keeping over- and undersized gears on separate cutters.

(*d*) Depending upon the number of available shaving cutters, and the number of gears involved in a given group, it is usually best to match a cutter with a gear whose characteristics are close to the gear for which the cutter was designed. If the meshing computations have shown that the cutter will meet the physical requirements of the gear, the same cutter design is then meshed with the other gears in the group range.

5. Compute Tight-Mesh Center Distance—After the shaving cutter and gear have been selected, it is necessary to compute the tight-mesh center distance between the two components. For this purpose equations follow. Because these equations were originally written for two gears, it becomes necessary to substitute the subscript $_G$ denoting the gear for the subscript $_1$ in the equations. Likewise the subscript $_c$ denoting the cutter must be substituted for the subscript $_2$. Therefore, $N_1 = N_G$ and $N_2 = N_c$ etc.

6. Equations for Tight-Meshed Gears:

Purpose: To find transverse operating-pressure angles of a given pair of involute gears when operating in tight mesh with their axes crossed.

NOMENCLATURE

Transverse Planes

N_1 and N_2	—Number of teeth
D_1 and D_2	—Given diameters
t_1 and t_2	—Circular-tooth thickness at D_1 & D_2
ϕ_1 and ϕ_2	—Pressure angle at D_1 & D_2
ψ_1 and ψ_2	—Helix angle at D_1 & D_2
ψ_{b1} and ψ_{b2}	—Base helix angle
ϕ_{r1} and ϕ_{r2}	—Operating pressure angle
ϕ_{t1} and ϕ_{t2}	—Transverse pressure angle at generating pitch diameter

Then follow these seven steps:

1 . . .

$$E = N_1\left[\frac{t_1}{D_1} + \text{inv } \phi_1\right] + N_2\left[\frac{t_2}{D_2} + \text{inv } \phi_2\right] - \pi$$

2 . . .

$$\text{inv } \phi_{r1} = \frac{E}{N_1 + N_2\left[\dfrac{\text{inv } \phi_{t2}}{\text{inv } \phi_{t1}}\right]}$$

Find sin ϕ_{r1}

3 . . .

$$\sin \phi_{r2} = \sin \phi_{r1} \frac{\cos \psi_{b1}}{\cos \psi_{b2}}$$

Find inv ϕ_{r2}

4 . . . First approximate check:

$$N_1 \text{ inv } \phi_{r1} + N_2 \text{ inv } \phi_{r2} \risingdotseq E$$

Note: Generally, the accuracy obtained with operating pressure angles ϕ_{r1} and ϕ_{r2} will be satisfactory if equation (4) differs from E by no more than 0.0010 in. However, if the difference is greater than 0.0010 in. or if greater accuracy is desired, it will be necessary to proceed as follows:

4A . . .

$$\text{inv } \phi'_{r1} = \frac{E}{N_1 + N_2\left[\dfrac{\text{inv } \phi_{r2}}{\text{inv } \phi_{r1}}\right]}$$

Find sin ϕ'_{r1}

4B . . .

$$\sin \phi'_{r2} = \sin \phi'_{r1} \frac{\cos \psi_{b1}}{\cos \psi_{b2}}$$

Find inv ϕ'_{r2}

4C . . . Check:

$$N_1 \text{ inv } \phi'_{r1} + N_2 \text{ inv } \phi'_{r2} \risingdotseq E$$

Note: Still greater accuracy may be obtained by repeating step 4A, using involute ϕ'_{r1} and ϕ'_{r2}. This is seldom necessary except perhaps when high crossed axes or high helix angles are being used.

5 . . . Then:

D_{b1} and D_{b2}	—Transverse base circle diameters
D_{r1} and D_{r2}	—Operating pitch diameters
ψ_{r1} and ψ_{r2}	—Operating helix angles
t_{nr1} and t_{nr2}	—Normal operating circular tooth thickness
p_{nr}	—Normal operating circular pitch

5A . . .

The following equations are used for both members:

$$D_r = \frac{D_b}{\cos \phi_r}$$

$$\tan \psi_r = \frac{D_r \tan \psi}{D}$$

Find cos ψ_r

$$t_{nr} = D_r \cos \psi_r \left[\frac{t}{D} + \text{inv } \phi - \text{inv } \phi_r\right]$$

$$^*p_{nr} = \frac{\pi D_r \cos \psi_r}{N}$$

*(Same on Both Members)

6 . . . Then:

$$t_{nr1} + t_{nr2} \risingdotseq p_{nr}$$

7 . . .

Normal distance between axes at the centerline of crossed axes (center distance):

$$C = \frac{D_{r1} + D_{r2}}{2}$$

Note: Closeness to equality in equation (6) determines the accuracy of the center distance from equation (7). For general practice, the summation of t_{nr1} and t_{nr2} should equal the normal operating circular pitch within ± 0.0005 in. However, almost any degree of accuracy may be obtained by repetition of step 4A (see step 4C) until the desired accuracy is obtained.

7. Compute *H* Factor—In order to transfer dimensions from the gear transverse plane to the cutter transverse plane, or in reverse order, compute the *H* factor. This is a function of the base helix angles:

$$H \text{ factor} = \frac{\cos \psi_{bG}}{\cos \psi_{bc}}$$

8. Line-of-Action Distance—Distance along the line of action in the transverse plane from the base diameter to:

(*a*) Operating pitch point $= A_r$

$$A_{rG} = D_{rG} \sin \phi_{rG}$$
$$A_{rc} = D_{rc} \sin \phi_{rc}$$

(*b*) Outside or limiting diameter $= A_o$

$$A_{oG} = \sqrt{(D_{oG})^2 - (D_{bG})^2}$$
$$A_{oc} = \sqrt{(D_{oc})^2 - (D_{bc})^2}$$

9. Lowest Point of Contact—Lowest points of contact in transverse planes for gear and cutter are:

(*a*) On gear = FIN

$$\text{FIN} = \frac{A_{rG} - H\,(A_{oc} - A_{rc})}{2}$$

$$D_{\text{FIN}} = \sqrt{(D_{bG})^2 + (2\ \text{FIN})^2}$$

(*b*) On cutter $= SAP_c$

$$SAP_c = \tfrac{1}{2}\left[A_{rc} - \frac{(A_{oG} - A_{rG})}{H}\right]$$

$$D_{cc} = \sqrt{(D_{bc})^2 + (2\ SAP_c)^2}$$

10. Length of Cutter Contact—Length of contact on the cutter Z_c equals the distance from the cutter tip down to SAP_c along the line of action:

$$Z_c = \frac{A_{oc}}{2} - SAP_c$$

Note: This value must be less than the available profile (AP_c) on the shaving cutter. The AP_c value is the line of action length of ground profile measured from the cutter tip down towards the base diameter. The value of AP_c can be determined from a profile checker or it can be computed. It is only necessary to check the difference between Z_c and AP_c in cases of very low operating pressure angles, extra long addendum on the gears or very large gears. The Z_c dimension is useful when modifications in the gear profile are being considered.

11. Extra Addendum "+A"—A computation is necessary to determine the amount of cutter addendum extending beyond the start of desired shaved profile (SAP_G). The term or symbol for this extra addendum is "+A". The cutter diameter for zero "+A" is known as "D_x".

$$D_x = \sqrt{\left(\frac{A_{rG} - 2\ SAP_G}{H} + A_{rc}\right)^2 + (D_{bc})^2}$$

$$+A = \frac{D_{oc} - D_x}{2}$$

The "+*A*" value is most important in determinging the usefulness of the shaving cutter being analyzed. It is analogous to the depth of profile shaved on the gear (point denoted as "FIN"). A positive value shows the cutter to be finishing beyond the SAP_G and a negative value denotes a "short addendum" on the cutter. On cutters already made the value can be shortened easily by reducing the cutter outside diameter. However, the value can be increased only by regrinding the cutter profile. Therefore, in this procedure, if the radial clearance (*c*) between cutter and gear root (computed below) is less than desired or if the "FIN" point is too deep in the gear (perhaps contacting the gear fillet), the amount of "+A" can be reduced to a point of acceptance. In any event the "+*A*" value should always be positive. A rule of thumb for general use is approximately 0.100 in./P_n.

12. Radial Root Clearance — To find the radial root clearance:

$$c = C - \left(\frac{D_{oc} + D_{RG}}{2}\right)$$

Note: The minimum amount of root clearance allowable depends entirely upon the conditions under which the cutter will operate.

13. Crossed-Axes Angle—The actual crossed axes angle, if desired, is available by obtaining the algebraic sum of the operating helix angles ψ_{rG} and ψ_{rc}.

Typical Calculation

A shaving cutter was designed for a 13-tooth pinion with standard tooth proportions. Problem: what is the tight-mesh condition when the cutter is used on a 14-tooth pinion, which is 20.3% oversize [as found by equations (1) and (2)]?

Data according to step 1:

P_n	= 8		
ϕ_n	= 20°		
N_G	= 14	N_c	= 73
ψ_G	= 25° L.H.	ψ_c	= 10° R.H.
D_G	= 1.9309	D_c	= 9.2657
ϕ_{tG}	= 21.8802°	ϕ_{tc}	= 20.2835°
D_{bG}	= 1.7918	D_{bc}	= 8.6911
T_{nG}	= 0.2119	T_{nc}	= 0.1095
D_{oG}	= 2.218	D_{oc}	= 9.207
SAP_G	= 0.125		
D_{RG}	= 1.643		
SRP_G	= 0.046		

Calculations for steps 6 to 13 inclusive are performed using the above data and the equations.

For steps 1 thru 7 under *Section 6*:

D_G	1.9309	D_c	9.2657
t_G	.2338	t_c	.1112
ϕ_G	21.8802°	ϕ_c	20.2835°
ψ_G	25°	ψ_c	10°
ψ_{bG}	23.3989°	ψ_{bc}	9.3912°

1 . . .

$$E = 14\left(\frac{.2338}{1.9309} + .0197\right) + 73\left(\frac{.1112}{9.2657} + .0155\right) - \pi$$

$E = .8425$

2 . . .

$$\text{inv } \phi_{rG} = \frac{.8425}{14 + 73\left(\frac{.0155}{.0197}\right)} = .0117$$

$\sin \phi_{rG} = .3177$

3 . . .

$$\sin \phi_{rc} = .3177\left(\frac{.9177}{.9865}\right) = .2955$$

$\text{inv } \phi_{rc} = .0093$

4 . . . First approximate check:

$E' = 14 \times .0117 + 73 \times .0093 = .8464$

$\triangle E = E' - E = .8464 - .8425 = .0038$

Since the error is greater than .0010″, steps 2 and 3 are repeated using functions of the calculated ϕ_{rG} and ϕ_{rc} in place of ϕ_{tG} and ϕ_{tc}.

4A . . .

$$\text{inv } \phi_{rG} = \frac{.8425}{14 + 73\left(\frac{.0093}{.0117}\right)} = .0117$$

$\sin \phi_{rG} = .3172$

4B . . .

$$\sin \phi_{rc} = .3172\left(\frac{.9177}{.9865}\right) = .2951$$

$\text{inv } \phi_{rc} = .0092974$

4C . . . Check:

$E'' = 14 \times .0117 + 73 \times .0092 = .8425$

$\triangle E = E'' - E = .8425 - .8425 = .0000$

5A . . . Find cos of ϕ_{rG} and ϕ_{rc} then: —

$$D_{rG} = \frac{1.7918}{.9483} = 1.8894$$

$$\tan \psi_{rG} = \frac{1.8894 \times .4663}{1.9309} = .4562$$

$\cos \psi_{rG} = .9097$

$$t_{nrG} = 1.8894 \times .9097\left[\frac{.2338}{1.9309} + .0197 - .0117\right]$$

$t_{nrG} = .2219$

$$D_{rc} = \frac{8.6911}{.9554} = 9.0963$$

$$\tan \psi_{rc} = \frac{9.0963 \times .1763}{9.2657} = .1731$$

$\cos \psi_{rc} = .9853$

$$t_{nrc} = 9.0963 \times .9853\left[\frac{.1112}{9.2657} + .0155 - .0092\right]$$

$t_{nrc} = .1638$

$$P_{nr} = \frac{\pi \times 9.0963 \times .9853}{73} = .3857$$

6 . . .

$P'_{nr} = .2219 + .1638 = .3857$

Difference or error = .0000″

7 . . .

$$C = \frac{1.8894 + 9.0963}{2} = 5.4928$$

$$H = \frac{.9177}{.9865} = .9302$$

8 . . .

a $\begin{cases} A_{rG} = 1.8894 \times .3172 = .5994 \\ A_{rc} = 9.0963 \times .2951 = 2.6846 \end{cases}$

b $\begin{cases} A_{oG} = \sqrt{(2.218)^2 - (1.7918)^2} = 1.3072 \\ A_{oc} = \sqrt{(9.207)^2 - (8.6911)^2} = 3.0384 \end{cases}$

9 . . .

(a) On gear: —

$$FIN = \frac{.5994 - .9302\ (3.0384 - 2.6846)}{2} = .1351$$

$$D_{fin} = \sqrt{(1.7918)^2 + (.2703)^2} = 1.8120$$

(b) On cutter: —

$$SAP_c = \frac{1}{2}\left[2.6846 - \frac{(1.3072 - .5994)}{.9302}\right] = .9618$$

$$D_{cc} = \sqrt{(8.6911)^2 + (1.9237)^2} = 8.9015$$

10 . . .

$$LC_c = \frac{3.0384}{2} - .9618 = .5573$$

11 . . .

$$D_x = \sqrt{\left(\frac{.5994 - .250}{.9302} + 2.6846\right)^2 + (8.6911)^2}$$

$D_x = 9.2142$

$$+A = \frac{9.207 - 9.2142}{2} = -.0036$$

12 . . .

$$c = 5.4928 - \frac{9.207 + 1.643}{2}$$

$c = .0678$

13 . . .

$\psi_{rG} = 24.5267°$ *LH*

$\psi_{rc} = 9.8207°$ *RH*

Crossed Axes = 14.7060°

Results of Analysis

The "*FIN*" calculated under section 9 and the negative "$+A$" value obtained in section 11 prove the shaving cutter to be "short on addendum." Therefore this cutter with its present tooth proportions should not be used on the 14-tooth pinion. However, it is possible to redevelop this cutter for use on the 14-tooth pinion by reducing the tooth thickness until the addendum is a minimum of 0.0150 in. longer.

#NG 14				
PN 8.000000	PAN 20.000000	#NG 14	HAG 25.000000	PD 1.930911
PATG (BASIC) 21.880232	TN 0.211900	TBG 0.140800	PIN (DIA.) 0.216000	DOP 2.263154
		#NC 73	HAC 10.000000	PD 9.265767
PATC (BASIC) 20.283559	TN 0.109543	TBC 0.027574	PIN DIA. 0.216000	O.D. DESIGN 9.207000
SD 0.076055	+A -0.003615	CLEAR 0.067944	FIN. T 0.135159	LCB 0.557353
	(11)	(12)	(9)	(10)

BD 1.791817	LEAD 13.008874	ELC 117 29 25.5	L-SET 3.187201	FROM 60.000000
BD 8.691183	LEAD 165.086886	ELC 8 36 40.2	L-SET 1.248640	FROM 90.000000
HOP 3.412504	HOD 3.353500	DOP 9.322900	O.D. 9.207000	TIP 1.519225
LC-1	LC 2	LC 3	CD 5.492900	PATRG 18.497770
			(7)	

Fig. 2-7 Computer printout of shaving cutter analysis made in Section 24.

Solution by Computer

Computers are being used extensively as a means for designing gear sets. They can also be utilized effectively to provide quick solutions to the laborious mathematical solution of gear calculations such as the one made in this section dealing with shaving cutter analysis to determine the compatability of a gear and shaving cutter design.

First the data is inserted in an input format such as that shown in Fig. 2-8. Then the data is transmitted to the computer. The solution as printed out in items No. 7, 9, 10, 11 and 12 in Fig. 2-7, agrees with the mathematical solutions shown on the opposite page.

1	Date		Customer's Name		Part#		C#, CC#, GH#, GE#	
	4-6-71		*Modern Methods*		*Example*			
2	Nor. Dia. Pitch		Nor. P.A.	Gear# Teeth	Gear H.A.		Gear O.D.	Des. Pin Dia.
	8.		*20.*	*14.*	*25.*		*2.218*	*.216*
3	Gear	1	Gear D.O.P.	Pin Dia.	Gear	1	SAP (Inches)	
		2	Nor. Cir. T.T.	Pitch Dia.		2	SAP (Degrees)	
		3	Corr. Add.	Chor. T.T.		3	SAP (Dia.)	Gear Ded.
	XIG	*2.*	*.2119*	*1.930911*	XIG1	*1.*	*.125*	*.144*
4	Modif. Points		Degrees (1)	Degrees (2)	Degrees (3)		Tr. Fin./Gear	Tool# Teeth
	XIX							*—73.*
5					Tool	1	Tool D.O.P.	Pin Dia.
						2	Nor. Cir. T.T.	Pitch Dia.
						3	H.O.P.	Pin Dia.
	Tool H.A.		Tool O.D.	Des. Pin Dia.		4	Corr. Add.	Chor. T.T.
	—10.		*9.207*	*.216*	XIC	*2.*	*.109540*	*9.265767*
6	Tool		Tool	Tool	Tool	0	Constant Fin.	
	Total T.T. Red.		T.T. Red./Grd.	Bore Dia.		1	Constant S.D.	
				2.500	XIC2	*0*		

Fig. 2-8 Input format for the computer solution of the shaving cutter analysis made in the printout in Fig. 2-7.

SECTION 25

determining the maximum crossed axes for shaving gears having restrictive shoulders

A shoulder is any integral part of a gear which is close enough to the gear face to prevent a shaving cutter at crossed axes from contacting the entire gear face width. This condition is illustrated by a cluster gear, where two or more gears are an integral part and where usually, two of the gears are quite close together. The larger gear constitutes a shoulder which limits the angle of "Crossed Axes" between the smaller gear and the shaving cutter. Sometimes, a flange, cam, or the tooling which holds the work piece can be the interference.

The shoulder gear layout shown in Fig. 2-9 may be used to determine the maximum crossed axes angle which will prevent interference between shaving cutter and shoulder. The maximum crossed axes angle can be calculated with the following equations. It must be realized that, at best, either method is an approximation, since the shaving cutter is treated as a circle in a section which is obviously elliptical.

Shaving cutter and other elements, used in either method, can be as obtained in the tight mesh calculations shown in Section 24; or, they can be approximated. The shoulder diameter (*SD*)

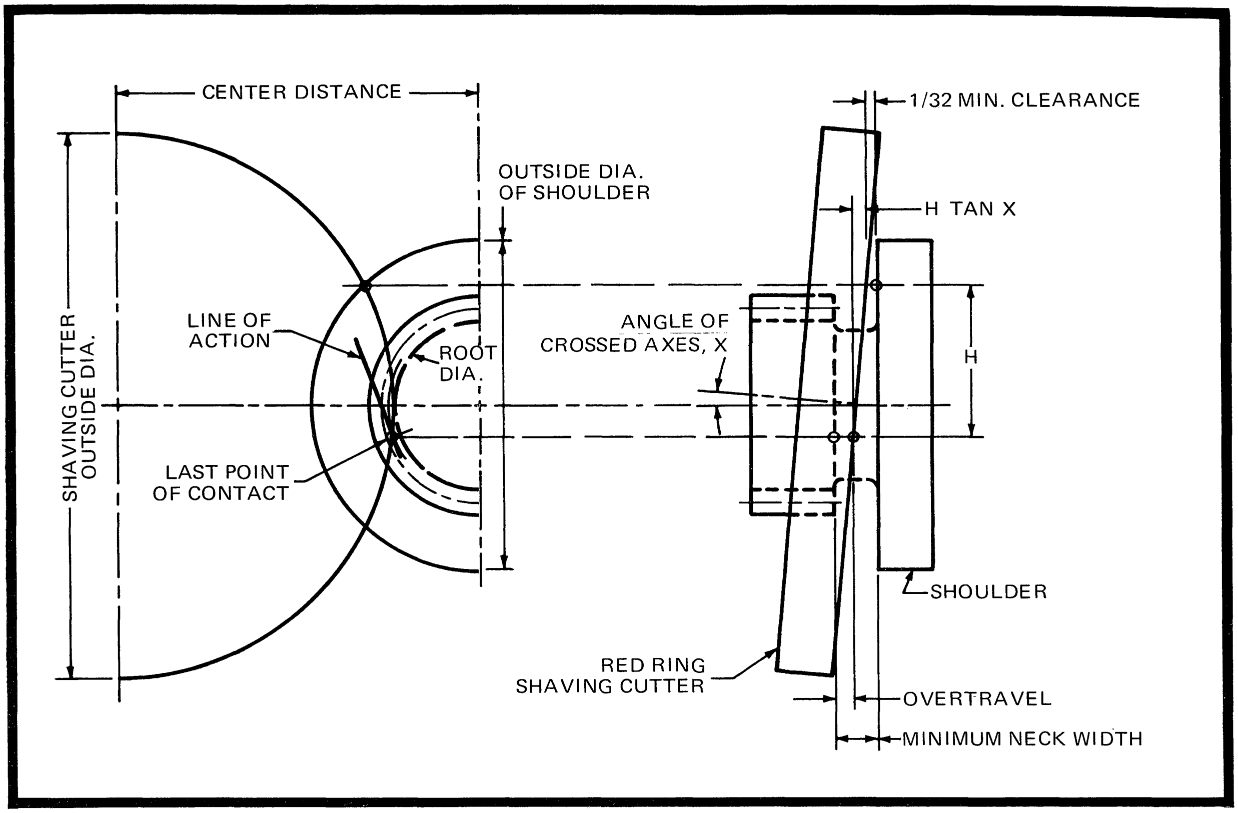


Fig. 2-9—Limitation of crossed axes angle on shoulder gears.

must be the maximum diameter to the interfering point. If not known, the approximated shaving cutter outside diameter (D_{OC}) should be the largest allowed in its design size class (3.5″, 6.0″, 8.0″, 9.5″ or 13.5″). Center distance C between cutter and gear can be approximated by adding the outside radius of the cutter to the root radius of the gear. The lowest point of shaving cutter contact on the gear (D_{FIN}) can be approximated by subtracting a reasonable amount (.010—.030) from the diameter to the lowest point of mating gear contact (D_c) or TIF diameter.

The neck width or gap between the gear face and face of the interfering shoulder should always be the minimum obtainable through the tolerance "stack up" of the component part. No less than the 1/32″ clearance shown in Fig. 2-9 should ever be used. The total of clearance plus the overtravel of the cutter face (hereafter referred to as C & O) should be as near as possible to that shown in Fig. 2-10. Shaving at crossed axes less than 3° is not recommended.

CROSSED AXES	MINIMUM C & O
3°	.080
4°	.100
5°	.120
6°	.140
7°	.160
8°	.180
9°	.200
10°	.220

Fig. 2-10—Minimum cutter clearance and overtravel (C & O) for various crossed-axes settings of work gear and cutter when shaving shoulder gears.

1. Calculation of dimension "H" used in the shoulder gear equations:

$$H = \sqrt{D_{oc}^2 - A^2} + \sqrt{D_{FIN}^2 - B^2}$$

Where

$$A = \frac{4C^2 - SD^2 + D_{oc}^2}{4C}$$

$$B = \frac{4C^2 + D_{FIN}^2 - D_{oc}^2}{4C}$$

Assuming all elements of shoulder interference are known.

2. For C & O to suit a desired crossed axes angle, X:

$$C \& O = (\text{Minimum neck width}) - \frac{H \tan X}{2}$$

3. For maximum crossed axes angle (X) to suit a given minimum neck width (C & O approximated):

$$\tan X = \frac{2 \text{ min. neck width} - (C \& O)}{H}$$

Although not nearly so prevalent, crossed axes restrictions do occur in internal gears. Such interferences can be treated in a similar manner as external gears with proper alterations to layout and equations.

Chapter THREE

Loading welded gear assemblies into a continuous furnace for a gas carburizing and modified marquenching sequence. Ends of gears are coated to prevent carburization and provide additional welding surfaces. *Courtesy Clark Equipment Co.*

Material Selection and Heat Treatment

Gear materials are selected to provide the optimum combination of properties at the lowest possible cost consistent with satisfying other requirements. Some of the important physical properties of gears are abrasion or wear resistance, toughness, static compression strength, shear strength, fatigue strength, and strength at elevated temperatures.

Because of widely varying requirements, gears are produced from a wide variety of materials. These materials include plastics such as nylon, powdered metals, brasses, bronzes, cast or ductile irons, and steels. Many types of steels, including stainless steel and tool steel, are used. Each of the materials mentioned will best satisfy some specific requirement such as corrosion resistance, extreme wear resistance, special damping qualities, ability to operate without lubrication, low cost, or producibility.

The majority of gears for automotive, aircraft, farm machinery, off-the-road equipment, and machine tool applications are produced from hardenable carbon or low-alloy steels or cast iron. Therefore, this chapter will cover only ferrous materials.

The choice of a gear material depends on four factors:

1. Mechanical properties
2. Metallurgical characteristics
3. Blank-forming method
4. Manufacturing process

Each of these factors must be evaluated with an eye to its effect on the three major performance criteria—durability, strength, and wear.

Mechanical Properties*

Before the optimum mechanical properties can be selected, the working stress must be deter-

**Abstracted from Machine Design, Gear Materials Design Guide, June 20, 1968.*

mined, based on recommended allowable stresses.

This article discusses durability and strength formulas adopted by the American Gear Manufacturers Association and widely used throughout the world, Table 3-1. There is a fundamental relationship among most formulas, so that working and allowable stresses determined by different formulas can be used or compared.

Table 3-1—AGMA Standards Useful in Selecting Gear Materials

Nomenclature		
110.03	Gear-Tooth Wear and Failure (USAS B6.12-1964)	Jan. 1962
Durability		
215.01	Information Sheet for Surface Durability (Pitting) of Spur, Helical, Herringbone and Bevel Gear Teeth	Sept. 1966
217.01	Information Sheet-Gear Scoring Design Guide for Aerospace Spur and Helical Power Gears	Oct. 1965
Strength		
225.01	Information Sheet for Strength of Spur, Helical, Herringbone and Bevel Gear Teeth	Dec. 1967
Inspection		
230.01	AGMA Standard-Surface Temper Inspection Process	March 1968
Materials		
241.02	Specification for General Industrial Gear Materials—Steel (Drawn, Rolled and Forged)	Jan. 1965
242.02	Cast Iron Blanks	Sept. 1946
244.02	Nodular Iron Gear Materials	July 1965
245.01	Specification for Cast Steel Gear Materials	Jan. 1964
246.01	Recommended Procedure for Carburized Industrial Gearing	Jan. 1965
247.01	Recommended Procedure for Nitriding, Materials and Process	Jan. 1965
248.01	Recommended Procedure for induction Hardened Gears and Pinions	Jan. 1964
249.01	Recommended Procedure for Flame Hardening	Jan. 1964

Working stress is usually based on the fatigue or yield strength of the material. Less frequently, impact resistance, tensile strength, or brittle-fracture characteristics of the material must be considered.

For most gear trains, the limiting design consideration is profile durability (pitting resistance), gear-tooth strength (resistance to fracture), or wear. Normally, durability is the limiting consideration; but sometimes all three are of nearly equal importance.

A number of other possible modes of failure may also limit gear performance, such as scoring associated with high speed and heavy loads, case crushing in carburized and hardened gearing, and micro-pitting. However, these are less likely to occur.

The AGMA gear-rating formulas for both strength and durability are basically the same for spur, helical, herringbone, and bevel gear teeth. Terms in both formulas are divided into four major groupings, associated with load, size, stress distribution, and stress.

The surface-durability or pitting-resistance formula—*AGMA* 215.01, *Sept. 1966, "Information Sheet for Surface Durability (Pitting) of Spur, Helical, Herringbone and Bevel Gear Teeth"*—is

$$s_c = C_p \sqrt{\underbrace{\left(\frac{W_t C_o}{C_v}\right)}_{\text{(Load)}} \underbrace{\left(\frac{C_s}{dF}\right)}_{\text{(Size)}} \underbrace{\left(\frac{C_f C_m}{I}\right)}_{\text{(Stress dist.)}}} \quad (1)$$

where

$$s_c \leqq s_{ac} \underbrace{\left(\frac{C_L C_H}{C_R C_T}\right)}_{\text{(Stress)}} \quad (2)$$

The "stress" term, Equation 2, is the most important consideration in selecting a gear material. Table 3-2 lists all symbols used in the strength and durability formulas. Allowable fatigue (contact) stresses, s_{ac}, are shown in Table 3-3.

Table 3-2—Gear-Rating Factors

Factor	Strength	Durability
Load		
Transmitted load	W_t	W_t
Dynamic factor	K_v	C_v
Overload factor	K_o	C_o
Size		
Pinion pitch diameter	—	d
Net face width	F	F
Transverse diametral pitch	P_d	—
Size factor*	—	C_s
Stress Distribution		
Load-distribution factor	K_m	C_m
Geometry factor	J	I
Surface-condition factor	—	C_f
Size factor*	K_s	—
Stress		
Calculated stress	s_t	s_c
Allowable stress	s_{at}	s_{ac}
Elastic coefficient	—	C_p
Hardness-ratio factor	—	C_H
Life factor	K_L	C_L
Temperature factor	K_T	C_T
Safety factor	K_R	C_R

**Size factor is placed either in the size or stress-distribution grouping, depending upon the importance of the effect.*

The mechanical properties for gear materials are almost always specified in terms of Brinell hardness rather than ultimate strength or test-bar properties. Location (normally the toothed portion) and number of hardness tests should also be specified. Maximum hardness is also specified in the interests of machinability and sometimes to insure a hardness difference between the mating gears; such a difference can increase wear resistance. Typical combinations are shown in Table 3-4.

Values of s_{ac} are based on 10^7 cycles, since most gear materials generally show the typical "knee" at or below this value. Hence, these

Table 3-3—Allowable Contact Stress (Durability)

Material	Minimum Surface Hardness	Contact Stress, s_{ac} (1000 psi)
Steel		
Through-hardened	180 Bhn	85-95
	240 Bhn	105-115
	300 Bhn	120-135
	360 Bhn	145-160
	440 Bhn	170-190
Case-carburized	55 R_c	180-200
	60 R_c	200-225
Flame or induction-hardened	50 R_c	170-190
Cast Iron		
AGMA Grade 20	—	50-60
AGMA Grade 30	175 Bhn	65-75
AGMA Grade 40	200 Bhn	75-85
Nodular Iron	165-300 Bhn	10% less than for steel with same hardness

values can be considered fatigue-strength stresses. If a gearset must operate for only a finite number of cycles, the s_{ac} values can be increased by life factor C_L, Fig. 3-1.

Pitting of gear teeth is considered a fatigue phenomenon. There are two kinds of pitting—initial and progressive (destructive). Corrective and non-progressive initial pitting is not con-

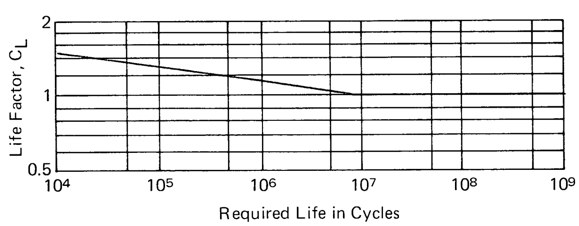


Fig. 3-1—Life factor for durability rating of gears.

sidered serious and is normally to be expected until imperfections, such as high spots, are worn in.

The stresses in Table 3-3 are satisfactory for gears with smooth profiles operating with adequate lubrication. Some "as-hobbed" or shaped gears, or those with rough-textured profiles, should use more conservative stresses. In this case, special attention should be given to the lubricant. Procedures for inducing favorable surface stresses for increased profile durability are not yet generally used.

Table 3-4—Typical Gear/Pinion Hardness Combinations

	Minimum Hardness (Bhn)*											
Gear	180	210	225	255	270	285	300	335	350	375	55R_c	58R_c
Pinion	210	245	265	295	310	325	340	375	390	415	55R_c	58R_c

*Maximum hardness is usually 35 to 40 Bhn higher.

Except for heavily loaded gears or those subjected to unusual environments, no substantial difference in allowable contact stress exists among the normal qualities of available commercial steels. Free-machining steels may be used to obtain improved machinability or a better surface finish.

Because pitting is a fatigue phenomenon, it displays a scatter which must be allowed for by a safety factor to ensure reliability. If this factor is not included in the basic design calculations, Table 3-5 can be used as a guide and the s_{ac}

Table 3-5—Recommended Safety Factors in Pitting

Required Reliability	Factor C_R
High	1.25+
Fewer than 1 failure in 100	1.00
Fewer than 1 failure in 3	0.80*

**At this value, plastic profile deformation might occur before pitting.*

values adjusted accordingly. It should be remembered that "failure" does not necessarily mean an immediate failure under applied load, but rather shorter life than expected.

If one of the mating gear elements is considerably harder than the other, the allowable stress of the softer element can be increased under certain conditions. Normally, the gear ratio must be high —over 8:1—and the gears large before any appreciable improvement is obtained. Typical of gears for which such a correction is normally made are those for large kilns, or for ball mills. The appropriate AGMA standards contain recommended values of the C_H factor used to rate gearsets having large differences in hardness.

Generally, temperature factor $C_T = 1$ when gears operate with oil or with gear-blank temperatures not exceeding 250F. In some instances, it is necessary to use $C_T > 1$ for carburized gears operating at oil temperatures above 180F. Tests have indicated a drop of several points R_c hardness for carburized steels subjected to 200F for 10,000 hr, as well as a 6 to 8% reduction in fatigue strength.

When gears are proportioned on the basis of allowable surface fatigue stress, surface yielding is seldom a problem. This is partially because the contact stress only increases as the square root of the transmitted load. Allowable overloads of 100% above the surface-fatigue rating are commonly specified. Greater amounts of overload are often successfuly carried. However, repeated overloading can cause plastic flow, which ripples or grooves the profile or extrudes a "wire edge" at the tip of the tooth. This extruded material can affect lubrication. Excessive plastic flow of the profile can induce abrasive wear because of the rough surface texture developed.

Surface yielding does not cause immediate or catastrophic failure and the use of heavier viscosities or extreme-pressure lubricants along with reduced loading can alleviate a troublesome situation when it is encountered.

The effect of impact or brittle-fracture properties of gear materials on surface stresses need not be considered, except for the peak stresses developed by impulse or impact loading.

Tooth Strength:

The tooth-strength rating formula—*AGMA* 225.01, *Dec. 1967, "Information Sheet for Strength of Spur, Helical, Herringbone and Bevel Gear Teeth"*—is

$$s_t = \underbrace{\left(\frac{W_t K_o}{K_v}\right)}_{\text{(Load)}} \underbrace{\left(\frac{P_d}{F}\right)}_{\text{(Size)}} \underbrace{\left(\frac{K_s K_m}{J}\right)}_{\text{(Stress dist.)}} \quad (3)$$

where

$$s_t \leqq s_{at} \underbrace{\left(\frac{K_L}{K_R K_T}\right)}_{\text{(Stress)}} \quad (4)$$

Again, allowable stress is the most important term in selecting a material.

The fatigue and yield resistance necessary to prevent the fracture of a gear tooth depend on more complex relationships between materials properties than those for profile durability. This is true primarily because the root-radius stress varies directly with the load. Fortunately, the root stress can be calculated fairly accurately by applying standard beam or plate theories.

Table 3-6—Allowable Stress (Strength)

Material	Minimum Hardness	Allowable Stress, s_{at} (1000 psi) Spur, Helical, & Herringbone	Bevel**
Steel			
Normalized	140 Bhn	19-25	11
Quenched & temp.	180 Bhn	25-33	14
Quenched & temp.	300 Bhn	36-47	19
Quenched & temp.	450 Bhn	44-59	25
Case carb.	$55R_c$	55-65	27.5
Case carb.	$60R_c$	60-70	30
Ind. or flame through-hardened	54 Bhn	45-55*	
Ind. or flame through-hardened	54 Bhn	22	13.5
Nitrided AISI 4140	$53R_c$†	37-42*	20
Cast Iron			
AGMA Grade 20	......	5	2.7
AGMA Grade 30	175 Bhn	8.5	4.6
AGMA Grade 40	200 Bhn	13	7
Nodular Iron			
ASTM Grade 60-40-18	Annealed	15	8
ASTM Grade 80-55-06		20	11
ASTM Grade 100-70-03	Normalized	26	14
ASTM Grade 120-90-02	Quenched & temp.	30	18.5

*Value for 6 DP and finer.

†Case hardness; core is 300 Bhn. For heavy gears, these hardnesses will be lower. Thus, lower values of s_{at} should be used.

**Allowable stress for bevel gears appear low, but these are modified, in the bevel-gear rating formula, by a size factor.

Some allowable fatigue stresses s_{at} used with the AGMA formula are listed in Table 3-6.

Allowable stress or fatigue strength is normally determined for 10^7 cycles, with adjustments for shorter finite life made by using the K_L factor given in Fig. 3-2. Unlike C_L, K_L depends on fatigue notch-sensitivity, which is somewhat proportional to hardness; hence the necessity for several curves for various gear hardnesses.

Experience suggests that a clearly defined knee is not always present in the gear fatigue stress/cycle plot. This phenomenon might be an inherent property, as it is with non-ferrous materials. But

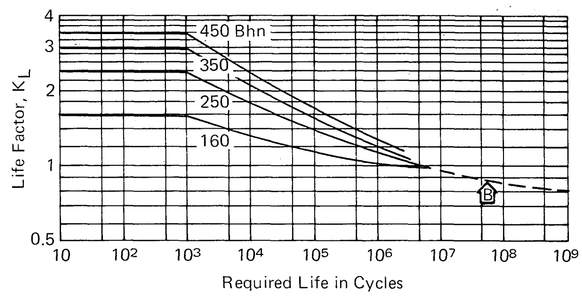


Fig. 3-2—Life factor for strength rating of gears.

it is more likely a result of wear or other service-developed changes which affect either the dynamic loading, load distribution, or stress system. For conservative design, sometimes s_{at} is reduced by the K_L factor determined from curve *B* in Fig. 3-2.

Normally the stress at the root of the tooth varies from zero to the maximum working tensile stress. If a fully reversed stress is present, as it is with reversing loads or with idler gears, the allowable stress should be 70% of the values shown in Table 3-6.

The quality of the material has a pronounced effect on the strength of gear teeth, more so than in the case of pitting resistance. The values shown in Table 3-6 are for commercially available steels. In the absence of a clear understanding of the criteria for judging quality, the lower values should be used. These are suitable for steels with a cleanliness typical of resulphurized or leaded steels.

The upper values in Table 3-6 would require steels produced under good melting and pouring conditions. Cast steels require adequate directional and progressive solidification. Rolled and forged steels will probably require vacuum treatment, with special instructions to prevent excessive reduction during rolling or forging; this procedure will provide a desirable direction of fiber flow lines without excessive reduction in transverse properties. Finally, heat treatment should be under rigid supervision in accurately controlled furnaces, followed by careful hardness and metallographic inspection. For high-speed gears, or for drives requiring maximum reliability, additional inspection, for example magnetic and ultrasonic inspection, is a necessity.

Fatigue: The fatigue strength of a gear tooth is significantly affected by size, surface finish, and residual stresses at the root radii. Residual stresses in a favorable direction can be developed by metallurgical processing such as case carburizing or nitriding. Under certain circumstances, flame or induction hardening, which do not develop a uniform hardness pattern on the profile or in the root area, can induce significant unfavorable stress patterns. Mechanical processing, such as shot peening, to induce favorable root stresses is used successfully.

The fatigue strength of gear teeth follows a statistical pattern, so that a safety factor should be used. Some recommended factors are listed in Tables 3-7.

It is important to remember that a tooth fracture, unlike profile pitting, is catastrophic and cannot be repaired or alleviated by reducing load or changing lubricant. Impact loads in geared systems are commonly two or more times the rated load. Because gears have backlash and are loosely coupled, both the electrical torque and inertial energy of an electric motor can be released simultaneously, so that peak torques of two to four times the nameplate rating are not uncommon. For this reason an adequate safety factor, K_R, and life factor, K_L, are important.

Table 3-7—Safety Factors for Fatigue and Yield Strength

Reliability	Factor K_R
Fatigue	
High	1.50+
Fewer than 1 failure in 100	1.00
Fewer than 1 failure in 3	0.70
Yield	
High	3.00+
Normal	1.33

Yield: Loads which develop a root stress above the yield strength of the material cause the gear tooth to bend permanently. Since gear-tooth dimensions are held to 0.001 or less, only a slight permanent bend or distortion can affect the geometric dimensions sufficient to cause interference or high dynamic loads. This will quickly result in a fatigue fracture.

The allowable yield stress for steel recommended for use with the AGMA rating formulas is shown in Fig. 3-3. The alloy composition, heat-treatment effectiveness, and section size affect the yield strength at any ultimate strength or hardness, so that more accurate yield-strength data can be used if it is available.

As a general rule, when peak loads exceed 200% of the allowable endurance stress (100% overload), it is necessary to consider yield in both the design calculations and material selection. Table 3-7 lists appropriate safety factors for use when selecting materials for yield strength.

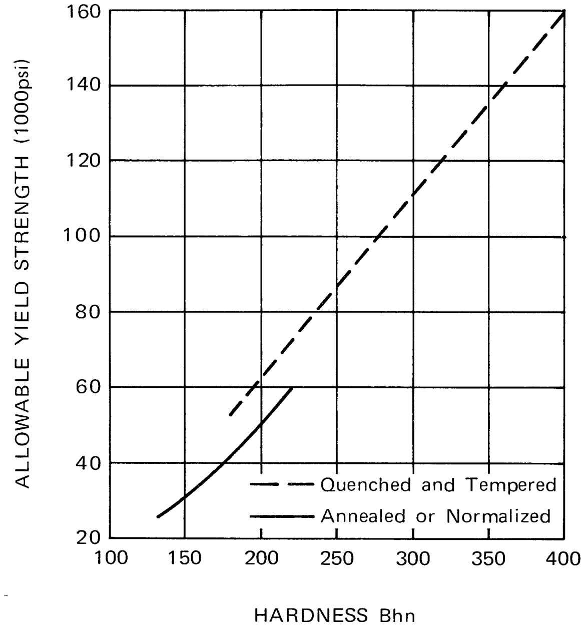


Fig. 3-3—Allowable yield stress recommended for use with AGMA formulas.

Operating temperatures can affect the allowable stress. According to AGMA Standards, when gears operate at oil or gear-blank temperatures not exceeding 250F temperature factor, generally $K_T = 1$. For case-carburized gears at temperatures above 160F, K_T may be found from

$$K_T = \frac{460 + T_F}{620} \qquad (5)$$

where T_F = peak operating oil temperature, deg F.

Fatigue failure initiated by a bent tooth caused by loads above the yield strength is not uncommon and should not be disregarded even for surface-hardened gears. This is particularly true when the design is based on allowable fatigue stresses which have been increased by favorable induced surface stresses.

Impact: Although impact is not significant in evaluating a material for surface durability, it can be important in tooth-strength considerations.

The first step is to determine whether or not destructive impact can be developed in the gear system. Normally, if the time to develop the peak stress is a significant proportion of the natural period of vibration, the stress can be calculated from loads deduced from the oscillatory characteristics of the mass elastic system. The material must then be selected on the basis of yield strength or, if sufficient cycles are involved, fatigue strength.

In some specific applications—mostly determined by field experience—impact properties should be considered as a matter of course. One example is service at low temperatures, which are known to reduce impact strengths. Here steels containing nickel and those having fully quenched and tempered metallographic structures are desirable. Lower hardness and sometimes lower carbon content are beneficial, although these must be considered with an eye to the required size of gearing and costs. Obviously the direction of grain flow, transverse properties, and area reduction during forging must be controlled.

Brittle fractures of gear teeth occur only occasionally. Efforts to correlate material properties with brittle-fracture analysis have not yet produced results of value to the designer, although some progress is being made.

Wear

The wear in contacting teeth varies from an unmeasurable amount with fully hydrodynamic lubrication, through intermediate and boundary stages, to dry metal-to-metal contact. Hydrodynamic lubrication is present when the ratio of film thickness to surface finish exceeds approximately 1.4; boundary lubrication when the ratio is between 0.05 to 0.2; and dry lubrication when the ratio is less than 0.01. These are approximate guides based on tests with straight mineral oils.

Normally, wear is encountered only when surface finishes are rough, speeds are very low, or loads and velocities are high. There are four types of wear: adhesive, abrasive, corrosive, and fatigue.

Generally, when ferrous metals are used for gears, the wear rate decreases with increasing hardness. Under certain conditions, softer gears will polish to a fine surface finish and a good load distribution; this provides improved lubrication. In some cases, the profile will be worn to a non-involute shape; once this "wearing-in" has occurred, little wear will follow. If this beneficial wear does not occur, high hardness, obtained by quenching or by a surface-hardening source such as induction or flame hardening, carburizing, or some form of nitriding must be used.

It is not uncommon for changes to occur in the dedendum of heavily loaded gears, particularly at slow speeds. A concave surface is developed which soon stabilizes and in no way affects the load-carrying capability of the gears. This phenomenon is due to a number of reasons, one of them being the fact that the rolling and sliding are in opposite directions on the dedendum.

In some carburized and hardened gears a frosted appearance develops. This ultimately results in surface spalling, which in turn can initiate a tooth fracture. This frosting is probably due to microscopic pitting. Better surface finish and accuracy, and improved lubrication—rather than changes in material—are required to solve this problem.

Scoring

In some heavily loaded or high-speed gearing, scoring may occur under boundary film conditions. This is believed to be caused by frictional heat which reduces the lubricant protection sufficiently to allow welding and tearing of the profile.

Materials selection alone will not prevent scoring; proper lubricants and design geometry are required. This difficulty is seldom encountered in the conventional industrial gear drive. *AGMA 217.01, Oct. 1967, "AGMA Information Sheet—Gear Scoring Design Guide for Aerospace Spur and Helical Power Gears"* provides helpful recommendations for avoiding scoring.

Metallurgical Characteristics*

The approximate tensile strength of any steel is measured by its hardness, Table 3-8. Since hardness is determined by both chemical composition and heat treatment, these are the two important metallurgical considerations in selecting gear steels.

Chemical Composition

Hardenable gear steels are of two types: Through-hardenable or case-hardenable. Through-hardenable steels contain alloying elements and usually have carbon content ranging from about 0.40 to 0.50-percent to give the desired hardness. Steels for case-hardening may or may not contain alloying elements, but have lower carbon content (usually less than 0.25-percent). The lower carbon content permits development of high surface hardness while retaining a softer, more ductile core.

An alloy steel, Table 3-9, is a type to which one or more alloying elements have been added to give it properties that cannot be obtained in carbon steel. Chromium is one of the most versatile and widely used alloying elements. It produces corrosion and oxidation resistance, and induces high hardness and wear resistance. It also intensifies the action of carbon, increases the elastic limit, increases tensile strength, and increases depth of hardness penetration.

Nickel increases shock resistance, elastic limit, and tensile strength of steel. Nickel steels are particularly suitable for case-hardening. This results in their frequent use for aircraft gears where strength-to-weight ratio must be high. The strong, tough case obtained with nickel steels combined with good core properties provides exceptional fatigue and wear resistance. Simplified hardening procedures and low distortion during heat treatment result from lower transformation temperature ranges and the relatively small difference between case and core transformation temperatures.

Molybdenum increases hardenability of steels and has a significant effect on softening of steels

**Implemented and reviewed by Harold A. Maloney, plant metallurgist, Clark Equipment Co.*

Table 3-8—Approximate Tensile Strength for Equivalent Hardness Numbers of Steel

Brinell Indentation Diameter, mm	Brinell Hardness Number, 3000-Kg 10 mm Tungsten Carbide Ball	Rockwell Hardness Number		Vickers Diamond Pyramid Hardness Number	Shore Scleroscope Hardness Number	Approx. Tensile Strength 1000 p.s.i.
		B-Scale 100-Kg Load 1/16 in. Ball	C-Scale 150-Kg Load Brale Penetrator			
2.25	745	—	65.3	840	91	—
—	710	—	63.3	780	87	—
2.35	682	—	61.7	737	84	—
2.40	653	—	60.0	697	81	—
2.45	627	—	58.7	667	79	—
2.50	601	—	57.3	640	77	—
2.55	578	—	56.0	615	75	—
2.60	555	—	54.7	591	73	298
2.65	534	—	53.5	569	71	288
2.70	514	—	52.1	547	70	274
2.75	495	—	51.0	528	68	264
2.80	477	—	49.6	508	66	252
2.85	461	—	48.5	491	65	242
2.90	444	—	47.1	472	63	230
2.95	429	—	45.7	455	61	219
3.00	415	—	44.5	440	59	212
3.05	401	—	43.1	425	58	202
3.10	388	—	41.8	410	56	193
3.15	375	—	40.4	396	54	184
3.20	363	—	39.1	383	52	177
3.25	352	(110.0)	37.9	372	51	170
3.30	341	(109.0)	36.6	360	50	163
3.35	331	(108.5)	35.5	350	48	158
3.40	321	(108.0)	34.3	339	47	152
3.45	311	(107.5)	33.1	328	46	147
3.50	302	(107.0)	32.1	319	45	143
3.55	293	(106.0)	30.9	309	43	139
3.60	285	(105.5)	29.9	301	—	136
3.65	277	(104.5)	28.8	292	41	131
3.70	269	(104.0)	27.6	284	40	128
3.75	262	(103.0)	26.6	276	39	125
3.80	255	(102.0)	25.4	269	38	121
3.85	248	(101.0)	24.2	261	37	118
3.90	241	100.0	22.8	253	36	114
3.95	235	99.0	21.7	247	35	111
4.00	229	98.2	20.5	241	34	109
4.05	223	97.3	(18.8)	234	—	104
4.10	217	96.4	(17.5)	228	33	103
4.15	212	95.5	(16.0)	222	—	100
4.20	207	94.6	(15.2)	218	32	99
4.25	201	93.8	(13.8)	212	31	97
4.30	197	92.8	(12.7)	207	30	94
4.35	192	91.9	(11.5)	202	29	92
4.40	187	90.7	(10.0)	196	—	90
4.45	183	90.0	(9.0)	192	28	89
4.50	179	89.0	(8.0)	188	27	88
4.55	174	87.8	(6.4)	182	—	86
4.60	170	86.8	(5.4)	178	26	84
4.65	167	86.0	(4.4)	175	—	83
4.70	163	85.0	(3.3)	171	25	82
4.80	156	82.9	(0.9)	163	—	80
4.90	149	80.8	—	156	23	—
5.00	143	78.7	—	150	22	—
5.10	137	76.4	—	143	21	—
5.20	131	74.0	—	137	—	—
5.30	126	72.0	—	132	20	—
5.40	121	69.8	—	127	19	—
5.50	116	67.6	—	122	18	—
5.60	111	65.7	—	117	15	—

The indentation and hardness values in the foregoing table are taken from Table 2, Approximate Equivalent Hardness Numbers for Brinell Hardness Numbers for Steel, pages 122 and 123 of 1952 SAE Handbook, Society of Automotive Engineers, Incorporated.

The values shown in parentheses are beyond the normal range of the test scale and are given only for comparison with other values.

Courtesy Republic Steel Corp.

Table 3-9—Basic AISI and SAE Numbering System for Steels

Numerals and Digits	Type of Steel and Average Chemical Contents, %
	CARBON STEELS
10XX	Plain Carbon (Mn 1.00% max)
11XX	Resulphurized
12XX	Resulphurized and Rephosphorized
15XX	Plain Carbon (max Mn range—over 1.00—1.65%)
	MANGANESE STEELS
13XX	Mn 1.75
	NICKEL STEELS
23XX	Ni 3.50
25XX	Ni 5.00
	NICKEL-CHROMIUM STEELS
31XX	Ni 1.25; Cr 0.65 and 0.80
32XX	Ni 1.75; Cr 1.07
33XX	Ni 3.50; Cr 1.50 and 1.57
34XX	Ni 3.00; Cr 0.77
	MOLYBDENUM STEELS
40XX	Mo 0.20 and 0.25
44XX	Mo 0.40 and 0.52
	CHROMIUM-MOLYBDENUM STEELS
41XX	Cr 0.50, 0.80 and 0.95; Mo 0.12, 0.20, 0.25 and 0.30
	NICKEL-CHROMIUM-MOLYBDENUM STEELS
43XX	Ni 1.82; Cr 0.50 and 0.80; Mo 0.25
43BVXX	Ni 1.82; Cr 0.50; Mo 0.12 and 0.25; V 0.03 minimum
47XX	Ni 1.05; Cr 0.45; Mo 0.20 and 0.35
81XX	Ni 0.30; Cr 0.40; Mo 0.12
86XX	Ni 0.55; Cr 0.50; Mo 0.20
87XX	Ni 0.55; Cr 0.50; Mo 0.25
88XX	Ni 0.55; Cr 0.50; Mo 0.35
93XX	Ni 3.25; Cr 1.20; Mo 0.12
94XX	Ni 0.45; Cr 0.40; Mo 0.12
97XX	Ni 0.55; Cr 0.20; Mo 0.20
98XX	Ni 1.00; Cr 0.80; Mo 0.25
	NICKEL-MOLYBDENUM STEELS
46XX	Ni 0.85 and 1.82; Mo 0.20 and 0.25
48XX	Ni 3.50; Mo 0.25
	CHROMIUM STEELS
50XX	Cr 0.27, 0.40, 0.50 and 0.65
51XX	Cr 0.80, 0.87, 0.92, 0.95, 1.00 and 1.05
501XX	Cr 0.50
511XX	Cr 1.02
521XX	Cr 1.45
	CHROMIUM VANADIUM STEELS
61XX	Cr 0.60, 0.80 and 0.95; V 0.10 and 0.15 minimum
	TUNGSTEN CHROMIUM STEELS
71XXX	W 13.50 and 16.50; Cr 3.50
72XX	W 1.75; Cr 0.75
	SILICON MANGANESE STEELS
92XX	Si 1.40 and 2.00; Mn 0.65, 0.82 and 0.85; Cr 0.00 and 0.65
	LOW ALLOY HIGH TENSILE STEELS
9XX	Various
	STAINLESS STEELS
	(Chromium-Manganese-Nickel)
302XX	Cr 17.00 and 18.00; Mn 6.50 and 8.75, Ni 4.50 and 5.00
	(Chromium-Nickel)
303XX	Cr 8.50, 15.50, 17.00, 18.00, 19.00, 20.00, 20.50, 23.00, 25.00 Ni 7.00, 9.00, 10.00, 10.50, 11.00, 11.50, 12.00, 13.00, 13.50, 20.50, 21.00, 35.00
	(Chromium)
514XX	Cr 11.12, 12.25, 12.50, 13.00, 16.00, 17.00, 20.50 and 25.00
515XX	Cr 5.00
	BORON INTENSIFIED STEELS
XXBXX	B denotes Boron Steel
	LEADED STEELS
XXLXX	L denotes Leaded Steel

NOTE: "XX" after numbers or letters in table indicates carbon percentage; i.e. 1040 indicates 0.40 percent carbon.

From SAE Iron and Steel Handbook Supplement 30 at tempering temperatures. It markedly retards softening of the hardened martensite at tempering temperatures above 450F.

Vanadium is used as an alloying element in steels for two reasons. First is the effect on grain size at elevated temperatures. Vanadium stabilizes the fine grain structure of austenitized steels and permits retention of excellent ductility and impact resistance while developing high tensile and yield strengths. The second reason is the ability to form carbides which remain stable at elevated temperatures.

Hardenability is the property of a steel which determines the depth and distribution of the hardness induced by quenching. The higher the hardenability of a steel, the greater the depth to which the steel can be hardened and the slower the quench which can be used. Hardenability should not be confused with hardness or maximum hardness which can be obtained by heat treatment, since that depends almost entirely on carbon content, Fig. 3-4. Also, section thickness has considerable influence on the maximum hardness obtained for a given set of conditions; the thicker the section, the slower the quench rate will be. Variations in test bar hardenability curves for various 0.20-percent carbon and alloy steels is shown in Fig. 3-5. Similar hardenability curves for 8600 alloy steels with various carbon contents is shown in Fig. 3-6. Maximum hardenability of case-hardened 8620 steel is achieved, Fig. 3-7, when the case carbon concentration is 0.80-percent.

H-Steels are guaranteed by the supplier to meet established hardenability limits for specific grades of steel. These steels are designated by an "H" following the composition code number, such as 8620H, Fig. 3-8. Hardenability of H-steels and a steel with the same chemical composition is not necessarily the same. Therefore, H-steels are often specified when it is essential that a given hardness be obtained at a given point below the surface of a gear tooth.

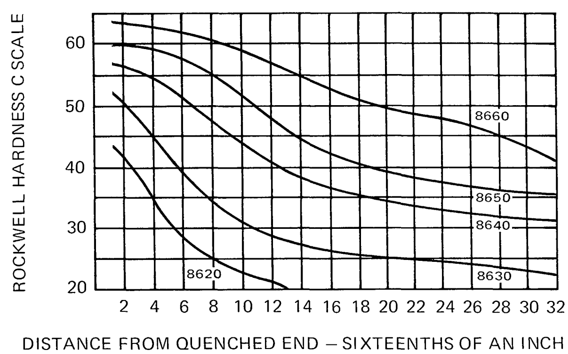


Fig. 3-6—Comparative hardenability of 8600 Alloy Steels. *Courtesy Republic Steel Corp.*

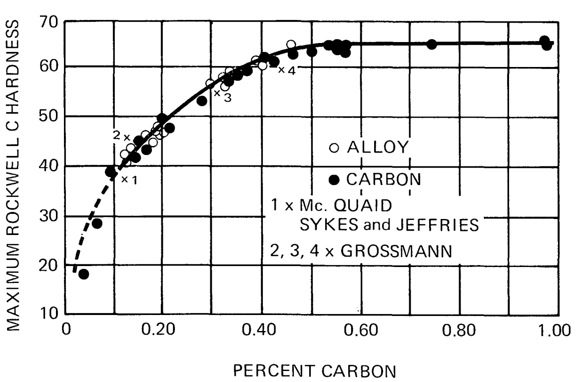


Fig. 3-4—Relationship of maximum quenched hardness of alloy and carbon steels to carbon content. *Courtesy Republic Steel Corp.*

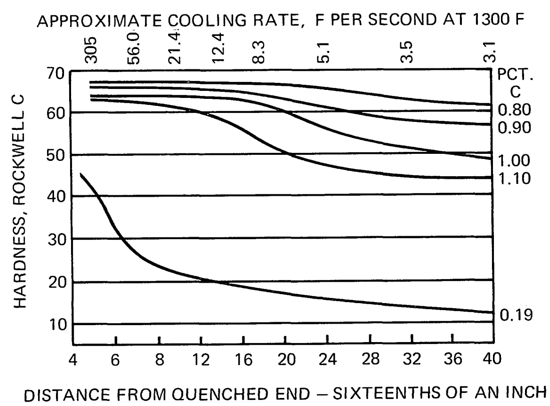


Fig. 3-7—Curves showing that maximum hardenability of 8620 Steel is achieved when case carbon concentration is at 0.80-percent carbon. *Courtesy Climax Molybdenum Co.*

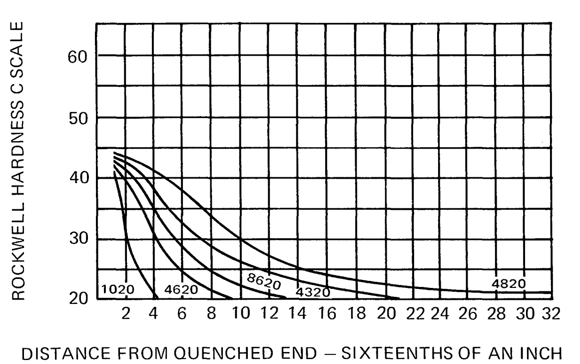


Fig. 3-5—Comparative hardenability of 0.20-percent carbon alloy steels. *Courtesy Republic Steel Corp.*

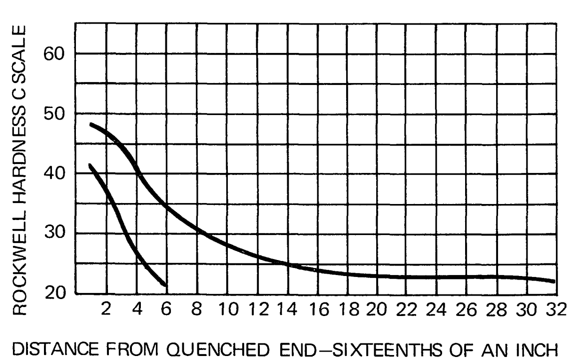


Fig. 3-8—Hardenability upper and lower curve limits for 8620H steel. *SAE Iron and Steel Handbook Supplement 30.*

Heat Treatment

Heat treating steel gears involves many different types of operations. They all have the common purpose of producing the micro-structure which will result in certain optimum properties in the gear.

Heat treating processes can be divided into two classes:

1. Those that tend to produce properties desirable for machining.
2. Those that tend to produce desirable properties in the finished gear.

Heat Treating for Machinability

Obviously, machinability is important in gear selection. It affects both tool life and manufacturing costs. A properly selected gear steel is machinable, has good mechanical properties, and has low heat treat distortion. Briefly, machinability is that property of a material which permits it to be cut economically to the size, shape and finish required in the end product.

Vacuum-Melted and Vacuum-Degassed Steels

Vacuum-melting and vacuum-degassing are processes used in steel production which do not affect composition but reduce gas content and minimize inclusions. Inclusions are small non-metallic particles which are always present to some degree in steels. They can have harmful effects during machining because of the non-uniform characteristics of the steel caused by the presence of inclusions.

The vacuum processes improve hot workability, improve ductility of low alloy steels, result in improved fatigue and impact properties, improve tensile strength, and improve stress-rupture properties. Improvement in mechanical properties can range from 5 to 50-percent for vacuum-degassed steels, and from 30 to 300-percent for vacuum-arc-melted steels.

Vacuum-degassed steels are priced lower than vacuum-melted steels. However, where one or more of the properties mentioned is important, they offer real advantages over air-melted steels in gear production. Vacuum-melted steels, of course, offer even more advantages at a somewhat higher price.

Regardless of the type of steel selected, fine-grained steels result in higher dynamic strength and less heat treat distortion than coarse-grained steels. ASTM grain sizes from 6 to 8, Fig. 3-9, are preferrable for gear applications.

Annealing

Steel is annealed to alter physical and mechanical properties, to improve machinability, to alter microstructure, to relieve stresses and to remove gases. There are a number of different annealing procedures. Terms such as full-annealing, isothermal-annealing and process annealing are used to distinguish them.

Full-Annealing: This is a softening process in which the steel is heated to a temperature above the transformation range (1400 to 1600F), held for sufficient time to complete the transformation to austenite and slowly cooled to a temperature below the transformation range. Steels containing up to 0.83-percent carbon will consist of ferrite and pearlite after annealing, Fig. 3-10. Steels containing more than 0.83-percent carbon will consist of pearlite and cementite.

Isothermal-Annealing: This is a modification of full-annealing that results in better control of the formation of pearlite. The pearlitic structure resulting, usually has better machinability in carbon and alloy steels containing 0.20 to 0.60-percent carbon. Machining operations which are facilitated by the pearlitic structure resulting from isothermal annealing include: Broaching, shaving, milling and drilling.

Spheroidizing

This operation consists of heating steel to a temperature just below the transformation temperture, holding it at that temperature for a specific length of time, and then allowing the steel to cool. Spheroidizing produces a spherical form of carbide in the steel structure, Fig. 3-11. Spheroidized steel has improved machinability in the higher carbon steels (0.040-percent or higher).

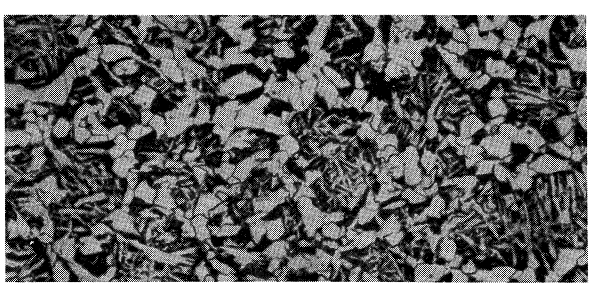


Fig. 3-9—A 100X magnification of the Widmanstatten structure of a No. 6 grain size alloy steel of hardness 87RB. *Courtesy Latrobe Steel Co.*

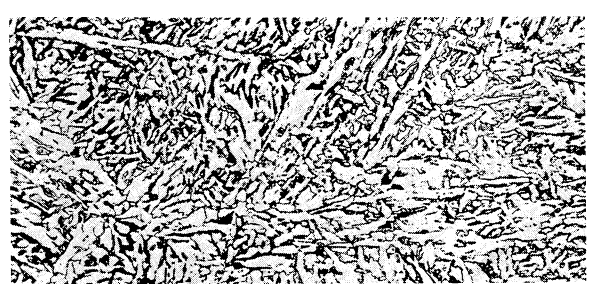


Fig. 3-10—A 250X magnification of 8640 steel structure with a full anneal. *Courtesy Latrobe Steel Co.*

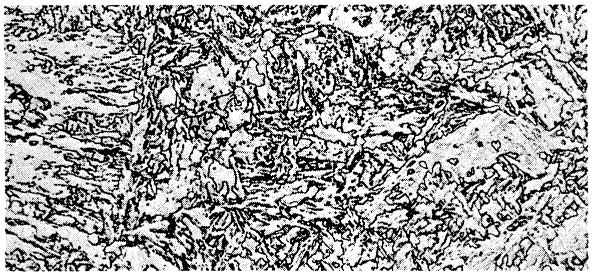


Fig. 3-11—A 250X magnification of 8640 steel structure with spheroidize anneal. *Courtesy Latrobe Steel Co.*

Stress-Relieving

This process reduces internal residual stresses that might cause problems in service, or improves dimensional stability. Stress-relieving is performed by heating to a suitable temperature and holding at that temperature for the proper time. Temperature and time at temperature depend upon the composition of the steel and requirements for stress-relief.

Stress-relieving may be performed between machining operations, and is sometimes performed after other heat treating processes to relieve stresses induced during the previous heat treatment.

Normalizing

In this process, the steel is heated to a temperature about 100F above the annealing, or upper transformation temperature, and then allowed to cool in air at room temperature. Normalizing's purpose is to obtain a homogeneous, unstressed structure of proper grain size. Structures obtained are pearlite and ferrite (low-carbon steel), Fig. 3-12; pearlite (medium-carbon steel); or pearlite and cementite (high-carbon steel).

With low-carbon steels, normalizing should leave the steel soft enough for machining and adequately prepare the steel for following processes such as carburizing or hardening. With medium to high carbon steels, it is usually necessary to follow normalizing with an annealing operation before machining.

Normalizing is similar to annealing but produces a harder, stronger, less ductile steel than annealing because of the faster cooling rate.

Final Heat Treatment

Two different types of final heat treatment processes are applied to steel gears. One type is through-hardening. The other is case-hardening. There are many different processes for case-hardening gears. Flame and induction hardening processes are also applied as final heat treatment methods.

Through-Hardening

This process is used with medium carbon steels (0.40 to 0.50-percent carbon) when a very high order of dimensional accuracy and stability combined with good physical properties is necessary. By through-hardening, in contrast to case-hardening, and then tempering to a machinable hardness before final machining, all final heat treat distortion is avoided.

Through-hardening is performed by first heating the entire gear in a furnace to the required temperature, quenching the gear, and then tempering to the required hardness.

Quenching: All hardening processes include a quenching operation. The objective of the quenching operation is to cool the steel fast enough to

Fig. 3-12—A 100X magnification of 4027 steel structure as normalized. *Courtesy Latrobe Steel Co.*

obtain a martensitic structure. If the rate of cooling is too slow, partial transformation will occur and the steel will contain other structures which will reduce hardness and alter the mechanical properties. Commonly used quenching media are air, oil, water or brine. Air has the slowest

Fig. 3-13—Quenching an 8620 alloy steel gear on a plug in a quenching press. *Courtesy Clark Equipment Co.*

cooling rate, and brine the fastest. The correct medium to use depends on the composition of the steel and the section thickness.

All surfaces of a gear being quenched should make contact with the quenching medium at as close to the same time as possible to prevent distortion. Therefore, none of the surfaces should be

Table 3-10—Mechanical Properties of Ferrous Gear Materials

Type	Grade	Heat Treatment	Hardness, Bhn	Max. Diam. or Section (in.)	Min. Tensile Strength (1000 psi)	Min. Yield Point (1000 psi)	Min. Elong. in 2 in. (%)	Min. Reduc. of Area (%)
Carbon Steels								
Hot-rolled & cold-finished bars	1020	None	—	Any size	55	25	20	48
Cold-drawn bars	1020	None	111 min	2-3	55	45	15	35
			121 min	1¼-2	60	50	15	35
			131 min	⅞-1¼	65	55	16	40
			143 min	⅝-⅞	70	60	16	40
Hot-rolled & cold-finished bars	1120	None	—	Any size	55	25	20	35
Cold-drawn bars	1120	None	111 min	2-3	55	45	15	35
			121 min	1¼-2	60	50	15	35
			131 min	⅞-1¼	65	55	16	35
			143 min	⅝-⅞	70	60	18	35
Hot-rolled & cold-finished bars or forged rounds; Open-die forgings; Closed-die forgings	1045	None	—	To 7½	75	40	23	36
		Annealed or normalized & tempered	—	7½-8	75	40	23	36
				8-12	75	40	21	35
				12-20	75	37	20	32
		Quenched & tempered	210-250	8-15	90	50	18	35
			225-265	6-8	95	55	16	35
			225-265	To 6	97	55	17	35
			245-285	4-6	100	75	15	35
			245-285	To 4	105	80	16	38
Cold-drawn bars	1045	None	163 min	2-3	80	70	10	30
			170 min	1¼-2	85	75	10	30
			179 min	⅞-1¼	90	80	11	30
			187 min	⅝-⅞	95	85	12	35
Hot-rolled & cold-finished bars or forged rounds; Open-die forgings; Closed-die forgings	1146	None	—	To 7½	75	40	23	36
		Annealed or normalized & tempered	—	7½-8	75	40	23	36
			—	8-12	75	40	21	35
			—	12-20	75	37	20	32
		Quenched & tempered	210-250	8-15	90	50	18	35
			225-265	6-8	95	55	16	35
			225-265	To 6	97	55	17	35
			245-285	4-6	100	75	15	35
			245-285	To 4	105	80	16	38
Cold-drawn bars	1146	None	163 min	2-3	80	70	10	30
			170 min	1¼-2	85	75	10	30
			179 min	⅞-1¼	90	80	11	30
			187 min	⅝-⅞	95	85	12	35
Cast Steel	0.40-0.50 Carbon	Annealed or normalized & tempered	160 min	Any size	70	35	18	25
		Quenched & tempered	Same as 1045					
Alloy Steels								
Hot-rolled bars or forged rounds; Open-die forgings; Closed-die forgings	4340 (Cr-Ni-Mo)	Normalized & tempered	225-265	To 10	100	75	18	40
			225-265	10-30	100	70	17	40
		Quenched & tempered	225-265	To 30	105	80	18	40
			245-285	To 30	115	90	17	40
			265-305	To 8	125	100	16	40
			265-305	8-30	115	90	15	35
			285-325	To 25	125	100	14	35
			310-350	To 15	135	110	13	35
			335-375	To 15	150	125	13	35
			360-415	To 15	160	130	12	35
			375-430	To 10	170	140	10	32
Open-die forgings	4350	Quenched & tempered	285-325	Over 25	125	100	14	35
			310-350	15-30	135	110	13	35
			335-375	15-30	150	125	13	35
			360-415	15-30	160	130	12	35
			375-430	10-25	170	140	10	32
			400-460	To 20	180	150	8	29
			430-495	To 20	195	165	6	25

From Machine Design, June 20, 1968

Table 3-10—Mechanical Properties of Ferrous Gear Materials *cont'd.*

Type	Grade	Heat Treatment	Hardness, Bhn	Max. Diam. or Section (in.)	Min. Tensile Strength (1000 psi)	Min. Yield Point (1000 psi)	Min. Elong. in 2 in. (%)	Min. Reduc. of Area (%)
Alloy Steels *cont'd.*								
Hot-rolled bars or forged rounds	4150 RS	Quenched & tempered	225-265	To 10	105	80	17	40
Open-die forgings			245-285	To 10	112	87	16	40
Closed-die forgings			265-305	To 9	120	95	15	38
			285-325	To 7	127	100	14	35
			310-350	To 6	135	105	12	30
			335-375	To 6	145	115	12	30
			360-415	To 6	155	123	10	30
			375-430	To 5	160	128	9	25
			400-460	To 5	176	137	8	25
Hot-rolled & cold-finished bars	4140	Quenched & tempered	210-250	To 4	100	80	17	40
Open-die forgings			225-265	To 4	95	75	18	40
Closed-die forgings			245-285	To 4	110	85	15	40
			265-305	To 4	120	95	14	36
			285-325	To 4	125	100	13	33
			310-350	To 4	135	105	12	30
			335-375	To 4	145	115	12	30
Cast Steel								
	0.30-0.40C 0.70-1.00Mn 0.15-0.25Mo	Annealed or normalized & tempered	160 min	Any size	80	45	23	35
		Quenched & tempered	210-250	To 4	90	60	15	35
			225-265	To 4	100	70	14	33
			245-285	To 3	110	80	13	31
				To 4	110	80	13	31
			265-305	To 2	120	90	11	28
				To 3	120	90	11	28
			285-325	To 1	130	100	10	26
				To 2	130	100	10	26
			300-340	1 in. max.	135	105	9	23
	0.27-0.37C 0.70-1.00Mn 0.60-0.90Cr 0.60-0.90Ni 0.30-0.40Mo	Quenched & tempered	210-250	4-15	90	60	15	35
			225-265	4-15	100	70	15	33
			245-285	3-15	110	80	13	31
			265-305	2-15	120	90	11	28
			285-325	1-15	130	100	10	26
			300-340	1-15	135	105	9	23
			335-375	To 15	150	120	8	20
			360-415	To 15	165	135	7	18

Cast (Gray) Iron

Class	Min. Tensile Strength (1000 psi)	Min. Hardness on Tooth, Bhn	Section Size or Diam. (in.)
20	20	—	None
30	30	175	9
35	35	185	7
40	40	200	5
50	50	215	2¼
60	60	220	1¼

Nodular Iron

Heat Treatment	Min. Hardness on Tooth, Bhn	Min. Tensile Strength (1000 psi)	Min. Yield Strength (1000 psi)	Min. Elong. in 2 in. (%)	Section Size or Diam. (in.)
Anneal or stress-relieve	165	65	45	10	Any size
Normalize & temper	180	70	55	7	Any size
	210	85	70	5	Check if section size exceeds 4 in.
	225	89	75	4	
	255	103	87	3	
	265	107	92	2	
	265	107	92	2	
	285	115	100	1.5	
	300	123	105	1	
	350	143	123	0.5	
Quench & temper	210	98	75	7	
	225	105	82	6	
	255	115	90	4	
	265	120	95	3.5	
	285	130	105	3	
	300	135	110	2.5	
	350	158	130	1	

shielded from the quenching medium by other objects. Gears are sometimes quenched on plugs, in clamps, in special fixtures and in presses to reduce or prevent distortion, Fig. 3-13.

In some high alloy steels there is a tendency to retain austenite after the quench. With these steels it is sometimes advantageous to deep-freeze the parts to minus 120F, and then temper them. This assures completion of the transformation of austenite to martensite.

Martempering: This special quenching method minimizes distortion and reduces internal strains in parts that are not of uniform cross-section. It consists of first quenching the part, which has been heated to transformation temperature, in hot oil or salt at 400 to 600F. Parts are then held at this temperature for from one to 10 minutes, according to size, and then allowed to cool slowly in air to room temperature. Modified martempering is also used. This is the same as martempering except for lower temperature of the quench medium, 200 to 300F. The lower temperature allows hardening of certain steels which would not fully harden at the higher temperature.

Material Selection: Through-hardened gear steels are selected on the basis of the alloy content required to get the proper hardness in a given diameter or section size. This diameter or section size refers to the minimum dimension of diameter, face width, or blank rim thickness (or shell dimension for bored pinions). For example, in a solid disc-type gear of 10-in. diameter and 2-in. face width; the 2-in. face is the section thickness. For a 4-in.-dia. pinion with an 8-in. face integral with a 2-in. shaft; the 4-in. diameter is the section thickness. When a bore diameter is less than 20-percent of the length of the bore, the diameter of the blank is taken as the section thickness.

There are many chemical compositions used for through-hardened gearing. In addition to the plain carbon steels, 4140 chrome-moly and the 4340 chrome-nickel-moly compositions are widely used. The carbon content of gear steels is normally held between 0.30 and 0.50-percent. Table 3-10 lists the most widely used steels and cast irons. Frequently, when the chosen alloy steel composition is marginal for either the section size or heat treatment, controlled-hardenability (H-grade) steels such as 4340H are used. Otherwise, the next higher grade alloy may be used.

Case-Hardening

Case-hardening is a term used to describe many different types of heat treating operations, all of which have the same objective: To obtain a hard wear-resistant surface on gear teeth while retaining a softer, more ductile core to absorb impact loads without breaking. Core mechanical properties for typical carburized steels are shown in Table 3-11.

Carburizing: Carburizing is probably the most widely used method of case-hardening. There are different types of carburizing: Gas, liquid and pack. In each type, the objective is to supply carbon to the surfaces to be hardened and thus increase the carbon content of the steel which permits higher hardness to be obtained. Ob-

Table 3-11—Core Mechanical Properties of Carburized Steels

(Based on Forged Round Bars, Normalized, Machined to 0.540-in. Diameter, Pseudocarburized at 1700° F for 8-Hr., Cooled to Room Temperature, Reheated, Oil Quenched and Tempered, Tested in—.505-in. Dia. Rounds)

Grade	Condition	Hardness (Bhn)	Tensile (1,000 psi)	Yield (1,000 psi)	Elongation (Percent)	Area Reduct. (Percent)
1022	1425F Reheat	170	80	55	30	68.5
	1475F Reheat	170	81	51.5	30	70.5
	1525F Reheat	179	82	58	29.5	72.5
	Quenched After Pseudocarburization	179	83	60	30	71
2317	1425F Reheat	277	132	103	15	50
	1475F Reheat	293	139	110	15	50
	1525F Reheat	302	142	115	16.5	55
	Quenched After Psuedocarburization	321	151	125	14.5	44
E3310	1425F Reheat	321	152	128	15.5	54
	1475F Reheat	363	171	150	16	54
	1525F Reheat	363	176	154	16	54
	Quenched After Pseudocarburization	375	185	159	15	52
4320	1425F Reheat	321	151	126	15	49
	1475F Reheat	331	157	130	16	50
	1525F Reheat	352	171	147	16	53
	Quenched After Pseudocarburization	375	180	148	14.5	46
4620	1425F Reheat	277	132	107	14	48
	1475F Reheat	293	139	114	16.5	52
	1525F Reheat	302	145	117	17	55
	Quenched After Pseudocarburization	311	147	122	16	50
4820	1345F Reheat	401	197.7	137.1	13.1	42.2
	1395F Reheat	388	192.6	133.4	13.8	43.8
	1535F Reheat	401	206.3	152.5	13.3	46
	Quenched After Pseudocarburization	415	210	165	12.3	43
8620	1425F Reheat	285	136	112	15	49
	1475F Reheat	321	151	123	15.5	50
	1550F Reheat	331	159	132	16.5	56
	Quenched After Pseudocarburization	331	161	134	15	53
E9310	1425F Reheat	331	155	130	15.5	52
	1475F Reheat	341	164	140	16	53
	1525F Reheat	363	174	153	16	53
	Quenched After Pseudocarburization	375	187	162	15	51

Courtesy Republic Steel Corp.

Table 3-12—Recommended Materials and Processes for Carburized Gears

Class	Service	Typical* A.I.S.I. Materials	Carburizing Methods	Min. Case Hardness, Rc	Min. Core Hardness, Rc
I	Light duty	1015, 1020, 1022, 1117, 1118	Pack, gas or salt bath	As specified	As specified
II	Moderate duty	8620, 4620, 4615 or equivalent	Pack, gas or salt bath	55	20
III	Heavy duty	4820, 4320, 2320 or equivalent	Pack, gas or salt bath	55 or 58	28
IV	Maximum perform-ance	E-3310, E-9310, E-2320 or equivalent	Pack, gas or salt bath	58 or 60	32

*For extremely large gears, it may be necessary to use the type of material recommended for the next higher class in order to meet case and core hardness requirements.

From Machine Design, June 20, 1968

viously, carburizing is employed with low carbon content steels, usually those with 0.10 to 0.25-percent carbon, Table 3-12.

Gas Carburizing: In gas carburizing, Fig. 3-14, the gears are heated to a temperature range of from 1650 to 1750F in an atmosphere containing carbonaceous gases. The gases are either supplied directly, or obtained by the vaporization of liquid hydro-carbons. The time during which the gears are exposed to this atmosphere determines the depth of the case. The method is used for both light and deep cases.

Gas Carbonitriding: This process is similar to gas carburizing. The difference is that both carbon and nitrogen are present in the furnace atmosphere. The nitrogen is provided by the introduction of ammonia into the furnace. Carbonitriding results in hard, wear-resistant case, usually 0.030-in. or less in depth. The case is harder than that obtained by carburizing and has higher resistance to softening during tempering operations.

Liquid Carburizing: This method of case-hardening steel consists of heating it above the transformation temperature in a molten salt bath, holding it at that temperature for the required length of time, and then quenching. While the gears are immersed in the bath, carbon is diffused into the metal. There are two types of liquid carburizing: Light-case and deep-case. For a light case, the bath temperature is from 1500 to 1650F. Case depth is up to 0.030 inch. With the deep case, bath temperatures go up to 1750F, and case depths are up to 0.250 inch. Advantages of this and other liquid-bath processes are: Extremely uniform carburizing, low distortion of parts, and no requirement for packing and unpacking parts as there is in certain other processes.

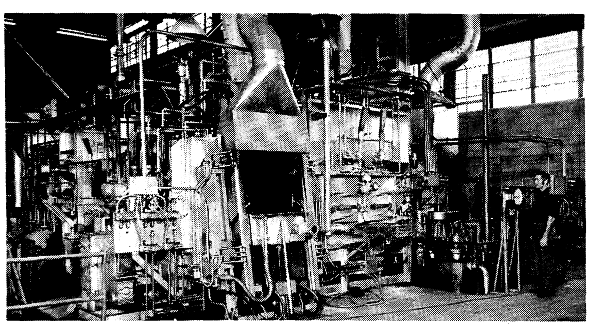


Fig. 3-14—A continuous gas carburizing furnace with alternate side flow feed for marquenching or press quenching. *Courtesy Clark Equipment Co.*

Liquid Carbonitriding: This process is similar to liquid carburizing; using a molten cyanide salt bath which supplies both carbon and nitrogen to the gears. This method is commonly used with alloy steels in the medium-carbon range (0.25 to 0.50-percent) to form an extremely hard case.

Pack Carburizing: In this method of carburizing, the gears are packed in a solid carbonaceous compound in heat-resistant alloy boxes before placing them in the heat-treat furnace. Temperature is raised to from 1650 to 1750F and held for the time required to produce the case depth wanted. During pack carburizing, carbon dioxide is formed by heating the solid carbonaceous material. The carbon dioxide immediately reacts with carbon in the compound to form carbon monoxide which supplies the carbon to the gear steel. The carbonaceous compound may be charcoal, coke or coal.

In all of the case-hardening methods discussed here, case depth increases both with temperature and time at temperature, Table 3-13. The greatest effect occurs during the first hour. The increase in case depth is progressively less as time at temperature increases.

Case Depth: There are two commonly-used methods of defining case depth: Total case depth

Table 3-13—Case Depths in Inches By Carburizing

Time in Hours	Temperature °F							
	1500	1550	1600	1650	1700	1750	1800	1850
1	0.012	0.015	0.018	0.021	0.025	0.029	0.034	0.040
2	0.017	0.021	0.025	0.030	0.035	0.041	0.048	0.056
3	0.021	0.025	0.031	0.037	0.043	0.051	0.059	0.069
4	0.024	0.029	0.035	0.042	0.050	0.059	0.069	0.079
5	0.027	0.033	0.040	0.047	0.056	0.066	0.077	0.089
6	0.030	0.036	0.043	0.052	0.061	0.072	0.084	0.097
7	0.032	0.039	0.047	0.056	0.066	0.078	0.091	0.105
8	0.034	0.041	0.050	0.060	0.071	0.083	0.097	0.112
9	0.036	0.044	0.053	0.063	0.075	0.088	0.103	0.119
10	0.038	0.046	0.056	0.067	0.079	0.093	0.108	0.126
11	0.040	0.048	0.059	0.070	0.083	0.097	0.113	0.132
12	0.042	0.051	0.061	0.073	0.087	0.102	0.119	0.138
13	0.043	0.053	0.064	0.076	0.090	0.106	0.123	0.143
14	0.045	0.055	0.066	0.079	0.094	0.110	0.128	0.149
15	0.047	0.057	0.068	0.082	0.097	0.114	0.133	0.154
16	0.048	0.059	0.071	0.084	0.100	0.117	0.137	0.159
17	0.050	0.060	0.073	0.087	0.103	0.121	0.141	0.164
18	0.051	0.062	0.075	0.090	0.106	0.125	0.145	0.169
19	0.053	0.064	0.077	0.092	0.109	0.128	0.149	0.173
20	0.054	0.066	0.079	0.094	0.112	0.131	0.153	0.178
21	0.055	0.067	0.081	0.097	0.114	0.134	0.157	0.182
22	0.056	0.069	0.083	0.099	0.117	0.138	0.161	0.186
23	0.058	0.070	0.085	0.101	0.120	0.141	0.164	0.190
24	0.059	0.072	0.086	0.103	0.122	0.144	0.168	0.195

Penetration data for various times and temperatures showing total case depth for all constructional and carbon steels in inches to base carbon.

Courtesy Republic Steel Corp.

and effective case depth. These two terms are sometimes confused. It is most important that they are not confused so that the desired case depth will be satisfied.

Total case depth is the perpendicular distance from the surface of the steel to the point where the differences in the physical and chemical properties of case and core can no longer be distinguished, Figs. 3-15, 16 and 17. Total case depth can always be used to specify a required case, but is usually used where a light or thin case is required.

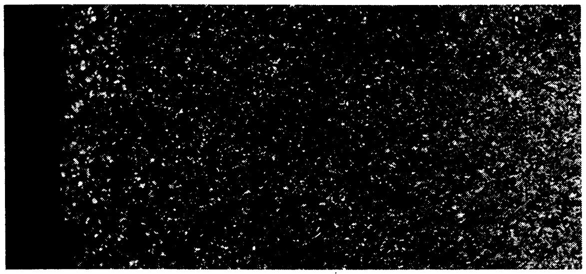


Fig. 3-15—A 75X magnification of the total case depth structure of 1113 steel. *Courtesy Latrobe Steel Co.*

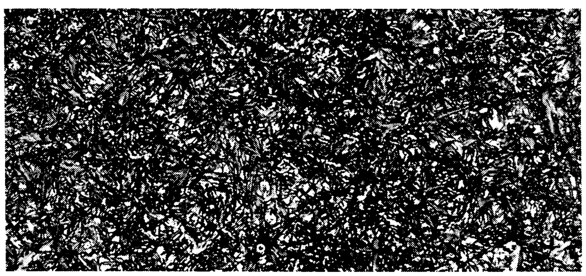


Fig. 3-16—A 250X magnification of the structure within the case of the 1113 steel specimen. *Courtesy Latrobe Steel Co.*

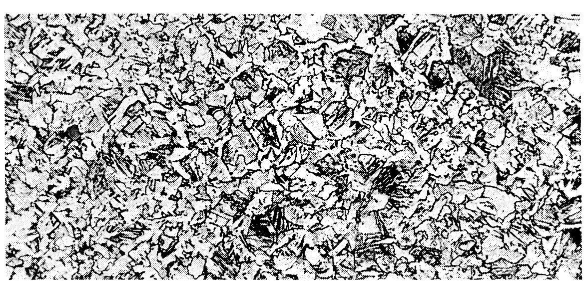


Fig. 3-17—A 250X magnification of the structure within the core of the 1113 steel specimen. *Courtesy Latrobe Steel Co.*

Effective case depth, Fig. 3-18, is the perpendicular distance from the surface of a hardened case to the greatest depth at which a specified hardness level (usually 50Rc) is maintained.

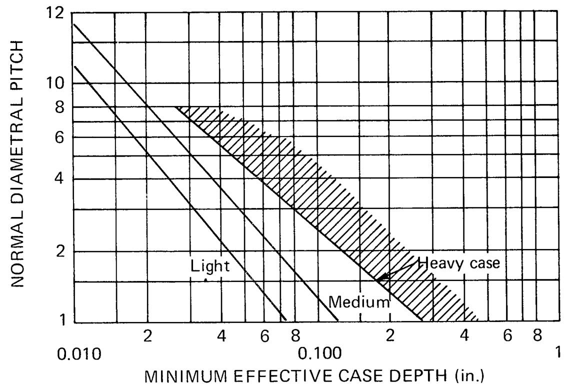


Fig. 3-18—Depth of effective case at pitch line for spur, helical and herringbone gears. *From Machine Design, June 20, 1968.*

Effective case depth will always be less than the total case depth when core hardness is less than 50 Rc.

Nitriding: This case-hardening process exposes the steel to a source of nascent nitrogen at a temperature of 950 to 1050F. Steels to be hardened by this method, Table 3-14 must contain alloying elements which will combine with nitrogen to form hard nitrides. Aluminum is the strongest nitride-forming element, with chromium, vanadium and molybdenum also being beneficial. Other alloy steels such as H-11, H-13, 4140, 4340, 5140 and 6150 can also be nitrided.

Nitriding provides improved wear resistance, anti-galling properties, higher fatigue resistance and less distortion than other case hardening processes. All hardenable steels should be hardened and tempered before nitriding. To maintain stability during the nitriding operation, the tempering temperature should be 50F above the nitriding temperature.

Required depth of nitrided cases depends on the nature of the gear, tooth thickness, size, wear resistance requirement, and other desired properties. Case depths from 0.004 to 0.035-in. are

Table 3-14—Composition of Various Nitriding Steels *Courtesy Republic Steel Corp.*

Element	MATERIAL DESIGNATION				
	AISI 7140 AMS 6470 E	AMS 6425	135 Type G	N	EZ
Carbon	0.38-0.43	0.21-0.26	0.30-0.40	0.20-0.27	0.30-0.40
Manganese	0.50-0.70	0.50-0.70	0.40-0.70	0.40-0.70	0.50-1.10
Silicon	0.20-0.40	0.20-0.40	0.20-0.40	0.20-0.40	0.20-0.40
Chromium	1.40-1.80	1.00-1.25	0.90-1.40	1.00-1.30	1.00-1.50
Aluminum	0.95-1.30	1.10-1.40	0.85-1.20	0.85-1.20	0.85-1.20
Molybdenum	0.30-0.40	0.20-0.30	0.15-0.30	0.20-0.30	0.15-0.25
Nickel	—	3.25-3.75	—	—	—
Selenium	—	—	—	—	0.15-0.25

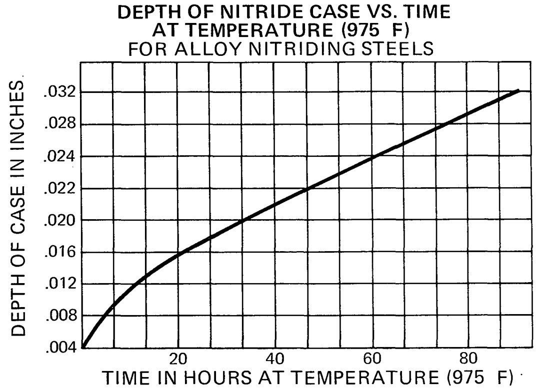


Fig. 3-19—Depth of case vs. time at 975F for alloy nitriding steels. *Courtesy Republic Steel Corp.*

produced in from 10 to 100-hours, Fig. 3-19. This rate of case formation is much slower than the formation of a carburized case. Typical surface hardness of nitrided steels are shown in Table 3-15.

Both liquid and gas nitriding processes are used. Temperature range is from 950 to 1050F with either method. Liquid nitriding uses molten cyanide as the hardening medium. This adds both nitrogen and a small amount of carbon to the steel surface.

Table 3-15—Typical Surface Hardnesses of Nitrided Steels

Steel	Hardness, Rc
4340	46
4140	48
2.5 Cr	56
Nitralloy 135 M	60

From Machine Design, June 20, 1968

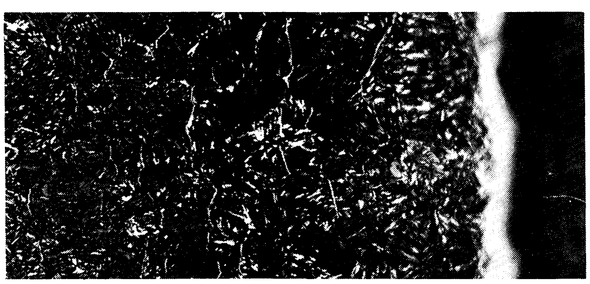


Fig. 3-20—A 250X magnification of the total case depth structure of a Nitralloy 135 specimen. *Courtesy Latrobe Steel*

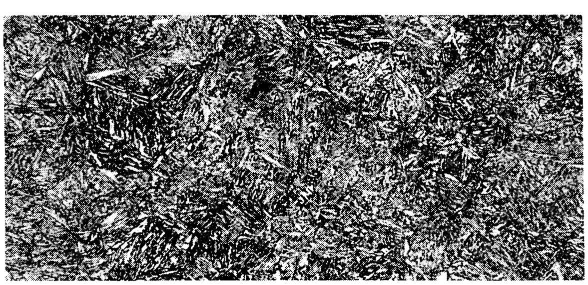


Fig. 3-21—A 250X magnification of the interior structure of the Nitralloy 135 specimen. *Courtesy Latrobe Steel Co.*

Gas nitriding employs a gas, usually ammonia, to provide the required nitrogen. Either a single or a double-stage process may be used when gas nitriding. In the single-stage process, the disassociation rate of the ammonia ranges from 15 to 30-percent. This process produces a weak, brittle, nitrogen-rich surface layer, known as a "white layer", Figs. 3-20 and 21. If this white layer is 0.0003-in. or less, it is not generally harmful to the gear. A thicker layer may, however, cause trouble. Often pickling, honing or grinding may be necessary to remove the white layer.

With the two-stage process, also known as the Floe process, it is possible to hold the white layer to a thickness of 0.0003-in. or less; which will cause no problems. The dissociation rate of the ammonia is increased to 80 to 85-percent during the second stage of the two-stage process.

A properly nitrided gear has little or no distortion. Generally, no machining is done after nitriding. If a nitrided gear is ground or honed, only a small amount of metal should be removed. Hardness of the nitrogen case decreases so much more rapidly than a carburized case that a nitrided case could be ruined by removing as much metal as would be quite permissible with a carburized case.

Despite the thin nitride case, nitrided gears have performed well in many critical applications. The extra-high hardness of the outer case compensates for the lack of case depth.

Salt Bath Nitriding: A typical proprietary process (Tufftride) for ferrous metals raises endurance limits and increases wear resistance. The salt bath consists primarily of cyanide and cyanate compounds that liberate specific quantities of carbon and nitrogen in the presence of ferrous (steel or cast iron) materials.

The process is performed after machining and heat-treating. In the salt bath, nitrogen diffuses into the work to improve fatigue properties while the carbon forms iron-carbide particles at or near the surface. These particles act as nuclei, precipitating some of the diffused carbon to form a wear-resistant compound zone of carbon-bearing epsilon iron nitride.

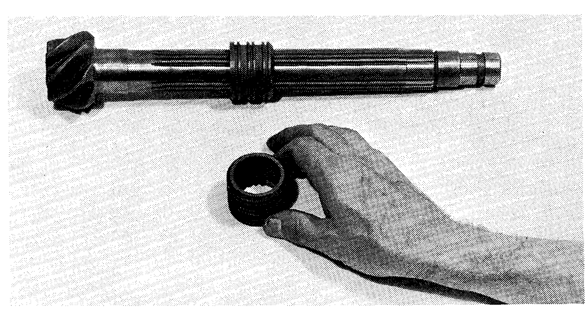


Fig. 3-22—The sliding sleeve on this lift truck transaxle assembly has a 0.001-in. hole tolerance. It is made from 1144 bar stock with 30-35 R_C core hardness. Tufftriding is done after finish-machining to reduce distortion and increase wear resistance in both internal spline and external piston ring grooves. *Courtesy Clark Equipment Co.*

In carbon and low-alloy steels, a 90-minute Tufftriding sequence produces a compound zone about 0.0004-in. thick, with a 0.035-in. deep total diffusion zone. A typical Tufftrided splined oil distribution sleeve with external piston ring grooves is shown in Fig. 3-22.

Tooth Distortion: Carburizing and carbonitriding processes always produce some changes in gear form. Both involute form and helix angle in helical gears will be affected. Consequently, these changes should be recognized and compensated for in the machining and shaving operations preceeding the final heat treatment. There are some instances where the changes in mating gears are such as to compensate for each other and thus require no compensation.

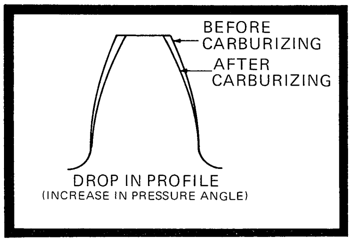


Fig. 3-23—Drop in involute profile during heat treatment of a case-hardened gear tooth.

Carburizing produces a drop in profile, Fig. 3-23, (increase in pressure angle) which varies with pitch and case depth as follows:

Diametral Pitch	Profile Drop, In.
14 to 18	0.0002 to 0.0003
8 to 12	0.0005 to 0.0007
4 to 6	0.0008 to 0.0010

Pressure angle may drop even more for gears of coarser pitch and deeper cases than those listed. The shrinkage of large-diameter gears must also be given special consideration, depending on the material.

Carburizing causes unwinding, or decrease in helix angle, in helical gears. Approximate values for this decrease in helix angle per inch of tooth length are as follows:

Helix Angle-Deg.	Lead Increase, In.
5 to 10	0.0002 to 0.0003
15 to 20	0.0004 to 0.0005
25 to 30	0.0006 to 0.0007
35 to 40	0.0008 to 0.0010

These values are only guidelines since material and the type of heat treatment produce variations. In general, the thicker the gear, the less movement or form change will occur.

Ring Gear Out-Of-Roundness: The correction of up to 75-percent of the out-of-roundness condition of case-hardened internal ring gears can be carried out on a special automated ring gear rounder, Chapter 10.

The machine burnishes the internal gear teeth and corrects out-of-roundness with a pair of burnishing gears. One gear is a fixed driver rotated by a motor drive. The other gear, which does not mesh with the driving gear is moved outward under the control of an air cylinder linkage to exert pressure on the ring as it is rotated.

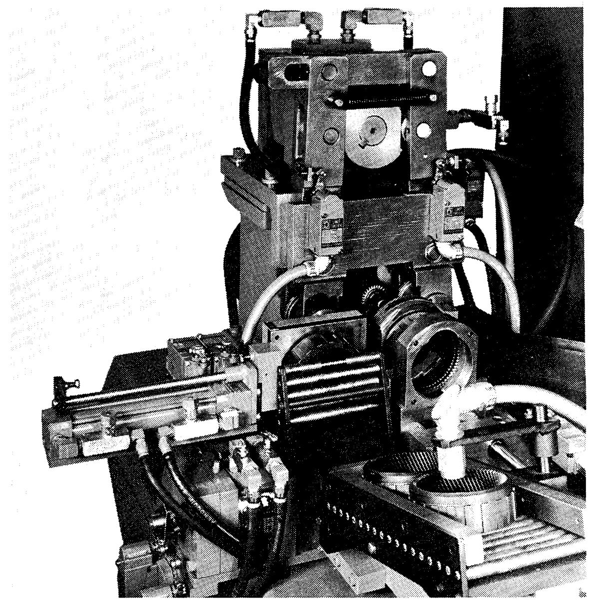


Fig. 3-24—Closeup of an automated ring rounding machine for correcting out-of-roundness of case-hardened internal gears at high production rates.

A pair of opposed plain rolls, mounted on an individual yoke member, are brought into pressure contact with the outside diameter of the ring gear as it is rotated.

The case-hardened automotive transmission input gear on the ring rounder in Fig. 3-24 is 1.690-in. wide and has a 4.797-in. outside diameter. It has 66 internal helical involute teeth on a 3.9688-in. pitch diameter. Ring gear out-of-roundness is corrected by the machine at a rate of up to 300 gears per hour.

Flame-Hardening

Flame-hardening is a heat treating process, Fig. 3-25, which permits localized or selective hardening of gear teeth, shafts or other parts where it may be desirable to harden only a portion of the part by selective application of a high-temperature flame to the area to be hardened. Temperature of the selected area is rapidly raised above the transformation temperature by heating. Parts are then cooled, or quenched, Fig. 3-26 to obtain the desired properties.

This process can be used with cast iron (Fig. 3-27), ductile iron, or any steel that is hardenable by furnace heat treatment. Plain carbon steels with a carbon content from 0.35 to 0.50-percent are the materials most often flame-hardened. This permits savings on material costs in many applications.

Equipment for flame-hardening may use any one of a number of gases such as natural gas, manufactured gas or acetylene depending upon availability and cost. Different types of burners and burner tips are commonly available; their

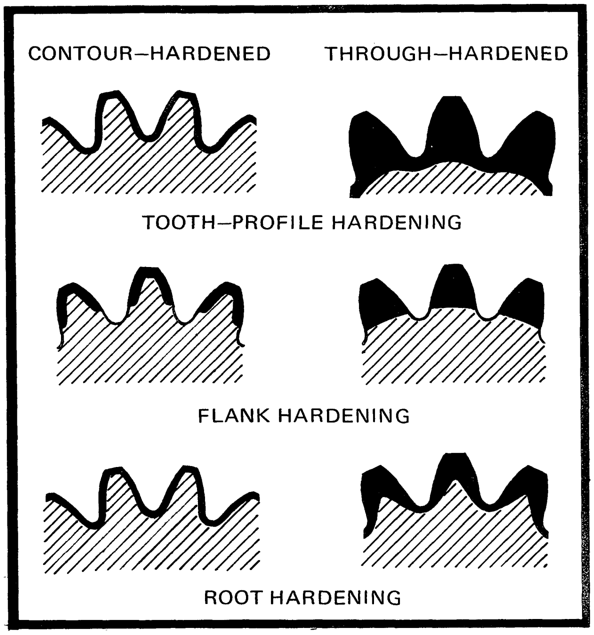


Fig. 3-25—Hardening patterns for flame and induction-hardened gear teeth. *From Machine Design, June 20, 1968.*

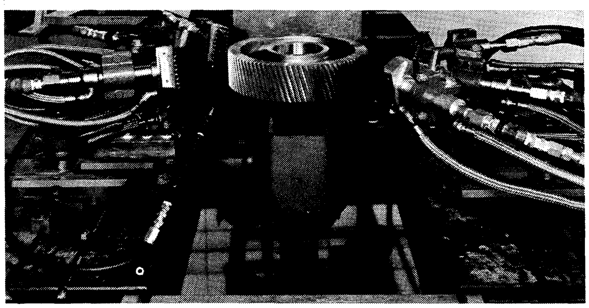


Fig. 3-26—A setup for flame-hardening the teeth on an external helical gear. *Courtesy Tocco Div.*

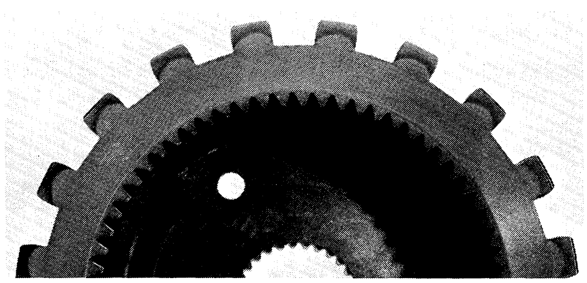


Fig. 3-27—Flame-hardening patterns on the external clutch teeth of a cast iron automatic transmission gear. *Courtesy Tocco Div.*

usage depending upon hardness, depth of case and transition zone requirements. Equipment for flame-hardening commonly incorporates a quench tank or a spray quench system as a part of the machine.

Compared with induction hardening, flame hardening requires lower capital investment. Flame hardening is more flexible for a range of part sizes, and can reach into isolated contours. However, flame hardening is slower than induction hardening, offers limited control of the heated area, and has a high energy cost per piece.

Induction-Hardening

This method, Fig. 3-28 also produces selective case hardening by localized heating. High frequency (1,000 cycles per second or higher) alternating current is used to induce a current flow and thus heating in the part. Because of an electrical phenomenon known as the "skin effect",

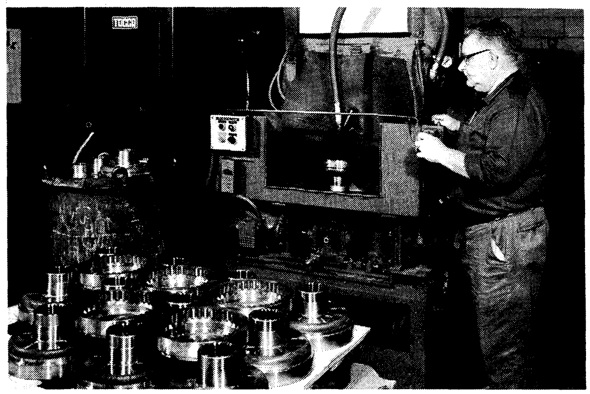


Fig. 3-28—A setup for induction-hardening the internal spur gear teeth on a 1045 steel wheel hub gear for a heavy-duty lift truck. *Courtesy Clark Equipment Co.*

the depth of the heated area is inversely proportional to the frequency used, Table 3-16. With fine-pitch gears, teeth are usually through-hardened.

Design of the heating coil is an important factor in successful induction-hardening and depends upon dimensions and configuration of the part to be heated, the heat pattern required, and the amount of power available, Fig. 3-29.

Induction heating is the fastest known way of heating a gear. Heating times range from 0.2 to 120-seconds. Because it is so fast, surfaces

Table 3-16—Approximate Minimum Depths of Hardness for Induction-Hardening of Steel Parts.

Frequency (Cycles per Sec.)	Approximate Depth (in.)
3,000	0.150
10,000	0.060 to 0.080
500,000	0.020 to 0.040
1,000,000	0.010 to 0.020

Courtesy Tocco Div.

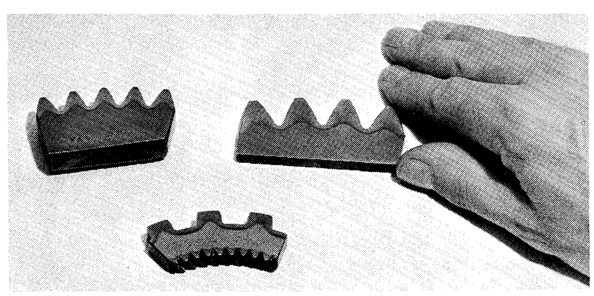


Fig. 3-29—Typical induction-hardening patterns on automatic transmission steel and cast iron gear teeth. *Courtesy Tocco Div.*

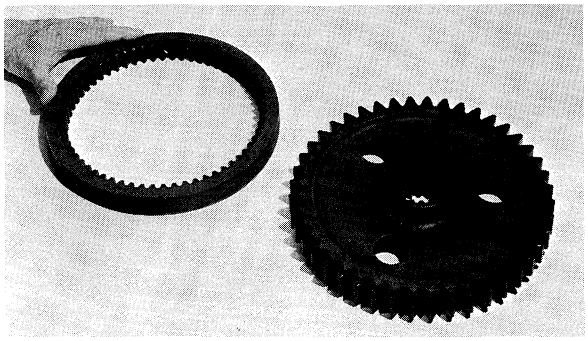


Fig. 3-30—Induction-hardened 5150 steel internal gear teeth, left, and 4140 steel external spur teeth, right. *Courtesy Tocco Div.*

remain clean and free from decarburization and scale; and core metal retains its original properties.

A relatively new method of induction-hardening is the Natco/Delapena process. It is performed with an intensifier that oscillates back and forth in the valley between gear teeth. The intensifier conforms very closely to the gear tooth profile. This is done with the gear submerged in a quench tank. The process is relatively slow because only one valley is processed at a time.

Like flame-hardening, induction-hardening is used with plain carbon or alloy steels, Fig. 3-30, that provide good case hardness combined with desirable core properties. Induction hardening can also be effectively applied to hardening the teeth on parts made of pearlitic malleable iron with a uniform structure and 0.50-percent carbon in solution, Fig. 3-31.

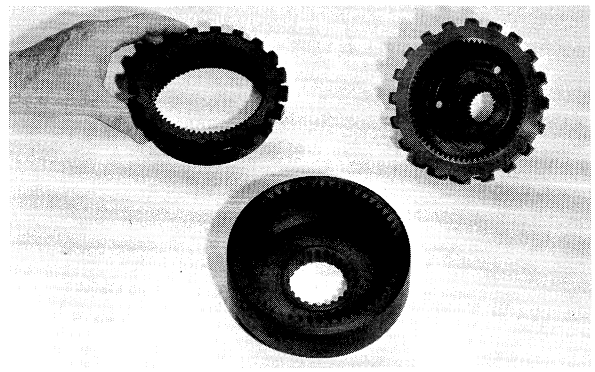


Fig. 3-31—Typical induction-hardened pearlitic malleable iron internal gear teeth on automotive automatic transmission gears. *Courtesy Tocco Div.*

Heat-Treating Definitions

Austenite—A form of steel structure which ordinarily does not exist at normal temperatures. Characterized by carbon in solid solution. Also, austenite is tough and non-magnetic. If steel is cooled from critical temperature very rapidly, some austenite may not be transformed during cooling. However, transformation will then occur over a long period of time and create stresses due to "growth" because the form into which it changes has lower density and higher volume.

Cementite—A constituent of fully-annealed carbon steels with carbon content higher than 0.83-percent. Cementite is the chemical compound iron carbide containing 6.67-percent carbon and 93.33-percent iron.

Critical Cooling Rate—Rate of cooling from transformation range which determines whether austenite changes into martensite, cementite, pearlite or ferrite. When cooling is faster than critical cooling rate, martensite is formed.

Draw—See Tempering.

Ferrite—A constituent of fully-annealed carbon steels with carbon content less than 0.83-percent. Ferrite is soft, magnetic iron.

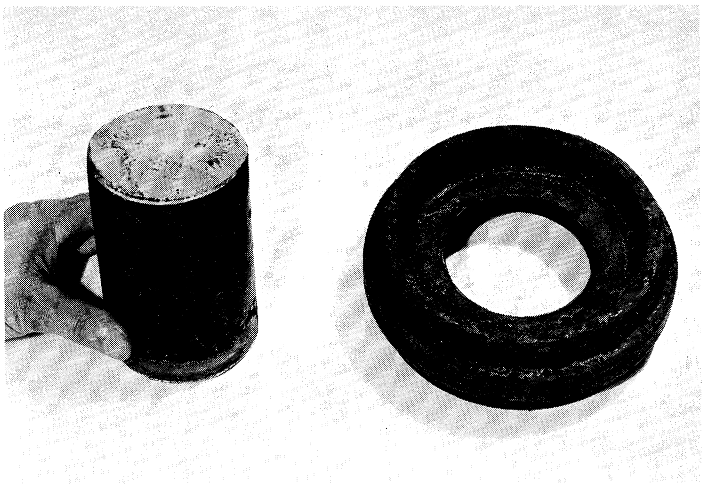
Martensite—Internal structure obtained in steel when critical cooling rate is exceeded. Martensite has a needle-like structure as viewed through a microscope and has the highest hardness of any structure obtained by decomposition of austenite.

Pearlite—An intimate mechanical mixture of cementite and ferrite which contains 0.83-percent carbon and 99.17-percent iron, neglecting impurities. Fully-annealed steel containing 0.83 percent carbon consists entirely of pearlite. The name came from mother-of-pearl-like appearance when viewed through a microscope.

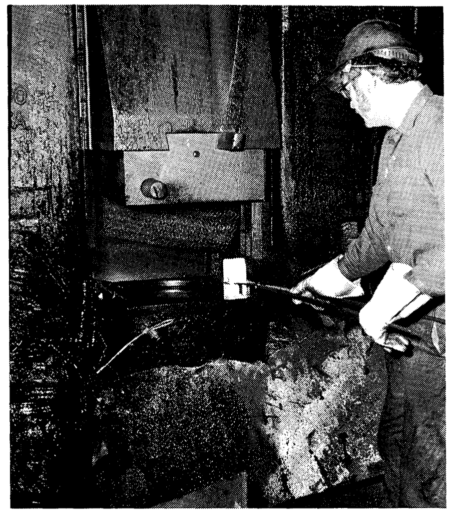
Quenching—Rapid cooling from a high temperature using liquid, gaseous, or solid material as a quenching medium. When the critical cooling rate is exceeded, martensite is formed rather than ferrite, pearlite, or cementite.

Tempering—Reheating a quench-hardened or normalized steel to some temperature below the transformation range and then cooling it to reduce brittleness reduce hardness, or remove internal strains caused by quenching.

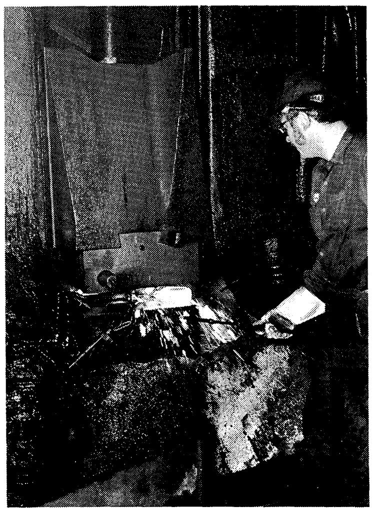
Transformation Range — Temperature range where austenite forms when steel is heated or disappears when cooled. Actually two different ranges depending upon whether steel is being heated or cooled. Ranges may overlap but are not the same. Actual temperature depends on composition and rate of temperature change, particularly when cooling.



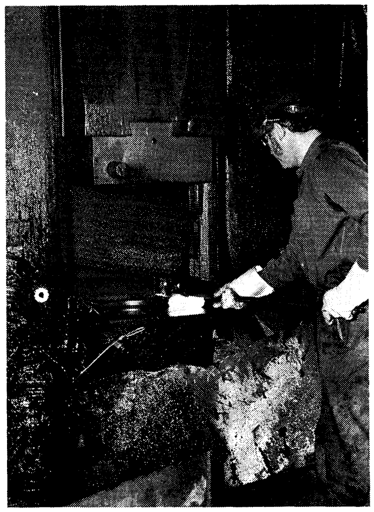
Sheared Slug and Finished Forging for a heavy duty transmission clutch gear. Forging is $7\frac{9}{32}$ in. dia. $2\frac{9}{16}$ in. thick and has a $3\frac{9}{16}$ in. dia. pierced hole.



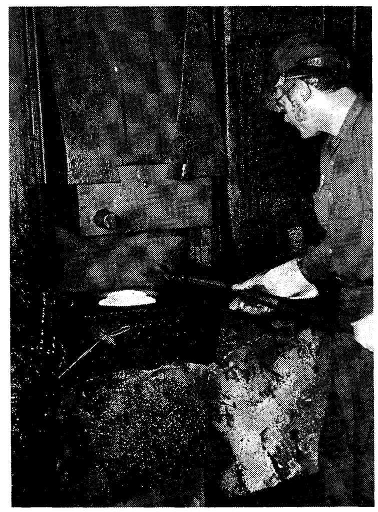
1. Slug is heated to 2350F in a gas furnace and hit once on side of die in steam hammer.



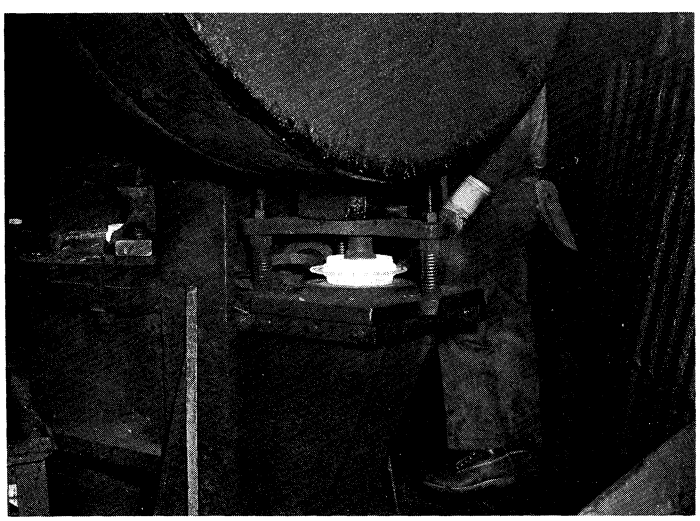
2. Shape of pancake after first hit in steam hammer.



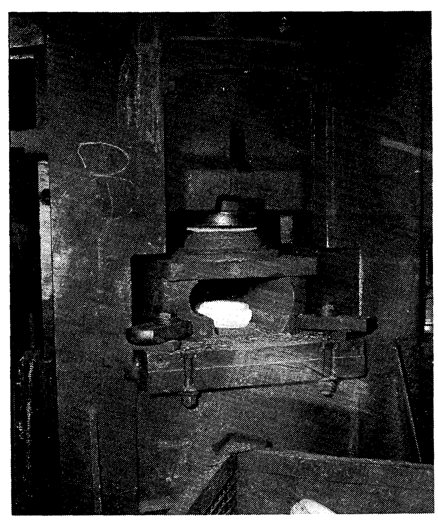
3. Second pancake hit in steam hammer.



4. Pancake is placed in steam hammer die and hit four times.



5. Forging is pierced in side arm on mechanical trim press.



6. Forging is trimmed in die on trim press.

Fig. 3-32—Typical hot forging sequence for producing an 8620 heavy-duty transmission clutch gear. *Courtesy Clark Equipment Co.*

Blank-Forming Method

Casting, rolling, hot-forging, cold-forging, pressing, welding, and extruding are among the processes used successfully for forming gear blanks. In general, the best method is the one which produces the most economical blank. This includes cost of the finishing operations. However, cost is not the only criterion. Also important is the effect of the process on mechanical properties; this involves a study of directional properties.

For a casting, there is little difference between the properties in various directions. Provided that the casting is sound, castings generally offer better mechanical qualities than forgings or rolled bars.

Rolled Bars

Rolled bars, which undergo considerable reduction from the ingot to the finished diameter, have significant directional properties. Banding or segregation of metallurgical constituents may

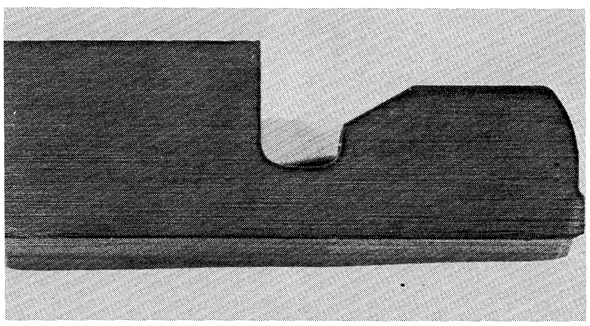


Fig. 3-33—Etched cross section of gear made from bar stock. *Courtesy Clark Equipment Co.*

occur. This can cause both a reduction in fatigue and reduced impact strength. Sometimes these directional influences are aggravated by "carry-over" of the original bloom or ingot shape into the final bar. In spite of these shortcomings, rolled bars have been widely and successfully used for gears.

When gears are to be cut from bar stock, Fig. 3-33, annealing or normalizing may be advisable; but these processes are often unnecessary.

Hot Forging

Hot forging techniques, Fig. 3-32, have a significant effect on many properties of gear blanks. Excellent gear design, ideal materials and perfect handling of all other phases of gear manufacture will not produce good gears if the forging is faulty. Forging practice affects machinability, ultimate strength, heat treat distortion, final finish, tool life, over-all manufacturing costs and service life of the finished gear.

Complete filling of a forging die is necessary to obtain the required density and eliminate porosity in forged gear blanks. This condition is indicated by sharp rather than rounded corners on the forging, Fig.3-34. Many difficulties in gear production and in service have been traced back to inadequate filling of forging dies.

Techniques used to assure complete filling of forging dies include: increasing forging temperature to assure adequate plastic flow; providing scale relief and flash pockets in the forging dies; increasing size of the forging blank, and using a piercing mandrel to force metal more thoroughly into the tooth section of the forged blank.

A good forged gear blank has maximum and highly uniform density. No scale or other foreign

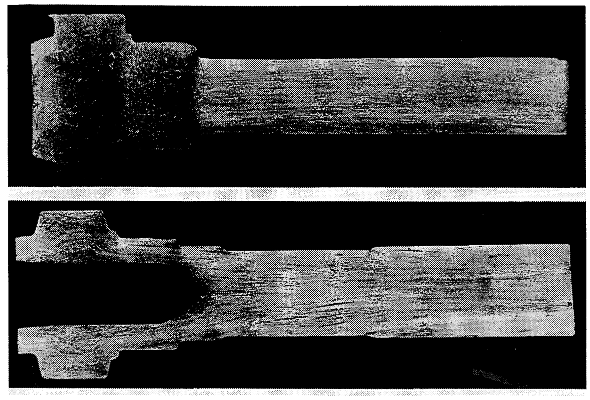


Fig. 3-34—Forging cross section showing bad unfilled condition.

matter may be included. The forging should be produced in such a manner that the material (grain) flow is at right angles to the direction of the stress the teeth will have to resist in service.

Uniform grain flow is also extremely important because it helps to minimize distortion during heat treatment. Uniform grain flow may be obtained by hammer forging, or upsetting. Where strength and minimum distortion are not essential, other forging methods may be used.

When the section of a gear forging is very large, particularly when it is made of deep hardening steel, such as 4340, it is essential that the forging be cooled slowly to avoid internal bursts. In general, all forgings should be normalized and annealed. This will refine the coarse structure resulting from forging temperatures, minimize distortion during subsequent heat treating operations and improve machinability.

Seamless Tubing

Seamless steel tubing is an economical hot-forged shape from which annular parts such as automatic transmission ring gears, Fig. 3-35, can be produced with minimum machining requirements. Tubing of this type is available in a range of sizes from ¾-in. diameter tubes with a 0.065-in. wall thickness, up to 11-in. diameter tubing with a 3¼-in. wall thickness.

Seamless tubing is made by heating solid round billets, piercing them into hollow shells, elongating the shells and rounding the tube.

The piercing operation, Fig. 3-36 is basically a forging operation in which the metal is worked from the inside as well as the outside. The large amount of metal displacement as the tubing is being pierced while spinning at high speed gives a fine grain structure and a uniform grain flow. Simultaneously the steel fibers are given a slight spiral twist that results in added strength and

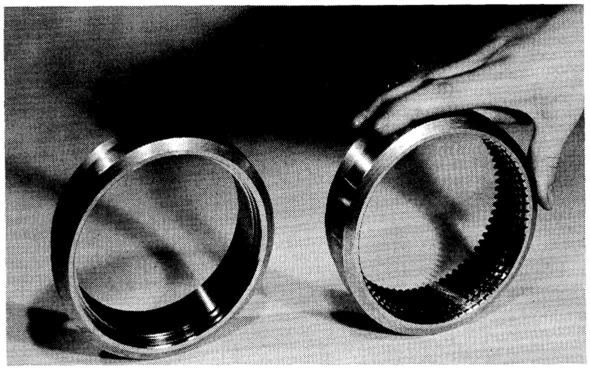


Fig. 3-35—A typical 4028 steel transmission ring gear made from seamless tubing. *Courtesy Ford Motor Co.*

ductility. During piercing, the tube rotates at 700-rpm and advances at a rate of one ft. per minute.

The seamless steel tubes are usually heat-treated and annealed to meet desired machinability requirements. Improved surface finish on seamless steel tubing can be achieved by cold drawing, roll-burnishing or rough-turning by the steel mill source.

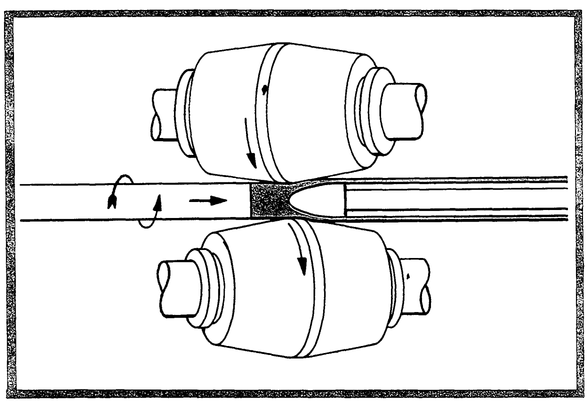


Fig. 3-36—Piercing a steel billet to produce seamless forged tubing. *Courtesy Timken Roller Bearing Co.*

Cold/Warm Forming

A relatively new development in the production of gear blanks is the cold forming method in which cold forming machines produce precision formed and pierced blanks from coil stock, bar stock or sheared blanks in an automated, high-production, multiple-station, die-forming process.

These gear blanks can be formed to close tolerances, Fig. 3-37, and effect considerable savings in material and machining costs. In one case, machining of a high carbon alloy steel planet

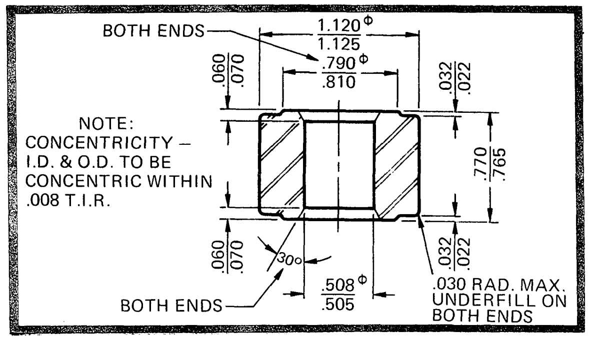


Fig. 3-37—Typical tolerances of a cold-formed 5130-H planet pinion produced on a six-die cold former at a rate of 55 pieces per minute.

pinion wasted over one-third of the material compared with less than nine percent by forming.

When cold forming is used exclusively to form a pinion blank, the finish-formed blank becomes extremely hot. An annealing operation usually follows the forming process.

The process may require intermediate heating (warming) of the blank, if the material carbon content with parts requiring backward extrusion exceeds 0.35-percent; or the blank size requires it. Stress-relieving operations usually follow warm-forming operations.

Planet pinion gear blanks are produced by a cold/warm forming process at a rate of 70 gear blanks per minute on a special five-die forming machine in the line shown in Fig. 3-38. This method combines cold and warm-forming in one machine.

The process starts with coiled hot-rolled 5140-H steel drawn to size just before entering the forming machine. In the forming machine, a blank is cut off, squared in the first die and upset in the second. After the second die, the upset blank drops into a chute leading to an elevator and hopper beside the forming machine. From the hopper and feeder, the blanks are pushed one after another through induction heating coils.

At the exit of the coils, an electric eye examines each blank and passes only those heated to specification. Blanks are automatically sorted three

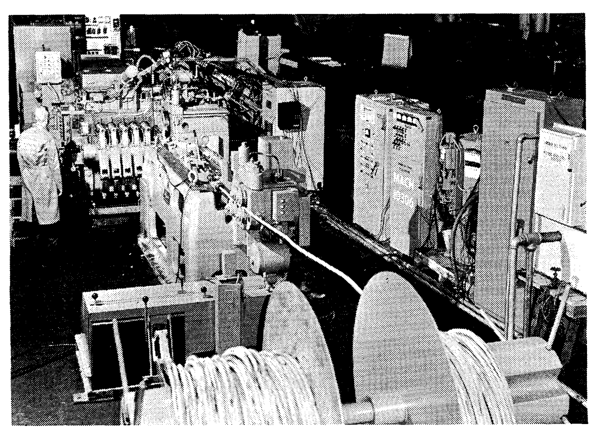


Fig. 3-38—A cold/warm forming line that produces 5140H planet pinion blanks from coiled bar stock. *Courtesy National Machinery Co.*

ways: Acceptable, too hot or too cold. Blanks with acceptable heat roll down tracks to a refeeding mechanism on the transfer of the forming machine. The gear blanks, Fig. 3-39, are formed warm in the last three dies.

The hole produced in cold or warm-formed blanks is usually broached, reamed or bored in subsequent machining operations.

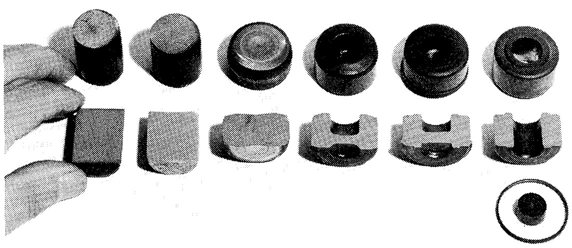


Fig. 3-39—Sequence for producing cold/warm formed 5140H planet pinions showing parts in half section. *Courtesy Hydramatic Div., GMC.*

Manufacturing Process

Once the mechanical properties and method of forming the blank have been decided on, the final step is to consider the manufacturing operations which might affect the selection, chemical composition, and heat treatment of the gear material.

Machining of Blank

In turning and boring gear blanks, the power, speed, feeds, and surface finish depend on the hardness and metallographic structure of the material. These also determine the amount of burring that will occur.

Machinability ratings for steels have been established using 1212 cold-drawn steel as a basis with a rating of 100-percent when turned at 165-fpm under normal cutting conditions Table 3-17. All other steels are rated either above or below this level, with most of them below (machinability less than 100 percent). It must be emphasized that these ratings are for turning only and are not necessarily the same for other types of machining operations.

The various machining processes differ in metal removal characteristics. Therefore, machinability ratings based on operations where chip flow is unrestricted cannot be applied where chip flow is restricted such as shaper-cutting or broaching.

If the gear teeth are to be finished before heat treatment, it is sometimes possible to specify either structure or chemical composition of the material for best machinability. If the gear blank can be roughed and heat-treated before the teeth are cut, it may be possible to reduce sections and therefore use a leaner alloy. Since machining operations are largely determined by the particular situation, it is difficult to generalize.

Tooth Forming

High-impact tooth-forming machining processes such as broaching, shaping and hobbing have machinability characteristics that are much more related to grain structure than to material hardness.

If the gear teeth are to be heat-treated after cutting, it is easier to obtain a more suitable structure for machining.

Many studies of broached internal gears and splines where material tearing has been encountered have revealed that the cause for tearing is the typical quenched and tempered sorbitic structure, Fig. 3-40.

When the material is normalized to form a blocky lamillar pearlitic structure, Fig. 3-41, smooth finishes can be achieved at normal broaching speeds.

Another type of structure called Widmanstatten gives tearing problems with broaching. This structure results from overheating of the material during the forging process. Needle-like ferrite particles identify the Widmanstatten structure, Fig. 3-42.

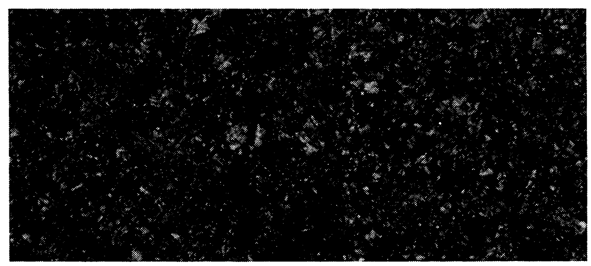


Fig. 3-40—A 100X magnification of a 5120 steel sample of 25Rc hardness showing typical quenched and tempered sorbitic structure that gives a poor machined finish.

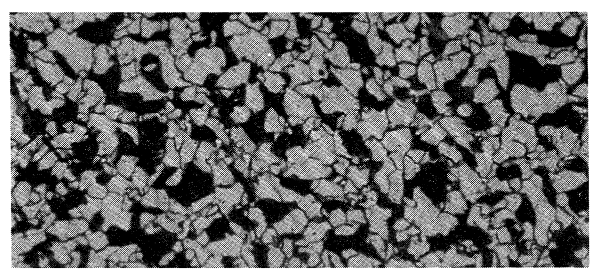


Fig. 3-41—A 100X magnification of a good normalized steel structure of zero Rc hardness and 5 to 6-grain size that machines to a good finish.

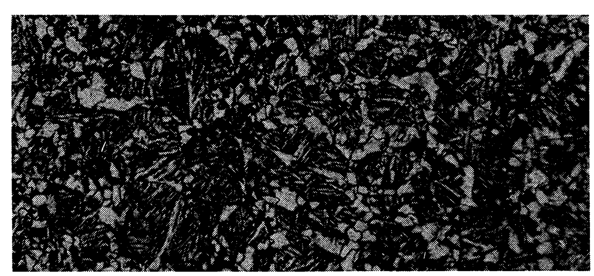


Fig. 3-42—A 65X magnification of a steel sample of 11-12 Rc hardness that shows Widmanstatten structure that gives tearing problems with broaching.

Table 3-17—Machinability

Speeds listed are approximate and are to be used only as a basis for calculating proper speeds for the part in hand. The figures represent the averages for the general run of parts made from cold finished steel. Any extraordinary features of the part to be made should be taken into consideration and speeds altered accordingly. (Grades not listed may be interpolated from the speed and feed values of the listed grades most similar in chemistry.)

A.I.S.I. GRADE	Surface Feet Per Minute	% Relative Speed Based on 1212 As 100%
CARBON STEELS		
1008	110	66
1010 (light feeds)	120	—
1011 (light feeds)	120	—
1012 (light feeds)	120	—
1013 (light feeds)	120	—
1015	120	72
1016	130	78
1017	120	72
1018	130	78
1019	130	78
1020	120	72
1021	130	78
1022	130	78
1023	125	76
1024	110	66
1025	120	72
1026	130	78
1027	110	66
1029	115	70
1030	115	70
1031	115	70
1033	115	70
1034	115	70
1035	115	70
1036	105	64
1037	115	70
1038	105	64
1039	105	64
1040	105	64
1041	95	57
1042	105	64
1043	95	57
1044	95	57
1045	95	57
1045 annealed	120	72
1046	95	57
1048	90	54
1049	90	54
1050	90	54
1050 annealed	115	70
1051	90	54
1052	80	49
1053	90	54
1054	90	54
1055 annealed	85	51
1059 annealed	85	51
1060 annealed	85	51
1061 annealed	85	51
CARBON STEELS—Cont'd.		
1064 annealed	80	49
1065 annealed	80	49
1066 annealed	80	49
1069 annealed	80	49
1070 annealed	80	49
1071 annealed	80	49
1072 annealed	80	49
1074 annealed	75	45
1075 annealed	75	45
1078 annealed	75	45
1080 annealed	70	42
1084 annealed	70	42
1085 annealed	70	42
1086 annealed	70	42
1090 annealed	70	42
1095 annealed	70	42
CARBON STEELS, LEADED		
10L18	150	92
10L45	109	66
10L45 annealed	138	84
10L50	99	60
10L50 annealed	131	78
RESULPHURIZED STEELS		
1108	135	81
1109	135	81
1110	135	81
1113	165	100
1115	135	81
1116	155	94
1117	150	91
1118	150	91
1119	165	100
1120	135	81
1125	135	81
1126	135	81
1132	125	76
1137	120	72
1138	125	76
1139	125	76
1140	120	72
1141	115	70
1141 annealed	135	81
1144	125	76
RESULPHURIZED STEELS,—Cont'd.		
1144 annealed	140	85
1145	110	66
1145 annealed	130	78
1146	115	70
1151	115	70
1151 annealed	135	81
1211	155	94
1212	165	100
1213	225	136
1215	225	136
RESULPHURIZED STEELS, LEADED		
11L17	172	104
11L37	138	84
11L41	131	79
11L41 annealed	155	94
11L44	144	87
11L44 annealed	161	98
12L13	260	158
12L14	260	158
12L15	260	158
ALLOY STEELS		
1330 annealed	100	60
1335 annealed	100	60
1340 annealed	95	57
1345 annealed	95	57
4012	130	78
4023	130	78
4024	130	78
4027	110	66
4028	120	72
4037 annealed	120	72
4047 annealed	110	66
4118	130	78
4130 annealed	120	72
4137 annealed	115	70
4140 annealed	110	66
4142 annealed	110	66
4145 annealed	105	64
4147 annealed	105	64
4150 annealed	100	60
4161 annealed	100	60
4320 annealed	100	60
ALLOY STEELS—Cont,d.		
4340 annealed	95	57
4419	130	78
4615	110	66
4620	110	66
4621	110	66
4626	100	60
4718	100	60
4720	100	60
4815 annealed	85	51
4817 annealed	80	49
4820 annealed	80	49
5015	130	78
5060 annealed	100	60
5120	125	76
5130	95	57
5132 annealed	120	72
5135 annealed	120	72
5140 annealed	115	70
5145 annealed	110	66
5147 annealed	110	66
5150 annealed	105	64
5155 annealed	100	60
5160 annealed	100	60
6118	110	66
6150 annealed	100	60
8615	115	70
8617	110	66
8620	110	66
8622	110	66
8625	105	64
8627	105	64
8630 annealed	120	72
8637 annealed	115	70
8640 annealed	110	66
8642 annealed	110	66
8645 annealed	105	64
8655 annealed	95	57
8720	110	66
8740 annealed	110	66
8822	105	64
9255 annealed	90	54
9260 annealed	85	51
ALLOY STEELS, LEADED		
41L40 annealed	127	73
41L50 annealed	115	70
86L20	127	77

Courtesy Republic Steel Corp.

Tooth Finishing

Teeth can be finished by shaving, roll finishing, grinding, honing, or lapping. Table 3-18 shows approximate machinability ratings for shaving.

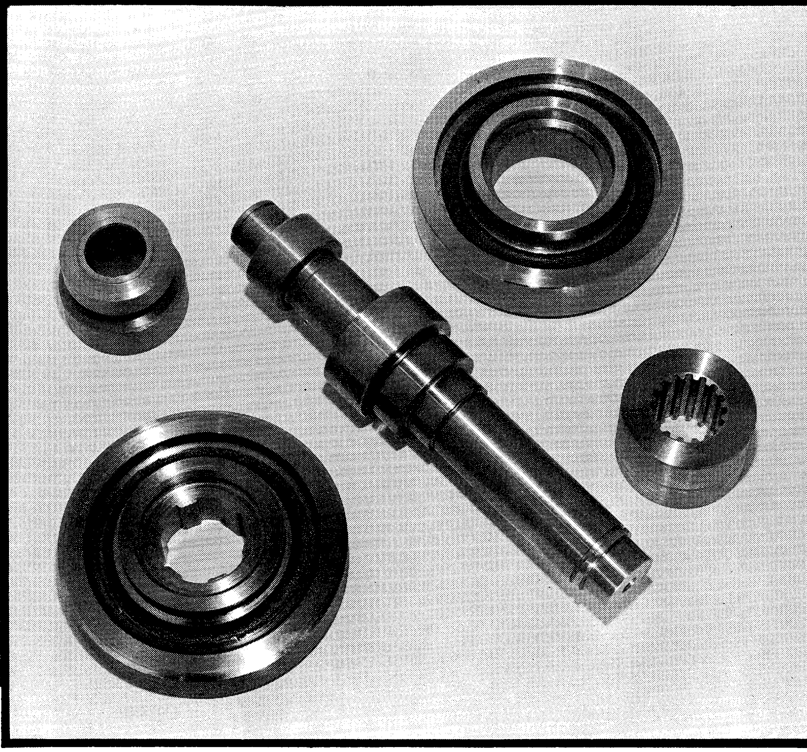
If the gears are to be finished by grinding, distortion should be minimized to reduce the amount of grinding necessary. This suggests the use of press quenching or, in certain instances, a special hot-quenching operation such as martempering.

Flame hardening can develop a rough surface, particularly if the heating torches are not carefully adjusted. Flame-hardened gears frequently require honing, grinding, or lapping to prevent accelerated wear. This is particularly true if the flame-hardened gear is meshed with a softer mate.

Table 3-18—Approximate Alloy Steel Machinability Ratings For Shaving, Percent.

Shaving is rarely recommended when machinability is less than 30-percent and never when below 25-percent.

Type of Steel	Rockwell C Hardness					
	30	31-35	36-40	41-45	46-48	48-50
Hy-Ten B-3-X	60	55	50	40	25	10
4640	45	40	35	25	15	None
8645	40	35	30	20	10	None
4145	40	35	30	20	10	None
3140	37	30	25	15	5	None
4340	35	30	25	15	5	None
5140	35	30	25	15	5	None
3250	30	25	20	10	None	None
2345	30	25	20	10	None	None
3440	30	25	20	10	None	None
6145	30	25	20	10	None	None



Chapter FOUR

Courtesy Clark Equipment Co.

Machining the Blank

The manufacture of quality gears begins with a good blank. Normally gear blanks are of four types: 1. Shaft-type with centers, 2. Flat-type with round or keyed holes, 3. Flat-type with splined holes and 4. Shoulder gears with round, keyed or splined holes.

Selection of Locating Surfaces

The first consideration in manufacturing a gear is to select the locating surfaces and use them throughout the processing sequence. Close relationship between the locating surface and the face of the gear itself must be held. Otherwise, when the teeth are cut and finished with tooling that necessarily contacts the gear faces, the teeth will be in improper relationship with the locating or related surface on which the gear operates. Gears that locate on round diameters, or spline teeth must fit the work arbors closely or these critical hole-to-face relationships will be destroyed.

Blank Tolerances

Often it is good manufacturing practice, particularly with shoulder gears, to provide a finish-turning and facing operation on the gear blank, Fig. 4-1 before the gear teeth are cut. This practice can avoid critical gear runout problems.

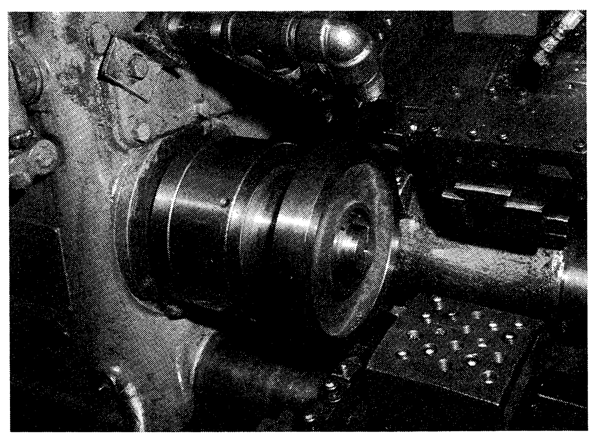


Fig. 4-1—Finish facing and turning operation performed in the green blank before tooth forming and finishing. *Courtesy Clark Equipment Co.*

Typical manufacturing tolerances for gear blanks prior to cutting of the teeth are shown in Table 4-1.

Shaft gears with centers are the simplest to locate. The center should be a protective type recessed design. The center should be conditioned by lapping after heat treatment.

Table 4-1—Typical Gear Blank Tolerances*

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

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

Flat-type gears with round or keyed holes should have the holes bored or green-ground to a close dimension to provide a good locating surface. Expanding arbors, Fig. 4-2, effectively hold gear blanks for finish-turning, gear cutting, gear finishing and gear inspection operations.

Spline Broaching

Gears with splined holes require more careful consideration. First, it should be determined what portion of the spline fits its mating member. Most involute spline fits are on the sides (pitch diameter) of the teeth, but others may locate on the major or minor diameter of the spline teeth.

The easiest way to process a gear with a splined hole is to locate the part with round arbors that contact the minor diameter of the spline teeth. If the spline in the gear blank is a side-bearing fit with its mating part, this means that the spline inside diameter must be broached concentric with the sides of the teeth.

Or if it is a major diameter fit, the spline minor diameter must be broached concentric with the major diameter and the sides of the teeth.

Normally nibbling-type spline broaches with alternate round and spline finishing sections are used to provide the required minor diameter concentricity. However, there are several reasons why such tools tend to drift off-center and do not always provide the required concentricity conditions. These include: back taper in the broach, relief on the sides of the teeth and breakdown of the cutting corners (which may number as many as 14,480), machine misalignment and improper face grinding techniques.

Concentricity Broaches

A unique design called the concentricity broach, Chapter 10, can correct these problems and assure absolute minor diameter, pitch diameter and major diameter concentricity. This broach, Fig. 4-3, has a floating full-form finishing shell on the end of a conventional nibbling-type broach section. The finishing shell has alternate round and full-form finishing teeth that produce accurate, concentric spline teeth with excellent surface finish.

Simultaneous cutting action on the critical surfaces by the full-form finishing shell insures that

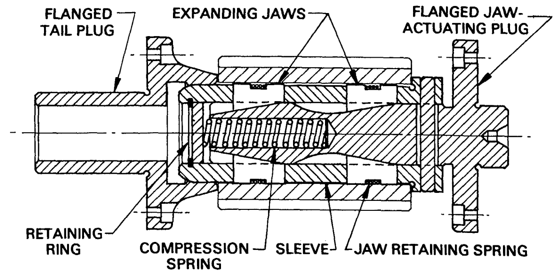


Fig. 4-2—A typical solid expanding arbor that can effectively hold round or splined-hole parts for machining.

the concentricity of the broached spline tooth dimensions will be equal to the accuracy ground in the tool itself. Perfect balance conditions in the part are also assured by the concentricity broach.

Avoiding Nicks and Burrs

Once the locating surfaces have been selected and conditioned, and the operation processing sequences have been specified to assure that gear faces are flat, parallel and square with the locating surface; only one problem remains in producing a good gear blank. That problem concerns the elimination of burrs produced by machining or improper handling.

If burrs or nicks remain on parts into the tooth forming and finishing operations, all of the care put into part conditioning can be lost when a burr causes improper part location. It is relatively simple to remove corner burrs by chamfering tools in the green blank finishing operation, Fig. 4-1.

The burrs caused by improper blank handling between process operations are the cause of most tooth runout problems. Gears should be handled and transported in such a way that they do not touch each other between operations.

An easy way to minimize the nicking of the tips of gear teeth is to specify semi-topping hobs as described in Chapter Five.

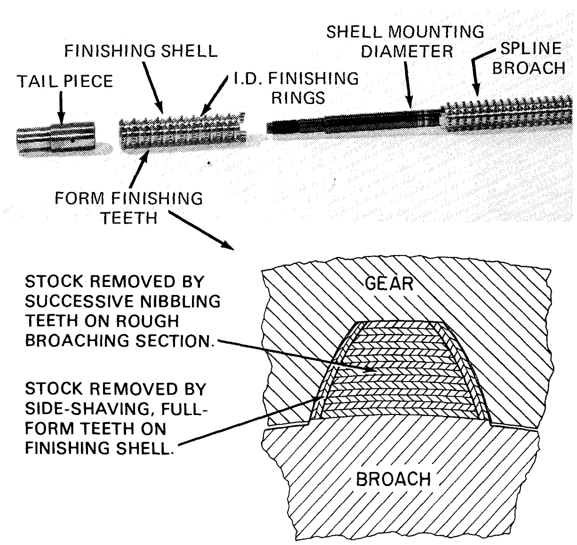
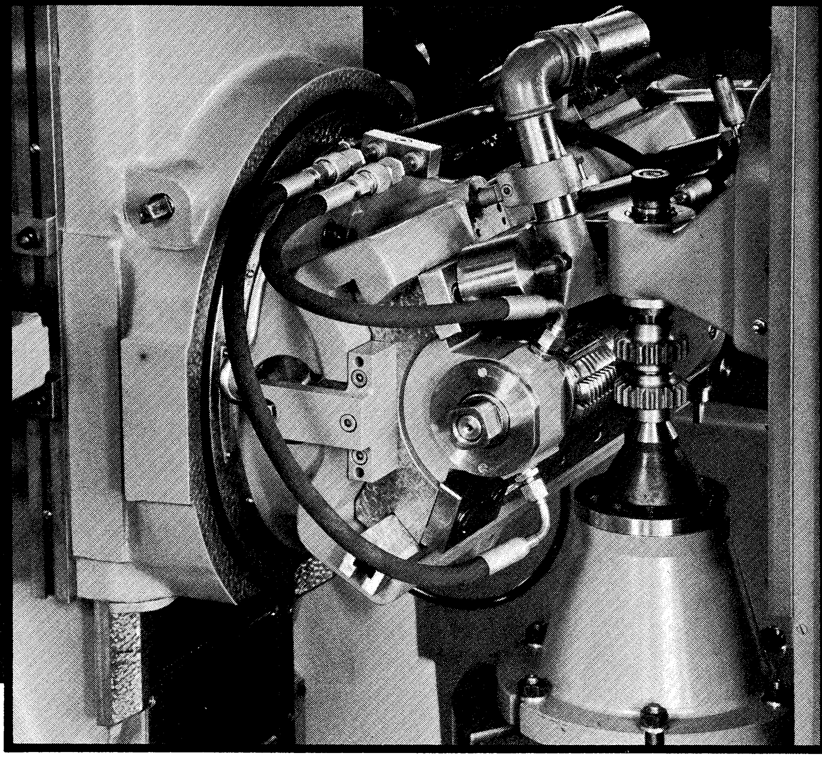


Fig. 4-3—Typical concentricity broach with a section drawing showing how the full-form finishing teeth produce smooth, accurate spline teeth.



Chapter FIVE

Hobbing an external helical gear on a vertical gear hobber. Cutting action is produced by feeding the angular, gashed, worm-shaped hob through the gear blank in proper timed relationship. *Courtesy Liebherr Machine Tool Div.*

Forming the Teeth

The forming of gear teeth has traditionally been a time-consuming heavy stock removal operation in which close tooth size, shape, runout and spacing accuracy are required. This is true whether the teeth are finished by a second forming operation or a shaving operation.

Originally gear teeth were produced with form-milling cutters on milling machines equipped with index heads. Later the popular gear hobbing process, photo above, was developed to produce external gears. The shaper-cutting method, Fig. 5-1, was developed primarily to produce internal gears and gears on blanks that would not permit passage of a hobbing tool.

Today internal gears are being broached at high production rates. External gears are also being produced at high production rates by pot broaching methods. Other methods such as high energy rate forming, and rolling of fine-pitch teeth from the solid are being applied and investigated.

Gear Hobbing and Shaping

One of the key problems in hobbing and shaping of gear teeth is the specification of a properly proportioned tooth form. Most of the problem occurs in the fillet area. However, when semi-topping hobs or shaper cutters are used to produce tip-protective chamfers, Fig. 5-2, a loss of active profile can result if the outside diameter of

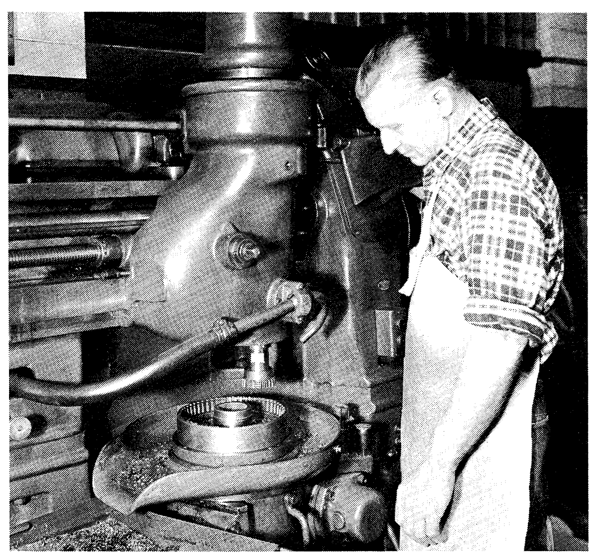


Fig. 5-1—Producing an internal spur gear with a gear shaper. Cutting action is achieved by reciprocating and infeeding the gear-like shaper cutter as the gear blank is rotated in proper timed relationship with the cutter.

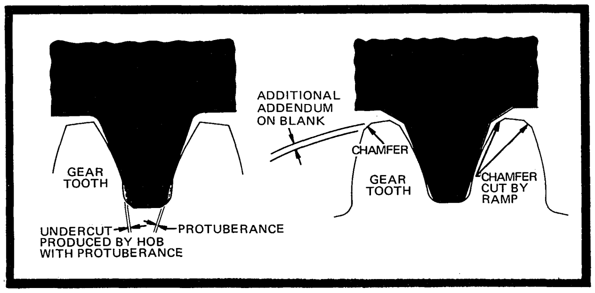


Fig. 5-2—Typical hob tooth shapes. Protuberance type is at left, semi-topping type at right. *Courtesy Star Cutter Co.*

the blank has not been increased beyond the theoretical outside diameter to provide additional stock for the chamfer.

If the fillet produced by hobbing or shaping is too high, finishing tool interference and breakage can result; and the accuracy of the produced profile can be affected. If the hob or shaper cutter tooth has a full radius form on the tip, maximum wear life of the tools is provided.

Referring to Fig. 5-3, it can be seen that forming of the teeth with a gear-shaped shaper cutter or a rack-shaped hobbing tool differs considerably from the in-fed form tool operation. Hobbing, gear-shaping, and rotary gear shaving have tooth tip paths which produce fillets that are actually trochoidal curves generated by the tip corner of each tooth.

As a result, the point of tangency between this curve and the generated involute profiles is higher than that of the radius on the form tool. The shaper-cut fillet tangency point is slightly higher than that produced by a hob of the same working depth. Thus, the shape of a fillet on a gear drawing is correctly specified as that produced by a specific hob or shaper cutter tooth

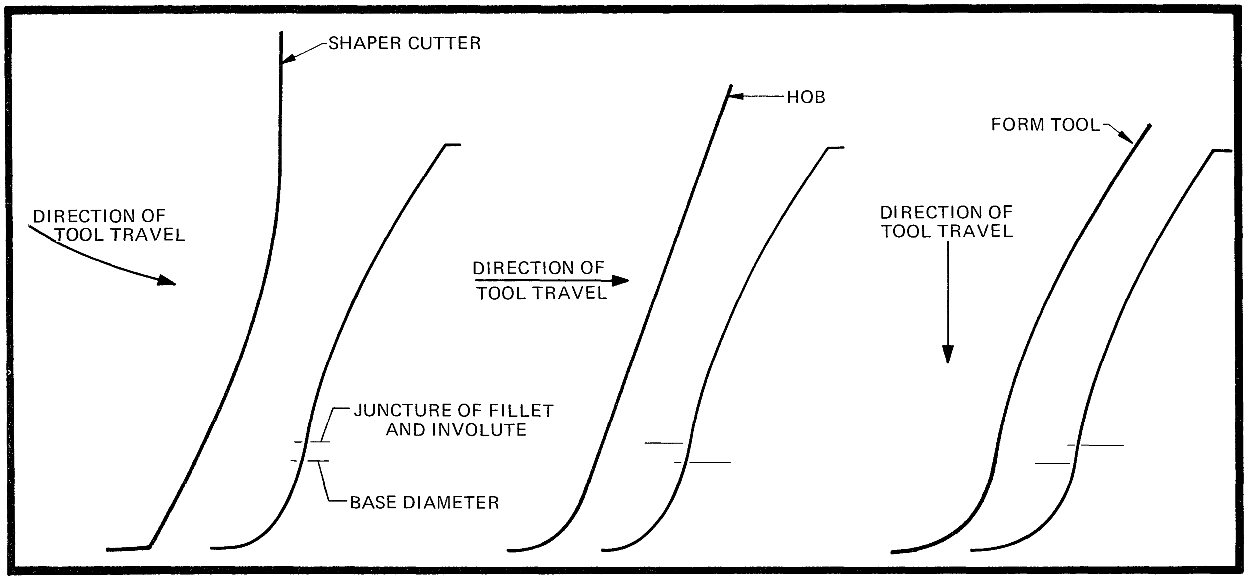


Fig. 5-3—Generating action of gear shaping, left, and hobbing, center; compared with index form-milling, right.

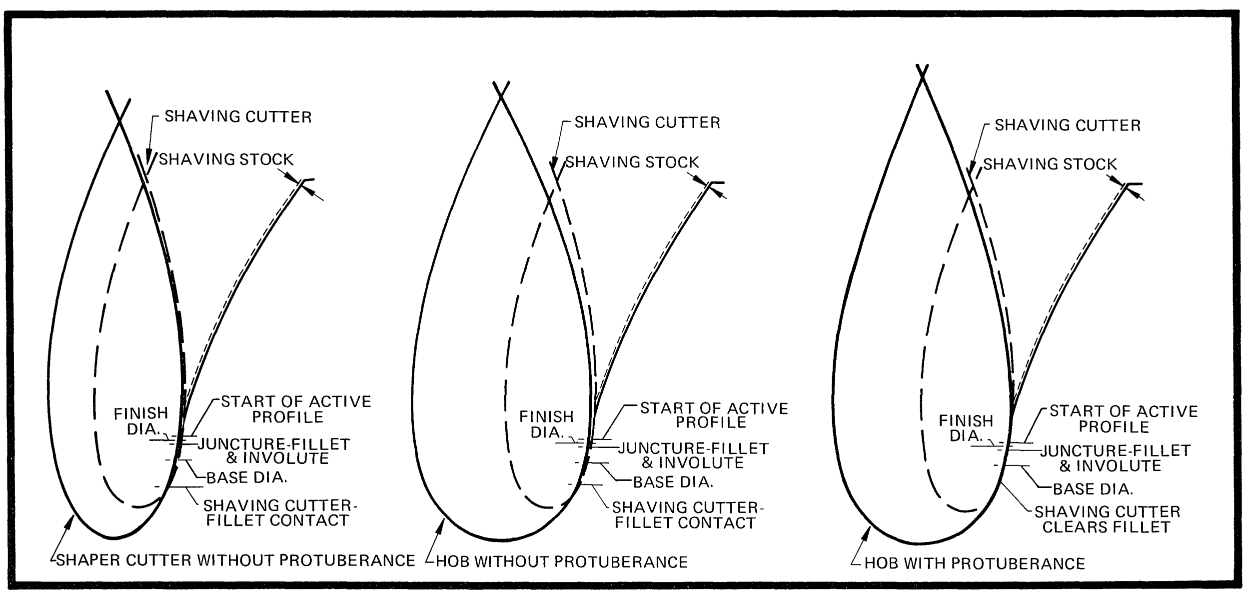


Fig. 5-4—Generating paths of shaper cutter, hobs and rotary-shaving cutters in fillet area on same tooth.

form with a specific tip radius or form.

The generating action of hobs and shaper cutters with and without protuberance to provide necessary shaving cutter tip clearance, Chapter 1, is illustrated in Fig. 5-4.

The amount of total undercut (Shaving stock plus 0.0005 to 0.001-in.) produced by pre-shaving, protuberance-type tools, Table 5-1, varies with the pitch of the gear teeth. Position of the protuberance-produced under-cut fillet produced by a specific hob or shaper cutter varies with the number of teeth in the gear. Usually the undercut will generate too high on gears with small numbers of teeth and reduce the necessary amount of involute profile. Use of the same tool on gears with large numbers of teeth will provide an undercut too low to serve any useful purpose.

Table 5-1—Recommended Shaving Stock and Total Undercut for Pre-Shave Gear Cutting Tools

Normal Diametral Pitch	Shaving Stock (In. per Side of Tooth)	Total Undercut (In. per Side of Tooth)
2 to 4	0.0015 to 0.0020	0.0025 to 0.0030
5 to 6	0.0012 to 0.0018	0.0023 to 0.0028
7 to 10	0.0010 to 0.0015	0.0015 to 0.0020
11 to 14	0.0008 to 0.0013	0.0012 to 0.0017
16 to 18	0.0005 to 0.0010	—
20 to 48	0.0003 to 0.0008	—
52 to 72	0.0001 to 0.0003	—

Theoretically protuberance - type hobs and shaper cutters should be designed for a gear with a specific number of teeth. However, this method is not economically feasible when a variety of gears with different tooth numbers are being processed. Often a tool with no protuberance may be used for gears with small numbers of teeth. This method makes use of the natural undercut produced by generating-type tools that extend below the base circle on gears with small tooth numbers. Figure 5-5, left, illustrates this condition.

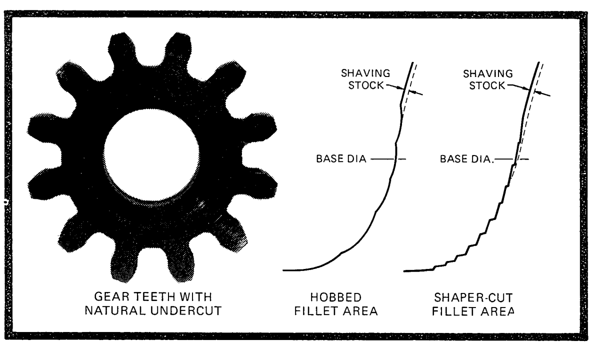


Fig. 5-5—A 12-tooth pinion, left, showing natural shaper-cutter undercut. Enlarged fillets, right, show type of finish generated by hobbing and shaping.

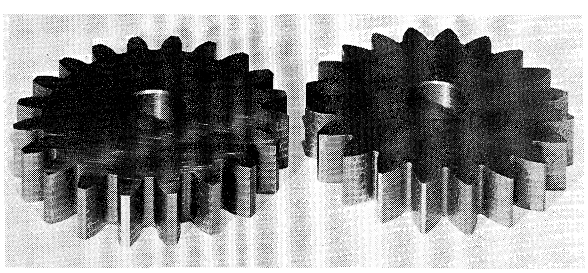


Fig. 5-6—Hobbed 4-DP, 5-in. PD, 20-tooth gears with 20° PA, left, and 30°-PA, right.

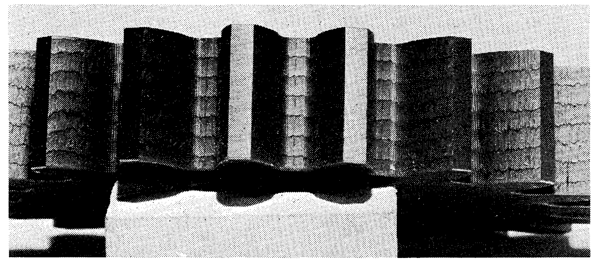


Fig. 5-7—Hobbed finish of left-hand gear in Fig. 5-6 as produced by a 4-in. dia., 10-flute, single-thread hob rotating at 71-rpm and fed at 0.150-in. per revolution.

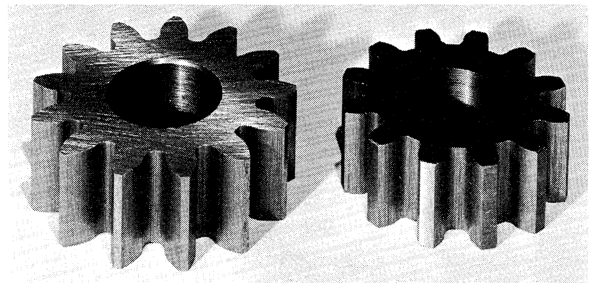


Fig. 5-8—Gear-Shaped 5-DP, 20°-PA, 13-tooth gear, left, produced by a 20-tooth, 4½-in. OD shaper cutter; compared with a 5/7-DP, 20°-PA, 12-tooth gear, right, produced by a 15-tooth, 3.325-in. OD cutter.

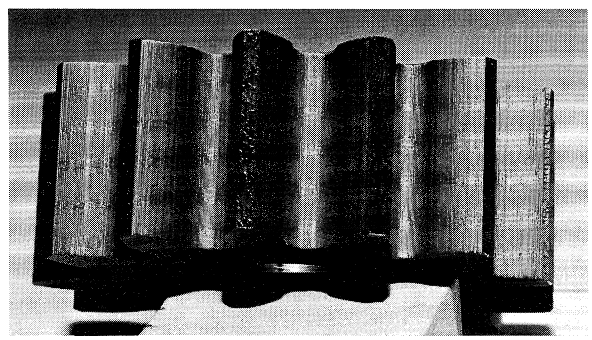


Fig. 5-9—Shaper-cut finish of left-hand gear in Fig. 5-8 as produced with the cutter making 121 strokes per minute and feeding at a rate of 0.001-in. per stroke.

On long and short-addendum gears, the amount and position of protuberance on hobs and shaper cutters must be carefully specified because of the different generating action in producing the teeth.

The fillet shapes of typical hobbed and shaped gears are shown in Figs. 5-6 and 5-8. The finish produced by these two generating forming methods is shown in Figs. 5-7 and 5-9.

The effect of the generating action of hobs and shaper cutters on the finish in the fillet area is shown in the two enlarged sketches in Fig. 5-5.

Applying the Processes

Careful consideration should be given to the of the tooling for hobbers and shapers. Where possible this tooling should locate on the rim or side of the gear blank, Fig. 5-1, just below the root diameter of the teeth. Proper mounting of hobs, including indication for runout within 0.0005-in., and careful machine setup for tooth size are most important for good results in the subsequent shaving operation.

Optimum machine performance and economy results when only sufficient stock is left for shaving to clean up the gear and assure the removal of semi-finishing errors or their reduction to specified tolerance limits. Leaving an excessive amount of stock to be removed by shaving unduly reduces cutter life, increases shaving time and may result in the shaving cutter hitting the fillet.

Table 5-1 shows the amount of stock left on each side of a tooth under average conditions for removal in the shaving operation.

It is also important that the involute profile and lead of a hobbed or shaped helical gear be held as close as possible to that of the gear as shaved if maximum shaving cutter life is to be attained. Uniform stock removal in the shaving operation equalizes cutter wear and results in more pieces shaved before the cutter has to be reground. This is not the case when the cutter has to correct too great an error in involute profile and excessive wear is concentrated on only part of the tool. This results in hollow spots on the cutter which in turn leave high spots on the shaved gear tooth profiles.

When hobbing helical gears that will subsequently be heat-treated, it is a simple matter to allow for heat-treat distortion as explained in Chapters 1 and 3.

It is good practice to process a pilot group of gears to the desired lead, heat treat them and then carefully check the amount of distortion caused by the heat treatment. The resulting average of this check will serve as a guide for compensating the lead in processing the remainder of the lot.

Changes in helix angle also produce changes in involute profiles. Thus, both must be adjusted in machining gears which are to be heat-treated. The gears should be hobbed or shaper cut as closely as possible to the adjusted lead. This is particularly true if maximum shaving cutter life is desired in producing wide face gears.

Clutch gears having rounded or pointed teeth should have all chips and burrs removed from their ends before they are shaved. Otherwise, these chips can become imbedded in the serrations of the cutter teeth and cause breakage.

Blank machining, hobbing or shaping speeds and feeds should not be so excessive that they cause cold working or burnishing of gear tooth surfaces. This practice will prolong shaving cutter life and avoid excessive heat treat distortion.

The selection of the type of hobbing tool has an important economic effect on the overall cost of gear processing. At one time, because they were used on finish-hobbing operations before the development of rotary gear shaving, only single-thread, Class "A", ground-form hobs were used as preshaving tools. With the advent of shaving, less-expensive single-thread, Class "B", ground-form pre-shaving hobs were successfully applied.

Today even lower-cost Class "C", accurate unground-form hobs are widely applied as preshaving tools. To reduce the required number of hobbing machines for roll-finished, fine-pitch helical transmission gears, multiple-thread, Class "C" accurate unground-form hobs are also being utilized as pre-shaving tools.

Multiple-thread hobs with straight gashes are usually larger in diameter than single-thread hobs because of the requirement for a low thread-angle. In actual production of 14 and 16-NDP helical transmission pinions, high production hobbing rates are being achieved by using 3-thread, 3-in. dia., Class "C" hobs instead of 2½-in. dia., single-thread hobs of the same class. The number of threads in multiple-thread hobs should not be prime with the number of teeth in the work gear.

Accurate unground-form, Class "C" single and multiple-thread hobs can be produced by rack form-tool methods to provide extremely close tolerances for such features as protuberance, semi-topping and full-radius fillet design.

Broaching Internal Gears

Internal spur and helical gears can be most economically produced in high production by a single pass of a full-form finishing broaching tool assembly, Fig. 5-10, and Chapters 4 and 10. A wide variety of automotive transmission internal running gears up to 6-in. pitch dia. with 6 to 20-DP teeth can be produced by this method, Fig. 5-11.

Full-form finish broaching provides fine surface finishes, precision involute form, accurate tooth thicknesses and precision tooth spacing and lead.

Internal helical gears are usually broached on

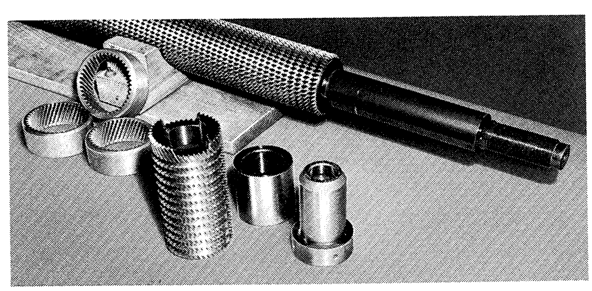


Fig. 5-10—Full-Form finishing broach showing roughing section, finishing shell, tailpiece and broached gears.

Fig. 5-11—Internal-broached spur and helical transmission pump and running gears ranging from 2 to 6-in. diameter.

vertical broaching machines. Accurate leads are produced by the action of a precision lead bar, follower nut, and associated gearing, Fig. 5-12, which rotate the broach as it is pulled through the blank.

Where close control of internal gear tip contact with mating pinions is desired, the broached tooth form can be notched as shown in Fig. 5-13 to provide absolute control of length of roll.

In one application, two fully-automated full-form finishing broaching machines produce the internal helical gear shown in Fig. 3-35, Chapter 3, at a rate of 180 pieces per hour. The internal gear has 72, 15.5-DP, 17½°-PA teeth with a 22° 11′, 30″ right hand helix angle. The gear blank has a 6-in. OD and is about 1-9/16-in. wide. Brinell hardness of the SAE 4028 blank is from 179 to 217. The broaching tool is 82-in. long and has a chip load of 0.0036-in. per tooth.

Originally the gear was shaped and shaved. It took 3-min. to shaper-cut the teeth and 1¼-min. to shave it. Each broaching machine makes a finished gear every 40 seconds. The former method required 28 gear shaper spindles and six rotary gear shavers. Total life of individual broaching tools is about 100,000 pieces.

Internal spur differential running gears with 5/7-DP teeth up to 9.400-in. pitch diameter have been produced by nibbling-type broaching tools, Fig. 5-14.

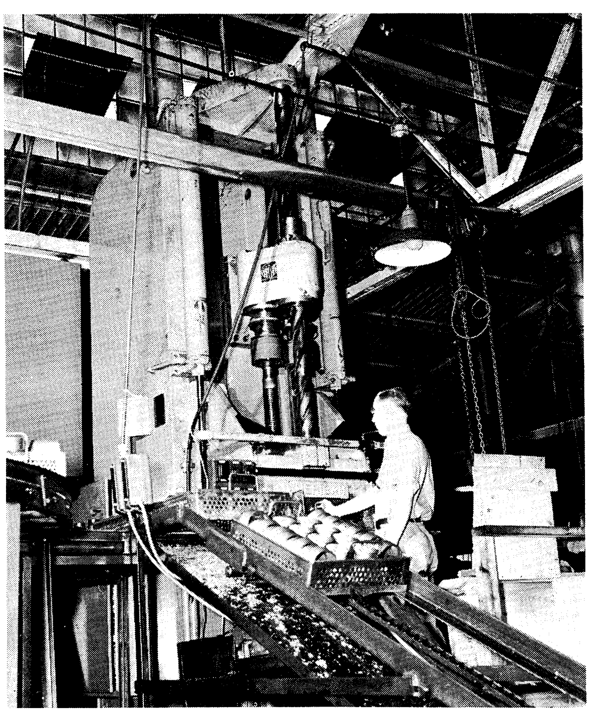


Fig. 5-12— Full-Form finish broaching of internal helical gears two-at-a-time on a vertical broach.

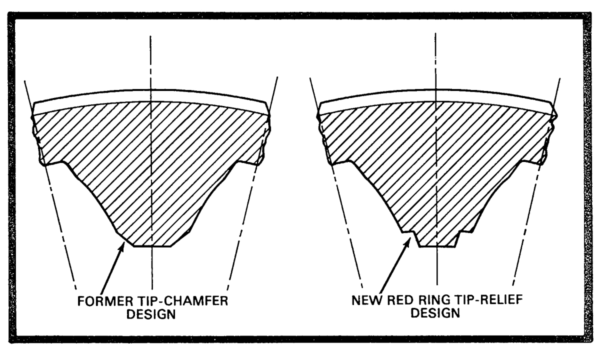


Fig. 5-13—Conventional broached internal gear tip chamfer and improved tip relief, length-of-roll-control design.

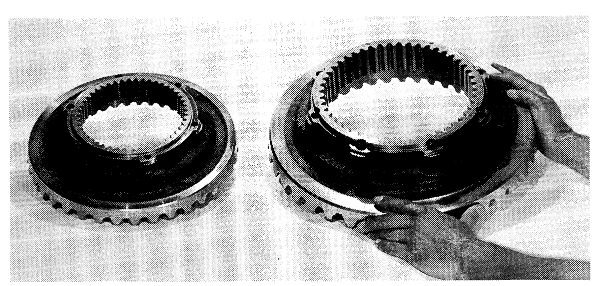


Fig. 5-14—Large internal spur differential running gears that are broached to precision tolerances.

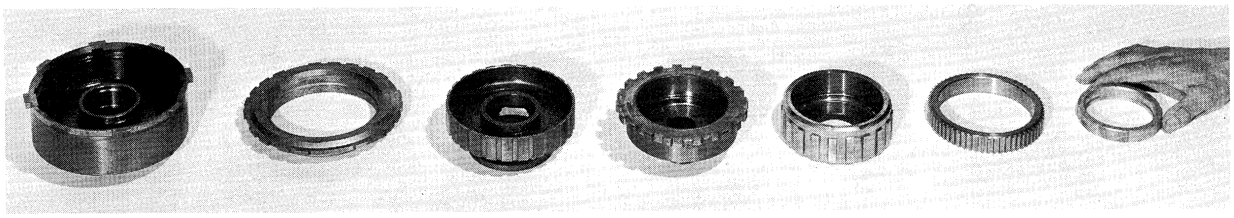


Fig. 5-15—A variety of external cast iron and steel clutches, cams and splines produce by push-up pot broaching.

Broaching External Gears

The fastest way to produce medium and high production external gears, splines and parts with specially formed teeth, Fig. 5-15, is by pot broaching, Chapter 10. A new process called push-up pot broaching uses a machine, Fig. 5-16, in which the part is pushed upward through a fixed pot broaching tool of either stick-type or wafer-type design to produce external teeth under ideal conditions that assure quick and complete chip removal from the broach teeth. Coolant is flushed into the tool area through a quick-disconnect coupling.

Fine finish and precision tooth form, size, and spacing are provided in gears and splines produced by push-up pot broaching.

The process is ideally adapted to full automation. Finished parts are ejected at the top of the pot broach where gravity force can help move them on to the next operation.

The 60-tooth, 12-DP, $14\frac{1}{2}$°-PA, SAE 5130 involute spline (second from the right in Fig. 5-15) has a 5-in. PD and is 0.800-in. long. The teeth are broached and the outside diameter finished with a ring-type broaching tool at a rate of 240 pieces per hour by pot broaching. Total life of the tool in this application is about 600,000 pieces.

External helical gears can also be produced by pot broaching, Fig. 5-17. The 4-in. OD, $\frac{3}{4}$-in. wide cast iron helical gear has eighty-seven, 24-DP, 22°-HA teeth.

The gears are produced on a special lead-bar-equipped vertical press by a solid HSS pot broaching tool in 15-sec. floor-to-floor time. Total tool life is 1,250,000 pieces.

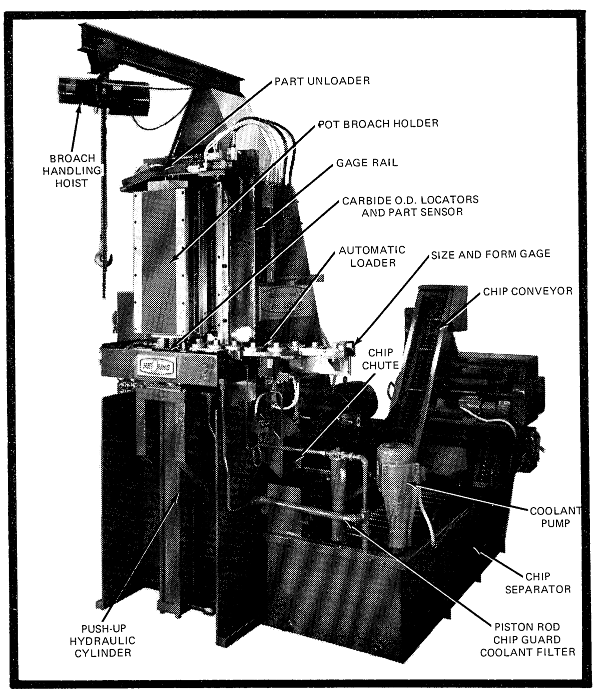


Fig. 5-16—A 25-ton automated push-up pot broaching machine that can produce external gears, splines and tooth forms at rates up to 450 pieces per hour.

Forming Teeth in Solid Blanks

Forming of fine-pitch gear teeth from the solid with gear rolling dies before roll-finishing is a process method that shows considerable promise. It is currently in the development stage.

High energy rate forging machines use high-pressure gas to drive a forming punch or die at speeds of up to 1,100-in. per second. Gears produced by this process are said to be 10 to 50-times stronger than those made by conventional forging and tooth-cutting methods.

To produce a blank with integrally-formed teeth, a raw billet is put in a blocker die to convert it into a preform. Then the preform is put into a finish die and is HERF-forged into a gear in a single blow. The gear is then trimmed to remove flash. Dies are 63Rc high-nickel, high-chrome, hardened steel.

Tooth grinding or rotary-shaving operations are performed after the forged blanks are machined. A typical HERF process makes thirty SAE 9310 gas turbine engine spur gears per hour. The gears have 10-DP, 25°-PA teeth on a $4\frac{1}{2}$-in. pitch diameter. HERF forging tolerances for the gears are plus or minus 0.005-in. with stock left for finish-shaving.

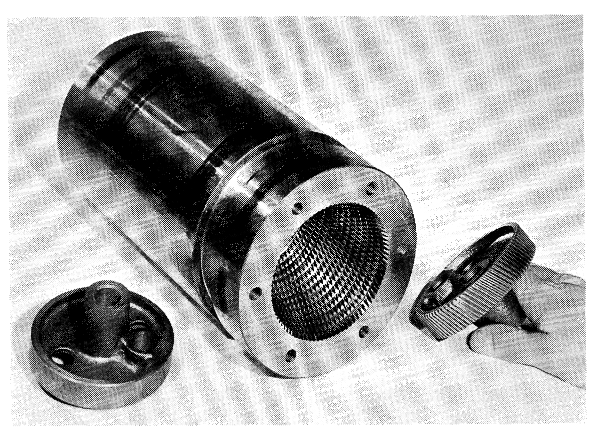
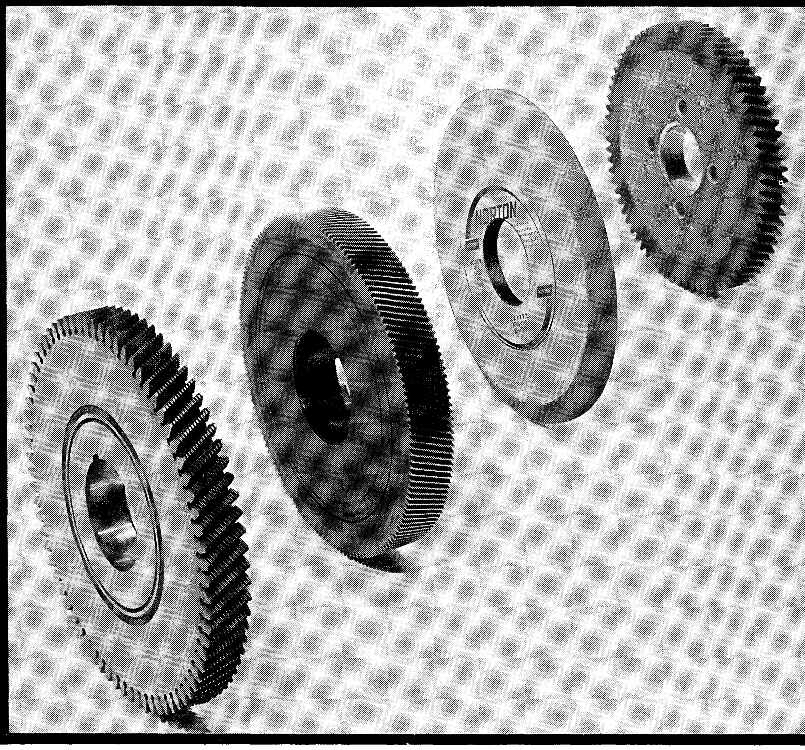


Fig. 5-17—A solid HSS pot broaching tool that produces external helical cast iron running gears.



Chapter SIX

Tools for finishing gears (left to right): A shaving cutter, a rolling die, a grinding wheel and a honing tool.

Finishing the Teeth

Four methods of finishing gear teeth are utilized following the generation of the teeth by conventional gear hobbing or gear shaping processes. These are: Shaving, roll-finishing, grinding and honing.

Although many gears are satisfactorily produced by hobbing or gear shaping without subsequent gear finishing operations, finishing operations become a necessity when high load-carrying capacities, high speeds, long wear life or quiet operation make surface finish and tooth accuracy of major design importance.

Tooth Modification

Modification of tooth form and shape by varying involute profile, lead or taper from theoretical curves has been found to have a marked effect on improving gear life and reducing operating sound levels by making allowances for deflection conditions under load. By far the most economical, and often the only way to achieve these important tooth modifications, is to make them by a finishing process.

The perfect tooth form for gears that run at high speeds, or are heavily loaded, is not likely to be the true involute trace. Some attempts have been made to calculate the proper amount and position of desired deviation from the true involute, but the majority of the modified forms have been developed by empirical methods.

The first reason for tooth modification is reduction of sound level due to normal gear operation. Two gears running together at high speed, even without loading, will make some noise. Normally, this noise level is less with helical gears than with spur teeth, since the overlap of the helix will minimize noise in helical gear trains.

In Fig. 6-1 (top), two teeth are in mesh. The next two teeth are about to engage. Contact that takes place at point *A*, is instantaneous. As a result there is an impact load, however minute it may be. Impact loads between two pieces of metal cause noise. Figure 6-1 (bottom), shows how the tooth form could be modified to ease the contact action.

The second reason for modification is reduction of noise level caused by heavy tooth loadings. Gear teeth bend when a load is applied. If the loading is high this adds to the operating noise level, by causing a tooth to enter mesh before its

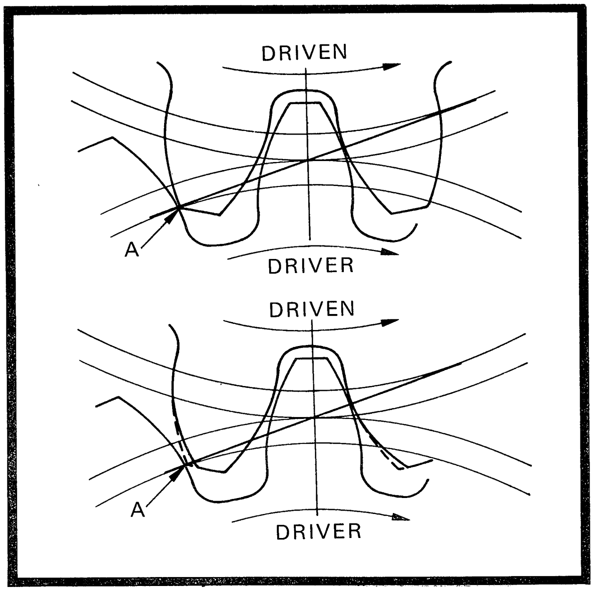


Fig. 6-1—Modification of tooth profile (bottom) to ease impact loading (top).

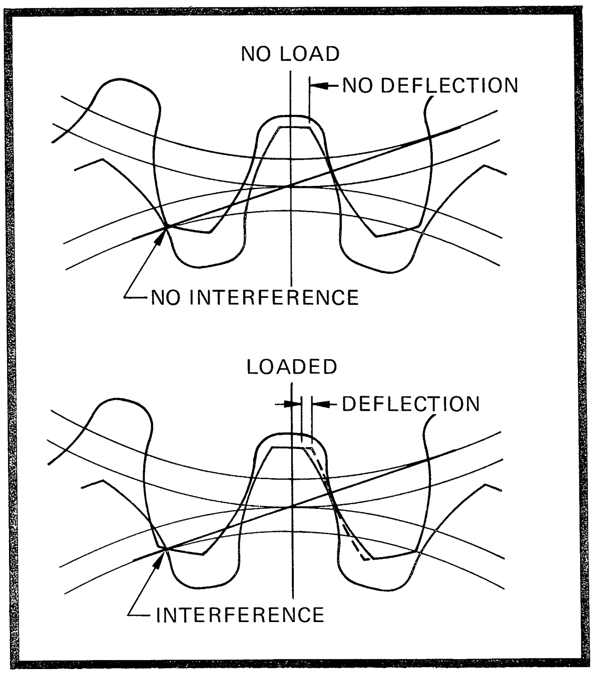


Fig. 6-2—Conditions of tooth interference without load (top) and with loading (bottom).

mate is in the proper location to receive the transmission of load. Figure 6-2, shows the same teeth in the same portion of their engagement, with and without loading. Notice that the bending of the load-bearing tooth shortens the distance between the two adjacent teeth along the line of contact. This means that the entering tooth will arrive too soon, and the impact of entry will be increased.

The attempt to mate two gears with differences in base pitch will result in cramping; and the cramping produces noise. The same slight modification which reduced noise in the no-load condition will not alleviate the cramping condition. The correct base pitch has to be restored by modifying the entering tooth in the same amount that the loaded tooth has deflected out of position.

The exact profiles of teeth which are used for quieting vary considerably if analyzed dimensionally, but all give the same ultimate result: A change in profile for easement of action. One or both gears of a pair may be modified. In a three-gear train such as a planetary set, only the pinion need be modified, since it mates with both the sun and ring gears. Figure 6-3 shows various approaches to profile modification.

Moderation in tooth modification is a necessity. If too little modification is bad from a noise standpoint, too much modification is infinitely worse. Enough true involute form must remain for continuous transfer of action from one pair of teeth to the succeeding pair; or the resulting loss of overlap will result in clatter, rough action, and impact destruction of the teeth. Helical gears are not as sensitive to this problem as spur gears.

The maintenance of the minimum required length of true involute profile can be solved by calculation. A typical involute profile modification is shown at the right in Fig. 6-3. This modification is in the magnitude of 0.0004-inch.

If more modification should be required, the flat area at the center of the profile would have to be longer, to insure continuous action. The amount and location of the modification will vary with application. Various attempts have been made to derive formulas for profile modification. However, an element of trial and error enters into

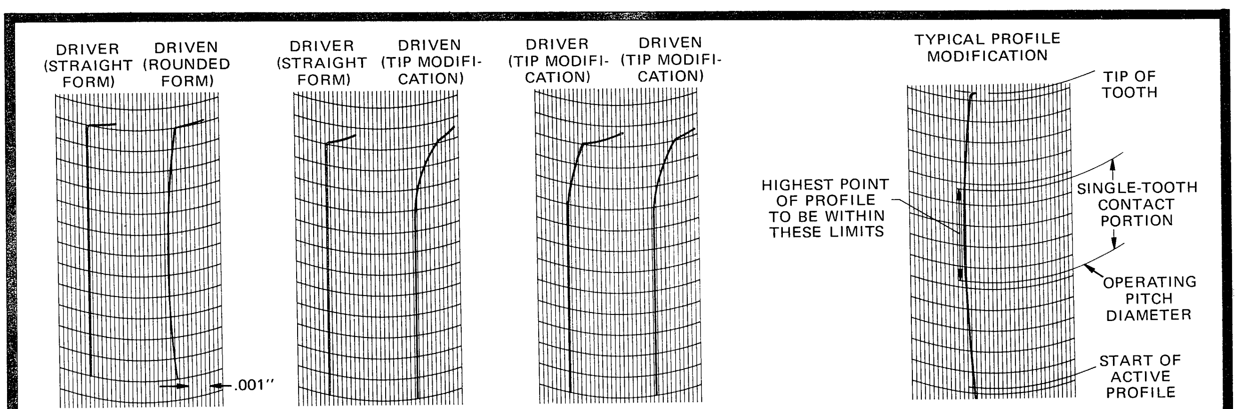


Fig. 6-3—Types of profile modification.

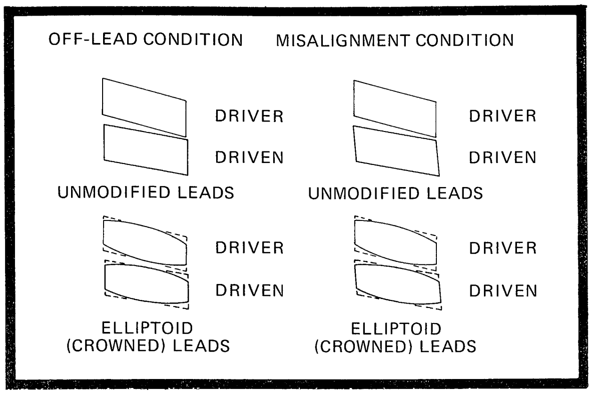


Fig. 6-4—How off-lead and misalignment conditions between a pair of mating gear teeth are corrected by the Elliptoid (crowned) tooth form.

nearly every case between the ideal of design and the practicality of production.

The lead or helix angle of the gear is subject to many of the same noise-producing problems as the involute profile. In addition there are the problems of gear case misalignment or deflection which can cause dimensionally perfect gears to perform as if they had errors. Gears have often been blamed for noises caused by defective mounting. Fig. 6-4 (top), shows a pair of gears mounted in a non-parallel condition, and another pair with helix angle errors. Both sets have end bearings, because they do not mate in a common plane. Gear tooth crowning (Elliptoid tooth form) corrects for these conditions, Fig. 6-4 (bottom).

Excessive crowning is as great an evil as no crowning. When the amount of crown is too great, effective face width is sacrificed. This condition is particularly detrimental to helical gears, which depend on helical overlap for smooth, quiet action. If the accumulated mounting errors or shaft deflection appear to call for gear tooth crowning in excess of 0.0005-in. per inch of face width on each tooth side, more rigid mounts, or stronger gear teeth should be considered.

Under certain conditions, either the gears or the gear housing may deflect under load to such an extent that normal lead modifications will not solve the problem. When corrected to solve a noise condition under load, these gears may be noisy when running without load. Or, conversely, gears that are quiet under no-load conditions, may be noisy when full load is applied. A combination of taper and lead crown in the gear teeth, Fig. 6-5, will satisfy both of these conditions.

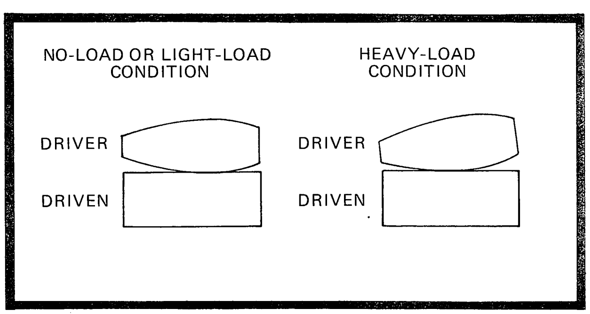


Fig. 6-5—How a combination of taper and lead crown can improve tooth mating conditions under heavy loading.

Although tooth modifications may affect calculated tooth contact ratios, so also do the detrimental results of deflection during operation. If the tooth modifications are properly applied, all detrimental contacts can be avoided. Some of the bad tooth contact conditions that occur when proper account is not taken of running conditions are shown in Fig. 6-6.

Proper tooth modification will keep contact out of the boundaries where it can affect sound quality and gear wear life.

Distortion of gears during heat treatment, although some of these effects can be reduced by proper blank design, is a problem that faces every gear manufacturer. Some distortion effects such as lead changes can be compensated for in the cutting operation. Involute profile compensation can also be made by shaving before heat treatment.

But if the effects of all heat treatment distortion are to be minimized, some type of gear finishing process must be specified. Depending on the machinability of the material, gears with a hardness of up to 40 Rockwell C scale maximum, can be shaved. But if material hardness exceeds this amount, either grinding or honing will have to be specified to eliminate the heat treatment errors, or remove objectionable nicks and burrs.

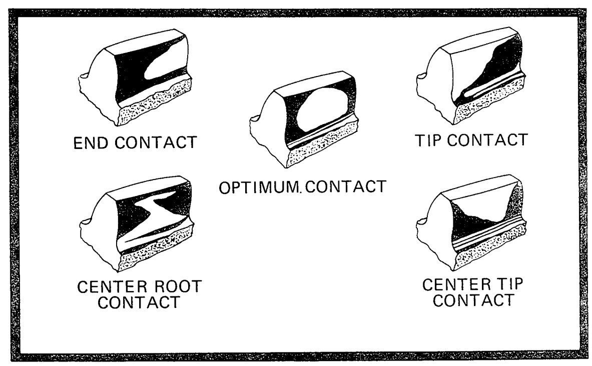


Fig. 6-6—Typical lead and involute patterns on gear teeth that occur when proper account is not taken of running conditions.

Production Economy

Overall production economy is another consideration that makes gear finishing operations attractive to a designer. It is possible, for example, to slow the gear hobbing process and use high-precision hobbing tools to produce gear teeth of reasonably high accuracy and good surface finish. But it is usually possible to produce gear teeth at lower cost by hobbing the gear at high feed rates with medium-accuracy gear hobs and follow with a rotary gear shaving operation. The resulting gear teeth produced by the high-production hobbing-shaving process are usually more accu-

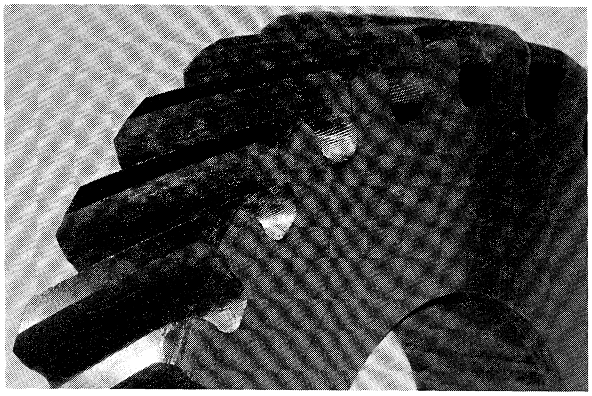


Fig. 6-7—Typical hobbed 10-D.P. helical gear teeth with a surface finish of from 65 to 70-microinches.

rate than those produced by a single slow-feed hobbing operation with precision hobbing tools.

Another production method that can be used to produce accurate gear teeth having improved tooth finish and accuracy is to rough-hob or shaper-cut the gear at high feed rates and finish-cut the teeth by hobbing or shaping at low feed with a precision tool. This method is still more costly, however, than applying hobbing or shaping followed by shaving.

Maximum economy and maximum quality of high performance, long-life, quiet gears is thus achieved by specifying a gear tooth finishing operation following the generation of the gear teeth by hobbing or shaping.

One important factor for the gear manufacturer to remember, however, is that gear finishing processes require fairly accurate hobbed or shaped gear teeth to assure that teeth of maximum quality can be produced.

In general, a gear shaving operation can remove from 65 to 80 percent of the errors in the hobbed or shaped gear, providing stock removal is held to from about 0.001-in. to 0.004-in. on tooth thickness depending on the diametral pitch of the gear teeth.

In grinding, it is extremely important that stock removal be kept to a minimum so that case-carburizing depths can be maintained, production cycles can be reduced, and detrimental surface-burning and drawing effects can be minimized.

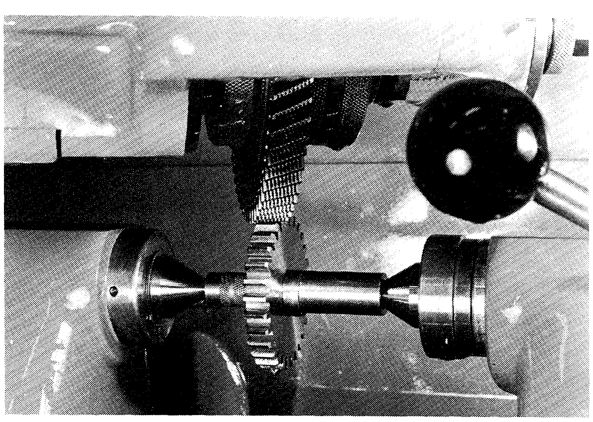


Fig. 6-8—Crossed-axes meshing of shaving cutter above the work gear with rotary shaving.

If a gear in the 4- to 6-in. dia., 7 to 10 diametral pitch range is to be finished by shaving, the hobbed or shaped gear teeth should not exceed the following tolerances: Runout, 0.001-in. to 0.003-in.; Involute Profile, 0.001-in.; Tooth-to-Tooth Spacing, 0.0005-in.; Tooth Lead or Parallelism, 0.001-in. per in. of face width; and Surface Finish, 100-mu to 200-mu. Roll-finished gears will require even closer hobbed or shaper-cut gear tolerances. Typical hobbed helical gear teeth are shown in Figure 6-7.

Selecting the Process

Rotary Shaving: Rotary gear shaving is a production process that utilizes a high-speed steel, hardened - and - ground, ultra - precision shaving cutter. The cutter is made in the form of a helical gear. It has gashes in the flanks of the teeth that act as the cutting edges.

The cutter is meshed with the work gear in crossed-axes relationship, Fig. 6-8, and rotated in both directions during the work cycle while the center distance is being reduced in small controlled steps. Simultaneously the work is traversed back and forth across the width of the cutter. The traverse path can either be parallel or diagonal to the work gear axis, depending on the type of work gear, the production rate and the finish requirements.

The gear shaving process can be performed at high production rates. It removes materials in the form of fine hair-like chips. Machines are available to shave external spur or helical gears up to 200-inches in diameter. Other machines are also available for shaving internal spur or helical gears.

For best results with shaving, the hardness of the gear teeth should not exceed 30 Rockwell C Scale. If stock removal is kept to recommended limits and the gears are properly qualified the shaving process will finish gear teeth in the 7 to 10-pitch range to the following accuracies: Involute Profile, 0.0002-in.; Tooth-to-Tooth Spacing, 0.0003-in. and Lead or Parallelism, 0.0002 in.

In any event it should be remembered that gear shaving can remove from 65 to 80 percent of the errors in the hobbed or shaped gear. It will make a good gear better. The quality of the shaved gear is dependent to a large degree upon having good hobbed or shaped gear teeth.

Excellent surface finish is achieved with gear shaving, Fig. 6-9. A value of about 25-mu, is the normal finish achieved with production gear shaving, although much finer finishes are possible by slowing the process.

In some cases, shaving cutters will finish up to 80,000 gears before they need sharpening. They may generally be sharpened from four to eight times.

To a gear designer, the shaving process offers attractive advantages in the ability to modify the

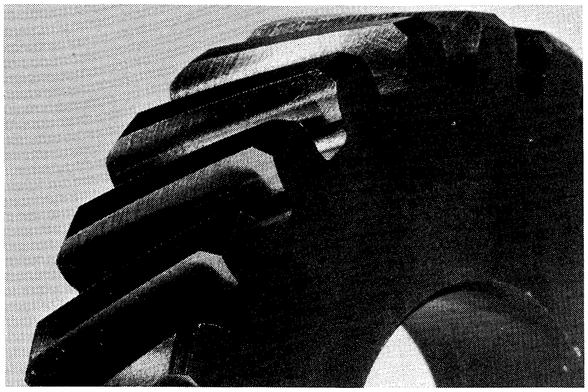


Fig. 6-9—Typical rotary-shaved 10-D.P. helical gear teeth with a surface finish of 16-microinches.

tooth form. If a crowned (Elliptoid) tooth form or a tapered tooth form are desired to avoid end bearing conditions, these can be easily provided by shaving.

If minute modifications are desired in the involute profile, these can be made by suitable modifications in the ground cutter tooth form.

If heat treatment distortions can be controlled to a minimum, the most inexpensive way to make an accurate, quiet, high-performance gear is to specify hobbing followed by either shaving or roll-finishing. The shaving and roll-finishing processes have a variety of standardized production equipment available (See Chapter 10); ranging from manual to highly-automated, automatically-loaded and automatically-gaged types. Cutter and rolling die modifications can be readily provided. Both cutters and dies are made in quantity to precision high standards of accuracy.

Roll-Finishing: Roll-finishing is a relatively new cold forming process. As applied on a double-die gear rolling machine, Fig. 6-10, the work gear is meshed in parallel-axes relationship between two hardened, high speed steel, precision-ground, power-driven gear rolling dies. The center distance between the dies is reduced during the work cycle to reduce the work gear tooth size and cold-flow the tooth flanks to produce a tooth form of high accuracy and excellent surface finish.

Roll finishing has been successfully applied to several high-production automotive transmission gears in the 8 to 20-pitch, 1-in. to 4-in. pitch diameter range.

Significant production economy with roll-finishing is the result of short work cycles and long rolling die life. Excellent surface finish, Fig. 6-11, of from 6 to 8-mu measured axially and from 15 to 20-mu measured along the profile is achieved. Other benefits of roll finishing are improved tooth strength and dimensional uniformity from piece-to-piece.

More than 1,000,000 pieces have been roll-finished by a set of dies before regrinding. It should be remembered that the hardness of the gears being rolled has a significant effect on the life of the dies. The die tooth forms can be ground

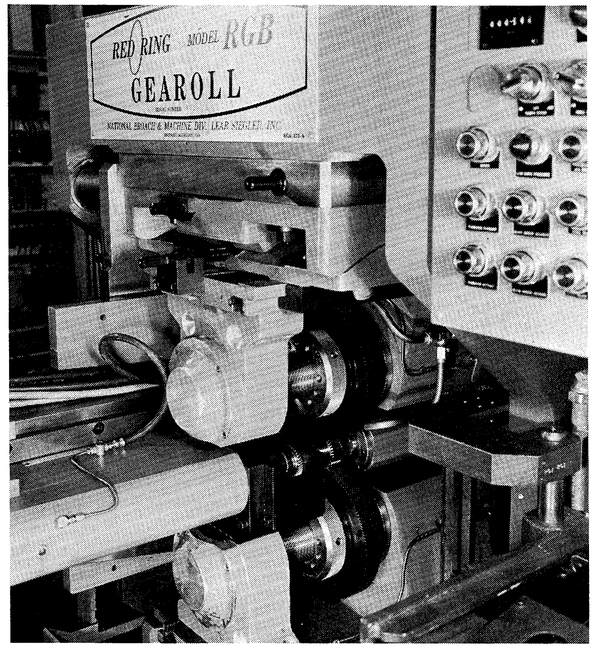


Fig. 6-10—Parallel-axes relationship in double-die roll-finishing of a transmission sun gear meshed between upper and lower dies.

to produce desired tooth form modifications, including crowning.

To successfully apply roll finishing, it is necessary to reduce by about one-half, the amount of stock traditionally left for gear shaving. Roll finishing will produce parts at a higher production rate than gear shaving.

In its current state of development, roll-finishing with double-die equipment seems better adapted to the smaller, high-production parts. Single-die roll finishing machines are better adapted for low and medium production gears, including those of larger diameter.

Gear Grinding: Grinding of gears is a slow and expensive production process. But when the absolute maximum in accuracy is demanded in a hardened steel gear, it may be the only method of finishing available. In the aerospace industry, where the weight of hardened parts has to be held to an absolute minimum, blank shapes are often

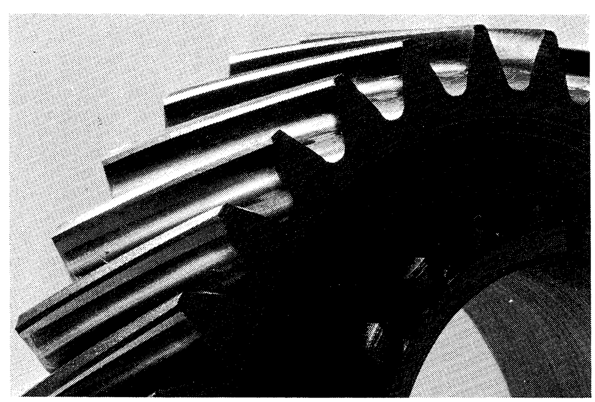


Fig. 6-11—Typical roll-finished 6-D.P. helical gear teeth with 6 to 8-mu finish along the lead and 15 to 20-mu finish measured along the profile.

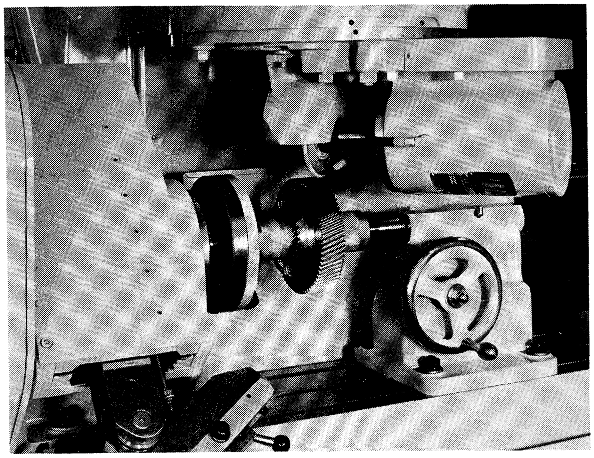


Fig. 6-12—Grinding wheel above work gear in angular relationship for form-grinding of the teeth.

in a form that distorts a considerable amount during heat treatment. As a result, gear grinding is widely applied in aircraft and missile power transmission components.

If the grinding process is carried out at excessive stock removal rates, tooth surface deterioration may result in the form of discoloration, hardness reduction and cracking. A method known as Nital Etch is used to determine any tempering or rehardening condition for high-performance gears where no surface deterioration is permitted. The Nital Etch method is a critical test, and if it is used as an inspection tool, the grinding process may have to be slowed.

Two different grinding methods are in present use. The first is called form grinding, Fig. 6-12. It uses a single-formed grinding wheel that is passed back and forth through the gear space to finish one or two tooth flanks at a time before the work is indexed to the next tooth space. The involute or modified-involute tooth form is generated on the grinding wheel by a dressing attachment.

The second grinding method is called generation grinding and can be carried out by either one or two single-formed wheels; or a multi-formed wheel resembling a worm. When a single-form grinding wheel is used, one tooth flank at a time is ground with an angular straight side of the wheel while the work gear is rotated in timed relationship with the wheel to generate the tooth form; and the work gear is slowly reciprocated along its axis. Then the work is indexed to the next tooth space. Two opposite, non-adjacent tooth sides can be ground simultaneously with a pair of single-tooth grinding wheels by the process.

When a multi-formed wheel is used in generation grinding, the work gear and wheel rotate in mesh in a manner similar to the action of a hobbing machine; and the operation takes place continuously without a separate indexing sequence.

The grinding wheel shape with generation grinding is either straight-sided or modified straight-side and is produced by a dressing attachment.

When single-form grinding wheels are utilized, the number of passes and the sequence of roughing and finishing operations depends upon the application and the desired accuracy. Similarly, the rate at which the multi-tooth wheel is fed into depth is also dependent on the application and work quality.

Surface finishes of form-ground gears, Fig. 6-13, measured across the profile are usually in the 20 to 30-mu range.

Gear Honing: This hard gear finishing method uses an abrasive-impregnated plastic or metal-bonded honing tool, made in the form of a helical gear. As in the gear shaving process, it is meshed with the work gear in crossed-axes relationship and rotated in both directions under controlled tight-mesh conditions. Slow relative reciprocation motion parallel to the axis of the work gear is imparted between the tool and the work gear during the honing process, Fig. 6-14.

The honing process is an excellent high-production method of reducing sound level of hardened gears by removing nicks and burrs, improving tooth surface finish and making minor corrections in tooth shape. It has been applied suc-

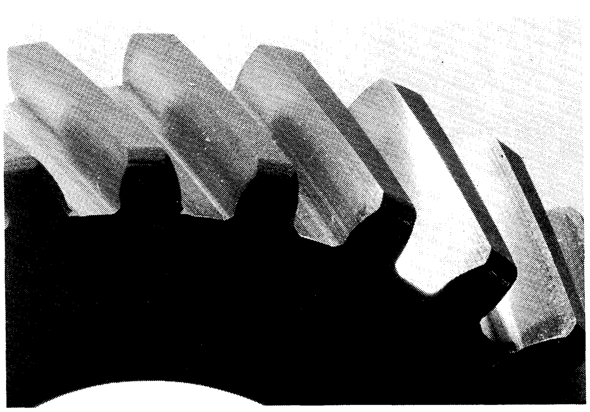


Fig. 6-13—Typical form-ground 10-D.P. helical gear having 12-mu finish along the lead and 20-mu on the profile.

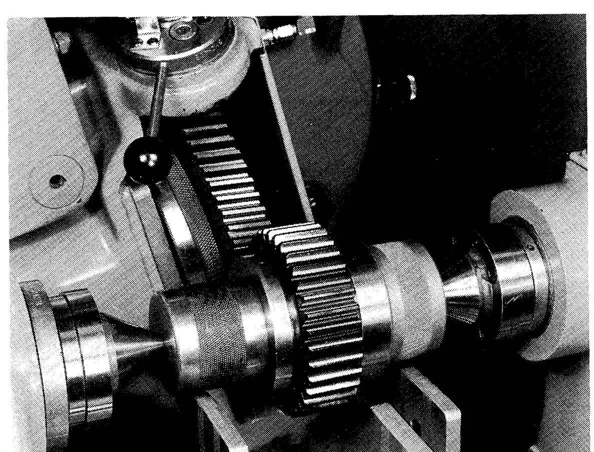


Fig. 6-14—Crossed-axes relationship of honing tool and work gear. The honing tool is at the rear.

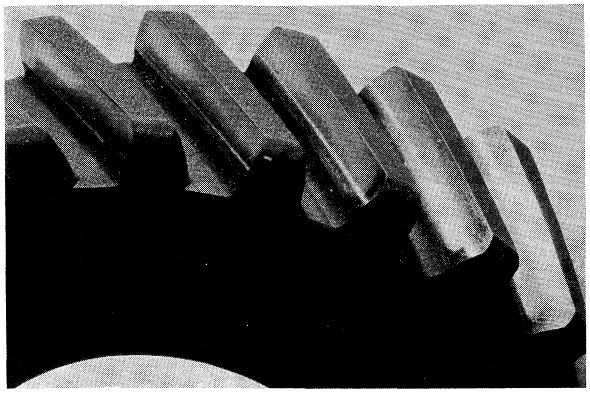


Fig. 6-15—Typical honed 10-D.P. helical gear teeth with 12-mu finish along the lead and 15-mu across the profile.

cessfully in nick and burr removal operations in which all gears are honed before assembly to avoid teardown operations. Like gear shaving, it is a process that can be highly automated and has short operating cycles.

The honing tool can have the tooth form made to produce gears with tooth modifications.

It has been found that when maximum wear life is desired from heavily-loaded, high-performance gears, quality of surface finish is a most important design consideration. The surface finishes achieved with honing are in the 10 to 15-mu range, Fig. 6-15. If the quality of the gear warrants the additional honing time investment, finishes down to 4-mu can be achieved.

Although gear honing was originally developed as a method of inexpensively and assuredly removing nicks and burrs from hardened gear teeth, it is now finding broad application in the improvement of tooth surface finish of both shaved and ground hardened gears.

Honing of a ground gear can improve its surface finish and also make small improvements in tooth-to-tooth spacing accuracy as a result of the same hunting tooth action as gear shaving.

Specifying the Process: The specification of tolerances and dimensions on gear teeth is a subject to which a great deal of discussion could be devoted. However, it is important to point out here that the mere specification of tolerances without some guidance as to the production processes required, may lead to much confusion between the engineering and manufacturing departments.

Gears are one power transmission component in which the broadest cooperation between the designer and the manufacturing department is required. If tolerances are too close, the cost of production can skyrocket. But if performance is the major consideration, the additional investment in one or more gear finishing processes may be justified, Table 6-1. For example, in some aerospace gearing, it has been found that without gear honing, little or no wear life can be achieved. Further, it has been found that capacities of gear trains in jet engine drives have been significantly increased by honing the gear teeth as a final production operation after grinding.

In automotive automatic transmissions, all gears are shaved or roll-finished before heat treatment and many are honed after heat treatment in a fast production operation to remove the nicks and burrs and lower noise levels. Gear noise has become a more and more objectionable quality in modern power transmission devices. Even most machine tool drive gears today are shaved, as are gearmotor reduction gears.

Once the designer and the manufacturing engineer have reached agreement on acceptable quality and performance levels in production gearing it is strongly recommended that the drawings indicate both the tolerances and the required finishing process.

Table 6-1—Typical Production Times for Various Gear Finishing Processes

(Times estimated for a 25-tooth, 2.9-in. pitch diameter, 5/8-in. wide steel gear having 10-diametral pitch, 20-deg pressure angle teeth with a 32-deg left-hand helix angle.)

Process	Production Time
Gear Shaving (Conventional Method)	43-sec.
Gear Shaving (Diagonal Method)	22-sec.
Gear Roll Finishing	10-sec.
Grinding (With Nital Etch Surface Control)	
Form-Type	16 to 24-min.
Generating Type (With Two Single-Form Wheels)	20 to 30-min.
Generating Type (With Multi-Form Wheel)	14 to 18-min.
Grinding (Without Nital Etch Control)	From 1/2 to 1/3 of the times indicated for grinding with Nital Etch control.
Gear Tooth Honing	21-sec.

The Gear Shaving Process

Gear Shaving is a free-cutting gear finishing operation which removes small amounts of metal from the working surfaces of the gear teeth. Its purpose is to correct errors in index, helical angle, tooth profile and eccentricity Fig. 6-16. The process can also improve tooth surface finish, and eliminate by crowned tooth forms the danger of tooth end load concentrations in service. Shaving provides for form modifications that reduce gear noise. These modifications can also increase a gear's load carrying capacity, its factor of safety and its service life.

Gear finishing (shaving) is not to be confused with gear cutting (roughing). They are essentially different. Any machine designed primarily for one cannot be expected to do both with equal effectiveness, or with equal economy.

Gear shaving is the logical remedy for the inaccuracies inherent in gear cutting. It is equally effective as a control for those troublesome fire distortions caused by heat-treatment.

The form of the shaving cutters can be reground to make crown and profile allowance for different heat treat movements due to varying heats of

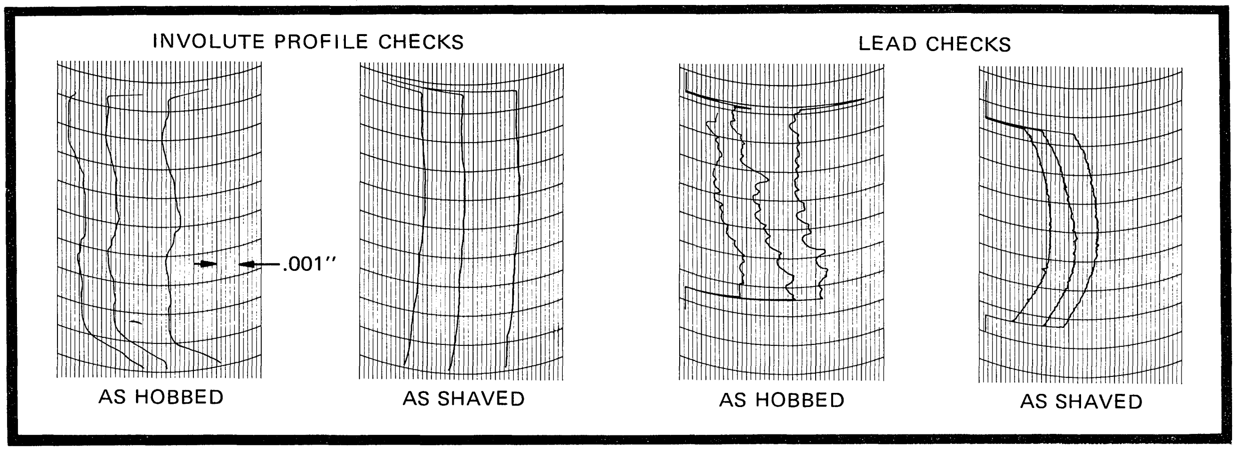


Fig. 6-16—Charts showing improvement in lead and profile when 8620, 5.7-D.P., 20-deg, P.A., 3.85-in. P.D. hobbed gears are crown-shaved with a stock removal of 0.011-in. over pins.

steel. The gear shaving machine can also be reset to make allowance for lead change in heat treatment.

Basic Principles

The Red Ring rotary gear shaving process is based on patented fundamental principles. This process uses a gashed rotary cutter, Fig. 6-17, in the form of a helical gear having a helix angle different from that of the gear to be shaved. The axes of cutter and work gear are crossed at a predetermined angle during the shaving operation.

When cutter and work gear are thus rotated in close mesh, the edge of each cutter gash as it moves over the surface of a work gear tooth shaves a fine hair-like chip, somewhat like that produced by a diamond boring tool, Fig. 6-18.

The finer the cut, the less the pressure required between tool and work, eliminating the

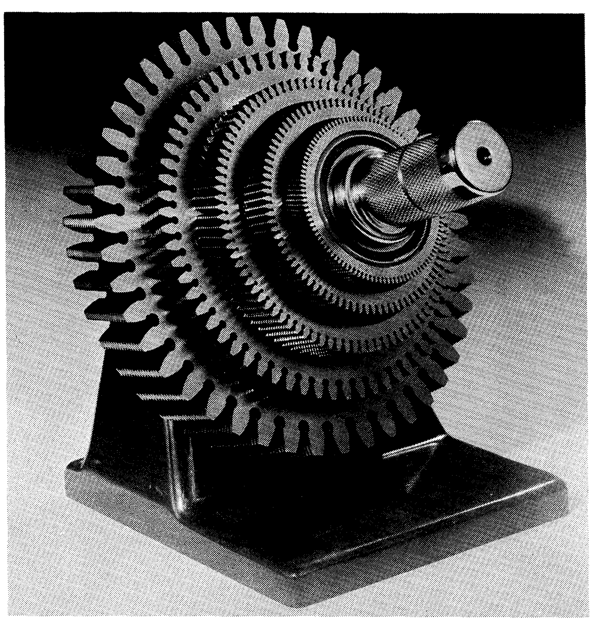


Fig. 6-17—An assortment of rotary gear shaving cutters with varying pitches and diameters.

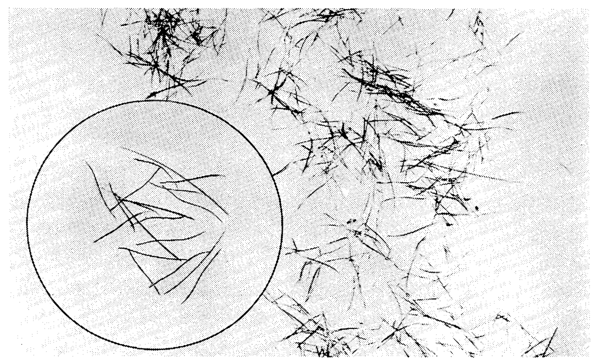


Fig. 6-18—Fine hair-like curled chips produced by rotary gear shaving.

tendency to cold-work the surface metal of the work gear teeth.

This process is utilized in a shaving machine, Figs. 6-19 & 20, which has a motor-driven cutter head and a reciprocating work table. The cutter head is adjustable to obtain the desired crossed axes relationship with the work. The work, carried between live centers, is driven by the cutter.

During the shaving cycle, the work is reciprocated parallel to its axis across the face of the cutter and up-fed an increment into the cutter with each stroke of the table. This shaving cycle (conventional) is one of several methods.

The Crossed-Axes Principle: To visualize the crossed-axes principle, consider two parallel cylinders of the same length and diameter, Fig. 6-21.

When brought together under pressure, their common contact surface is a rectangle having the length of a cylinder and a width which varies with contact pressure and cylinder diameter.

When one of these cylinders is swung around so that the angle between its axis and that of the other cylinder is increased up to 90 degrees, their common contact plane remains a parallelogram but its area steadily decreases as the axial angle increases.

The same conditions prevail when, instead of the two plain cylinders, a shaving cutter and a

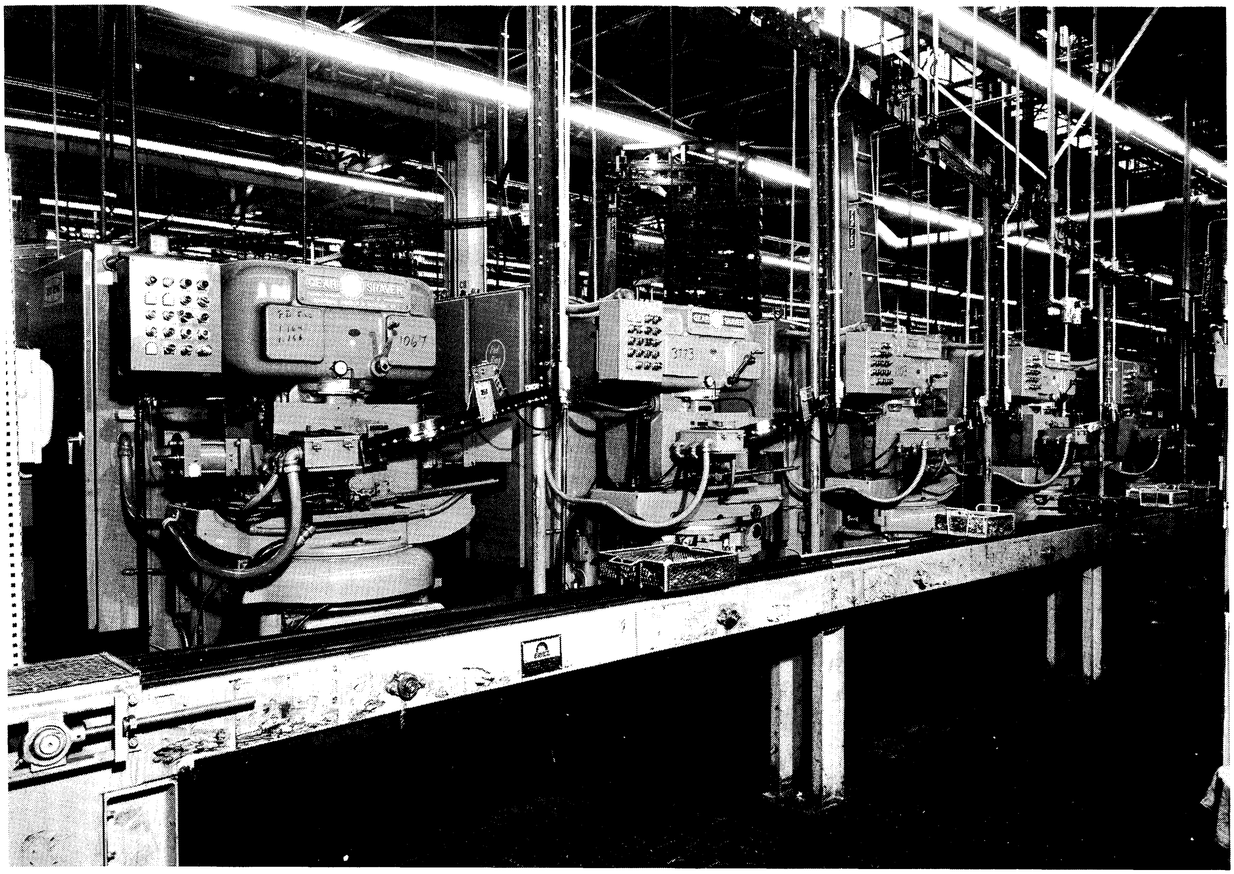


Fig. 6-19—A battery of gear shavers for finishing automotive transmission planet pinions.

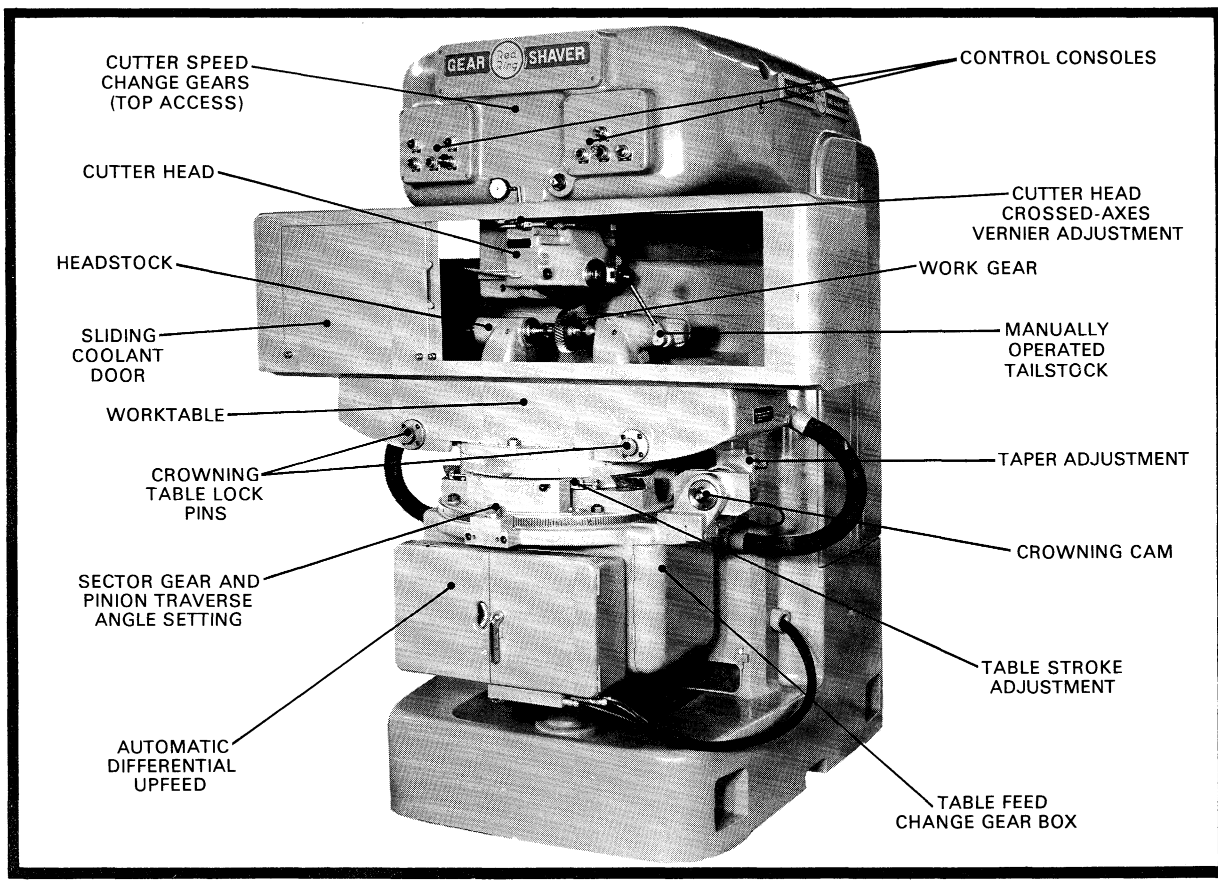


Fig. 6-20—Operating components of a knee-and-column type universal gear shaving machine equipped for manual loading.

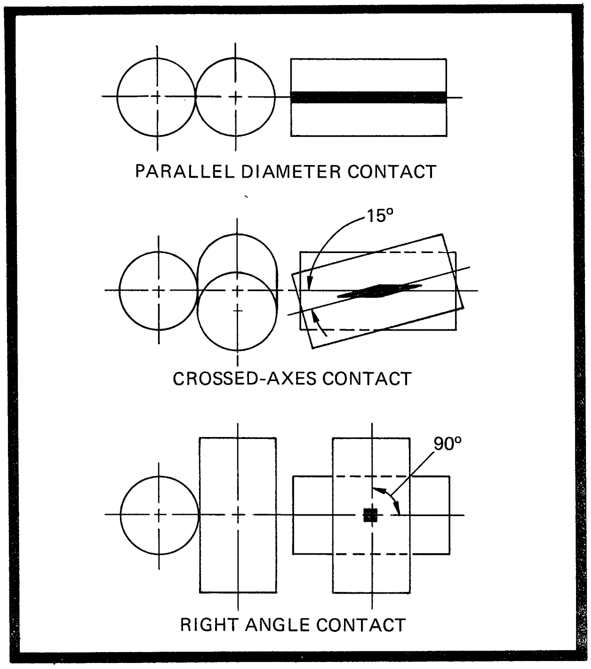


Fig. 6-21—How the contact between cylinders changes as the angle between their axes is varied.

work gear are meshed together. When the angle between their axes is from 10-deg. to 15-deg., tooth surface contact is reduced and pressure required for cutting is small.

As the work gear is moved away axially from the point of intersection of the axes, backlash develops. Conversely, as it is returned to the point of axial intersection, backlash decreases until the two members engage in tight mesh with the teeth of the cutter wedging between those of the work gear. Thus, each succeeding cutting edge sinks deeper into the work gear tooth until the point of axial intersection is reached.

For shaving, the cutter and work gear axes are crossed at an angle usually in the range of 10-deg. to 15-deg. or approximately equal to the difference in their helical angles.

Crossing of the axes produces reasonably uniform diagonal sliding action from the tip of the teeth to the root. This not only compensates for the non-uniform involute action typical of gears in mesh on parallel axes, but also provides the necessary shearing action for metal removal.

Relation Between Cutting and Guiding Action: Increasing the angle between cutter and work gear axes increases cutting action but, as this reduces the width of the contact zone, guiding action is sacrificed. Conversely, guiding action can be increased by reducing the angle of crossed axes but at the expense of cutting action. At zero angle there is practically no cutting.

The spur and helical gears at the left in Fig. 6-22 were shaved without table reciprocation. They show the band of cutter contact less than the gear face width; and also a deeper cut in the middle of the band than on either side.

The gears in the center of Fig. 6-22 were shaved with only about ¼-in. of table reciprocation. The profiles of these teeth are perfect over the distance of reciprocation but fade out at each end. The chordal thickness along this ¼-in. length is less than that at the ends.

Gears at the right in Fig. 6-22 were shaved with full table reciprocation and therefore have been finished to full depth over the entire face. Tooth

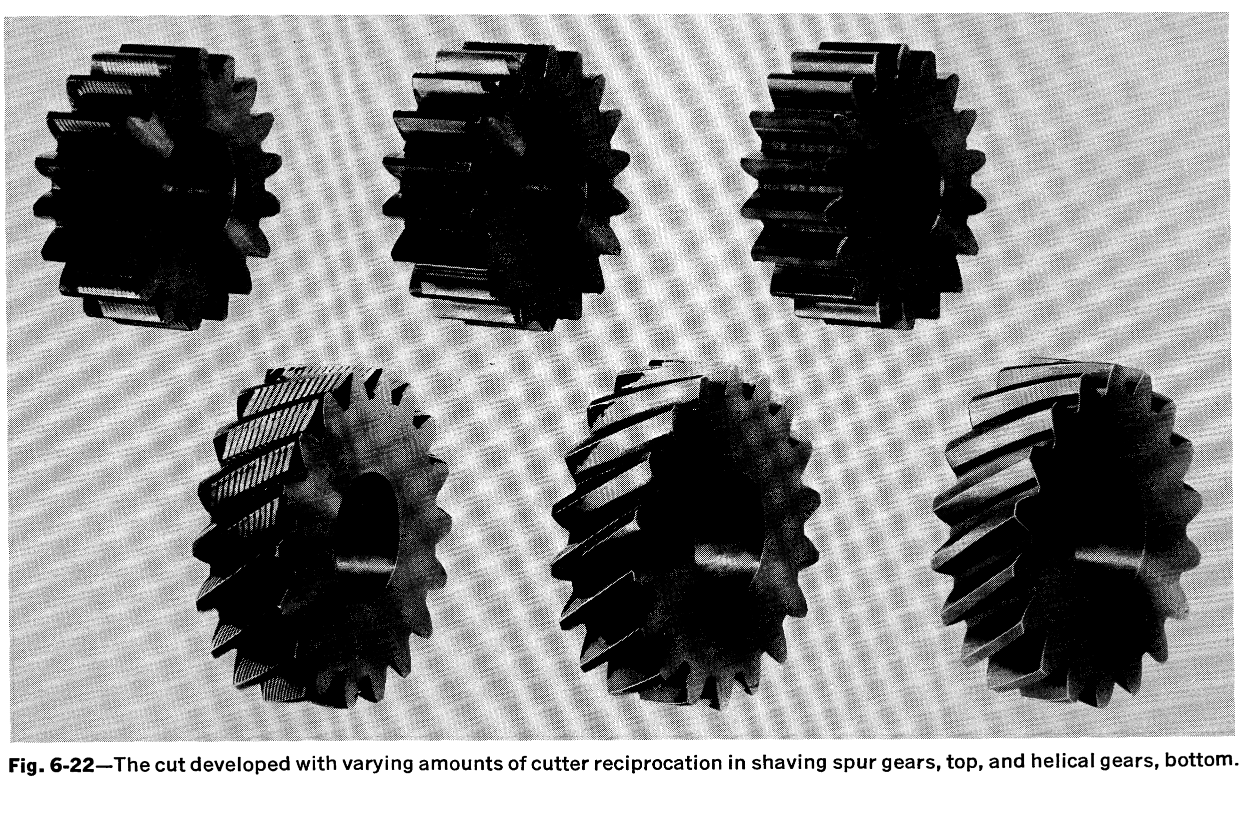


Fig. 6-22—The cut developed with varying amounts of cutter reciprocation in shaving spur gears, top, and helical gears, bottom.

thickness may be decreased by increasing the up-feed of work toward the cutter.

Shaving Methods

Conventional rotary shaving, Fig. 6-23 is widely used in low and medium production operations. Diagonal shaving not only increases production rates substantially but also has other advantages for high-production operations.

The principal difference between conventional and Diagonal shaving is in the direction of reciprocation of the work through and under the cutter. In conventional shaving, reciprocation is parallel with the work gear axis. In Diagonal shaving it is at an angle with that axis, Fig. 6-24.

Although conventional shaving requires a number of table strokes, each with its increment of up-feed, Diagonal shaving of finer-pitch gears may be done in just two strokes with no up-feed and a fixed center distance between cutter and work. Diagonal shaving, however, is not restricted to a 2-stroke cycle as explained later.

Traverse Angle and Cutter Selection: The traverse angle for Diagonal shaving is the angle between the direction of traverse and the work gear axis.

Relative face width of the gear and the shaving cutter has an important relationship with the diagonal traverse angle. A wide-face work gear and a narrow-faced shaving cutter restrict the diagonal traverse to a small angle. Increasing the face width of the shaving cutter permits an increase in diagonal traverse angle.

When the shaving cutter face width is increased to slightly greater than the work gear face width, traverse angles up to 90-deg. are permissible. Traverse angles above 60-deg. require a special cutter having differential serrations.

The maximum theoretical diagonal traverse angle is determined as follows;

$$\text{Tangent Max. Traverse Angle} = \frac{\text{Cutter Face Width} \times \text{Sine Crossed-Axes Angle}}{\text{Gear Face} - (\text{Cutter Face} \times \text{Cosine Crossed-Axes Angle})}$$

In most cases the diagonal traverse angle will vary from 30 to 60-degrees. Usually it is in the

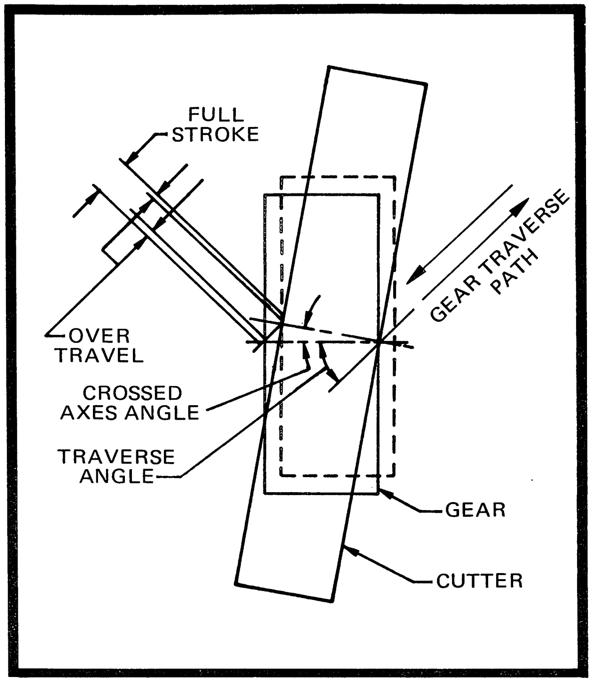


Fig. 6-24—Diagonal shaving.

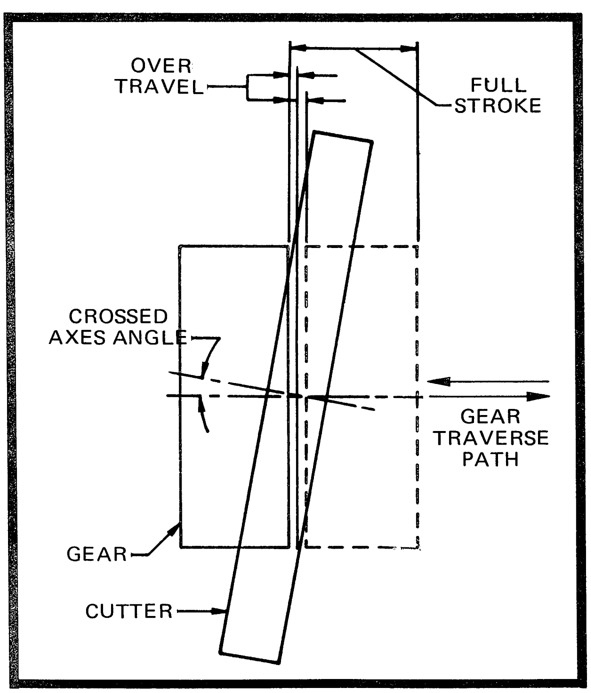


Fig. 6-23—Conventional shaving.

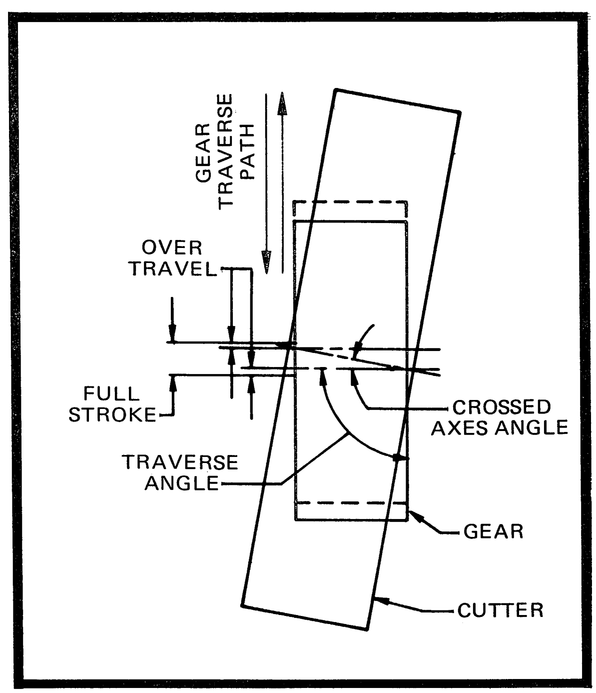


Fig. 6-25—90-Deg. traverse shaving.

Table 6-2—Typical Shaving Production Rates for Different Gears

Name of Gear	Material	No. of Teeth	Diam. Pitch	Pitch Dia., in.	Pressure Angle, deg.	Helix Angle, deg.	Face Width, in.	Stock Removal, in. Over Pins	Production Rates, Loading Method, and Shaving Process
Planet pinion	Steel	12	13.5	1⅛	20°	27°16′2″	1	0.005	240 per hr, auto. load, Diagonal
Timing Gear	Cast iron	28	10	2⁵⁄₁₆	14°30′	23°17′49″	1¹⁄₁₆	0.010	130 per hr, semiauto. load, Diagonal
Shoulder gear (small gear on two gear cluster)	Steel	14	6/8	2⁹⁄₁₆	20°	Spur	¹¹⁄₁₆	0.006	160 per hr, semiauto. load, Diagonal
Cluster gear (one of large gears on a four-gear cluster)	Steel	29	9.25	4	16°30′	35°56′14″	¹¹⁄₁₆	0.010	95 per hr, semiauto. load, Diagonal
Timing gear	Aluminum	56	10	6½	14°30′	23°17′49″	1	0.010	95 per hr, semiauto. load, Diagonal
Bull gear	Steel	158	7	24	20°	20°	4½	0.010	2 per hr, manual load, conventional
Bull gear	Steel	400	6	72	14°30′	22°30′	12	0.001-0.002″ total on tooth thickness	5 hr each, manual load, conventional
Marine propulsion gear	Steel	1,128	6	200	14°30′	30°	36 (herringbone)	0.001-0.002″ total on tooth thickness	130 hr, manual load, conventional

range of 40-deg. to obtain optimum conditions of cutting speed and work gear quality.

A full 90-deg. traverse may be used, Fig. 6-25, providing the cutter has special (differential) serrations and its face width exceeds that of the work gear. The 90-deg. traverse method is ideally adapted to the shaving of close-shoulder gears.

In the traverse angle range of 30 to 60-deg., the standard Diagonal cutter without specially designed serrations will suffice. In many cases, the conventional shaving type of cutter may be used, providing a slight amount of crown in the gear is permissible.

Advantages of Diagonal Shaving: Diagonal shaving is much faster than conventional shaving. In many cases it is up to 50-percent faster. The process also offers greater latitude in handling shoulder gears with critical clearance between the gear and the shoulder.

With diagonal traverse, cutting is not restricted to a small zone of the cutter as it is in conventional shaving, but is migrated across the entire cutter face. Consequently, cutter life is extended.

Multi-Stroke Diagonal Shaving: An automatic up-feed mechanism on the shaving machine materially enlarges the scope of Diagonal shaving by making it available also for multi-stroke operations. This device feeds the work into the cutter in a series of small increments, synchronized with table reciprocation.

Removing stock from the work gear in a series of small increments instead of two larger increments, further increases cutter life. It also makes the process feasible for gears requiring more stock removal than can be handled on a two-stroke cycle.

When up-feed is completely automatic, there can be no danger of an error in selecting feed rates. Inasmuch as the cycle starts and stops in a position of maximum backlash, loading and unloading can be very fast.

Table 6-2 gives typical shaving production rates for various gears and traverse methods.

The Shaving Cutter

Rotary shaving cutters, Fig. 6-26 are high-precision, hardened-and-ground, high speed steel generating tools held to Class "A" and "AA" tolerances Table 6-3 in all principal elements.

The gashes in the Red Ring Cutter extend the full length of the tooth, terminating in a clearance

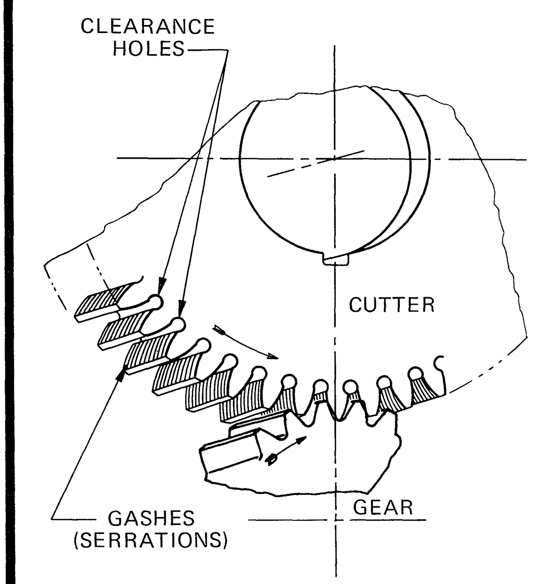


Fig. 6-26—Work gear in crossed-axes mesh with rotary shaving cutter mounted above it.

space at the bottom. These clearance spaces provide unrestricted channels for a constant flow of coolant to promptly dispose of chips. They also permit uniform depth of serration penetration and increase cutter life.

The shaving cutter is rotated at high speeds up to 400 and more surface feet per minute. Feed is fine and the tool contact zone is restricted.

Cutter life depends on several factors: Oper-

Table 6-3—Tolerances for Rotary Gear Shaving Cutters

Tolerance Specification	Cutter Specifications					
	4 Thru 19.999 NDP 13° Helix Angle & Over Thru 9.499″ PD		4 Thru 19.999 NDP 13° Helix Angle & Over 9.500 Thru 13.499″ PD		20 Thru 120 NDP 20° Max. Helix Angle 2 Thru 4″ PD	
	Class A (In.)	Class AA (In.)	Class A (In.)	Class AA (In.)	Class A (In.)	Class AA (In.)
Involute Profile (True Involute Forms)—Active Length—tiv						
Thru .177 Working Depth	.00020	.00015	.00020	.00015	.00020	.00015
.178 thru .395 Working Depth	.00025	.00020	.00025	.00020		
.396 thru .610 Working Depth	.00030	.00025	.00030	.00025		
Lead—(Uniformity—tiv Per Inch of Face)	.0004	.0003	.0004	.0003	.0003	.0002
Parallelism—(Opposite Sides of Same Tooth Alike Within)	.0003	.0002	.0003	.0002	.0003	.0002
Helix Angle—(Deviation from True Angle —Per Inch of Face)	.0010	.0005	.0010	.0005	.0005	.0003
Tooth Spacing—(Adjacent Teeth at PD)	.0002	.00015	.0002	.00015	.0002	.00015
Circular Pitch—(Variation—tiv)	.0004	.0002	.0004	.0002	.0004	.0002
Spacing Accumulation—(Over 3 Consecutive Teeth)	.0004	.0003	.0004	.00025	.0003	.0002
Runout—(tiv at PD)						
Thru 4.499″ PD						
Over 19° NPA	.0007	.0005				
13° thru 19° NPA	.0011	.0007				
4.500 thru 9.499″ PD						
Over 19° NPA	.0009	.0006				
13° thru 19° NPA	.0012	.0008				
9.500 through 13.499″ PD						
Over 19° NPA			.0010	.0007		
13° thru 19° NPA			.0014	.0010		
2.000 through 4.000″ PD						
Over 19° NPA					.0007	.0005
13° thru 19° NPA					.0009	.0006
Face Runout—(tiv below Teeth)	.0002	.0002	.0003	.0002	.0002	.0002
Tooth Thickness	−.0010	−.0010	−.0010	−.0010	−.0010	−.0010
Hole	+.0002	+.0002	+.0002	+.0002	+.0002	+.0002
Outside Diameter	+.010 −.040	+.005 −.020	+.010 −.040	+.005 −.020	+.010 −.040	+.005 −.020

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ating speed, feed, material and hardness of the work gear, its required tolerances, type of coolant, and the size ratio of cutter to work gear.

Shaving Cutter Selection: Theoretically one shaving cutter will shave all gears of a given pitch and pressure angle regardless of the number of teeth. A slight involute variation due to the difference in sliding action may be noticeable on gears in the coarser pitches having small numbers of teeth. Therefore, the range of gears that can be shaved with one cutter, will depend on the allowable involute tolerances.

A shaving cutter that is ground to produce a true involute for a 30-tooth, 8-DP gear, might produce a plus involute profile on a 20-tooth gear and a minus involute on a 60-tooth gear. The variation although slight (only a few "tenths") could be harmful, depending on the service function of the gears in question. In this example, however, with a 20-tooth pinion driving a 60-tooth gear, satisfactory function could be expected from these high and low profiles.

Gears with the smaller numbers of teeth are more critical from a production and sound standpoint than those with greater numbers of teeth. Likewise, spur gears are generally more critical on both scores than helical gears.

In jobbing work, it is generally preferable to use two or three shaving cutters to cover the full range of gears. One company in this line uses two cutters for gears of 16-DP and finer. One of these is ground for a range of 14 to 22-teeth (approx.) while the second cutter covers the range higher than 22-teeth. One of the chief reasons for the use of two cutters is to accommodate the difference in fillet condition on the gears having small numbers of teeth. Involute variation in fine pitches is negligible.

In another case involving 8-DP to 10-DP gears, three shaving cutters are used: The first for 14

Table 6-4—Tooth Ranges of Shaving Cutters for Jobbing Work

Diametral Pitch	RANGE NUMBER				
	1	2	3	4	5
	Teeth in Gears				
4 and 5	15 to 17	17 to 25	24 to 37	35 to 60	55 and up
6 and 7	15 to 17	17 to 26	23 to 40	35 to 60	55 and up
8 to 10	14 to 18	17 to 29	27 to 60	60 and up	
12 to 18	14 to 18	17 to 34	25 to 60	60 and up	
20 to 30	13 to 22	20 and up			
32 to 64	12 to 24	20 and up			

to 18-teeth, the second for 17 to 29-teeth and the third for gears having 27-teeth or more.

Approximate ranges of teeth that can be shaved by one cutter are shown in Table 6-4. The ranges shown are average for general work. They are overlapping and flexible depending on the largest and especially the smallest number of teeth to be shaved. The exact limits depend on tolerances, material hardness and pre-shaving operations and are determined by trial.

Gears with numbers of teeth smaller than those shown in the table may require additional cutters. The table is based on full-pitch spur gears with pressure angles from 14 to 22-deg. and cut with hobs or shaper cutters that maintain root and fillet clearance for the shaving cutter tip. Long and short addendums, over-size, stub tooth, and special pressure angle gears require special consideration.

Right and left-hand cutters are usually required for a range of helical gears. However, gears ranging from 5-deg. to 18-deg. helix angle, right and left-hand, may be shaved with one straight spur cutter. The range of helical angles which one cutter can handle is determined by the crossed-axes angle between cutter and work gear. The most desirable range of crossed-axes angles is from 5-deg. to 15-deg. Thus, one cutter having a 15-deg. helix angle will shave spur gears and opposite-hand helical gears from 0-deg. to 10-deg. helix angle and from 20-deg. to 30-deg. approximately.

Occasionally, it becomes necessary to modify the involute form of the shaving cutter in order to produce teeth of the desired form on the work gear as, for instance, a tip relief or a rounded involute form. The effect of such modification may vary somewhat for gears of different numbers of teeth shaved with the same cutter.

On high production operations where involute form variation may be necessary to rectify either heat-treat distortion or troublesome noise conditions, it is desirable to use separate cutters for each gear or limited group of gears.

In setting up for jobbing work, it is preferable to start with a minimum of cutters, perhaps one in each pitch for spur gears or one right and one left-hand to suit helicals and spurs.

Cutter Design: Shaving cutters are designed in much the same manner as other helical involute gears. The serrations on the tooth profiles, in conjunction with the crossing of the axes of cutter and work gear, make it a cutting tool. In designing Red Ring Shaving Cutters, the following points are considered:

1. Normal Diametral Pitch and Normal Pressure Angle must be the same as those of the gears to be shaved.
2. Helix Angle is chosen to give a desired crossed-axes angle between cutter and work. The crossed-axes angle is the difference between the helix angle of the shaving cutter and that of the work gear. The desired range is from 5-deg. to 15-deg. for spur gears and from 5-deg. to 12-deg. for helical gears. Experience has established these as the most desirable ranges although in some cases it is permissible to extend them to from 3-deg. to 18-degrees. At the lower end of this extended range, cutting action is reduced and at its upper end, guiding action is somewhat sacrificed.
3. The Number of Teeth is chosen to give the approximate pitch diameter required, considering helix angle and diametral pitch. Hunting tooth conditions, and machine capacity are also important factors to consider:
4. Cutter Size is determined by diametral pitch and shaving machine capacity. The basic sizes are 3-in., 5-in., 7-in., 9-in. and 12-in. pitch diameter. From Table 6-5, cutter sizes for external work gears may be approximated. Occasionally, because of the shape of the part to be shaved (such as a gear which has a large adjacent shoulder), or because the gear is of small diameter, a larger cutter would be used in order to reach the gear and stay within the capacity of the machine.
5. Face Width is determined by the nature of the operation. The face width of shaving cutters used for Diagonal shaving must be calculated to obtain desired traverse angle. Open-type, no-hub cutters for internal gears are usually 0.625-in. face width. The width of cutters for gears with an adjacent shoulder is usually narrower than that of cutters for open gears. Face width of Red Ring Cutters is also varied to control cutting action.
6. The Operating Condition is based on an engineering analysis of the gears to be shaved, tempered with previous experience. Initially, it is the amount that the cutter is undersize or oversize which governs the operating pres-

Table 6-5—Shaving Cutter Characteristics

Nominal Cutter Size (in.)	Theoretical Pitch Diameter Range	Approximate Diametral Pitch Range	Std. Width—Conventional Shaving (in.)	Hole Size (in.)
3	2.500 3.499	20 to 64	½	1.500
5	4.500 5.499	16 to 36 (12 to 64 maximum)	⅝	1.500
7	6.500 7.499	7 to 16 (6 to 24 maximum)	¾	2.500
9	8.500 9.499	4 to 10 (3 to 14 maximum)	1	2.500
12	11.500 12.499	2 to 4 (2 to 8 maximum)	1	2.500

sure angle between cutter and gear. The cutter must be designed so that it may be sharpened several times. The Operating Condition for Red Ring shaving cutters varies with pitch, pressure angle and number of work gear teeth. Generally, cutters for low pressure angles are enlarged and those for higher pressure angles reduced, depending on the actual condition of the gears to be shaved. If the gears are undersize or oversize, the operating condition is affected and is adjusted in accordance with past experience.

7. Tooth Thickness of the cutter is the difference between the normal circular tooth thickness and the normal circular pitch of the work gear, based on the actual operating condition of the cutter. This involves the enlargement or reduction of the cutter and variations when the cutter is reground.

8. The Addendum is always calculated so that the shaving cutter will finish the gear profile slightly below the lowest point of contact with the mating gear. This addendum may be varied to suit fillet (root radius, conditions of undercut, protuberance) and the size of the mating gear. Tooth thickness and addendum of the cutter are not necessarily given to the theoretical pitch diameter.

9. Cutter Serrations are lands and gashes in the involute profile at a right angle to the cutter axis and parallel to its sides. They extend from top to bottom of the tooth, clearing into an oil hole at its base. They have an involute form and a constant depth throughout. Their width or size is determined from past experience and with consideration given to the type of work gears to be shaved. The gashes are usually a little wider than the lands on a new shaving cutter. Very often the lands are tapered on one side for strength and to facilitate washing the chips out. The depth of serrations is determined by their strength with allowance for resharpening.

   Differential serrations with a controlled lead are produced on shaving cutters used for plunge shaving operations on internal gears, Diagonal shaving with traverse angles over 55-deg., and for 90-deg. traverse shaving.

10. Oil Holes are drilled between and at the base of the teeth of Red Ring shaving cutters to provide clearance for chips formed by the shaving operation. An unrestricted flow of coolant passes through the serrations and holes to keep the tool clear of these chips. The holes are calculated so that the distance between them is seldom less than the hole diameter.

11. The Hole Circle Diameter is chosen to allow clearance between the tips of the gear teeth being shaved and the point of intersection between cutter profile and oil hole.

12. The Involute Profile of the shaving cutter tooth is not always a true involute. Very often, it must be modified to produce the desired involute form or modifications in the profile of the gears being shaved.

Handling Shaving Cutters: Gear shaving cutters should receive the best of care in transit to the customer, in handling by the customer, and in shipping back to the supplier for regrinding. They are precision tools that can easily be damaged by nicking or their useful life reduced by corrosion.

Rotary gear shaving cutters are usually packed by the supplier in special containers. They are greased, wrapped, and tightly wedged in the containers. When a small-quantity user receives the cutters, he should keep them in the containers. If the customer carries a large stock of cutters and has many cutters in process, he can take them out of the containers and store them in wooden storage bins with individual pockets for each cutter.

The cutters should never touch each other nor should they come in contact with metal shelving. If stored in wooden bins, the cutters should be thoroughly greased to prevent rust or corrosion of all surfaces including the hole. When returned to the supplier for resharpening, they should be repacked in the manner and in the containers in which they were received.

Gear shaving cutters should be stored in a room at moderate temperatures from 70 to 80F. They should not be stored in an unheated area in the winter or too close to an outside window. Neither should they be stored over a radiator or steam pipes or near magnetic sources. Similar considerations should be given to summer storage.

Shaving cutters should be thoroughly cleaned and checked for magnetism before installation on the shaving machine.

Sharpening Shaving Cutters: The shaving cutter, like other tools, dulls with use. To determine when to sharpen and to tell when a cutter is dull are important factors in the useful life expectancy of the tool. Methods proven successful are:

1. Graph lines on involute charts of the shaved gears show when deviations start, usually from the cutter becoming dull.
2. Periodic examination of cutter teeth.
3. Set a predetermined number of gears that may be shaved and then remove the cutter for sharpening. Experience will establish the number of parts the cutter may shave before it starts to dull.

In sharpening, minimum stock is removed from tooth faces. With normal dullness the resharpening operation usually reduces the tooth thickness approximately 0.005-in. An excessively dull or damaged tool must be ground until all traces of dullness or damage are removed. If cutter teeth are broken, the supplier can often insert new

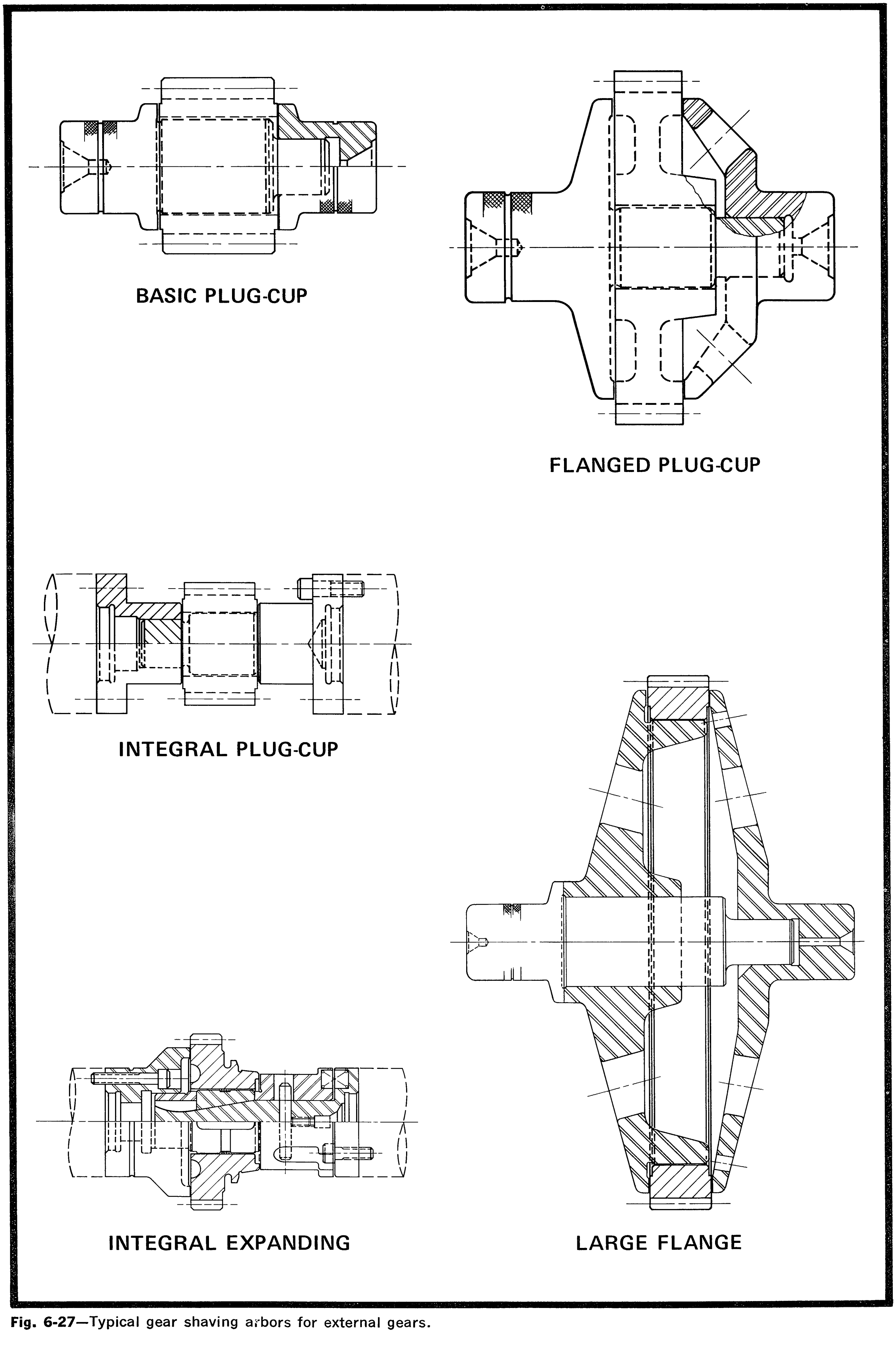


Fig. 6-27—Typical gear shaving arbors for external gears.

ones, thereby adding considerable life to the cutter. Each cutter is demagnetized after sharpening.

The number of sharpenings varies with pitch and available depth of serrations. Usually a cutter can be sharpened until the depth of serrations on the cutter teeth has been reduced to approximately 0.012 to 0.015-inch.

New cutters are fully inspected and often tested by shaving a gear before leaving the supplier's plant. A complete record by serial number, dates, size and special characteristics, is kept on each cutter.

Setting Up the Machine

Mounting the Work Gear: The work gear should be shaved from the same locating points or surfaces used in the preshaving operation. Locating faces should be clean, parallel and square with the gear hole. Gears with splined holes may be located from major diameter, side of splined teeth or minor diameter. When shaved from centers, the true center angle should be qualified and the surfaces should be free of nicks, scale and burrs.

Locating points on work arbors and fixtures should be held within a tolerance of 0.0002-in. The arbor should fit the gear hole snugly. Head and tailstock centers should run within 0.0002-in.

For most dependable results, gears should be shaved from their own centers whenever possible. If this is not possible, rigid, hardened and ground arbors having large safety centers should be used, Fig. 6-27. Locating faces should be the same as those used in hobbing or shaper cutting.

Integral tooling, Fig. 6-27, is another method which is becoming popular, especially in high production shops. This consists of hardened and ground plugs instead of centers, on the head and tailstocks. These plugs are easily detached and replaced when necessary. They locate in the bore and against the faces of the gear. It is, therefore, essential that the gear faces be square and bore tolerance held to assure a good slip fit on the plugs.

Mounting The Cutter: The greatest care is required in handling the cutter to avoid any accidental contacts between its teeth and other hard objects. The slightest bump may nick a tooth. Until the cutter is placed on its spindle it should lie flat and away from other objects.

Cutter spindle and spacers should be thoroughly cleaned and the spindle checked before the cutter is mounted. The spindle should run within 0.0002-in. on the O.D. and 0.0002-in. on the flange, full indicator reading.

The cutter is designed for a sliding fit on its spindle. If, in mounting, the fit seems too tight, the key should be examined for burrs. A hammer should never be used to seat the cutter.

After mounting, the cutter face should be indicated to check mounting accuracy. Face runout should not exceed 0.0008-in. for a 12-in. cutter, 0.0006-in. for a 9-in. cutter or 0.0004-in. for a 7-in. cutter.

Before a newly mounted cutter contacts a work gear it should be cycled several times to assure thorough wetting by the coolant. This will prevent the possibility of galling.

Feeds and Speeds: Shaving cutter spindle speeds will vary with the gear material, hardness, finish and size of part. Normally, when using a 7-in. cutter on a 10 pitch gear having a 3-in. pitch diameter, spindle speed will be approximately 200 rpm; or, using a 9-in. cutter, 160 rpm. This speed figured on the pitch circle is close to 400 surface feet per minute and this generally produces good results.

The following are formulas for determining cutter and gear speeds (rpm):

$$\text{Cutter Rpm} = \frac{\text{Desired Surface Ft. Per. Minute}}{\frac{\text{Cutter Diameter (In.)} \times \pi}{12}}$$

$$\text{Gear Rpm} = \text{Cutter Rpm} \times \frac{\text{No. of Teeth in Cutter}}{\text{No. of Teeth in Gear}}$$

For conventional shaving, about 0.010-in. per revolution of gear is considered a good starting point and becomes a factor in the following formula:

$$\text{Table Feed (ipm)} = 0.010 \times \text{Gear Rpm}$$

For Diagonal shaving, an "Effective Feed Rate" of approximately 0.040-in. per revolution of gear is considered a good starting point. Effective feed rate is the speed at which the point of crossed-axes migrates across the face of the gear and shaving cutter.

The following is the formula for determining the table traverse rate (ipm) to produce 0.040-in. effective feed rate:

$$\text{Table Traverse Rate (ipm)} = \frac{0.040 \times \text{Gear Rpm}}{R_f}$$

$$\text{Where } R_f = \frac{\text{Sine Traverse Angle}}{\text{Tangent Crossed-Axes Angle}} + \text{Cosine of Traverse Angle}$$

These suggested feed rates may be varied depending on individual operating conditions.

If higher production is desired, the table feed can be increased, but this may result in some sacrifice in the quality of tooth finish.

Where surface finish is very important as with aviation and marine gears, table feeds are reduced below the amounts indicated. In some cases, notably large tractor applications, feeds considerably in excess of those indicated are used.

Vertical table feed, or the increment by which center distance between cutter and work gear is progressively decreased during the shaving cycle,

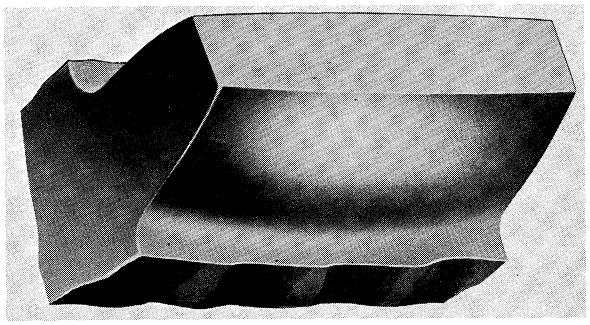


Fig. 6-28—The Elliptoid (crowned) surface as produced by rotary gear shaving.

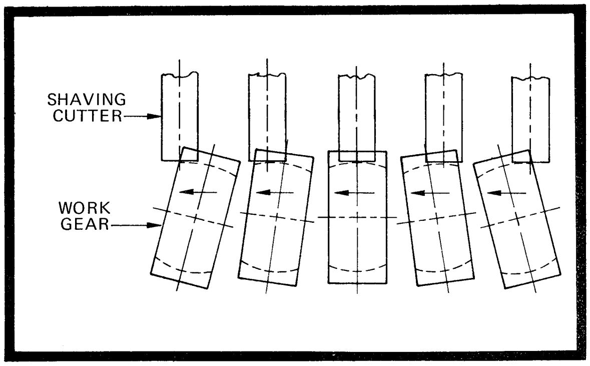


Fig. 6-29—Rocking table action with conventional shaving shows zone of cutting through center of cutter.

is selected in accordance with the type of shaving being used. Vertical up-feed is always used for conventional shaving and usually for Diagonal shaving.

Coolants: It is very important to use the proper cutting oil or coolant for gear shaving, the basic essentials of which will be found in the following:

For steels, use a sulphur base oil having a sulphur content of 3 to 3.5-percent.

For bronze, cast iron and aluminum use a mixture of 8 parts of kerosene and one part of light machine oil. Some types of quenching and honing oils are also satisfactory.

For plastics, known under various trade names such as Celeron and Micarta, use a water-soluble oil mixture of approximately 1 part oil and 20 parts water.

The type of coolant and the degree of its contamination directly affect cutter service life and the finish of the shaved tooth surfaces. Avoid the use of a cutting oil that is too thin as this will cause chip scratch on gear teeth faces. Chip baskets should be cleaned periodically if a supplementary filter is not used.

The cutting oil should have a viscosity of about 135 S.S.U. at 100F. A magnetic chip separator in the coolant circuit will help reduce contamination.

Crowning on Gear Shavers: The Elliptoid tooth form, Fig. 6-28 is produced on Red Ring Shaving Machines with conventional shaving by rocking the work table as it is reciprocated, Fig. 6-29. This rocking table motion causes the cutter teeth to sink deeper into the work gear teeth at their ends than it does in the middle, thus thinning the teeth progressively toward their ends. Both the point of maximum tooth thickness and the amount of crowning are readily controlled.

With Diagonal shaving and traverse angles up to 60-deg., a rocking table motion in the diagonal plane permits a cutter with straight teeth to produce a crowned tooth on the work gear.

When traverse angles are from 60 to 90-deg. a cutter with reverse-crowned teeth is utilized to provide the crowned tooth form.

Shaving Internal Gears

Internal gears can be shaved on special machines in which the work drives the cutter, Fig. 6-30, or by internal cutter head attachments on external shaving machines, Chapter 10. Typical air and hand-operated pot chucks for holding internal work gears are shown in Fig. 6-31.

Because of the crossed-axes relationship between the cutter and the work gear in internal

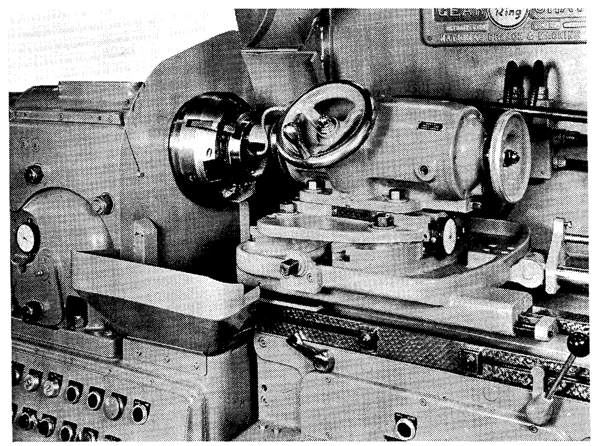


Fig. 6-30—Shaving of an internal gear on a special machine in which the work drives the cutter.

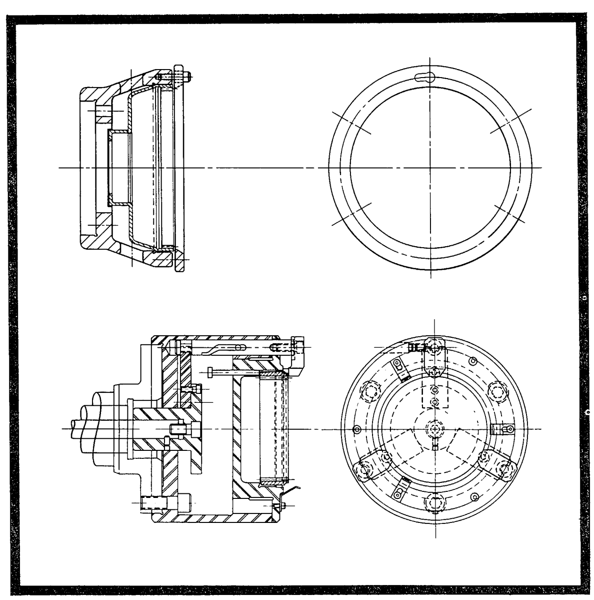


Fig. 6-31—Typical work holding fixtures for shaving internal gears on a rotary gear shaver.

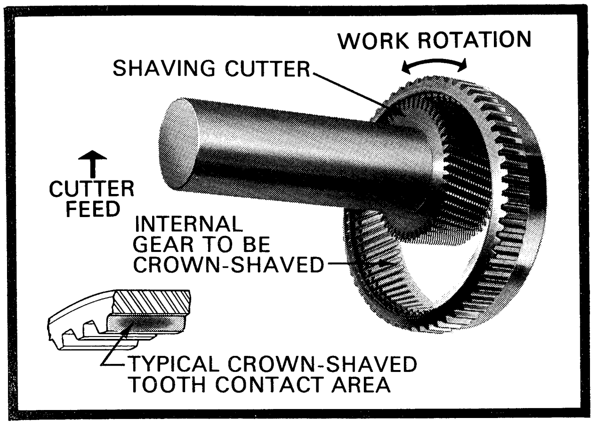


Fig. 6-32—Motions of internal work gear and rotary shaving cutter for plunge-feed, crown-shaving.

shaving, the cutter requires a slight amount of crown in the teeth to avoid interference with the work gear teeth.

Crowning of the teeth on gears over ¾-in. wide is best achieved by a rocking action of the work head similar to the rocking table action with external gear shaving.

When internal gears are ¾-in. wide and under, or shoulder interferences limit the work reciprocation and crossed-axes angle, a plunge-shaving method can be applied. Here the cutter is provided with differential serrations and plunge-fed upward into the work, Fig. 6-32. If crowning is desired, a reverse-crowned cutter is used with the plunge-feed shaving process.

Loading Methods

Three methods are available for loading work gears into gear shavers. The first is manual loading in which the operator places the work gear between centers or on an arbor in mesh with the cutter, hand-clamps the tailstock, closes the coolant doors and initiates the shaving work cycle.

The second and faster method is called semi-automatic loading, and applies an approximate locator, power tailstock and cylinder-controlled coolant doors. Thus the operator only has to place the work on the locator in mesh with the cutter and push a button to initiate the balance of the cycle that clamps the gear, closes the coolant door and starts the shaving sequence. At the end of the semi-automatic work cycle, he merely unloads the finish-shaved work from the approximate locator.

With automatic loaders (the fastest loading method), shaving machines can run continuously as long as the magazines are supplied with gears. One unskilled operator can easily keep a battery of automatic-loaded machines filled. Or they can be filled automatically from a factory conveyor system.

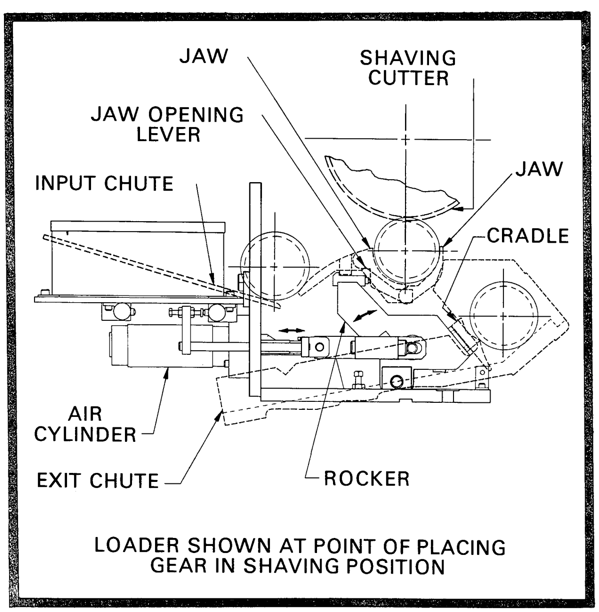


Fig. 6-33—Basic type of rocker-type automatic loader for rotary gear shavers.

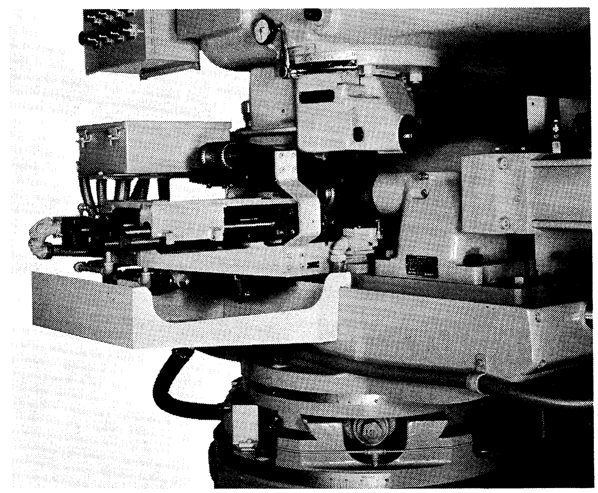


Fig. 6-34—Basic type of slide-type automatic loader for rotary gear shavers.

Automatic loaders operate as follows: At the start of the continuous cycle, the head and tailstocks are retracted. The loader picks up a work gear from the end of the magazine and advances it into mesh with the cutter. The tailstock advances and clamps the work gear in shaving position. The loader moves back to the magazine, picks up another work gear, and the shaving cycle begins.

When shaving is completed, the tailstock automatically retracts at the end of the shaving cycle, lowering the gear onto rails that carry it out of the work area to the front of the machine. Then a new cycle starts automatically.

The magazine that feeds the gears to the loader can be equipped with a pair of master gears through which gears to be shaved are passed. Oversized or incomplete gears are thus prevented from entering the loader.

The loader, and tailstock are air-operated and designed so that they will remain positively locked in case of any air line failure during the work cycle. Actual time for loading and unloading of a gear by an automatic loader on a shaving machine is about two seconds.

For internal gears, the automatic loading principles are somewhat different than those described here.

Automatic loaders have been successfully applied on gear shavers to handle all types of internal and external automotive and truck transmission and engine gears, including cluster gears, stem gears, camshaft gears and timing gears.

The design of automatic loaders varies with the type of gear to be shaved and the overall blank dimensions.

Two general types of automatic loaders are normally applied on gear shavers. These include the rocker type, Fig. 6-33, and the slide type, Fig. 6-34.

Shaving Conical Involute Gears

Conical involute spur or helical gears (tapered-tooth gears that mesh on non-parallel axes) can be shaved by the high speed Diagonal method, Fig. 6-35. Here a steering gear segment is mounted on an arbor with steel dummy teeth. The cutter and work gear are set at a slight crossed axes relationship for the shaving operation in which

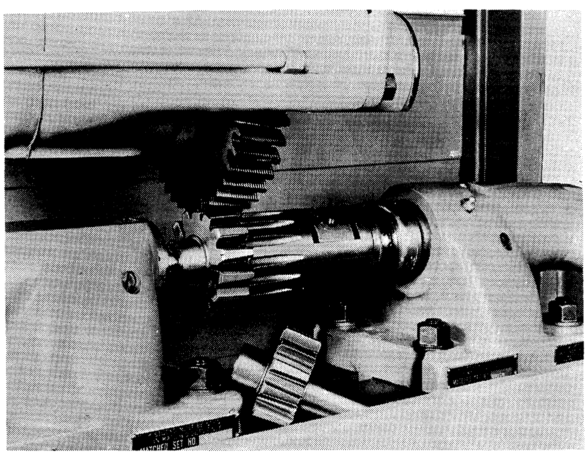


Fig. 6-35—Shaving a conical involute sector gear segment.

0.004-in. of stock is removed from the tooth thickness by the shaving operation. The finished segment work gear will mesh with a 5-DP rack that has 20-deg. pressure angle teeth.

The special 9-in. dia. 1⅜-in. wide shaving cutter has conical involute teeth. Such cutters can also be applied to shave conical involute marine reduction gears that are mounted on non-parallel shafts.

Gear Roll-Finishing

In this discussion of gear roll-finishing, particular attention is called to the special tooth nomenclature resulting from the interaction between the rolling die teeth and the gear teeth. To eliminate confusion, the side of a gear tooth that is in contact with the

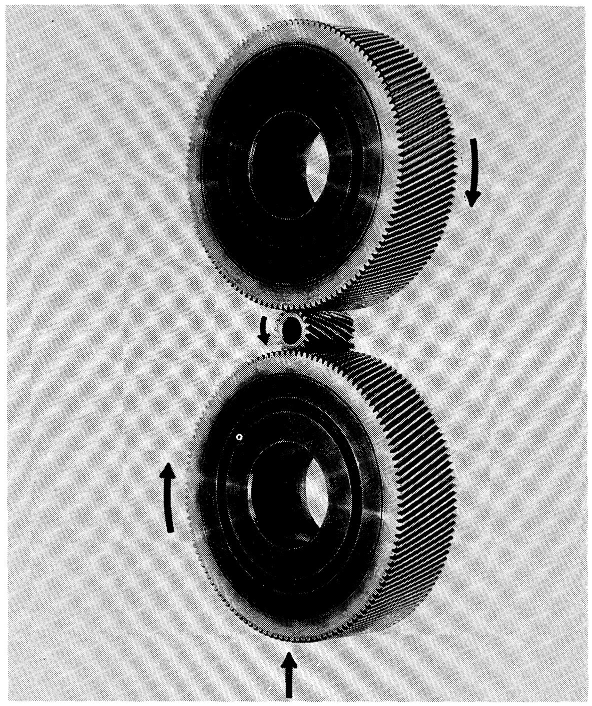


Fig. 6-36—Operating principle of double-die gear roll-finishing.

"approach" side of a rolling die tooth is also considered to be the approach side. The same holds true for the "trail" side. Thus, the side of the gear tooth that is in contact with the trail side of a rolling die is also considered to be the trail side.

Gear roll-finishing, Fig. 6-36, is much different from gear shaving in that a flow of material is involved, rather than a removal of material. A study of gear tooth action is required to analyze the material flow in the rolling process. In Fig. 6-37 it can be seen that as a gear rolling die tooth engages the approach side of a workpiece tooth, sliding action occurs along the line of action in the arc of approach in a direction from the top of the gear tooth toward the pitch point where instantaneous rolling action is achieved. As soon as the contact leaves the pitch point, sliding action occurs again, but in the opposite direction toward the pitch point in the arc of recession.

What is more interesting, however, is that the

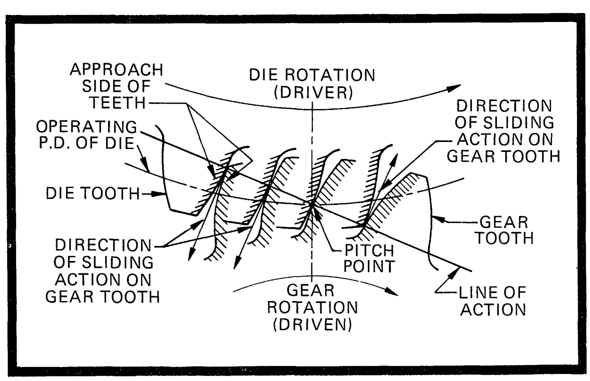


Fig. 6-37—Contact action between one tooth of a workpiece and the approach side of a rolling die tooth.

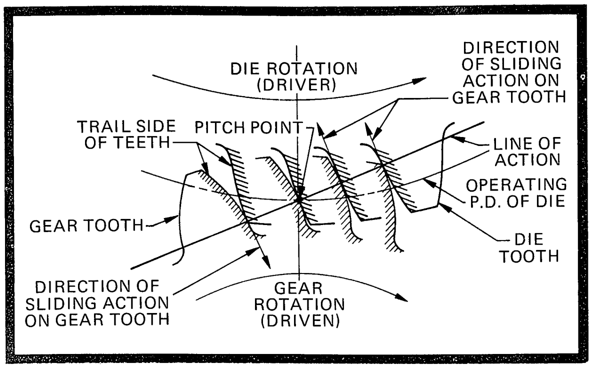


Fig. 6-38—Contact action between the tooth of a workpiece and the trail side of a rolling die tooth.

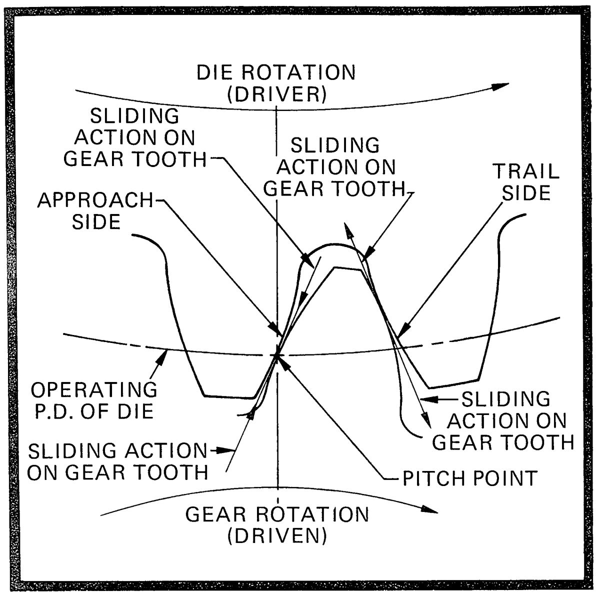


Fig. 6-39—Differing flow directions induced by each side of a die tooth with gear roll-finishing.

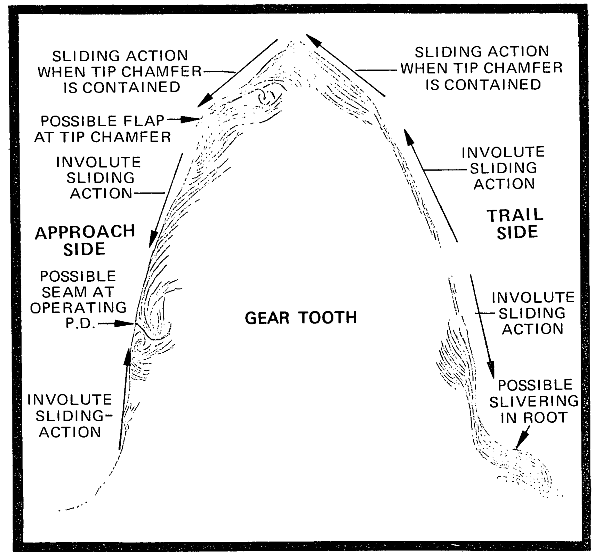


Fig. 6-40—Tooth flow pattern that results when too much stock is left for roll-finishing, or when material hardness is excessive.

contact between the die and work gear teeth on the trail side, Fig. 6-38, produces exactly the opposite direction of sliding to that on the approach side. The result of these changing directions of sliding is that material is being compressed toward the pitch point on the approach side and extended away from the pitch point on the trail side, Fig. 6-39.

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

To be successful, it appears that for roll-finishing an undercut is desirable near the root section like conventional pre-shave tooth forms. Since most production gears are also provided with a tip chamfer, the material will tend to be pulled up into the chamfer on the trail side and down away from the chamfer on the approach side.

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

If too much stock is left for gear roll-finishing, or if the gear material tends to be too hard (above approximately $20R_C$), several conditions may result. The sliding action on the approach side of the tooth may cause a "seaming" of material that builds-up in the area of the pitch point. On the

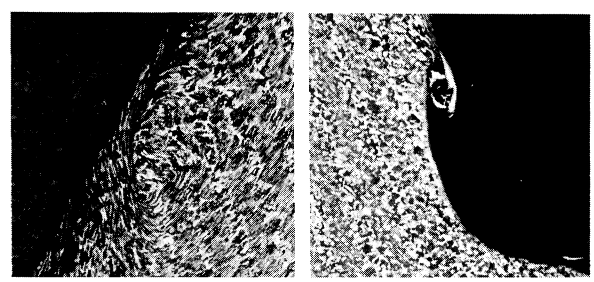


Fig. 6-41—Photomicrographs of a gear tooth with high hardness. The approach side, left, has a seam in the area of the pitch diameter. The trail side, right, shows where excessive stock has caused cold-working and a sliver near the root.

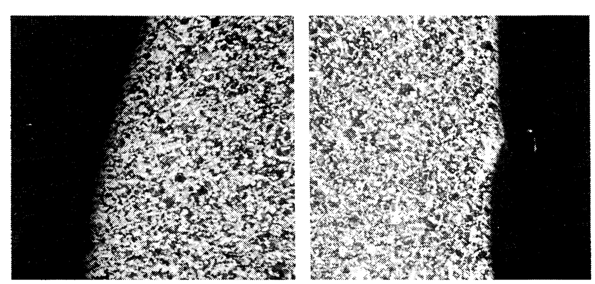


Fig. 6-42—Photomicrographs of a properly roll-finished gear tooth. The approach side, left, has no seaming. The trail side, right, shows no slivering or cold-working.

trail side, the flow of excess material may result in a burr on the tip of the gear tooth and a "slivering" of material into the root area. Figure 6-40 shows the condition of a roll-finished gear tooth when too much stock is flowed, or high hardness conditions are encountered.

In Fig. 6-41, photomicrographs show the conditions encountered when stock removal is excessive and material hardness is too high. A seam is evident in the approach side of the tooth at the left in the area of the operating pitch diameter. The trail side photomicrograph at the right in Fig. 6-41 shows slivering in the root portion with about 0.004-in. of lapped-over metal, and about 0.002-in. deep surface cold-working of the material.

In contrast, photomicrographs in Fig. 6-42, show the excellent tooth structure that can be achieved with roll-finishing if stock reduction is held to a minimum and the material is not too hard. No evidence of cold-working or seaming is seen in the approach side at the left. In the trail side at the right in Fig. 6-42, no evidence of slivering or cold-working is seen.

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

Table 6-6—Tolerances for Gear Roll-Finishing Dies

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

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

Gear Rolling Dies

Since roll-finishing involves material flow rather than metal removal it should be expected that the tooth form on the die would not be faithfully reproduced on the workpiece tooth due to minute material springback and material flow conditions.

Even with gear shaving it has been found necessary to modify the shaving cutter teeth profiles somewhat to produce a desired form on the work gear teeth. Experience to date has shown that a

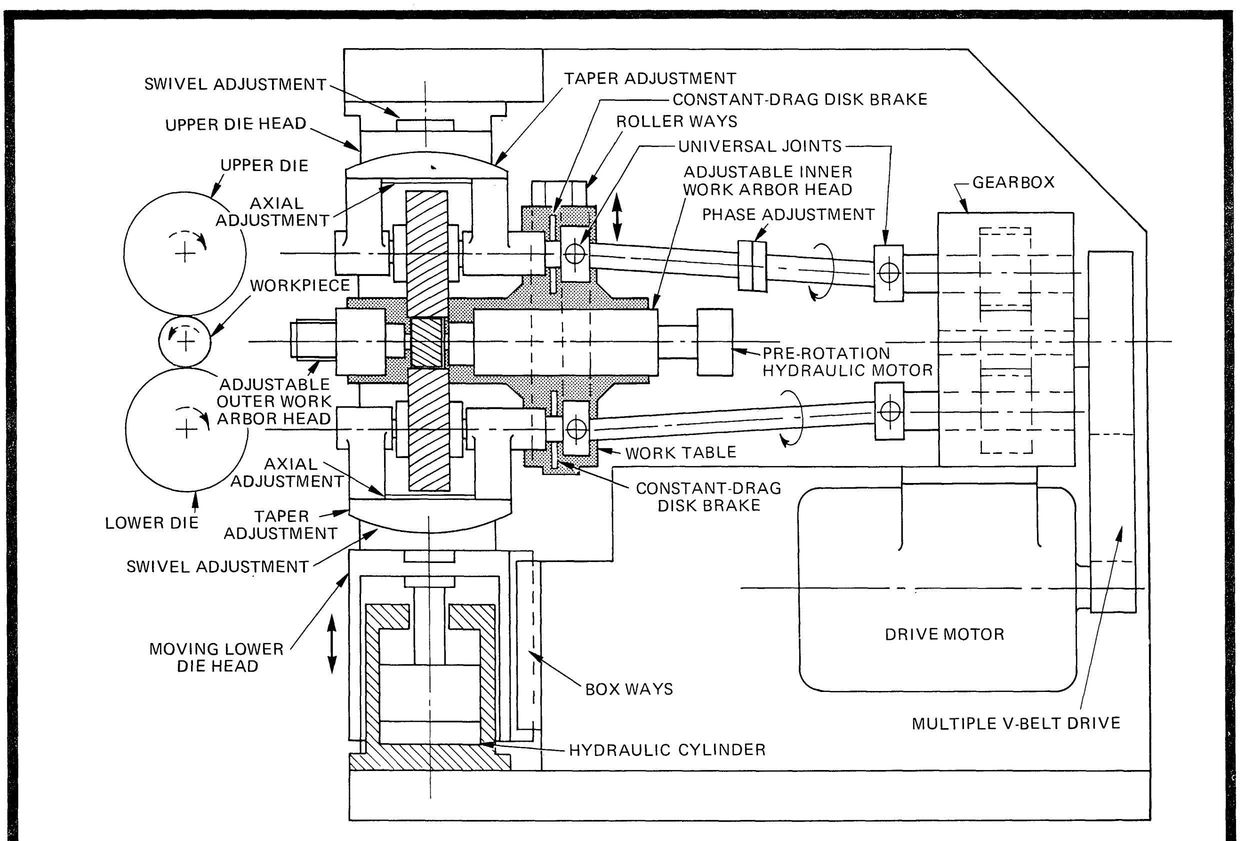


Fig. 6-43—Operating components of a double-die gear roll-finishing machine.

different type of tooth form modification is required for gear rolling dies than for gear shaving cutters. The correct amount of gear rolling die tooth form modification, as with gear shaving cutters, is determined from an extensive development program. Less-rigid gear roll-finishing machines usually require greater and varying die form modifications.

Gear rolling dies are made from a special fatigue and impact-resistant high speed steel to the tolerances shown in Table 6-6.

Gear Rolling Machines

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

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

Double-Die Gear Rolling: The Red Ring double-die GeaRoll machine shown in Fig. 6-43 and Chapter 10, is a vertical design with the dies mounted one above the other. Such a design provides maximum rigidity, requires minimum floor space and also gives maximum accessibility for a hinged automatic work loader as well as die head positioning adjustments.

It is designed to handle gears up to 4-in. wide and 6-in. diameter. Die speeds are from 40 to 160-rpm. Dies up to 4½-in. wide and 9⅝-in. diameter can be mounted in the machine.

The welded steel frame has 2-in. thick steel side members that will withstand forces up to 100,000 pounds. This large frame force capacity was built-in to utilize the machine in future roll-forming operations in which gear teeth are expected to be rough-formed from solid steel blanks.

Phasing of the two die heads, so that the teeth are in proper timing with the teeth on the workpiece, is accomplished by moving and locking a phase adjuster in one of the two die head drive-shafts.

The upper die head is fixed and the lower die is fed upward by a hydraulic cylinder. The work gears are fed into rolling position by a magazine-fed, slide-type shuttle, air-operated automatic loader. Here they are picked up on a work arbor that is advanced by a hydraulic rotary actuator utilizing a gear rack arrangement. The work gears are advanced against a pneumatically-loaded cup.

The work arbor is pre-rotated by a hydraulic motor at a speed slightly slower or faster than the

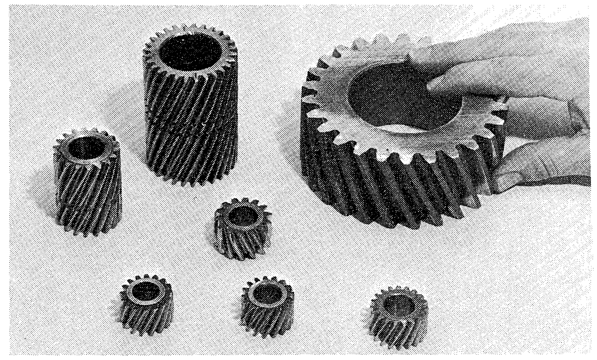


Fig. 6-44—Gears that have been successfully roll-finished.

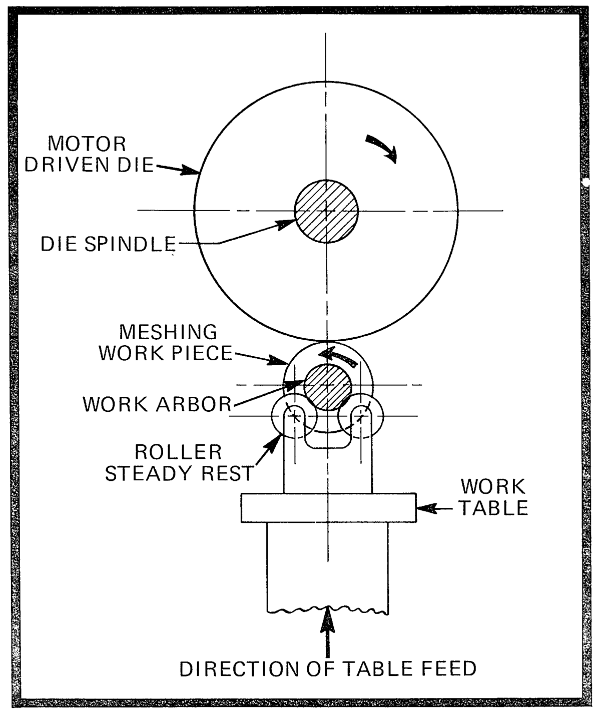


Fig. 6-45—Operating principle of single-die gear roll-finishing.

Table 6-7—Data on Roll-Finished Gears

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

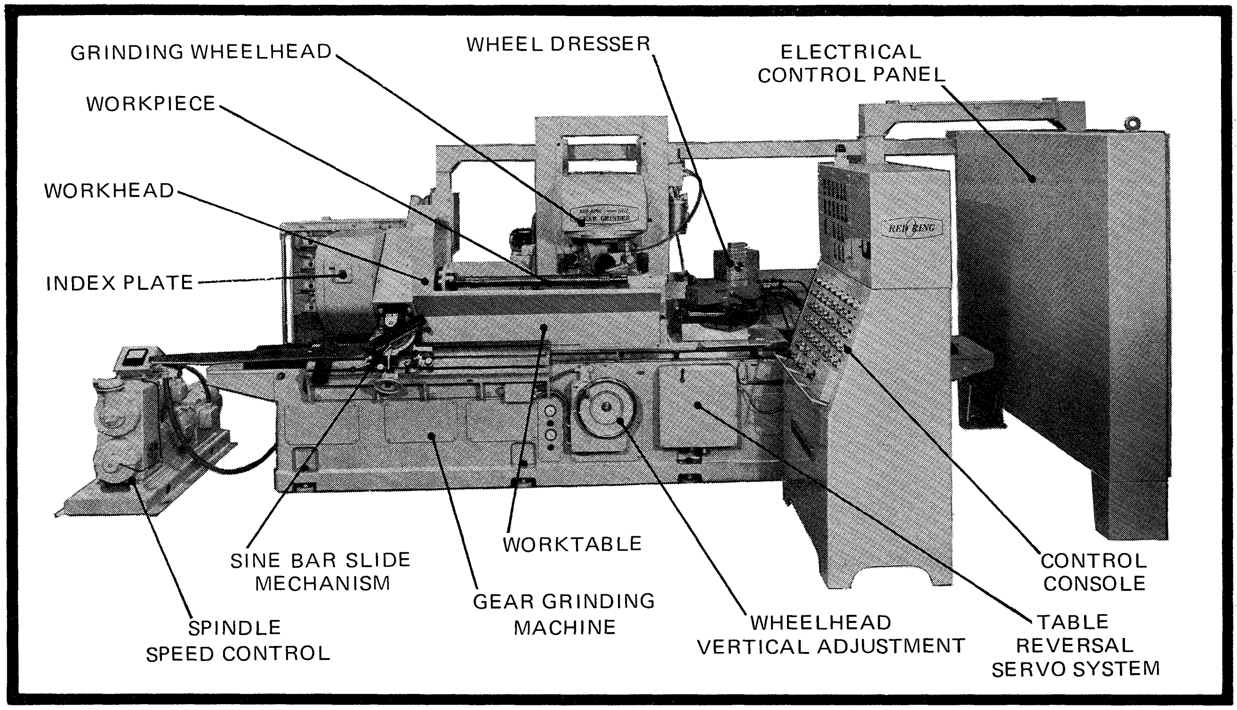


Fig. 6-46—A gear form-grinding center with major operating components indicated.

die speed to ensure clash-free engagement. The lower die then feeds upward to a predetermined operating position to control finish-rolled gear size.

Independent adjustments are provided for taper, face alignment and swivel positioning of each of the die heads. Crowning is produced by a reverse-crown that is form-ground in the dies.

Table 6-7 and Fig. 6-44, illustrate the range of gearing for which gear rolling dies have been produced for finish-rolling production applications.

Single-Die Gear Rolling: Red Ring Rollshave and Uniroll finishing machines (Chapter 10) use a single-die gear rolling process that is ideally adapted for low and medium-production finishing operations where both roll-finishing and shaving operations, or roll-finishing only are done economically.

A single gear rolling die is mounted in a heavy-duty gear head above the workpiece, Fig. 6-45. The die is driven by an electric motor to provide rotation of the workpiece that meshes with it. Normally semi-automatic loading methods are utilized on single-die roll finishing machines whose work cycles are somewhat longer than those of the fully-automatic, double-die machines.

The workpiece is mounted on an arbor between head and tailstocks. In operation, the table supporting the head and tailstock is fed upward by a unique, air-powered, heavy-duty radial feed system. The continuous upfeed of the table provides the large force necessary to roll-finish the gear teeth.

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

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

Form Grinding

Form grinding is one of the oldest methods of finishing the teeth on hardened gears. The grinding wheel is dressed to a precision form and the work is passed back and forth under the grinding wheel. The work is indexed from space-to-space and the wheelhead is fed down in decreasing increments. A spark-out operation completes the sizing and finishing of the tooth.

The traditional form-grinding machine has required the ultimate in operator skill. If too much stock is removed too quickly, gear tooth accuracy and surface quality are impaired. Or, if there is too much distortion, a case-hardened type of tooth surface may have a portion of its hardened surface removed before the teeth are cleaned up and finish size is reached.

Since tooth grinding operations are normally performed on gears to provide the highest possible dimensional precision and finish quality, the scrapping of any gear at this point is a most expensive and critical consideration.

As a result of these critical quality, cost and operator skill requirements, the Red Ring form-grinding machine today has reached a form of sophisticated automatic control whereby such machine tools are commonly designated as

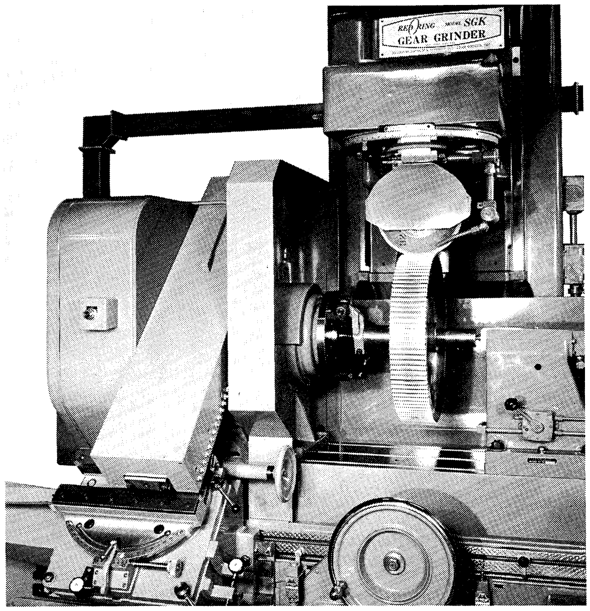


Fig. 6-47—Grinding the teeth in a large spur gear.

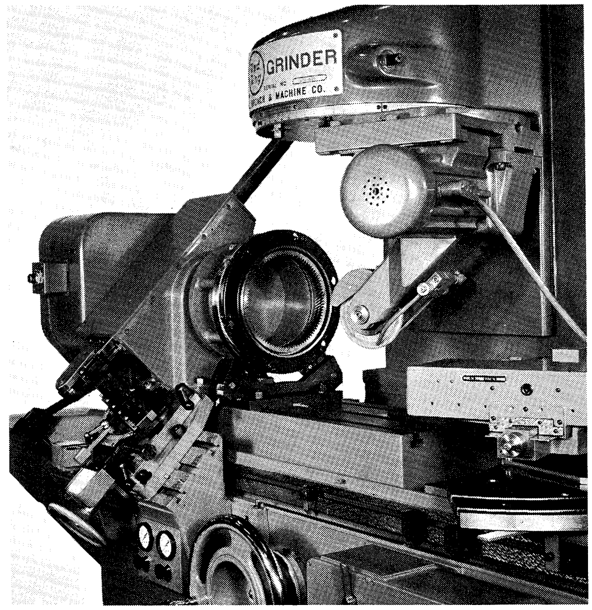


Fig. 6-48—Grinding the teeth on an internal helical gear.

'grinding centers', Fig. 6-46.

A wide variety of hardened work gears can be ground in these form-grinding machining centers including external spur gears, Fig. 6-47; external helical gears; internal spur gears; internal helical gears, Fig. 6-48; or shoulder gears.

Several basic requirements have to be met in a precision form-grinding machine before attention is given to the automation of the work cycle. First a precision indexing system is required in which the index plate is of considerably larger diameter than the work gear, and made to even more precise tolerances than those specified for the work piece.

The second important machine requirement is the application of an adjustable, accurate means of rotating the work gear to provide accurate leads on helical gears, Fig. 6-49.

A diamond dresser that will dress super-precision true or modified involute forms in either a manual or automatic sequence, Fig. 6-50, is a basic requirement for accurate form-ground gears.

An automatic control system is the final key link in the automating of a precision form-grinding sequence.

Typical Applications of Grinding Centers

Long-Shaft Helical Gears: The grinding center illustrated in Fig. 6-50 has an automatic control system in which "dial-in" controls are provided for all phases of a tooth grinding cycle including index; length and position of grinding stroke, roughing, semi-finishing and finishing grinding phases, sparkout, number of feed increments, frequency of wheel dressing, amount of feed for individual roughing, semi-finishing and finishing operations, and infinite grinding wheel speed adjustments over a 3 to 1 range. Wheel down-feed

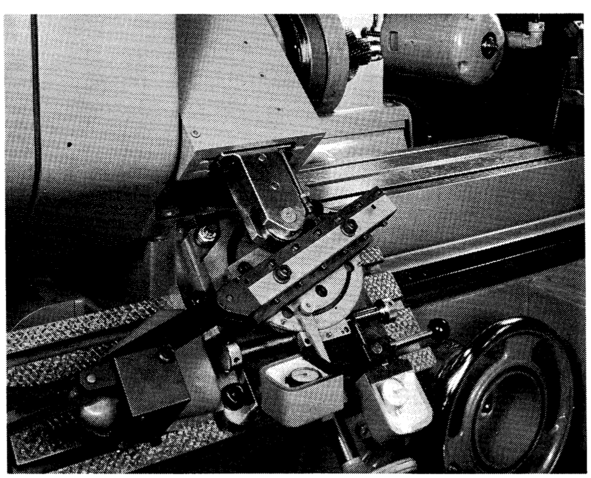


Fig. 6-49—Sine bar mechanism for rotating the work gear in timed relationship with the table stroke to produce helical lead on a shoulder gear.

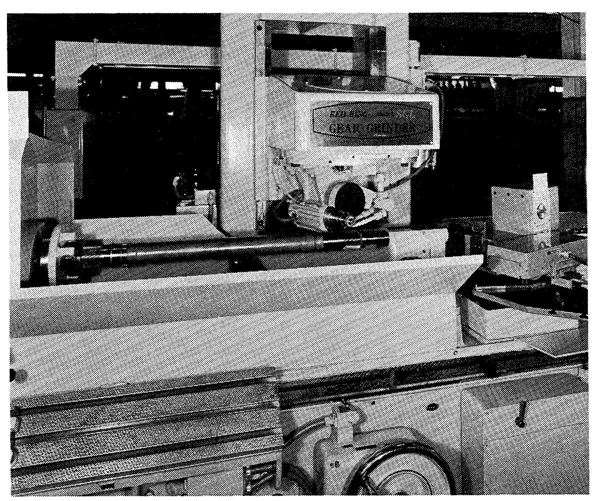


Fig. 6-50—Grinding a long-shaft helical gear in a grinding center. The automatic wheel dresser is at the right.

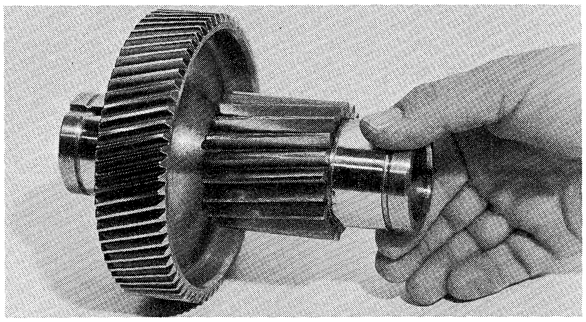


Fig. 6-51—A shoulder gear for a jet engine reduction set whose smaller helical gear is ground, Fig. 6-49.

increments are adjustable from 0.00025-in. to 0.00225-inch. Total downfeed range is 0.090-inch.

The automatic wheel dresser has dial-in controls for number of dress strokes; number of dress feeds; amount of feed for individual roughing, semi-finish and finish grinding; and also provides for dwell strokes without feed, as well as individual dress speeds for roughing and finishing.

Four grinding phases are selective 'in or out' of the grinding cycle. These include roughing, semi-finish, finish and sparkout. In roughing, multiple downfeeds and table strokes between indexes are possible. In semi-finish and finish, the grinding wheel downfeeds once for each revolution of the work gear. In sparkout, downfeeds are eliminated.

The part being ground on this machine, Fig. 6-50, is a 39½-in. long steel shaft on which helical gears are ground on each end. The hardened gears have 42, 16/32-D.P. teeth with a 2.8-in. O.D.; and 18-deg., 58-min. L.H. helix angle. Tolerances are: 0.0024-in. size over pins; 0.0005-in. involute profile; 0.001-in. accumulative spacing; and 0.0005-in. lead.

Shoulder Gears: The gear illustrated in Figs. 6-49 and 6-51 is a shoulder-type planet helical gear for a jet engine reduction set. The smaller gear on the blank has 20, 2-in. pitch diameter, 10.0685-pitch, 20-deg pressure angle teeth with a 6-deg, 54-min. and 28-sec. helix angle. Grinding of the smaller helical gear, which is closely restricted by the rim of the larger 67-tooth helical gear on the blank is performed with a 1¼-in. dia. grinding wheel that rotates at 20,000-revolutions per minute.

Tolerance on the gear, which is held in radial relationship with the larger gear to within 0.0006-in., is 0.0037-in. over pins for size, 0.0002-in. for spacing and 0.0002-in. per in. of the 2-in. length for lead. The grinding center is capable of producing teeth with even closer tolerances, where desired.

Complex Gear Sections: Grinding centers are now built with sufficient control flexibility that super-precision parts formerly impossible to produce accurately by any grinding method can be economically produced.

Such a typical part is shown in Fig. 6-52. This cam nut is a 9.660-in. long, 4.875-in. O.D. hardened steel tubular part having a number of complex cam tracks on one side and a portion of a gear on the other side. The gear is buried on the O.D. of the part in such a manner that it is impossible to produce it by hobbing or gear shaping.

The gear is a 14-space portion of what would be a 37-tooth, 7.6630-pitch, 25.4062-deg pressure angle, 39.9768-deg helix angle helical gear. The tooth spaces are of varying lengths and blind on both ends.

Specifications on the part drawing call for the gear teeth to be hardened and then ground to a tooth thickness tolerance of 0.002-in., a lead tolerance of plus or minus 0.0005-in. and to have a spacing accuracy of 0.0004-inch. In addition, the teeth have to be positioned in relationship with a drilled hole on the rim to within plus or minus 0.001-inch.

Roughing of the teeth is done with an involute gear cutter on a milling machine equipped with a helical milling attachment. The problem of grinding the teeth after heat treatment was one of providing a helical gear grinding machine with automatic work cycle and table stroke controls that would produce the precision cam nut gear in a minimum of time. The gear portion can be produced by varying the stroke of the table on the grinder from about 1-in. to 4¾-in. in a properly controlled sequence.

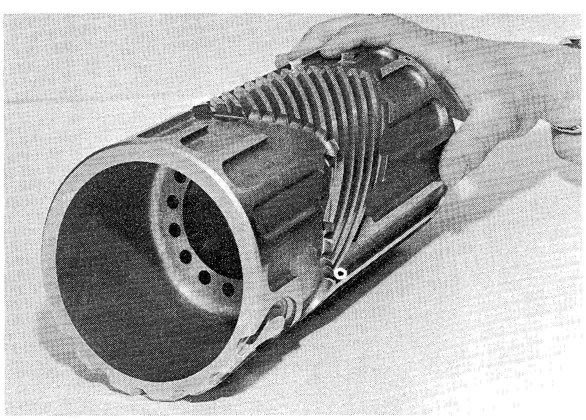


Fig. 6-52—A cam nut whose gear segment teeth are ground.

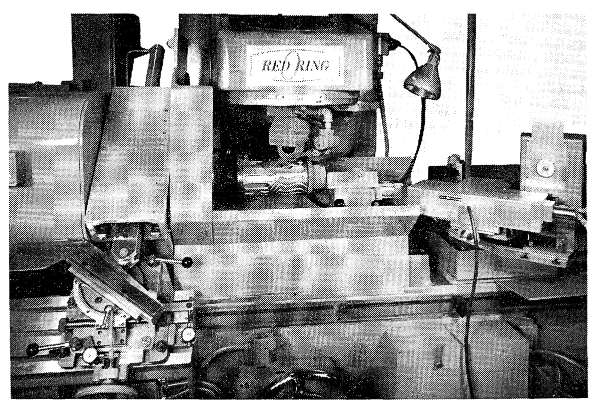


Fig. 6-53—Close-up view of cam nut gear segment automatic tooth grinding operation.

The helical gear grinder for producing the cam nut, Fig. 6-53 has an adjustable sine bar unit for generating the lead, an index-plate precision index system, a variable-speed grinding spindle drive, a semi-automatic wheel dresser, and an electronic table drive servo control system that selects in proper sequence the thirteen necessary table stroke lengths for the fourteen tooth spaces.

The control panel has the counters for the grinding functions at the top, and a series of adjustable potentiometers for controlling each end of the stroke for the thirteen required stroke sequences. Also on the control panel are a variety of push-button controls for setup, automatic cycle, and automatic preset wheel dressing functions.

The machine has the 5-in. dia grinding wheel mounted on a motorized spindle. The totally-enclosed motor is cooled by circulating coolant through its jacket. The index mechanism has an added auxiliary hydraulic cylinder-controlled rack and pinion mechanism that returns the gear to start position at the completion of the fourteen-space grinding sequences.

The cam nut is mounted on an arbor, having a pin detail that locates the part radially to maintain tooth-to-hole location. In the grinding operation, a total of 0.011-in. is removed from each side of the teeth. Once the part has been set up, the operator can initiate an automatic cycle of operation.

The first portion of the automatic grinding cycle is a roughing sequence in which a tooth-space is rough-ground in twenty-five successive 0.001-in. downfeed increments and then indexed to the next space. The wheel is automatically trimmed at the completion of seven spaces. In the next semi-finishing sequence, all of the tooth spaces are ground in the first pass without downfeed and then on a second pass with a 0.0005-in. downfeed.

In the finishing sequence, the tooth spaces are ground in one pass with a 0.0005-in. downfeed.

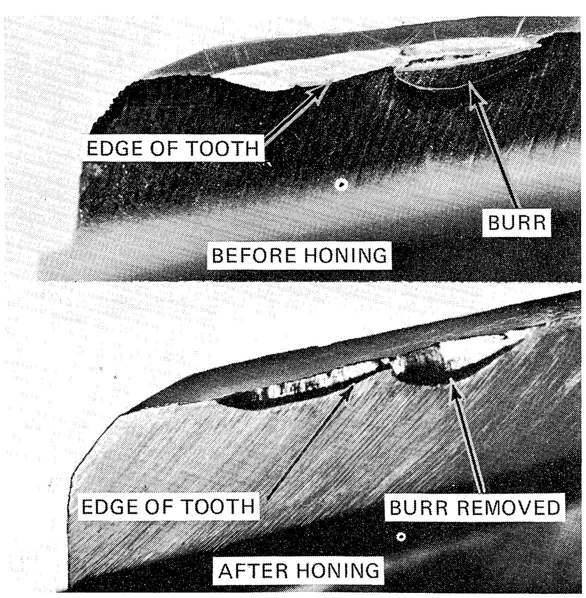


Fig. 6-54—Removal of nicks from gear teeth by the gear honing process.

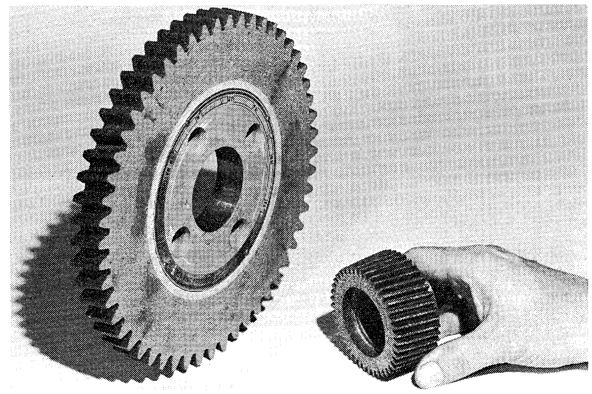


Fig. 6-55—Typical gear honing tools for external gears (left) and internal gears (right).

Total time for the automatic precision grinding operation is approximately 60-minutes. Both right and left-hand cam nut helical gear portions can be ground in this grinding center.

Gear Honing

Gear tooth honing is a hard gear finishing process that was developed to improve the sound characteristics of hardened gears by 1. Removing nicks and burrs, 2. Improving surface finish and 3. Making minor corrections in tooth irregularities caused by slight heat-treat distortion.

The process was originally developed to remove nicks and burrs, Fig. 6-54, that are often unavoidably encountered in production gears because of careless handling. Further development work with the process has shown that minor corrections in tooth irregularities and surface finish quality improvement can be achieved. These latter improvements can add significantly to the wear life and sound qualities of both shaved and ground hardened gears.

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

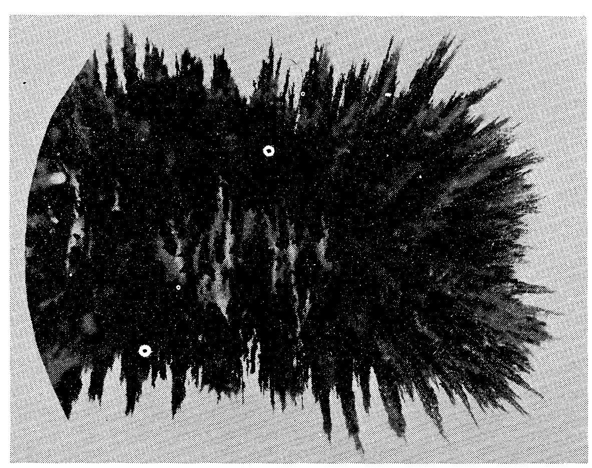


Fig. 6-56—Honing particles in a magnetic field.

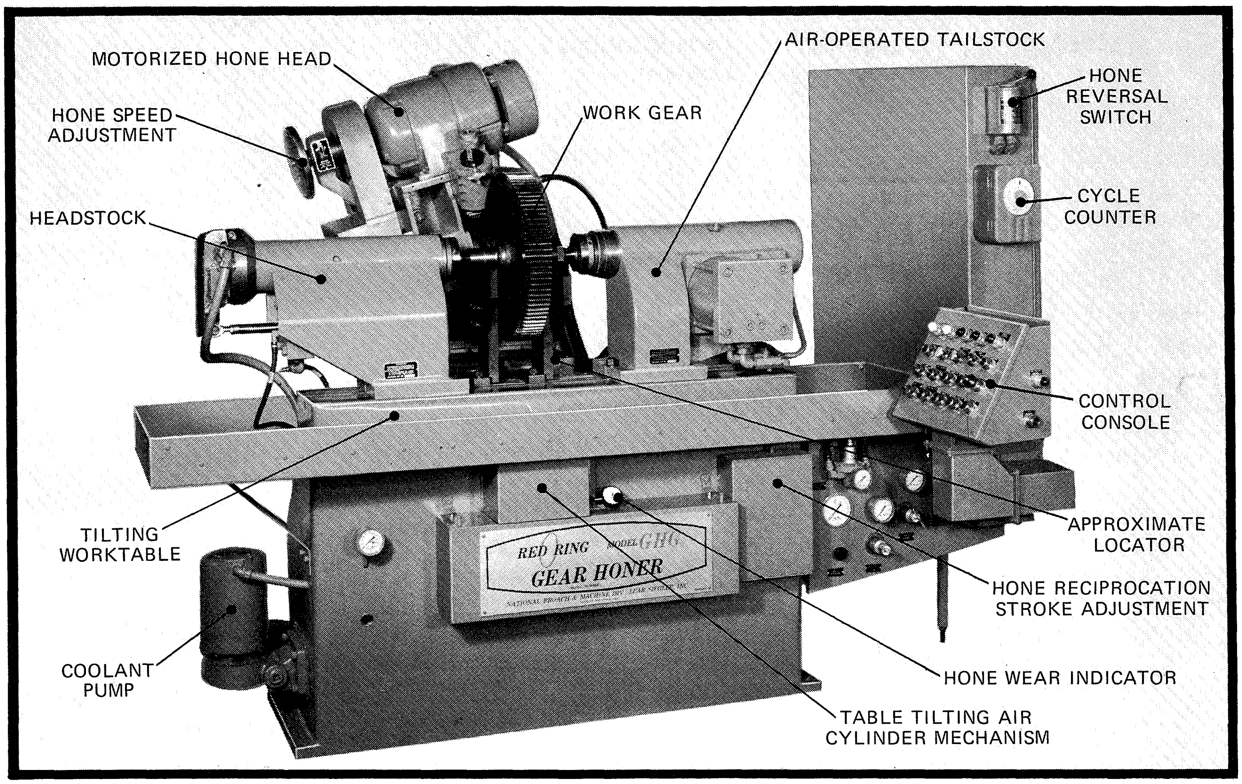


Fig. 6-57—Operating components of a 24-in. gear tooth honing machine.

terial, nor does it generate internal stresses.

Honing can be applied to both external and internal spur and helical gears utilizing a variety of specialized types of honing machine tools, Chapter 10. Both taper and crown honing operations can be carried out if desired.

How the Process Works

The process uses an abrasive-impregnated, helical gear-shaped tool Fig. 6-55. This tool is generally run in tight mesh with the hardened work gear in crossed-axes relationship under low, controlled center distance pressure.

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

The honing tool is a throw-away type that is discarded at the end of its useful life. It is not unusual for a tool to hone as many as from 5,000 to 8,000 pieces before it is replaced. Particles removed by honing are shown in Fig. 6-56. The process is carried out with conventional honing oil as a coolant.

The honing tool teeth are thinned as the tool wears. This tooth thickness reduction can continue until root or fillet interference occurs with the work gear. Then the hone O.D. can be reduced to provide proper clearance.

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

Usually the amount of stock removed from gear teeth by honing ranges from 0.0005 to 0.002-in. measured over pins.

External Gear Honing Machines

A typical 24-in. Red Ring external gear honing machine, Fig. 6-57 has the motor-driven honing tool mounted at the rear of the work spindle. The work spindle is mounted on a tilting table that can be positioned to provide four selective modes of operation.

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

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

The third and most generally applied mode of operation is called constant-pressure. The work gear is held in mesh with the honing tool at a constant pressure. This method removes nicks

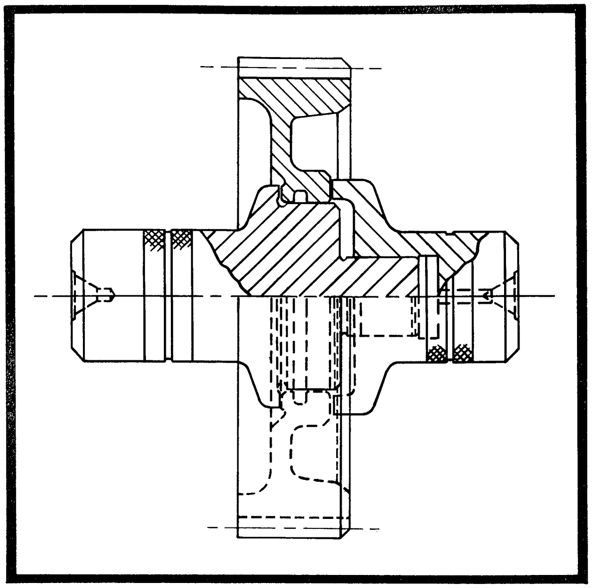


Fig. 6-58—A typical work holding arbor for honing.

and burrs and provides maximum surface finish improvement in a minimum time.

The fourth mode of operation is called differential-pressure. A pre-selected low-pressure is present between the hone and the low point of an eccentric gear; and a pre-selected increased amount of pressure is present between the hone and the high point of eccentricity. This method has all of the desirable features of the constant-pressure method plus the ability to slightly correct eccentricity.

The amount of eccentricity in the gears with differential-pressure honing may cause the hone to wear faster than the constant-pressure method.

Red Ring Model GHG machines can be equipped for manual loading (low production), semi-automatic loading (medium production), and fully-automatic loading (high production) in much the same manner as rotary gear shaving machines.

Arbors for mounting work gears on honing machines, Fig. 6-58 do not normally need to provide support out to the gear rim as is required with shaving.

External gears up to 180-in. diameter can be honed on large gear shaving machines especially equipped for honing. All four modes of honing operation are possible on these special machines.

Internal Gear Honing Machines

Internal gears from 4 to 12-in. diameter can be honed on special Model GHC honers, Figs. 6-59 and 60, in which the motorized workhead drives the honing tool. The head has a tilting arrangement that provides all four modes of operaion.

The Model GHE honing machine shown in Fig. 6-61 will hone internal gears from 6-in. to 24-in. diameter. The motor-driven work gear meshes with the honing tool in crossed-axes relationship. A pressure control mechanism is built into the hone head. This unit permits application of all four

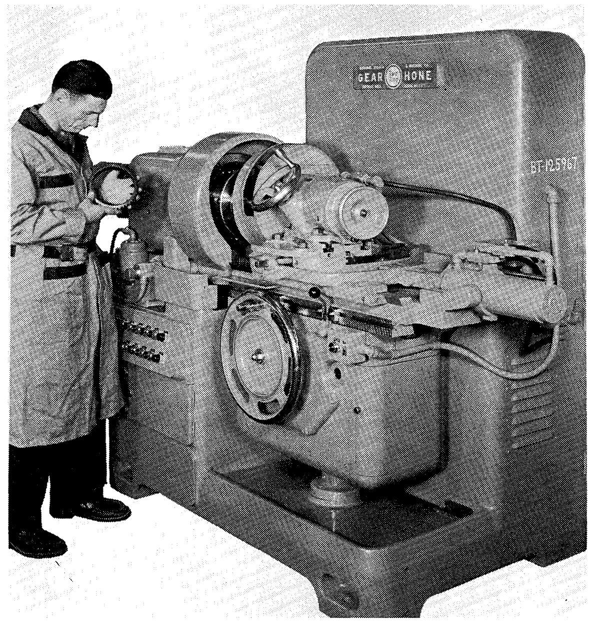


Fig. 6-59—A machine for honing internal gears up to 12-in. diameter.

modes of honing operation.

Backlash-honing with the addition of controlled torque resistance on the hone spindle is a popular method of honing internal gears. Both of these machine types are equipped with an adjustable electric torque brake on the honing spindle.

Gear Honing Tools

Honing tools are a mixture of plastic resins and abrasive grains such as silicon carbide, that is formed in a precision mold. They are made in a wide variety of mix numbers with grits ranging from 46 to 500, to suit special production and part requirements (Chapter 10).

Normally, molded hones are used for nick and burr removal operations. When ground, hardened; or precision-shaved and hardened gears are honed, special AA tolerance hones are applied.

Metal-bonded hones having a steel body and tooth surfaces with bonded abrasive coatings are often applied for honing fine pitch gears, and gears where hone breakage is a problem. These

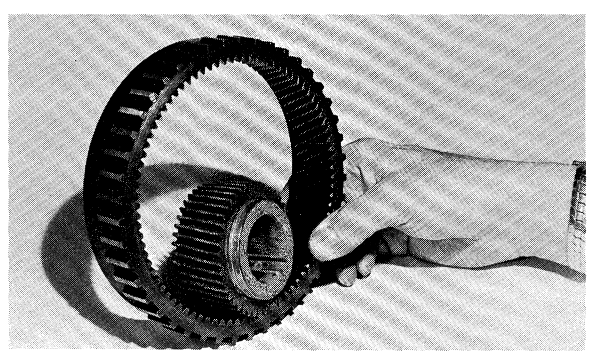


Fig. 6-60—Relationship of honing tool and work gear with internal gear tooth honing.

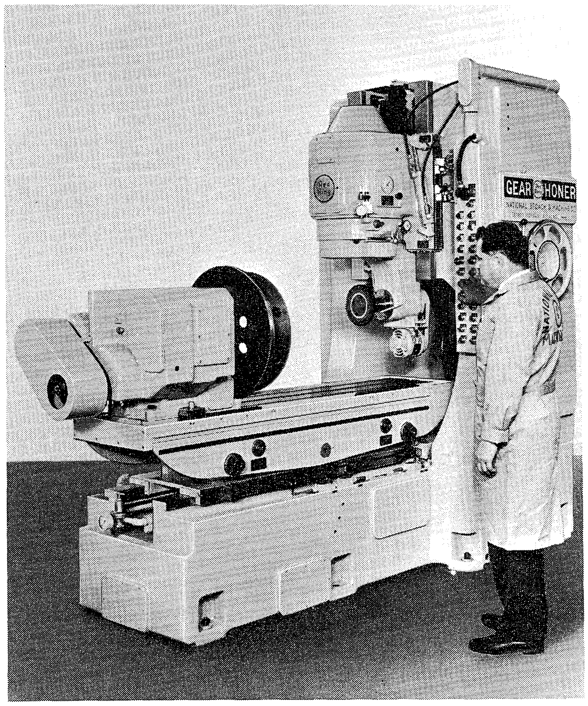


Fig. 6-61—A honing machine for internal gears up to 24-in. diameter.

honing tools are also available in a variety of grit sizes.

Removing Nicks and Burrs

Before gear honing was invented, the method used to remove nicks and burrs in production lines was to sound-test gears against masters in sound testing machines or hand-roll the gears with a master gear on a rolling fixture and observe center distance variations. Any nicks indicated by sound or indicator "jumps", required a time-consuming visual detection of the nick locations. Once found, the nicks were removed by a pencil grinder and sound-tested a second time.

Finally, any nicks that were not found by this method, showed up in the assembly test fixtures; requiring costly teardown and a second round of nick-searching operations.

With gear honing, one company has put all of its hardened truck transmission gears through a honing operation, completely eliminating the sound testing and nick removal operations. The overall result of this processing method has been a cost saving of 35-percent, and complete elimination of transmission teardowns for noisy gears in finished assemblies.

Correcting Heat-Treat Distortion

Earlier it was mentioned that gear honing can make minor corrections in tooth form that can improve overall sound characteristics and gear quality. The charts in Fig. 6-62 show significant improvements in profile, lead and eccentricity achieved by removing 0.002-in. measured over pins from shaved heavy-duty truck transmission, second-speed gears with carburized-and-hardened teeth on a 4¾-in. outside diameter. The gear, Fig. 6-63, has 23, 17½-degree pressure angle teeth on a 26-degree helix angle.

Typical honing production performances are shown in Table 6-8.

Use of the gear honing process as a salvage operation is a matter of economics. If gears to be salvaged have relatively large tooth shape errors, honing time is increased, and tool life is decreased. However, when gears that can be salvage-honed represent a large investment in dollars, honing may be an economical way to recoup this investment.

Honing Shaved Gears

Traditionally tooth surface finishes in the range

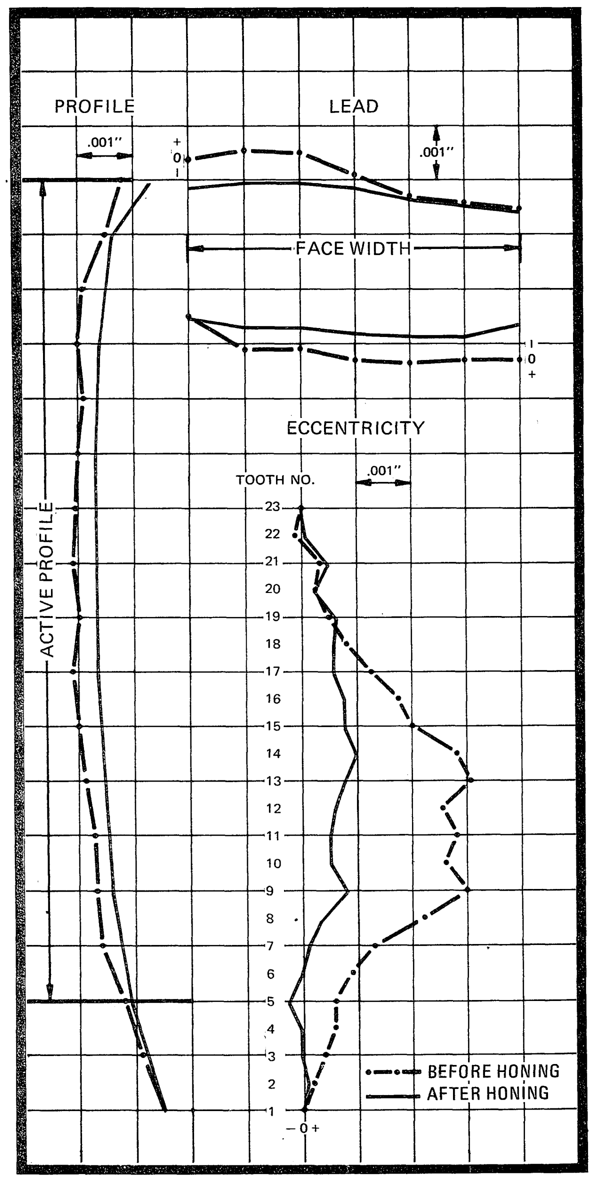


Fig. 6-62—Charts showing improvement in lead, involute profile and eccentricity by honing.

Work Gear	Accomplished	Pitch Dia. (in.)	No. Teeth	Rc Hardness	Stock Removed (in.)	Time (sec.) Machine	Floor to Floor
(A) Internal	Lead error reduced from .0008″ to .0001″	5.7832	83	58-63	.0015	40	70
(B) Planet Pinion	Runout improved .0020″	.790	15	56	.0005	27	31
(C) Rear Sun Gear	Notable improvement in surface finish and tooth kickout	2.3726	45	54	.0006	30	34
(D) Rear Sun Gear	Involute variation improved .0002″	2.508	45	58	.001	24	30
(E) Spur Gear	Pin Dim. reduced .004″ and all nicks removed	6.000	46	60	.004		205

Note: Gear "B"—Hone life 6000 to 8500 pieces; Gear "C"—Hone life 3000 to 4000 pieces; Gear "E"—Essentially a salvage operation.

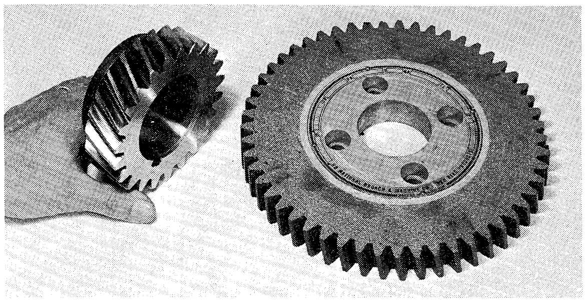


Fig. 6-63—A heavy-duty truck transmission and the tool that honed it to the improvements shown in Fig. 6-62.

from 10 to 40-mu have been provided by the rotary gear shaving process. The honing process, because it is not basically a heavy stock removal or tooth correction process cannot substitute for the gear shaving operation, which is performed on the soft gear following generation by hobbing, shaper-cutting, or broaching. In fact the tendency of a hone to charge a soft gear under 40 Rockwell C hardness with abrasive particles makes the honing of soft gears a questionable application.

However, because a gear has to be heat-treated, a process that usually roughens the tooth surface to a degree, the honing process tends to restore the hardened tooth surface finish to its original as-shaved condition and actually improve it. Table 6-9 shows effects of honing on the surface finish of hardened gears that were shaved before hardening. In all cases, the honed surface finish is better than the surface finish before honing.

This table shows that a wide range of gears, honing tools, different honing methods and honing speeds were utilized in these tests. To hone production gears, economy dictates that one grit of tool and a relatively short honing cycle be

Fig. 6-64—A honed herringbone marine gear.

Table 6-9—Gear Tooth Surface Finishes Before and After Honing

No. Teeth	Normal Diametral Pitch	Pressure Angle, (deg.)	Helix Angle, (deg and hand)	Grit of Hone	Honing Time, (sec)	Honing Pressure, (lb)	Honing Speed, (sfpm)	Surface Finish before Honing, (mu)	Surface Finish after Honing, (mu)
44	18	20	18½ R.H.	100	12	25	622	8-10	6
44	18	20	18½ R.H.	100	12	25	622	10-12	8
30	6	17½	26° 10′ 37″ L.H.	60	90	30	836	15-18	10
19 (Long Pinion)	15.5	17½	22° 11′ 30″ R.H.	60	28	25	353	18-22	8-10
19 (Short Pinion)	15.5	17½	22° 11′ 30″ R.H.	60	20	25	353	16-19	13-15
15	20	20	18½ R.H.	100	22	zero backlash	553	14-16	9-11
26	9.25	16½	26° 40′ L.H.	60	23	30	800	30-35	21-26
26	9.25	16½	26° 40′ L.H.	100	23	30	800	30-35	10-15
19	10	20	29° 45′ 47″ R.H.	60	24	28	900	26	18-20
27	5	20	Spur	60	64	35	1064	25-27	6-7

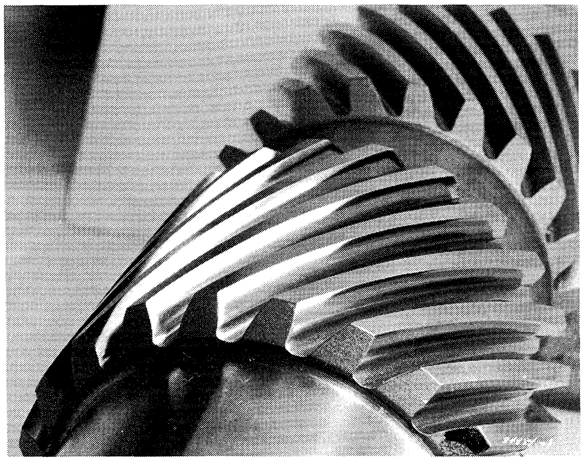


Fig. 6-65—Close-up view of 2 to 4-mu surface finish on the honed herringbone pinion that mates with gear in Fig. 6-64.

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

Honing Ground Gears

The herringbone gears shown in Fig. 6-64 and 6-65 are a set of marine gears upon which a closely controlled honing process was carried out. The larger gear has 65 teeth and the smaller has 22-teeth. The teeth are 4.6 D.P., 17½-deg pressure angle and have a 31½-deg helix angle. Each gear has a 4½-in. face width. The carburized-and-hardened alloy steel gears, which had pits, heat-treat scale, and distortion; were first ground to remove 0.012-in. of stock on tooth thickness from the large gear and 0.015-in. of stock from the pinion.

After grinding, the gears were stress-relieved, Magnafluxed and shot-peened. The ground gear surfaces had a 20-mu surface finish. In honing, approximately 0.002-in. stock, measured over pins was removed. Figure 6-65 shows the finish obtained by the honing process, which measured between 2 and 4 microinches. This finish was obtained by using a 180-grit honing tool followed by a cycle with a 500-grit tool.

The tremendous impact of honing on the load-carrying capacity and wear life of hardened and ground aerospace gearing (Chapter 10) has put new emphasis on the value of accuracy and surface finish on overall gear quality.

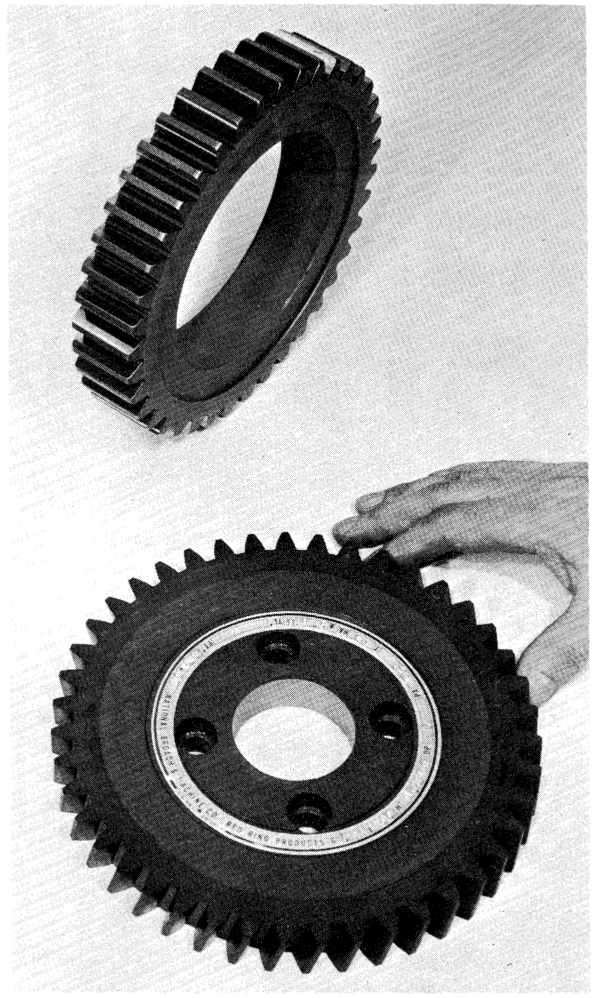


Fig. 6-66—A helicopter drive gear whose ground teeth were honed to improve wear life and load-carrying capacity.

The Proficorder check of one of these gears, Fig. 6-66, before and after honing is shown in Fig. 6-67. The spur gear in Fig. 6-66, is a 39-tooth, 5-D.P., 25-deg P.A., 7.800-in. P.D., helicopter drive gear whose ground teeth have been honed to an 8-mu finish. Tooth-to-tooth spacing is held to 0.0003-in., accumulative tooth spacing to 0.001-in., and parallelism to 0.0003-inch.

In the years ahead, when reduction of noise will undoubtedly be a matter of government legislation, the gear honing process may well offer the prime production solution to noise pollution where quieter-operating gears are necessary.

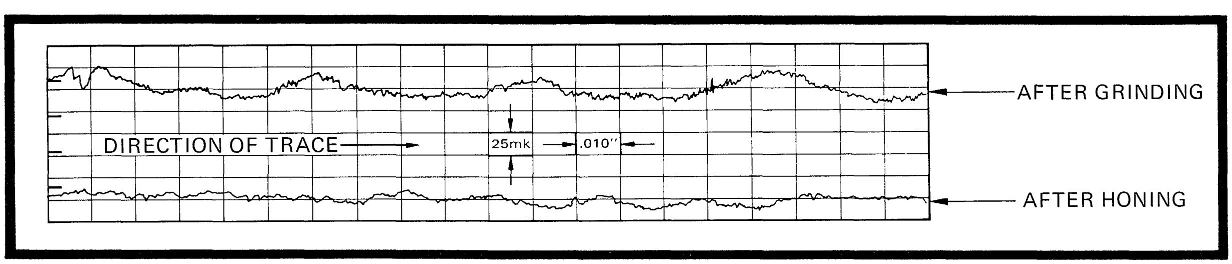
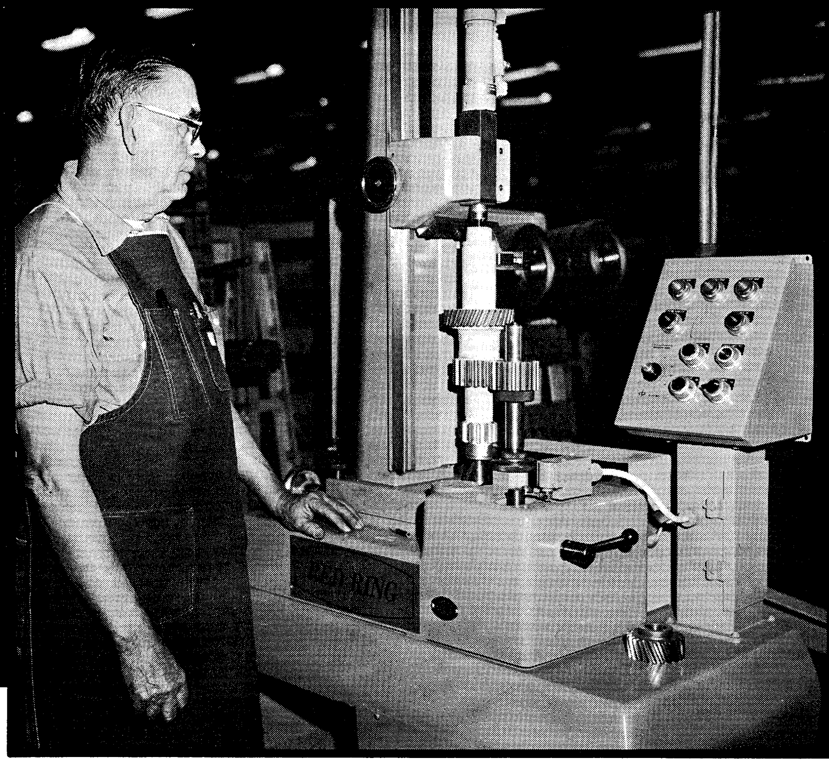


Fig. 6-67—Proficorder checks of ground gear teeth in Fig. 6-66 before and after honing.



Chapter SEVEN

Checking cluster gear teeth after heat treatment to determine nick condition prior to honing. *Courtesy Clark Equipment Co.*

Inspecting the Teeth

The dimensional inspection of gear teeth is one of the key functions in the production of gears. It is the measurement function that controls initial machine set-up accuracy, makes the necessary allowances for material heat treat movements, controls the overall level of quality and pin-points tooth production problems and solutions.

Involute gear teeth are subject to the following errors, Fig. 7-1: Profile, tooth thickness, spacing, lead (helical gears), parallelism (spur gears) and runout. Table 7-1 shows typical tolerances for automotive passenger car transmission gears. It can be seen that the measurement of gear tooth dimensions is a precision operation.

The Gear Laboratory

Since gear measurements are so precise, it is common practice for these to be carried out in a soundproof, air-conditioned room; assuring that measurements of maximum dimensional accuracy can be quickly made.

Normal practice to establish green tooth forms for a given heat of steel is to process a small batch of gears quickly through the green production operations, inspect them and then feed them on through heat treatment. Final measurements are then made to determine if allowances for heat treat movements are correct.

Table 7-1—Typical Gear Tooth Dimension Tolerances For Automotive Passenger Car Transmissions. (Variations From Established Dimensions).

Dimension	Pre-Shaved Gear	Shaved Gear
Involute Profile	0.0005-in. ave., 0.001-in. max.	0.0003-in. ave. 0.0008-in. max.
Lead	0.001-in. per in., 0.0012-in. max.	0.0004-in. per in. 0.0008-in. max.
Size	0.004-in. over pins	0.002-in. over pins
Eccentricity	0.003-in.	0.002-in.
Spacing	0.0005 to 0.001-in., tooth-to-tooth	0.0003 to 0.00006-in., tooth-to-tooth

The laboratory is responsible for checking each new gear-cutting or finishing machine setup before the operator can proceed with production. A spot check of shaved gears about twice every eight hours provides adequate quality control

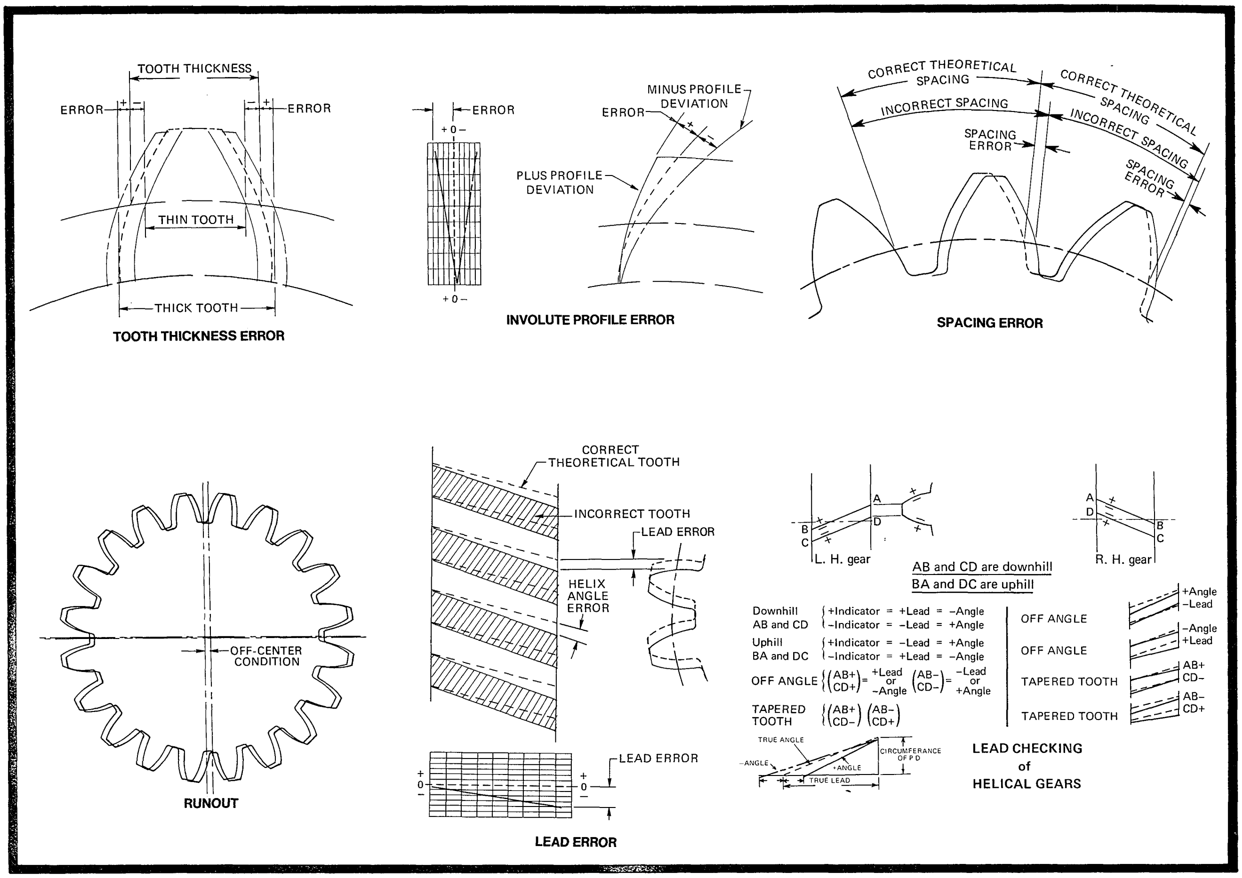


Fig. 7-1—Typical errors for gear teeth for which tolerances must be established and inspection measurements made.

between cutter changes.

When checking equipment on the production floor indicates that bad gears are being produced, it is the responsibility of the gear laboratory to quickly analyze the gears and find out the problem.

To make a composite check of a gear in a gear laboratory, it is put on a rolling fixture, Fig. 7-2, where the gear is run in mesh with a master gear. Displacement of the gear spindle as it rolls with the master is indicated by a line that is inked on a moving chart.

Inspectors experienced in operating rolling fixtures can quickly spot general roughness, eccentricity or spacing errors. Following the rolling fixture check, a series of individual checks may be required, Figs. 7-3 & 4, such as spacing, lead and profile measurements to point out necessary corrective measures.

Sound testers, Fig. 7-5, are also used as a check of accuracy of both green and hardened gears in the gear laboratory.

Production Inspection

The inspection of general gear accuracy in production has been traditionally a matter of rolling the gear with a precision master gear and measuring the change in center distance by allowing either the work gear or master gear mounting to move during the rolling operation. Such a composite check gives an indication of the total variation, size and eccentricity of the gear teeth. All errors in the various tooth characteristics are reflected into a single reading.

Modern bench-type rolling fixtures, Chapter 10, are now equipped with motor drives, and in some cases, electric indicator light arrangements, to avoid operator variables and the reading of indicators.

In low production, the composite checker is a sufficient means of checking a gear, with such other elements as helix angle, outside diameter or involute profile being spot-checked on individual gaging equipment. Multi-function gages that check all of the important gear elements are applied in a single final inspection operation in high-production operations.

The installation of a trunnion mounting for the master gear in a rolling fixture, for example, permits the twist position of the master to be indicated, thus providing a simultaneous measurement of size, eccentricity and helix angle.

High-Production Gaging

It has been found most desirable to provide 100-percent inspection of gears in high production lines to avoid teardowns at assembly and maintain a high performance level of quiet gear operation.

As a result, the bench-type rolling fixture has been supplanted by other types of automated multi-function gages that can handle the gears

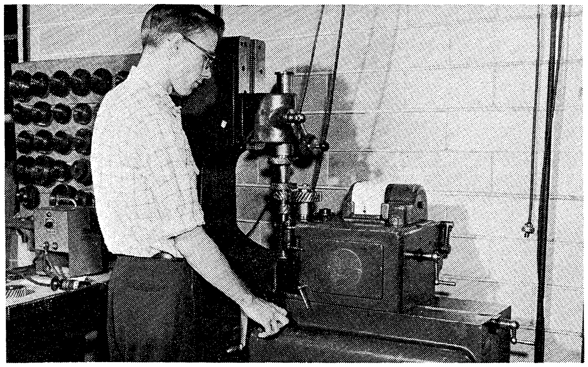


Fig. 7-2—Charting a composite check of a cluster gear on a Red Liner. *Courtesy Warner Gear Div.*

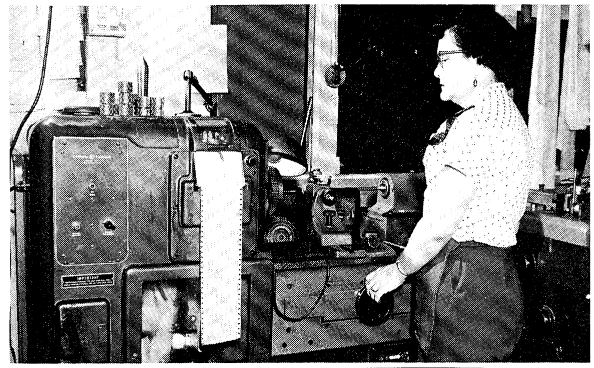


Fig. 7-3—Measuring and charting the lead on a helical gear on a lead checker. *Courtesy Warner Gear Div.*

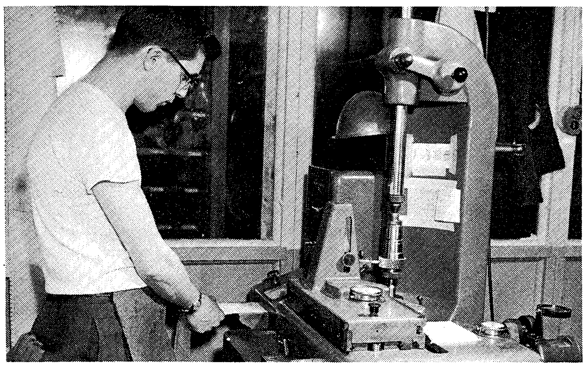


Fig. 7-4—Charting gear involute profile accuracy on an involute checker. *Courtesy Warner Gear Div.*

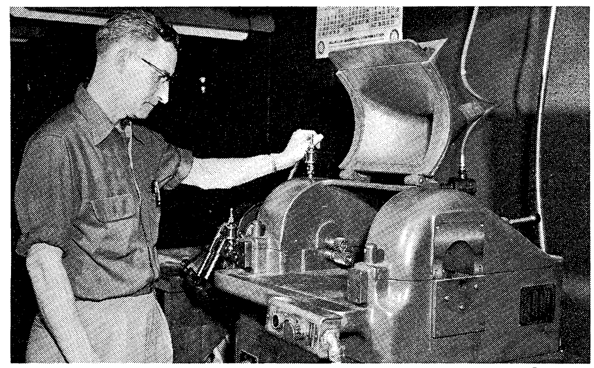


Fig. 7-5—Sound-testing of transmission gears in a sound-proof room. *Courtesy Warner Gear Div.*

fast and make quick inspection and rejection decisions.

The multi-function gages of the single-station type have all of the necessary inspection checks on a gear or pinion carried out in a single gaging station. Thus, the gear is inspected without movements to different gaging stations, and the possiblitity of errors resulting from different locating conditions is avoided. Single-station gear gaging equipment provides the ultimate in simplicity and carries out the inspection function in a minimum space.

Detecting Nicks

One of the most frequent causes for noisy gear operation and teardowns at assembly is gear nicks. A nick will show up quickly on a sound testing machine, and can be removed with a pencil grinder by hand. However, if the nick can be detected in a multi-function gage, considerable production economy can result.

To adapt a motorized rolling fixture to detect a nick, a patented Red Ring inertia-type system has been devised in which nick and eccentricity movements induced into the master gear slide can be separated for measurement. This is achieved, Fig. 7-6, left, by reed suspension of a weight that has sufficient inertial capacity to prevent its following any abrupt or rapid motions caused by a nick. The weight slowly follows and eliminates any movement caused by eccentricity in the work gear. The electric indicator therefore only registers the rapid motions induced by nicks.

To measure eccentricity, Fig. 7-6, right, the master gear slide has a friction-controlled motion whose indicator only reads center distance changes caused by work gear eccentricity, and is not affected by the gear's pitch diameter size. The friction bar that operates the eccentricity indicator only moves when there is eccentricity of the rotating work gear.

The friction bar is pushed forward by the master gear slide as the low point of eccentricity of the work gear contacts the master gear. The slide moves away from the bar when the high point of eccentricity contacts the master. The amount that the slide moves away from the sliding friction bar is measured by the indicator as eccentricity (total indicator reading).

Often it may be feasible to check a finished gear only for nicks before it is assembled in a transmission.

The nick detector shown at the start of this chapter has a rolling fixture that utilizes the inertia checking principle to analyze gears for nick condition. The column on the detector supports gears on centers.

The nick detector has a master gear that is driven at constant speed of about 20-rpm through an adjustable speed drive. The hardened work gear is meshed with the pivoted rotating master gear whose deflections are measured through the inertia checking system to detect large nicks (over 0.0025-in.-high), small nicks (up to 0.0025-

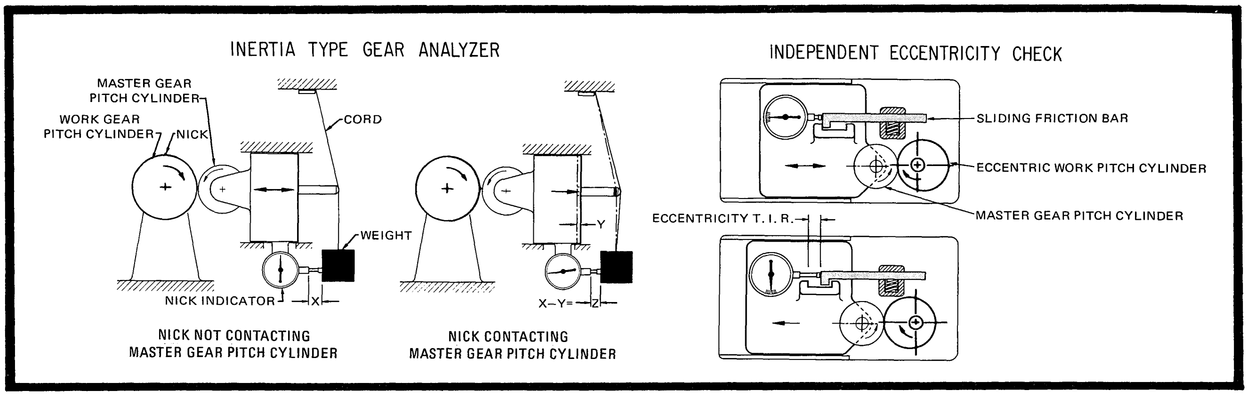


Fig. 7-6—The patented Red Ring inertia principle provides separate nick (left) and eccentricity (right) measurements.

in. high), or nick-free condition (under 0.0015-in. high).

The work cycle is initiated by operation of a switch by the master gear shaft which causes a timer-operated sequence to start. The nick checking operation is made in about two revolutions to assure complete peripheral contact between the master gear and the work gear.

Panel lights tell the inspector the condition of the gear being checked.

Gear Inspection Centers

When a wide variety of measuring, sorting and reject operations are carried out automatically on multi-function gear gages, they are called gear inspection centers. Where gears are mounted on arbors in single-station gear inspection centers it is possible to air-gage the bore size simultaneously with the tooth measurement checks. Further it is possible to include the inertia checking feature in the inspection centers to provide nick detection.

One of the more sophisticated gear inspection lines is shown in, Fig. 7-7, where five centers check automotive automatic transmission planet pinions, each at a rate of 400 pinions per hour.

The fully-automated inspection center first burnishes each gear with a triple burnishing gear arrangement and then inspects it for tooth size, eccentricity, helix angle, tooth action and nicks as well as bore size, length and face runout.

After each gear has been inspected it rolls into an exit chute where trap doors drop it into one of nine semi-circular storage troughs. A rotary actuator unit feeds the gears axially along the trough-type units.

Two of the storage troughs are for non-salvable reject gears. Three troughs accommodate salvable gears. The four other storage troughs are for gears classified into different tooth size ranges for selective close-backlash assembly.

Gages That Protect Tools

Breakage of gear shaving cutters or rolling dies due to incorrect or incomplete hobbed or shaper-cut teeth can be completely avoided by automatically gaging the parts at rates up to 720 pieces per hour in a single station ahead of the shaving or rolling operation.

The standardized automated gear inspection center in Fig. 7-8, inspects gears for tooth size, outside diameter accuracy, and malformed teeth before feeding into gear shavers or gear rolling machines.

Gears to be checked are stored in a magazine. Parts are fed one at a time into mesh with a motor-driven rotating master gear and an O.D. contact roller. A one-revolution check of the gear is made.

Movement of the pivoted master gear and O.D. contact roller is measured by electrical gaging units, whose signals operate trap doors in the exit chute. Reject gears pass on through the exit chute. Gears that pass the inspection tests drop out of the chute through the trap doors.

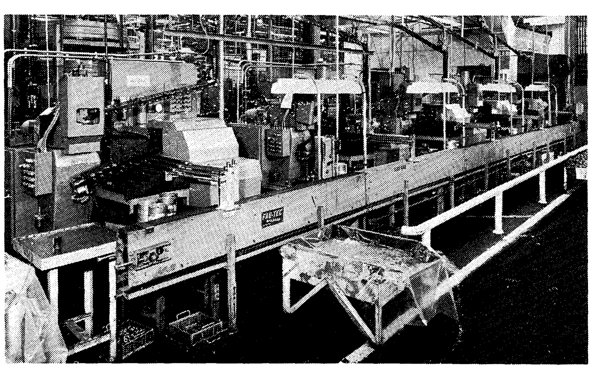


Fig. 7-7—A fully-automated line of five gear inspection centers for checking planetary transmission pinions.

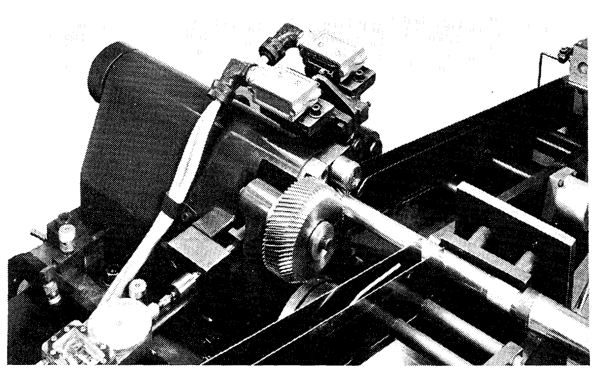
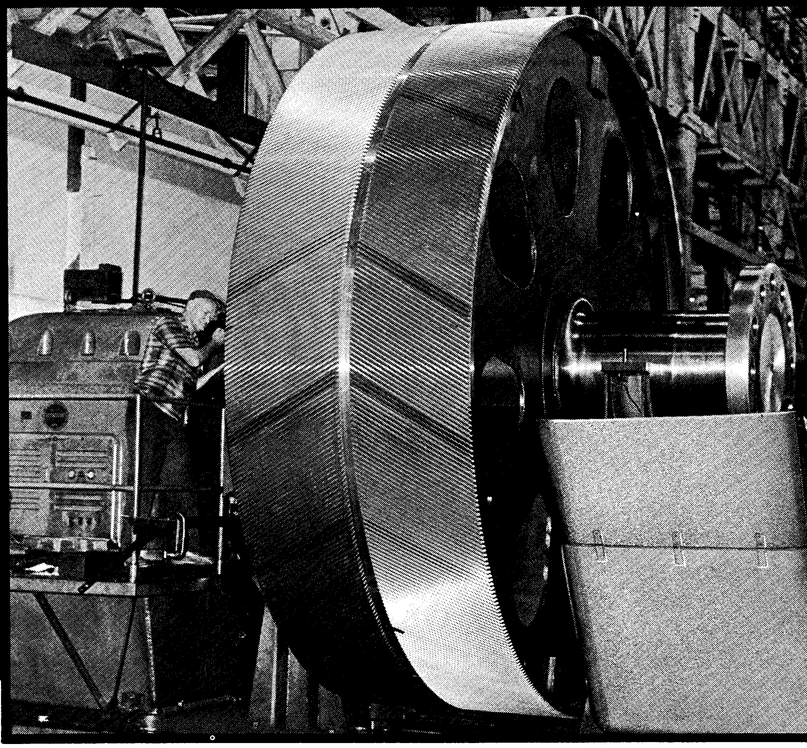


Fig. 7-8—A standardized gear inspection center that checks gears before being fed into shavers or gear rollers.



Chapter EIGHT

Shaving a 459-tooth, 164-in. dia., double-helical, 28,500-hp. steam turbine ship propulsion gear on a Red Ring gear shaver.

Courtesy Marine Div., Westinghouse Electric Corp.

Modern Gear Designs

In this chapter a wide variety of modern gear designs are discussed in detail. They range from tiny instrument spur gears up to huge double-helical reduction gears for steam turbines. Each of the applications gives gearset functional information followed by detailed gear tooth data including pitch, pressure angle, helix angle, number of teeth, material and processing sequence.

Helicopter Planetary Gear Transmission

This two-stage planetary gear set reduces input speed of 7,335-rpm to 240-rpm to drive the aft or rearward rotor of a twin-rotor helicopter. Input torque is 34,000 in.-lb; output torque is 1,043,000 in.-lb. A bevel gear set drives the first stage sun gear or input member, and the first stage planet carrier drives the second stage sun gear. Output to the rotor is from the second stage planet carrier. Both ring gears are held stationary.

Gear Data: 5-DP; 25°-PA (Spur).

Pitch Diameters: Ring Gear, 21.2-in.; 1st Stage Sun Gear, 5.6-in.; 1st Stage Planet Gear (4 Req'd), 7.8-in.; 2nd Stage Sun Gear, 8.0-in.; 2nd Stage Planet Gear (6 Req'd.); 6.6-in.

Gear Material: SAE 9310 (Carburized).

Processing: Hobbed, Heat Treated, Ground and Honed.

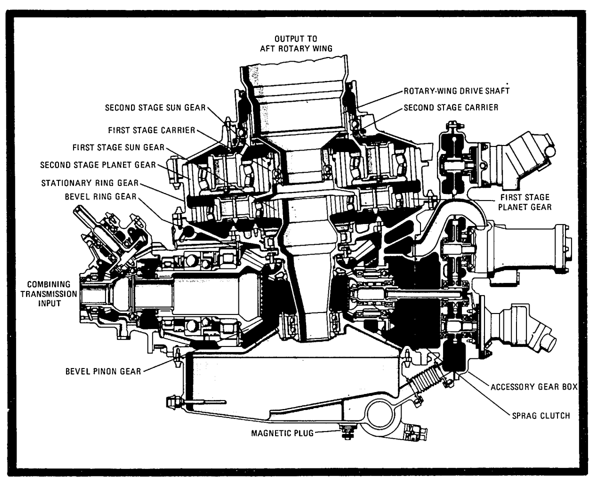


Fig. 8-1—*Courtesy The Boeing Co., Vertol Division.*

Marine Reversing Reduction Gears

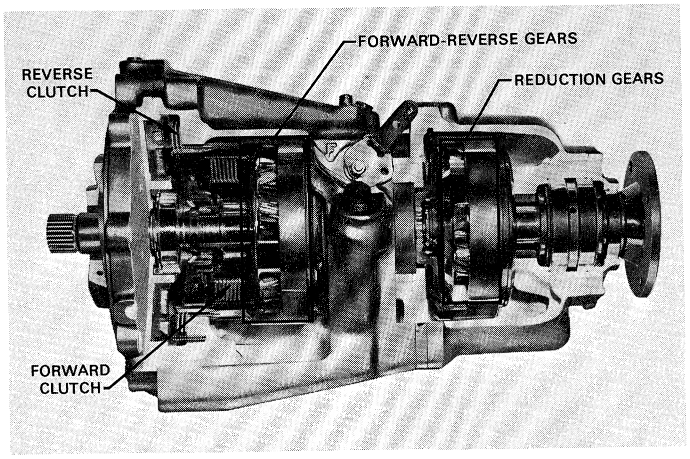


Fig. 8-2—*Courtesy Warner Gear/Warner-Motive.*

Two planetary gearsets, one engaged and disengaged by an oil-operated clutch pack, provide forward-neutral-and-reverse control plus fixed gear reduction to provide most efficient propeller speed from fairly fast-turning engines. Three different models of the same basic design, differing in size, are available in four models up to 560-hp at 4,200 revolutions per minute. Overall reduction ratios are as shown in the following table.

Overall Reduction Ratios

Forward	Reverse
1.52	1.52
1.52	1.67
1.91	1.91
1.91	2.10
2.10	2.10

Forward	Reverse
2.10	2.31
2.57	2.57
2.57	2.83
2.91	2.91
2.91	3.20

Planetary Gearset Design No. 1
Gear Data: 11.39-NDP, 20°-NPA, 15.106°- R. & L.H. Helix Angle.
No. Teeth: Sun Gears, 30, 33, 42; Pinions, 13; Ring Gears, 63, 66.
Material: Sun Gears SAE 4027, 4047; Pinions, SAE 4047, 5140; Ring Gears, 4027, 8620.

Planetary Gearset Design No. 2
Gear Data: 15.5-NDP, 17°-NPA, 22°11′30″-R. & L. H. Helix Angle.
No. Teeth: Sun Gear, 36; Pinion 15, Ring Gear, 72.
Material: Sun Gear, SAE 4027; Pinion and Ring Gear SAE 4047.

Planetary Gearset Design No. 3
Gear Data: 9.47-NDP, 22°-NPA; 11° 8′ 58.73″-R. & L.H. Helix Angle.
No. Teeth: Sun Gear, 31; Pinion 14, Ring Gear 62.

Gear Material: Sun Gear, SAE 4047; Pinion, SAE 4047, 5140; Ring Gear, SAE 4027, 8620.

Processing: Reduction Ring Gears, Broached; Reversing Ring Gears, Shaped and Shaved; Sun Gears and Pinions, Hobbed & Shaved.

Two-Speed Rear Axle Gears

Four helical gears integrated with the axle gears in the rear axle housing offer a choice of two overall reduction ratios so that recreational vehicles and vehicles used for towing may use the ratio best suited to operating conditions. For example, a passenger car which is sometimes used to haul a fairly heavy trailer would use the higher ratio for performance when towing the trailer and the lower ratio for economy when not towing the trailer. The unit includes as standard a limited-slip differential with a 9¾ in. dia. ring gear. Direct drive ratio is the axle gear ratio. Some typical overall gear ratios are: 3.54:1, 3.73:1, 4.10:1, 4.56:1 and 4.87:1.

By engagement of a sliding clutch, the helical gear train is engaged to provide overdrive, or underdrive, depending on the selection of helical gears. In overdrive or underdrive, the overall ratio is the product of the axle gear ratio and the ratio of the helical gear train. Some of the typical ratios for the helical gear train are: 0.67:1, 0.74:1, 0.82:1 or 0.90:1. Horsepower and torque capacity of the overdrive gear train and axle gears exceed the requirements of all existing gasoline engines used in passenger cars, pickup trucks and recreational vehicles.

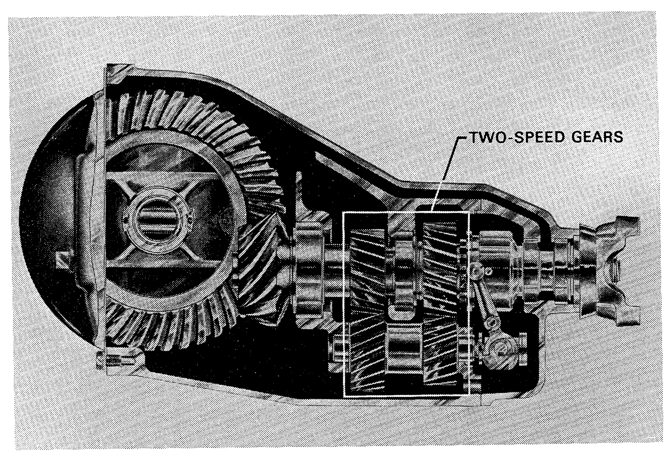


Fig. 8-3—*Courtesy Spicer Axle Div., Dana Corp.*

Gear Data: 8-NDP, 20°-NPA, 21° 20′-R & L. H. Helix Angle.

Pitch Diameters: 2.488 to 3.512-Inches.

Gear Material: SAE 8620.

Processing: Hobbed, Shaved, Heat-Treated and Honed.

The Torqueflite Automatic Transmission

This unit, typical of many of today's automobile automatic transmissions, uses two planetary gearsets operating individually or in combination to give overall gear ratios of 2.45:1 (Low or First Gear), 1.45:1 (Second Gear), 1:1 (High Gear), and 2.2:1 (Reverse Gear). The unit is designed to operate with a maximum of 470 hp input, or at input speeds to 7200 rpm with maximum rated torque input of 1080 lb-ft.

Gear Data: 14-NDP, 20°-NPA, 22°30′-R. & L. H. Helix Angles.

Sun Gear: 28-Teeth; SAE 4023 or 4034.

Planet Pinions (3 per set): 17-Teeth; SAE 4024.

Ring Gears: 62-Teeth, SAE 4027.

Processing: Sun Gear, Hobbed and Shaved; Planet Pinions, Hobbed and Shaved; Front Ring Gear, Broached and Shaved; Rear Ring Gear, Broached.

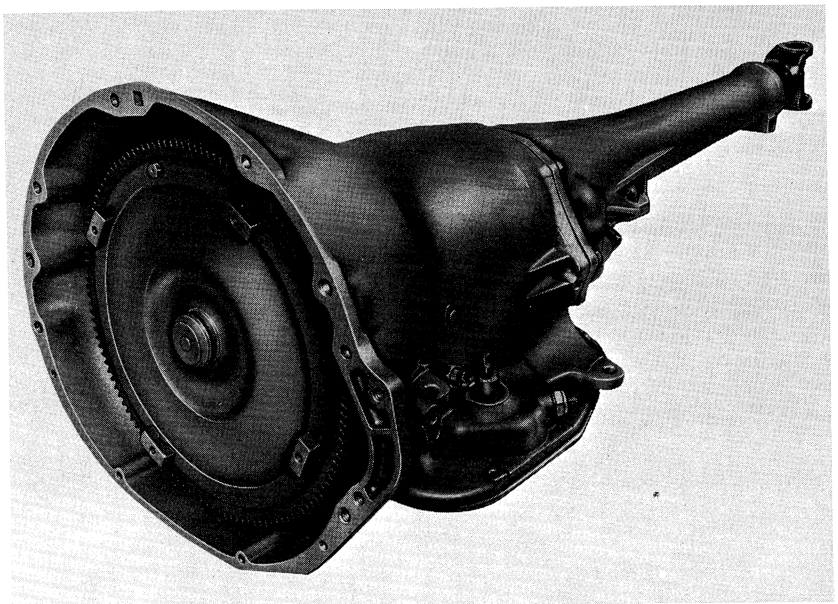
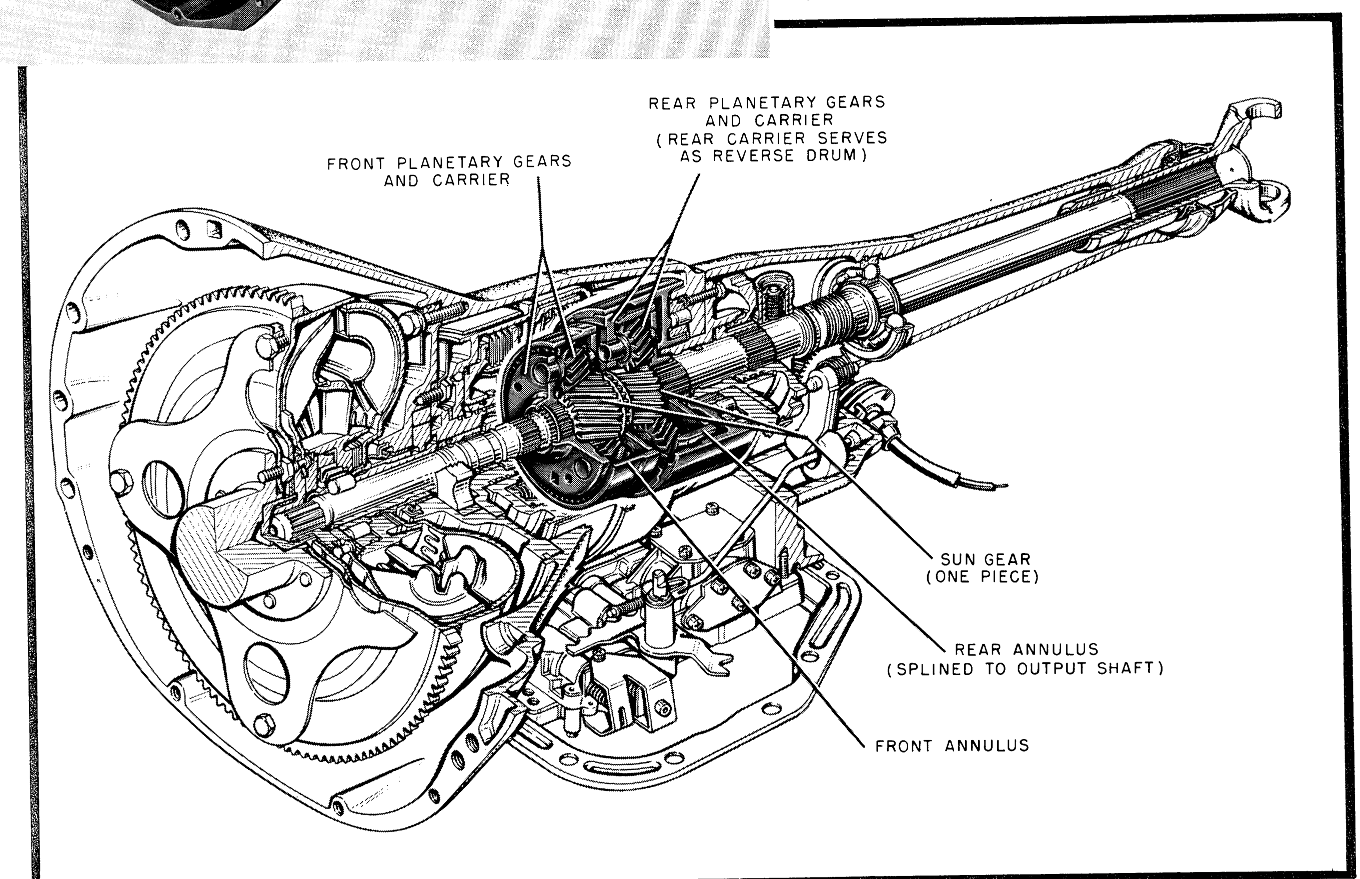


Fig. 8-4—*Courtesy Chrysler Corporation.*



Heavy-Duty Five-Speed Truck Transmission

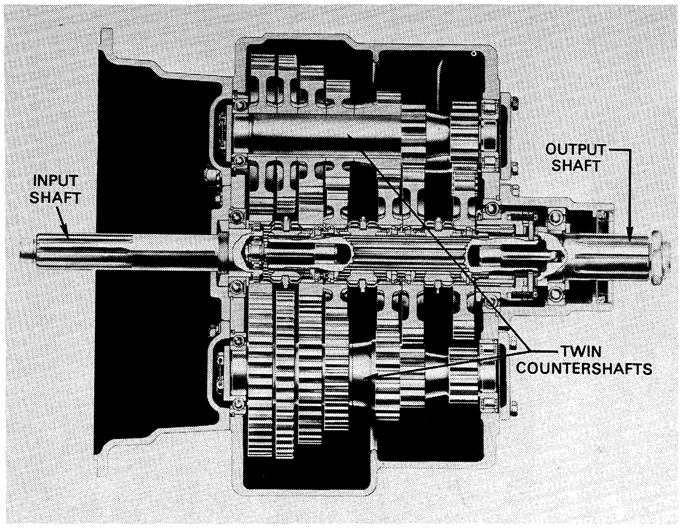


Fig. 8-5—*Courtesy Fuller Transmission Div., Eaton Yale & Towne, Inc.*

This twin countershaft transmission splits engine torque equally between the two countershafts, allowing a 40-percent reduction in gear widths, thus providing a shorter, lighter transmission. These units are used on a wide variety of on- and off-highway equipment.

Maximum input speed is 2800-rpm; maximum input torque is 900 ft-pounds. The units are available in several models which differ in available gear ratios as shown by the following table.

Gear Ratios

Model No.	1st	2nd	3rd	4th	5th	Reverse
1	6.35	3.75	2.38	1.54	1.00	6.48
2	4.12	2.44	1.54	1.00	.65	4.21
3	6.35	3.75	2.04	1.16	1.00	6.48
4	5.47	3.23	1.76	1.00	.86	5.58
5	7.23	3.92	2.16	1.18	1.00	7.12
6	6.11	3.31	1.82	1.00	.84	6.01
7	7.23	3.92	2.16	1.45	1.00	7.12
8	4.97	2.69	1.48	1.00	.69	4.90
9	6.92	3.75	2.38	1.54	1.00	6.48
10	7.23	4.10	2.56	1.61	1.00	7.12
11	5.45	2.67	1.76	1.32	1.00	5.58

Gear Data: 5 or 6-DP, 20°-PA, Spur.

Gear Material: SAE 8620; Carburized, Hardened.

Processing: Hobbed or Shaper-Cut, Shaved, Heat-Treated and Honed.

Diesel Engine Timing Gears

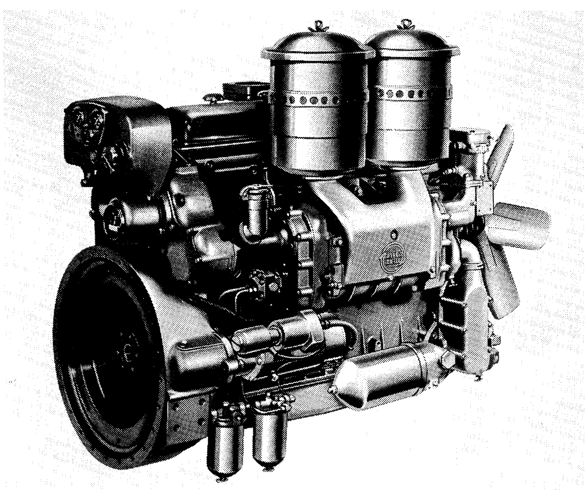


Fig. 8-6—*Courtesy Detroit Diesel Allison Div., GMC.*

The timing gear train on these two-cycle diesel engines consists of five gears to drive the camshaft, balance shaft, and a Roots-type blower. The important function of this drive is accurately synchronizing cam shaft rotation with crankshaft rotation so that fuel injection occurs at the proper point relative to the piston position in the cylinder. Long, trouble-free life is also important. Crankshaft, camshaft and balance shaft rotate at the same speed; the blower is driven at 1.95 times crankshaft speed.

Gear Data: 10-NDP, 16°-NPA, 19°41′-R. & L.H. Helix Angle.

Numbers of Teeth: Crankshaft Gear, 78; Idler Gear, 68; Camshaft Gear, 78; Balance Shaft, 78; Blower Drive, 40.

Gear Material: Pearlitic Malleable Iron.

Processing: Hobbed or Shaper-Cut, Crown-Shaved.

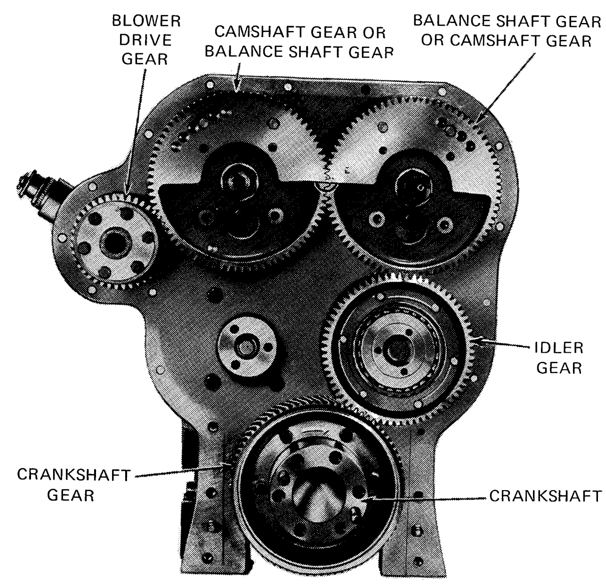


Fig. 8-7—End view of Series 71 diesel engine timing gears.

Helicopter Turbine Engine Reduction Gearing

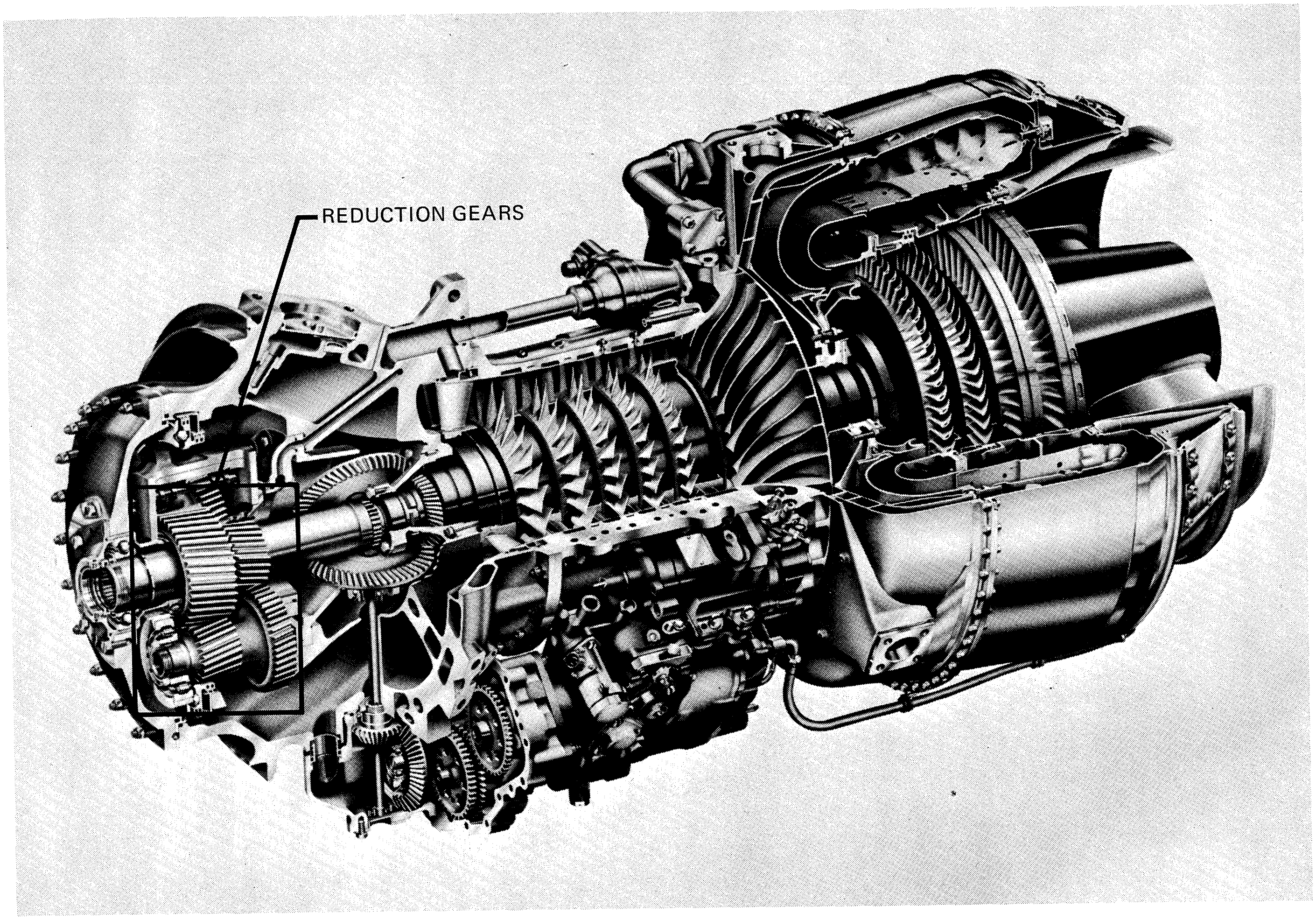


Fig. 8-8—*Courtesy Avco Lycoming Div.*

A compound planetary gear set reduces power turbine speed of 20,226-rpm to approximately 6,300-rpm output speed to drive a helicopter transmission. Overall reduction ratio is 3.21:1. Horsepower of the turbine is 1,400. This gear arrangement produces high torque capacity, compactness and a desirable coaxial arrangement of the gearing.

The power turbine shaft drives the input sun gear. The sun gear drives four primary planet gears. Four secondary planet gears, which are integral with the primary planet gears, drive the output.

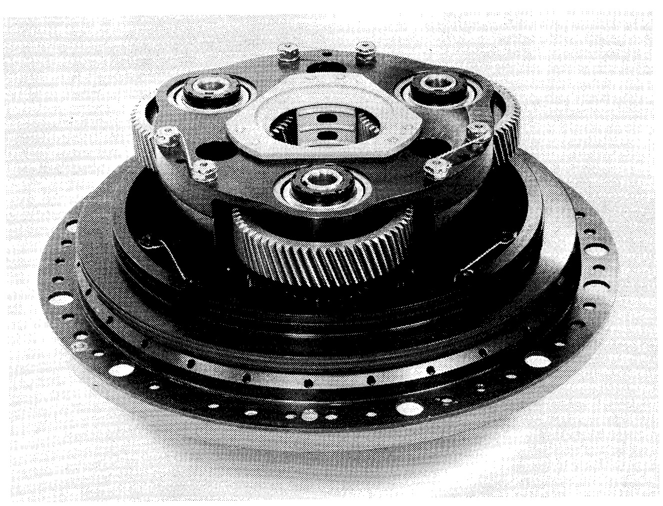


Fig. 8-9—Planetary gearset viewed from input end.

Input Gearing:
Gear Data: 15.09-TDP; 20°-NPA; 18°-R & L. H. Helix Angle.
Sun Gear: 38-Teeth; 2.52-in. P.D.
Primary Planet Pinions (3): 61-Teeth, 4.04-in. P.D.

Output Gearing:
Gear Data: 20.28-TDP; 20°-NPA; 10°-R. & L. H. Helix Angle.
Secondary Planet Pinions (3): 27-Teeth; 10.02-in. P.D.
Output Sun Gear: 54-Teeth; 4.37-in. P.D.

Gear Material: SAE 9310.

Processing: Input Sun Gear, Primary Planet Pinions and Output Sun Gear: Hobbed, Carburized, Hardened and Ground. Primary Planet Pinions and Output Sun Gear Honed After Grinding. Secondary Planet Pinions Shaped, Carburized, Hardened and Ground.

Self-Propelled Combine Transmission

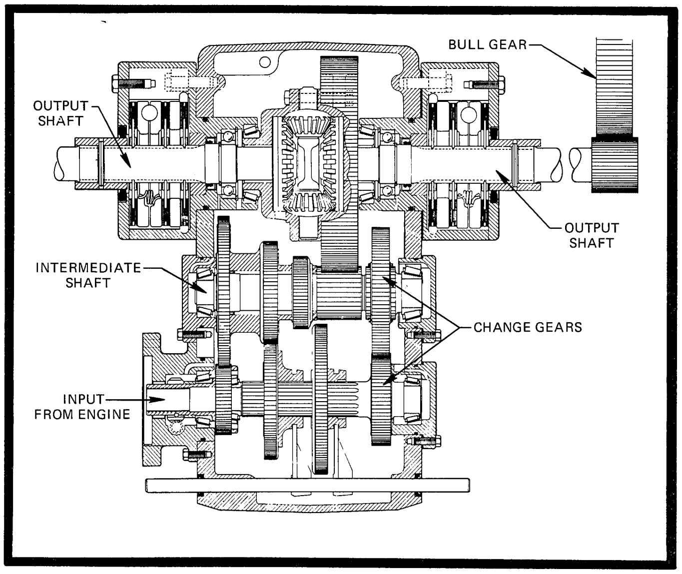


Fig. 8-10—*Courtesy International Harvester Co.*

This self-propelled combine transmission provides three output speeds by using the shift lever. A pair of change gears provides three ranges as shown in the following table. Input to the transmission is the output of an internal combustion engine with a rating of 38-hp at 1,400 rpm; maximum torque of the engine is 142 foot-pounds.

Reduction Ratios

Gear	Change Gear Tooth Numbers		
	37/36	41/32	43/30
1st	52.8	65.6	73.4
2nd	29.1	36.1	40.3
3rd	10.9	13.6	15.2

Gear Data: Input and Intermediate Shaft Gears, 7.0-DP, 20°-PA; Differential Ring Gear Set, 5.5-DP, 20°-PA; Change Gears, 6.0-DP, 20°-PA; Final Drive Gears 4.5-DP, 25°-PA. All Spur Gears.

Gear Material: SAE 8620 and 4118; Bull Gear SAE 1053, Induction-Hardened.

Processing: Hobbed, Shaved, Hardened and Honed

Floor Machine Planetary Reduction Gears

Fig. 8-11—*Courtesy Clarke Floor Machines.*

This floor maintenance machine may be used for scrubbing, waxing, polishing, buffing and steel wooling. It is applicable to wood, tile, linoleum, cork, terrazo, cement and other types of floors. A 1-hp, 1,750-rpm electric motor powers the unit. A planetary gear set with an overall ratio of 10:1 reduces speed and provides a compact, in-line drive set-up.

Gear Data: 24-NDP, 22½-PA; 20°4′36″-R. & L. H. Helix Angles

Gear Material: Sun Gear, SAE 4140, 38-43 Rc; (2) Planet Pinions, SAE 4140, 30-35Rc; Internal Gear, Meehanite Cast Iron.

Processing: Sun Gear and Planet Pinion, Hobbed and Shaved after Hardening; Internal Gear, Shaped.

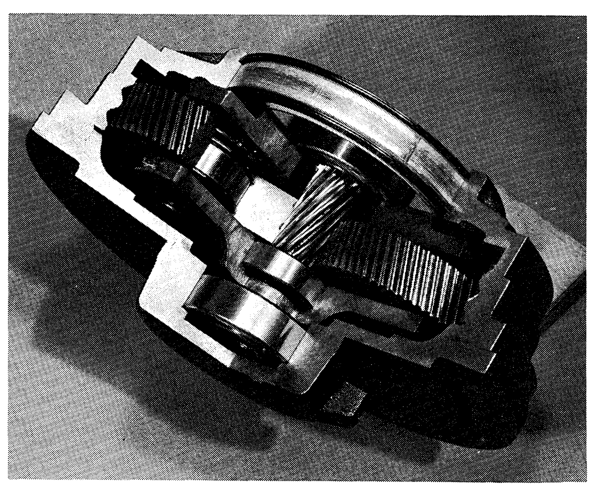


Fig. 8-12—Cutaway view of planetary gear reduction.

Heavy-Duty, Full-Power-Shift Transmission

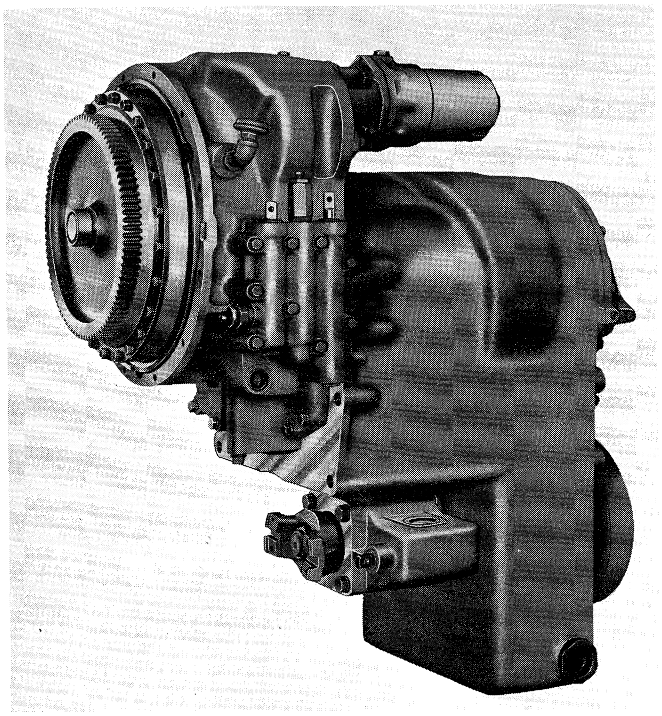


Fig. 8-13—*Courtesy Clark Equipment Co.*

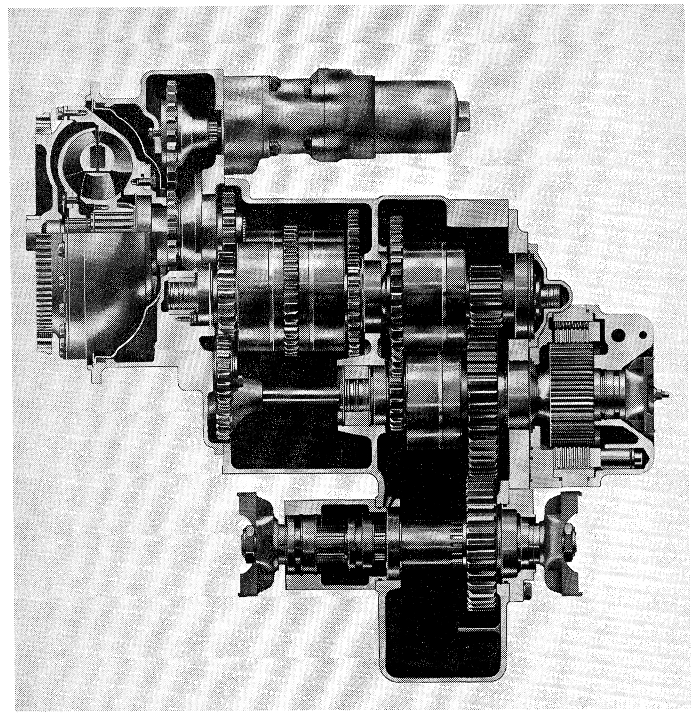


Fig. 8-14—Cutaway view of power-shift transmission.

The heavy-duty transmission pictured here is one of a series offering full-power-shifting by means of constant-mesh gears and oil-operated clutches. The series includes models offering two, three, four, or six speeds in both forward and reverse. The model shown is the four-speed version. All models have rated input up to 200-hp and 350 ft-lb of torque. All models are designed for use with 11, 12, or 13-in. dia. torque converters and are available in versions for engine mounting, midship mounting, or remote mounting.

Gear Ratios (Forward and Reverse)
1st Gear: 5.18, 4.84 or 4.52 to 1
2nd Gear: 2.45, 2.28 or 2.13 to 1
3rd Gear: 1.41, 1.32 or 1.23 to 1
4th Gear: 0.78, 0.73 or 0.685 to 1

Gear Data: 6-DP, 20°-PA, Spur.

Gear Material: SAE 8620.

Processing: Hobbed or Shaper-Cut, Shaved, Carburized, Hardened and Honed. Clutch Drum Internal Splines Shaped and Carbonitrided.

Fig. 8-15—How welding methods are used to assemble a clutch drum unit. Gears and shafts are forgings; drums are seamless tubing.

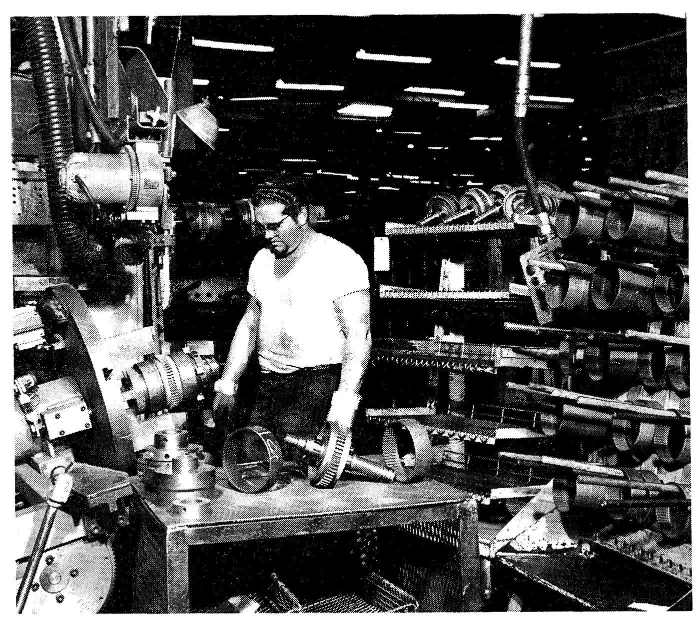


Fig. 8-16—Submerged-arc welding of (2) clutch drums to welded gear assembly as shown in Fig. 8-15.

Precision Boring-Milling Machine Spindle Drive

Thirty-eight spindle speeds in two separate ranges are provided by this multiple gear transmission. There are 19 speeds in the standard range—from 14 to 1,450-rpm; and 19 speeds in the high range—from 17 to 1,660 revolutions per minute. Gears are all helical and in constant mesh. There are 16 gears on four shafts. Speed changes are made by a pushbutton that energizes a previously-selected solenoid valve to actuate and de-actuate hydraulically-operated clutches. The spindle drive motor is a standard 1,800-rpm, a.c. motor with rated output of 20-horsepower.

Gear Data: 10-NDP; 20°NPA; 18°-R. & L. H. Helix Angles.

Gear Material: Hy-10 B3X.

Processing: Hobbed, Martempered and Shaved.

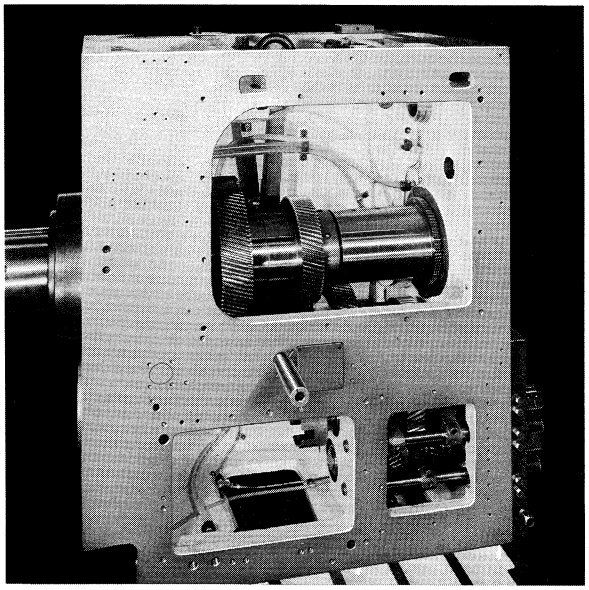


Fig. 8-17—*Courtesy DeVlieg Machine Co.*

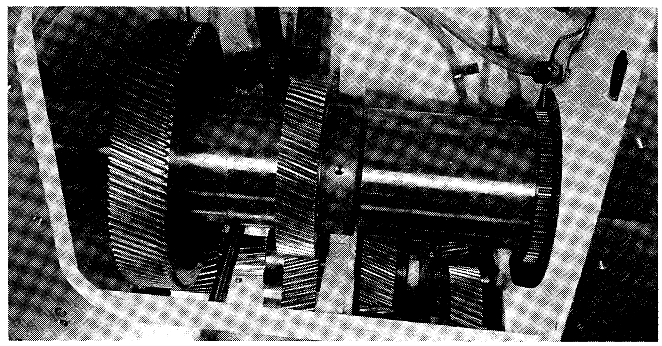


Fig. 8-18—Closeup view of transmission gearing in boring machine spindle head.

Crawler Tractor Power Shift Transmission

Five planetary gear sets provide three speeds in both forward and reverse directions in the Powershift Transmission used in the Caterpillar D9 Series G Tractor.

Shifting is accomplished by a single lever which controls clutches that are an integral part of the transmission. The control system allows changes in speed range or direction to be made without braking, decelerating or pausing in neutral. Input to the transmission is the output of a turbocharged diesel engine that develops 385 flywheel hp. at 1,330 revolutions per minute.

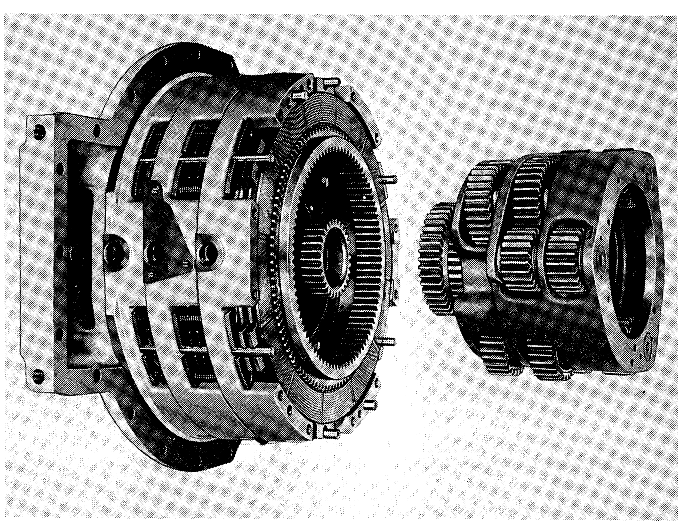


Fig. 8-19—*Courtesy Caterpillar Tractor Co.*

Gear Reductions

Speed	Forward	Reverse
1st	1.688 to 1	1.359 to 1
2nd	0.951 to 1	0.766 to 1
3rd	0.604 to 1	0.487 to 1

Gear Data: 6-DP; 20°-PA; Spur.

Gear Material: Low-Carbon Alloy Steel. Internal Gears Nitrided, All Others Carburized and Hardened.

Processing: External Teeth Hobbed, Shaved, Heat-Treated, Honed. Internal Teeth Shaper Cut, Shaved, Heat-Treated and Honed.

Gas Turbine Reduction and Accessory Drive Gears

This prototype gas turbine engine, intended for truck and bus use, develops 375 hp at a power turbine speed of 31,650-revolutions per minute. Gears reduce this speed to 3,000-rpm input to the remainder of the vehicle power train and also to provide a maximum speed of 6,700-rpm for operation of accessories.

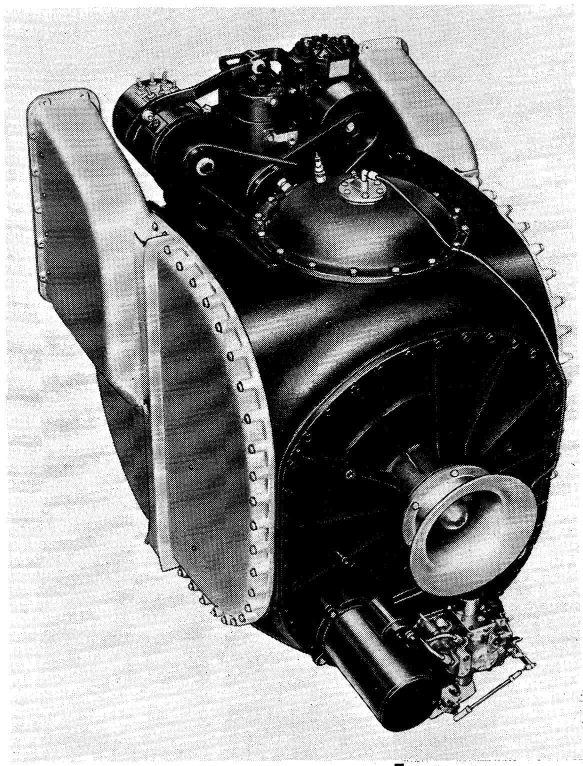


Fig. 8-20—Compared with conventional heavy truck engines of equivalent power, the 375-hp gas turbine engine, above, is said to be 2,000-lb. lighter, occupy less space, and offer low noise and negligible exhaust emission levels. It weighs 1,700-lb. installed with a length of 40-in. and height of 39-inches.

Gear Data: 19.50-NDP; 20°-NPA; 19°36′41″—R. & L. H. Helix Angle.

Number of Teeth: Driving Pinion, 26-Teeth; Reduction Gear, 274-Teeth; Accessory Drive Gear, 123-Teeth.

Gear Material: Pinion and Reduction Gear, SAE 6150; Accessory Gear, SAE 1065.

Processing: Pinion and Gear Hobbed, Shaved, Nitrided and Honed. Accessory Gear Hobbed, Crown-Shaved and Tufftrided.

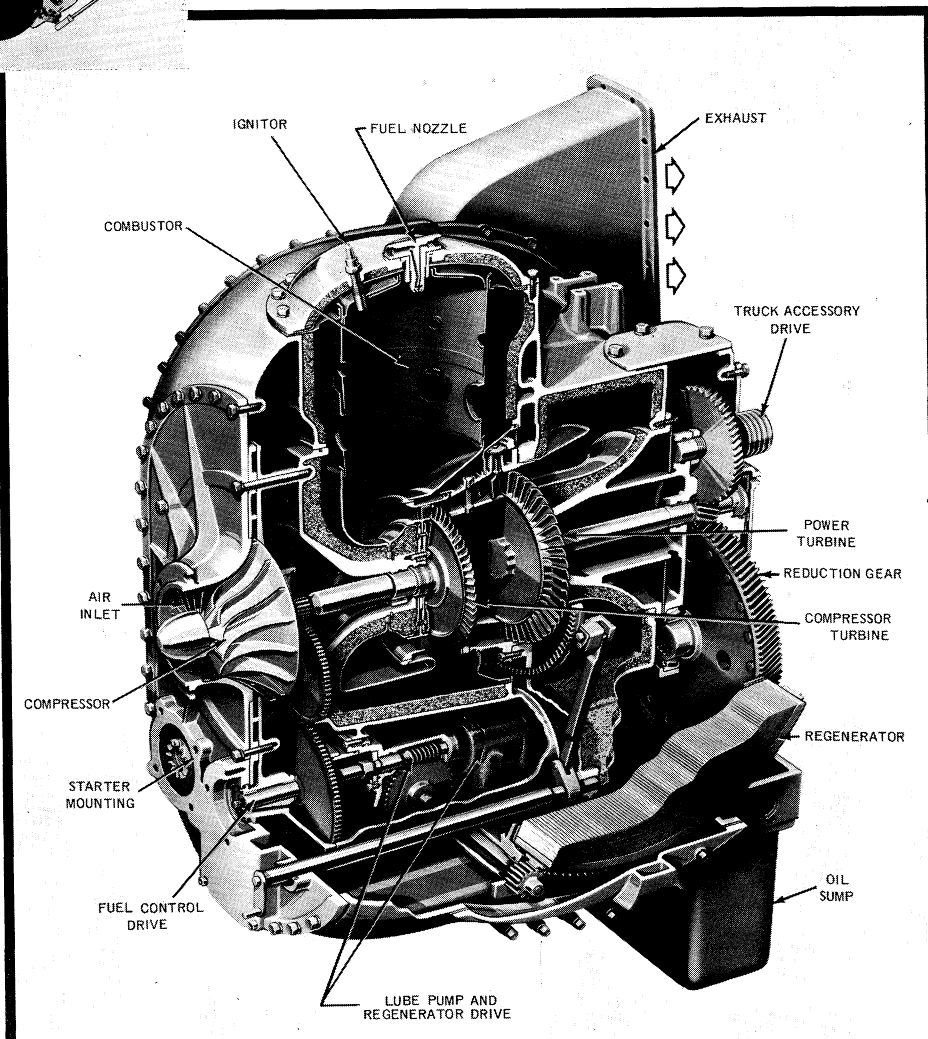


Fig. 8-21—Cutaway view of 375-hp gas turbine engine, right, shows relative simplicity of design and location of reduction and accessory drive gear described here. *Courtesy Ford Motor Co.*

Four-Speed Manual Automobile Transmission

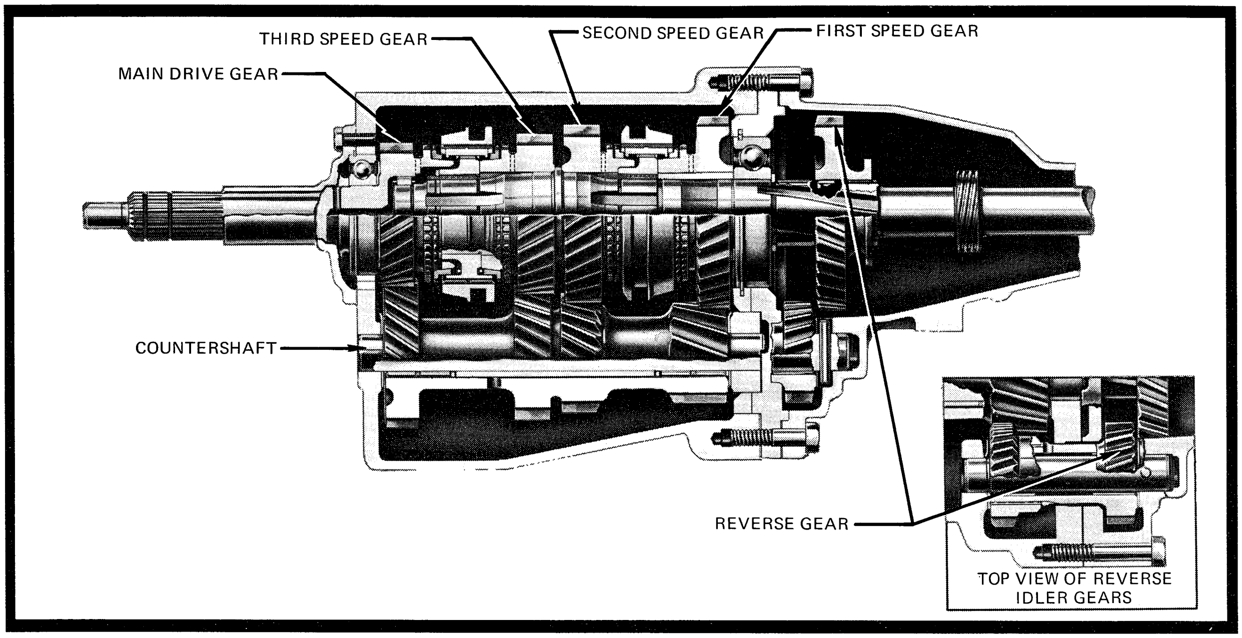


Fig. 8-22—*Courtesy Chevrolet Motor Div., GMC.*

Four speeds forward and a single-speed reverse are provided by this constant-mesh, all-helical, full-synchromesh transmission. Designed for use with gasoline engines with output speeds up to 8,000-rpm, rated torque input is 400 ft.-pounds. There are two choices of gear ratios, as follows: 1st Speed, 2.25 or 2.20 to 1; 2nd Speed 1.88 or 1.64 to 1; 3rd Speed, 1.46 or 1.28 to 1; 4th Speed, 1.00 to 1 and Reverse, 2.59 or 2.27 to 1.

Gear Design Data:
Main Drive Gearset: 9.36-NDP; 16°-NPA, 39.35°-R. & L. H. Helix Angle.
Reverse Drive Gearset: 9.11-NDP; 16°-NPA, 25.48°-R. & L. H. Helix Angle.
First Speed Gearset: 9.33-NDP; 14½°-PA, 26.38°-R. & L. H. Helix Angle.
Second Speed Gearset: 8.95-NDP; 14½°-PA, 31.45°-R. & L. H. Helix Angle.
Third Speed Gearset: 9.46-NDP; 14½°-PA, 36.07°-R. & L. H. Helix Angle.
(All Gears Operate on Spread Center Distance).

Gear Material: SAE 8620 Modified Steel.

Processing: Hobbed or Shaped, Crown-Shaved, Carburized, Hardened (59-63Rc), Roots and Teeth Shot-Peened.

Turboprop Engine Power Turbine Gear Train

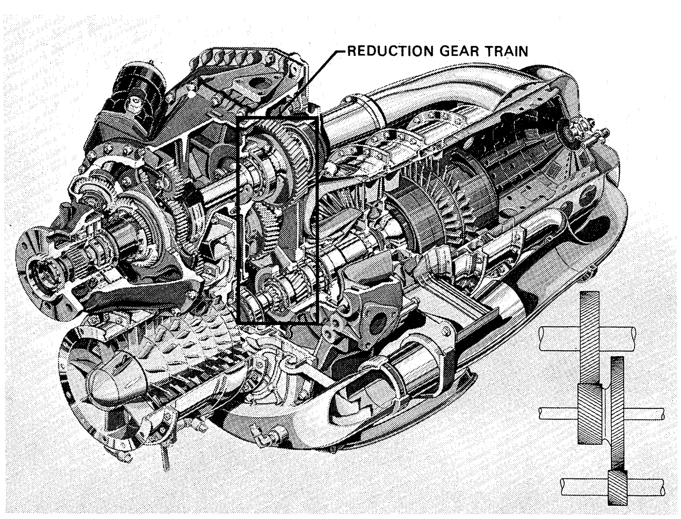


Fig. 8-23—*Courtesy Detroit Diesel Allison Div., GMC.*

Maximum speed of the power turbine in the 317-hp. Model 250 Turboprop Engine is 35,000 revolutions per minute. Reduction gearing permits either a propeller or helicopter rotor to operate at an efficient speed. Reduction from 35,000-rpm to 6,000-rpm is achieved with two pairs of helical gears. Overall reduction ratio is 5.833 to 1; first reduction is 3.5 to 1 and second reduction is 1.665 to 1.

Gear Data: 6-NDP; 22½°-PA; 25°-R. & L. H. Helix Angle.

Gear Material: SAE 9310.

Processing: Hobbed or Shaper-Cut, Carburized, Hardened, Ground & Honed.

Marine Turbine Reduction Gears

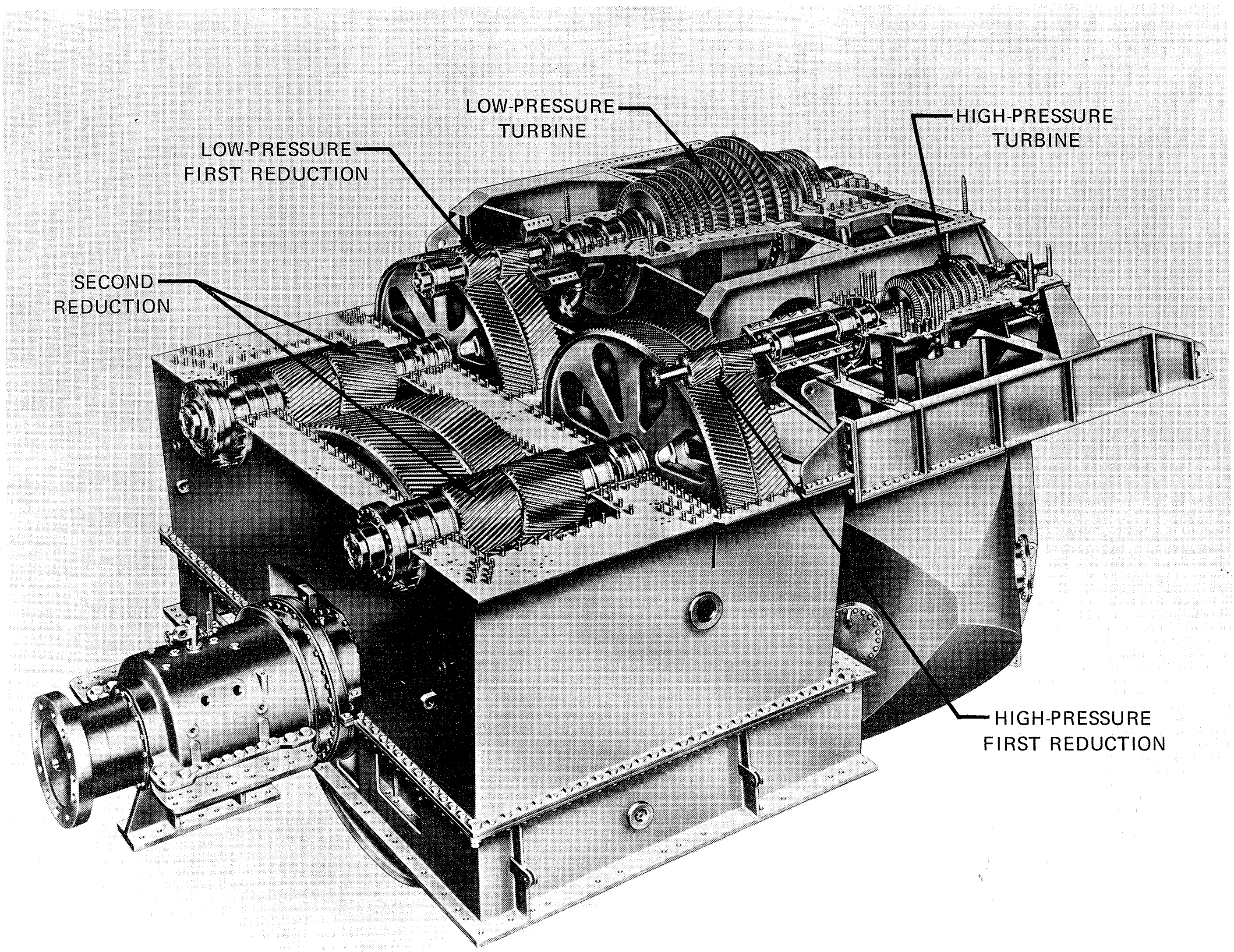


Fig. 8-24—*Courtesy Marine Div., Westinghouse Electric Corp.*

Large double-helical gearsets are used to combine the inputs from a high-pressure, high-speed steam turbine and low-pressure, high speed steam turbine and reduce speed to a value which is efficient for driving the ship's propeller. The combined horsepower input is 22,000. The high-pressure input speed is 6,437-rpm; low-pressure input speed is 4,379-rpm; and output speed to the propeller is 115-revolutions per minute.

Reduction is performed in two steps. First reduction ratio for the high-pressure turbine is 7.456 to 1; first reduction ratio for the low-pressure turbine is 5.073 to 1. Second reduction ratio. where both inputs have been combined, is 7.507 to 1. Overall reduction is 55.972 to 1 for the high-pressure turbine and 38.083 to 1 for the low-pressure turbine.

Gear Data:
First Reduction (High and Low-Pressure): 5-NDP; 14½°-NPA; 30°-R. & L. H. Helix Angle.
Second Reduction: 3½-NDP; 14°-NPA 30°-R. & L. H. Helix Angle.
Number of Teeth: High Pressure first reduction, 46 and 343; Low Pressure First Reduction, 55 and 279; Second Reduction, 71 and 533.

Gear Material: Carbon-Molybdenum, Vanadium Alloy Steel Forgings.

Processing: Rough & Finish Hobbed, Shot-Peened, Crown-Shaved.

Fine-Pitch Instrument Gears

The Model 4068 Attitude-Director used in the United States' Lunar Excursion Modules provides the astronauts with pitch, roll and yaw attitude information during lunar flight, lunar landing and rendezvous with the Command Module. The system includes special anti-backlash gears where the total face width of the gear is actually made up of two thin gears which are spring loaded to exert pressure in opposite directions and thus take up all back-lash.

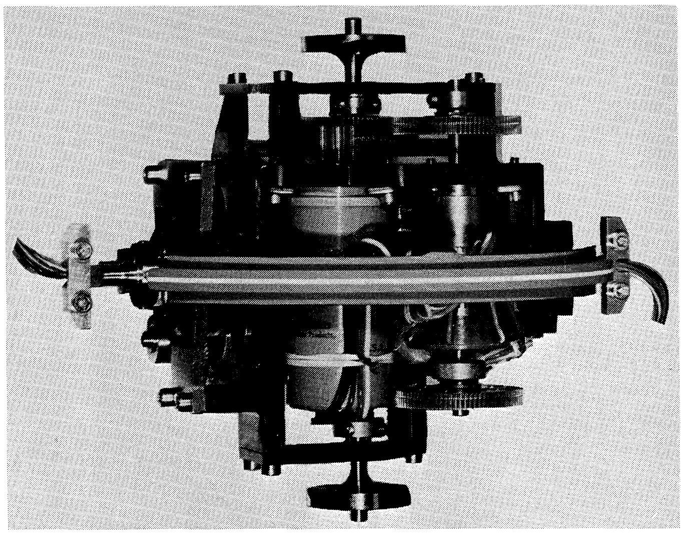


Fig. 8-25—*Courtesy Instrument Div., Lear Siegler, Inc.*

Typical of the gearing used, is the train shown at the top of the accompanying photo. A small gearhead motor with maximum output speed of 70 rpm and stall torque of 0.136 oz-in. drives a meshing anti-backlash gear; ratio of pinion and gear is 7:1 so the gear turns at a maximum speed of 10 rpm. Another gear on the same shaft as the pinion-driven gear meshes in 1:1 ratio with another anti-backlash gear on the shaft of a resolver. In operation, the drive motor is started by an error signal from the resolver. The motor then drives until the resolver shaft is turned to a position such that the error signal is neutralized. In the process, an indicator moves to show the degree of yaw, pitch or roll corresponding to the original error signal.

Gear Data: 96-DP; 20°-PA; Spur.

Gear Material: Stainless Steel and Aluminum.

Processing: Hobbed.

Heavy-Duty 18-Speed Dumper-Mixer Transmission

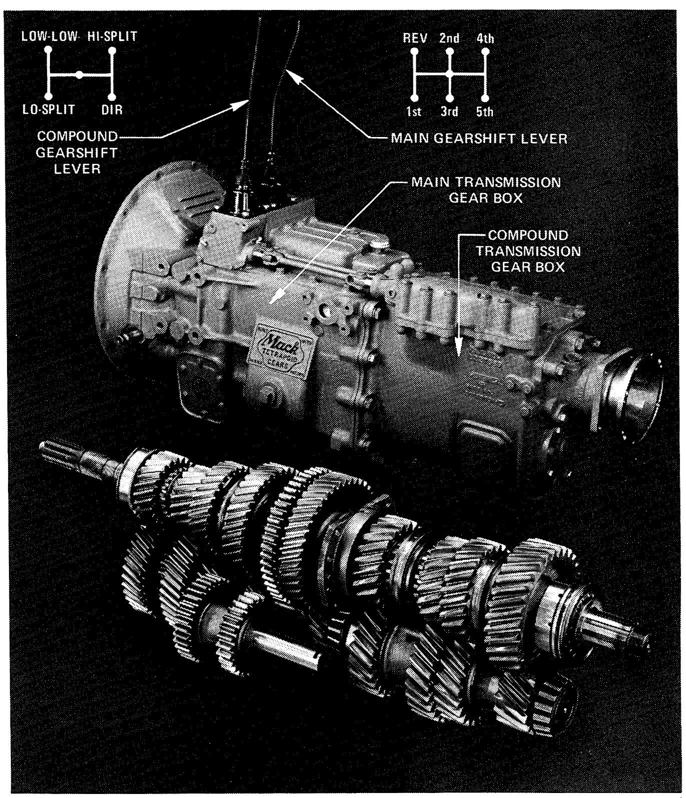


Fig. 8-26—*Courtesy Mack Trucks, Inc.*

This heavy-duty dumper-mixer transmission combines main and compound gearsets in one compact unit to give 18 forward and four reverse speeds for over-the-road and off-the-road use in vehicles produced by its builder. The transmission is used with engines developing approximately 250-hp at output speeds of 2,100-revolutions per minute.

Transmission Speed Ratios:

Main Trans. Shift Positions	Compound Shift Positions			
	High Split	Direct	Low Split	Low Low
5th	0.70	0.84	1.01	2.13
4th	0.84	1.00	1.20	2.53
3rd	1.47	1.76	2.10	4.44
2nd	2.61	3.13	3.74	7.92
1st	4.55	5.45	6.52	13.80
Rev	4.55	5.45	6.52	13.80

Gear Design Data: 4 to 6-NDP; 20 to 25°-PA; 23 to 27°-R. & L. H. Helix Angle.

Gear Material: SAE 4620H, 4817H, 8620H.

Processing: Hobbed or Shaper-Cut, Shaved, Carburized, Hardened, Honed.

Chapter NINE

Shaving the small gear on a one-piece HY-10-B3X steel shoulder gear to a 0.0005-in. runout with a 7-in. dia. cutter mounted on a special internal cutter head on an external rotary gear shaver. The 30-teeth are 10-DP, 20°-PA.

Special Gear Developments

The manufacture of special gears for research, development and prototype work requires a special production facility with a variety of lathes, grinders, and gear cutting, tooth rounding and finishing equipment as well as specialized gear tooth dimensional and surface measurement inspection equipment. Another important requirement for such work is an in-house heat-treatment department to provide all types of controlled heat treatment, annealing, and surface conditioning processes such as carburizing, carbonitriding, and Tufftriding, to make high performance gears in small quantities with maximum accuracy.

Over the years thousands of special gear sets have been manufactured and developed in the Red Ring development laboratory. When such gear development work is properly carried out, realistic performance tests can be made, final tooth forms and heat treatment methods can be established, and processing sequences and equipment can be recommended.

This chapter gives gear data, material, heat treatment processes and tooth processing sequences for a selected variety of prototype gear developments.

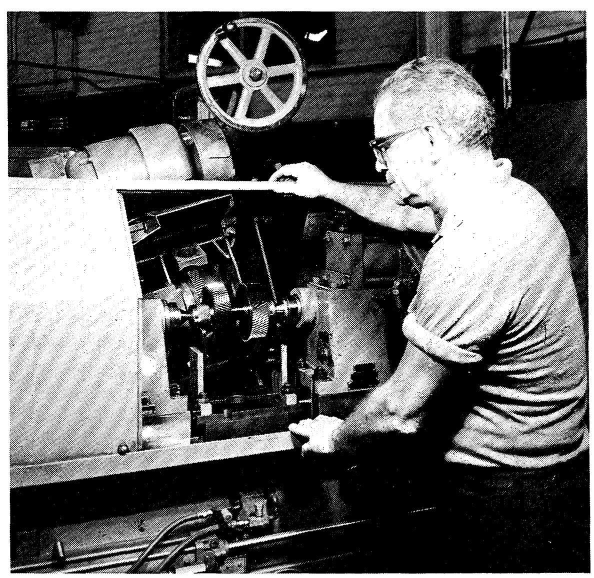


Fig. 9-1—Honing the teeth on an SAE 9310 double-helical gear that is hobbed; carburized to provide a 0.030-in. case; and hardened to 58-62Rc hardness. Then the teeth are ground. Honing the ground gear teeth provides the accuracy and fine surface finish required for quiet underwater operation. The 73 gear teeth are 14-NDP, 20°-NPA and 22°18′ helix angle.

Truck Turbine Planetary Reduction Gearing

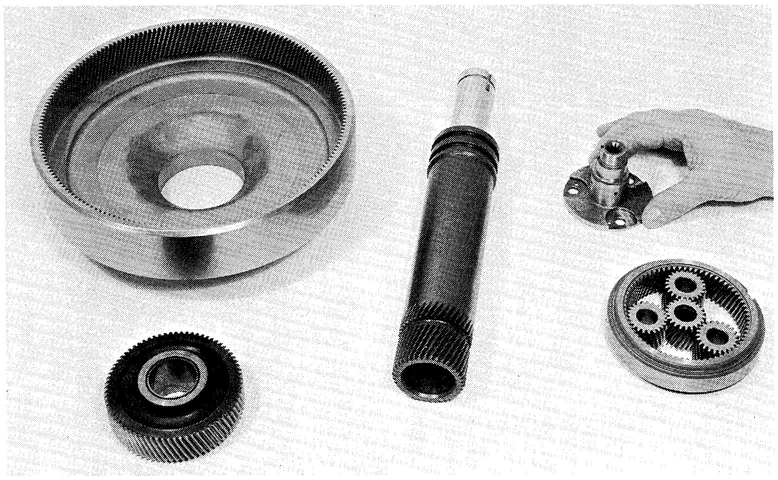


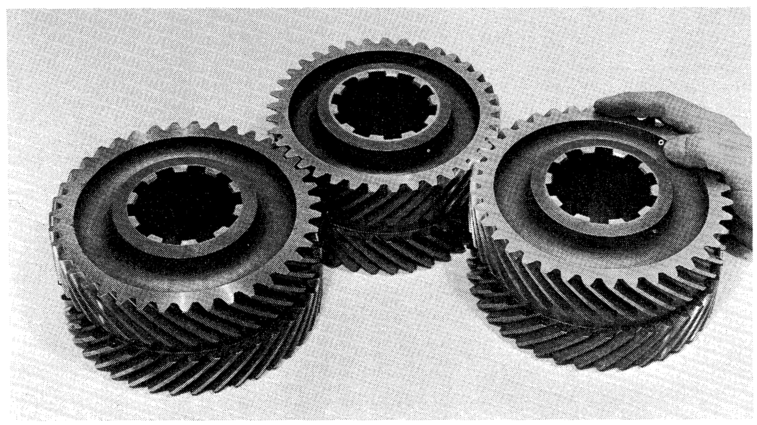
Fig. 9-2—These two sets of planetary reduction gears are applied in a split torque arrangement in a gas turbine for truck and bus applications.

Gear Data: 20-NDP; 20°-NPA; 18°30′ Helix Angle.
Power Drive Train, Left, (32,000-rpm Input)
Sun Gear: 38-Teeth; SAE 4140, 31-34Rc; Shaved; Tuftrided; Honed.
Planet Pinion: 73-Teeth; SAE 4140, 31-34Rc; Crown-Shaved; Tufftrided;Crown-Honed.
Internal Ring Gear: 184-Teeth; SAE 4140; Heat-Treated to 35Rc; Shaped and Shaved.
Gasifier Accessory Drive, Right, (35,000-rpm Input)
Sun Gear: 22-Teeth; SAE 4140, 31-34Rc; Hobbed; Shaved; Carbonitrided; Honed.
Planet Pinion: 23-Teeth; SAE 8620; Hobbed; Crown-Shaved; Carbonitrided; Crown-Honed.
Internal Ring Gear: 68-Teeth; SAE 4140, 31-34Rc; Shaper-Cut; Shaved; Nitrided; Honed.

Double-Helical Truck Turbine Transfer Gears

Fig. 9-3—These gears were designed to operate in a transfer box between a 600-hp. truck gas turbine and a three-speed power-shift transmission. The gears operate at 3,600 revolutions per minute. They have splined holes for running the truck accessory drives.

Gear Data: 37, 38 and 39-Teeth; 8-NDP; 17½°-PA; 45°-R. and L. H. Helix Angles.
Material: SAE 4150, Heat-Treated to 34-38Rc Core Hardness.
Processing: Teeth Shaved, Nitrided and Honed.



Wankel Engine Gearing

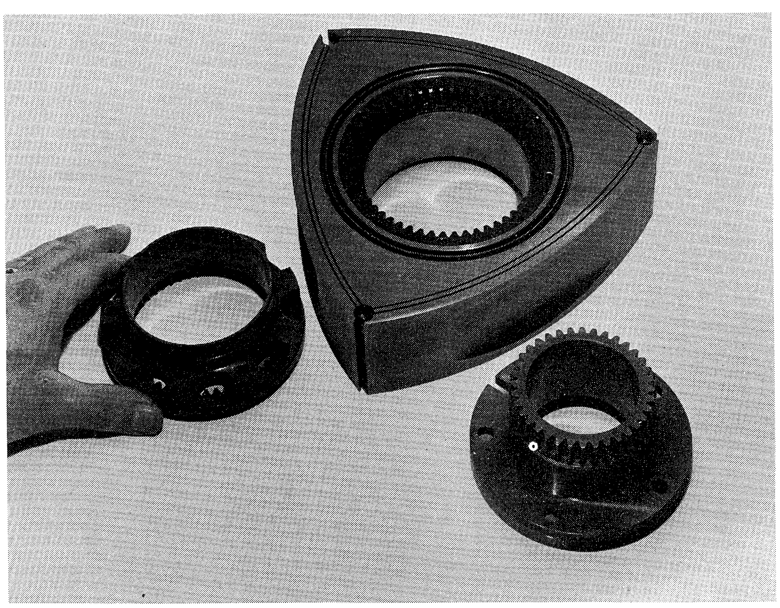


Fig. 9-4—Experimental parts for Wankel rotary internal combustion engines. The triangular rotor has its internal gear bolted into a bored recess. The other internal and external gears are made in one piece.

Gear Data: 12.8205 and 14.5143-DP, 20°-PA.
Material: SAE 6150, 35Rc; Tufftrided After Shaving.
Processing: Teeth Shaper-Cut and Shaved.

Fine-Pitch Automotive Transmission Gearing

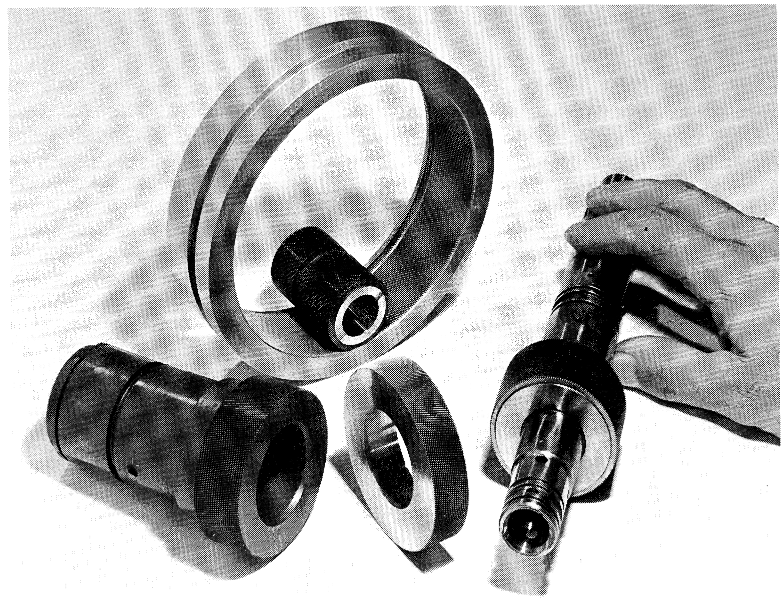


Fig. 9-5—This unusual automotive automatic transmission planetary gearset has teeth so fine that its operation was questioned. Actually the gear set functioned properly and the transmission performed well on the test track.

Gear Data: 96-DP, 20° PA.
Material: SAE 8620, Carbonitrided, Heat-Treated.
Processing: Hobbed or Shaper-Cut; Lapped After Carbonitriding. (Lapping Tool at Front Center of Illustration).

Automotive Gas Turbine Reduction Gears

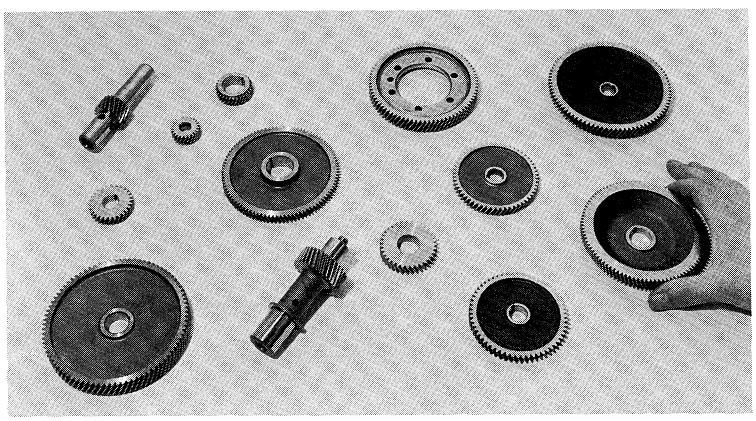


Fig. 9-6—These thirteen gears were made for a research gas turbine designed to drive a small, light car. The gears include power turbine 1st and 2nd reduction gears (two in each set), accessory drive 1st and 2nd reduction gears (two in each set), accessory drive 3rd and 4th reduction gears, and three accessory gears.

Gear Data: 20 to 86-Teeth; 20 and 24-NDP; 20°-NPA; 21° Approximate Helix Angle.
Power Turbine and Accessory Drive Reduction Sets
Material: SAE H-13 Steel, Core Hardness 30-35Rc.
Processing: Teeth Shaved, Nitrided and Honed.
Accessory Drive and Accessory Gears
Material: SAE 6150 Steel, Core Hardness 32-35Rc.
Processing: Teeth Shaved, Tufftrided and Honed.

Stationary Gas Turbine Planetary Gear Reduction

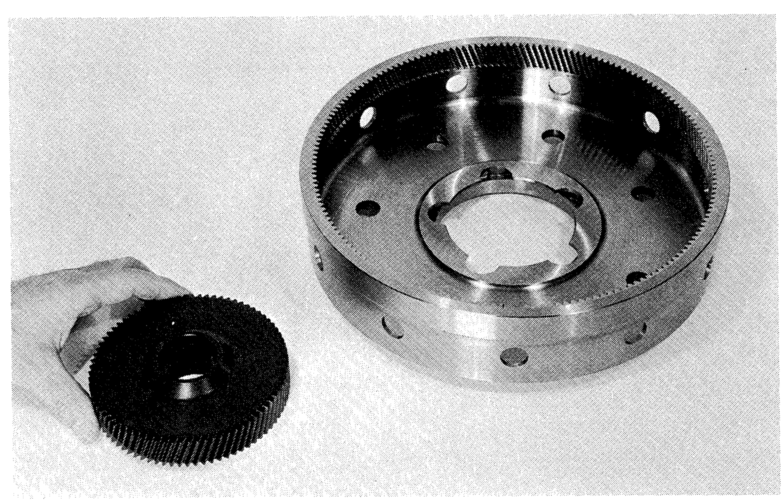


Fig. 9-7 —This planetary gear reduction has three planet pinions, left, and an internal gear, right. It is used on a stationary gas turbine-electric generator in-line application.

Gear Data: 16-NDP; 25°-PA; 20°-Helix Angle.
Material (Internal Gear and Planet Pinion): Through-Hardened, High Carbon Steel; 32-38Rc.
Processing: Internal Gear Shaper-Cut and Shaved After Heat Treatment; Planet Pinion Hobbed, Heat-Treated, Ground and Honed.

Winocircondu (Circular-Arc) Gear Set

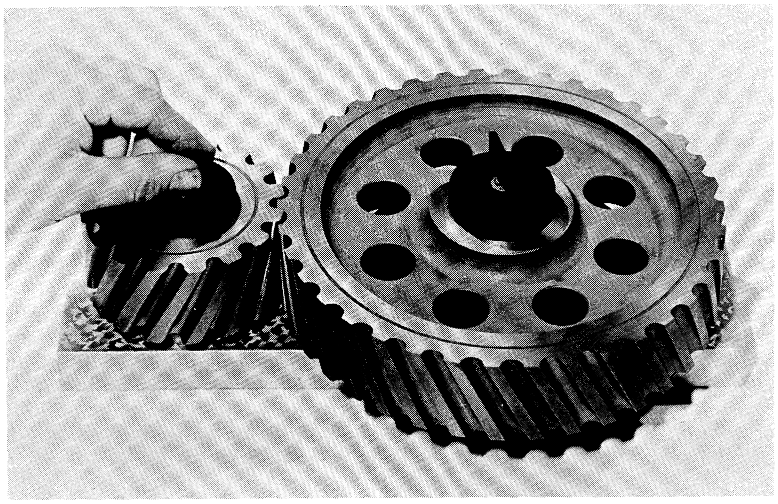


Fig. 9-8—Circular-arc gears are reputed to carry from three to five times the tooth flank loading of gears with involute tooth shapes. They are also said to retain up to 10 times the thickness of oil film compared with involute teeth.

Tooth overlap is achieved by helical action only and the gears are sensitive to center distance variation. As a result, these gears require the ultimate in form, spacing and runout accuracy to perform efficiently, effectively and quietly.

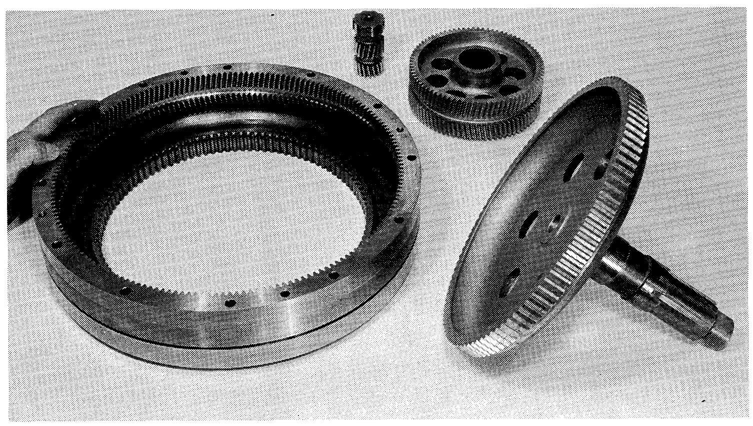
The term 'Winocircondu' was coined by National Broach engineers. It is derived from the inventors Wildhaber and Novikov; and the trade names Circarc, Conformall and Duracurve.

The set illustrated was built for a helicopter drive test application.

Gear Data:
Pinion: 16-Teeth; 4.5-NDP; 25°-PA; 25°13′-R. H. Helix Angle at 3.556-in. PD.
Gear: 38-Teeth; 4.5-NDP; 25°-PA; 25°13′-L. H. Helix Angle at 8.444-in. PD.
Material: SAE 9310.
Processing: Hobbed, Carburized, Hardened and Ground.

Double-Helical Planetary Gear Set

Fig. 9-9—This experimental planetary gear set was designed for a 30,000-rpm sun gear input speed and a 3,000-rpm output ring gear speed. The internal ring gear is assembled by bolting three pieces together: A right and left-hand steel internal ring gear, and a pearlitic malleable iron body that is, splined to mount on a pearlitic malleable flanged output shaft.



Gear Data: 14-NDP, 20°-NPA, 10°-Helix Angle.
Sun Gear: 18-Teeth; SAE 8620-H Steel; Shaper-Cut, Shaved, Carburized and Hardened.
Planet Pinion: 81 Teeth; Same as Sun Gear.
Internal Ring Gears: 180-Teeth; Right and Left-Hand Helix Angle; SAE 4140, 34-38Rc; Shaper-Cut and Shaved.
Output Shaft: Pearlitic Malleable Iron; 105-Tooth Hobbed External Involute Spline.

Cluster Gear For 5-Speed Tractor Transmission

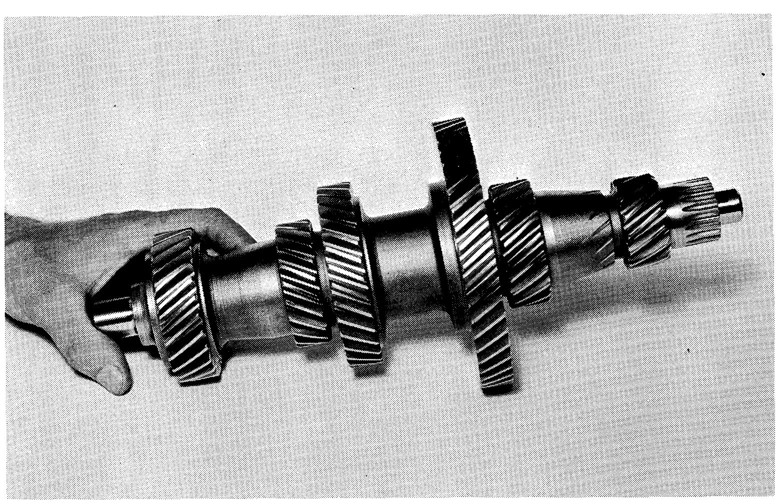


Fig. 9-10—This 13¾-in. long cluster gear for a five-speed tractor transmission challenges the ingenuity of any gear manufacturer. It has six gears and a spline that must run concentric within 0.003-in. true indicator reading. As manufactured in prototype quantities, all tooth finishing operations were completed before heat treatment.

Gear Data: 10-NDP, 20°-NPA.
No. Teeth, Size and Helix Angle:

Gear (L. to R.)	Pitch Dia. (In.)	No. Teeth	Helix Angle
Third	3.009	26	30° 13′ 53″
Reverse	2.461	21	31° 25′ 26″
Fourth	3.703	32	30° 13′ 53″
Fifth	5.507	47	31° 25′ 26″
Second	2.461	21	31° 25′ 26″
First	1.715	15	28° 59′ 40″

Material: SAE 5140, Carburized, and Hardened.
Processing: Hobbed and Shaved Before Heat Treatment.

RED RING

Chapter TEN

Products and Services for Gear Manufacturers

INDEX

ABOUT NATIONAL BROACH . . .

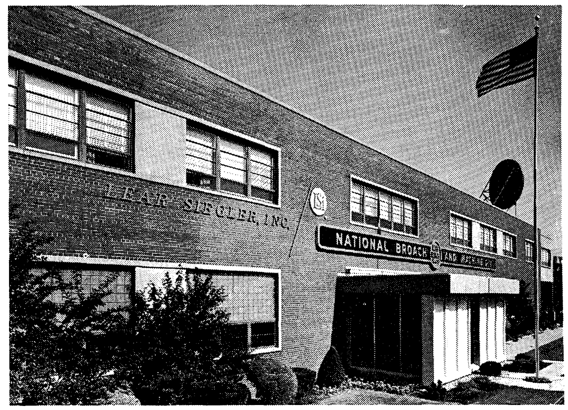


Fig. 10-1 Entrance to the National Broach & Machine Div. main office lobby.

National Broach and Machine Division began operations in 1925 as the National Broach Company in Dayton, Ohio. For several years National Broach designed and built only broaching tools.

In 1929 the company's name was changed to National Broach & Machine Company. Operations were moved from Dayton into the present plant location at 5600 St. Jean Avenue in Detroit.

The company name-change was the result of the development of a new crossed-axes principle of gear finishing that led to the creation of the rotary gear shaving process in the 1930's.

In 1968 National Broach & Machine Co. was acquired by Lear Siegler, Inc. Today National Broach & Machine Division operates as a member of the Commercial and Industrial Group of Lear Siegler, Inc.

The company's plant facilities have been continually expanded over the years until it occupies about 130,000 square feet of floor space in three buildings. Included in these facilities are machine shops, machine assembly, extensive equipment for heat-treating high speed steel; broach grinding; gear research and development; air-conditioned, temperature-controlled rooms for shaving cutter and rolling die grinding and inspection; engineering department; and general office.

National Broach designs and builds gear production equipment in a wholly-owned subsidiary, Precision Gear Machines and Tools, Ltd., in Coventry, England.

National Broach Contributions to Gear Practice

Rotary Crossed-Axes Gear Lapping
Rotary Crossed-Axes Gear Shaving
Elliptoid (Crowned) Tooth Form
Gear Sound-Testing
Precision Gear Broaching
Shaving Marine Propulsion Gears
Universal Diagonal Shaving
Automatic Loaders for Shavers
Unitized Broaching Fixtures
Automatic Gear Inspection
Rotary Hard Gear Honing
Helical Gear Pot Broaches
Full-Form Finish Gear Broaches
Concentricity Gear and Spline Broaches
Diagonal Crown-Shaving
Electronic Gear Gaging Centers
Precision Expanding Arbors
Gear Grinding Centers
Annular Ring-Type Gear Pot Broaching
Ring Rounding of Internal Gears
High-Speed Gear Burnishing
Vertical Double-Die Gear Rolling
Large Gear Honing
Single-Die Gear Rolling
Push-Up Pot Broaching
Combination Shaver-Roller

Fig. 10-2 Aerial view of the National Broach Detroit, Michigan plant facilities.

Rotary Gear Shaving

The rotary gear shaving process was introduced to industry by National Broach & Machine Division in 1932. The process uses a helical gear-shaped, high speed steel, hardened-and-ground cutter having tooth flanks with multiple gashes (serrations) that act as cutting edges. The cutter is meshed and rotated in crossed axes relationship with external or internal spur or helical gears while the work gear is reciprocated across the face of the cutter. The center distance between the cutter and work is reduced to remove metal from the work gear tooth surfaces in the form of minute, hair-like chips.

Rotary shaving is a low-pressure, free-cutting finishing operation that can be applied to finish pre-formed, hobbed or shaped gear teeth before heat treatment to:

1. Correct eccentricity and errors in index, helix angle and tooth profile.
2. Improve tooth surface finish.
3. Maintain tooth size.
4. Eliminate tooth end bearing conditions by producing an Elliptoid (crowned) tooth form.

Machine Types

Red Ring gear shavers are made in four basic Types: 1. The knee and column type for small gears up to 18-in diameter, 2. The tunnel type for gears up to 24-in. diameter, 3. The vertical type for gears up to 100-in. diameter, and 4. The horizontal type for gears up to 180-in. diameter.

Work Diameter Range (In.)	External Gears	Internal Gears
½ to 8	GCU-8 In. (Fine Pitch)	
1 to 8	GCU-8 In.	
3 to 8		*GCU-8 In.
1 to 12	GCU-12 In. GCY-12 In. GFC-12 In.	
3 to 12		GCR-12 In. *GCU-12 In. *GCY-12 In.
3 to 18	GCU-18 In. GCY-18 In. GFC-18 In.	*GCU-18 In. *GCY-18 In.
3 to 24	GCX-24 In.	
6 to 24		*GCX-24 In.
4 to 36	GCM-36 In.	*GCM-36 In.
	GCJ-36 In.	*GCJ-36 In.
5 to 36	GCQ-36 In.	GCQ-36 In.
4 to 48	GCM-48 In.	*GCM-48 In.
	GCJ-48 In.	*GCJ-48 In.
22 to 48	GCQ-48 In.	GCQ-48 In.
24 to 96	GCK-96 In.	
25 to 100	GCT-100 In.	GCT-100 In.
48 to 120	GCK-120 In.	
30 to 180	GCK-180 In.	*Special Attachments

Fig. 10-5 Selection table for Red Ring rotary gear shaving machines.

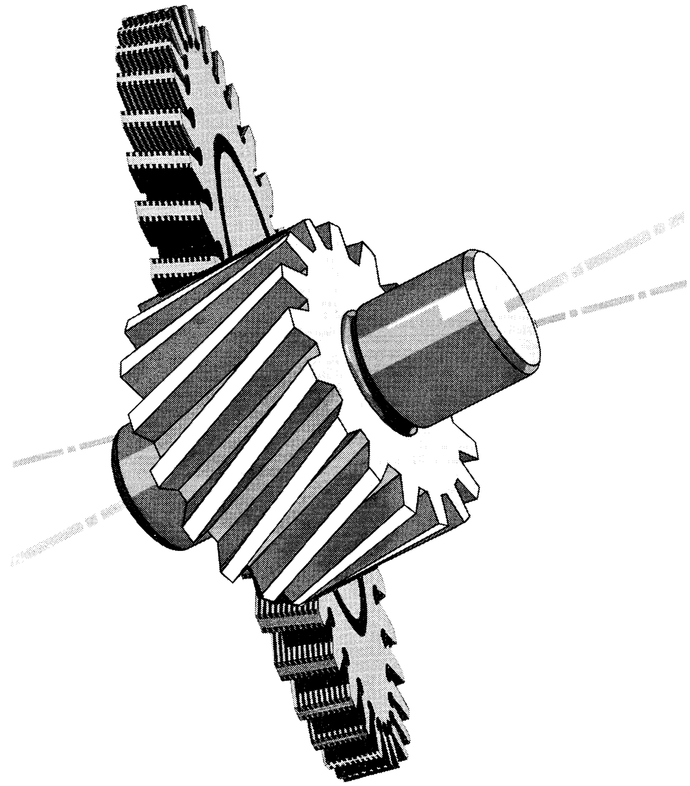


Fig. 10-3 The crossed-axes relationship between a shaving cutter and the work gear with rotary gear shaving.

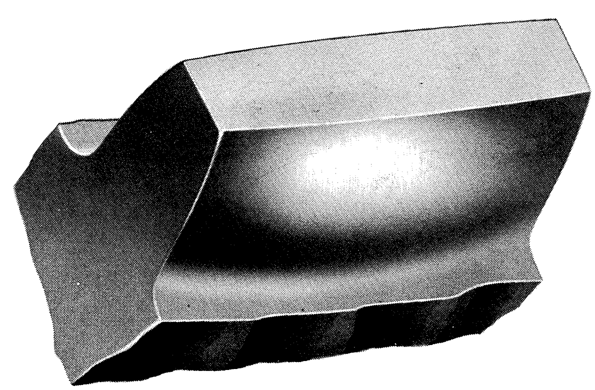


Fig. 10-4 The Red Ring Elliptoid (crowned) tooth form.

Knee and Column Machines for External Gears

Red Ring knee and column types of gear shaving machines have the shaving cutter mounted above the work gear. This design enables the cutter to be mounted in a small power-driven cutter head above the larger and heavier up-feed reciprocating table that supports the head and tailstocks.

In this arrangement, gravity forces act to minimize table clearances. The table assembly, for which running clearances are provided, is always in compression. In addition, there is no danger of dropping a gear on the cutter and damaging it during loading. Also, chips fall down away from the cutter during shaving and do not clog the serrations.

In the rotary shaving process on knee and column shaving machines, the work is up-fed into the shaving cutter at both ends of the strokes, and the direction of cutter rotation is also reversed at each stroke end.

Shaving Methods

Three methods of traversing the work gear back and forth across the cutter are available on these machines. These are:

1. The conventional (axial traverse) method in which the work is reciprocated in a path parallel to the work axis. This method is used in medium and wide-faced gears. Crowning is by rocking table action in the plane of the work axis.
2. The diagonal (angular traverse) method in which the work gear is traversed across the cutter in a path at an angle to work gear axis. This method is used for high production application and uses a cutter that is wider than work gear. Crowning is by rocking table action in the plane of the work axis.
3. The tangential (right angle traverse) method in which the work table does not reciprocate and the work is passed across the face of the cutter at right angles to the work gear axis. Crowning is provided by using a shaving cutter with a reverse-crowned tooth form. This method uses a shaving cutter with differential serrations and is applied to finish shoulder gears.

Loading Methods

Three loading methods are applied to knee and column machines:

1. Manual. Here the work gear is loaded by the operator into the shaver, the tailstock is manually advanced and clamped, the coolant door is manually closed and the machine cycle pushbutton is operated.
2. Semi-Automatic. With this method the work is loaded by the operator on an approximate locator and the cycle pushbutton is operated. The work is automatically clamped by a power tailstock, the coolant door is closed by an air cylinder and the automatic machine cycle is initiated.
3. Fully Automatic. Here the shaving machine is equipped with an automatic loader whose magazine is kept full of gears by the machine operator or filled from an automatic line.

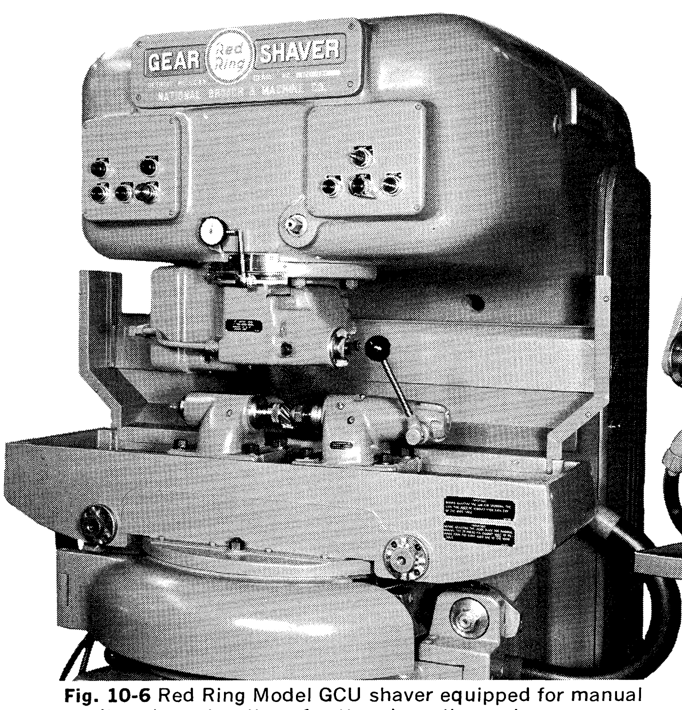


Fig. 10-6 Red Ring Model GCU shaver equipped for manual loading shows location of cutter above the work.

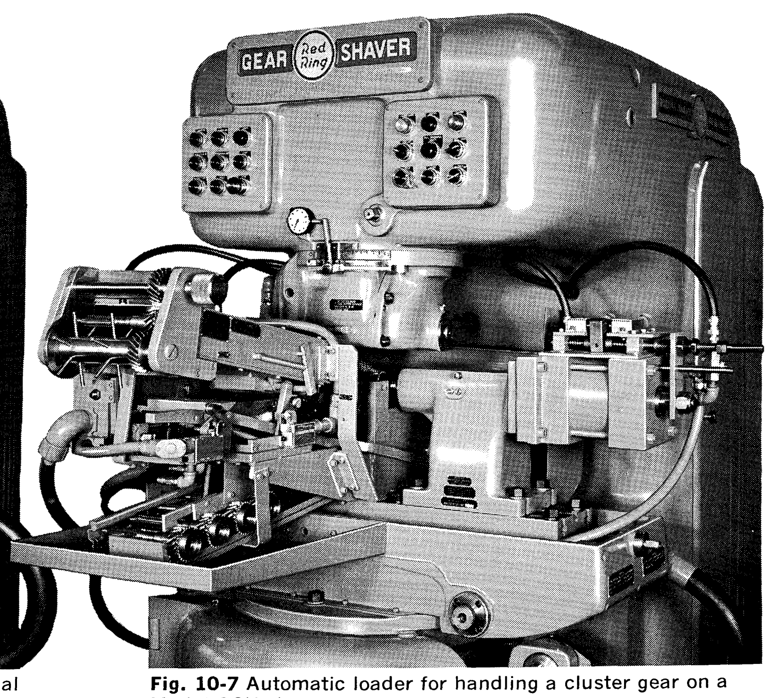


Fig. 10-7 Automatic loader for handling a cluster gear on a Model GCU shaver.

MODEL GCU UNIVERSAL DIAGONAL GEAR SHAVER

This rotary gear shaver can handle all types of spur and helical gears from ½-in. to 18-in. diameter having wide or narrow faces, close shoulders, plunged-hobbed teeth or long integral shafts.

It will shave straight, crowned, tapered or conical tooth forms; provides conventional, diagonal or tangential shaving methods; and can have manual, semi-automatic or fully automatic loading.

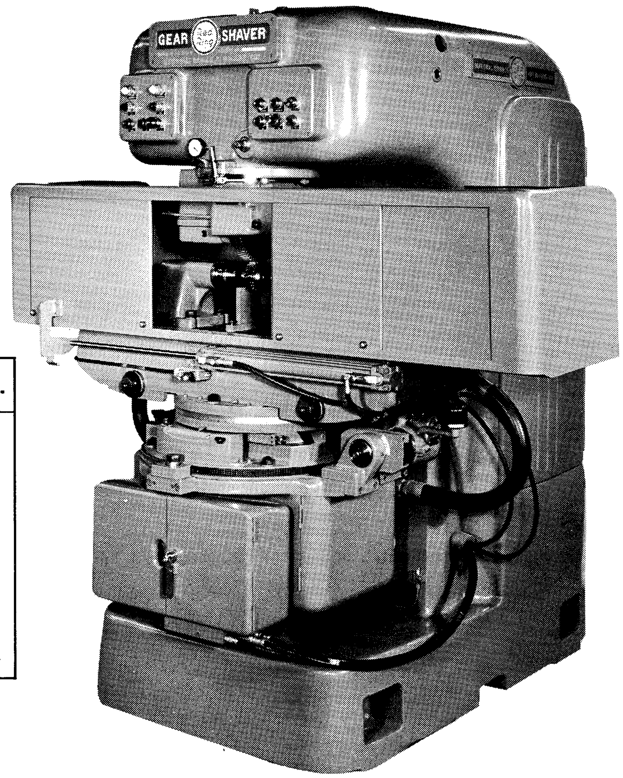


Fig. 10-8 Model GCU shaver equipped for semi-automatic loading.

Specifications

Model	GCU-8 in. (Fine Pitch)	GCU-8 in.	GCU-12 in.	GCU-18 in.
P. D. capacity (In.)	½ to 8	1 to 8	1 to 12	3 to 18
Max. O.D. (In.)	9¼	8¾	12¾	19¼
Pitch Range (D.P.)	10 to 48	4 to 16	4 to 16	3 to 16
Max. Table Travel (In.)	6	6	6	6
Max. Distance between work centers (In.)	23¾	24	23⅝	23⁷⁄₁₆
Distance Between Cutter and Headstock centerlines (In.)	2⁹⁄₁₆ to 8⅞	4¹⁄₃₂ to 9⅛	4¹⁄₃₂ to 11⅛	5¹⁄₁₆ to 14

MODEL GCY UNIVERSAL GEAR SHAVER

The Model GCY gear shaver is designed to reduce changeover time from one shaving job to the next. Dial-set shift-type transmissions replace conventional change gears for selecting table reciprocation speeds and cutter speeds.

Constant, low 45-in. work height is achieved by down-feeding the cutter head rather than up-feeding the worktable. Selective adjustable down-feed increments provide maximum flexibility in the shaving feed sequences.

The Model GCY can provide the three different shaving methods and can crown or taper-shave gear teeth.

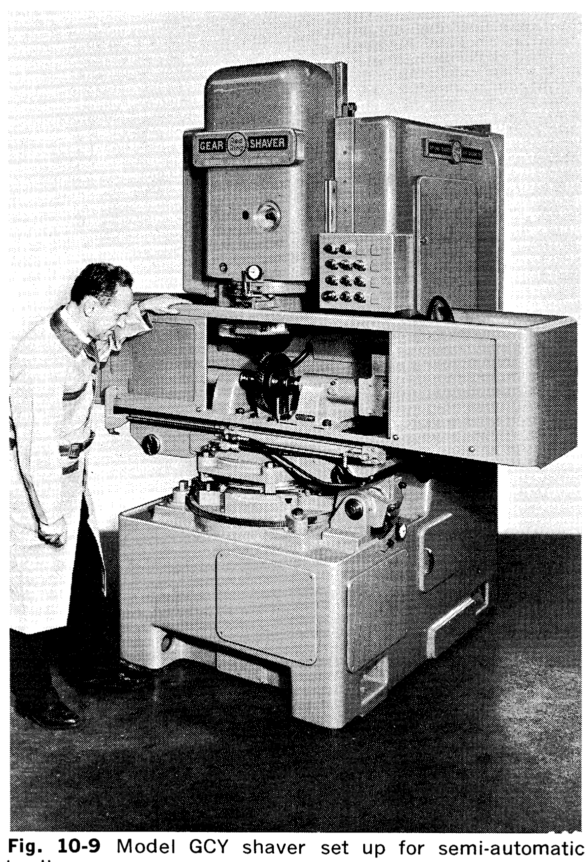


Fig. 10-9 Model GCY shaver set up for semi-automatic loading.

Specifications

Model	GCY-12 in.	GCY-18 in.
P. D. Capacity (In.)	1 to 12	3 to 18
Max. O.D. (In.)	12¾	19¼
Pitch Range (D.P.)	4 to 16	3 to 16
Max. Table Travel (In.)	6	6
Max. Distance Between Work Centers (In.)	17	18¹⁵⁄₁₆
Distance Between Cutter and Headstock Centerlines (In.)	4¹⁄₃₂ to 12¼	4½ to 15⅛

Knee and Column Machines—continued

MODEL GFC ROLLSHAVE

The Red Ring Model GFC ROLLSHAVE gear finishing machine is designed to perform either rotary gear shaving or roll-finishing operations by changing tooling and operating selector switches. It has all of the operating capabilities of the Model GCU gear shaver, but is provided with a special heavy-duty welded steel frame, heavy-duty cutter drive gearing and heavy-duty head and tailstock to resist the large forces produced by roll-finishing processes.

The knee is a special design that includes a unique combination upfeed mechanism that will produce the necessary large continuous upward forces for roll-finishing coarse-pitch gears and the incremental up-feed for gear shaving.

Specifications

(Shaving Only, see also Gear Rolling Machines)

Model	GFC-12 In.	GFC-18 In.
P. D. Capacity (In.)	1 to 12	3 to 18
Max. O.D. (In.)	12¾	19¼
Pitch Range (D. P.)	4 to 16	3 to 16
Max. Table Travel (In.)	6	6
Max. Distance Between Work Centers (In.)	17½	17½
Distance Between Cutter and Headstock Centerlines (In.)	4¹⁄₃₂ to 13⅜	5⁄16 to 14⅜

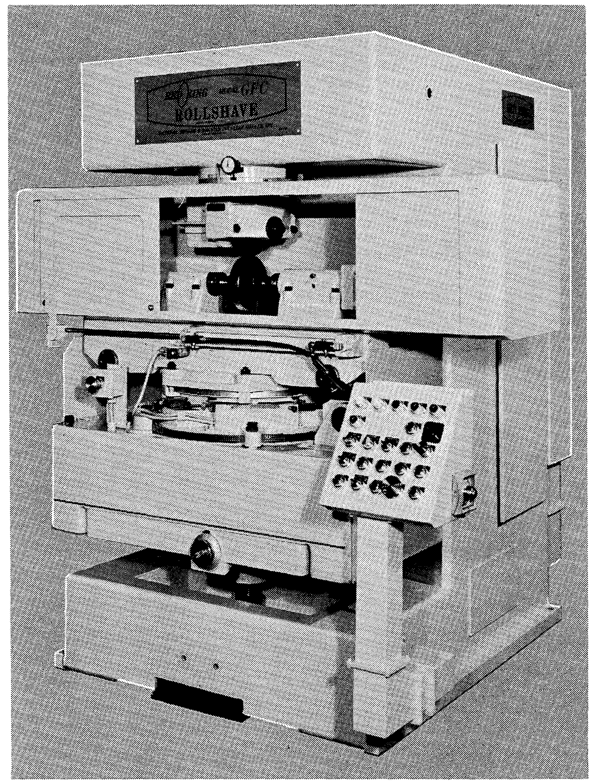


Fig. 10-10 Red Ring Model GFC ROLLSHAVE equipped for semi-automatic loading.

Tunnel-Type Machines For External Gears

MODEL GCX GEAR SHAVER

The Model GCX gear shaver is designed to facilitate the shaving of gears that are too large to load by hand. It has a 72-in. reciprocating table and a heavy cast iron tunnel-type column that straddles the table. The table carries a power-driven headstock and an adjustable tailstock. The column carries the cutter head and the vertical downfeed and traverse assembly.

At the end of the shaving cycle the worktable moves out to the end of its travel for work unloading without overhead machine obstruction.

The machine shaves by the conventional method and can provide crowned or tapered tooth forms.

Specifications

Model	GCX-24 In.
P.D. Capacity (In.)	3 to 24
Max. O.D. (In.)	25½
Pitch Range (D.P.)	2 to 16
Max. Table Stroke (In.)	8(with Crowning) 10(without Crowning)
Max. Distance Between Work Centers (In.)	33½
Distance Between Cutter and Headstock Centerlines (In.)	5⁄16 to 19½

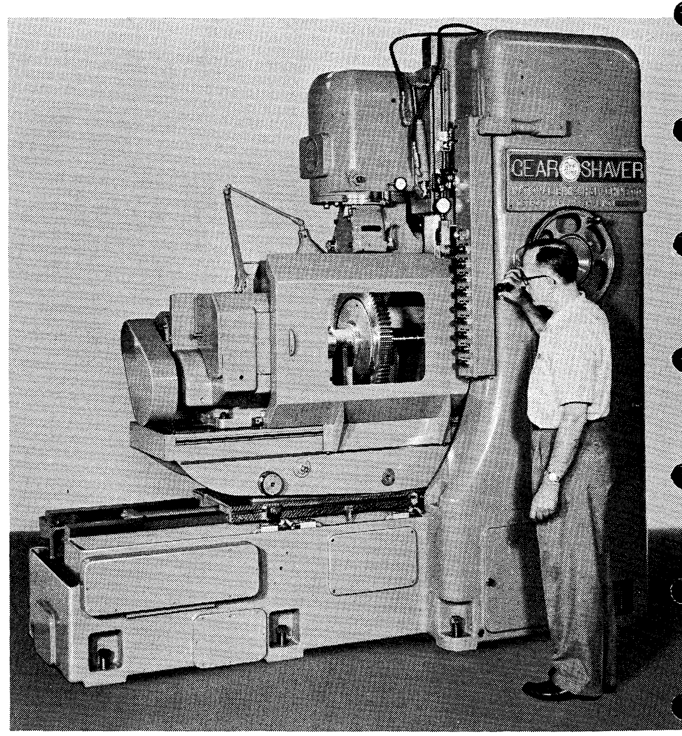


Fig. 10-11 Model GCX gear shaver with table in shaving position.

Internal Gear Shavers

MODEL GCR INTERNAL GEAR SHAVER

The Model GCR gear shaver has been developed especially for shaving only internal gears. It is essentially a horizontal machine with the work mounted on a power driven workhead that is pivoted upward to facilitate work loading and unloading of wide-face gears and those with integral shafts. The shaving cutter is on a slide opposite the work head that reciprocates the cutter in crossed axes relationship for conventional shaving.

During the shaving process the cutter slide is fed upward by an automatic differential upfeed mechanism similar to that used on knee and column gear shavers.

The machine will conventionally taper and crown-shave. It can also be used in a plunge-cut cycle where upfeed of the cutter slide is made without cutter reciprocation in conjunction with a cutter provided with differential serrations, and (if required) a ground crowned tooth form.

Automatic loaders are also available for Model GCR machines.

Specifications

Model	GCR-12 In.
P. D. Capacity (In.)	3 to 12
Max. O.D. (In.)	20
Pitch Range (D. P.)	4 to 48
Max. Face Width (In.)	$2\frac{1}{2}$ (Conventional) 1 (Plunge)
Distance Between Spindle and Crossed-Axes Centerline (In.)	$1\frac{11}{16}$ to $8\frac{3}{4}$
Distance Between Cutter and Headstock Centerlines (In.)	Zero to 5

SHAVING INTERNAL GEARS ON EXTERNAL GEAR SHAVERS

Model GCU, GCY and GCX Red Ring external gear shavers can be equipped with special internal cutter head assemblies to permit the shaving of internal spur or helical gears. The following table gives general capacities of these arrangements. The factors that determine exact internal gear shaving capacities of external gear shavers are the part configuration and the work holding tooling.

External Shaver Model	Size (In.)	Internal Gear P.D. Capacity (In.)
GCU	8	3 to 8
GCU	12	3 to 12
GCU	18	3 to 18
GCY	12	3 to 12
GCY	18	3 to 18
GCX	24	6 to 24

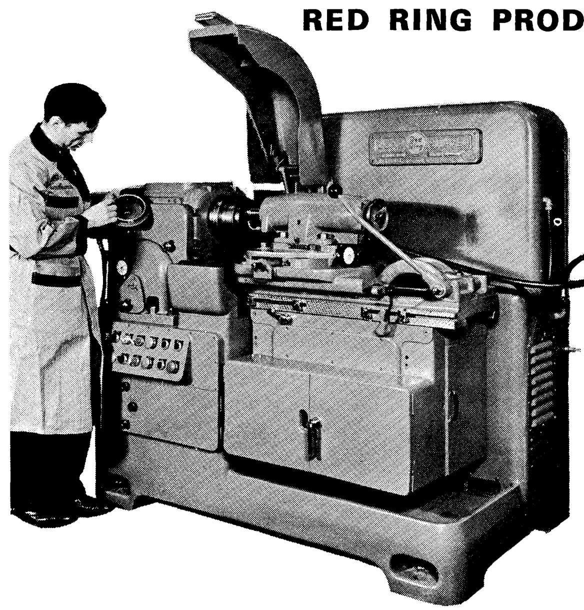


Fig. 10-12 The Red Ring Model GCR internal gear shaver.

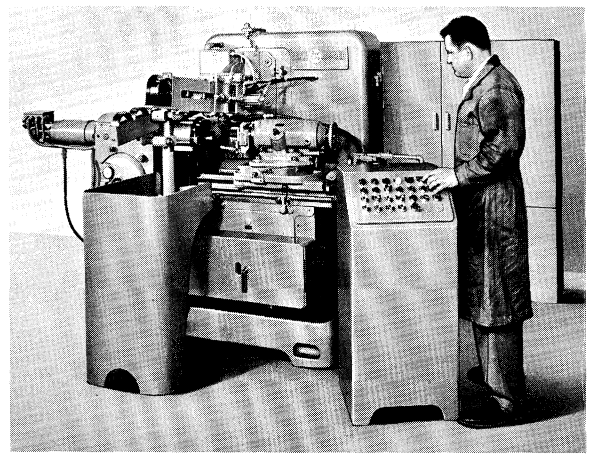


Fig. 10-13 A model GCR internal gear shaver equipped with an automatic loader.

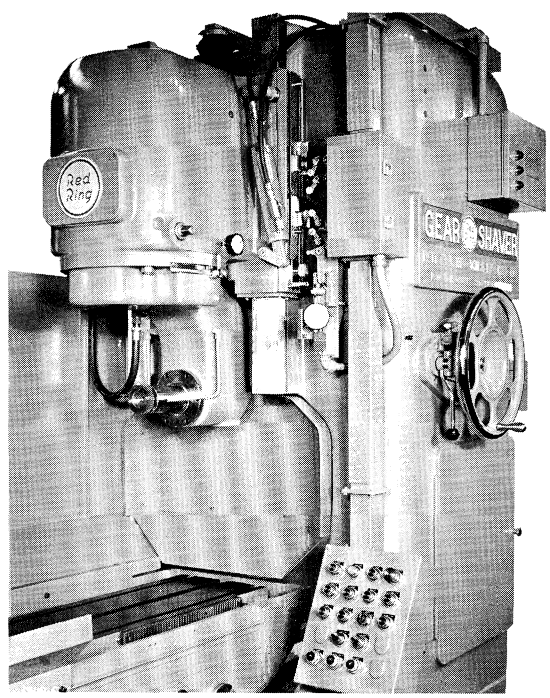


Fig. 10-14 An internal cutter head attachment on a Model GCX Red Ring gear shaver.

Large Vertical Axis Shavers

MODEL GCQ AND GCT SHAVERS

When gears exceed 24-in. diameter, their size and weight require that the utmost in rigidity and precision be built into the machines that will finish the teeth. The Red Ring Model GCQ and GCT vertical axis gear shavers will finish either internal or external gears.

The work gear is mounted on a power-driven rotary table and the cutter is mounted on a slide-mounted head in crossed axes relationship with the work gear axis. The cutter reciprocates vertically while being fed into the work automatically at each end of the stroke. Tooth crowning is not available on these machines.

Specifications

	GCQ-36 In. (Small)	GCQ-36 In. (Std.)	GCQ-48 In.	GCT-100 In.
P. D. Capacity (In.)	5 to 30	12 to 36	22 to 48	25 to 100
Max. O.D. (In.)	40	40	54	116
Pitch Range (D. P.)	4	4	4	2
Max. Gear Width (In.)	8	8	8	43
Max. Distance Center of Cutter to Center of Work (In. External)	5/16 to 23	3 to 23	7½ to 30	15½ to 54½
Max. Distance Between Cutter and Table Centerlines (In. Internal)	5/16 to 23	3 to 23	7½ to 30	8 to 47

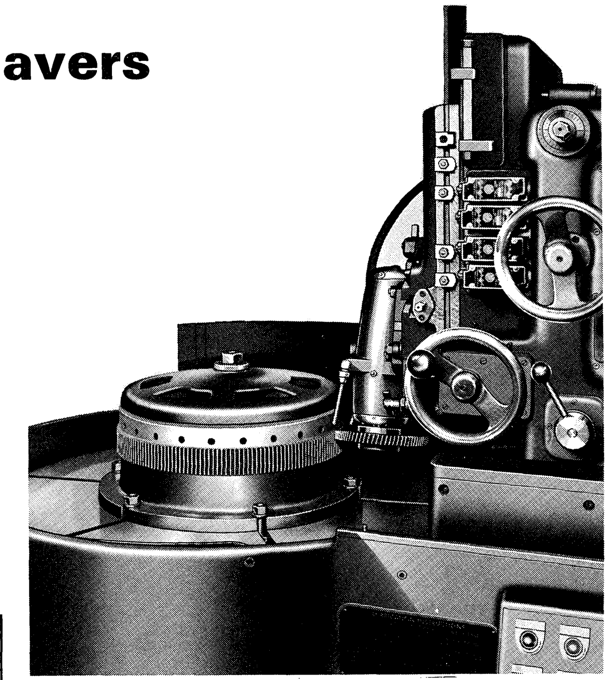


Fig. 10-15 Shaving an external gear on the Model GCQ machine.

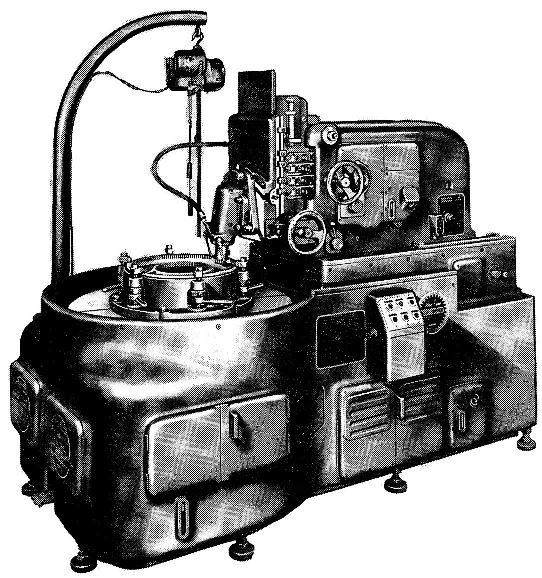


Fig. 10-16 Model GCQ-36 in. Red Ring gear shaver processing an internal gear.

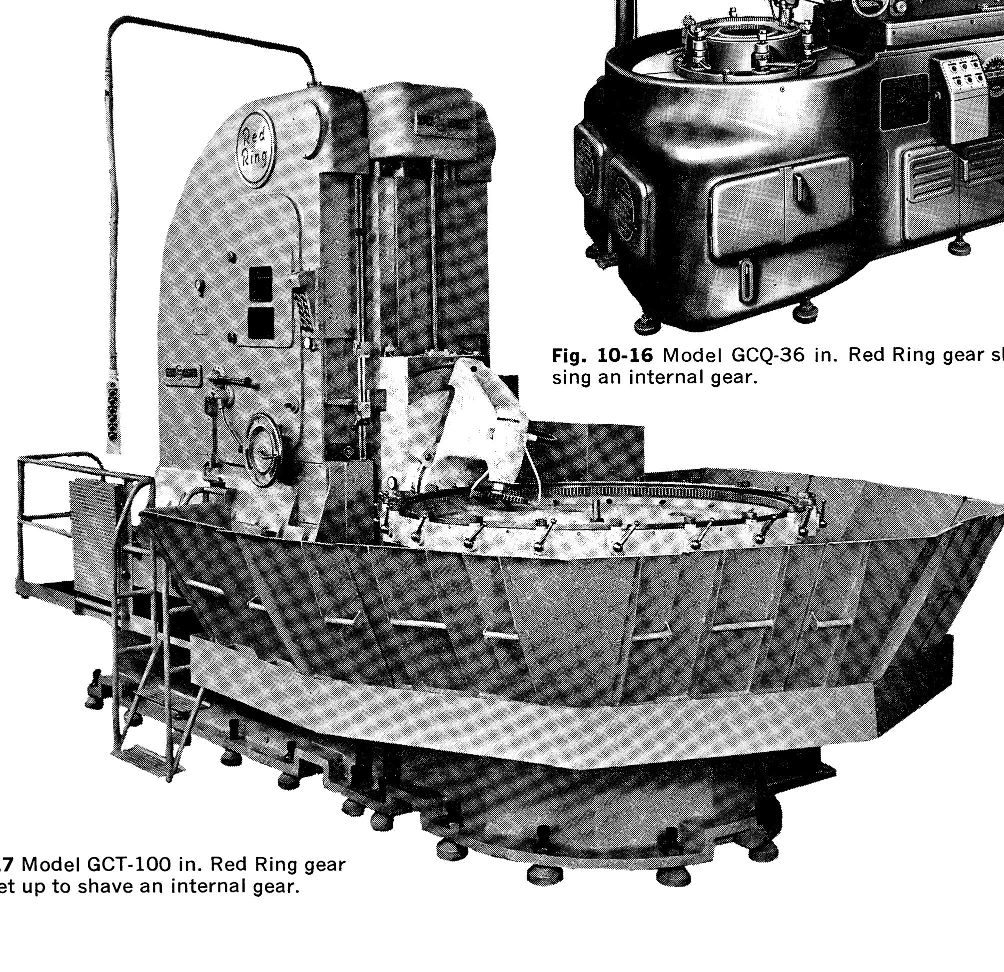


Fig. 10-17 Model GCT-100 in. Red Ring gear shaver set up to shave an internal gear.

Large Horizontal Axis Gear Shavers

MODEL GCK, GCJ AND GCM SHAVERS

Model GCK Red Ring shavers are made in both single and double power-driven work spindle designs. The double-spindle designs apply one work spindle for large gears and another for smaller gears. The cutter head is mounted on a saddle and includes two cutter slides with a journal alignment checker between them. When herringbone gears are shaved, each head can be set for its own gear portion. Work gears are located on centers or journals.

Model GCM and GCJ Machines are similar in principle and action to the GCK machines with the exception that they have single cutter heads, feed cycles can be automatic, and they have live center tailstocks. As in the larger machines, the work gear drives the cutter and the saddle reciprocates the cutter across the face of the work gear.

The Model GCJ machines are similar to the GCM in all features with the exception that crown shaving can be provided with a rocking table action. This machine can also be provided with an internal cutter head to shave large internal gears.

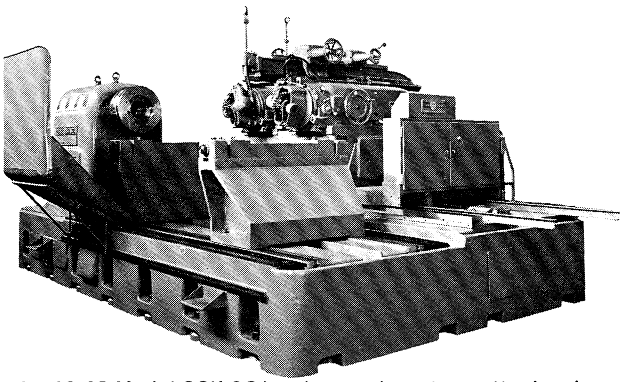


Fig. 10-18 Model GCK-96 in. shaver whose two cutter heads permit shaving both sides of a herringbone gear.

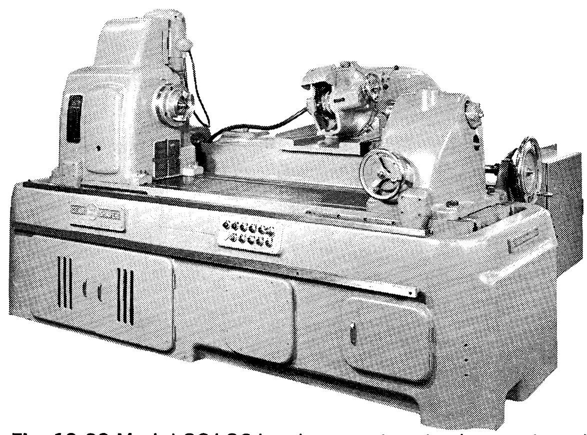


Fig. 10-20 Model GCJ-36 in. shaver set up to shave external gears.

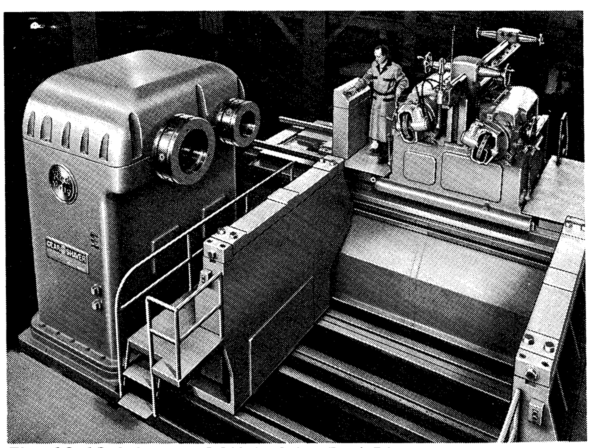


Fig. 10-19 A Model GCK-180 in. Red Ring shaving machine equipped with two headstocks and two cutter heads.

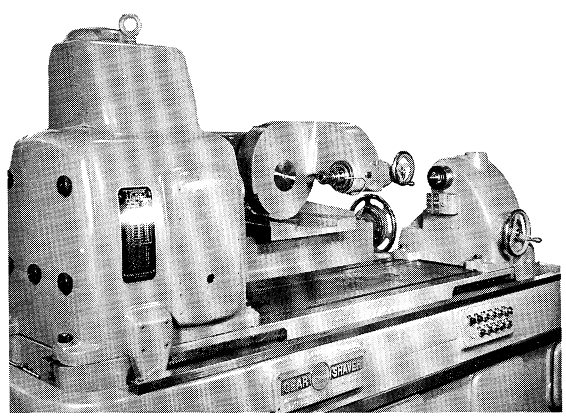


Fig. 10-21 An internal cutter head used to shave internal gears on the Model GCJ-36 in. shaver.

Specifications

Model	GCK			GCM		GCJ	
Size (In.)	96	120	180	36	48	36	48
P. D. Capacity, (In.)	24 to 96	48 to 120	30 to 180	4 to 36	4 to 48	4 to 36	4 to 48
Max. O.D. (In.)	97	121	181	38	50	38	50
Pitch Range (D.P.)	2 to 16	2 to 16	2 to 16	2 to 16	2 to 16	2 to 16	2 to 16
Max. Stroke (In.)	137	137	137	36	36	36	36
Std. Distance Headstock To End of Work Bed. (In.)	170	170	198	82	130	82	82
Distance, Center of Cutter to Center of Work (In.)	16¼ to 58	28 to 70	56 to 97	6¼ to 28	6¼ to 34	6¼ to 28	6¼ to 34

Gear Shaving Cutters

Red Ring rotary gear shaving cutters are super-precision high speed steel cutting tools that are computer-designed, forged from special material, serrated on special machine tools, heat-treated to 64-66 R_c in the National Broach heat-treat department, ground on proprietary tooth grinding machines, and inspected and charted on special measuring equipment. Final accuracy is proven by shaving a test piece. Most of the dimensions on a Red Ring shaving cutter are held to 0.0002-in. and less.

Normally these cutters are sent back to National Broach for regrinding where they are processed through a rigid sequence of inspection, analysis, and production operations.

First the bore is air-gaged and then chromium-plated and reground if it is more than 0.0002-in. oversize. Then tool records are studied to determine proper form for the tooth resharpening process. The amount of stock removal is selected to assure that a maximum number of parts will be produced. A carefully determined amount of stock is then removed from the outside diameter by grinding to provide the proper start of active profile for the shaved tooth form.

The tooth form on the shaving cutter can be precisely modified during resharpening to suit the design and operating conditions of a particular set of gears. It is this ability to produce modified tooth forms that makes the shaving process ideally adaptable to modern gear design and production practice.

Since 1930, National Broach & Machine Division has manufactured nearly 125,000 shaving cutters. More than 365,000 shaving cutter regrinds have been made with less than 1-percent rejection. Rotary shaving cutters are available in tooth sizes from 120 to 1¼ diametral pitch with outside diameters up to 14-in. and widths up to 4¼-inches.

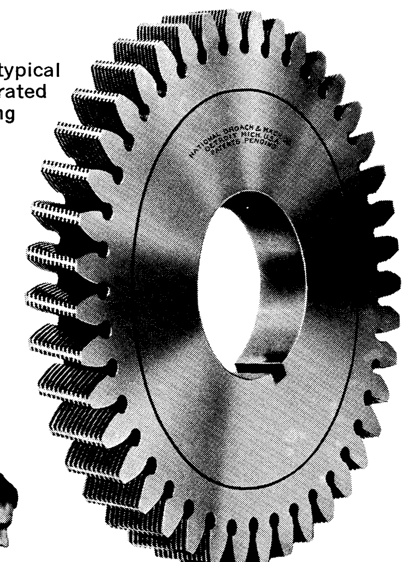


Fig. 10-22 A typical Red Ring serrated helical shaving cutter.

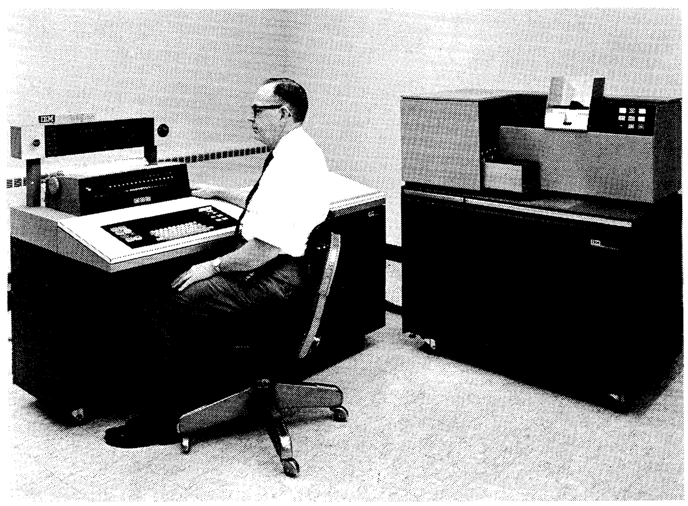


Fig. 10-23 The IBM 1130 computer in the engineering department.

Fig. 10-24 Inspecting and charting involute profile on a shaving cutter on a special Red Ring involute checker.

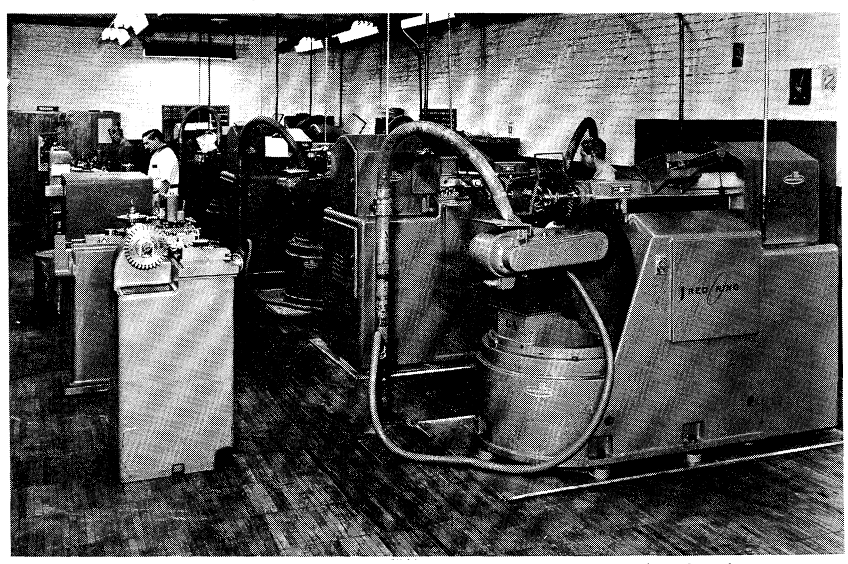


Fig. 10-25 A portion of the special automatic machines that grind and resharpen Red Ring shaving cutters.

Gear Grinding Machines

MODEL SGL AND SGK GEAR GRINDING CENTERS

Red Ring Gear grinding centers are designed to produce the ultimate in precision tooth grinding in preset, fully-automated gear grinding sequences.

They will handle internal and external spur and helical gears, shoulder gears, and splines.

Straight, tapered, crowned or modified involute tooth forms can be effectively produced in experimental low or high production quantities.

Following the setting of dials, micrometers and indicators all of the machine functions are automatic including lead generation, indexing by a large diameter master index plate system, table speed, stroke length, wheel downfeed sequences (rough, semi-finish, finish and sparkout), and wheel dressing frequency, speed and dwell.

The machines can be supplied without the automatic sequencing equipment for hand operation if desired.

Specifications

Model	SGL				SGK	
Size (In.)	12		18		24	
Type	Spur	Helical	Spur	Helical	Spur	Helical
Max. Work P. D., External (In.)	12	12	18	18	24	24
Max. Work O.D.	21	21	24	24	31	31
Max. Work P. D., Internal, (In.)	13	13	16	16	24	24
Table Stroke (In.)	32	32	32	32	48	48
Max. Distance Between Centers (In.)	19½	19½	19½	19½	24	24
Center of Wheel to Center of Work, External (In.)	2½ to 12	½ to 10	5½ to 15	3½ to 13	6 to 19	6 to 19

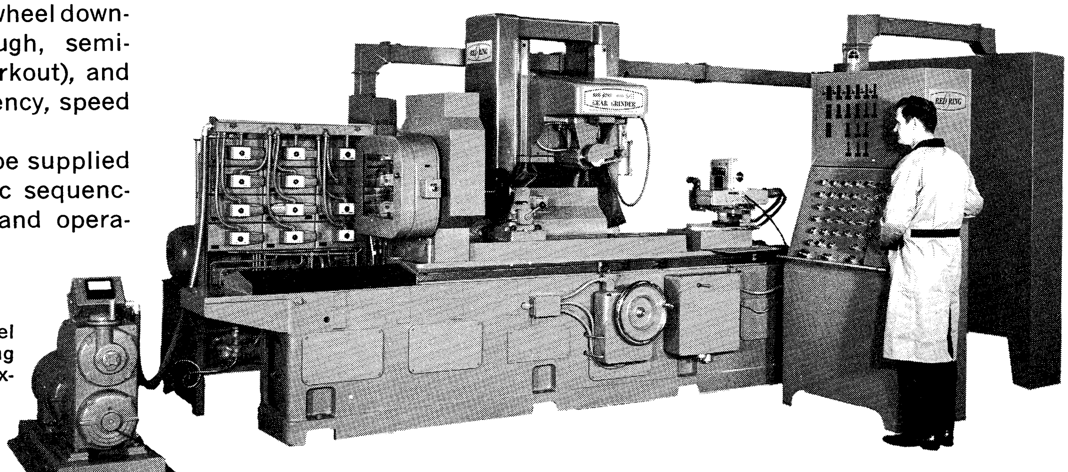


Fig. 10-26 Red Ring Model SGL-12 in. gear grinding center for internal or external spur gears.

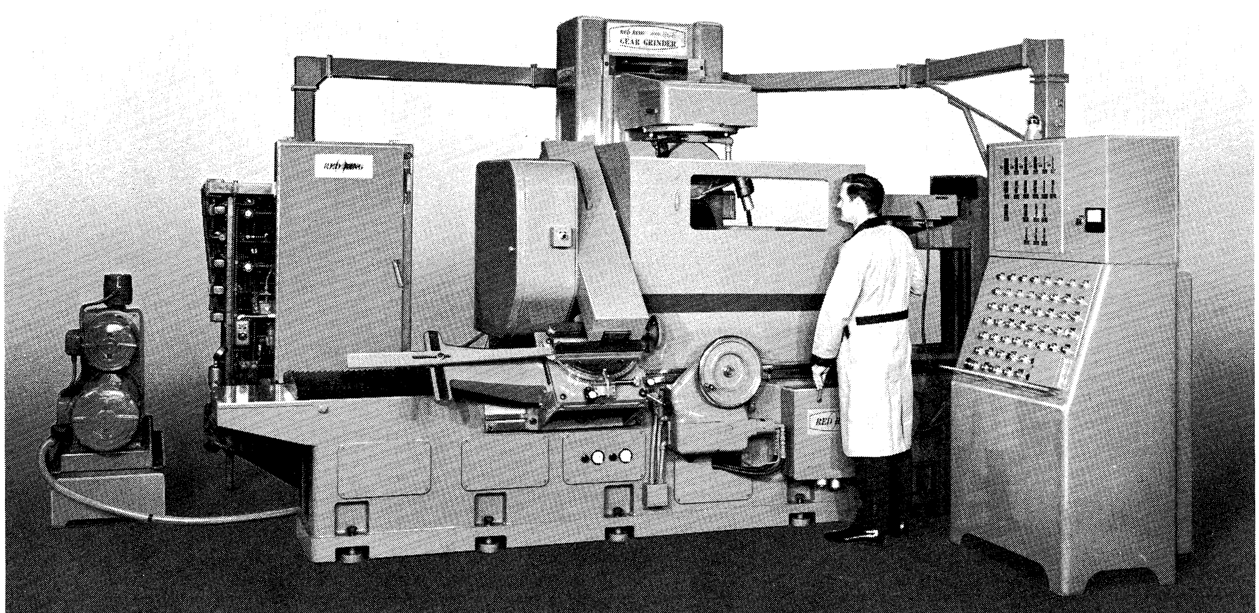


Fig. 10-27 Model SGK-24 in. gear grinding center for external or internal helical gears.

Gear Rolling Machines

Gear rolling is a method of forming or finishing gear teeth by meshing a work gear with a hardened steel die in the form of a mating gear, and reducing the center distance between the two gears to cold flow and form either a gear tooth shape or finish a tooth surface.

The method has been successfully applied to produce the teeth on splines and speedometer worms from the solid and is expected to be used to produce larger gear teeth from the solid by rough-rolling.

Finish-rolling of pre-formed (hobbed, rolled, broached or shaped) gear teeth is being successfully applied to gear teeth to provide excellent surface finish, improved tooth strength and dimensional uniformity from piece to piece.

The finish rolling process is normally faster than rotary gear shaving. Gear rolling dies can roll-finish up to 1,500,000-pieces in the smaller diametral pitch ranges without reconditioning.

Red Ring gear rolling machines are made in two types: a double-die model RGB GeaRoll for rough rolling applications and high-production roll finishing; and a single-die model RGE UNIROLL for low and medium production applications. The Model GFC ROLLSHAVE single-die finishing machine (See Fig. 10-10) performs either rotary shaving or roll-finishing operations; and has the same roll-finishing capacity as the Model RGE UNIROLL (See specification table on opposite page). All machines follow the Red Ring tradition of vertical design in a knee-and-column type of machine.

The forming or finishing of a gear tooth form by gear rolling is considerably different from the cutting action achieved by crossed-axes rotary gear shaving. As a result, the design considerations for both machine and tools are completely different.

Gear roll finishing requires close control of amount of preform stock left for finishing. Whereas excess stock for finishing usually has only a dulling effect on a shaving cutter, it can cause complete breakdown of the tooth form produced by roll-finishing.

MODEL RGB GEAROLL

The Model RGB GeaRoll is a heavy-duty, high production, precision double-die gear rolling machine that can be used for rough rolling of gear teeth from the solid or roll finishing of the tooth surfaces of rolled, hobbed broached or shaped gears.

Following National Broach knee and column shaving machine concepts, the Model RGB is a compact vertical design with one hardened and ground rolling die mounted above the other. Work is fed into and out of rolling position by an automatic loader. Hydraulic pressure raises the lower die to a preset center distance between the two dies to either rough-form or roll-finish a gear. Both dies are power-driven, and provided with accessible adjustments for taper, helix angle and face alignment.

Specifications

Max. Work O.D. (In.)	6
Max. Gear Width (In.)	4
Max. Frame Force Capacity (Lb.)	100,000
Distance Between Die Centers (In.)	9 to 15

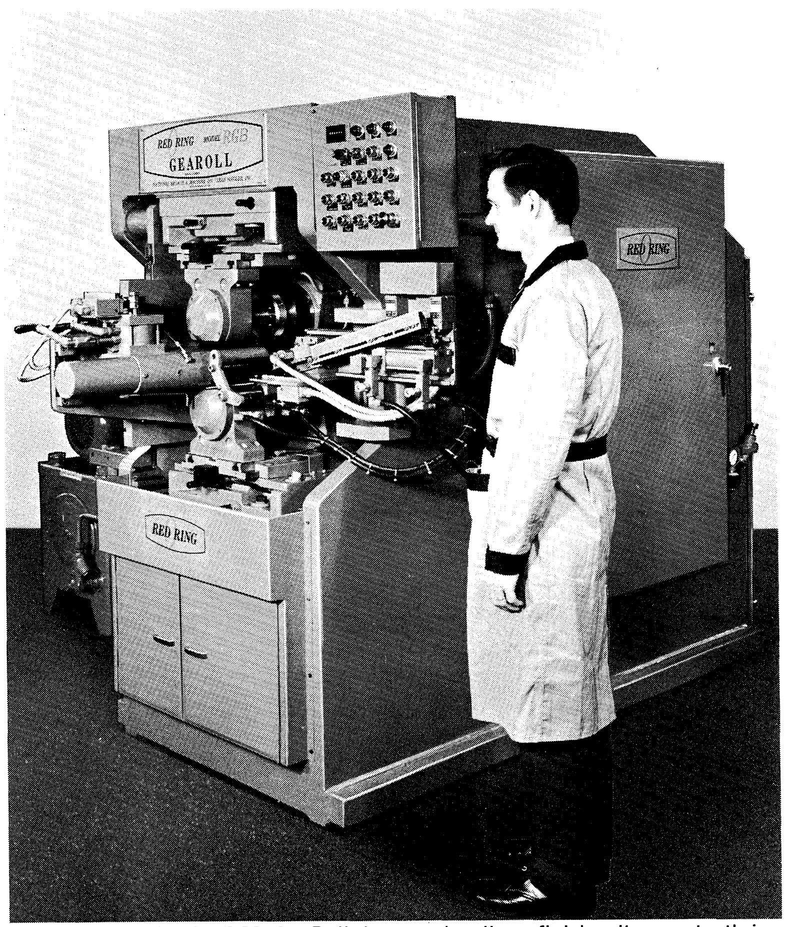


Fig. 10-28 The Model RGB GeaRoll that rough-rolls or finish-rolls gear teeth in high production.

MODEL RGE UNIROLL

The Model RGE UNIROLL is a single-die gear roll finishing machine designed especially for low and medium production. It is basically a Model GFC ROLLSHAVE machine that performs only roll-finishing operations.

The UNIROLL has no table slide or increment feed system as required in shaving. The table mounting the head and tailstocks feeds upward through an air cylinder and toggle mechanism to bring the work into large-force contact with the single rolling die, which is rotated in either one or both directions during the automatic work cycle. Loading can be manual, semi-automatic or fully automatic.

Specifications

[UNIROLL and ROLLSHAVE (Rolling)]

Model	RGE-12 In. and GFC-12 In.	RGE-18 In. and GFC-18 In.
Max. Work O.D. (In.)	$12\frac{3}{4}$	$19\frac{1}{4}$
Max. Gear Width (In.)	$2\frac{3}{4}$	$2\frac{3}{4}$
Max. Frame Force Capacity (Lb.)	20,000	20,000
Distance Between Die and Work Centerline (In.)	$4\frac{1}{32}$ to $13\frac{3}{8}$	$5\frac{1}{16}$ to $14\frac{3}{8}$

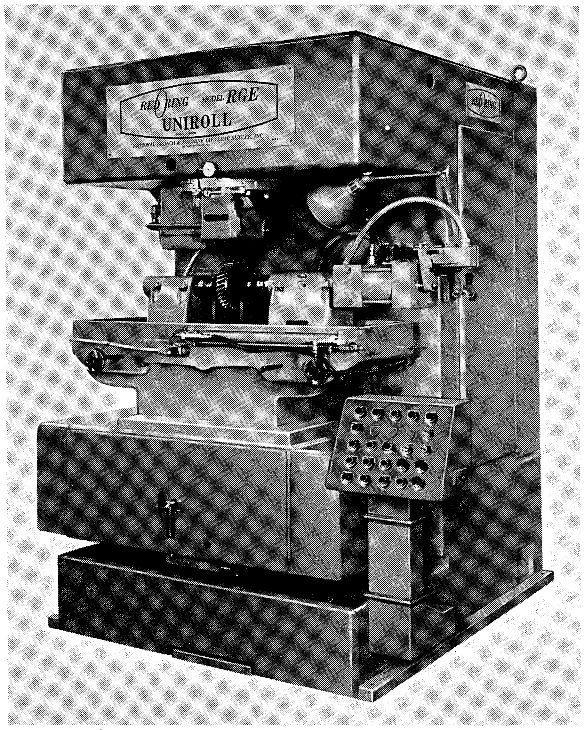


Fig. 10-29 The Model RGE UNIROLL for low and medium production roll finishing.

Gear Rolling Dies

Red Ring gear rolling dies are super-precision, hardened and ground high speed steel. They are machined, heat-treated and ground by National Broach to exacting tolerances of 0.0002-in. and less.

A great amount of development work goes into the determination of the form to grind in a gear rolling die. More modification in die tooth form is required to produce a given gear tooth form than has been traditional in shaving cutter design.

The dies are made of a special fatigue and impact-resistant high speed steel that is made in the form of a pancake forging. The forging is mill-annealed, machined, the teeth are hobbed, and the blank is heat-treated to 59-61 R_c hardness in a four-step process that includes heating, tempering, sub-zero treatment and a final tempering.

Grinding is carried out on special Red Ring automatic grinders. Following grinding, all tooth dimensions are checked and charted.

Red Ring gear rolling dies are made in diameters up to 10-in. with tooth forms of 5-diametral pitch and finer.

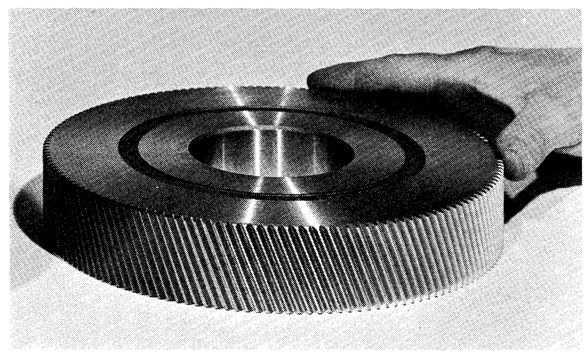


Fig. 10-30 Typical Red Ring gear rolling dies.

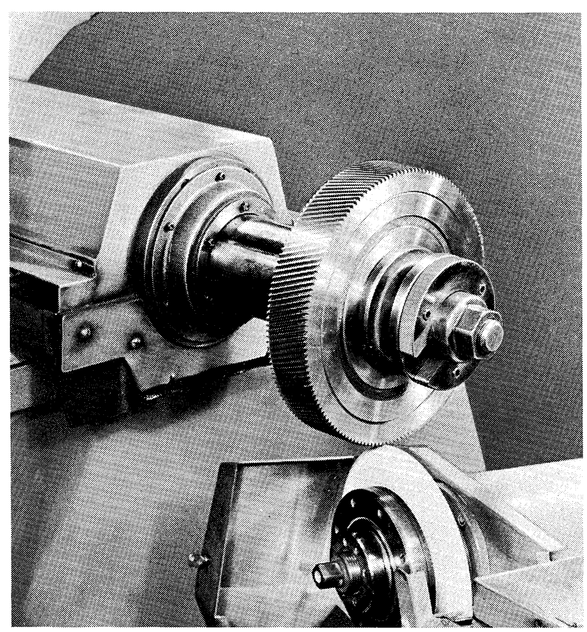


Fig. 10-31 Grinding a rolling die on a special automatic grinder.

Gear Honing Machines

Gear honing is a hard gear finishing process that was introduced by National Broach & Machine Div. in 1956. Originally conceived as a process to improve the sound characteristics of production gears, it has now been expanded into use by the aerospace industry as a proven method of increasing the service life and load-carrying capacity of precision ground gears.

The process uses an abrasive-impregnated, helical gear-shaped tool that is meshed in crossed-axes relationship with a work gear and run in both directions at high speeds while the hone is traversed back and forth across the face of the gear in a plane parallel to the work axis.

Improvement in sound characteristics by honing a hardened gear results from the removal of nicks and burrs, improvement in surface finish and minor corrections in tooth form.

In production of transmission gears, the process has completely eliminated transmission teardown and subsequent nick search and removal operations to remove nick and burr noise.

In aerospace gearing, honing of ground helicopter drive gears has in some cases increased service life by 1,000 percent and load carrying capacity by up to 30-percent. This life increase has been achieved by reducing the 16 to 32-mu surface finish on the ground gears down to 8-mu and finer by honing.

The Red Ring Model GHG gear honer is made in three sizes to perform honing operations. Automatic loaders can be applied to perform high-production honing operations.

The Model GHG has a patented tilting table that supports the head and tailstock. This tilting table provides a choice of four different honing methods as follows:

1. Honing in loose backlash with the hone and work gear positioned in fixed center distance location.
2. Zero backlash with pressure control. The work gear is positioned with the table locked in zero backlash with a pre-selected pressure against the hone.
3. Constant-pressure operation with the work gear held in mesh with the hone at a constant pressure.
4. Constant pressure with overload relief. Here constant pressure is maintained between the hone and work gear with extra back-up protection provided by an overload relief system.

Gears with crowned tooth forms can be honed on the Model GHG by any of the above four methods by rocking the work table in the horizontal plane while the work gear is running in mesh with the honing tool.

Specifications

Model	GHG-12 In.
P. D. Capacity (In.)	1 to 12
Max. O.D.	14¾
Pitch Range (D. P.)	2 to 20
Max. Stroke (In.)	5½
Max. Distance Between Work Centers (In.)	26$^{15}/_{16}$
Center of Tool to Center of Work	4¾ to 14½
Model	**GHG-18 In.**
P. D. Capacity (In.)	1 to 18
Max. O.D.	19¼
Pitch Range (D. P.)	2 to 20
Max. Stroke (In.)	5½
Max Distance Between Work Centers (In.)	26$^{15}/_{16}$
Center of Tool to Center of Work	4¾ to 14½
Model	**GHG-24 In.**
P. D. Capacity (In.)	3 to 24
Max. O.D.	25¼
Pitch Range (D. P.)	2 to 20
Max. Stroke (In.)	5½
Max. Distance Between Work Centers (In.)	26$^{15}/_{16}$
Center of Tool to Center of Work	5½ to 19½

MODEL GHG GEAR HONER

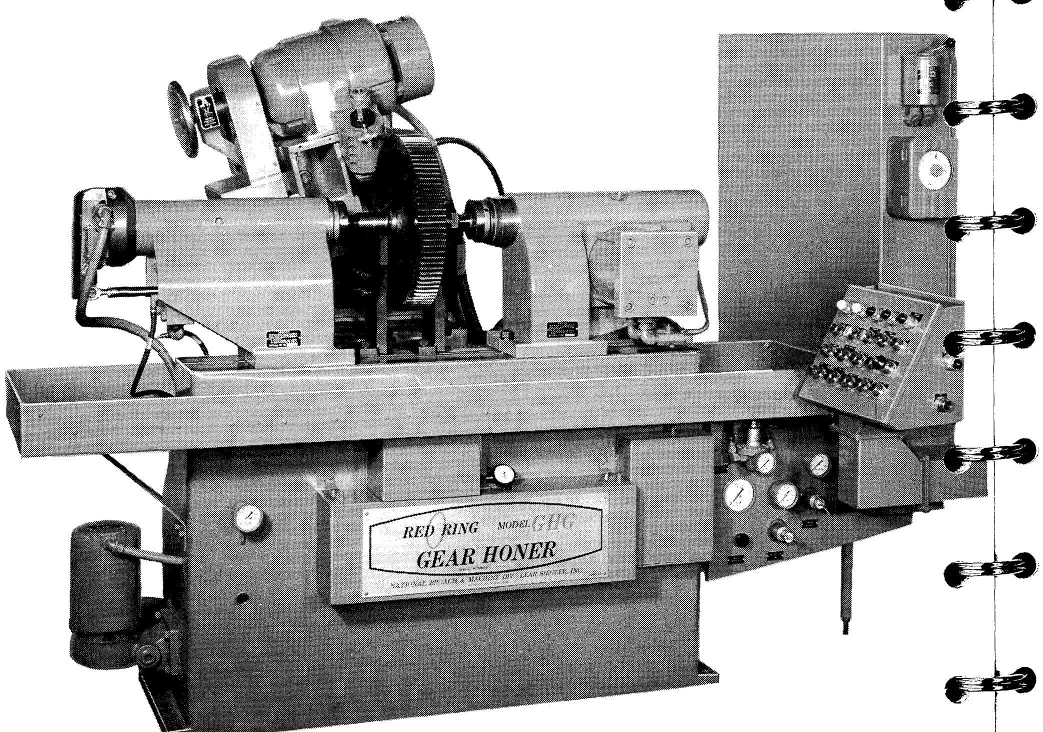
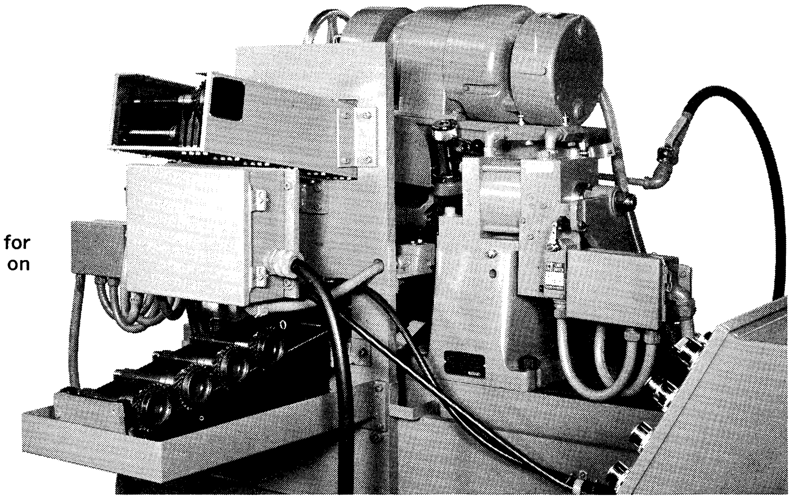


Fig. 10-32 The Red Ring Model GHG gear tooth honing machine.

AUTOMATIC LOADER

Fig. 10-33 An automatic loader for honing transmission clutch gears on the Model GHG honing machine.



Gear Honing Tools

Red Ring honing tools are molded from a mixture of plastic resins and abrasive materials in the 46 to 500-grit particle size range.

The hones are made in two tolerance ranges. The commercial molded tolerance hones are used for nick and burr removal on hardened gears. Hones having AA super-precision tolerances (all critical dimensions within 0.0002-in.) are applied for finishing ground hardened gears of aerospace tolerance quality.

Honing tools are made in 10-in. diameter sizes with widths of about 1-inch.

Three different types of resin-abrasive mix are provided to suit various production applications. Mix No. 7 is used for honing precision-ground gears to improve surface finish and make minor tooth corrections.

Mix No. 32 is a general-purpose, long wear life honing tool that provides fast, heavy stock removal from hardened gears whose teeth have been finished by shaving.

Mix No. 56 provides high impact strength and fast cutting action. It can be applied to meet high-production nick removal requirements where hone tooth breakage has been a problem.

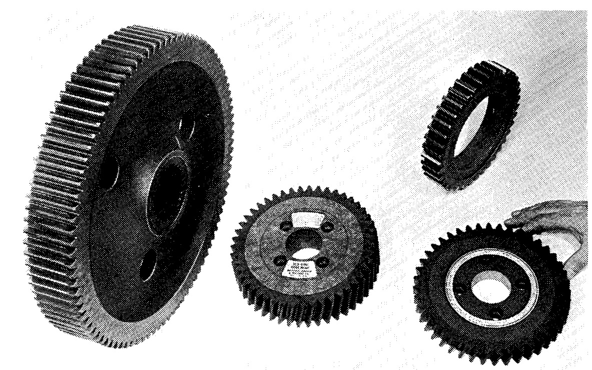


Fig. 10-34 A large spur gear (left) with its No. 32 Mix Red Ring honing tool used to correct parallelism and size. At the right is a helicopter drive gear whose adjacent No. 7 mix AA-tolerance honing tool provided an 8-mu surface finish and 0.0003-in. accumulative tooth spacing error.

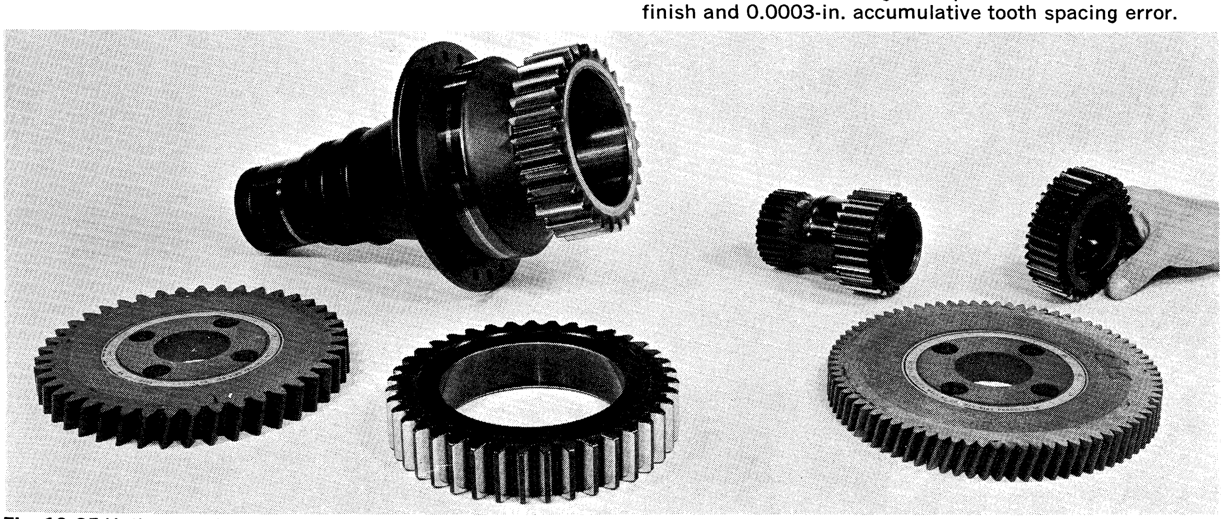


Fig. 10-35 Helicopter drive gears along with their AA super-precision Red Ring honing tools which were applied to increase load-carrying capacity and prolong wear life.

Gear Checking Machines

MODEL SIC, ICB, SIG, LCA & LCB CHECKERS

A variety of Red Ring checking machine combinations are available for inspecting spur or helical gear tooth properties. External gears can be checked as follows: Index and spacing, ICB head; helix angle wobble and size, SIC head; eccentricity and size, ECA head; tooth parallelism and crown, CPA head; lead and tooth parallelism, LCB head.

The SIC head is mounted on a base with SIC head and tailstocks to serve as a universal gear checker with interchangeable SIC and ICB heads. The SIC-ICB universal checker is of similar design and has the ICB head mounted at the rear on a table extension.

The Model SIG internal gear checker inspects internal spur or helical gears for spacing, size and eccentricity.

The LCB lead comparator and checker has a base and table with an SIC tailstock, LCB master lead headstock and an LCA lead comparator slide assembly. The LCB-ICB lead comparators and checkers have a rear table extension that can mount interchangeably the ICB spacing head or SIC angle and eccentricity head.

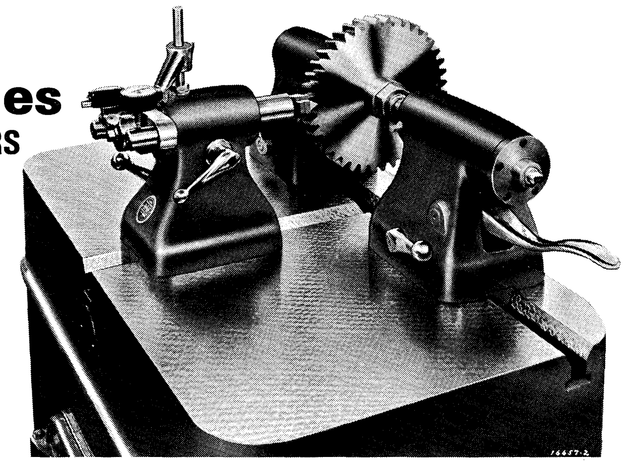


Fig. 10-36 The SIC head that checks for helix angle, wobble and eccentricity.

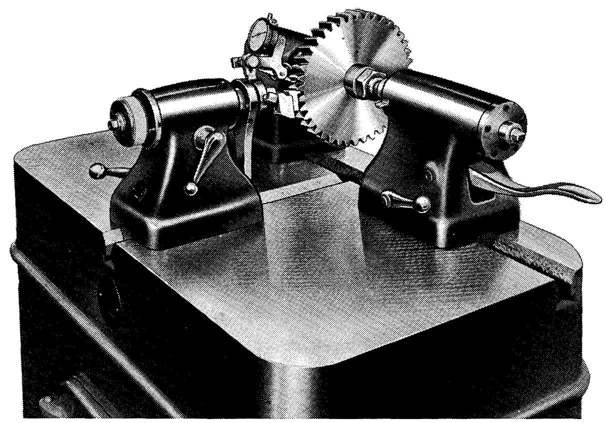


Fig. 10-37 The ICB head that checks for index or tooth spacing.

Specifications

Model	Model Size-In.	Diameter Range-In.	Pitch	Max. Between Centers-In.
SIC or	12	1/4 to 12 11/16	2 to 48	18 1/2
SIC-ICB	18	2 to 18 7/8	2 to 48	17 1/8
Universal Checkers	24	3 to 25 1/2	2 to 48	11 3/8
SIG (Internal)	18	2 to 18 (26-in. Max. O.D.)	2 to 48	2 to 10
LCB or	12	1/4 to 12 11/16	2 to 48	14 1/16
LCA-LCB	18	2 to 18 7/8	2 to 48	13 5/16
Lead Comparator and Checker	24	3 to 25 1/2	2 to 48	12 5/16

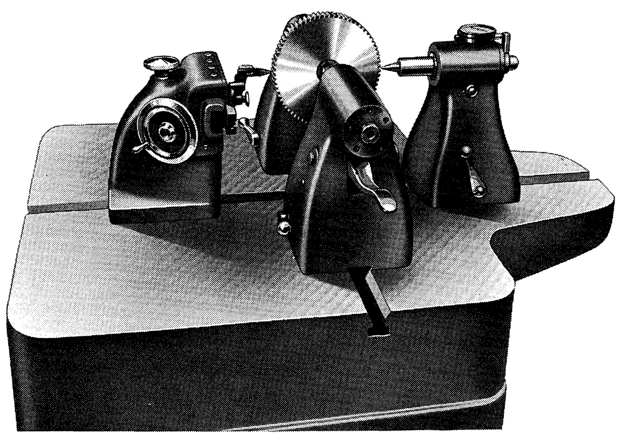


Fig. 10-38 The CPA head at the left on this SIC-ICB universal checker base measures tooth parallelism and crown. The ECA head at the right measures eccentricity and size.

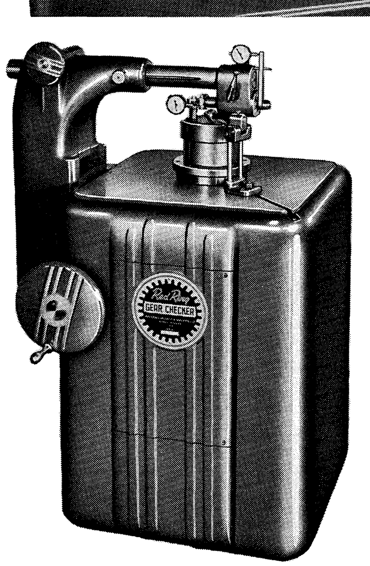


Fig. 10-39 The Model SIG internal gear checker that checks spacing, size or eccentricity.

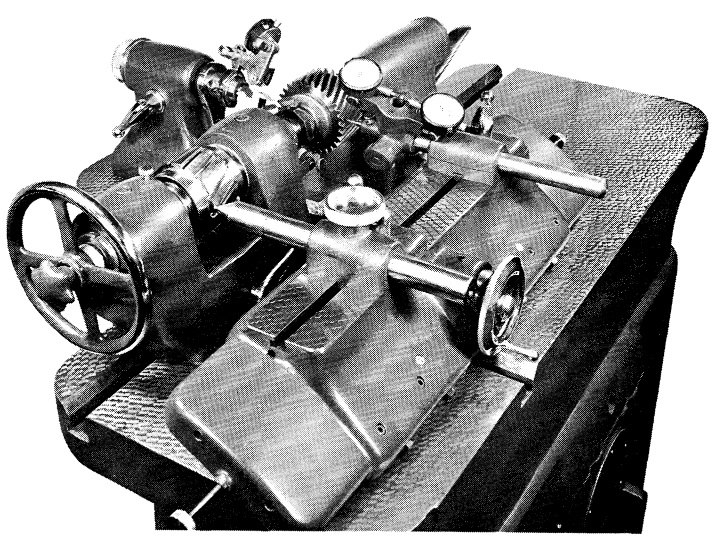


Fig. 10-40 The Red Ring Model LCB-ICB lead comparator and checker with the ICB head mounted on the table behind the work spindle.

Gear Rolling Fixtures

MODEL GRH & GRJ ROLLING FIXTURES

Specifications

Model	Attachment	Center Distance (In.)	Max. O.D. (In.)	Max. Length Between Centers (In.)
GRH		$1\frac{7}{8}$ to $10\frac{3}{8}$		
GRJ	Small Stub Arbor Head	$1\frac{7}{8}$ to $12\frac{5}{16}$		
GRJ	Large Stub Arbor Head	$2\frac{9}{16}$ to $12\frac{11}{16}$		
GRJ	Small Gooseneck Head	$1\frac{13}{16}$ to $11\frac{5}{16}$	8	$4\frac{3}{16}$
GRJ	Large Gooseneck Head	$1\frac{13}{16}$ to $8\frac{5}{16}$	$9\frac{1}{2}$	$19\frac{1}{8}$

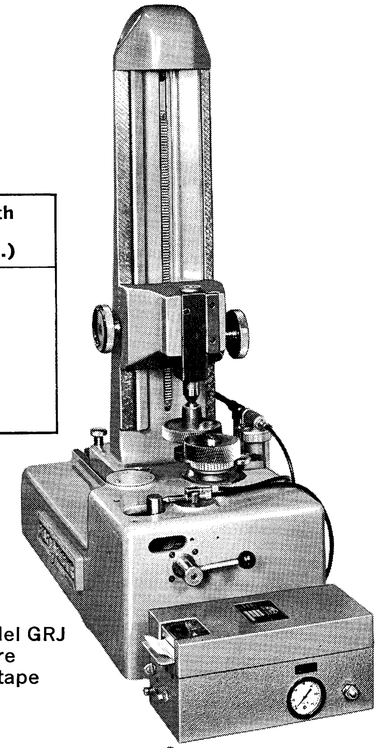


Fig. 10-41 A Model GRJ gear rolling fixture equipped with a tape recorder.

Red Ring gear rolling fixtures measure with a dial indicator the center distance variation of a work gear in tight mesh with a master gear to give a rapid composite check of all gear tooth characteristics simultaneously.

Two different Models are available: The GRH that checks spur and helical gears, and the Model GRJ that checks spur and helical gears with or without long shafts, or spur and helical internal gears.

All models can be furnished with or without motor drives or a tape recorder for making a permanent record of the inspection results.

A check of work gear eccentricity and tooth nick condition independent of the center distance variation size check can be made by equipping either Model with a patented Red Ring inertia checking unit. This rolling fixture can be motorized and provided with an indicator light panel unit to provide a compact, versatile gear analyzer.

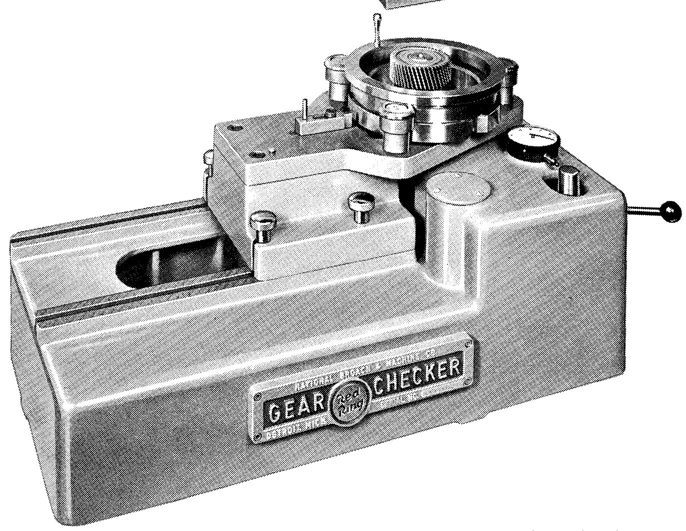


Fig. 10-42 The Model GRJ rolling fixture adapted to check internal gears.

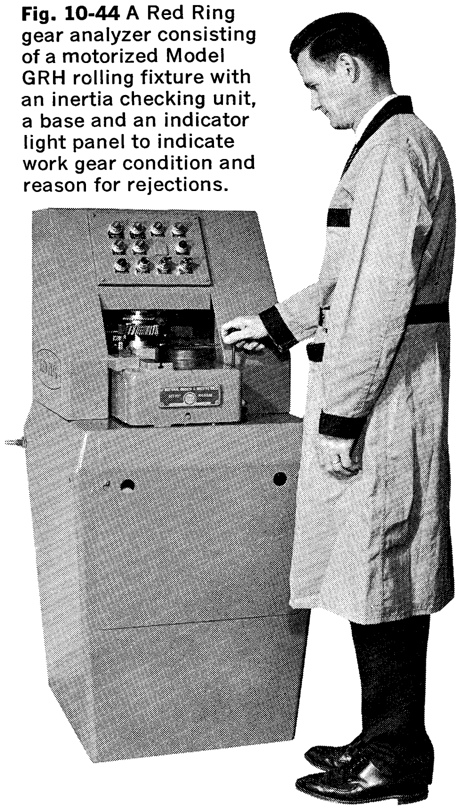


Fig. 10-44 A Red Ring gear analyzer consisting of a motorized Model GRH rolling fixture with an inertia checking unit, a base and an indicator light panel to indicate work gear condition and reason for rejections.

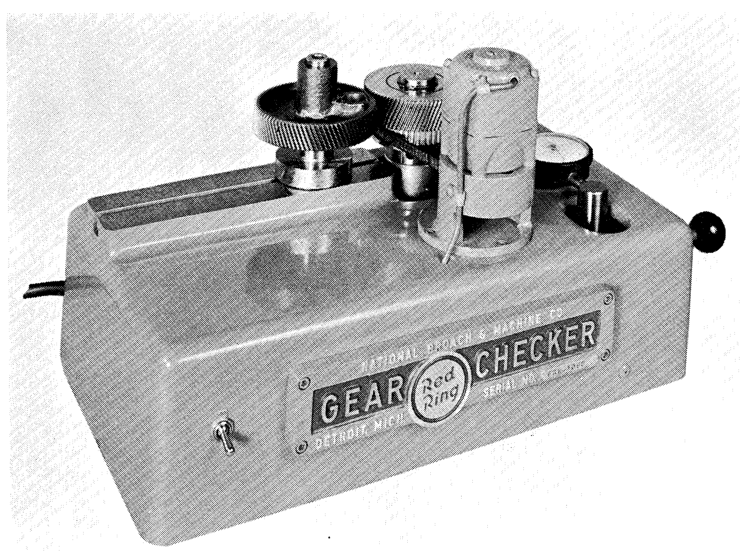


Fig. 10-43 The Model GRH rolling fixture equipped with a motor drive.

Gear Sound Testers

MODEL GSC, GSD, GSJ & GSS TESTERS

Red Ring sound testing machines run finished production gears in mesh at controlled speeds in both directions of rotation both with and without a brake load to ascertain the character and volume of the produced sound.

This procedure prevents gears with nicks or objectionable noise characteristics from being assembled before corrective measures are taken. Gear tooth contact pattern can also be observed on sound testing equipment to help analyze causes of specific gear noise.

Red Ring sound testers have two precision spindles, a means of adjusting the spindle and clamping for accurate center distance setting, a quiet multiple-speed power drive unit and reverse drive, a device for varying brake loads, and provision for easy loading and unloading.

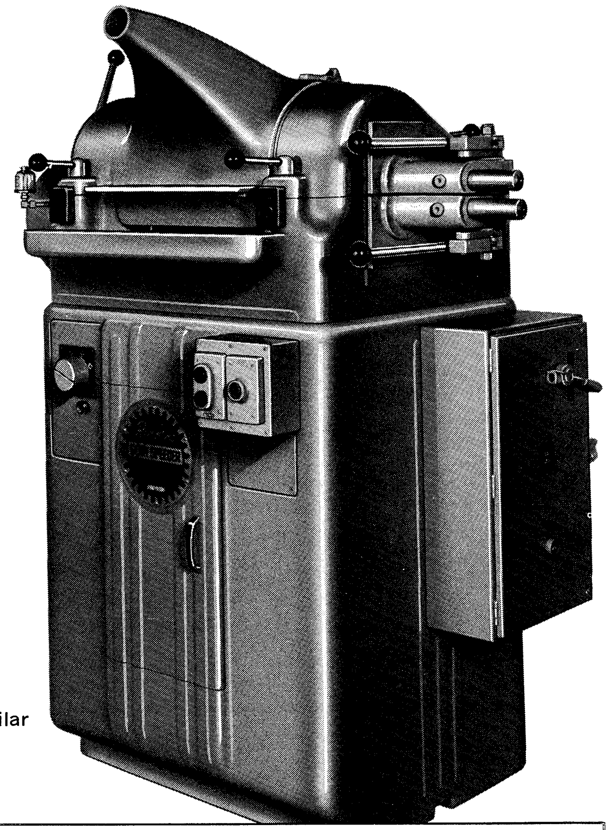


Fig. 10-45 Red Ring Model GSC sound tester that is similar in design to the Model GSJ tester.

Specifications

Model	GSC-10 In. (External Gears)	GSJ-18 In. (External Gears)	GSD-14 In. (Internal or External Gears)	GSD-24 In.	GSS-24 In. (External Gears)
Max. O.D. (In.)	10	19¼	14	24	25⅜
Max. Stem O.D. (In.) (Hollow Spindle Machines)	1½	1⅞	2½	2½	
Spindle Center Distance (In.)	2½ to 5½	3¼ to 10¾	12 Max.	12 Max.	4½ to 24
Max. Length	11	14	18	18	4½ to 24¼ (spindle face to brake head face) 25 1/16 (spindle face to tailstock center)
No. Spindle Speeds	4	4	4	4	Variable
Speed Range (RPM)	750 to 2,250	750 to 2,250	650 to 1,950	650 to 1,950	50 to 875

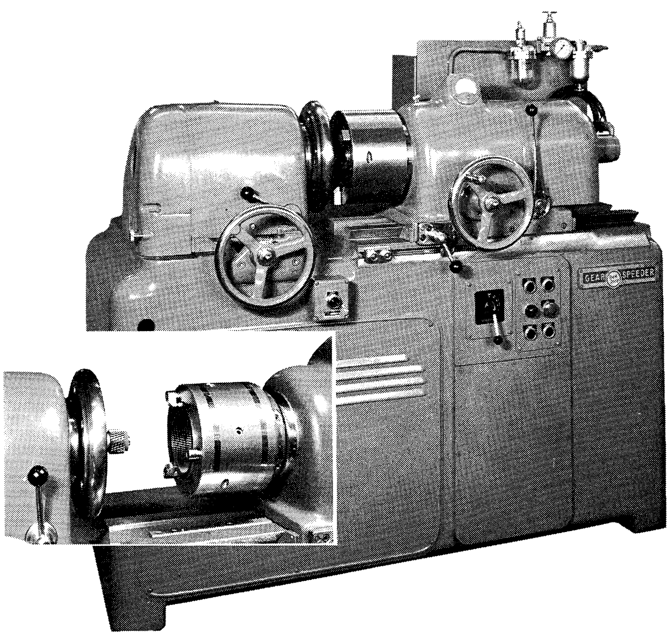


Fig. 10-46 Model GSD sound tester for either internal or external gears.

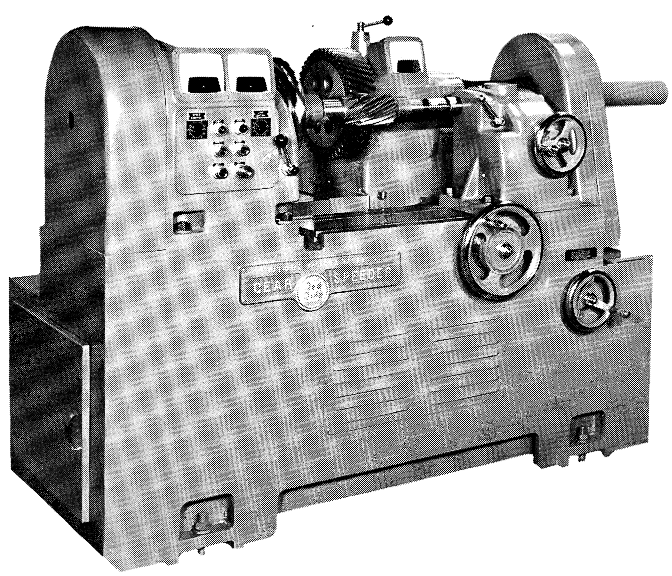


Fig. 10-47 Model GSS-24 in. tester for external gears has indicators for both speed and torque.

Automatic Gear Inspection Machines

Red Ring automatic gear inspection machines make a 360-degree check of a motor-driven workpiece in contact with one or more master gears mounted on Red Ring inertia checking units. These gear gaging centers can be fed automatically and provided with exit chutes having trap doors that can classify parts for size, or sort reject parts into several salvable and non-salvable categories.

In addition to checking parts for size, helix angle, eccentricity, nicks and tooth action, they can also be used to simultaneously check certain size dimensions of the gear body.

Each Red Ring gear gaging center is made to suit specific part inspection and production requirements. They can be fed automatically or hand-fed to initiate an automatic inspection and light indication sequence.

These gaging centers can be designed to operate in conjunction with gear production machines to shut them down after a certain number of rejects are produced, prevent infeed of out-of-tolerance parts into a machine, or sort and classify parts fed to assembly areas.

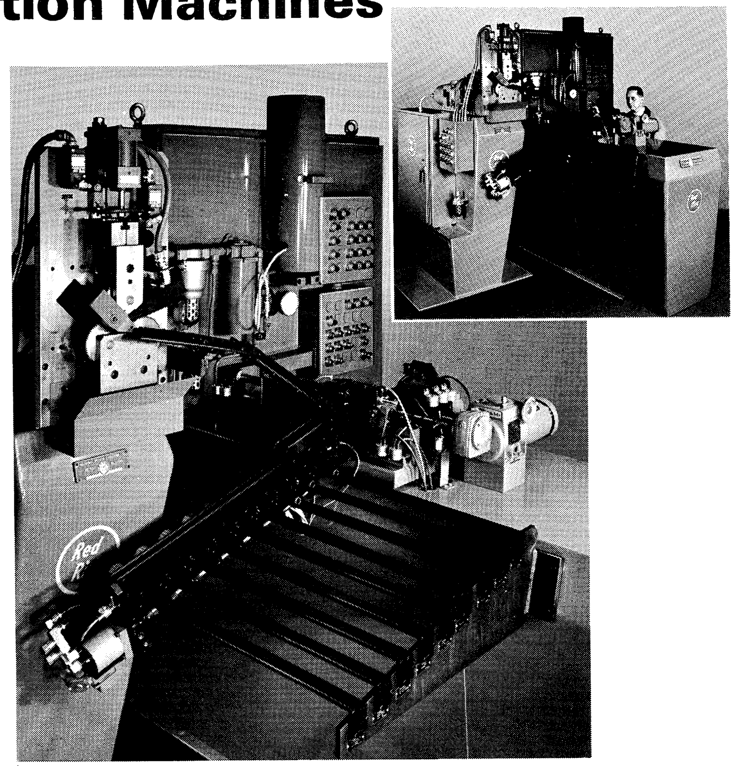


Fig. 10-48 A Red Ring automatic gear gaging center that burnishes transmission pinions, checks them for eight variables and sorts rejects into nine categories at a 400-per-hour rate.

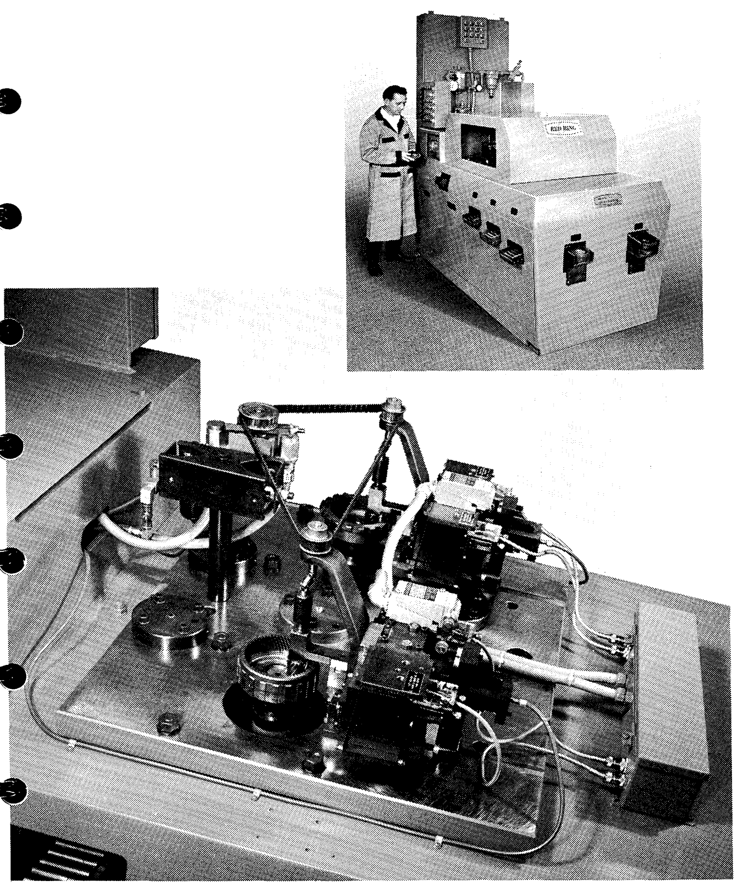


Fig. 10-49 A twin-station Red Ring Automatic gear gaging center that simultaneously checks two different part shapes with the same internal helical gear for five variables and sorts rejects for three different categories at a rate of 250 per hour of each gear.

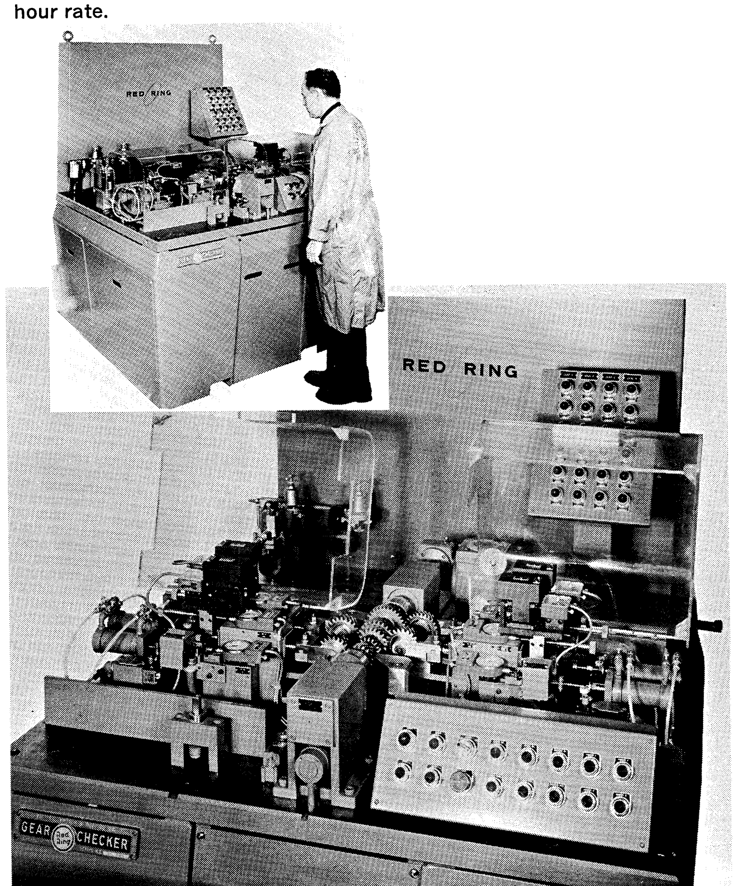


Fig. 10-50 A semi-automatic inspection machine that simultaneously checks four gears on a transmission cluster for size, eccentricity and nicks at 200 per hour. Indicator lights shows errors.

Horizontal and Vertical Broaching Machines

The Red Ring-American Line of broaching machines includes horizontal hydraulic broaching machines, four types of hydraulic convertible vertical broaching machines, three-way vertical hydraulic broaching machines and vertical hydraulic presses.

This line of time-tested, proven broaching machines was acquired by National Broach & Machine Division from Sundstrand Corporation in 1970.

The horizontal broaching machines, are a rugged design that provides maximum rigidity, smooth operation, high accuracy and easy change-over for low and medium production internal and surface broaching operations. They can also be equipped with automated feed systems to meet high production broaching requirements.

The Type T, 3-way vertical hydraulic broaching machines can be used for either internal or surface broaching operations. They can be applied for 'push' or 'pull' internal broaching operations as well as surface broaching operations. The machines can also be applied for arbor-pressing, assembly or straightening work.

The 4-way convertible vertical hydraulic broaching machines feature a modular construction concept that provides four different types of broaching operations. By proper selection and assembly of a line of standardized modular machine components, conversion assemblies and optional accessories; individual machines can be built up for push-down, surface, pull-down or pull-up broaching operations. Dual-ram surface machines in this design have a wider column with mountings for two sets of guideways and two main cylinders.

The vertical hydraulic presses are ideally adapted to economically carry out a wide variety of internal and surface broaching operations. They can also be utilized for arbor press, assembly and light straightening work.

They have a large, open area between the ram and the rest table plus deep throat clearance that provide ample room for adaptation of work fixtures and tooling.

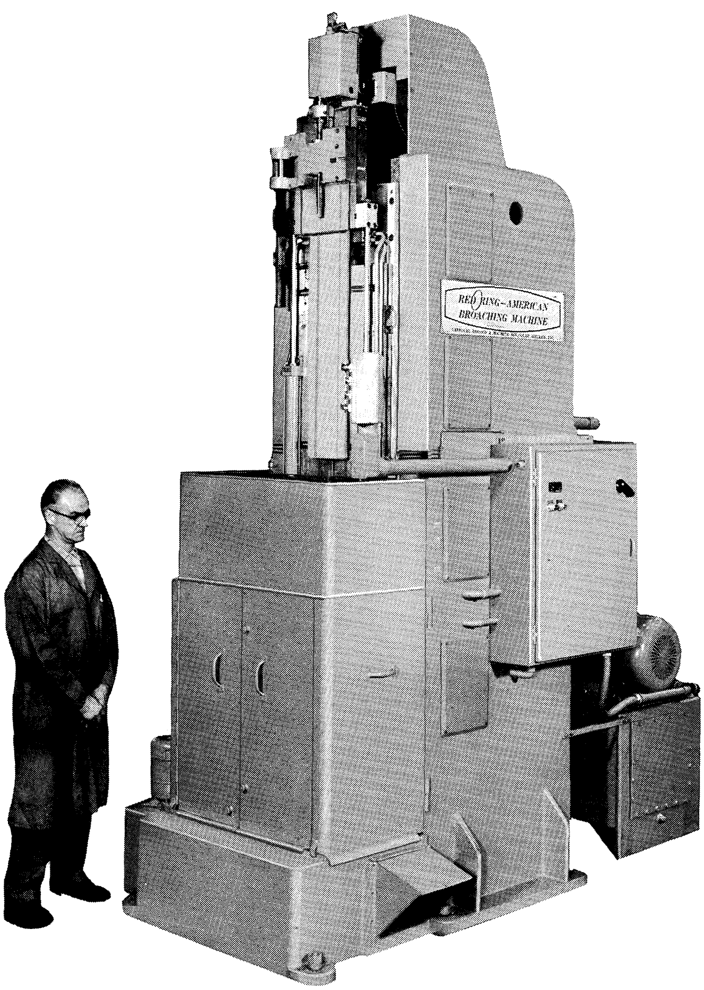


Fig. 10-50A Red Ring-American Four-way convertible broaching machine arranged for 'pull-down' operations.

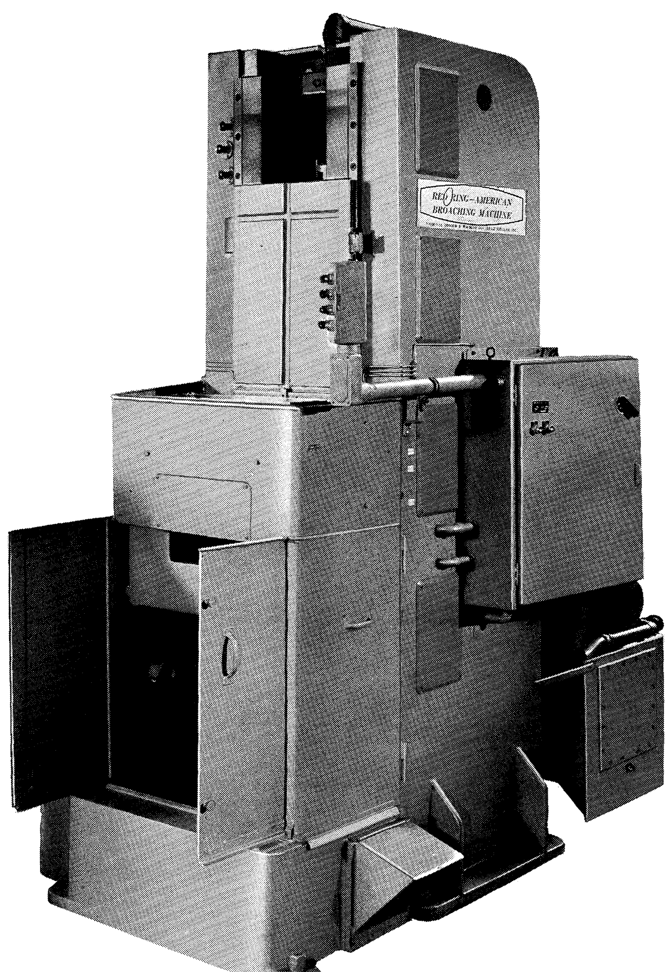


Fig. 10-50B Four-way convertible broaching machine arranged for 'push-down' or surface broaching operation.

Specifications

Type	Model	No. of Models	Capacities (Tons)	Strokes (In.)
Horizontal	HD, HDE	10	4 to 40	30 to 90
Vertical 4-Way Convertible	Pull-Down	21	5 to 50	30 to 90
	Pull-Up	21	5 to 50	30 to 90
	Single-Ram Surface	21	5 to 50	30 to 90
	Dual-Ram Surface	21	5 to 50	30 to 90
	Push-Down	21	5 to 50	30 to 90
Vertical 3-Way	Type T (Push-Down, Pull-Up or Surface)	3	6, 10 & 15	32, 42 & 48
Vertical Hydraulic Press	Push-Down or Surface	3	4, 6 & 12	20, 26 & 26

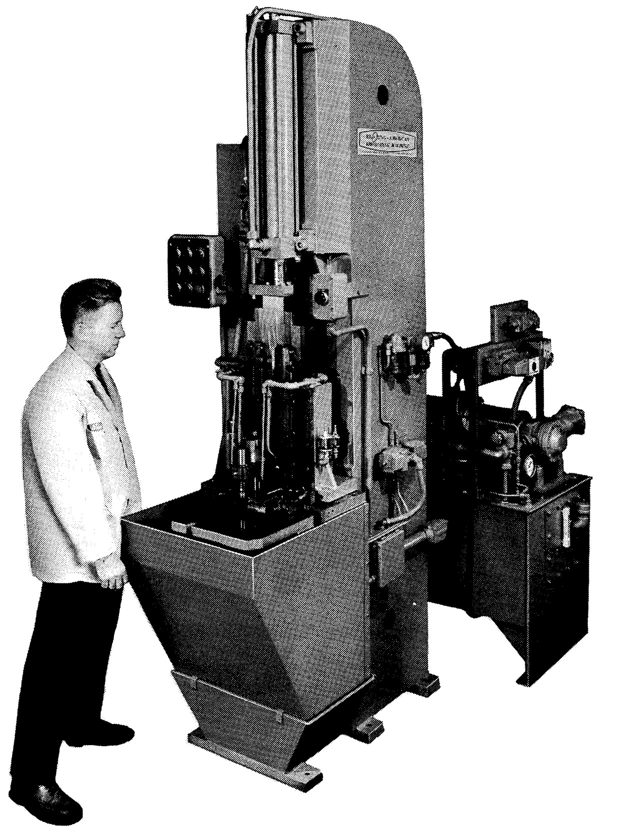


Fig. 10-50C Vertical hydraulic press has a four-station fixture for surface broaching connecting rods and caps.

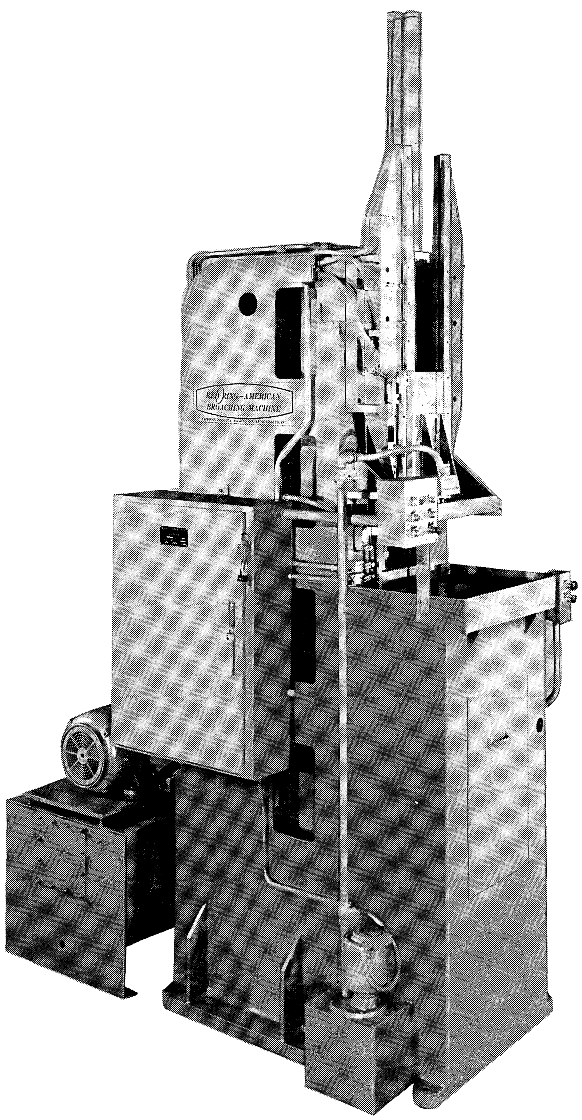


Fig. 10-50D Red Ring-American Type T, three-way broaching machine for 'push-down', 'pull-down' or surface broaching.

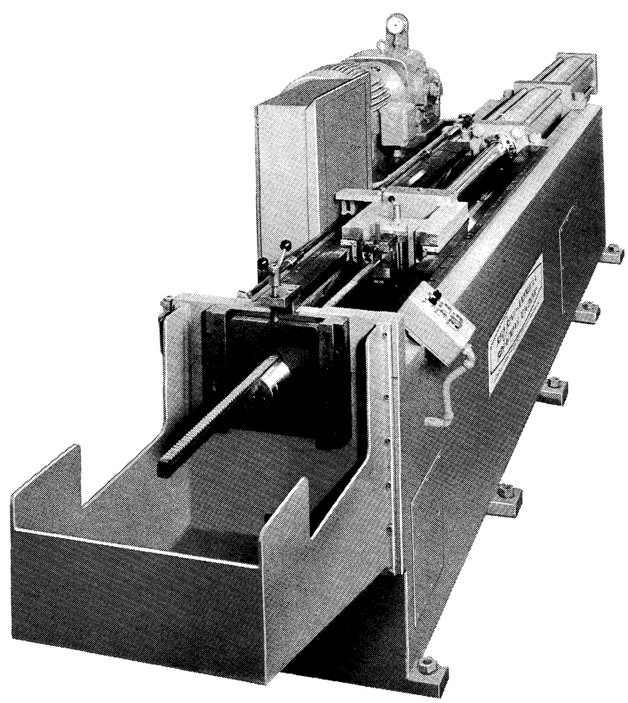


Fig. 10-50E Horizontal broaching machine with adjustable faceplate for multiple-pass keyway broaching.

Pot Broaching Machines

Pot broaching is a method of economically producing precision, high-production external involute gear teeth, splines, and special external toothed forms by pushing a workpiece through a pot broach having internal cutting teeth.

Thus, in one single pass of the workpiece, all of the teeth on the outside diameter of a part can be accurately produced from the solid.

National Broach pioneered the pot-broaching of external helical running gears in 1958 when a 4-in. diameter, ¾-in. wide cast iron helical gear with 24-pitch teeth was produced with a special solid high speed steel pot broaching tool in 5-seconds.

Since that time the process has been successfully applied to a wide variety of splines, running gears, clutches and parts with special toothed forms in a broad range of materials.

Today National Broach and Machine Division offers a line of two pot broaching machines and two different basic types of built-in pot broaching tools to meet broad requirements for accuracy, low cost and easy maintenance.

All Red Ring pot broaching machines have the broaching tool mounted above the workpiece so that gravity will assist in removing the chips from both the broaching tool cutting edges and the workpiece.

The Model VBA Red Ring Pot Broach is a heavy-duty machine that is rated at up to 50-ton capacity. It has the workpiece mounted on a long post and the pot broaching tool is fed down over the part by an overhead hydraulic cylinder-driven ram that is guided by ways on all four corners.

The economical Model VBB Red Ring pot broaching machine applies a unique push-up concept in which the pot broaching tool remains stationary above the part and the workpiece is fed up through it by a hydraulic cylinder underneath the tool. The Model VBB machine has a maximum capacity of 25 tons and is made in sizes and configurations to suit specific production requirements.

Both the Model VBA and Model VBB machines are ideally adapted for automatic loading and unloading, and are made with strokes up to 50-ins.

MODEL VBA POT BROACH

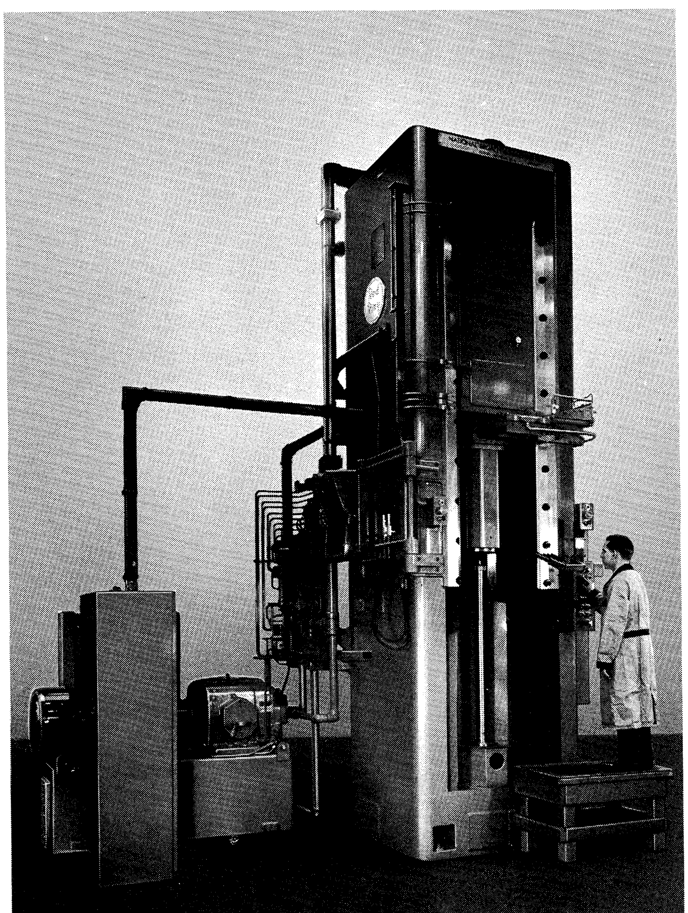


Fig. 10-51 Red Ring Model VBA heavy-duty push-down pot broaching machine.

MODEL VBB POT BROACH

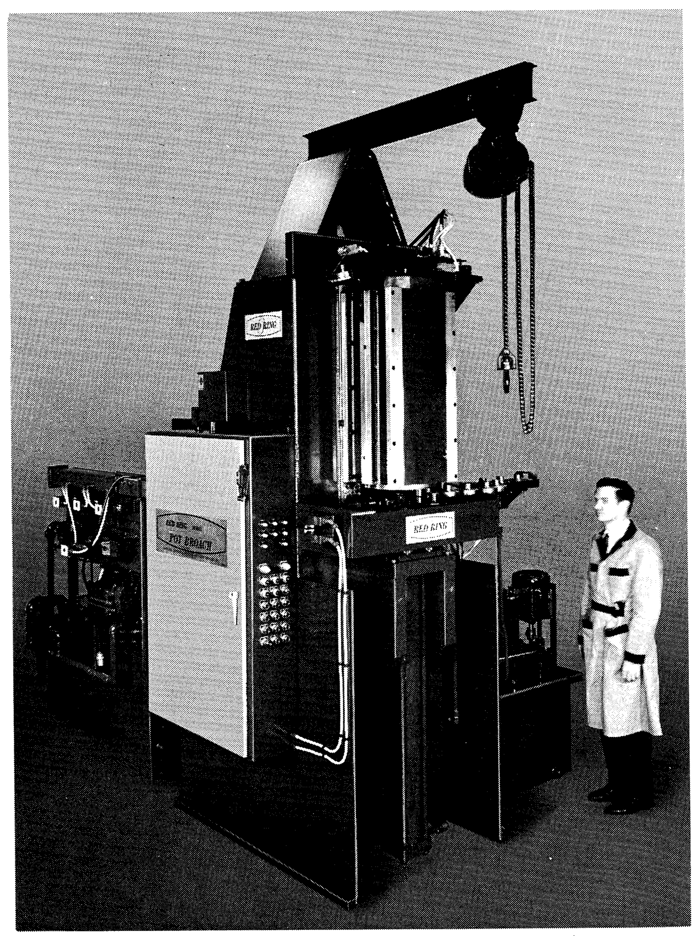


Fig. 10-52 The Model VBB Red Ring push-up pot broaching machine.

POT BROACHING TOOLS

Red Ring pot broaching tools are made in two different types: The ring-type for parts where tooth form and spacing are critical, and the lower-cost stick type which is used where accuracy permits.

The annular-ring type pot broach was a pioneer development of National Broach & Machine Division in 1963. The pot broaching tool is made up of a holder with a series of individual precision ground keyed high-speed steel rings or wafers, each of whose internal cutting teeth are individually backed off. These rings can be sharpened by either the conventional face-sharpening method or the progressive method in which several face sharpenings are performed followed by I.D. grinding of all rings. In the progressive method, the first ring is perished and a new ring added to provide a 200-percent increase in broach life over face sharpening methods.

The stick-type pot broaching tool, which has lower cost individual high speed steel slab broach inserts, includes a holder that supports a series of keyed ground rings which locate the broach sticks in internal ground slots. On thin parts, the teeth in the stick inserts can have wide, long-life tooth lands; and can also be staggered axially to provide a helical pattern to balance the cutting forces.

Combination stick and ring-type Red Ring pot broaches can also meet certain part specifications.

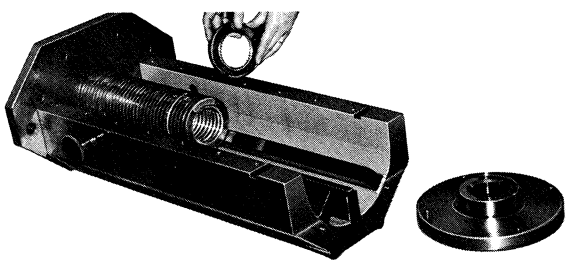


Fig. 10-53 Assembling a Red Ring ring-type pot broach.

Fig. 10-54 A 16-pitch cast iron running gear whose 0.0002-in. tooth profiles are assured by using the illustrated full-form finishing rings at the rear of the ring-type broach assembly. The part is produced in 22-seconds on a Model VBA machine.

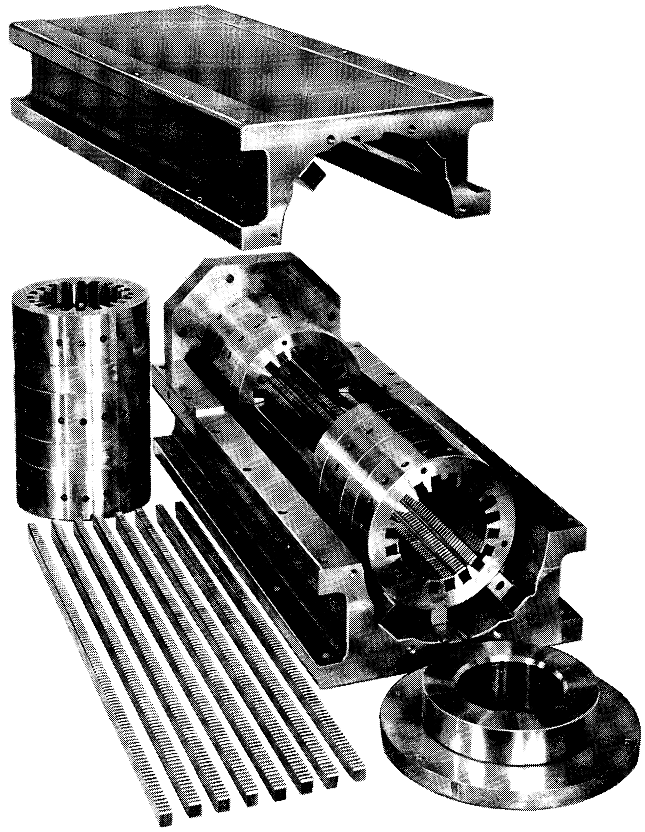


Fig. 10-55 Stick-type Red Ring pot broaching tools disassembled to show construction.

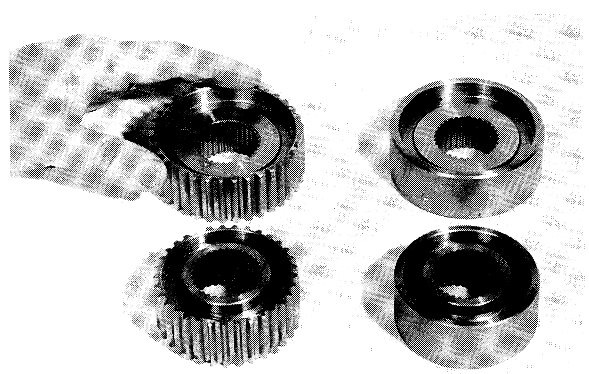


Fig. 10-56 Two different steel synchronizer gears whose 11.4-pitch outside diameter spur teeth have varying designed tooth thicknesses. They are pot-broached on a Model VBA machine at 240 per hour.

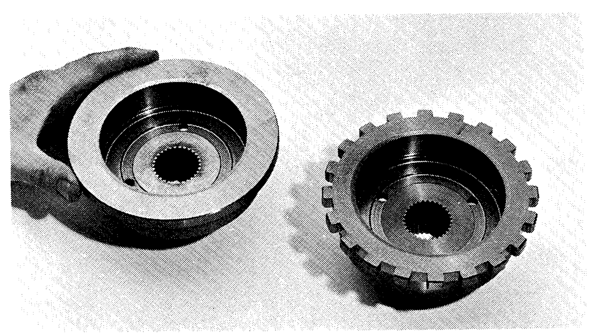


Fig. 10-57 A malleable iron output ring gear whose special-formed 3.666-pitch brake pawl teeth are formed in a 0.40-in. section on a Model VBB at 240 pieces per hour.

Broaching Tools

RED RING BROACH PRODUCTION FACILITIES

National Broach & Machine Division has the largest facilities in the world for machining, heat-treating, grinding and inspecting high speed steel broaches up to 84-in. long and 12-in. diameter.

Heat treating operations are performed in a proprietary heat treating department, permitting close liason between metallurgical and manufacturing processing.

All broaches are heat-straightened instead of peening after hardening to retain the ground dimensional accuracy.

Close control of heat treatment assures that uniform, consistent martensitic grain structures are maintained to provide maximum wear life.

Four grinding operations: Roughing, semi-finishing, primary finishing and secondary finishing are performed to produce the teeth from the solid after hardening.

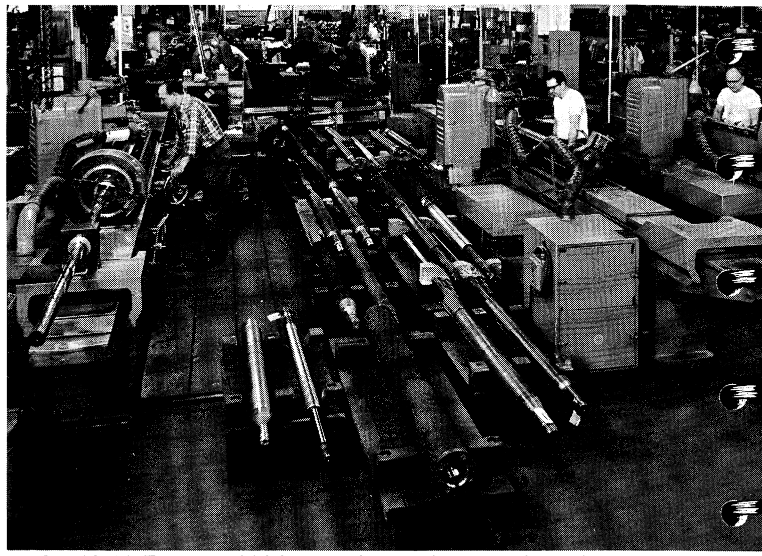


Fig. 10-58 Thirteen 100-in. grinders with 14-in. dia index plates, precision lead bars and ultra-precision dresser templates produce Red Ring large gear and spline broaches.

Naloy Surface Treatment

Finally the Red Ring Naloy Surface Treatment is applied to give maximum abrasive resistance. This exclusive process adds from 50 to 700-percent to broach service life when the broach is not reground on the contact edges, eliminates soft-skin ground surface conditions, increases corrosion resistance and surface hardness, and reduces friction between tool and work to minimize galling and metal pickup.

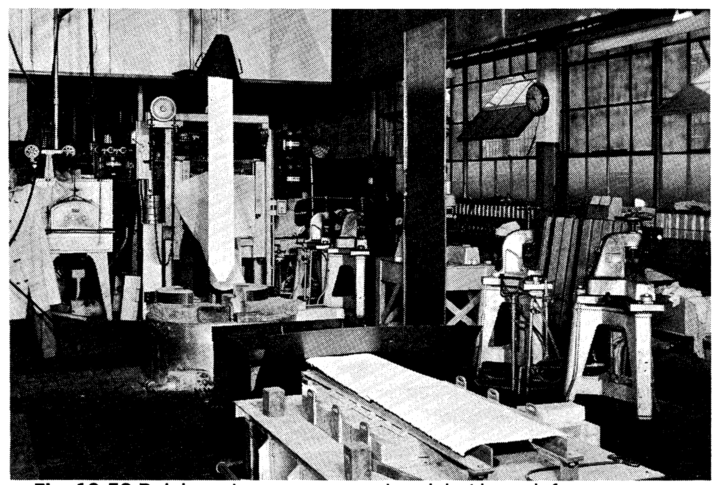


Fig. 10-59 Raising a large, orange-colored, hot broach from a vertical pit furnace at National Broach and Machine Div. before lowering into the adjacent quench tank.

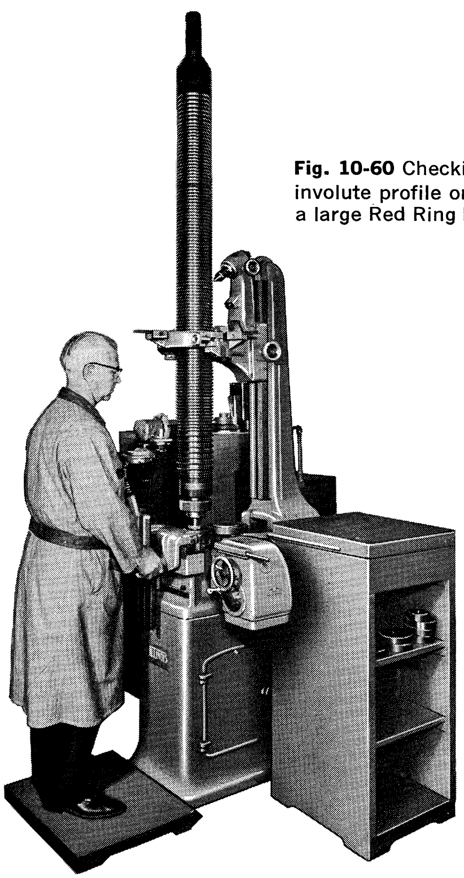


Fig. 10-60 Checking involute profile on a large Red Ring broach.

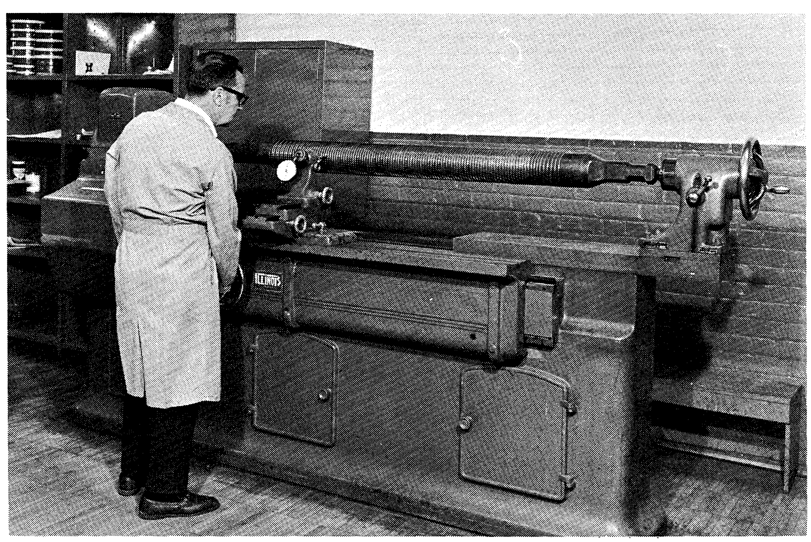


Fig. 10-61 Checking lead on a large Red Ring helical gear broach.

FULL-FORM FINISHING BROACHES

Today the broached internal involute helical running gears in automotive automatic transmissions are produced exclusively by Red Ring full-form finishing high speed steel broaches.

Developed by National Broach in 1959, this patented design has a removable floating shell-type finishing section whose teeth have a full involute form and finish the generated form produced by the front roughing portion of the broaching tool.

This broaching tool concept provides an ultra-precise, ultra-smooth finished tooth form. It eliminates the need for finish broaching operations, since the internal spur or helical gear is produced in a single pass of the tool.

Normally one finishing shell outlasts three of the roughing sections and can give a total tool life of up to 150,000 pieces. Ultra-precise internal splines can also be produced with Red Ring full-form finishing broaches.

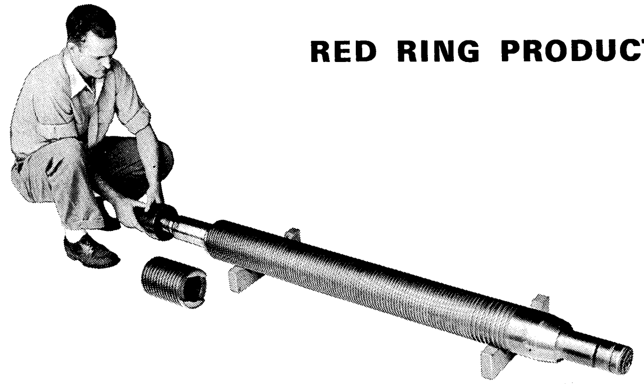


Fig. 10-62 A Red Ring full-form finishing broach with the shell-type finishing section removed.

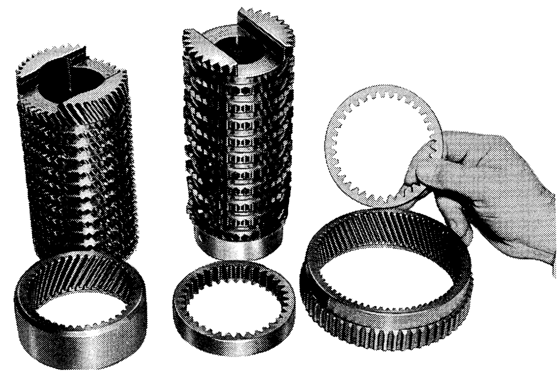


Fig. 10-63 Spur and helical gears whose internal teeth are produced in one pass by broaches equipped with full-form finishing shells.

CONCENTRICITY BROACHES

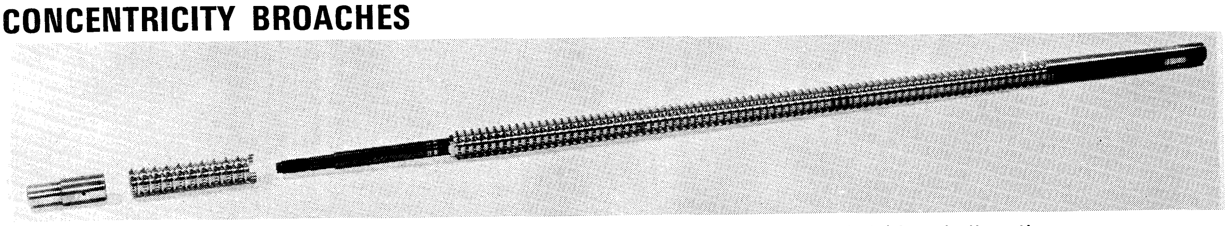


Fig. 10-64 An 80-in. long Red Ring Concentricity broach in exploded view to show the finishing shell section.

In 1961, National Broach engineers expanded the full-form-finishing broaching tool concept by developing a finishing shell with alternate round rings and full-form involute teeth. Called the Red Ring concentricity broach, this tool provides an internal gear or spline with a smooth broached inside diameter that is truly concentric with the pitch diameter of the teeth.

Thus, in one pass of a broaching tool a spline or gear is produced whose inside diameter can be used for processing through its subsequent manufacturing operations on plain or expanding round arbors.

The concentricity broach avoids the errors in inside diameter and pitch diameter concentricity produced by one-piece broaches having alternate round and spline sections. These errors result from back taper, tooth side relief, cutting corner breakdown, machine misalignment or improper face grinding.

In sizes under 1½-in. diameter Red Ring concentricity broaches are made in an exclusive one-piece design that incorporates the full-form finishing features.

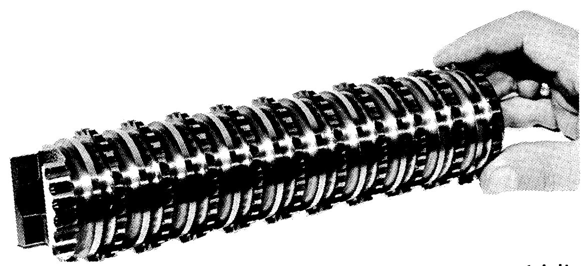


Fig. 10-65 The finishing shell on a Red Ring concentricity broach for a 2.1667-in. dia truck transmission gear splined hole.

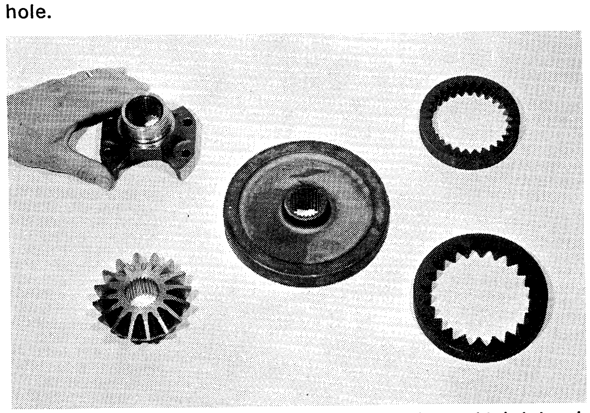


Fig. 10-66 Typical internal pump gears, universal joint, truck gear and bevel side gear whose processing has been simplified and improved by using Red Ring concentricity broaches.

SPECIAL BROACHES

National Broach has extensive design and manufacturing experience in the development of special high speed steel and carbide broaching tools in both slab and built-up designs.

This experience includes the manufacture of the associated broach holders as well as the slab broach detail.

Experience with involute and spline tooth forms has led to capability in the development of special tools for surface-broaching toothed members and contoured shapes.

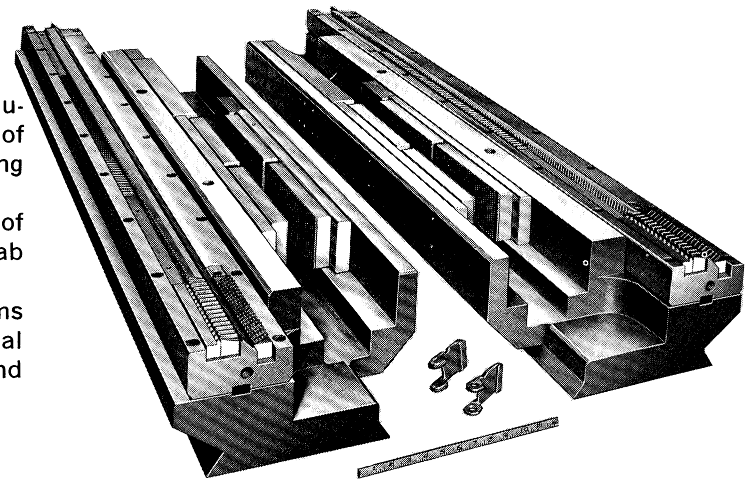
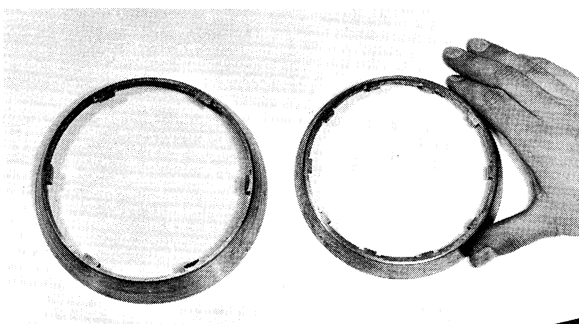


Fig. 10-67 A Red Ring surface broach assembly for producing a bracket. The assembly includes a main holder, subholder and Naloy broach inserts.



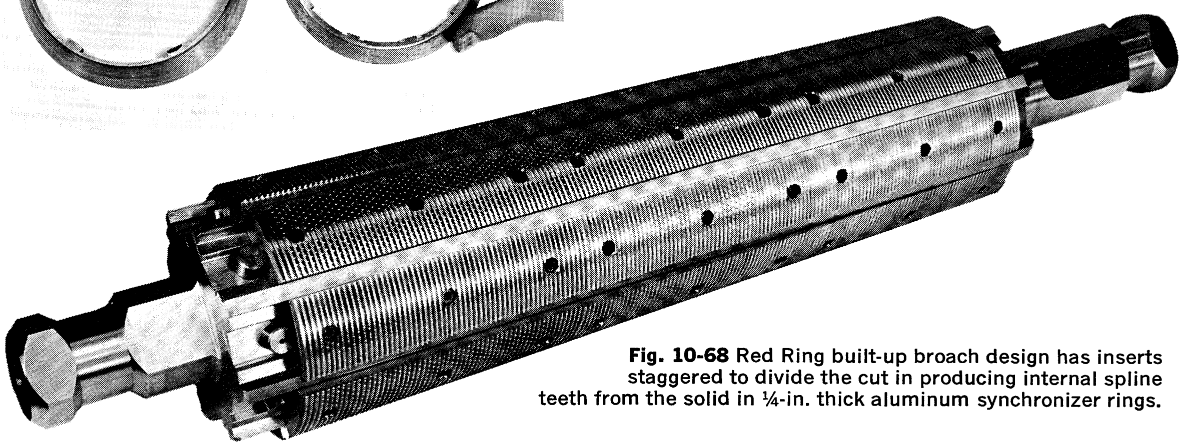


Fig. 10-68 Red Ring built-up broach design has inserts staggered to divide the cut in producing internal spline teeth from the solid in ¼-in. thick aluminum synchronizer rings.

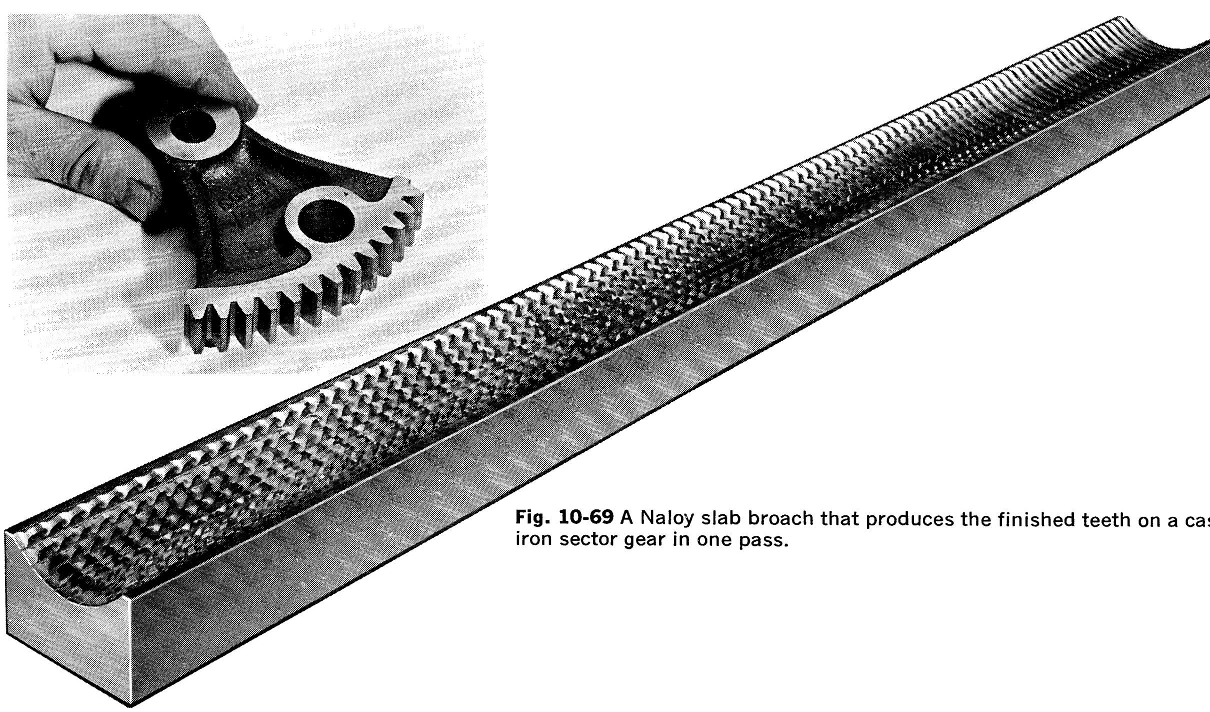


Fig. 10-69 A Naloy slab broach that produces the finished teeth on a cast iron sector gear in one pass.

Broach Fixtures

National Broach & Machine Div. can design and build the tooling for all types of broaching machines including the work holding and/or work-feeding fixtures. This experience includes hydraulic operation, magazine feeding and hydraulic clamping.

A standardized Red Ring unitized, self-contained air or hydraulic-powered broaching fixture has also been developed to economically handle the broaching of small parts in low, medium and high production. This is a versatile bench-type broaching unit.

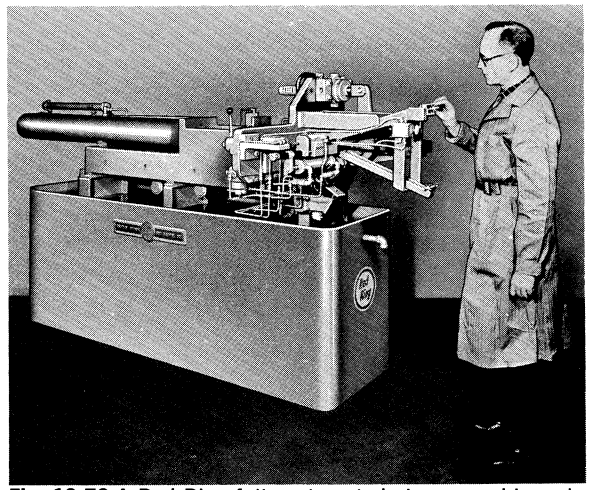


Fig. 10-70 A Red Ring fully-automated air-powered broaching fixture that broaches flats on the ends of motor shafts at a rate of 720 pieces per hour.

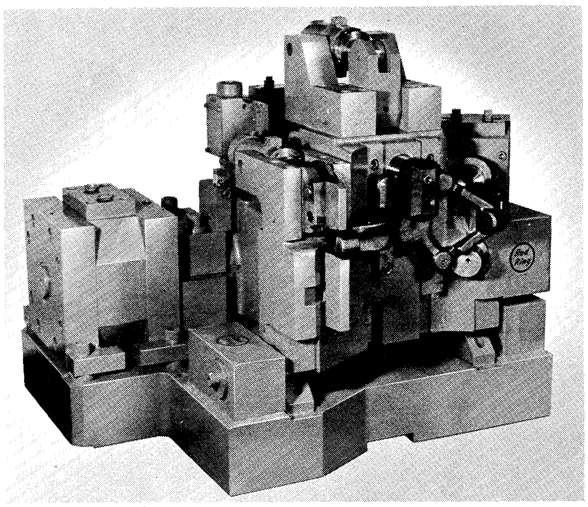
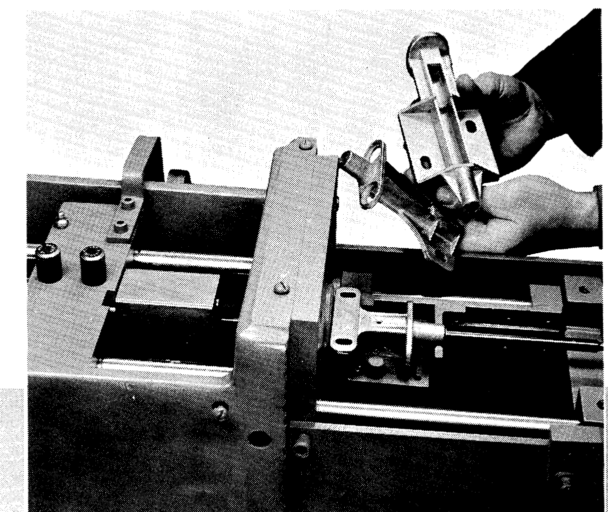


Fig. 10-71 Red Ring fixture for surface-broaching a ball joint steering knuckle uses hydraulic actuators and cylinders to clamp the part, and tip up the fixture for unloading.



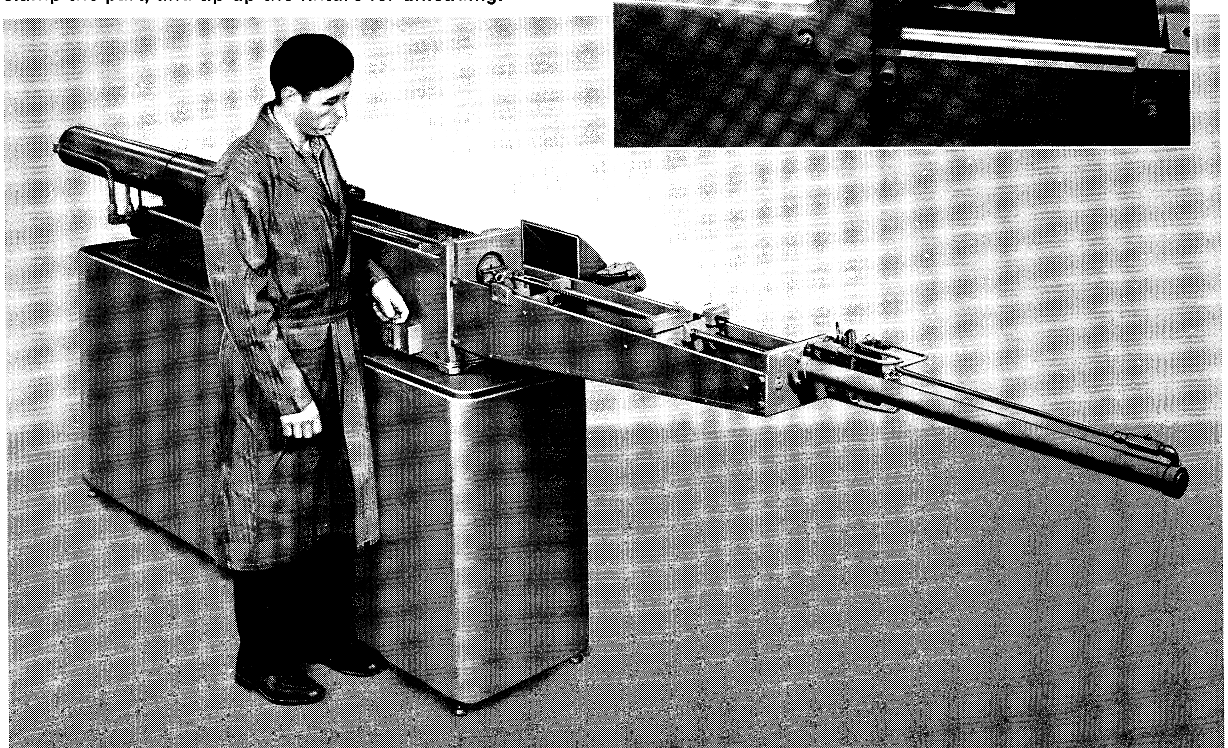


Fig. 10-72 Round holes in windshield wiper bracket bushings are broached in this Red Ring air-operated semi-automatic, self-contained broaching fixture in an 8-sec. machining cycle. Air cylinders operate the broach retriever and a part ejector.

Ring Gear Rounders

MODEL RRA GEAR ROUNDER

Ring gear rounding is a process developed by National Broach & Machine Div. to provide close control of the roundness of hardened internal spur or helical gears. The internal work gear is meshed with a pair of external burnishing gears. One gear is driven and the other is moved outward under pressure to roundup the work gear as it is rotated in a preset time sequence. Up to 75-per-cent of the out-of-roundness condition can be corrected by the process.

Red Ring Model RRA ring gear rounders have an automatic load and unload system. They will burnish the teeth and correct the out-of-round-ness condition of case-hardened internal ring gears up to 6-in. diameter at rates up to 300-gears per hour.

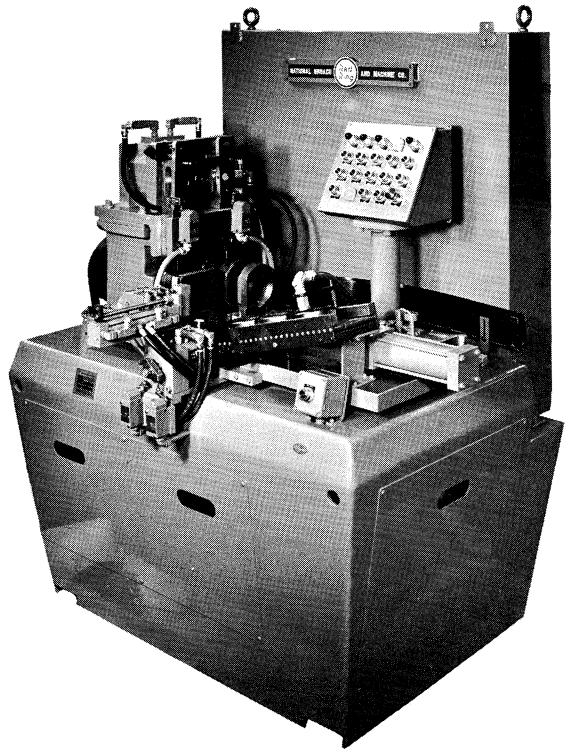


Fig. 10-73 Model RRA ring gear rounding machine.

Index Plates

Red Ring super-precision index plates can be used on machine tools, work fixtures and inspection equipment. These plates are made in three basic types up to 20-in. diameter. High accuracy is the result of more than 20 years of development work in designing and building index plates for proprietary broach, shaving cutter and master gear grinding equipment.

Guaranteed tooth spacing accuracy of Red Ring index plates is 0.0001-in. tooth-to-tooth and 0.0003-in. accumulative.

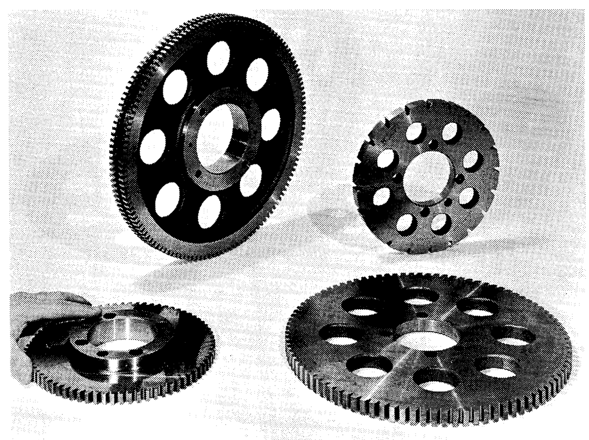


Fig. 10-74 Red Ring super precision index plates of (left to right) split-tooth, single-tooth hub, and single-tooth plain design.

Lead Bars

Red Ring ultra-precision lead bars, follower nuts and associated drive gearing are widely applied on broaching machines utilizing Red Ring helical broaches. They can also be used on spline grinders, special machine tools and inspection fixtures.

The lead bars are ground from solid high speed steel on special grinding equipment in which leads of 100-in. long bars can be held to an accuracy of less than 0.001-inch.

The follower nuts are cast with babbitt bearing material on precision-ground molding mandrels to provide backlash-free sliding and rotating motion.

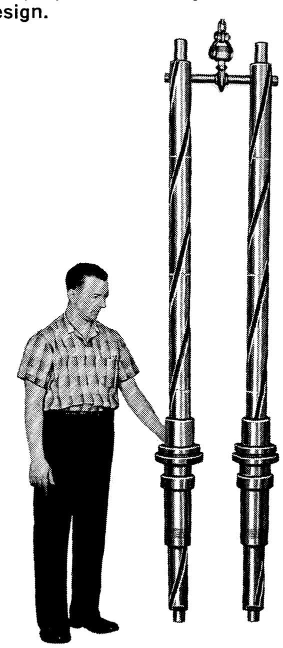


Fig. 10-75 Two Red Ring 100-in. long, 3¾ in diameter lead bars whose helical groove total lead accuracy is 0.0006-inch.

Expanding Arbors

Red Ring expanding arbors assure that gears of high quality are produced by utilizing workpiece bore locating and clamping methods that hold the gear blank firmly and concentric; and operate with consistent chucking repeatability.

These precision expanding arbors are ideally adapted to applications on gear cutting and gear finishing machines. Master Red Ring expanding arbors are used for precision inspection operations.

Machine expanding arbors feature a solid design in which tailstock advance causes a tapered internal plug to expand four jaws into contact with the workpiece plain or splined bore. These arbors are made in sizes from ½-in. dia. and up with one or more sets of jaws to clamp specific parts. Clamping repeatability is 0.0003-in. and less.

The master arbors are made in single wrench-operated four-jaw designs in four sizes from ⅜ to 1¼-in. diameter with clamping repeatability of 0.0002-in. and total expansion of 1/16-in.

Both types can be returned to National Broach for complete reconditioning.

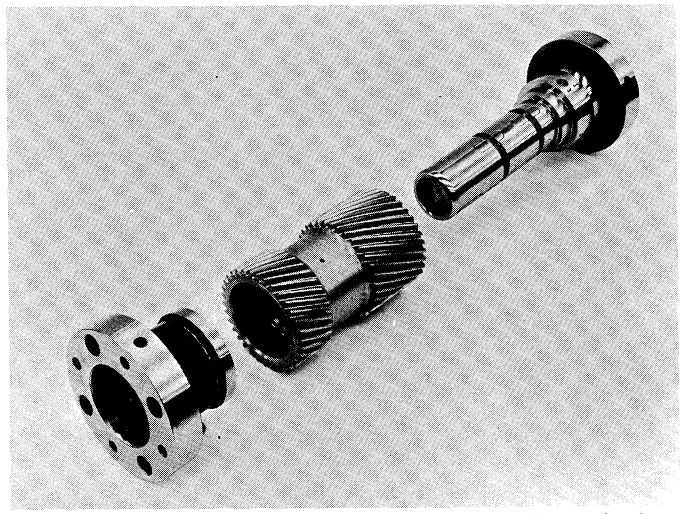


Fig. 10-76 A Red Ring expanding arbor with two sets of clamping jaws for mounting transmission sun gears on a rotary gear shaver.

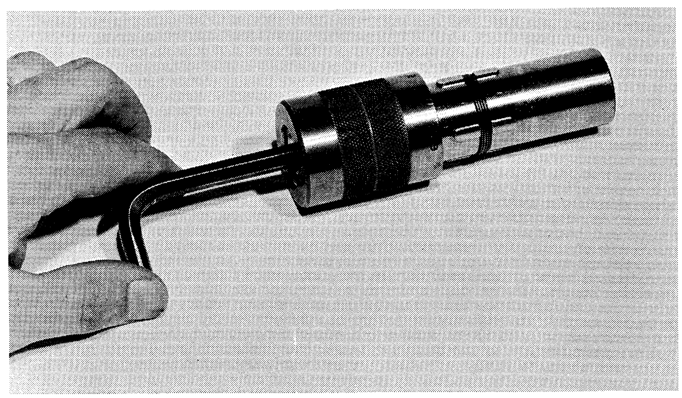


Fig. 10-77 Red Ring Master expanding arbor for inspection operations.

Gear Development

National Broach & Machine Division maintains one of the finest gear research and development facilities in the world for manufacturing, heat-treating, finishing and inspecting gears, transmissions and highly sophisticated, high performance drive gearing. Most of the new automotive, tractor, helicopter drive and gas turbine transmissions have been built in prototype and pilot production quantities at National Broach. Much aerospace and U.S. government gearing development work is done here.

The Red Ring gear development laboratory has equipment for turning and hobbing gears up to 48-in. dia.; shaping gears up to 18-in. dia.; shaving and honing gears up to 36-in. diameter, grinding gears up to 12-in. dia.; and inspecting both lead and involute profile of gearing up to 40-in. diameter.

Fig. 10-78 A portion of the extensive Red Ring gear development laboratory production and inspection equipment.

National Broach & Machine Division Patents

Certain features of the machines, methods and tools described in this book are covered by the following patents:

UNITED STATES

2,683,919	2,916,971	3,020,644	3,135,136	3,299,521
2,684,612	2,917,973	3,021,006	3,170,453	3,299,577
2,686,956	2,924,884	3,034,219	3,177,623	3,299,578
2,686,993	2,931,274	3,046,705	3,178,800	3,299,753
2,692,535	2,942,389	3,048,066	3,182,558	3,300,833
2,692,536	2,943,383	3,054,225	3,195,409	3,301,134
2,725,871	2,944,343	3,054,226	3,199,172	3,314,157
2,730,793	2,944,373	3,059,278	3,200,504	3,319,526
2,733,641	2,945,424	3,059,385	3,202,057	3,320,821
2,738,569	2,945,425	3,059,544	3,206,893	3,324,736
2,758,363	2,948,952	3,060,811	3,214,842	3,327,589
2,803,066	2,955,514	3,064,809	3,214,843	3,331,115
2,803,342	2,960,421	3,067,733	3,217,383	3,332,129
2,804,734	2,968,998	3,068,619	3,217,602	3,335,639
2,815,579	2,969,697	3,068,713	3,220,311	3,337,964
2,819,532	2,970,408	3,068,714	3,221,608	3,353,391
2,851,930	2,972,982	3,068,759	3,231,962	3,353,392
2,856,637	2,977,726	3,069,977	3,233,331	3,370,385
2,859,508	2,980,966	3,071,862	3,233,518	3,389,476
2,863,360	2,981,035	3,072,116	3,247,301	3,429,195
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2,877,658	2,987,801	3,092,934	3,267,552	3,461,526
2,882,798	2,990,657	3,092,935	3,267,581	3,487,516
2,886,990	2,990,658	3,095,782	3,269,020	3,492,914
2,887,014	2,994,988	3,095,980	3,272,041	3,494,752
2,887,015	2,994,989	3,097,566	3,272,075	3,505,763
2,893,744	2,995,941	3,097,567	3,276,099	3,505,847
2,898,670	3,006,060	3,099,882	3,276,100	3,505,911
2,904,938	3,006,117	3,106,805	3,280,467	3,546,760
2,905,062	3,007,563	3,110,132	3,280,675	3,552,167
2,905,320	3,008,218	3,110,224	3,284,909	3,553,095
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REISSUED U.S. PATENTS

Re. 26,248

CANADA

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548,437	606,248	651,664	684,501	782,225
558,313	606,987	651,776	688,204	794,094
568,692	610,656	655,048	693,081	794,095
574,625	619,906	656,774	697,920	803,800
576,712	625,408	656,786	701,248	805,899
580,481	629,003	659,306	702,265	819,377
580,572	635,837	661,827	709,136	828,409
587,660	636,086	662,101	715,301	838,928
591,016	637,110			850,889

ENGLAND

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797,282	848,275	902,921	987,973	1,145,403
800,680	864,688	907,924	1,002,378	1,146,901
806,399	865,355	911,216	1,005,153	1,151,547
811,341	865,847	911,218	1,006,014	1,173,794
811,419	871,184	919,784	1,006,015	1,182,945
812,867	877,462	943,478	1,030,270	1,188,234
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823,985	877,726	953,401	1,096,223	1,191,403
828,046	881,478		1,107,562	1,195,807

FRANCE

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1,153,954	1,186,895	1,247,300	1,326,180	1,478,734
1,160,740	1,198,903	1,248,317	1,327,570	1,479,700
P/A 72,672	1,206,740	1,248,338	1,355,171	1,487,969
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	1,208,485	1,272,339	1,403,305	1,510,637
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P/A 72,573	1,216,784	1,278,975	1,429,574	1,562,470
				1,601,424

GERMANY

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ITALY

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563,927	597,477	626,856	639,638	861,046
584,433	615,782	627,941	655,310	

JAPAN

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252,813	268,450	295,040	318,646	404,710
252,974	270,504	300,071	319,260	405,222
254,998	272,392	300,860	400,463	458,521
259,941	272,803	301,702	401,475	486,304
263,830				505,418

SWEDEN

196,376 302,232

SWITZERLAND

458,883

National Broach & Machine Division Trademarks

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