GEAR MATERIALS, PROPERTIES, AND MANUFACTURE

Edited by

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Preface

Gears, because of their unique contribution to the operation of so many machines and mechanical devices, have received special attention from the technical community for more than two millennia. New developments in gear technology, particularly from the materials point-of-view, have also been covered in detail by ASM International for many years. Numerous forums, conference proceedings, books, and articles have been devoted to the understanding of gear performance by examining gear tribology, failure modes, the metallurgy of ferrous gear materials, heat treatment, gear manufacturing methods, and testing. All of these important technical aspects of gear technology are brought together in the present offering, *Gear Materials, Properties, and Manufacture*.

Chapter 1, "Basic Understanding of Gears," discusses the various types of gears used, important gear nomenclature, and applied stresses and strength requirements associated with gears. It also provides an overview of several important topics that are covered in greater detail in subsequent chapters, namely, gear materials, gear manufacture, and heat treatment. Gear tribology and lubrication is covered in Chapter 2. Lubrication-related failures (pitting, wear, and scuffing), elastohydrodynamic lubrication, lubricant selection, and gear lubricant application are among the subjects described.

Chapters 3 and 4 describe both metallic (ferrous and nonferrous alloys) and plastic gear materials, respectively. Emphasis in Chapter 3 has been placed on the properties of carburized steels, the material of choice for high-performance power transmission gearing. The increasing use of plastics for both motion-carrying and power transmission applications is covered in Chapter 4.

Chapters 5, 6, and 7 address methods for manufacturing gears including metal removal processes (machining, grinding, and finishing), casting, forming, and forging (including recent advances in near-net shape forging of gears), and powder metallurgy processing. Injection molding, another important method for the manufacture of plastic gears, is covered in Chapter 4.

The heat treatment of gears is reviewed in Chapters 8 through 12. Both through hardening and surface hardening methods are reviewed. Again, emphasis has been placed on carburizing, the most common heat treatment applied to gear steels. It should be noted that some of the material presented in these chapters was adapted, with the kind permission of the author, from *Heat Treatment of Gears: A Practical Guide for Engineers*, by A.K. Rakhit (ASM International, 2000). Dr. Rakhit's book is an excellent resource for those seeking a more in-depth reference guide to gear heat treatment.

Failure analysis, fatigue life prediction, and mechanical testing are examined in Chapters 13, 14, and 15, respectively. In Chapter 13, "Gear Failure Modes and Analysis," emphasis has been placed on two of the most common types of gear failure—bending fatigue and contact fatigue. Bending fatigue of carburized steels is also discussed in depth in Chapter 3.

In summary, this book is intended for gear metallurgists and materials specialists, manufacturing engineers, lubrication technologists, and analysts concerned with gear failures who seek a better understanding of gear performance and gear life. It supplements other gear texts that emphasize the design, geometry, and theory of gears.

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CHAPTER 1

Basic Understanding of Gears

GEARS are machine elements that transmit rotary motion and power by the successive engagements of teeth on their periphery. They constitute an economical method for such transmission, particularly if power levels or accuracy requirements are high. Gears have been in use for more than three thousand years and they are an important element in all manner of machinery used in current times. Application areas for gears are diverse and include—to name a few:

- Small, low-cost gears for toys
- Gears for office equipment
- Bicycle gears
- Appliance gears
- Machine tool gears
- Control gears
- Automotive gears
- Transportation gears
- Marine gears
- Aerospace gears
- Gears in the oil and gas industry
- Gears for large mills that make cement, grind iron ore, make rubber, or roll steel

Gears range in size from recently developed micrometer-sized gears for electric motors no bigger than a grain of sand to gears as large as 30 m (100 ft) in diameter. Gear materials range from lightweight plastics to ultrahigh-strength heat-treated steels.

Gear Nomenclature

Before discussing the various types of gears used, this section will review some of the terms used in the gear industry to describe the design of gears and gear geometries. Figure 1 shows schematically typical gear nomenclature. It

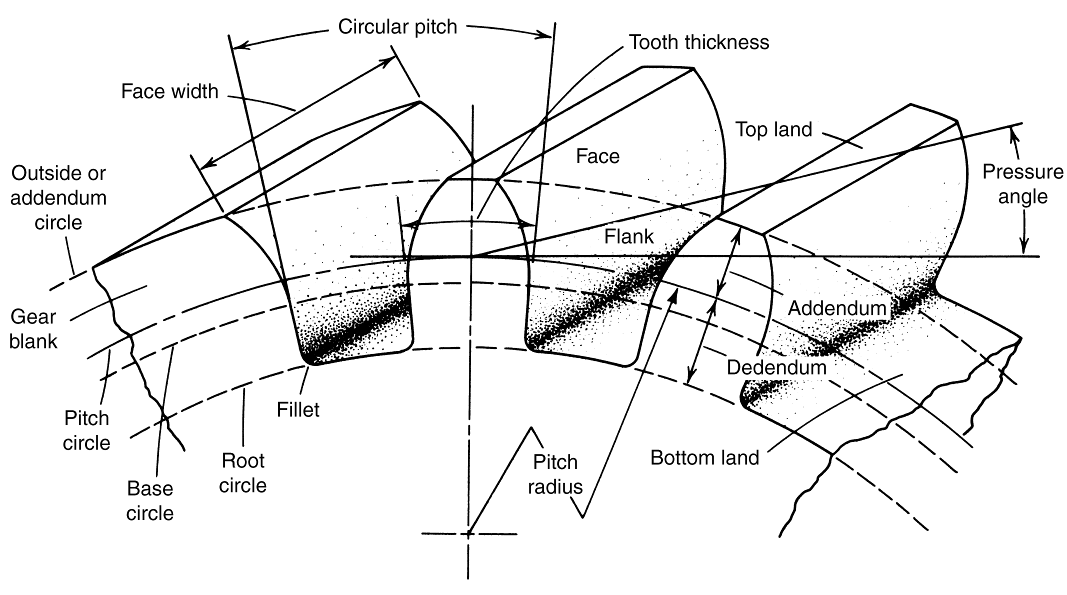

Fig. 1 Schematic of typical gear tooth nomenclature

should be noted that only the most common terms are discussed below. More detailed information on gear nomenclature can be found in various standards published by the American Gear Manufacturers Association (AGMA), most notably AGMA 1012-F90, "Gear Nomenclature, Definitions of Terms with Symbols."

active profile. The part of the gear tooth profile that actually comes in contact with the profile of the mating gear while in mesh (Fig. 2).

addendum. The height of the tooth above the pitch circle (Fig. 1).

backlash. The amount by which the width of a tooth space exceeds the thickness of the engaging tooth on the operating pitch circle (Fig. 3).

base circle. The circle from which the involute tooth profiles are generated (Fig. 1).

bottom land. The surface at the bottom of a tooth space adjoining the fillet (Fig. 1 and 2).

center distance. The distance between the axes of rotation between two mating gears.

circular pitch. Length of arc of the pitch circle between corresponding points on adjacent teeth (Fig. 1).

circular thickness. The length of arc between the two sides of a gear tooth at the pitch circle.

dedendum. The depth of the tooth below the pitch circle (Fig. 1).

diametral pitch (DP). A measure of tooth size in the English system. In units, it is the number of teeth per inch of pitch diameter. As the tooth size increases, the diametral pitch decreases (Fig. 4). Diametral pitches range from 0.5 to 200. Coarse pitch gears are those with a diametral pitch of ≤20. Fine pitch gears are those with a diametral pitch of >20. Table 1 shows the various tooth dimensions for different diametral pitches of spur gears.

face width. The length of the gear teeth in an axial plane (Fig. 1).

fillet radius. The radius of the fillet curve at the base of the gear tooth (Fig. 2).

gear. A geometric shape that has teeth uniformly spaced around the circumference. In general, a gear is made to mesh its teeth with another gear.

gear blank. The workpiece used for the manufacture of a gear, prior to machining the gear teeth.

gear pinion. When two gears mesh together, the smaller of the two is called the pinion; the larger is called the gear (Fig. 5).

gear quality numbers. AGMA gear quality numbers ranging from 3 to 15 identify the

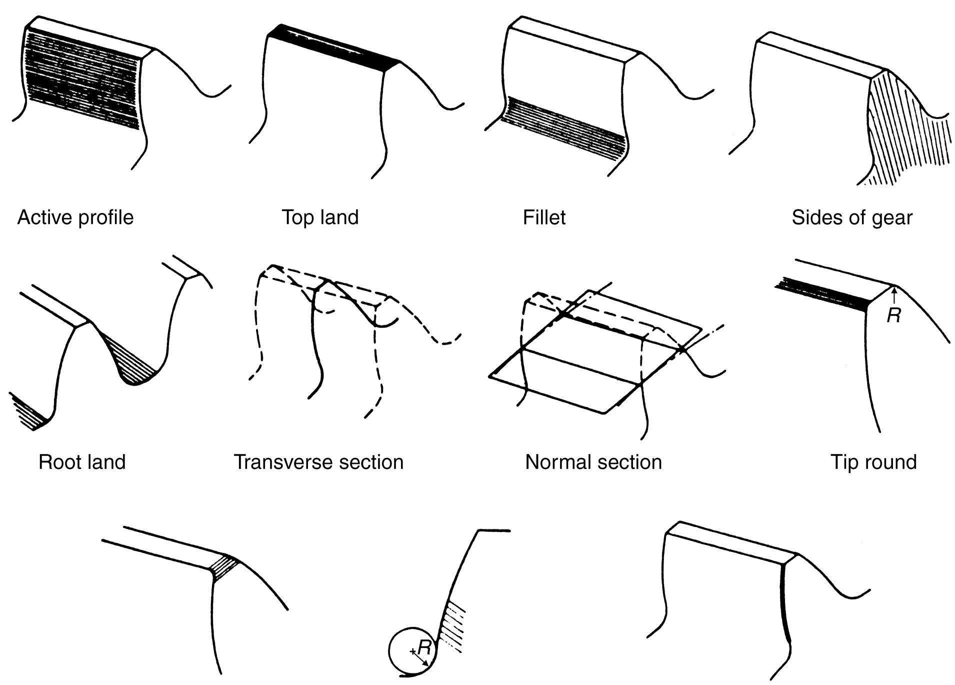

Fig. 2 Nomenclature of gear contact areas and boundary zones

accuracy level of the tooth element tolerances permissible in the manufacture of a particular gear in terms of its specialized use. The higher the number, the greater the level of accuracy. Numbers 3 through 7 are for commercial applications such as appliances, numbers 8 through 13 are for precision applications, and numbers 14 and 15 are for ultraprecision applications. The permissible tolerances for the different quality numbers may be obtained from the AGMA standards, which show the type of gear and the permissible tolerances and inspection dimensions.

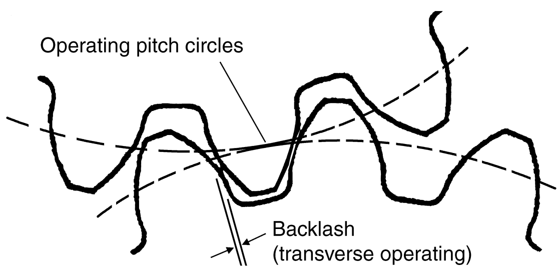

Fig. 3 Schematic of gear backlash. Source: Ref 1

gear ratio. The ratio of the larger to the smaller number of teeth in a pair of gears.

helix angle. The angle between any helix and an element of its cylinder. In helical gears and worms, it is at the standard pitch circle unless otherwise specified.

involute gear tooth. A gear tooth whose profile is established by an involute curve outward from the base circle (Fig. 6).

normal section. A section through a gear that is perpendicular to the tooth at the pitch circle (Fig. 2).

pitch circle. The circumference of a gear measured at the point of contact with the mating gear (Fig. 1 and 7).

pitch diameter. The diameter of the pitch circle.

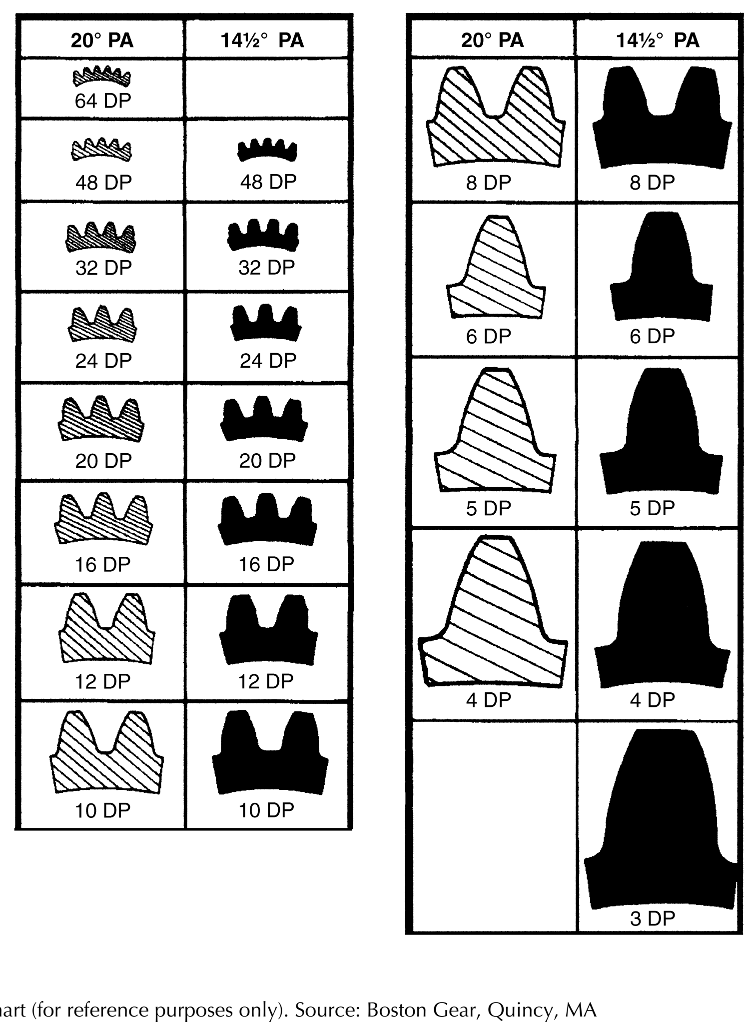

Fig. 4 Tooth gage chart (for reference purposes only). Source: Boston Gear, Quincy, MA

pitch line. In a cross section of a rack, the pitch line corresponds to the pitch circle in the cross section of the gear (Fig. 7).

pitch point. The tangency point of the pitch circles of two mating gears (Fig. 7).

pitch radius. The radius of the pitch circle in a cross section of gear teeth in any plane other than a plane of rotation (Fig. 1).

pressure angle. The angle between a tooth profile and a radial line at its pitch point (Fig. 1). The pressure angle of an involute gear tooth is determined by the size ratio between the base circle and the pitch circle (Fig. 8). Common pressure angles used by the gear industry are 14.5, 20, and 25°.

rack. A rack is a gear having a pitch circle of infinite radius. Its teeth lie along a straight line on a plane. The teeth may be at right angles to the edge of the rack and mesh with a spur gear (Fig. 5 and 9b), or the teeth on the rack may be at some other angle and engage a helical gear (Fig. 10b).

Table 1 Gear cutting table showing various tooth dimensions for different diametral pitches of spur gears

Diametral pitch	Circular pitch, in.	Thickness of tooth on pitch line, in.	Depth to be cut in gear(a), in.	Addendum, in.
3	1.0472	0.5236	0.7190	0.3333
4	0.7854	0.3927	0.5393	0.2500
5	0.6283	0.3142	0.4314	0.2000
6	0.5236	0.2618	0.3565	0.1667
8	0.3927	0.1963	0.2696	0.1250
10	0.3142	0.1571	0.2157	0.1000
12	0.2618	0.1309	0.1798	0.0833
16	0.1963	0.0982	0.1348	0.0625
20	0.1571	0.0785	0.1120	0.0500
24	0.1309	0.0654	0.0937	0.0417
32	0.0982	0.0491	0.0708	0.0312
48	0.0654	0.0327	0.0478	0.0208
64	0.0491	0.0245	0.0364	0.0156

(a) Hobbed gears. Source: Boston Gear, Quincy, MA

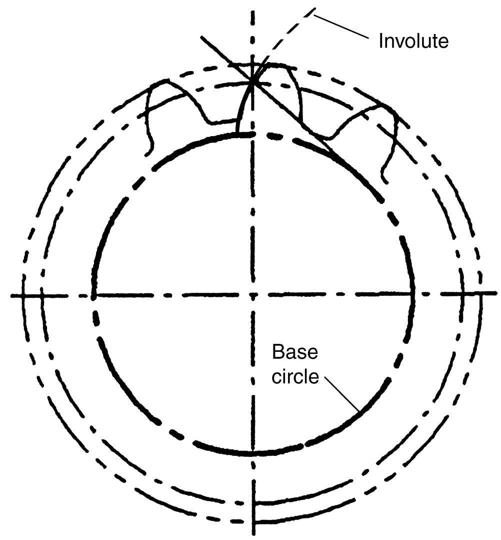

Fig. 6 Schematic of an involute gear tooth. Source: Ref 1

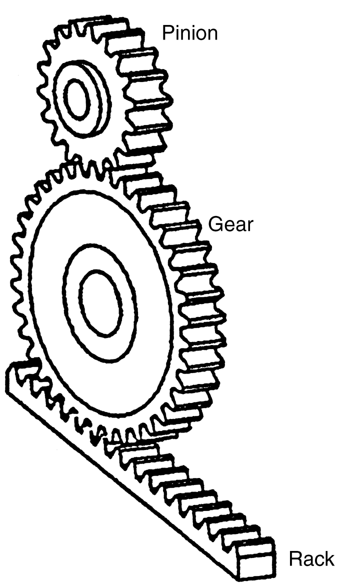

Fig. 5 The pinion, gear, and rack portions of a spur gear. Source: Ref 1

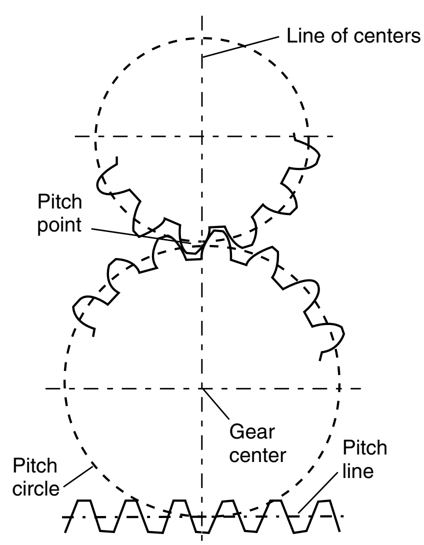

Fig. 7 Schematic of pitch nomenclature. Source: Ref 1

root circle. The root circle coincides with the bottoms of the tooth spaces (Fig. 1).

tooth thickness. The thickness of the tooth measured at the pitch circle (Fig. 1).

top land. The surface of the top of a tooth (Fig. 1 and 2).

transverse section. A section through a gear perpendicular to the axis of the gear (Fig. 2).

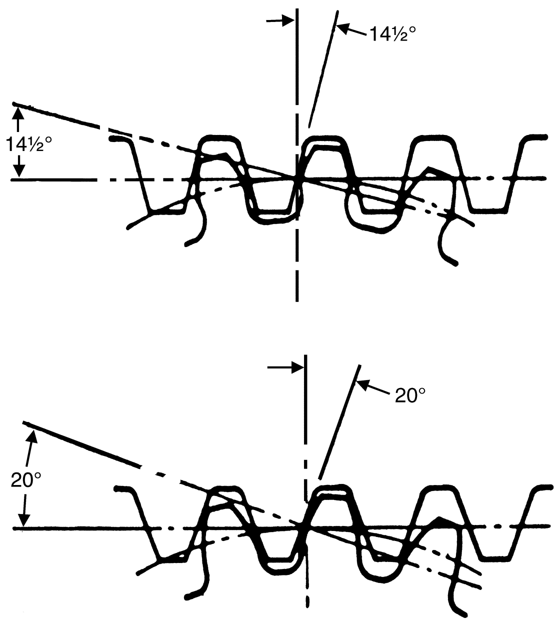

Fig. 8 Schematic of two common pressure angles. Source: Boston Gear, Quincy MA

Types of Gears

There is a wide variety of types of gears in existence, each serving a range of functions. In order to understand gearing, it is desirable to classify the more important types in some way. One approach is by the relationship of the shaft axes on which the gears are mounted. As listed in Table 2, shafts may be parallel, intersecting, or nonintersecting and nonparallel.

Types of Gears that Operate on Parallel Shafts

Spur gears (Fig. 5 and 9) are used to transmit motion between parallel shafts or between a shaft and a rack. The teeth of a spur gear are radial, uniformly spaced around the outer periphery, and parallel to the shaft on which the gear is mounted. Contact between the mating teeth of a spur gear is in a straight line parallel to the rotational axes, lying in a plane tangent to the pitch cylinders of the gears (a pitch cylinder is the imaginary cylinder in a gear that rolls without slipping on a pitch cylinder or pitch plane of another gear).

Helical gears (Fig. 10a) are used to transmit motion between parallel or crossed shafts or between a shaft and a rack by meshing teeth that lie along a helix at an angle to the axis of the shaft. Because of this angle, mating of the teeth occurs such that two or more teeth of each gear are always in contact. This condition permits smoother action than that of spur gears. How-

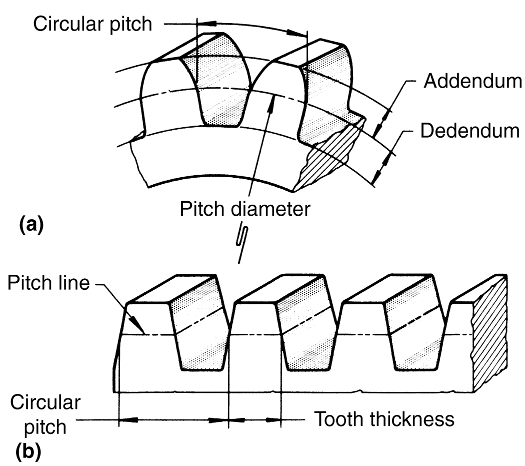

Fig. 9 Sections of a spur gear (a) and a spur rack (b)

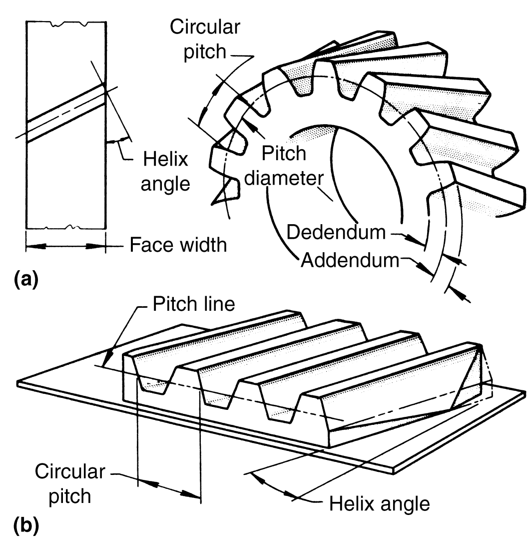

Fig. 10 Sections of a helical gear (a) and a helical rack (b)

ever, unlike spur gears, helical gears generate axial thrust, which causes slight loss of power and requires thrust bearings.

Herringbone gears (Fig. 11), sometimes called double helical gears, are used to transmit motion between parallel shafts. In herringbone gears, tooth engagement is progressive, and two or more teeth share the load at all times. Because they have right-hand and left-hand helixes, herringbone gears are usually not subject to end thrust. Herringbone gears can be operated at higher pitch-line velocities than spur gears.

Internal gears are used to transmit motion between parallel shafts. Their tooth forms are similar to those of spur and helical gears except that the teeth point inward toward the center of the gear. Common applications for internal gears include rear drives for heavy vehicles, planetary gears, and speed-reducing devices. Internal gears are sometimes used in compact designs because the center distance between the internal gear and its mating pinion is much smaller than that required for two external gears. A typical relation between an internal gear and a mating pinion is shown in Fig. 12.

Table 2 Types of gears in common use

Parallel axes
Spur external
Spur internal
Helical external
Helical internal
Intersecting axes
Straight bevel
Zerol bevel
Spiral bevel
Face gear
Nonintersecting and nonparallel axes
Crossed helical
Single-enveloping worm
Double-enveloping worm
Hypoid
Spiroid

Types of Gears that Operate on Intersecting Shafts

Bevel gears transmit rotary motion between two nonparallel shafts. These shafts are usually at 90° to each other.

Straight bevel gears (Fig. 13a) have straight teeth that, if extended inward, would intersect at the axis of the gear. Thus, the action between mating teeth resembles that of two cones rolling on each other (see Fig. 14 for angles and terminology). The use of straight bevel gears is generally limited to drives that operate at low speeds and where noise is not important.

Spiral bevel gears (Fig. 13b) have teeth that are curved and oblique. The inclination of the teeth results in gradual engagement and continuous line contact or overlapping action; that is, more than one tooth will be in contact at all times. Because of this continuous engagement, the load is transmitted more smoothly from the driving to the driven gear than with straight bevel gears. Spiral bevel gears also have greater load-carrying capacity than their straight counterparts. Spiral bevel gears are usually preferred to straight bevel gears when speeds are greater than 300 m/min (1000 sfm), and particularly for very small gears.

Zerol bevel gears (Fig. 13c) are curved-tooth bevel gears with zero spiral angle. They differ from spiral bevel gears in that the teeth

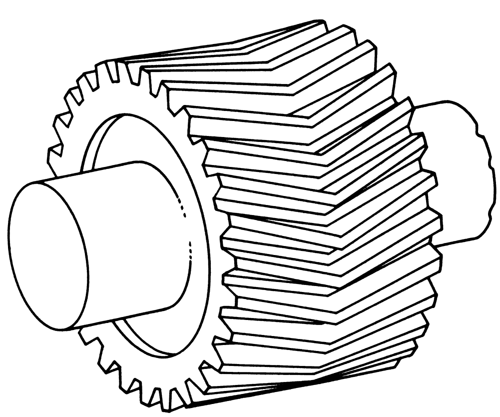

Fig. 11 A typical one-piece herringbone gear. The opposed helixes permit multiple-tooth engagement and eliminate end thrust.

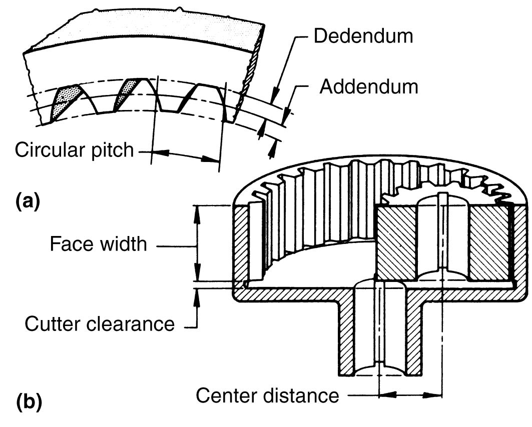

Fig. 12 Section of a spur-type internal gear (a) and relation of internal gear with mating pinion (b)

are not oblique. They are used in the same way as spiral bevel gears, and they have somewhat greater tooth strength than straight bevel gears.

Face gears have teeth cut on the end face of a gear, as the term face gear implies. They are not ordinarily thought of as bevel gears, but functionally they are more akin to bevel gears than to any other type.

A spur pinion and a face gear are mounted (like bevel gears) on shafts that intersect and have a shaft angle (usually 90°). The pinion bearings carry mostly radial load, while the gear bearings have both thrust and radial load. The mounting distance of the pinion from the pitch-cone apex is not critical, as it is in bevel or hypoid gears. Figure 15 shows the terminology used with face gears.

The pinion that goes with a face gear is usually made spur, but it can be made helical if necessary. The formulas for determining the dimensions of a pinion to run with a face gear are no different from those for the dimensions of a pinion to run with a mating gear on parallel axes. The pressure angles and pitches used are similar to spur gear (or helical gear) practice.

The gear must be finished with a shaper-cutter that is almost the same size as the pinion. Equipment for grinding face gears is not available. The teeth can be lapped, and they can be shaved without too much difficulty, although ordinarily shaving is not used.

The face gear tooth changes shape from one end of the tooth to the other. The face width of the gear is limited at the outside end by the radius at which the tooth becomes pointed. At the inside end, the limit is the radius at which

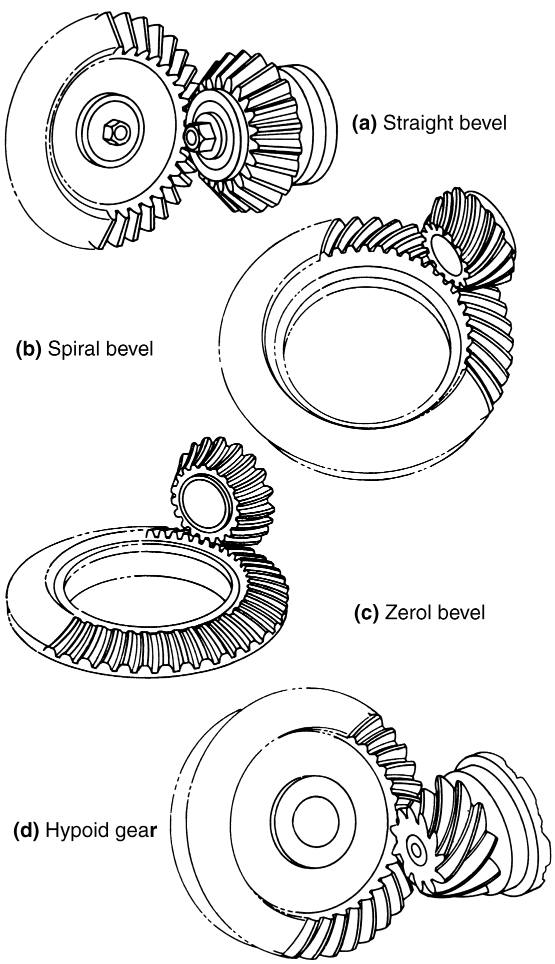

Fig. 13 Three types of bevel gears and a hypoid gear

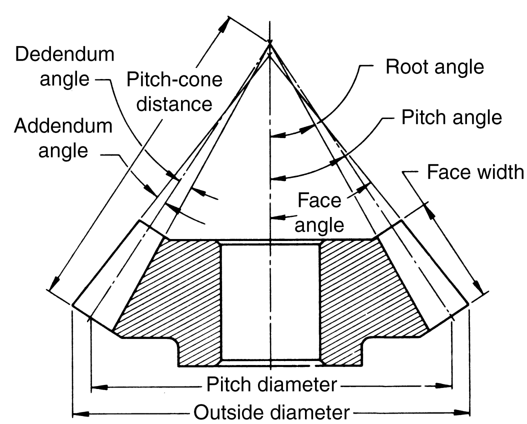

Fig. 14 Angles and terminology for straight bevel gears

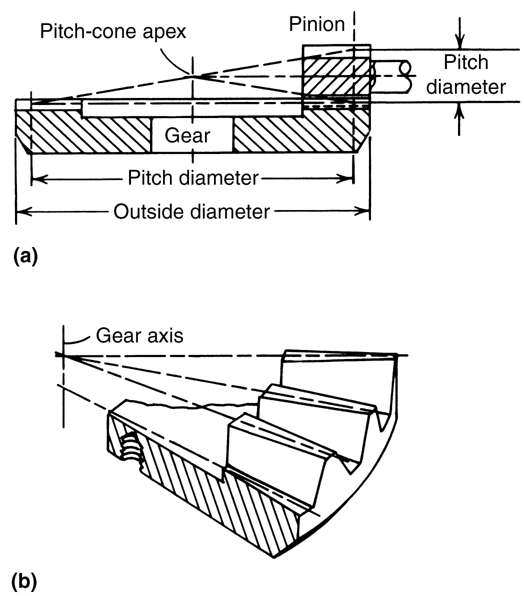

Fig. 15 Face gear terminology. (a) Cross-sectional view showing gear and pinion positions. (b) Relationship of gear teeth to gear axis

undercut becomes excessive. Due to practical considerations, it is usually desirable to make the face width somewhat short of these limits. The pinion to go with a face gear is usually made with a 20° pressure angle.

Types of Gears that Operate on Nonparallel and Nonintersecting Shafts

Worm gear sets are usually right-angle drives consisting of a worm gear (or worm wheel) and a worm. A single-enveloping worm gear set has a cylindrical worm, but the gear is throated (that is, the gear blank has a smaller diameter in the center than at the ends of the cylinder, the concave shape increasing the area of contact between them) so that it tends to wrap around the worm. In the double-enveloping worm gear set, both members are throated, and both members wrap around each other. A double-enveloping worm gear set is shown in Fig. 16. Worm gear sets are used where the ratio of the speed of the driving member to the speed of the driven member is large, and for a compact right-angle drive.

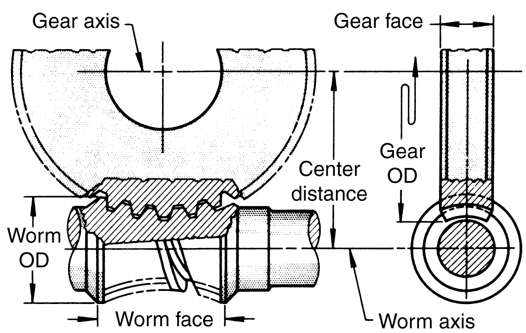

Fig. 16 Mating of worm gear (worm wheel) and worm in a double-enveloping worm gear set

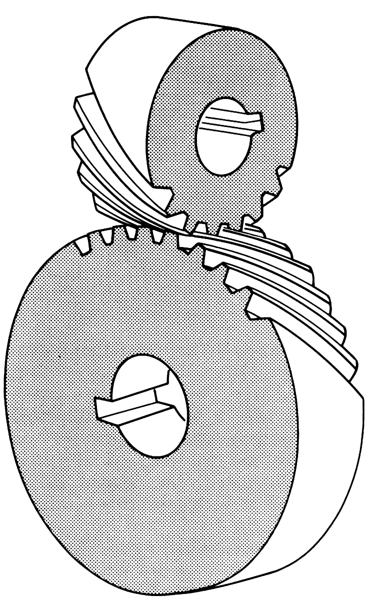

Fig. 17 Mating crossed-axes helical gears

Crossed-helical gears are essentially nonenveloping worm gears, that is, both members are cylindrical (Fig. 17). The action between mating teeth has a wedging effect, which results in sliding on tooth flanks. These gears have low load-carrying capacity, but are useful where shafts must rotate at an angle to each other.

Hypoid gears (Fig. 13d) are similar to spiral bevel gears in general appearance. The important difference is that the pinion axis of the hypoid pair of gears is offset somewhat from the gear axis. This feature provides many design advantages. In operation, hypoid gears run even more smoothly and quietly than spiral bevel gears and are somewhat stronger.

Spiroid gears consist of a tapered pinion that somewhat resembles a worm (Fig. 18) and a gear member that is a face gear with teeth curved in a lengthwise direction; the inclination to the tooth is like a helix angle, but not a true helical spiral. The combination of a high gear ratio in compact arrangements, low cost when mass produced, and good load-carrying capacity makes these gears attractive in many applications.

Proper Gear Selection

The first step in designing a set of gears is to select the correct type. In many cases, the geometric arrangement of the apparatus that needs a gear drive will considerably affect the selec-

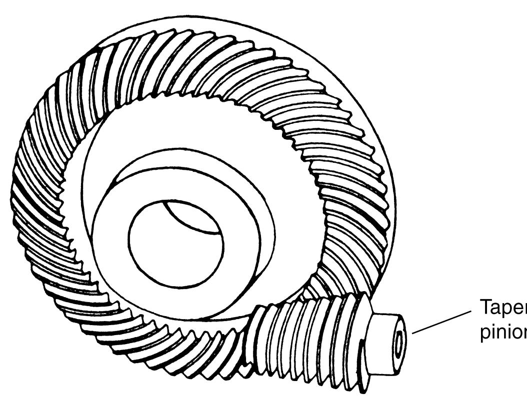

Fig. 18 Spiroid gear design

tion. If the gears must be on parallel axes, then spur or helical gears are appropriate. Bevel and worm gears can be used if the axes are at right angles, but they are not feasible with parallel axes. If the axes are nonintersecting and nonparallel, then crossed-helical gears, hypoid gears, worm gears, or Spiroid gears can be used. Worm gears, though, are seldom used if the axes are not at right angles to each other.

There are no dogmatic rules that tell the designer which gear to use. The choice is often made after weighing the advantages and disadvantages of two or three types of gears. Some generalizations, though, can be made about gear selection.

External helical gears are generally used when both high speeds and high horsepowers are involved. External helical gears have been built to carry as much as 45,000 kW (60,000 hp) of power on a single pinion and gear. Larger helical gears could also be designed and built. It is doubtful if any other type of gear could be built and used successfully to carry this much power on a single mesh.

Bevel and Hypoid Gears. Bevel gears are ordinarily used on right-angle drives when high efficiency is needed. These gears can usually be designed to operate with 98% or better efficiency. Hypoid gears do not have as good efficiency as bevel gears, but hypoid gears can carry more power in the same space, provided the speeds are not too high.

Worm gears are ordinarily used on right-angle drives when very high ratios (single-thread worm and gear) are needed. They are also widely used in low-to-medium ratios (multiple-thread worm and gear) as packaged speed reducers. Single-thread worms and worm gears are used to provide the mechanical indexing accuracy on many machine tools. The critical function of indexing hobbing machines and gear shapers is nearly always done by worm gear drive. Worm gears seldom operate at efficiencies above 90%.

Spur gears are relatively simple in design and in the machinery used to manufacture and check them. Most designers prefer to use them wherever design requirements permit.

Spur gears are ordinarily thought of as slow-speed gears, while helical gears are thought of as high-speed gears. If noise is not a serious design problem, spur gears can be used at almost any speed that can be handled by other types of gears. Aircraft gas-turbine precision spur gears sometimes operate at pitch-line speeds above 50 m/s (10,000 sfm). In general, though, spur gears are not used much above 20 m/s (4000 sfm).

Basic Applied Stresses (Ref 2)

The loads applied to one tooth by the action of its mating tooth are at any moment of time a line contact at the most; or, at the least, a point contact (more detailed information on gear tooth contact can be found in Chapter 14, "Fatigue and Life Prediction"). As the loads are increased, the line may lengthen or even broaden, or the point may expand to a rounded area.

The basic stresses applied to a gear tooth include the six types listed in Fig. 19; often, a combination of two or three types are applied at a time. Commonly they are tensile, compressive, shear (slide), rolling, rolling-slide, and torsion. Each type of gear tooth will have its own characteristic stress patterns.

Spur Gear. As the contacting tooth moves up the profile of the loaded tooth, a sliding-rolling action takes place at the profile interface. At the pitchline, the stresses are pure rolling. Above the pitchline, the rolling-sliding action again takes over, but the sliding will be in the opposite direction. Keep in mind that the action on the profile of the contacting tooth is exactly the same as the loaded tooth except in reverse

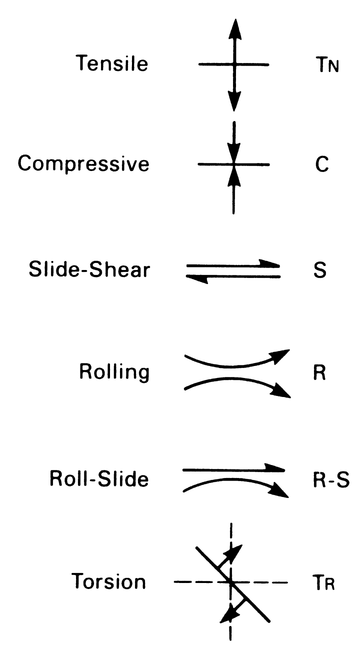

Fig. 19 Basic stresses that are applied to gear teeth. Often, two or three are simultaneously applied to a specific area. Source: Ref 2

order (see Fig. 20). The sliding action of two surfaces, when lubricated properly, will have no problem. However, surface disparities, insufficient lubrication, improper surface hardness, higher temperatures, and abrasive or adhesive foreign particles will contribute to a breakdown during a sliding contact. At the same time, there is a tensile stress at the root radius of the loaded side of the tooth and a compressive stress at the root radius of the opposite side.

Helical Gear. The helical gear tooth receives the same contact action as the spur gear; i.e., a rolling-sliding action from the lowest point of active profile up to the pitchline, rolling over the pitchline, then sliding-rolling from the pitchline over the addendum. An additional stress is being applied to the helical tooth; a lateral sliding action is applied at all contact levels, including the pitchline. The force component at 90° to the direction of rotation increases as the helical angle increases. Resultants of this side thrust are often overlooked (see Fig. 21). The web between the center shaft hub and the outer gear rim is constantly undergoing a cycle of bending stress; it is not uncommon for a relatively thin web to fail in bending fatigue. If the hub of the gear faces against a thrust bearing, the bearing itself is under a constant thrust load. The shaft carrying the gear undergoes a continual rotation bending stress. It is also not uncommon to have such a shaft fail by rotational bending fatigue. The above secondary stresses are found only in a gear of a single-helix pattern. A double-helical gear or a herringbone gear will not have a side thrust component of stress; therefore, the entire stress load will be absorbed by the teeth.

One additional stress that should be discussed at this time is a stress common to all gearing because it involves rolling surfaces. It is a shear stress running parallel to the surface at a distance from 0.18 to 0.3 mm (0.007 to 0.012 in.) below the surface. The distance below the surface given above is the average depth for a normal loading condition. The actual depth of maximum shear could be deeper, depending on the radius of curvature of the mating surfaces and the tangential forces being applied. In one instance there has been evidence of rolling loads above the shear strength as deep as 0.86 mm (0.034 in.). The subsurface shear stress is most often the originator of initial line pitting along the pitchline of gear teeth, line pitting low on the profile due to tooth tip interference, line pitting along the tooth tip due to the same tooth tip interference, and subsurface rolling contact fatigue. The subject of rolling contact fatigue is discussed more fully in Chapters 3, "Ferrous and Nonferrous Alloys" and 13, "Gear Failure Modes and Analysis."

Straight Bevel Gear. The straight bevel gear undergoes the same stresses as discussed above, including a very slight helical action laterally. The larger sliding action component is parallel to the axis of the gears and tends to push

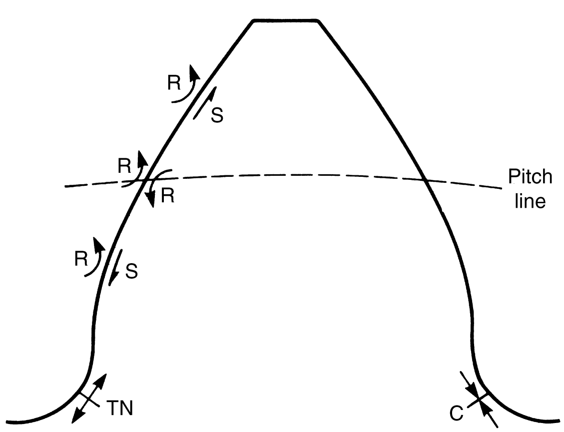

Fig. 20 Diagramatic stress areas on basic spur gear tooth. Source: Ref 2

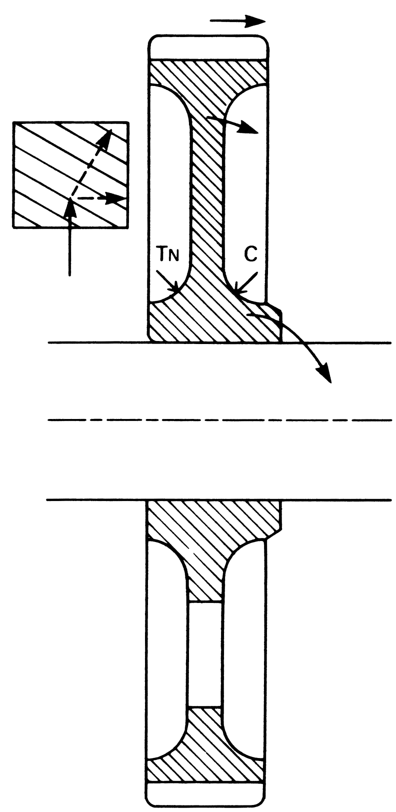

Fig. 21 Secondary stresses set up in associated parameters of a helical gear due to the side thrust action of the helix. Source: Ref 2

the gears apart, causing a higher profile contact, and to exert a rotational bending stress in the web of the part as well as in the shaft.

Spiral Bevel Gear. Aside from all the stresses applied above, a spiral bevel gear has a resultant peculiar to itself. As the rolling-sliding stress tends to move in a straight line laterally, the progression of the points along the stress line moves in a bias across the profile of the tooth. As long as at least two teeth are in contact, the resulting load per unit area is well within reasonable limits. However, there are circumstances (and it may be only momentary) when there is a 1-to-1 tooth contact. This very narrow line contact may be accepting an extremely high load per unit area, and a line of pitting will result early in the life of the tooth. Careful attention should be given to the design characteristics of these parts, such as spiral angle and pressure angle.

Hypoid Gear. The hypoid gearing has the same applied stresses as those discussed for the spiral bevel, but sliding becomes the more predominant factor. This predominance increases as the axis of the pinion is placed farther from the central axis of the gear, and is maximum when the set becomes a high-ratio hypoid.

Strength (Ref 2)

The strength of any component is measured by the amount of stress that can be tolerated before permanent strain (deformation) takes place.

Strain, or deflection under load, is a constant for steel regardless of hardness or heat treatment. The amount of deflection under load of a thin gear web or the shank of a pinion cannot be changed by heat treatment or by use of a stronger material. Hooke's law is the same: A change of deflection can be accomplished only by a change of design.

Bending strength of a gear tooth is the amount of load per unit area acceptable at the root radius to the point of permanent deformation. Permanent deformation of a carburized tooth is usually accompanied by a crack at the root radius, whereas with a noncarburized tooth, actual bending may occur. The root radius is mentioned as the point of deformation because it is the area of greatest stress concentration in tension. Also, stress (load per unit area) calculations assume that the load is applied at the pitchline or the mid-height of the tooth. Actually, the realistic stress at the root radius varies from approximately one-half, when the load is applied low on the active profile, to double, when the load is applied near the tooth tip. Bending strength of the root radius is a function of the surface hardness and the physical condition of the surface, such as smoothness, sharpness of radius, and/or corrosive pitting.

The strength of the core material—i.e., the basic material under the carburized steel case—is generally to be considered as compressive strength rather than tensile strength. It measures the ability to withstand surface pressures that may crush through the case and/or brinell (indent) the surface.

Torsional strength of a pinion shank or of a shaft is a bit more complex. The maximum tensile stress is at the surface in a direction 45° from the central axis or longitudinal direction. The maximum shear stress, also at the surface, is longitudinal (parallel to the central axis) and transverse (90° across the central axis) (see Fig. 22). The strength at the surface is a function of surface hardness; therefore, surface-originated torsional tensile failures of carburized parts are rare unless a specific type of stress raiser is present at the surface. This is not so with through hardened or non-heat treated parts since the strength is uniform throughout the part. Under this condition, torsional tensile failure is expected to originate at the surface. In most instances of torsional failure of carburized or induction hardened shafts and pinion shanks, the initial fracture is along the

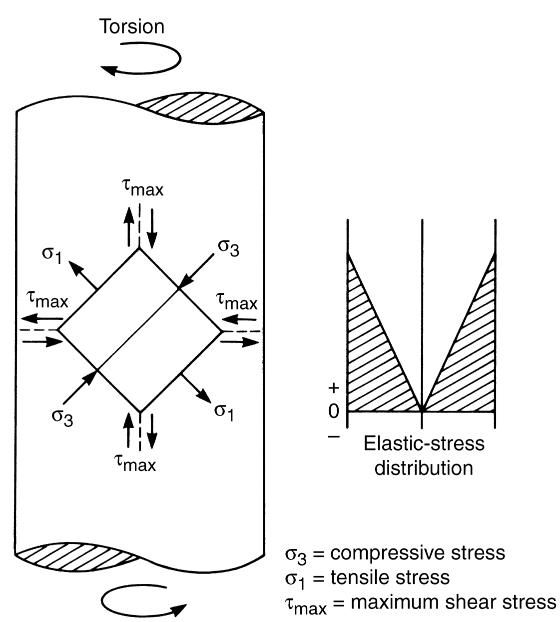

Fig. 22 Free-body diagram of maximum tensile and shear stress orientation on a surface element of a shaft in a torsional mode. Both maximums are at the surface. Stress is considered to be zero at the central axis. Source: Ref 2

shear plane (i.e., longitudinal or transverse), and not at the 45° angle. This means that the shear strength of the subsurface material is the controlling factor. Shear strength is considered to be only about 60% of the tensile strength. The area most vulnerable to the origin of torsional shear failure of a shaft is the transition zone between the case and the core of either a carburized or an induction hardened part. The maximum applied stress often exceeds the shear strength of the material at this area and initiates a start of subsurface failure.

Gear Materials

A variety of cast irons, powder-metallurgy materials, nonferrous alloys, and plastics are used in gears, but steels, because of their high strength-to-weight ratio and relatively low cost, are the most widely used gear materials for heavy duty, power transmission applications. Consequently, steel gears receive primary consideration in this chapter.

Among the through-hardening steels in wide use are 1040, 1060, 4140, and 4340. These steels can also be effectively case hardened by induction heating. Among the carburizing steels used in gears are 1018, 1524, 4026, 4118, 4320, 4620, 4820, 8620, and 9310 (AMS 6260). Table 3 lists specific application areas for commonly used gear steels. Many high-performance gears are carburized. Some special-purpose steel gears are case hardened by either carbonitriding or nitriding. Other special-purpose gears, such as those used in chemical or food-processing equipment, are made of stainless steels or nickel-base alloys because of their corrosion resistance, their ability to satisfy sanitary standards, or both. Gears intended for operation at elevated temperatures may be made of tool steels or elevated-temperature alloys.

Most gears are made of carbon and low-alloy steels, including carburizing steels and the limited number of low-alloy steels that respond favorably to nitriding. In general, the steels selected for gear applications must satisfy two basic sets of requirements that are not always compatible—those involving fabrication and processing and those involving service. Fabrication and processing requirements include machinability, forgeability, and response to heat treatment as it affects fabrication and processing. Service requirements are related to the ability of the gear to perform satisfactorily under the conditions of loading for which it was designed and thus encompass all mechanical-property requirements, including fatigue strength and response to heat treatment (see the section "Selection Guidelines" presented below).

Table 3 Recommended steels for various applications and gear types

Typical industrial application	Gear design type	Typical material choice
Differentials		
Automotive	Hypoid, spiral/straight bevel	4118, 4140, 4027, 4028, 4620, 8620, 8622, 8626
Heavy truck	Hypoid, spiral/straight bevel	4817, 4820, 8625, 8822
Drives		
Industrial	Helical, spur rack and pinion, worm	1045, 1050, 4140, 4142, 4150, 4320, 4340, 4620
Tractor-accessory	Crossed-axis helical, helical	1045, 1144, 4118, 4140
Engines		
Heavy truck	Crossed-axis helical, spur, worm	1020, 1117, 4140, 4145, 5140, 8620
Equipment		
Earth moving	Spiral/straight bevel, zerol	1045, 4140, 4150, 4340, 4620, 4820, 8620, 9310
Farming	Face, internal, spiral/straight bevel, spur	4118, 4320, 4817, 4820, 8620, 8822
Mining, paper/steel mill	Helical, herringbone, miter, spur, spur rack and pinion	1020, 1045, 4140, 4150, 4320, 4340, 4620, 9310
Starters		
Automotive	Spur	1045, 1050
Transmissions		
Automotive	Helical, spur	4027, 4028, 4118, 8620
Heavy Truck	Helical, spur	4027, 4028, 4620, 4817, 5120, 8620, 8622, 9310
Marine	Helical, helical conical, spiral bevel	8620, 8622
Off highway	Helical, internal, spiral/straight bevel, spur	1118, 5130, 5140, 5150, 8620, 8822, 9310
Tractor	Herringbone, internal, spur	4118, 4140, 8822

Source: Ref 3

Because resistance to fatigue failure is partly dependent upon the cleanness of the steel and upon the nature of allowable inclusions, melting practice may also be a factor in steel selection and may warrant selection of a steel produced by vacuum melting or electroslag refining. The mill form from which a steel gear is machined is another factor that may affect its performance. Many heavy-duty steel gears are machined from forged blanks that have been processed to provide favorable grain flow consistent with load pattern rather than being machined from blanks cut from mill-rolled bar.

Selection Guidelines. In all gears the choice of material must be made only after careful consideration of the performance demanded by the end-use application and total manufactured cost, taking into consideration such issues as machining economics. Key design considerations require an analysis of the type of applied load, whether gradual or instantaneous, and the desired mechanical properties, such as bending fatigue strength or wear resistance, all of which define core strength and heat treating requirements.

Different areas in the gear tooth profile see different service demands. Consideration must be given to the forces that will act on the gear teeth, with tooth bending and contact stress, resistance to scoring and wear, and fatigue issues being paramount. For example, in the root area, good surface hardness and high residual compressive stress are desired to improve endurance, or bending fatigue life. At the pitch diameter, a combination of high hardness and adequate surface strength are necessary to handle contact stress and wear and to prevent spalling.

Numerous factors influence fatigue strength, including:

- Hardness distribution, as a function of case hardness, case depth, core hardness
- Microstructure, as a function of retained austenite percentage, grain size, carbides (size, type, distribution), nonmartensitic phases
- Defect control, as a function of residual compressive stress, surface finish, geometry, intergranular toughness

More detailed information on the factors that influence the properties of gear steels can be found in Chapter 3, "Ferrous and Nonferrous Alloys."

Gear Manufacturing Methods

Gears can be made by a variety of manufacturing processes. This section will briefly review these processes. For the manufacture of metal gears, the reader should consult Chapters 5, "Machining, Grinding, and Finishing," 6, "Casting, Forming and Forging," and 7, "Powder Metallurgy." Methods for making plastic gears are described in Chapter 4, "Plastics."

Metal Removal Processes

As shown in Fig. 23, gear blanks can be shaped by a number of cutting (machining) and finishing processes. Often blanks are processed

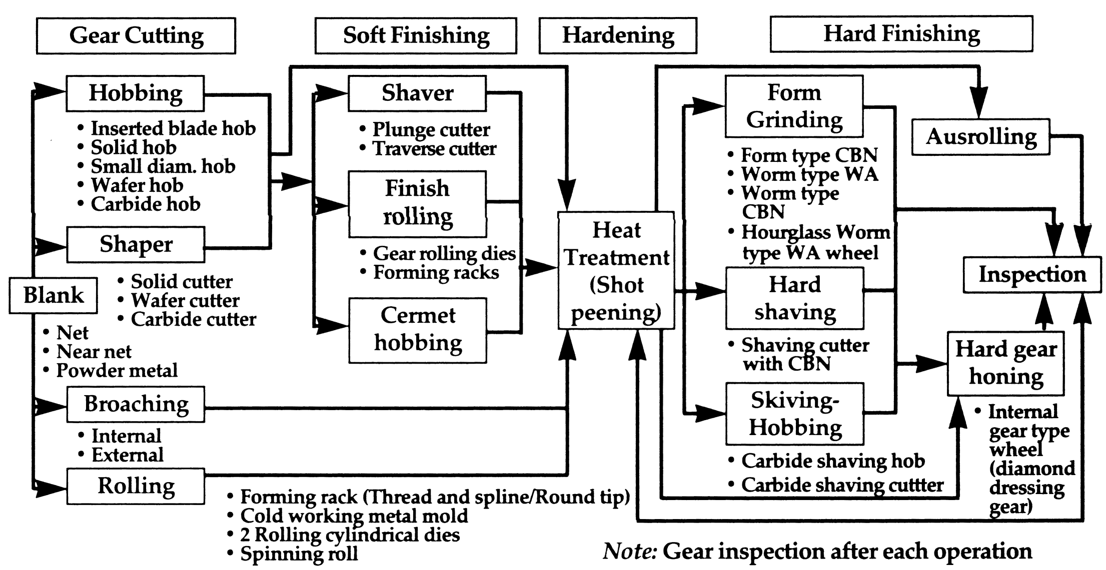

Fig. 23 Examples of various gear manufacturing processes. Source: Ref 3

by a series of rough cutting and finishing operations. Gears made by machining/finishing have the highest AGMA quality tolerance levels than gears made by competing processes (see Table 1 in Chapter 6). Machining and finishing processes make up about 60% of gear manufacturing costs (Fig. 24).

Broaching is a machining operation which rapidly forms a desired contour in a workpiece by moving a cutter, called a broach, entirely past the workpiece. The broach has a long series of cutting teeth that gradually increase in height. The broach can be made in many different shapes to produce a variety of contours. The last few teeth of the broach are designed to finish the cut rather than to remove considerably more metal. Broaches are often used to cut internal gear teeth, racks, and gear segments on small gears, and usually are designed to cut all teeth at the same time.

Grinding is a process that shapes the surface by passes with a rotating abrasive wheel. Grinding is not a practical way to remove large amounts of metal, so it is used to make very fine-pitch teeth, or to remove heat treat distortion from large gears that have been cut and then fully hardened. Many different kinds of grinding operations are used in gear manufacture.

Hobbing. This is a gear cutting method that uses a tool resembling a worm gear in appearance, having helically spaced cutting teeth. In a single-pitch hob, the rows of teeth advance exactly one pitch as the hob makes one revolution. With only one hob, it is possible to cut interchangeable gears of a given pitch of any number of teeth within the range of the hobbing machine.

Honing is a low-speed finishing process used chiefly to produce uniform high dimensional accuracy and fine finish. In honing, very thin layers of stock are removed by simultaneously rotating and reciprocating a bonded abrasive stone or stick that is pressed against the surface being honed with lighter force than is typical of grinding.

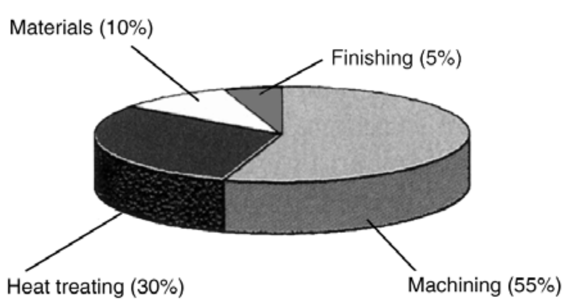

Fig. 24 Typical gear manufacturing costs. Source: Ref 3

Lapping is a polishing operation that uses abrasive pastes to finish the surfaces of gear teeth. Generally a toothed, cast iron lap is rolled with the gear being finished.

Milling is a machining operation which removes the metal between two gear teeth by passing a rotating cutting wheel across the gear blank.

Shaping is a gear cutting method in which the cutting tool is shaped like a pinion. The shaper cuts while traversing across the face width and rolling with the gear blank at the same time.

Shaving is a finishing operation that uses a serrated gear-shaped or rack-shaped cutter to shave off small amounts of metal as the gear and cutter are meshed at an angle to one another. The crossed axes create a sliding motion which enables the shaving cutter to cut.

Skiving is a machining operation in which the cut is made with a form tool with its face so angled that the cutting edge progresses from one end of the workpiece to the other as the tool feeds tangentially past the rotating workpiece.

Casting, Forming, and Forging Processes

Casting is a process of pouring or injecting molten metal into a mold so that the metal solidifies and hardens into the desired shape. Casting is often used to make gear blanks that will have cut teeth. Small gears are frequently cast complete with teeth by the die casting process, which uses a precision mold of tool steel and low-melting-point alloys for the gears.

Stamping is a fast inexpensive method of producing small gears from thin sheets of metal. The metal is sheared by a punching die which stamps through the sheet stock into a mating hole.

Gear rolling is a process which rapidly shapes fine gear teeth or worm threads by high-pressure rolling with a toothed die.

Powder Metallurgy (P/M) Processing. In its most basic and widely used form, the P/M process consists of pressing a powder to the desired shape, followed by heating (sintering) at an elevated temperature below its melting point. There are a number of variations of the P/M process that are applicable to gears (see Fig. 1 in Chapter 7). The P/M process is suitable for high-volume production of small gears. It is not economical for low-to-medium volume production.

Injection molding is a method of forming a plastic to the desired shape by forcing the heat-softened plastic into a relatively cool cavity (die) under pressure. It is widely used for high-volume production of thermoplastic resin (e.g., acetals and nylons) gears. Often lubricants are added to the thermoplastic material to further improve the inherent lubricity of the material.

Forging. The forging process has long been used to create blanks that will be subsequently shaped into gears by metal removal methods. However, it is increasingly being used for the production of near-net shape and net shape gears for demanding applications where great strength and durability are required. The forging process is carried out hot with metal preheated to a desired temperature under intense pressure until it fills the die cavity. The resultant grain flow which smoothly follows gear tooth contours makes forged gears stronger than those made by other processes.

Alternative or Nontraditional Gear Manufacturing

Although the gear manufacturing processes discussed above are by far the most prevalent methods for gear production, there are a number of other processes that are being used increasingly by the gear industry.

Laser Machining. While sometimes slower than traditional machining techniques, depending on the material, lasers can cut complex shapes such as gears with great precision and very little material waste. This conservation comes from the ability of the computerized-numerically-controlled (CNC) machines controlling the lasers to reuse cutting paths, getting as many gears from a single sheet as possible. Also, the computer control means that laser machining is also low maintenance. The setup and first runs are closely supervised, but the actual production runs don't need any real supervision due to the CNC programming.

Limitations of the laser machining process are as follows:

- Pieces cut with a laser have heat affected zones, areas where the metal is heated beyond a critical transformation point, and recast. These zones are limited, however, to the edges of the cuts—minimizing, but not eliminating heat distortion and the need for further machining. Post production grinding and honing are common.
- Lasers are limited to cutting metals ≤19 mm (≤0.75 in.). Cutting thicker materials requires too much power.
- Lasers are limited to nonreflective or semi-reflective metals. Metals like aluminum and brass that are highly reflective are difficult to cut.
- Lasers are limited, like stamping, to flat forms such as spur gears.

Electrical Discharge Machining (EDM). The EDM process uses electricity to melt or vaporize the material being cut. Many of the attributes and limitations outlined for laser machining are also applicable to EDM.

Abrasive water jet machining is a hydrodynamic machining process that uses a high-velocity stream of water laden with fine abrasive particles as a cutting tool. The process typically produces burr-free edges with heat-affected zones, can easily handle heat treated material, and, unlike lasers, can cut through stacks of material to create multiple parts at the same time, saving time and money. Abrasive waterjets have been used to:

- Cut lapping machine gears made from difficult-to-machine plastic composites
- Titanium rack and pinion components for commercial jet pilot seats
- Process phenolics into machinery gear components
- Cut spring steel into gears with tightly spaced teeth

Inspection

As noted at the bottom of Fig. 23, inspection is an integral part of the gear manufacturing process. As with all manufactured products, gears must be checked to determine whether the resulting product meets design specifications and requirements. Because of the irregular shape of gears and the number of factors that must be measured, such inspection is somewhat difficult. Among the factors to be checked are the linear tooth dimensions (thickness, spacing, depth, and so on), tooth profile, surface roughness, and noise. Several special devices, most of them automatic or semiautomatic, are used for this inspection.

Gear tooth vernier calipers can be used to measure the thickness of gear teeth on the pitch circle. However, inspection is usually done by special machines, which in one or a series of operations check several factors, including eccentricity, variations in circular pitch, variations in pressure angle, fillet interference, and lack of continuous action. The gear is usually mounted and moved in contact with a master gear. The movement of the latter is amplified and recorded on moving charts, as shown in Fig. 25.

Noise level is important in many applications, not only from the standpoint of noise pollution but also as an indicator of probable gear life. Therefore, special equipment for its measurement is quite widely used, sometimes integrated into mass-production assembly lines.

Dimensional variations in gears result in noise, vibration, operational problems, reduced carrying ability, and reduced life. These problems are compounded at higher gear-operating speeds. AGMA has incorporated the critical dimensions of pitch, concentricity, tooth profile, tooth thickness, and tooth surface finish into a number of standards which the interested reader should refer to. Additional information can also be found in Ref 4 and 5.

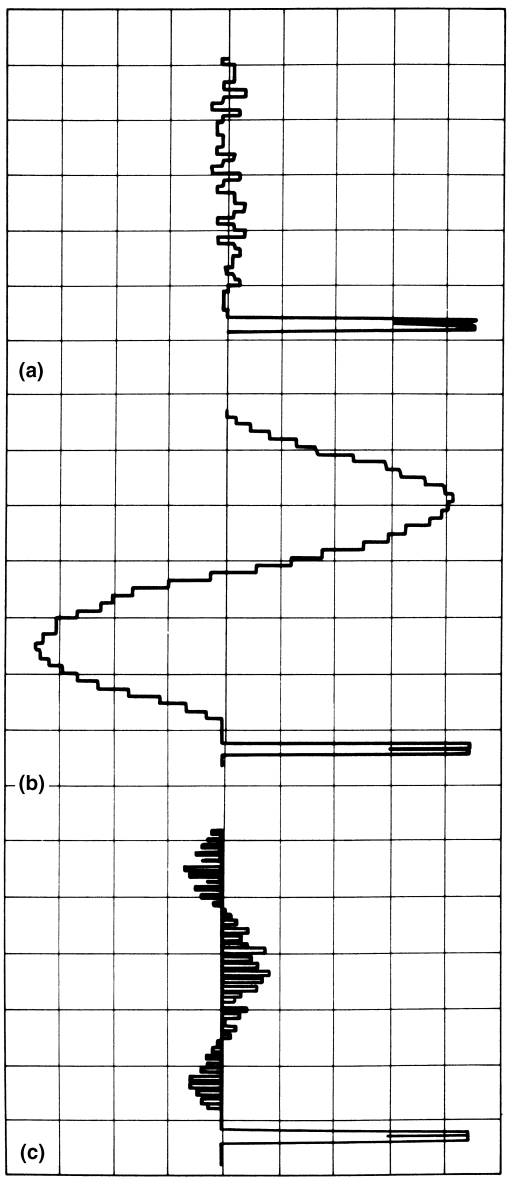

Fig. 25 Typical data obtained on charts generated by automated gear-checking machines. (a) Tooth-to-tooth pitch error. (b) Accumulated pitch error. (c) Spacing error

Heat Treating (Ref 3)

Heat treating is one of the most important steps in the manufacture of precision gearing. Its contribution is vitally important for cost control, durability, and reliability. As shown in Fig. 24, heat treating represents about 30% of a typical gear manufacturing cost. If not properly understood and controlled, it can have a significant impact on all aspects of the gear manufacturing process. This section will briefly review the following heat treating processing steps: prehardening processes, through hardening and case hardening processes, applied energy hardening, and post-hardening processes. More detailed information on these processes can be found in the following chapters:

- Chapter 8, "Through Hardening"
- Chapter 9, "Carburizing"
- Chapter 10, "Nitriding"
- Chapter 11, "Carbonitriding"
- Chapter 12, "Induction and Flame Hardening"

Prehardening Processes

Several heat treatments are normally performed during the gear manufacturing process to prepare the part for the intended manufacturing steps. These are essential to the manufacture of a quality gear.

Annealing consists of heating to and holding at a suitable temperature followed by cooling at an appropriate rate, primarily intended to soften the part and improve its machinability. Supercritical or full annealing involves heating a part above the upper critical temperature (Ac_3), that is the temperature at which austenite begins to transform to ferrite during cooling, and then slowly cooling in the furnace to around 315 °C (600 °F). Intercritical annealing involves heating the part to a temperature above the final transformation temperature (Ac_1), the temperature at which austenite begins to form during heating. Subcritical annealing heats the part to just below the Ac_1 point followed by a slow cool in the furnace. The rate of softening increases rapidly as the annealing temperature approaches the Ac_1 point.

Normalizing involves heating the part above the upper critical temperature and then air cooling outside the furnace to relieve residual stresses in a gear blank and for dimensional stability. Normalizing is often considered from both a thermal and microstructural standpoint. In the thermal sense, normalizing is austenitizing followed by cooling in still or slightly agitated air or nitrogen. In a microstructural sense, normalizing produces a more homogenous structure. A normalized part is very machinable but harder than an annealed part. Normalizing also plays a significant role in the control of dimensional variation during carburizing.

Stress relieving involves heating to a temperature below the lower transformation temperature, as in tempering, holding long enough to reduce residual stress and cooling slowly enough, usually in air, to minimize the development of new residual stresses. Stress relief heat treating is used to relieve internal stresses locked in the gear as a consequence of a manufacturing step.

Through Hardening and Case Hardening

Various heat treatment processes are designed to increase gear hardness. These usually involve heating and cooling and are typically classified as through hardening, case hardening, and hardening by applied energy which will be discussed separately in a subsequent section.

Through or direct hardening refers to heat treatment methods, which do not produce a case. Examples of commonly through hardened gear steels are AISI 1045, 4130, 4140, 4145, 4340, and 8640. It is important to note that hardness uniformity should not be assumed throughout the gear tooth. Since the outside of a gear is cooled faster than the inside, there will be a hardness gradient developed. The final hardness is dependent on the amount of carbon in the steel; the depth of hardness depends on the hardenability of the steel as well as the quench severity.

Through hardening can be performed either before or after the gear teeth are cut. When gear teeth will be cut after the part has been hardened, surface hardness and machinability become important factors especially in light of the fact that machining will remove some or most of the higher hardness material at the surface. The hardness is achieved by heating the material into the austenitic range, typically 815 to 875 °C (1500 to 1600 °F), followed by quenching and tempering.

Case hardening produces a hard, wear resistant case or surface layer on top of a ductile, shock resistant interior, or core. The idea behind case hardening is to keep the core of the gear tooth at a level around 30 to 40 HRC to avoid tooth breakage while hardening the outer surface to increase pitting resistance. The higher the surface hardness value, the greater the pitting resistance. Bending strength increases for surface hardness up to about 50 HRC, after which the increase in bending strength is offset by an increase in notch sensitivity.

Carburizing is the most common of the case hardening methods. A properly carburized gear will be able to handle between 30% to 50% more load than a through hardened gear. Carburizing steels are typically alloy steels with approximately 0.10% to 0.20% carbon. Examples of commonly carburized gear steels include AISI 1018, 4320, 5120, 8620, and 9310 as well as international grades such as 20MnCr5, 16MnCr5, ZF-7B, 20MoCr4, and V2525.

Carburizing can be performed in the temperature range of 800 to 1090 °C (1475 to 2000 °F). Common industry practice today finds the majority of carburizing operations taking place at 870 to 1010 °C (1600 to 1850 °F). Carburizing case depths can vary over a broad range, 0.13 to 8.25 mm (0.005 to 0.325 in.) being typical. However, it is common to use the carbonitriding process for case depths below 0.4 mm (0.015 in.).

Carbonitriding is a modification of the carburizing process, not a form of nitriding. This modification consists of introducing ammonia into the carburizing atmosphere to add nitrogen to the carburized case as it is being produced. Examples of gear steels that are commonly carbonitrided include AISI 1018, 1117, and 12L14.

Typically, carbonitriding is done at a lower temperature than carburizing, or between 700 to 900 °C (1300 to 1650 °F), and for a shorter time. Because nitrogen inhibits the diffusion of carbon, what generally results is a shallower case than is typical for carburized parts. A carbonitrided case is usually between 0.075 to 0.75 mm (0.003 to 0.030 in.) deep.

Nitriding is another surface treatment process that increases surface hardness. Since rapid quenching is not required, dimensional changes are kept to a minimum, which is a major advantage. It is not suitable for all gear materials. One of its limitations is the extremely high surface hardness or "white layer" produced, which has a more brittle nature than the surface produced by carburizing. Despite this, nitriding has proved to be a viable alternative for numerous applications. Commonly nitrided gear steels include AISI 4140, 4150, 4340, 7140, 8640, and AMS 6475 (Nitralloy N).

Nitriding is typically done in the 495 to 565 °C (925 to 1050 °F) range. Three factors that are extremely critical in producing superior and consistent nitrided cases and predictable dimensional change are steel composition, prior structure, and core hardness. Case depth and case hardness properties vary not only with the duration and type of nitriding being performed but are also influenced by these factors. Typically case depths are between 0.20 to 0.65 mm (0.008 to 0.025 in.) and take from 10 to 80 hours to produce.

ACKNOWLEDGMENT

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REFERENCES

1. *Gear Nomenclature, Definitions of Terms with Symbols,* ANSI/AGMA 1012-F90, American Gear Manufacturerers Association, 1990
2. L.E. Alban, *Systematic Analysis of Gear Failures,* American Society for Metals, 1985
3. F.J. Otto and D.H. Herring, Gear Heat Treatment: Part I, *Heat Treating Progress,* June 2002, p 55–59
4. D.W. Dudley, *Handbook of Practical Gear Design,* McGraw Hill Book Company, 1984
5. D.P. Townsend, Ed., *Dudley's Gear Handbook: The Design, Manufacture, and Application of Gears,* 2nd ed., McGraw Hill, Inc., 1991

Gear Materials, Properties, and Manufacture
J.R. Davis, editor, p19-38
DOI:10.1361/gmpm2005p019

CHAPTER 2

Gear Tribology and Lubrication

GEAR PERFORMANCE is profoundly influenced by tribology, which can be defined as the science and technology of interacting surfaces in relative motion, or, alternatively, the science and technology of friction, lubrication, and wear. This chapter reviews current knowledge of the field of gear tribology and is intended for both gear designers and gear operators. Gear tooth failure modes are discussed with emphasis on lubrication-related failures. Equations for calculating lubricant film thickness which determines whether the gears operate in the boundary, elastohydrodynamic, or full-film lubrication range are given. Also given is an equation for Blok's flash temperature, which is used for predicting the risk of scuffing. In addition, recommendations for lubricant selection, viscosity, and method of application are discussed. Finally, a case history demonstrates how the tribological principles discussed in the chapter can be applied in a practical manner to avoid gear failure. Table 1 lists the nomenclature used in this chapter and the units required to calculate Eq 1 to 24.

Gear Tribology

Because gears are such common machine components, they may be taken for granted. It is not generally appreciated that they are complex systems requiring knowledge from all the engineering disciplines for their successful design. Gear design is a process of synthesis in which gear geometry, materials, heat treatment, manufacturing methods, and lubrication are selected to meet the requirements of a given application. The designer must design the gearset with adequate strength, wear resistance, and scuffing resistance. To do this, he or she must consider gear tribology. The choice of lubricant and its application method is as important as the choice of steel alloy and heat treatment. The interrelationship of the following factors must be considered:

- Gear tooth geometry
- Gear tooth motion (kinematics)
- Gear tooth forces (static and dynamic)
- Gear tooth material and surface characteristics (physical and chemical)
- Lubricant characteristics (physical and chemical)
- Environmental characteristics (physical and chemical)

Lubrication Regimes

Throughout this chapter a number of different types of *lubrication regimes* are discussed. Lubrication regimes are the ranges of operating conditions for lubricated tribological systems that can be distinguished by their frictional characteristics and/or by the manner and amount of separation of the contacting surfaces (Fig. 1). A brief review of the various types of lubrication regimes is discussed below. More detailed information on this subject can be found in the article "Lubrication Regimes" in *Friction, Lubrication, and Wear Technology,* Vol 18, of the *ASM Handbook* (see pages 89 to 97).

Terminology. When two surfaces are in contact with each other, the load is carried by many high points, or asperities, on the surfaces. During sliding, the total tangential force required to shear these asperity junctions is usually high, causing unacceptable friction, wear, and surface damage. To reduce the frictional force and thus allow easier sliding, a lubricant is deliberately introduced to separate the asperities either totally or partially (Fig. 1).

The use of liquid or gas lubricants is known as *fluid-film lubrication. Full-film lubrication,* also known as thick-film lubrication, refers to the total separation of asperities by a lubricant film thickness many times larger than the size of the lubricant molecules. If this condition exists only partially—that is, if part of the load is carried by the fluid pressure and the rest is borne by contacting asperities separated by a molecularly thin lubricant film—the term *thin-film lubrication* is used. In the most severe form of thin-film lubrication, the entire load is carried by asperities lubricated by surface films of molecularly thin liquids, gases, or solids; this condition is known as *boundary lubrication.* The exclusive use of solid lubricants is called *solid lubrication.*

For highly loaded contacts, lubricant pressure can cause elastic deformation of the surfaces that is on the same order as the lubricant film thickness. When this occurs, the influence of deformation on lubrication performance becomes a significant parameter. Contacts operating under this condition are in the regime of *elastohydrodynamic lubrication.*

Lubricated contacts such as gear teeth and rolling bearings have surface deformation com-

Table 1 Nomenclature and units of measure used in this chapter to discuss friction, lubrication, and wear of gears

Symbol	Description	Units	Symbol	Description	Units
B_M	Thermal contact coefficient	lbf/(in. · s$^{0.5}$ · °F)	W_{Nr}	Normal operating load	lbf
b_H	Semiwidth of Hertzian contact band	in.	w_{Nr}	Normal unit load	lbf/in.
c	Constant (see Table 5)	hp/(gal/min)	X_w	Welding factor	. . .
c_M	Specific heat per unit mass	lbf · in./[lb · °F]	X_Γ	Load sharing factor	. . .
d	Operating pitch diameter of pinion	in.	α	Pressure-viscosity coefficient	in.^{2}/lbf
E_1	Modulus of elasticity of pinion	lbf/in.^{2}	λ	Specific film thickness	. . .
E_2	Modulus of elasticity of gear	lbf/in.^{2}	λ_{min}	Minimum specific film thickness	. . .
E_r	Reduced modulus of elasticity	lbf/in.^{2}	λ_M	Heat conductivity	lbf/[s · °F]
h_{min}	Minimum film thickness	in.	μ_m	Mean coefficient of friction	. . .
L_{min}	Minimum contact length	in.	μ_0	Absolute viscosity	reyns (lbf · s/in.^{2})
n	Pinion speed	rev/min	ν_1	Poisson's ratio of pinion	. . .
P	Transmitted power	hp	ν_2	Poisson's ratio of gear	. . .
q	Oil flow rate	gal/min	ν_{40}	Kinematic viscosity at 40 °C (105 °F)	cSt
S	Average root-mean-square (rms) surface roughness	μin.	ρ_1	Transverse radius of curvature of pinion	in.
			ρ_2	Transverse radius of curvature of gear	in.
T_b	Bulk temperature	°F	ρ_M	Density	lb/in.^{3}
$T_{b_{test}}$	Bulk temperature of test gears	°F	ρ_n	Normal relative radius of curvature	in.
T_c	Contact temperature	°F	σ	Root-mean-square composite surface roughness	μin.
T_f	Flash temperature	°F			
$T_{f_{test}}$	Maximum flash temperature of test gears	°F	σ_1	Root-mean-square surface roughness of pinion	μin.
T_s	Scuffing temperature	°F	σ_2	Root-mean-square surface roughness of gear	μin.
V	Operating pitch line velocity	ft/min			
V_e	Entraining velocity	in./s	ψ_b	Base helix angle	degrees
V_{r1}	Rolling velocity of pinion	in./s	ω_1	Angular velocity of pinion	rad/s
V_{r2}	Rolling velocity of gear	in./s	ω_2	Angular velocity of gear rad/s	rad/s

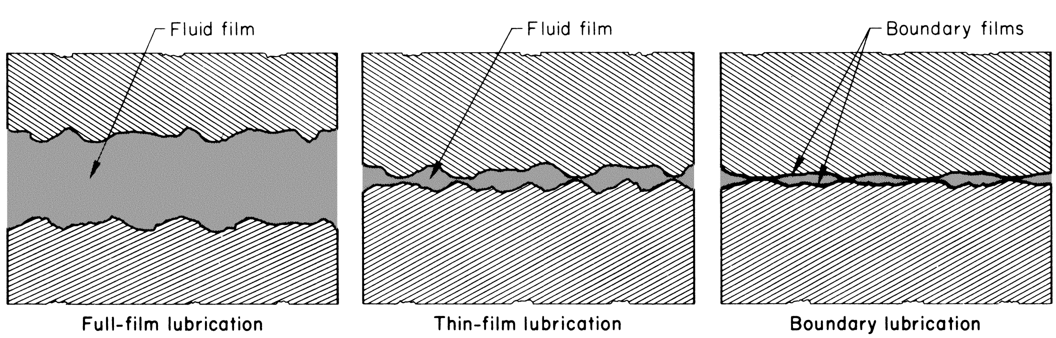

Fig. 1 Schematic showing three lubrication regimes

parable to or exceeding the lubricant film thickness. Therefore, elastohydrodynamic (EHD) lubrication is extremely important in determining friction and wear in many mechanical components and will be emphasized throughout this chapter (see, for example, the section "Elastohydrodynamic Lubrication").

Classification of Gear Tooth Failure Modes

To obtain optimum minimum-weight gearsets, the gear designer must be aware of the intricate details of many competing modes of failure. Erichello (Ref 1) has classified gear failures into two broad groups:

- Nonlubrication-related failure
- Lubrication-related failure

As shown in Table 2, nonlubrication-related failures include both overload and bending fatigue types of failure. Lubrication-related failures, the focal point in this chapter, include Hertzian fatigue, wear, and scuffing. Failure modes are further discussed in Chapter 13, "Gear Failure Modes and Analysis," and in Ref 2 through 6.

Many gear failures are known by several names and qualifying terms, such as initial, moderate, destructive, and so on. The term "scoring" has been used in the past in the United States, while the term "scuffing" is used in Europe to describe the severe form of adhesive wear that involves the welding and tearing of the surfaces of gear teeth. To agree with current usage, the term scuffing will be used in this chapter when referring to this failure mode. The term scoring implies scratching, and it will be used to describe abrasive wear rather than scuffing.

Lubrication-Related Failure

This chapter is concerned with gear tooth failures that are influenced by friction, lubrication, and wear. Pitting or scuffing may cause the gear teeth to deteriorate and generate dynamic forces, which in turn cause the gear teeth to fail by bending fatigue. In these cases, the bending failure is secondary and not directly related to lubrication, whereas pitting or scuffing are the primary failure modes, and both are definitely influenced by lubrication. The failure analyst must discern the difference between primary and secondary failure modes because the wrong corrective action is likely to be recommended if a secondary failure mode is mistaken for the primary failure mode. For example, increasing the size of the gear teeth to prevent reoccurrence of the above-mentioned bending failure would only make the situation worse by lowering the pitting and scuffing resistance. Godfrey (Ref 7) gives a good description of lubrication-related failure modes.

Table 2 Basic failure modes of gear teeth

Nonlubrication-related failures		Lubrication-related failures		
Overload	**Bending fatigue**	**Hertzian fatigue**	**Wear**	**Scuffing**
Brittle fracture	Low-cycle fatigue (≤1000 cycles to failure)	Pitting	Adhesion	Scoring
Ductile fracture	High-cycle fatigue (>1000 cycles to failure)	Initial	Normal	Galling
Plastic deformation		Superficial	Running-in	Seizing
Cold flow		Destructive	Mild	Welding
Hot flow		Spalling	Moderate	Smearing
Indentation		Micropitting	Severe	Initial
Rolling		Frosting	Excessive	Moderate
Bruising		Gray staining	Abrasion	Destructive
Peening		Peeling	Scoring	
Brinelling		Subcase fatigue (case crushing)	Scratching	
Rippling (fish scaling)		Plowing	Cutting	
Ridging			Plowing	
Bending (yielding)			Gouging	
Tip-to-root interference			Corrosion	
			Fretting corrosion	
			Electrical discharge damage	
			Polishing (burnishing)	

On the basis of the above considerations, overload and bending fatigue are judged to be unrelated to lubrication and are eliminated from further discussion together with another category, subcase Hertzian fatigue. Although corrosion, fretting-corrosion, cavitation, and electrical discharge damage are influenced by lubrication, they will not be discussed in this chapter because these failure modes occur relatively rarely in gear teeth. Hence, only the following failure modes are discussed in this chapter:

- Hertzian fatigue (including pitting and micropitting)
- Wear (including adhesion, abrasion, and polishing)
- Scuffing

Hertzian Fatigue

Pitting and micropitting are two lubrication-related gear tooth failure modes caused by Hertzian fatigue.

Pitting is a common failure mode for gear teeth because they are subjected to high Hertzian contact stresses and many stress cycles. For example, through-hardened gears are typically designed to withstand contact stresses of approximately 700 MPa (100 ksi), while the contact stresses on carburized gears may reach 2100 MPa (300 ksi). In addition, a given tooth on a pinion that is revolving at 3600 rev/min accumulates over 5×10^6 stress cycles every 24 h.

Pitting is a fatigue phenomenon (Ref 8) that occurs when a fatigue crack initiates either at the surface of the gear tooth or at a small depth below the surface. The crack usually propagates for a short distance in a direction roughly parallel to the tooth surface before turning or branching to the surface. When the cracks have grown to the extent that they separate a piece of the surface material, a pit is formed. If several pits grow together, the resulting larger pit is often referred to as a "spall." There is no endurance limit for Hertzian fatigue, and pitting occurs even at low stresses if the gears are operated long enough. Because there is no endurance limit, gear teeth must be designed for a suitable finite lifetime.

To extend the pitting life of a gearset, the designer must keep the contact stress low and the material strength and lubricant specific film thickness high. There are several geometric variables, such as diameter, face width, number of teeth, pressure angle, and so on, that can be optimized to lower the contact stress. Steels and heat treatment are selected to obtain hard tooth surfaces with high strength. Maximum pitting resistance is obtained with carburized gear teeth because they have hard surfaces, and carburizing induces beneficial compressive residual stresses that effectively lower the load stresses. The drawbacks to using carburized gear teeth are that they are relatively expensive to produce and that they must be finished by grinding. The details for obtaining high lubricant specific film thickness will be explained later when EHD lubrication is discussed, but general recommendations are to use an adequate supply of cool, clean, and dry lubricant that has adequate viscosity and a high pressure-viscosity coefficient.

Pitting may initiate at the surface or at a subsurface defect, such as a nonmetallic inclusion. With gear teeth, pits are most often of the surface-initiated type because the lubricant film thickness is usually low, resulting in relatively high metal-to-metal contact. The interaction between asperities or contacts at defects, such as nicks or furrows, creates surface-initiated, rather than subsurface-initiated cracks. For high-speed gears with smooth surface finishes, the film thickness is greater and subsurface-initiated pitting, rather than surface-initiated, may predominate. In these cases, pitting usually starts at a subsurface inclusion, which acts as a point of stress concentration. Cleaner steels, such as those produced by vacuum melting, prolong the pitting life by reducing the number of inclusions.

Contamination from water in the lubricant is believed to promote pitting through hydrogen embrittlement of the metal, and abrasive particles in the lubricant cause pitting by indenting the tooth surfaces, causing stress concentrations, and disrupting the lubricant film. At present, the influence of lubricant additives on pitting is unresolved.

Prevention of Pitting. The following recommendations serve as guidelines for preventing the onset of pitting in gearsets:

- Reduce contact stresses by reducing loads or optimizing gear geometry
- Use clean steel, properly heat treated to high hardness, preferably by carburizing
- Use smooth tooth surfaces produced by careful grinding or honing
- Use an adequate amount of cool, clean, and dry lubricant of adequate viscosity

Micropitting. On relatively soft gear tooth surfaces; such as those of through-hardened gears, Hertzian fatigue forms large pits with dimensions on the order of millimeters. With surface-hardened gears (for example, carburized, nitrided, induction hardened, and flame hardened), pitting may occur on a much smaller scale, typically only 10 μm (400 μin.) deep. To the naked eye, the areas where micropitting has occurred appear frosted, and "frosting" is a popular term for micropitting. Japanese researchers (Ref 9) have referred to the failure mode as "gray staining" because the light-scattering properties of micropitting give the gear teeth a gray appearance. With scanning electron microscopy (SEM), it is immediately evident that micropitting proceeds by the same fatigue process as classical pitting, except the pits are extremely small.

In many cases, micropitting is not destructive to the gear tooth surface. It sometimes occurs only in patches and may stop after the tribological conditions have been improved by running-in. The micropits may actually be removed by mild polishing wear during running-in, in which case the micropitting is said to "heal." However, there have been examples (Ref 9–11) where micropitting has escalated into full-scale pitting, leading to the destruction of the gear teeth.

The specific film thickness is the most important parameter that influences micropitting. Damage seems to occur most readily on gear teeth with rough surfaces, especially when they are lubricated with low-viscosity lubricants. Gears finished with special grinding wheels to a mirrorlike finish (Ref 12) have effectively eliminated micropitting. Slow-speed gears are prone to micropitting because their film thickness is low.

Prevention of Micropitting. To prevent micropitting, the specific film thickness should be maximized by using smooth gear tooth surfaces, high-viscosity lubricants, and high speeds. Experiments (Ref 10) have shown that flame-hardened and induction-hardened gears have less resistance to micropitting than carburized gears of the same hardness. This is probably due to the lower carbon content of the surface layers of the flame-hardened and induction-hardened gears.

The following recommendations serve as guidelines for preventing the onset of micropitting in gearsets:

- Use smooth tooth surfaces produced by careful grinding or honing
- Use an adequate amount of cool, clean, and dry lubricant of the highest viscosity permissible
- Use high speeds if possible
- Use carburized steel with proper carbon content in the surface layers

Wear

Gearsets are susceptible to wear caused by adhesion, abrasion, and polishing.

Adhesion. Adhesive wear is classified as "mild" if it is confined to the oxide layers of the gear tooth surfaces. If, however, the oxide layers are disrupted and bare metal is exposed, the transition to severe adhesive wear usually occurs. Severe adhesive wear is termed scuffing and will be discussed in the section "Scuffing" in this chapter. Here we assume that scuffing has been avoided through proper design of the gears, selection of the lubricant, and control of the running-in process.

When new gear units are first operated, the contact between the gear teeth is not optimum because of unavoidable manufacturing inaccuracies. If the tribological conditions are favorable, mild adhesive wear occurs during running-in and usually subsides with time, resulting in a satisfactory lifetime for the gears. The wear that occurs during running-in is beneficial if it smooths the tooth surfaces (thereby increasing the specific film thickness) and if it increases the area of contact by removing minor imperfections through local wear. To ensure that the wear rate remains under control, new gearsets should be run-in by being operated for at least the first ten hours at one-half load.

The amount of wear considered tolerable depends on the expected lifetime for the gears and requirements for control of noise and vibration. Wear is considered excessive when the tooth profiles wear to the extent that high dynamic loads occur or the tooth thickness is reduced to the extent that bending fatigue becomes possible.

Many gears, because of practical limits on lubricant viscosity, speed, and temperature, must operate under boundary-lubricated conditions in which some wear is inevitable. Highly loaded, slow speed (<30 m/min, or 100 ft/min), boundary-lubricated gears are especially prone to excessive wear. Tests with slow-speed gears (Ref 10) have shown that nitrided gears have good wear resistance, whereas carburized and through-hardened gears have similar but lower wear resistance. Winter and Weiss (Ref 10) concluded that lubricant viscosity has the greatest effect on slow-speed adhesive wear, and that high-viscosity lubricants reduce the wear rate significantly. Winter and Weiss (Ref 10) also found that sulfur-phosphorus additives can be detrimental with slow-speed (<3 m/min, or 10 ft/min) gears, giving very high wear rates.

A few gear units operate under ideal conditions with smooth tooth surfaces, high pitch line speed, and high lubricant film thickness. For example, turbine gears that operated almost continuously at 9000 m/min (30,000 ft/min) pitch line speed still had the original machining marks on their teeth, even after operating for 20 years. Most gears, however, operate between the boundary and full-film lubrication regimes, under EHD conditions. In the EHD regime, with the proper type and viscosity of lubricant, the wear rate usually reduces during running-in and adhesive wear virtually ceases once running-in is completed. If the lubricant is properly maintained (cool, clean, and dry), the gearset should not suffer an adhesive wear failure.

Prevention of Adhesive Wear. The following recommendations serve as guidelines for preventing the onset of adhesive wear in gearsets:

- Use smooth tooth surfaces
- If possible, run-in new gearsets by operating the first 10 hours at one-half load
- Use high speeds if possible. Otherwise, recognize that highly loaded slow-speed gears are boundary lubricated and are especially prone to excessive wear. For these conditions, specify nitrided gears and the highest permissible lubricant viscosity
- For very slow-speed gears (<3 m/min, or 10 ft/min), avoid using lubricants with sulfur-phosphorus additives
- Use an adequate amount of cool, clean, and dry lubricant of the highest viscosity permissible

Abrasion. Abrasive wear on gear teeth is usually caused by contamination of the lubricant by hard, sharp-edged particles. Contamination enters gearboxes by being built-in, internally generated, ingested through breathers and seals, or inadvertently added during maintenance.

Many gear manufacturers do not fully appreciate the significance of clean assembly; it is not uncommon to find sand, machining chips, grinding dust, weld splatter, or other debris in new gearboxes. To remove built-in contamination, the gearbox lubricant should be drained and flushed before start-up and again after the first 50 h of operation, refilled with the recommended lubricant, and a new oil filter should be installed.

Internally generated particles are usually wear debris from gears or bearings due to Hertzian fatigue pitting or adhesive and abrasive wear. The wear particles are especially abrasive because they become work-hardened when they are trapped between the gear teeth. Internally generated wear debris can be minimized by using accurate surface-hardened gear teeth (with high pitting resistance), smooth surfaces, and high-viscosity lubricants.

Breather vents are used on gearboxes to vent internal pressure, which may occur when air enters through seals, or when air within the gearbox expands (or contracts) during the normal heating and cooling of the gear unit. The breather vent should be located in a clean, nonpressurized area and should have a filter to prevent ingress of airborne contaminants. In especially harsh environments, the gearbox can be completely sealed, and the pressure variation can be accommodated by an expansion chamber with a flexible diaphragm.

All maintenance procedures that involve opening any part of the gearbox or lubrication system must be carefully performed to prevent contamination of the gearbox system.

Abrasive wear due to foreign contaminants, such as sand or internally generated wear debris, is called three-body abrasion and is a common occurrence. Two-body abrasion also occurs when hard particles or asperities on one gear tooth abrade the opposing tooth surface. Unless the tooth surfaces of a surface-hardened gear are smoothly finished, they will act like files if the mating gear is appreciably softer. This is the reason that a worm pinion is polished after grinding before it is run with a bronze worm wheel. Manufacturers of computer disk drives have found that stainless steel pinions mated with anodized aluminum racks have excessively high wear rates. The anodized layer of the aluminum rack is extremely thin and brittle, and it breaks up and impregnates the relatively soft stainless steel pinion. The aluminum oxide particles then act like emery paper and wear the teeth of the rack very quickly.

The lubrication system should be carefully maintained and monitored to ensure that the gears receive an adequate amount of cool, clean, and dry lubricant. For circulating-oil systems, fine filtration removes contamination. Filters as fine as 3 μm (120 μin.) have significantly increased gear life. For oil bath gearboxes, the lubricant should be changed frequently to remove contamination. Under normal operating conditions, the lubricant should be changed at least every 2500 h of operation or every six months, whichever occurs first. For critical gearboxes, a regular program of lubricant monitoring can help prevent gear failures by showing when maintenance is required. The lubricant monitoring should include spectrographic and ferrographic analysis of contamination, along with analysis of acid number, viscosity, and water content.

Prevention of Abrasive Wear. The following guidelines should be observed to prevent abrasive wear in gearsets:

- Remove built-in contamination from new gearboxes by draining and flushing the lubricant before start-up and again after the first 50 h of operation. Refill with the recommended lubricant and install a new filter
- Minimize internally generated wear debris by using surface-hardened gear teeth, smooth tooth surfaces, and high-viscosity lubricants
- Minimize ingested contamination by maintaining oil-tight seals and using filtered breather vents located in clean, nonpressurized areas
- Minimize contamination that is added during maintenance by using good housekeeping procedures
- For circulating-oil systems, use fine filtration
- For oil bath systems, change the lubricant at least every 2500 h or every six months
- Monitor the lubricant with spectrographic and ferrographic analysis together with analysis of acid number, viscosity, and water content

Polishing Wear. If the extreme-pressure (EP) antiscuff additives in the lubricant are too chemically reactive, they may cause polishing of the gear tooth surfaces until they attain a bright mirror finish. Although the polished gear teeth may look good, polishing wear is undesirable because it generally reduces gear accuracy by wearing the tooth profiles away from their ideal form. Antiscuff additives used in lubricants to prevent scuffing, such as sulfur and phosphorus, will be covered when scuffing is discussed. They function by forming iron-sulfide and iron-phosphate films on areas of the gear teeth where high temperatures occur. Ideally, the additives should react only at temperatures where there is a danger of welding. If the rate of reaction is too high, and there is a continuous removal of the surface films caused by very fine abrasives in the lubricant, the polishing wear may be excessive (Ref 13).

Prevention of Polishing Wear. Polishing wear can be prevented by using less chemically active additives. As an alternative to sulfur-phosphorus additives, antiscuff lubricants are available with dispersions of potassium borate (Ref 14) that deposit EP films without chemically reacting with the metal. Removing the abrasives in the lubricant by using fine filtration or frequent oil changes is helpful.

In summary, the following guidelines should be observed to prevent polishing wear in gearsets:

- Use less chemically active antiscuff additives (for example, borate)
- Remove abrasives from the lubricant by using fine filtration or frequent oil changes

Scuffing

Scuffing is defined as localized damage caused by solid-phase welding between sliding surfaces. It is accompanied by the transfer of metal from one surface to another due to welding and tearing. It may occur in any sliding and rolling contact where the oil film is not thick enough to separate the surfaces. The symptoms of scuffing are microscopically rough, matte, and torn surfaces. Surface analysis that shows transfer of metal from one surface to the other is proof of scuffing.

Scuffing can occur in gear teeth when they operate in the boundary lubrication regime. If the lubricant film is insufficient to prevent significant metal-to-metal contact, the oxide layers that normally protect the gear tooth surfaces may be broken through, and the bare metal surfaces may weld together. The sliding that occurs between gear teeth results in tearing of the welded junctions, metal transfer, and catastrophic damage.

In contrast to pitting and bending fatigue, which only occur after a period of running time, scuffing may occur immediately upon start-up. In fact, gears are most vulnerable to scuffing when they are new and their tooth surfaces have not yet been smoothed by running-in. For this reason, it is wise to run-in a new gearbox under one-half load for at least 10 h to reduce the surface roughness of the teeth before applying full load. The gear teeth can be coated with iron-manganese phosphate or plated with copper or silver to protect them from scuffing during the critical running-in period.

The basic mechanism of scuffing is not clearly understood, but by general agreement it is believed to be caused by intense frictional heating generated by the combination of high sliding velocity and intense surface pressure. Blok's (Ref 15) critical temperature theory is believed to be the best criterion for predicting scuffing. It states that scuffing will occur in gear teeth that are sliding under boundary-lubricated conditions when the maximum contact temperature of the gear teeth reaches a critical magnitude. For mineral oils without antiscuff/EP additives, each combination of oil and rubbing materials has a critical scuffing temperature that is constant, regardless of the operating conditions (Ref 16). The critical scuffing temperatures are not constant for synthetic lubricants and lubricants with antiscuff additives; they must be determined from tests that closely simulate the operating conditions of the gears.

Today, most antiscuff additives are sulfur-phosphorus compounds, which form boundary-lubricating films by chemically reacting with the metal surfaces of the gear teeth at local points of high temperature. Antiscuff films help prevent scuffing by forming solid films on the gear tooth surfaces and inhibiting true metal-to-metal contact. The films of iron sulfide and iron phosphate have high melting points, allowing them to remain as solids on the gear tooth surfaces even at high contact temperatures. The rate of reaction of the antiscuff additives is greatest where the gear tooth contact temperatures are highest. Because of the rubbing action of the gear teeth, the surface films are repeatedly scraped off and reformed. In effect, scuffing is prevented by substituting mild corrosion in its place. Occasionally, antiscuff additives (for example, sulfur) are too chemically active, causing polishing wear and necessitating a change to less aggressive additives. Lubricants with antiscuff additives of potassium borate do not cause polishing wear because they deposit glasslike boundary films without reacting with the metal.

For mineral oils without antiscuff additives, the critical scuffing temperature increases with increasing viscosity and ranges from 150 to 300 °C (300 to 570 °F). The increased scuffing resistance of high-viscosity lubricants is believed to be due to differences in chemical composition rather than increases in viscosity. However, a viscosity increase also helps to reduce the risk of scuffing by increasing the lubricant film thickness and reducing the contact temperature generated by metal-to-metal contact.

Scuffing is controlled by the total contact temperature, T_c, which consists of the sum of the gear bulk temperature, T_b, and the flash temperature, T_f:

$$T_c = T_b + T_f \qquad \text{(Eq 1)}$$

The bulk temperature is the equilibrium temperature of the surface of the gear teeth before they enter the meshing zone. The flash temperature is the local and instantaneous temperature rise that occurs on the gear teeth due to the frictional heating as they pass through the meshing zone.

Anything that reduces either the bulk temperature or the flash temperature will reduce the total contact temperature and lessen the risk of scuffing. Higher viscosity lubricants or smoother tooth surfaces help by increasing the specific film thickness, which in turn reduces the frictional heat and, therefore, the flash temperature. Also, the lubricant performs the important function of removing heat from the gear teeth. The lubricant must be supplied to the gear teeth in such a way that it removes heat rapidly and maintains a low bulk temperature. A heat exchanger can be used with a circulating-oil system to cool the lubricant before it is sprayed at the gears.

Prevention of Scuffing. The gear designer can maximize scuffing resistance by optimizing the gear geometry so that the gear teeth are as small as possible, consistent with bending strength requirements, to reduce the temperature rise caused by sliding. Figure 2 shows that the rolling velocity of the pinion, V_{r1}, and the rolling velocity of the gear, V_{r2}, linearly increase from zero at the interference points to a maximum at each end of the path of contact. The sliding velocity is represented by the distance between the V_{r1} and V_{r2} lines. The amount of sliding is proportional to the distance from the pitch point, *P*, and is zero when the gear teeth contact at the pitch point, and largest at the ends of the path. Addendum modification can be used to balance and minimize the temperature rise that occurs in the addendum and dedendum of the gear teeth. The temperature rise may also be reduced by modifying the tooth profiles with slight tip and/or root relief to ease the load at the start and end of the engagement path where the sliding velocities are the greatest. Also, the gear teeth must be accurate and held rigidly in good alignment to minimize tooth loading and, therefore, the temperature rise.

Gear materials should be chosen with their scuffing resistance in mind. Nitrided steels, such as Nitralloy 135M, are generally found to have the highest resistance to scuffing, whereas stainless steels are liable to scuff even under near-zero loads. The thin oxide layer on stainless steel is hard and brittle and breaks up easily under sliding loads, exposing the bare metal and thus promoting scuffing. Like stainless steel, anodized aluminum has a low scuffing resistance. Hardness does not seem to be a reliable indication of scuffing resistance.

In summary, the following guidelines should be observed to prevent scuffing in gearsets:

- Use smooth tooth surfaces produced by careful grinding or honing
- Protect the gear teeth during the critical running-in period by coating them with iron-manganese phosphate or plating them with copper or silver. Run-in new gearsets by operating the first 10 h at one-half load
- Use high-viscosity lubricants with antiscuff additives, such as sulfur, phosphorus, or borate
- Cool the gear teeth by supplying an adequate amount of cool lubricant. For circulating-oil systems, use a heat exchanger to cool the lubricant
- Optimize the gear tooth geometry by using small teeth, addendum modification, and profile modification
- Use accurate gear teeth, rigid gear mountings, and good helix alignment
- Use nitrided steels for maximum scuffing resistance. Do not use stainless steel or aluminum for gears if there is a risk of scuffing

Elastohydrodynamic Lubrication

Gear teeth are subjected to enormous contact pressures on the order of the ultimate tensile strength of hardened steel, yet they are quite successfully lubricated with oil films that are <1 μm (<40 μin.) thick. This is possible because lubricants have a fortuitous property that causes their viscosity to increase dramatically with increased pressure. Figure 3 depicts the region of contact between mating gear teeth. It shows the shape of the elastically deformed teeth and the pressure distribution developed within the contact zone.

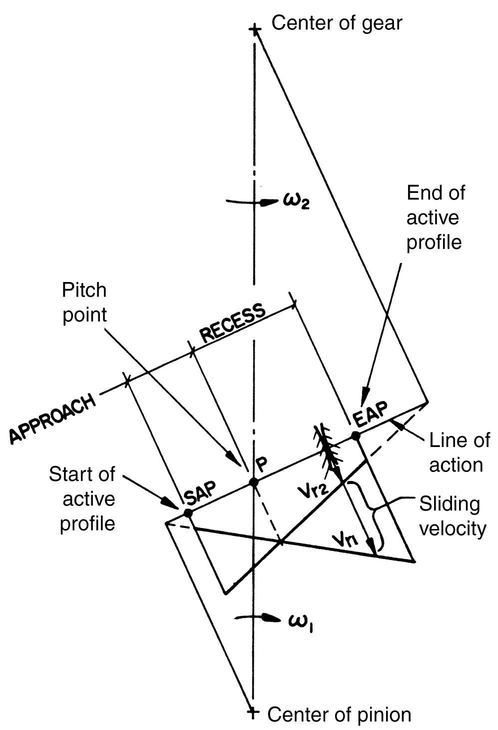

Fig. 2 Graphical representation of rolling velocities of pinion (V_{r1}) and gear (V_{r2}) in relation to sliding velocity

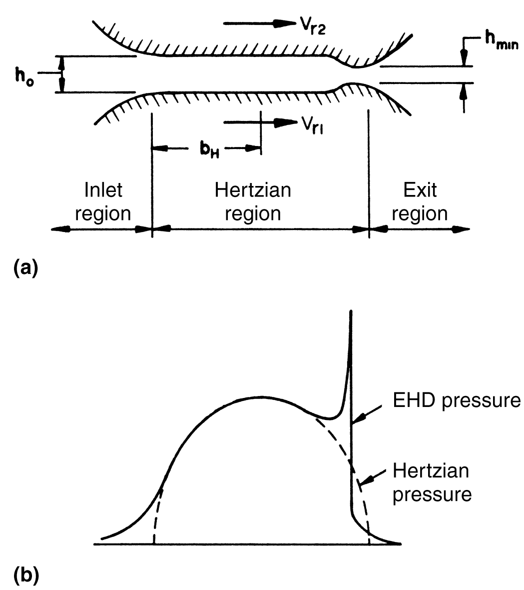

Fig. 3 Regions of elastohydrodynamic contact between the mating gear teeth of a gearset. (a) Schematic showing three distinct regions on pinion and gear tooth surfaces and key parameters determining oil film lubrication. (b) Plot of pressure distribution within contact zone. b_H, semiwidth of Hertzian contact band; h_0, central film thickness; h_{min}, minimum film thickness

The molecular adsorption of the lubricant onto the gear tooth surfaces causes it to be dragged into the inlet region of the contact, where its pressure is increased due to the convergence of the tooth surfaces. The viscosity increase of the lubricant caused by the increasing pressure helps to entrain the lubricant into the contact zone. Once it is within the high-pressure Hertzian region of the contact, the lubricant cannot escape because its viscosity has increased to the extent where the lubricant is virtually a rigid solid.

The following equation, from Dowson and Higginson (Ref 17) gives the minimum film thickness that occurs near the exit of the contact. The minimum film thickness, h_{min}, is obtained from:

$$h_{min} = \frac{1.63\alpha^{0.54}\,(\mu_0 V_e)^{0.7}\,\rho_n^{0.43}}{(X_\Gamma w_{Nr})^{0.13}\,E_r^{0.03}} \quad \text{(Eq 2)}$$

where α is the pressure-viscosity coefficient (in.^{2}/lbf), μ_0 is the absolute viscosity (reyns, or lbf · s/in.^{2}), and V_e is the entraining velocity, in which:

$$V_e = V_{r1} + V_{r2} \quad \text{(Eq 3)}$$

where V_{r1} and V_{r2} are the rolling velocities of pinion and gear, respectively, given by:

$$V_{r1} = \omega_1\,\rho_2 \quad \text{(Eq 4a)}$$

$$V_{r2} = \omega_1\,\rho_2 \quad \text{(Eq 4b)}$$

in which ω_1 and ω_2 are the angular velocities of pinion and gear, respectively, and ρ_1 and ρ_2 are the transverse radii of curvature of the pinion and gear, respectively.

The normal relative radius of curvature, ρ_n, is given by:

$$\rho_n = \frac{\rho_1\rho_2}{(\rho_2 \pm \rho_1)\cos\psi_b} \quad \text{(Eq 5)}$$

where ψ_b is the base helix angle. Referring back to Eq 2, X_Γ is the load-sharing factor, and w_{Nr} is the normal unit load given by:

$$w_{Nr} = \frac{W_{Nr}}{L_{min}} \quad \text{(Eq 6)}$$

in which W_{Nr} is the normal operating load and L_{min} is the minimum contact length.

E_r is the reduced modulus of elasticity, given by:

$$E_r = 2\left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right)^{-1} \quad \text{(Eq 7)}$$

where ν_1 is the Poisson's ratio of pinion, ν_2 is the Poisson's ratio of gear, E_1 is the modulus of elasticity of pinion, and E_2 is the modulus of elasticity of gear.

Figure 4 gives average values of absolute viscosity, μ_0, versus bulk temperature for typical mineral gear lubricants with a viscosity index of 95.

The pressure-viscosity coefficient, α, ranges from 0.5×10^{-4} to 2×10^{-4} in.^{2}/lbf for typical gear lubricants. Data for pressure-viscosity coefficients versus bulk temperature for typical gear lubricants are given in Fig 5.

The specific film thickness, λ, is given by:

$$\lambda = \frac{h_{min}}{\sigma} \quad \text{(Eq 8)}$$

where h_{min} is the minimum film thickness and σ is the composite surface roughness given by:

$$\sigma = (\sigma_1^2 + \sigma_2^2)^{1/2} \quad \text{(Eq 9)}$$

in which σ_1 is the root-mean-square (rms) surface roughness of the pinion and σ_2 is the rms surface roughness of the gear.

Load-Sharing Factor. The load-sharing factor, X_Γ, accounts for load sharing between succeeding pairs of teeth as affected by profile modification (tip and root relief) and whether the pinion or gear is the driver. Figure 6 gives plots of the load-sharing factors for unmodified and modified tooth profiles.

As shown by the exponents in Eq 2, the film thickness is essentially determined by the entraining velocity, lubricant viscosity, and pressure-viscosity coefficient; the elastic properties of the gear teeth and the load have relatively small influences. In effect, the relatively high stiffness of the oil film makes it insensitive to load, and an increase in load simply increases the elastic deformation of the tooth surfaces and widens the contact area, rather than decreasing the film thickness.

Blok's contact temperature theory (Ref 15) states that scuffing will occur in gear teeth which are sliding under boundary-lubricated conditions when the maximum contact temperature of the gear teeth reaches a critical magnitude. The contact temperature is the sum of two components: the bulk temperature and the flash temperature, as described in Eq 1.

Blok's flash temperature equation as formulated in AGMA 2001-B88, Appendix A (Ref 18) for spur and helical gears is:

$$T_f = \frac{0.8\mu_m X_\Gamma w_{Nr}\left|(V_{r1})^{0.5} - (V_{r2})^{0.5}\right|}{B_M(b_H)^{0.5}} \quad \text{(Eq 10)}$$

where μ_m is the mean coefficient of friction, B_M is the thermal contact coefficient, and b_H is the semiwidth of the Hertzian contact band.

Mean Coefficient of Friction. The following equation gives a typical value of $0.06 < \mu_m < 0.18$ for the mean coefficient of friction for gears operating in the partial EHD regime ($\lambda < 1$). It may give values too low for boundary-lubricated gears, where μ_m may be greater than 0.2, or too high for gears in the full-film regime ($\lambda > 2$), where μ_m may be less than 0.01:

$$\mu_m = 0.06\left(\frac{50}{50 - S}\right) \quad \text{(Eq 11a)}$$

where:

$$\left(\frac{50}{50 - S}\right) \leq 3.0 \quad \text{(Eq 11b)}$$

in which S is the average rms surface roughness given by:

$$S = \frac{\sigma_1 + \sigma_2}{2} \quad \text{(Eq 12)}$$

Thermal Contact Coefficient. The thermal contact coefficient, B_M, is given by:

$$B_M = (\lambda_M \rho_M c_M)^{0.5} \quad \text{(Eq 13)}$$

where λ_M is the heat conductivity, ρ_M is the density, and c_M is the specific heat per unit mass. For typical gear steels, B_M is ~43 lbf/(in. · s$^{0.5}$ · °F).

Semiwidth of the Hertzian Contact Band. The semiwidth of the Hertzian contact band, b_H, is defined as:

$$b_H = \left(\frac{8X_\Gamma w_{Nr}\rho_n}{\pi E_r}\right)^{0.5} \quad \text{(Eq 14)}$$

Bulk Temperature. The gear bulk temperature, T_b, is the equilibrium bulk temperature of the gear teeth before they enter the meshing zone. In some cases, the bulk temperature may be significantly higher than the temperature of the oil supplied to the gear mesh. In a test with ultrahigh-speed gears (Ref 19), the pinion bulk

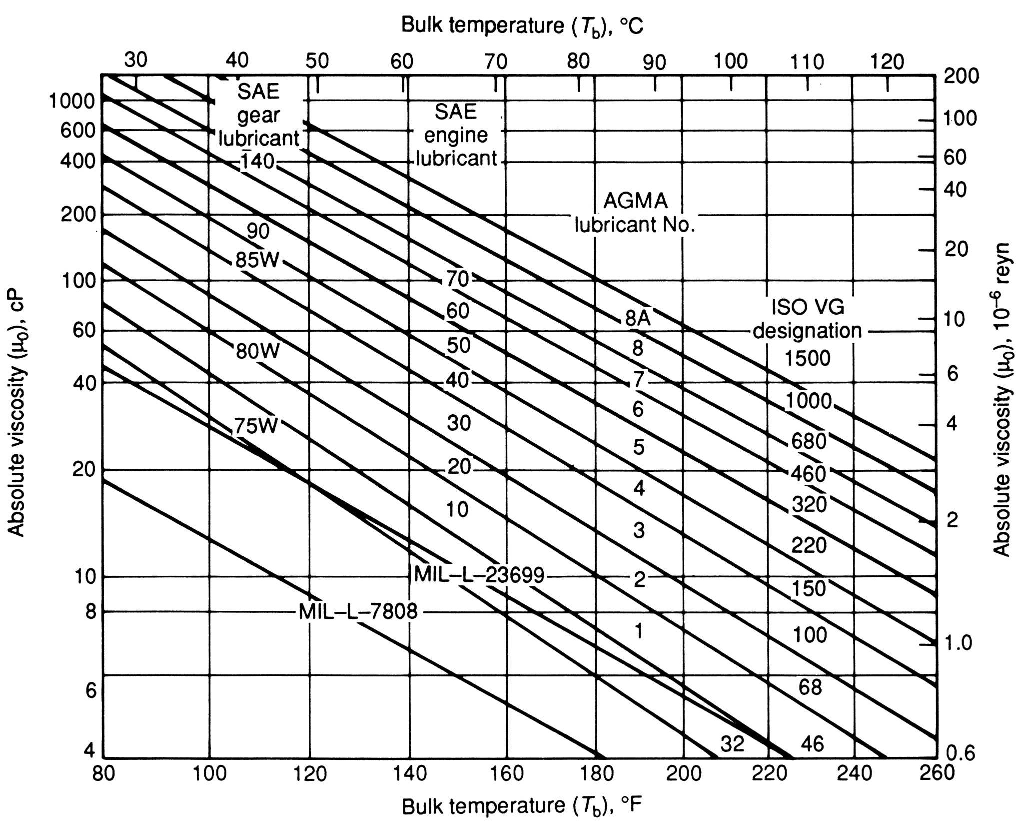

Fig. 4 Plot of absolute viscosity versus bulk temperature for selected mineral oil gear lubricants having a viscosity index of 95

temperature was 135 °C (275 °F), which is 95 °C (171 °F) hotter than the oil inlet temperature. For turbine gears at lower speeds, the bulk temperature rise of the gear teeth over the inlet oil temperature may range from 11 °C (20 °F) at 3700 m/min (1.2×10^4 ft/min) pitch line velocity to 22 °C (40 °F) at 4900 m/min (1.6×10^4 ft/min). At similar speeds, the bulk temperature rise of aircraft gears with less oil flow may range from 22 to 33 °C (40 to 60 °F).

Scuffing Temperature. The scuffing temperature, T_s, is the contact temperature at which scuffing is likely to occur with the chosen combination of lubricant and gear materials.

For mineral oils without antiscuff additives or for mineral oils with low concentrations of antiscuff additives, the scuffing temperature is independent of the operating conditions for a fairly wide range. For these oils, the scuffing temperature may be correlated with the composition of the oil. The viscosity grade is a convenient index of the composition and, thus, of the scuffing temperature.

For nonantiscuff mineral oils, the mean scuffing temperature (50% chance of scuffing) in °F is given by:

$$T_s = 146 + 59 \ln \nu_{40} \qquad \text{(Eq 15)}$$

where ν_{40} is the kinematic viscosity at 40 °C (105 °F) in centistokes.

For mineral oils with low concentrations of antiscuff additives, the mean scuffing temperature in °F is given by:

$$T_s = 245 + 59 \ln \nu_{40} \qquad \text{(Eq 16)}$$

The scuffing temperature determined from Forschungsstelle für Zahnräder und Getriebebau (FZG, or Technical Institute for the Study of Gears and Drive Mechanisms) test gears for mineral oils without antiscuff additives or with low concentrations of antiscuff additives may be extended to different gear steels, heat treatments, or surface treatments by introducing an empirical welding factor:

$$T_s = T_{b_{test}} + X_w T_{f_{test}} \qquad \text{(Eq 17)}$$

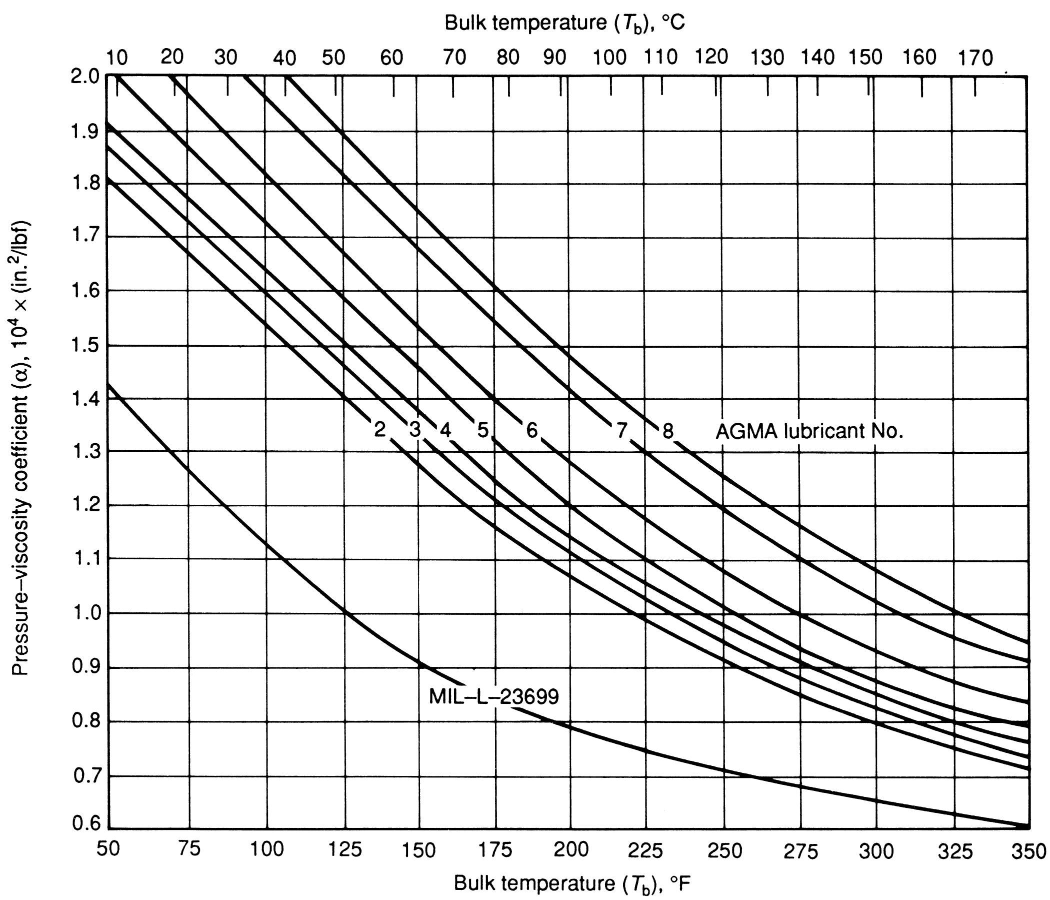

Fig. 5 Plot of pressure-viscosity coefficient versus bulk temperature for selected mineral oil gear lubricants

where X_w is the welding factor (see Table 3), $T_{b_{test}}$ is the bulk temperature of the test gears, and $T_{f_{test}}$ is the maximum flash temperature of the test gears.

Scuffing temperatures for synthetic lubricants typically used with carburized gears in the aerospace industry are shown in Table 4.

For mineral oils with high concentrations of antiscuff additives (for example, hypoid gear oils), research is still needed to determine whether the scuffing temperature is dependent on the materials and operating conditions. Special attention has to be paid to the correlation between test conditions and actual or design conditions.

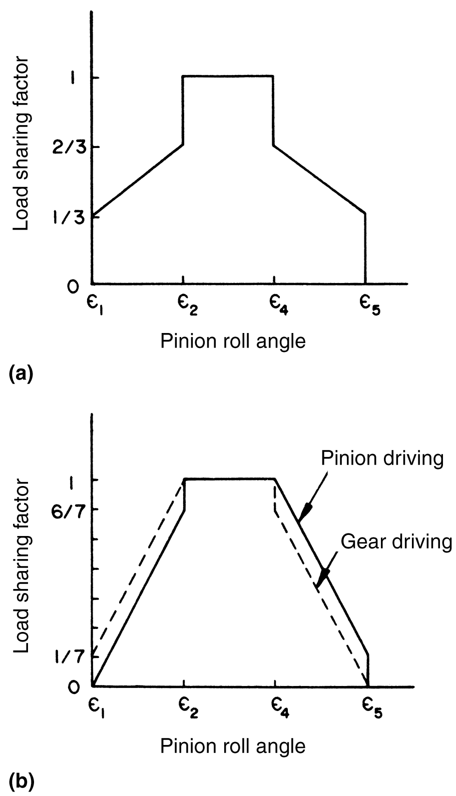

Fig. 6 Plot of load-sharing factor (X_Γ) versus pinion roll angle. (a) Unmodified tooth profiles. (b) Modified tooth profiles

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Table 3 Welding factor for selected gear steels

Material	Welding factor, X_W
Through-hardened steel	1.00
Phosphated steel	1.25
Copper-plated steel	1.50
Nitrided steel	1.50
Carburized steel	
Content of austenite < average	1.15
Content of austenite = average	1.00
Content of austenite > average	0.85
Stainless steel	0.45

Source: Ref 18

Lubricant Selection

The choice of lubricant depends on the type of gearing and enclosure, operating speed and load, ambient temperature, and method of lubricant application. Most gears are lubricated with one of the following types:

- Oil
- Synthetic lubricant
- Grease
- Adhesive open-gear lubricant
- Solid lubricant

The optimum lubricant for any application is the product that is the least expensive, considering both initial cost and maintenance costs, and meets the requirements.

Oil is the most widely used lubricant because it is readily distributed to gears and bearings and has both good lubricating and cooling properties. In addition, contamination may be readily removed by filtering periodically or draining and replacing the oil. However, it requires an oil-tight enclosure provided with adequate shaft seals.

Synthetic lubricants are used for applications (for example, aircraft gas turbines) where the oil must operate over a wide temperature range and have good oxidation stability at high temperature. Ester and hydrocarbon synthetic lubricants have high viscosity indices, giving them good fluidity or low viscosities at very low temperatures and acceptable viscosities at high temperatures. The volatility of esters is lower than that of mineral oils of the same viscosity,

Table 4 Mean scuffing temperature for synthetic lubricants typically used for operating carburized gears in aerospace applications

	Mean scuffing temperature, T_S	
Lubricant	°C	°F
MIL-L-6081 (grade 1005)	129	264
MIL-L-7808	205	400
MIL-L-23699	220	425
DERD2487	225	440
DERD2497	240	465
DOD-L-85734	260	500
Mobil SHC624	280	540
Dexron II	290	550

Source: Ref 20

thus reducing oil loss at high temperature. Despite their long service life, the extra cost of synthetic lubricants generally cannot be justified for oil-bath systems unless there are extreme temperatures involved, because the oil must be changed frequently to remove contamination.

Grease is suitable only for low-speed, low-load applications because it does not circulate well, and it is a relatively poor coolant. Grease-lubricated gears are generally boundary lubricated because the grease is either pushed aside or thrown from the gear teeth. Contamination from wear particles or other debris is usually trapped in the grease and requires costly maintenance to eliminate. Grease is often used to avoid leakage from enclosures that are not oil tight. However, if all the factors are considered, it is usually found that an oil lubricant is more economical and reliable than a grease lubricant for gear lubrication.

Open-gear lubricants are viscous adhesive semifluids used on large low-speed open gears, such as those used in iron ore and cement mills, antenna drives, bridge drives, cranes, and so on. Gears in these applications run slowly, and they are therefore boundary lubricated. The lubricant must bond strongly to resist being thrown off the gear teeth. However, the squeezing and sliding action of gear teeth tends to push the lubricant into the roots of the gear teeth where it is relatively ineffective. These lubricants are applied by hand brushing or by automatic systems that deliver an intermittent spray. Some open-gear lubricants are thinned with a quick-evaporating solvent/diluent to make them easier to apply. Open-gear lubricants share the disadvantages of grease lubrication, and they are especially costly (in addition to being messy) to maintain. For these reasons, the trend is away from open gears and toward the use of enclosed oil-lubricated gearboxes whenever possible.

Solid lubricants, usually in the form of bonded dry films, are used when the following conditions are encountered:

- The temperature is too high or too low for an oil or grease
- Leakage cannot be tolerated
- The gears must operate in a vacuum

These lubricants are usually molybdenum disulfide (MoS_2) or graphite in an inorganic binder, which is applied to the gear teeth and cured to form a dry film coating. Polytetrafluoroethylene (PTFE) and tungsten disulfide (WS_2) coatings are also used. Solid lubricants are expensive to apply and have limited wear lives. However, in many applications, such as spacecraft, they are the only alternative and can provide excellent service.

Oil Lubricant Applications

Of the above-mentioned lubricants, only oil will be discussed in greater detail in this chapter. Oil should be used as the lubricant unless the operating conditions preclude its use. Generally, the simplest and least expensive lubrication system for gears is a totally enclosed oil bath of mineral oil.

Spur, Helical, and Bevel Gears. The lubrication requirements of spur, helical, straight-bevel, and spiral-bevel gears are essentially the same. For this class of gears, the magnitudes of the loads and sliding speeds are similar, and requirements for viscosity and antiscuff properties are virtually identical. Many industrial spur and helical gear units are lubricated with rust and oxidation inhibited (R & O) mineral oils. The low-viscosity R & O oils, commonly called turbine oils, are used in many high-speed gear units where the gear tooth loads are relatively low. Mineral oils without antiscuff additives are suitable for high-speed lightly loaded gears where the high entraining velocity of the gear teeth develops thick EHD oil films. In these cases, the most important property of the lubricant is viscosity. Antiscuff/EP additives are unnecessary because the gear teeth are separated, eliminating metal-to-metal contact and the scuffing mode of failure. Slower speed gears, especially carburized gears, tend to be more heavily loaded. These gears generally require higher viscosity lubricants with antiscuff additives.

Hypoid gears, such as those used for automotive axles, are especially prone to scuffing because they are heavily loaded and have high sliding velocities. For these reasons, hypoid gear oils have the higher concentrations of antiscuff additives.

For critical applications, the contact temperature should be calculated with Blok's equation (see Eq 1) (Ref 15) and compared to the scuffing temperature of the lubricant. This quantitative method is effective for selecting a lubricant with adequate scuffing resistance.

Worm gears have high sliding velocity, which generates significant frictional losses. Fortunately, their tooth loads are relatively light, and they are successfully lubricated with mineral oils that are compounded with lubricity additives. These oils contain 3 to 10% fatty oil or low-acid tallow. The polar molecules of the additive form surface films by physical adsorption or by reaction with the surface oxide to form a metallic soap that acts as a low shear strength film, improving the "lubricity," or friction-reducing property.

Selection of Gear Lubricant Viscosity

The recommendations of AGMA 250.04 (Ref 21) should be followed when selecting lubricants for enclosed gear drives that operate at pitch line velocities ≤1500 m/min (≤5000 ft/min). AGMA 421.06 (Ref 22) should be consulted for high-speed drives (>1500 m/min, or 5000 ft/min).

Viscosity is one of the most important lubricant properties, and the higher the viscosity, the greater the protection against the various gear tooth failures. However, the viscosity must be limited to avoid excessive heat generation and power loss from churning and shearing of the lubricant by high-speed gears or bearings. The operating temperature of the gear drive determines the operating viscosity of the lubricant. If the lubricant is too viscous, excessive heat is generated. The heat raises the lubricant temperature and reduces its viscosity, reaching a point of diminishing returns where increasing the starting viscosity of the lubricant leads to a higher operating temperature and a higher oxidation rate, without a significant gain in operating viscosity.

Gear drives operating in cold climates must have a lubricant that circulates freely and does not cause high starting torques. A candidate gear lubricant should have a pour point at least 5 °C (9 °F) lower than the expected minimum ambient start-up temperature. Typical pour points for mineral gear oils are –7 °C (20 °F), whereas synthetic gear lubricants have significantly lower pour points of about –40 °C (–40 °F). Pour point depressants are used to tailor pour points of mineral lubricants for automotive hypoid gears to be as low as –40 °C (–40 °F).

The pitch line speed of the gears is a good index of the required viscosity. An empirical equation for determining required viscosity is:

$$\nu_{40} = \frac{7000}{(V)^{0.5}} \qquad \text{(Eq 18)}$$

where ν_{40} is the lubricant kinematic viscosity at 40 °C (105 °F) (in cSt) and V is the operating pitch line velocity (in ft/min) given by:

$$V = 0.262\, d \cdot n \qquad \text{(Eq 19)}$$

where d is the operating pitch diameter of pinion (in inches) and n is the pinion speed (rev/min).

Caution must be used when using AGMA recommendations for viscosity. For example, there was an application where two gear drives were considered to be high speed. The pinion speed was 3625 rev/min, qualifying the gear units as high-speed gear drives per AGMA 421.06. The gear drives were supplied with oil having the recommended viscosity (per AGMA 421.06) of ISO 68. However, because the pinion was relatively small, its pitch line velocity was only 1000 m/min (3000 ft/min). This qualifies the gear drives as slow speed per AGMA 250.04, which recommends a viscosity of ISO 150. Both gear drives failed within weeks of start-up because of pitting fatigue. The empirical equation (Eq 18) for this application gives:

$$\nu_{40} = \frac{7000}{(3000)^{0.5}} = 128 \text{ cSt} \qquad \text{(Eq 20)}$$

This indicates that the viscosity per AGMA 421.06 (68 cSt, or 6.8×10^{-5} m^2/s) is much too low, and the viscosity per AGMA 250.04 (150 cSt, or 1.5×10^{-4} m^2/s) is appropriate. Hence, definitions of high-speed versus slow-speed gear drives must be carefully considered, and pitch line velocity is generally a better index than shaft speed. The gear drives were rebuilt with new gearsets and the ISO VG 68 oil was replaced with ISO VG 150. The gear drives then operated without overheating, and the pitting was eliminated.

For critical applications, the specific film thickness, λ, should be calculated with Dowson and Higginson's equation (see Eq 8) (Ref 16). The specific film thickness is a useful measure of the lubrication regime. It can be used with Fig. 7 as an approximate guide to the probability of wear-related surface distress. Figure 7 is based on the data of Wellauer and Holloway (Ref 23), which were obtained from several hundred laboratory tests and field applications of gear drives.

Application of Gear Lubricants

The method of applying the lubricant to the gear teeth depends primarily on the pitch line velocity.

Splash lubrication systems are the simplest, but they are limited to a pitch line velocity of ~1000 m/min (~3000 ft/min). The gears should dip into the oil bath for about twice the tooth depth to provide adequate splash for pinions and bearings and to reduce losses due to churning. The gear housing should have troughs to capture the oil flowing down the housing walls, channeling it to the bearings.

The range of splash lubrication can be extended to ~1500 m/min (~5000 ft/min) by using baffles and oil pans to reduce churning. However, at velocities >1000 m/min (>3000 ft/min), providing auxiliary cooling with fans and improving heat transfer by adding fins to the housing are usually necessary.

Pressure-Fed Systems. Above 1500 m/min (5000 ft/min), most gears are lubricated by a pressure-fed system. For gearboxes with antifriction bearings, spraying the oil at the gear mesh only and relying on splash to lubricate the bearings is permissible up to a maximum pitch line velocity of 2100 m/min (7000 ft/min). Above this speed and for gear drives with journal bearings, both the gears and bearings should be pressure-fed.

The oil jets should be placed on the incoming side of the gear mesh for pitch line velocities ≤2600 m/min (≤8000 ft/min). Above 2600 m/min (8000 ft/min), more oil is needed for cooling than for lubricating, and the oil flow removes heat best by being directed at the outgoing side of the gear mesh where the oil jets can strike the hot drive-side of the gear teeth.

For very high-speed gears (Ref 19) (>5300 m/min, or 16,000 ft/min), there is a danger that the amount of oil carried to the incoming side of the gear mesh may be inadequate, and it is prudent to add a supplementary flow at the incoming side of the gear mesh. Generally, about two-thirds of the oil flow should be supplied to the outgoing side of the mesh for cooling, and one-third of the flow directed at the incoming side for lubrication. The placement of the oil jets is a crucial factor when pitch line velocities are >6100 m/min (>20,000 ft/min). At speeds this high, experiments are required to find the optimum number and location for the oil jets.

In pressure-fed systems, the following parameters must be considered to ensure adequate lubrication and cooling of the gear mesh:

- Quantity of flow
- Jet size
- Feed pressure
- Number of jets

There are general guidelines, based on experience and experimentation, for specifying these parameters, but each application must be evaluated independently based on its particular operating conditions and requirements.

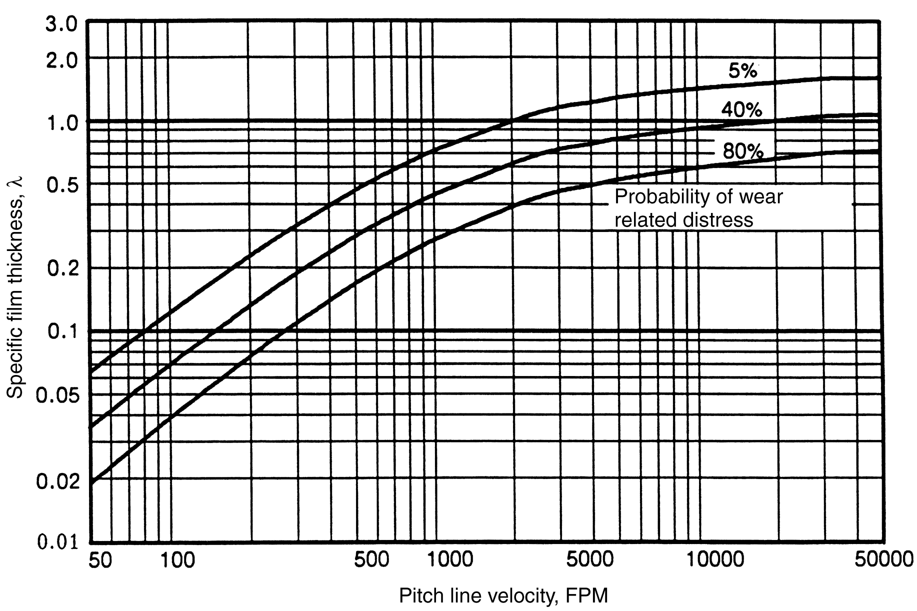

Fig. 7 Probability of wear distress as a function of specific film thickness and pitch line velocity (in ft/min). Source: Ref 18

An empirical equation used to calculate the quantity of oil flow, q, is:

$$q = P/c \quad \text{(Eq 21)}$$

where q is the oil flow rate (in gal/min), P is the transmitted power (in hp), and c is taken from Table 5.

For a typical industrial application transmitting 150 kW (200 hp), where weight is not critical, the designer might choose a value of c = 200 hp/(gal/min), resulting in a copious flow of 4 L/min (1 gal/min). On the other hand, for a high-efficiency aviation application transmitting 150 kW (200 hp), where weight is critical, a value of c = 800 hp/(gal/min) might be chosen, resulting in a lean flow of 1 L/min (0.25 gal/min). Some applications may require different flow rates than those given by Table 5. For instance, wide-face, high-speed gearing may require a higher flow rate to ensure uniform cooling and full-face coverage.

The proper jet size, feed pressure, and number of jets must be determined to maintain the proper flow rate, jet velocity, and full-face coverage.

Jet Size. The diameter of a jet can be calculated for a given flow rate and pressure based on the viscosity of the oil at the operating temperature (Ref 24). There are practical limitations on jet size, and the minimum recommended size is 0.8 mm (0.03 in.). If a jet smaller than this is used, contaminants in the oil may clog it. Typical jet diameters range from 0.8 to 3.0 mm (0.03 to 0.12 in.).

Feed Pressure. The feed pressure determines the jet velocity, which in turn determines the amount of oil that penetrates the gear mesh. Typical feed pressures range from 20 to 100 psig. Industrial application feed pressures are typically 30 psig, and high-speed aerospace applications are typically 100 psig. In general, the higher the pressure, the greater the cooling (Ref 25), but the higher the pressure, the smaller the jet diameter. Therefore, pressure is limited by the minimum recommended jet diameter of 0.8 mm (0.03 in.).

Jet Quantity. The number of jets should be sufficient to provide complete lubrication coverage of the face width. More than one jet for each gear mesh is advisable because of the possibility of clogging. The upper limit on the number of jets is determined by the flow rate and jet diameter; too many jets for a given flow rate will result in a jet diameter less than the minimum recommended.

Table 5 Recommended values of constant *c* based on oil flow and gear mesh specifications

c, hp/(gal/min)	Flow conditions	Applications
200	Copious	General industrial
400	Adequate	Typical aviation
800	Lean	Lightweight, high-efficiency aviation
1000	Starved	Only for unusual conditions

Case History: Failure of a 24-Unit Speed-Increaser Gearbox

This case history demonstrates many of the principles of gear lubrication described in earlier sections of this chapter. Lubricant viscosity, EHD film thickness, contact temperature, and design considerations are addressed.

Background. In an industrial application, 24 speed-increaser gearboxes were used to transmit 258 kW (346 hp) and increase speed from 55 to 375 rev/min. The gears were parallel shaft, single helical, carburized, and ground. The splash lubrication system used a mineral oil without antiscuff additives with ISO 100 viscosity. After about 250 h of operation, two gearboxes failed by bending fatigue. The gear tooth profiles were so badly worn that determining the primary failure mode was impossible. Three other gearboxes with less service were selected for inspection. One had logged 15 h, and the other two had operated for 65 h each. Upon disassembly, no broken teeth were found, but all three gearboxes had scuffed gear teeth. The primary failure mode was scuffing, and the earlier bending fatigue failures were caused by dynamic loads generated by the worn gear teeth. Subsequent inspection of the remaining gearboxes revealed that all had scuffing damage, which probably had occurred immediately upon start-up because the loads were not reduced during run-in.

Analysis. Fortunately, a prototype gearbox had been run at one-half load for about 50 h. When these gears were inspected, no signs of distress were seen on any of the gear teeth. The tooth profiles were smooth with surface roughness, R_q, estimated to be 0.5 μm (20 μin.), and the contact pattern indicated 100% face contact. This gearbox was reassembled and run under one-half load until its oil sump temperature reached equilibrium at 95 °C (200 °F). For this application, the ambient temperature was in the range of 10 to 50 °C (50 to 125 °F). The center distance of the gears was 405 mm (16 in.) and the pitch line velocity was 120 m/min (400 ft/min). Referring to AGMA 250.04 (Ref 21), the recommended viscosity for these conditions is ISO 150 or ISO 220.

Using the empirical equation (Eq 18), the following is obtained:

$$\nu_{40} = \frac{7000}{(400)^{0.5}} = 350 \text{ cSt} \qquad \text{(Eq 22)}$$

Hence, the empirical equation recommends a viscosity close to ISO 320. It is apparent that the viscosity that was originally supplied (ISO VG 100) was too low.

The EHD film thickness was calculated with a special computer program (Ref 26). The gear bulk temperature was assumed to be 110 °C (230 °F), which is 17 °C (30 °F) hotter than the measured oil sump temperature. The following data for the ISO VG 100 lubricant was obtained from Fig. 4 and 5: $\mu_0 = 6.6$ cP $= 0.96 \times 10^{-6}$ reyns and $\alpha = 1.02 \times 10^{-4}$ in.^{2}/lbf.

Figure 8 shows a plot of the film thickness versus position on the pinion tooth. The minimum film thickness occurs low on the pinion tooth near the lowest point of single-tooth contact (LPSTC), where h_{min} is 0.053 μm (2.1 μin.). The specific film thickness, based on R_q of 0.5 μm (20 μin.) for both profiles, is $\lambda = 0.073$. Figure 7 shows that the gears operate in the boundary lubrication regime. The program predicts that the probability of wear is >95%.

The contact temperature was also calculated with the program. The scuffing temperature for the ISO VG 100 lubricant was calculated with the equation for nonantiscuff mineral oils:

$$T_s = 146 + 59 \ln (100) = 418 \text{ °F} \qquad \text{(Eq 23)}$$

Figure 9 shows a plot of the contact temperature versus position on the pinion tooth. The maximum contact temperature occurs high on the pinion tooth near the highest point of single-tooth contact (HPSTC), where T_c is 226 °C (439 °F). The program predicts that the probability of scuffing is 63%. This is considered to be a high risk of scuffing. The relatively high-temperature peak near the tip of the pinion tooth was caused by the geometry of the gears.

The designer selected a long-addendum tooth for the pinion. Long-addendum pinions perform well in speed reducers, where they increase the amount of recess action and decrease the amount of approach action of the gear mesh. Because recess action is much smoother than approach action, long-addendum pinions give speed reducers smooth meshing characteristics. When operated as a speed increaser, however, the approach and recess portions of the gear mesh reverse, making a long-addendum pinion rough running and vulnerable to scuffing.

To explore the possibilities for reducing the scuffing risk, new gear tooth geometry was proposed with the pinion and gear addenda designed to minimize the flash temperature rise. The new gearset, analyzed with the program, assumed the lubricant was a mineral oil with antiscuff additives, with a viscosity of ISO 220, and with the following properties: $\mu_0 = 10$ cP $= 1.45 \times 10^{-6}$ reyns, $\alpha = 1.09 \times 10^{-4}$ in.^{2}/lbf, and

$$T_s = 245 + 59 \ln (220) = 563 \text{ °F} \qquad \text{(Eq 24)}$$

Figure 10 shows that the film thickness increases to h_{min} of 0.068 μm (2.7 μin.), and the specific film thickness increases to $\lambda = 0.097$. Figure 7 shows that the gears still operate in the

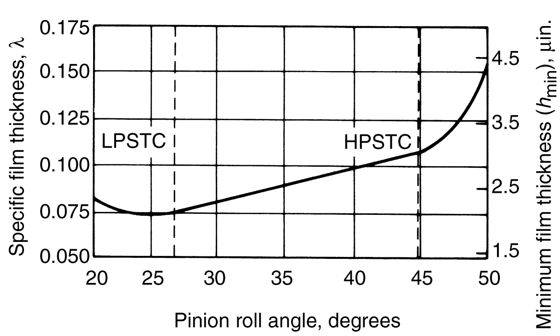

Fig. 8 Plot of film thickness versus pinion roll angle for gear tooth geometry of a scuffed gearset. Minimum specific film thickness, λ_{min}, 0.073; probability of wear, >95%

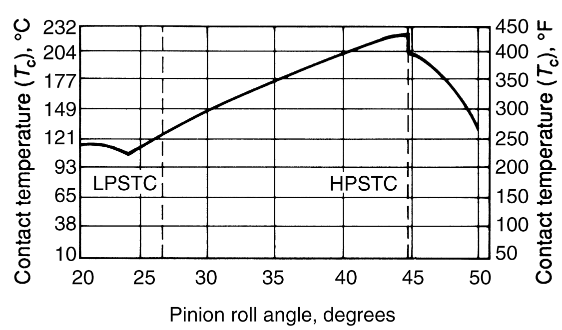

Fig. 9 Plot of contact temperature versus pinion roll angle for gear tooth geometry of scuffed gearset. Maximum T_c, 226 °C (439 °F); scuffing probability, 63%

boundary lubrication regime; however, the probability of wear is reduced to 94%. Figure 11 shows that the optimized gear geometry reduced the maximum contact temperature to a T_c of 150 °C (302 °F). The combination of reduced contact temperature and the increased scuffing resistance provided by the higher viscosity mineral oil with antiscuff additives reduces the scuffing probability to <5%.

Summary. Typical of many gear failures, this case history shows that several factors contributed to the failures:

- The lubricant viscosity was too low
- No antiscuff additives were used
- A gearbox designed as a speed reducer was used as a speed increaser
- The gear teeth were not provided with a coating or plating to ease running-in
- The gears were not run-in properly under reduced loads

Gear failures, as exemplified by this case history, can be avoided if designers and operators recognize that the lubricant is an important component of a gearbox and appreciate that the tribology of gearing requires the consideration and control of many interrelated factors.

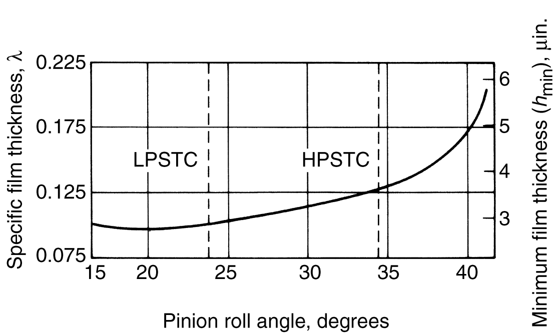

Fig. 10 Plot of film thickness versus pinion roll angle for gear tooth geometry that was optimized for maximum scuffing resistance. λ_{min}, 0.097; probability of wear, 94%

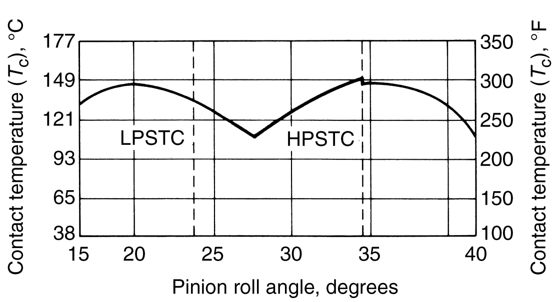

Fig. 11 Plot of contact temperature versus pinion roll angle for gear tooth geometry that was optimized for maximum scuffing resistance. Maximum T_f, 150 °C (302 °F); scuffing probability, <5%

ACKNOWLEDGMENT

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REFERENCES

1. R. Errichello, The Lubrication of Gears, *Lubrication Engineering,* Jan–April, 1990
2. E.E. Shipley, Gear Failures, *Mach. Des.,* 7 Dec 1967, p 152–162
3. D.W. Dudley, Gear Wear, *Wear Control Handbook,* American Society of Mechanical Engineers
4. P.M. Ku, Gear Failure Modes—Importance of Lubrication and Mechanics, *ASLE Trans.,* Vol 19 (No. 3), 1975, p 239–249
5. D.J. Wulpi, *Understanding How Components Fail,* American Society for Metals, 1985
6. L.E. Alban, Failures of Gears, *Failure Analysis and Prevention,* Vol 11, 9th ed., *Metals Handbook,* ASM International, 1986, p 586–601
7. D. Godfrey, Recognition and Solution of Some Common Wear Problems Related to Lubrication and Hydraulic Fluids, *Lubr. Eng.,* Feb 1987, p 111–114
8. W.E. Littman, The Mechanism of Contact Fatigue, *Interdisciplinary Approach to the Lubrication of Concentrated Contacts,* SP-237, National Aeronautics and Space Administration, 1970, p 309–377
9. T. Ueno et al., "Surface Durability of Case-Carburized Gears—On a Phenomenon of Grey Staining of Tooth Surface," Paper No. 80-C2/DET-27, American Society of Mechanical Engineers, 1980, p 1–8
10. H. Winter and T. Weiss, "Some Factors Influencing the Pitting, Micropitting (Frosted Areas) and Slow Speed Wear of Surface-Hardened Gears," Paper No. 80-C2/DET-89, American Society of Mechanical Engineers, 1980, p 1–7
11. E.E. Shipley, "Failure Analysis of Coarse-Pitch, Hardened, and Ground Gears," Paper

No. P229.26, American Gear Manufacturers Association, 1982, p 1–24
12. S. Tanaka et al. "Appreciable Increases in Surface Durability of Gear Pairs with Mirror-Like Finish," Paper No. 84-DET-223, American Society of Mechanical Engineers, 1984, p 1–8
13. A. Milburn, R. Errichello, and D. Godfrey, "Polishing Wear," Paper No. 90 FTM 5, American Gear Manufacturers Association, 1990, p 1–13
14. J.H. Admas and D. Godfrey, Borate Gear Lubricant-EP Film Analysis and Performance, *Lubr. Eng.,* Vol 37 (No. 1), Jan 1981, p 16–21
15. H. Blok, "Les Temperatures de Surface dans les Conditions de Graissage sons Pression Extreme," Second World Petroleum Congress (Paris), June 1937
16. H. Blok, The Postulate about the Constancy of Scoring Temperatures, *Interdisciplinary Approach to the Lubrication of Concentrated Contacts,* SP-237, National Aeronautics and Space Administration, 1970, p 153–248
17. D. Dowson, Elastohydrodynamics, Paper No. 10, *Proc. Inst. Mech. Eng.,* Vol 182, PT3A, 1967, p 151–167
18. "Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth," 2001-B88, American Gear Manufacturers Association, 1988
19. M. Akazawa, T. Tejima, and T. Narita, "Full Scale Test of High Speed, High Powered Gear Unit—Helical Gears of 25,000 PS at 200 m/s PLV," Paper No. 80-C2/DET-4, American Society of Mechanical Engineers, 1980
20. R.J. Drago, "Comparative Load Capacity Evaluation of CBN-Finished Gears," Paper No. 88 FTM 8, American Gear Manufacturers Association, Oct 1988
21. "AGMA Standard Specification—Lubrication of Industrial Enclosed Gear Drives," No. 250.04, American Gear Manufacturers Association, Sept 1981
22. "Practice for High Speed Helical and Herringbone Gear Units," No. 421.06, American Gear Manufacturers Association, Jan 1969
23. E.J. Wellauer and G.A. Holloway, Application of EHD Oil Film Theory to Industrial Gear Drives, *J. Eng. Ind. (Trans. ASME),* Vol 98, Series B (No. 2), May 1976, p 626–634
24. R.J. Drago, *Fundamentals of Gear Design,* Butterworths, 1998
25. L. Akin and D. Townsend, "Study of Lubricant Jet Flow Phenomena in Spur Gears," TMX-71572, National Aeronautics and Space Administration, Oct 1974
26. SCORING+, computer program, GEARTECH Software, Inc., Copyright 1985–1992

SELECTED REFERENCES

- R.H. Boehringer, Grease, *Friction, Lubrication, and Wear Technology,* Vol 18, *ASM Handbook,* ASM International, 1992, p 123–131
- H.S. Cheng, Lubrication Regimes, *Friction, Lubrication, and Wear Technology,* Vol 18, *ASM Handbook,* ASM International, 1992, p 89–97
- R.S. Fein, Liquid Lubricants, *Friction, Lubrication, and Wear Technology,* Vol 18, *ASM Handbook,* ASM International, 1992, p 81–88
- S.Q.A. Rizvi, Lubricant Additives and Their Functions, *Friction, Lubrication, and Wear Technology,* Vol 18, *ASM Handbook,* ASM International, 1992, p 98–112
- H.E. Sliney, Solid Lubricants, *Friction, Lubrication, and Wear Technology,* Vol 18, *ASM Handbook,* ASM International, 1992, p 113–122

Gear Materials, Properties, and Manufacture
J.R. Davis, editor, p39-76
DOI:10.1361/gmpm2005p039

CHAPTER 3

Ferrous and Nonferrous Alloys

GEAR MATERIALS can be broadly classified into two groups: nonmetallic and metallic materials. Nonmetallics are the plastics, both thermoplastic and thermosetting, used for gearing. These materials are described in Chapter 4, "Plastics." Metallic gear materials can be further subdivided into ferrous, or iron-base alloys, and nonferrous alloys. The most commonly used ferrous alloys are the wrought surface-hardening and through-hardening carbon and alloy steels. In fact, these steels are the most widely used of all gear materials and will be the emphasis of this chapter. Other ferrous alloys used for gears are cast irons, cast steels, powder metallurgy (P/M) irons and steels, stainless steels, tool steels, and maraging steels.

Although a number of nonferrous alloys have been used for gears, by far the most commonly employed are copper-base alloys. Die cast aluminum-, zinc-, and magnesium-base alloys are also sometimes used. More recently titanium alloy Ti-6Al-4V has been used in some specialized applications.

Wrought Gear Steels

Wrought steel is the generic term applied to carbon and alloy steels which are mechanically worked into form for specific applications. The standard wrought steel forms are round bar stock, flat stock, and forgings. Forgings reduce machining time and are available in a wide range of sizes and grades.

In general, there are two types of wrought gear steels: surface-hardening and through-hardening grades. The surface-hardened steels are hardened to a relatively thin case depth and include carburizing, nitriding, and carbonitriding steels. Surface-hardening steels include plain carbon and alloy steels with carbon content generally not exceeding 0.25% C. Table 1 lists compositions of surface-hardening steels.

Through-hardening steels may be comparatively shallow hardening or deep hardening, depending on their chemical composition and method of hardening. Through-hardening steels include plain carbon and alloy steels with carbon content ranging from 0.30 to approximately 0.55% C. Table 2 lists the compositions of through-hardening gear steels. Table 3 lists the mechanical properties of both surface- and through-hardening steels in various conditions.

The steels selected for gear applications must satisfy two basic sets of requirements that are not always compatible—those involving fabrication and processing and those involving service. Fabrication and processing requirements include machinability, forgeability, and response to heat treatment as it affects fabrication and processing. Service requirements are related to the ability of the gear to perform satisfactorily under the conditions of loading for which it was designed and thus encompass all mechanical-property requirements, including fatigue strength, response to heat treatment, and resistance to wear.

Because resistance to fatigue failure is partly dependent upon the cleanness of the steel and upon the nature of allowable inclusions, melting practice may also be a factor in steel selection and may warrant selection of a steel produced by vacuum induction melting followed by vacuum arc remelting (VIM/VAR) or by electroslag refining. The mill form from which a steel gear is machined is another factor that may effect its performance. Many heavy-duty steel gears are machined from forged blanks that have been processed to provide favorable grain flow consistent with load pattern rather than being machined from blanks cut from mill-rolled bar.

Surface-Hardening Steels

General Properties. Carburizing or nitriding grades of steel are usually specified where maximum wear resistance is required for bearing surfaces. Carburized case-hardened gears are best suited for heavy-duty service, for example, transmission gears, and offer high resistance to wear, pitting, and fatigue. Surfaces must be sufficiently hard to resist wear and of sufficient depth to prevent case crushing. A rough rule for case depth is that it shall not exceed one-sixth of the base thickness of the tooth. Figure 1 shows the typical structure of a case-hardened gear steel.

A case-hardened gear provides maximum surface hardness and wear resistance and at the same time provides interior toughness to resist shock. In general, case-hardened gears can withstand higher loads than through-hardened gears, although the latter are quieter and less expensive because of the simpler heat treatment required (Ref 1).

Selection Factors. The following factors must be considered when selecting a case-hardened gear (Ref 1):

- High tooth pressures will crack a thin case.
- Too soft a core will not provide proper backing for a hard case.

Table 1 Chemical compositions of carburizing steels

The carburizing grades commonly used for gears are described in text.

	Composition, %					
Steel	C	Mn	Ni	Cr	Mo	Other
Carbon steels						
1010	0.08–0.13	0.30–0.60	. . .	. . .	. . .	(a), (b)
1015	0.13–0.18	0.30–0.60	. . .	. . .	. . .	(a), (b)
1018	0.15–0.20	0.60–0.90	. . .	. . .	. . .	(a), (b)
1019	0.15–0.20	0.70–1.00	. . .	. . .	. . .	(a), (b)
1020	0.18–0.23	0.30–0.60	. . .	. . .	. . .	(a), (b)
1021	0.18–0.23	0.60–0.90	. . .	. . .	. . .	(a), (b)
1022	0.18–0.23	0.70–1.00	. . .	. . .	. . .	(a), (b)
1025	0.22–0.28	0.30–0.60	. . .	. . .	. . .	(a), (b)
1524	0.19–0.25	1.35–1.65	. . .	. . .	. . .	(a), (b)
1527	0.22–0.29	1.20–1.50	. . .	. . .	. . .	(a), (b)
Resulfurized steels						
1117	0.14–0.20	1.00–1.30	. . .	. . .	. . .	0.08–0.13 S
1118	0.14–0.20	1.30–1.60	. . .	. . .	. . .	0.08–0.13 S
Alloy steels						
3310	0.08–0.13	0.45–0.60	3.25–3.75	1.40–1.75	. . .	(b), (c)
4023	0.20–0.25	0.70–0.90	. . .	. . .	0.20–0.30	(b), (c)
4027	0.25–0.30	0.70–0.90	. . .	. . .	0.20–0.30	(b), (c)
4118	0.18–0.23	0.70–0.90	. . .	0.40–0.60	0.08–0.15	(b), (c)
4320	0.17–0.22	0.45–0.65	1.65–2.00	0.40–0.60	0.20–0.30	(b), (c)
4615	0.13–0.18	0.45–0.65	1.65–2.00	. . .	0.20–0.30	(b), (c)
4620	0.17–0.22	0.45–0.65	1.65–2.00	. . .	0.20–0.30	(b), (c)
4815	0.13–0.18	0.40–0.60	3.25–3.75	. . .	0.20–0.30	(b), (c)
4820	0.18–0.23	0.50–0.70	3.25–3.75	. . .	0.20–0.30	(b), (c)
5120	0.17–0.22	0.70–0.90	. . .	0.70–0.90	. . .	(b), (c)
5130	0.28–0.33	0.70–0.90	. . .	0.80–1.10	. . .	(b), (c)
8617	0.15–0.20	0.70–0.90	0.40–0.70	0.40–0.60	0.15–0.25	(b), (c)
8620	0.18–0.23	0.70–0.90	0.40–0.70	0.40–0.60	0.15–0.25	(b), (c)
8720	0.18–0.23	0.70–0.90	0.40–0.70	0.40–0.60	0.20–0.30	(b), (c)
8822	0.20–0.25	0.75–1.00	0.40–0.70	0.40–0.60	0.30–0.40	(b), (c)
9310	0.08–0.13	0.45–0.65	3.00–3.50	1.00–1.40	0.08–0.15	(b), (c)
Special alloys						
CBS-600	0.16–0.22	0.40–0.70	. . .	1.25–1.65	0.90–1.10	0.90–1.25 Si
CBS-1000M	0.10–0.16	0.40–0.60	2.75–3.25	0.90–1.20	4.00–5.00	0.40–0.60 Si 0.15–0.25 V
Pyrowear Alloy 53	0.10	0.35	2.00	1.00	3.25	1.00 Si, 2.00 Cu, 0.10 V

(a) 0.04 P max, 0.05 S max. (b) 0.15–0.35 Si. (c) 0.035 P max, 0.04 S max

- Compressive stresses in the case improve fatigue durability, and a high case hardness increases wear resistance.
- If the ratio of case depth to core thickness is too small, excessive stresses in subsurface layers can produce poor fatigue life.
- Residual tensile stresses are highest with low core hardness and increase with increasing case depth. These stresses can be relieved by tempering.

Grain size variations have an important effect on core properties. These variations are influenced by the type of steel and the method of heat treatment used subsequent to carburizing. Section thickness also influences core properties.

A tough tooth core may not be required in applications where a gear will not be subjected to impact loading. In these applications, core properties are relatively unimportant, provided the core is sufficiently hard to support the case. Considering the case alone, it is important that the surface resist wear and fatigue bending, because bending stresses vary from a maximum at the surface to zero near the tooth center.

Carburizing Steels. Carburizing is a very important commercial heat-treating operation that is used to modify the surface chemistry of components manufactured from ferrous alloys by the processes of carbon absorption and diffusion. The process is carried out at a temperature sufficient to render the steel austenitic (generally between 850 and 950 °C, or 1560 and 1740 °F), followed by quenching and tempering to form a high-carbon martensitic structure. The increase in carbon content of a carburized surface layer results in a substantial change in the properties of the effected volume of material. For example, the carburized case will be harder, will be more resistant to abrasive wear, and will exhibit improved fatigue properties compared with the uneffected core. These variations in properties are quite useful in applications where a hard, wear-resistant surface is needed and where a softer, more ductile core is required to prevent catastrophic failure of the component.

Carburizing methods include:

- Gas carburizing
- Vacuum carburizing

Table 2 Chemical compositions of through-hardening gear steels

Steel	Composition, %					
	C	Mn	Ni	Cr	Mo	Other
Carbon steels						
1035	0.32–0.38	0.60–0.90	. . .	. . .	. . .	(a)(b)
1040	0.37–0.44	0.60–0.90	. . .	. . .	. . .	(a)(b)
1045	0.43–0.50	0.60–0.90	. . .	. . .	. . .	(a)(b)
1050						
Free-cutting (resulfurized) carbon steels						
1137	0.32–0.39	1.35–1.65	. . .	. . .	. . .	0.08–0.13 S
1141	0.37–0.45	1.35–1.65	. . .	. . .	. . .	0.08–0.13 S
1144	0.40–0.48	1.35–1.65	. . .	. . .	. . .	0.24–0.33 S
Alloy steels						
1340	0.38–0.43	1.60–1.90	. . .	. . .	. . .	(c)(d)
3140	0.38–0.43	0.70–0.90	1.10–1.40	0.55–0.75	. . .	(c)(d)
4042	0.40–0.45	0.70–0.90	. . .	. . .	0.20–0.40	(c)(d)
4130	0.28–0.33	0.40–0.60	. . .	0.80–1.10	0.15–0.25	(c)(d)
4140	0.38–0.43	0.75–1.00	. . .	0.80–1.10	0.15–0.25	(c)(d)
4142	0.40–0.45	0.75–1.00	. . .	0.80–1.10	0.15–0.25	(c)(d)
4145	0.41–0.48	0.75–1.00	. . .	0.80–1.10	0.15–0.25	(c)(d)
4150	0.48–0.53	0.75–1.00	. . .	0.80–1.10	0.15–0.25	(c)(d)
4340	0.38–0.43	0.60–0.80	1.65–2.00	0.70–0.90	0.20–0.30	(c)(d)
4350	0.48–0.53	0.60–0.80	1.65–2.00	0.70–0.90	0.20–0.30	(c)(d)
5140	0.38–0.43	0.70–0.90	. . .	0.70–0.90	. . .	(c)(d)
6145	0.41–0.48	0.70–0.90	. . .	0.80–1.10	. . .	(c)(d) 0.15 V min
8640	0.38–0.43	0.75–1.00	0.40–0.70	0.40–0.60	0.15–0.25	(c)(d)
8740	0.38–0.43	0.75–1.00	0.40–0.70	0.40–0.60	0.20–0.30	(c)(d)
9840	0.38–0.43	0.70–0.90	0.85–1.15	0.70–0.90	0.08–0.15	(c)(d)

(a) 0.04 P max, 0.05 S max. (b) 0.10–0.60 Si. (c) 0.035 P max, 0.04 S max (d) 0.15–0.35 Si

- Plasma carburizing
- Salt bath carburizing
- Pack carburizing

These methods introduce carbon by the use of gas (atmospheric-gas, plasma, and vacuum carburizing), liquids (salt bath carburizing), or solid compounds (pack carburizing). All of these methods have advantages and limitations, but gas carburizing is used most often for large-scale production because it can be accurately controlled and involves a minimum of special

Table 3 Mechanical properties of selected gear steels

		Tensile strength		Yield strength				
Steel	Condition	MPa	ksi	MPa	ksi	Elongation, in 50 mm, %	Reduction in area, %	Hardness, HB
Carbon steel bar(a)								
1015	Hot rolled	345	50	190	27.5	28	50	101
	Cold drawn	385	56	325	47	18	40	111
1018	Hot rolled	400	58	220	32	25	50	116
	Cold drawn	440	64	370	54	15	40	126
1020	Hot rolled	380	55	205	30	25	50	111
	Cold drawn	420	61	350	51	15	40	121
1022	Hot rolled	425	62	235	34	23	47	121
	Cold drawn	475	69	400	58	15	40	137
1025	Hot rolled	400	58	220	32	25	50	116
	Cold drawn	440	64	370	54	15	40	126
1040	Hot rolled	525	76	290	42	18	40	149
	Cold drawn	585	85	490	71	12	35	170
1045	Hot rolled	565	82	310	45	16	40	163
	Cold drawn	625	91	530	77	12	35	179
	Annealed, cold drawn	585	85	505	73	12	45	170
	Spheroidized annealed, cold drawn	650	94	500	72.5	10	40	192
1117	Hot rolled	425	62	235	34	23	47	121
	Cold drawn	475	69	400	58	15	40	137
1118	Hot rolled	450	65	250	36	23	47	131
	Cold drawn	495	72	420	61	15	40	143
Low-alloy steels(b)								
4130	Normalized at 870 °C (1600 °F)	670	97	435	63	25.5	59.5	197
	Annealed at 865 °C (1585 °F)	560	81	460	67	21.5	59.6	217
	Water quenched from 855 °C (1575 °F) and tempered at 540 °C (1000 °F)	1040	151	979	142	18.1	63.9	302
4140	Normalized at 870 °C (1600 °F)	1020	148	655	95	17.7	46.8	302
	Annealed at 815 °C (1500 °F)	655	95	915	60	25.7	56.9	197
	Water quenched from 845 °C (1550 °F) and tempered at 540 °C (1000 °F)	1075	156	986	143	15.5	56.9	311
4150	Normalized at 870 °C (1600 °F)	1160	168	731	106	11.7	30.8	321
	Annealed at 830 °C (1525 °F)	731	106	380	55	20.2	40.2	197
	Oil quenched from 830 °C (1525 °F) and tempered at 540 °C (1000 °F)	1310	190	1215	176	13.5	47.2	375
4340	Normalized at 870 °C (1600 °F)	1282	186	862	125	12.2	36.3	363
	Annealed at 810 °C (1490 °F)	745	108	470	68	22.0	50.0	217
	Oil quenched from 800 °C (1475 °F) and tempered at 540 °C (1000 °F)	1207	175	1145	166	14.2	45.9	352
5140	Normalized at 870 °C (1600 °F)	793	115	470	68	22.7	59.2	229
	Annealed at 830 °C (1525 °F)	570	83	290	42	28.6	57.3	167
	Oil quenched from 845 °C (1550 °F) and tempered at 540 °C (1000 °F)	972	141	841	122	18.5	58.9	293
8620	Normalized at 915 °C (1675 °F)	635	92	360	52	26.3	59.7	183
	Annealed at 870 °C (1600 °F)	540	78	385	56	31.3	62.1	149
8630	Normalized at 870 °C (1600 °F)	650	94	425	62	23.5	53.5	187
	Annealed at 845 °C (1550 °F)	565	82	370	54	29.0	58.9	156
	Water quenched from 845 °C (1550 °F) and tempered at 540 °C (1000 °F)	931	135	850	123	18.7	59.6	269
8740	Normalized at 870 °C (1600 °F)	931	135	605	88	16.0	47.9	269
	Annealed at 815 °C (1500 °F)	696	101	415	60	22.2	46.4	201
	Oil quenched from 830 °C (1525 °F) and tempered at 540 °C (1000 °F)	1225	178	1130	164	16.0	53.0	352
9310	Normalized at 890 °C (1630 °F)	910	132	570	83	18.8	58.1	269 HRB
	Annealed at 845 °C (1550 °F)	820	119	450	65	17.3	42.1	241 HRB
	Aged sheet 6 mm (0.25 in.)	2169	315	2135	310	7.7	35	55.1 HRC

(a) All values are estimated minimum values; type 1100 series steels are rated on the basis of 0.10% max Si or coarse-grain melting practice; the mechanical properties shown are expected minimums for the sizes ranging from 19 to 31.8 mm (0.75 to 1.25 in.). (b) Most data are for 25 mm (1 in.) diam bar.

handling. More detailed information on various carburizing methods can be found in Chapter 9, "Carburizing."

The core carbon content of carburized gears is usually within the range of 0.10 to 0.25%. A lower carbon content is usually used to obtain maximum ductility, and a higher carbon content is used to obtain maximum core strength. Some representative SAE-AISI carburizing steels used for gears include:

- Plain carbon steels: 1015, 1018, 1020, 1022, and 1025
- Free-machining steels: 1117 and 1118
- Alloy steels: 4020, 4026, 4118, 4320, 4620, 4820, 5120, 8620, 8720, and 9310

Many other standard (SAE-AISI) and proprietary carburizing steels are also available.

The nickel-bearing carburizing steels are used chiefly where exceptional core toughness combined with the highest degree of wear resistance and greatest surface compressive strength is required. These steels include the nickel-molybdenum steels (4600 and 4800 series) and the nickel-chromium-molybdenum steels (4300, 8600, 8700, and 9300 series). The carbon-molybdenum steels (4000 series) are used where exceptional toughness and good resistance to temper embrittlement are required.

Another advantage of the more highly alloyed steels is the ability of heavy sections to harden more completely. This greater hardenability promotes better core strength properties than can be achieved with shallow hardening steels quenched in the same size section.

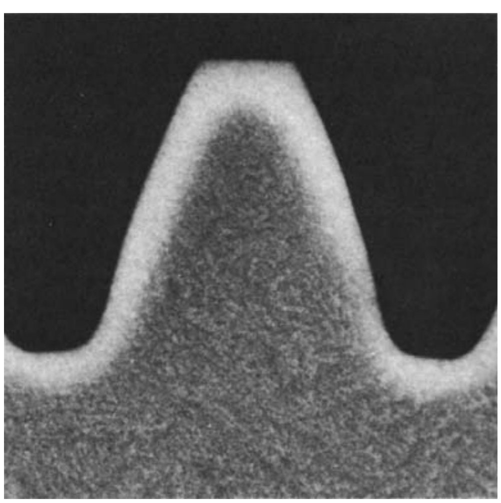


Fig. 1 Uniform case depth on a 40-tooth gear made from Fe-0.16C-0.6Mn-0.37Si-1.65Cr-3.65Ni steel that was produced by ion carburizing

Nitriding Steels. Nitriding is a surface-hardening heat treatment that introduces nitrogen into the surface of steel at a temperature range of 500 to 550 °C (930 to 1020 °F) while it is in the ferritic condition. Thus, nitriding is similar to carburizing in that surface composition is altered, but different in that nitrogen is added into ferrite instead of austenite. Because nitriding does not involve heating into the austenite phase field and a subsequent quench to form martensite, nitriding can be accomplished with a minimum of distortion and with excellent dimensional control. Process methods for nitriding include gas, liquid (salt bath), and plasma (ion) nitriding. Details on these processes can be found in Chapter 10, "Nitriding."

Nitriding steels can be used in many gear applications where a hard, wear-resistant case, good fatigue strength, low notch sensitivity, and some degree of corrosion resistance are desired. In addition, nitriding steels make it possible to surface harden the teeth of large gears having thin sections that might be impractical to carburize and quench.

Nitrided gears are relatively free from wear up to the load at which surface failure occurs, but at this load they become badly crushed and pitted. Thus, nitrided gears are generally not suitable for applications where overloads are likely to be encountered.

Nitrided steels are generally medium-carbon (quenched-and-tempered) steels that contain strong nitride-forming elements such as aluminum, chromium, vanadium, and molybdenum. The most significant hardening is achieved with a class of alloy steels (Nitralloy steels as described below) that contain about 1% Al. When these steels are nitrided, the aluminum forms AlN particles, which strain the ferrite lattice and create strengthening dislocations.

Table 4 lists chemical compositions of Nitralloy gear steels. Nitralloy N, a nickel-bearing (3.5% Ni) nitriding steel, is a precipitation-hardening alloy that attains a core strength and hardness after nitriding that are considerably in excess of its original properties. Both Nitralloy N and Nitralloy 135M are outstanding for heavy-duty gears that are highly stressed. Little change in tensile strength of nitriding steels occurs if the tempering temperature used for treating the core is at or above the nitriding temperature. However, because of the increased hardness of the case, the elongation, ductility, and impact strength of both alloys are considerably reduced after tempering, though not to the same extent; Nitralloy N develops a tougher and softer surface and a stronger core than Nitralloy 135M.

Any of the SAE-AISI steels that contain nitride-forming elements, such as chromium, vanadium, or molybdenum, can also be nitrided. The steels most commonly nitrided are 4140, 4340, 6140, and 8740. In some applications, the 0.50% C grades are also used.

Through-Hardening Steels

By virtue of their higher carbon content, through-hardening steel gears possess greater core strength than carburized gears. They are not, however, as ductile or as resistant to surface compressive stresses and wear as case-hardened gears. Hardness of gear surfaces may vary from 300 to 575 HB. Through-hardened steels may also be effectively surface hardened by induction heating or by flame hardening.

Typical of the relatively shallow-hardening carbon steel gear materials are SAE-AISI types 1035, 1040, 1045, 1050, 1137, 1141, 1144, and 1340. These steels are water-hardening, but not deep-hardening types that are suitable for gears requiring only a moderate degree of strength and impact resistance.

In general, the more highly alloyed through-hardening steels harden more completely when quenched in heavy sections. This greater hardenability provides greater strength than can be attained with shallow-hardening steels quenched in the same size section.

Typical of the low-alloy, medium-to-deep hardening gear materials are (in order of increasing hardenability): 4042, 5140, 8640, 3140, 4140, 8740, 6145, 9840, and 4340. These steels, as well as many other alloy steels with the proper hardenability characteristics and a carbon content of 0.35 to 0.50%, are suitable for gears requiring medium-to-high wear resistance and high load-carrying capacity. Other standard (SAE-AISI) and proprietary through-hardening steels are also available.

When selecting a through-hardening steel, it should be considered that a higher carbon and alloy content is accompanied by greater strength and hardness (but lower ductility) of the surface and the core. Fully hardened and tempered medium-carbon alloy steels possess an excellent combination of strength and toughness at room temperature and at lower temperatures. However, toughness can be substantially decreased by temper embrittlement by slow cooling through the temperature range of 450 to 540 °C (850 to 1000 °F), or by holding or tempering in this range. Because of their good hardenability and immunity to temper brittleness, molybdenum steels have been widely used for gears requiring good toughness at room and low temperatures.

Gear Steel Requirements

Some of the more important requirements for gear steels are their:

- Processing characteristics (for example, hardenability and machinability)
- Response to heat treatment. This subject is addressed in Chapters 8 through 12 which cover through-hardening, carburizing, nitriding, carbonitriding, and induction and flame hardening, respectively.
- Resistance to tooth bending fatigue—both low-cycle ($\leq 10^5$ cycles to failure) and high-cycle ($>10^5$ cycles to failure) fatigue. Because carburized steels for high-performance gear applications are subjected to cyclic loading, this is one of the most important properties or measure of gear performance. The section on "Bending Fatigue Strength of Carburized Steels" deals with this subject in detail.

Table 4 Nominal chemical compositions for aluminum-containing low-alloy steels commonly gas nitrided

Steel			Composition, %							
SAE	AMS	Nitralloy	C	Mn	Si	Cr	Ni	Mo	Al	Se
...	...	G	0.35	0.55	0.30	1.2	...	0.20	1.0	...
7140	6470	135M	0.42	0.55	0.30	1.6	...	0.38	1.0	...
...	6475	N	0.24	0.55	0.30	1.15	3.5	0.25	1.0	...
...	...	EZ	0.35	0.80	0.30	1.25	...	0.20	1.0	0.20

- Resistance to surface-contact (pitting) fatigue. This subject is addressed in Chapter 2, "Gear Tribology and Lubrication."
- Resistance to rolling contact fatigue
- Resistance to wear. This subject is briefly discussed in this chapter, but more detailed information on adhesive wear, abrasive wear, and scuffing of gears can be found in Chapter 2.
- Their hot hardness
- Their bending strength and bend ductility
- Their toughness, both impact toughness and fracture toughness

Each of these will be described in subsequent sections.

Processing Characteristics of Gear Steels

Hardenability refers to the ability of a steel to be transformed partially or completely from austenite to martensite at a given depth when cooled under prescribed conditions. This definition reflects the empirical nature of steel hardenability, and, as discussed in Ref 2, many types of experiments have been devised to measure or describe the hardenability of various kinds of steel.

Martensite is the microstructure usually desired in quenched carbon and low-alloy steels. The cooling rate in a quenched part must be fast enough so that a high percentage of martensite is produced in critically stressed areas of the part. Higher percentages of martensite result in higher fatigue and impact properties after tempering.

Hardenability should not be confused with hardness as such or with maximum hardness. The maximum attainable hardness of any steel depends solely on carbon content. Also, the maximum hardness values that can be obtained with small test specimens under the fastest cooling rates of water quenching are nearly always higher than those developed under production heat-treating conditions, because hardenability limitations in quenching larger sizes can result in less than 100% martensite formation.

Basically, the units of hardenability are those of cooling rate—for example, degrees per second. These cooling rates, as related to the continuous-cooling-transformation behavior of the steel, determine the hardness and microstructural outcome of a quench. In practice, these cooling rates are often expressed as a distance, with other factors such as the thermal conductivity of steel and the rate of surface heat removal being held constant. Therefore, the terms Jominy distance (J) and ideal critical diameter (D_1) derived from the Jominy end-quench test can be used.

The hardenability of steel is governed almost entirely by the chemical composition (carbon and alloy content) at the austenitizing temperature and the austenite grain size at the moment of quenching. In some cases, the chemical composition of the austenite may not be the same as that determined by chemical analysis, because some carbide may be undissolved at the austenitizing temperature. Such carbides would be reflected in the chemical analysis, but because the carbides are undissolved in the austenite, neither their carbon nor alloy content can contribute to hardenability. In addition, by nucleating transformation products, undissolved carbides can actively decrease hardenability. This is especially important in high-carbon (0.50 to 1.10%) and alloy carburizing steels, which may contain excess carbides at the austenitizing temperature. Consequently, such factors as austenitizing temperature, time at temperature, and prior microstructure are sometimes very important variables when determining the basic hardenability of a specific steel composition. Certain ingot casting and hot reduction practices may also develop localized or periodic inhomogeneities within a given heat, further complicating hardenability measurements. The effects of all these variables are discussed in Ref 2.

Table 5 provides a qualitative rating of the hardenability of gear steels. Additional information on hardness and hardenability can be found in Chapter 9, "Carburizing."

Machinability. The term *machinability* is used to indicate the ease or difficulty with which a material can be machined to the size, shape, and desired surface finish. The terms *machinability index* and *machinability rating* are used as qualitative and relative measures of the machinability of a steel under specified conditions. There are no clear-cut or unambiguous meanings for these terms and no standard or universally accepted method of measuring machinability. Historically, machinability judgments have been based on one or more of the following criteria:

- *Tool life:* Measured by the amount of material that can be removed by a standard cutting tool under standard cutting conditions before tool performance becomes unacceptable or tool wear reaches a specified amount

- *Cutting speed:* Measured by the maximum speed at which a standard tool under standard conditions can continue to provide satisfactory performance for a specified period
- *Power consumption:* Measured by the power required to remove a unit volume of material under specified machining conditions
- Comparisons with a standard steel based on experience in machine shops
- Quality of surface finish
- Feeds resulting from a constant thrust force

Some of the test criteria are best suited to laboratory studies intended to elicit information about the effects of small changes in microstructure, composition, or processing history on machinability. Other types of tests are useful for studying the effects of geometry changes or cutting tool composition.

Figure 2 is a plot of cutting speed for one-hour tool life versus the Brinell hardness of various gear steels. This figure indicates a wide range correlation. For example, at a hardness range of 175 to 200 HB, the cutting speed for one-hour tool life varies from 23 to 43 m/min (75 to 140 surface feet per minute). The figure indicates decreased cutting speed for one-hour tool life as the hardness increases. In Table 6, the machinability of some of the more common gear steels is listed on the basis of good, fair, and poor machinability. With good machinability as a base, a fair rating would add 20 to 30% to the machining cost, and a poor rating would add 40 to 50%.

Bending Fatigue Strength of Carburized Steels

Bending fatigue of carburized steel components is a result of cyclic mechanical loading. The bending produces stresses, which are tensile at the surface, decrease with increasing distance into the component, and at some point become compressive. Such loading is a characteristic of the roots of gear teeth. Carburizing produces a high-carbon, high-strength surface layer, or high-strength case, on a low-carbon, low-strength interior or core and is therefore, an ideal approach to offset the high surface tensile stresses associated with bending. Thus when the design of a component maintains operating stress gradients below the fatigue strength of the case and core microstructures, excellent bending fatigue resistance is established.

There are, however, many alloying and processing factors that produce various microstructures, and therefore variable strength and fracture resistance, of the case regions of carburized

Table 5 Hardenability characteristics of commonly used gear steels

Common alloy steel grades	Common heat treat practice(a)	Hardenability
Noncarburizing grades		
1045	T-H, I-H, F-H	Low hardenability
4130	T-H	Marginal hardenability
4140	T-H, T-H&N, I-H, F-H	Fair hardenability
4145	T-H, T-H&N, I-H, F-H	Medium hardenability
8640	T-H, T-H&N, I-H, F-H	Medium hardenability
4340	T-H, T-H&N, I-H, F-H	Good hardenability in heavy sections
4150	I-H, F-H, T-H, T–H&N	Quench crack sensitive Good hardenability
4142	I-H, F-H, T-H&N	Used when 4140 exhibits Marginal hardenability
4350	T-H, I-H, F-H	Quench crack sensitive, excellent Hardenability in heavy sections
Carburizing grades		
1020	C-H	Very low hardenability
4118	C-H	Fair core hardenability
4620	C-H	Good case hardenability
8620	C-H	Fair core hardenability
4320	C-H	Good core hardenability
8822	C-H	Good core hardenability in heavy sections
3310	C-H	Excellent hardenability (in heavy sections) for all three grades
4820	C-H	
9310	C-H	

(a) C-H = Carburize harden, F-H = Flame harden, I-H = Induction harden, T-H = Through harden, T-H&N = Through harden then nitride. Source: American Gear Manufacturers Association steels. When applied surface stresses exceed the surface strength, surface fatigue crack initiation and eventual failure will develop. When the surface strength is adequate, depending on the steepness of the applied stress gradients in relationship to the case/core strength gradient, subsurface fatigue cracking may develop.

Bending fatigue performance of carburized steels can vary significantly. One study reported values of experimentally measured endurance limits ranging from 200 to 1930 MPa (29 to 280 ksi), with most values between 700 and 1050 MPa (100 and 152 ksi) (Ref 3). As will be described subsequently, this wide variation in fatigue performance is a result of variations in specimen design and testing, alloying, and processing interactions that produce large variations in carburized microstructures and the response of the microstructures to cyclic loading.

Bending Fatigue Testing

Data Presentation and Analysis. Most bending fatigue data for carburized steels are presented as plots of maximum stress, *S,* versus the number of cycles, *N,* to fracture for a specified stress ratio (*R*), which is the ratio of minimum (or compressive) stress to maximum tensile stress (R = min-S/max-S). Figure 3 shows an example of typical *S-N* plots for a series of carburized alloy steels (Ref 4). The *S-N* curve consists of two parts: a straight section with negative slope at low cycles and a horizontal section at high cycles. The horizontal line defines the fatigue limit or endurance limit, which is taken to be the maximum applied stress below which a material is assumed to be able to withstand an infinite number of stress cycles without failure. Pragmatically, the endurance limit is taken as the stress at which no failure occurs after a set number of cycles, typically on the order of 10 million cycles. The low-cycle portion of the *S-N* plot defines various fatigue strengths or the stresses to which the material can be subjected for a given number of cycles. The more cycles at a given strength, the better the low-cycle fatigue resistance of a material.

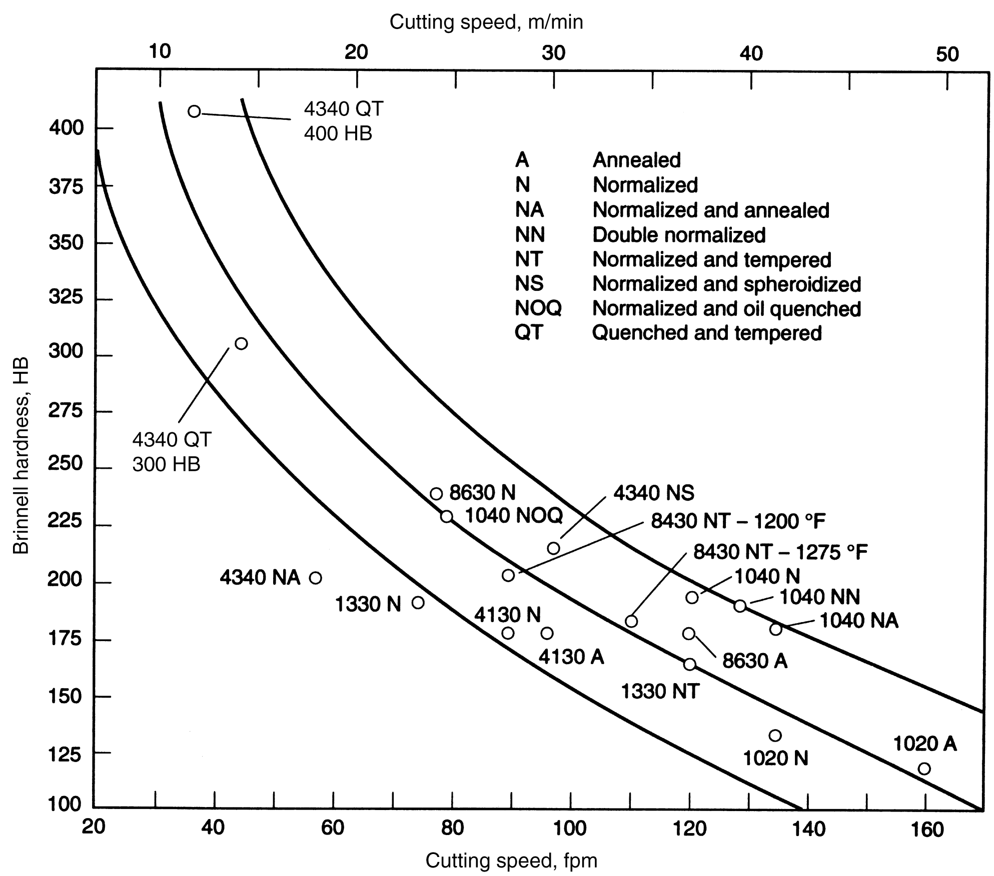


Fig. 2 Cutting speed for one-hour tool life versus Brinell hardness number for various through-hardened carbon and alloy steels

Analysis of bending fatigue behavior of carburized steels based on *S-N* curves represents a stress-based approach to fatigue and assumes that the carburized specimens deform nominally only in an elastic manner (Ref 5). This assumption is most valid at stresses up to the endurance limit and is useful when machine components are designed for high-cycle fatigue. However, as maximum applied stresses increase above the endurance limit, plastic strain becomes increasingly important during cyclic loading, and fatigue is more appropriately analyzed by a strain-based approach. In this approach, the total strain range is the sum of the applied elastic and plastic strains, and the strain amplitude is plotted as a function of the strain reversals required for failure at the various levels of strain.

According to the strain-based approach, low-cycle fatigue behavior is determined by plastic strains, while high-cycle fatigue behavior is determined by elastic strains. In particular, ductile materials with microstructures capable of sustaining large amounts of plastic deformation have better low-cycle fatigue resistance, while high-strength materials with high elastic limits and high yield strengths have better high-cycle fatigue resistance. Figure 4 shows the results of strain-based bending fatigue testing of uncarburized and carburized 4027 steel (Ref 6). The more ductile, low-strength uncarburized specimens show better fatigue resistance at low cycles than do the carburized specimens. The performance is reversed at high cycles, where the carburized specimens with their high-strength surfaces show better fatigue resistance, especially those specimens with the deeper cases.

Specimen Design. Many types of specimens have been used to evaluate bending

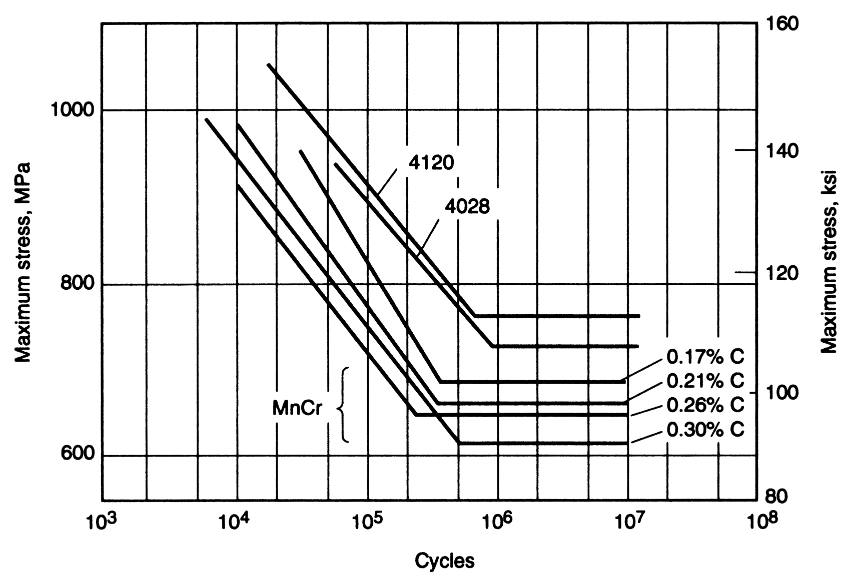
Table 6 Machinability of commonly used gear materials. Refer also to the text describing this table

Material	Remarks
Low-carbon carburizing steel grades(a)	
1020	Good machinability, as rolled, as forged, or normalized.
4118 4620 8620 8822	Good machinability, as rolled, or as forged. However, normalized is preferred. Inadequate cooling during normalizing can result in gummy material, reduced tool life and poor surface finish. Quench and temper as a prior treatment can aid machinability. The economics of the pretreatments must be considered.
3310 4320 4820 9310	Fair to good machinability if normalized and tempered, annealed or quenched and tempered. Normalizing without tempering results in reduced machinability.
Medium-carbon through-hardening steel grades(a)	
1045 1141 1541	Good machinability if normalized.
4130 4140 4142	Good machinability if annealed, or normalized and tempered to approximately 255 HB or quenched and tempered to approximately 321 HB. Over 321 HB, machinability is fair. Above 363 HB, machinability is poor. Inadequate (slack) quench with subsequent low tempering temperature may produce a part which meets the specified hardness, but produces a mixed microstructure which results in poor machinability.
4145 4150 4340 4345 4350	Remarks for medium carbon alloy steel (above) apply. However, the higher carbon results in lower machinability. Sulfur additions aid the machinability of these grades. 4340 machinability is good up to 363 HB. The higher carbon level in 4145, 4150, 4345, and 4350 makes them more difficult to machine and should be specified only for heavy sections. Inadequate (slack) quench can seriously affect machinability in these steels.
Other gear materials	
Gray Irons	Gray cast irons have good machinability. Higher strength gray cast irons [above 345 MPa (50 ksi) tensile strength] have reduced machinability.
Ductile Irons	Annealed or normalized ductile cast iron has good machinability. The "as cast" (not heat treated) ductile iron has fair machinability. Quenched and tempered ductile iron has good machinability up to 285 HB and fair machinability up to 352 HB. Above 352 HB, machinability is poor.
Gear Bronzes and Brasses	All gear bronzes and brass have good machinability. The very high strength heat treated bronzes [above 760 MPa (110 ksi) tensile strength] have fair machinability.
Austenitic Stainless Steel	All austenitic stainless steel grades only have fair machinability. Because of work hardening tendencies, feeds and speeds must be selected to minimize work hardening.

(a) Coarse grain steels are more machinable than fine grain. However, gear steels are generally used in the fine grain condition since mechanical properties are improved, and distortion during heat treatment is reduced. Increasingly cleaner steels are now also being specified for gearing. However, if sulfur content is low, less than 0.015%, machinability may decrease appreciably. Source: American Gear Manufacturers Association fatigue in carburized steels. Rotating beam, unnotched four-point bend, notched four-point bend, and cantilever beam specimens have all been used, and they have in common a maximum applied surface tensile stress and decreasing tensile stress with increasing distance into the specimen. Axial fatigue testing of carburized specimens, which applies the maximum tensile stresses uniformly over the cross section of a specimen, invariably results in subsurface initiation and propagation of fatigue cracks in the core of carburized specimens, and therefore it does not permit evaluation of the resistance of case microstructures to fatigue.

Brugger was the first to use cantilever bend specimens to evaluate the fracture and fatigue of carburized steels (Ref 7). Figure 5 shows a cantilever bend specimen that has evolved from the Brugger specimen. The radius between the change in section simulates the geometry at the root of gear teeth and results in maximum applied surface stresses just where the cross section begins to increase, as shown in Fig. 6 (Ref 8). An important feature of this specimen is the rounding of the corners of the beam section. If the corners are square, the carbon introduced into the corner surfaces cannot readily diffuse into the interior of the specimen. As a result, the corner microstructures may have significantly elevated levels of retained austenite and coarse carbide structures—both microstructural features that influence bending fatigue resistance (Ref 8).

Mean Stress and Stress Ratio. The maximum applied surface tensile stress is the testing parameter plotted in *S-N* curves that characterize bending fatigue. However, the applied stress ranges between maximum and minimum values during a fatigue cycle, and two other parameters, mean stress and stress ratio, are important for the characterization of fatigue. The mean stress is the algebraic average of the maximum and minimum stresses in a cycle, and as discussed, the stress ratio, *R,* is the ratio of the minimum stress to the maximum stress in a cycle. Thus *R* values may range from –1, for fully reversed loading that ranges between equal maximum tensile and compressive stresses, to positive values where



SAE steel grade	Composition, wt% C	Mn	Cr	Mo
4120	0.19	1.03	0.52	0.19
4028	0.28	0.80	...	0.25
PS59	0.17	1.20	0.90	...
EX60	0.21	1.18	0.90	...
PS61	0.26	1.18	0.90	...
EX62	0.30	1.12	0.88	...

Fig. 3 Typical maximum stress (*S*) vs. number of cycles (*N*) bending fatigue plots for 6 carburized steels. $R = -1$. Source: Ref 4

the stress is cycled between two tensile values (Ref 5). Much of the bend testing of cantilever specimens described subsequently is performed with *R* values of 0.1 in order to preserve details of the fracture surface.

Figure 7 shows a typical allowable-stress diagram that plots fatigue strength versus mean stress for a given material (Ref 9). The diagram shows that the most severe condition for fatigue is for fully reversed testing with $R = -1.0$. As the mean stress increases, the fatigue strength in terms of maximum applied stress increases, but the allowable stress range decreases. Zurn and Razim (Ref 10) have examined the effect of notch severity and retained austenite on allowable-stress/mean-stress diagrams of carburized steels, and they conclude that carburizing, relative to the use of through-hardened steels, is especially effective for parts with sharp notches. In the absence of notches, carburizing is most suitable for parts subjected to fatigue loading at low values of mean stress.

The testing of actual machine components is another important approach to the fatigue evaluation of carburized steels. An example of component testing is the bending fatigue testing of single teeth in gears (Ref 11). Gears are fabricated, carburized, and mounted in a fixture so that one tooth at a time is subjected to cyclic loading. More recently, identically carburized specimens of the same steel were subjected to cantilever bend and single tooth bending fatigue testing (Ref 12). The

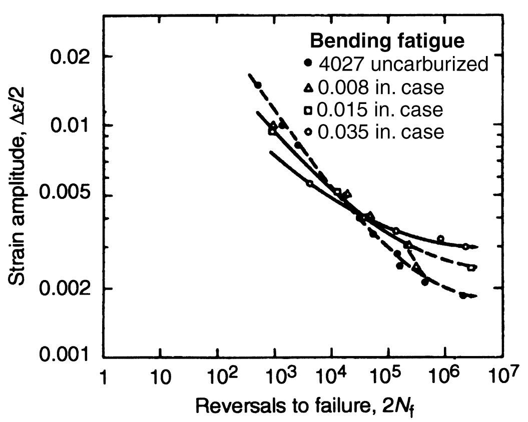


Fig. 4 Strain amplitude vs. reversals to failure for uncarburized (solid symbols) and carburized (open symbols) 4027 steel (0.80% Mn, 0.28% Si, 0.27% Mo). Source: Ref 6

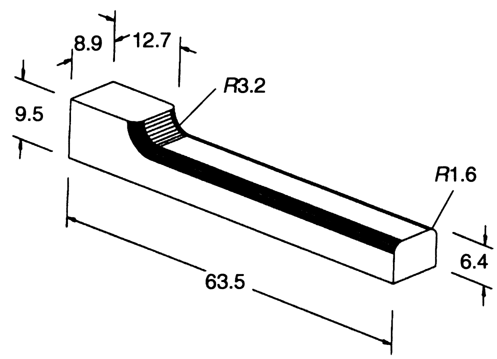


Fig. 5 Example of a cantilever specimen used to evaluate bending fatigue of carburized steels. Specimen edges are rounded and maximum stress is applied at the location shown in Fig. 6. Dimensions in millimeters

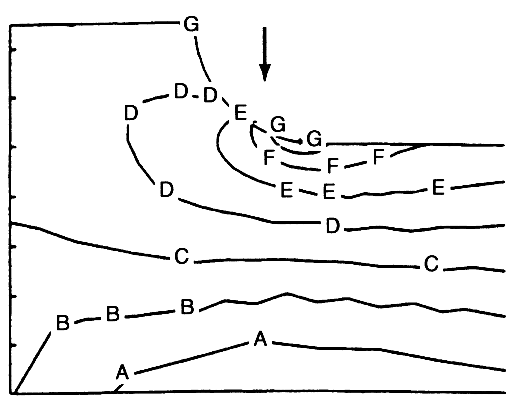


Fig. 6 Location of maximum stress on the cantilever bend specimen shown in Fig. 5 as determined by finite element modeling

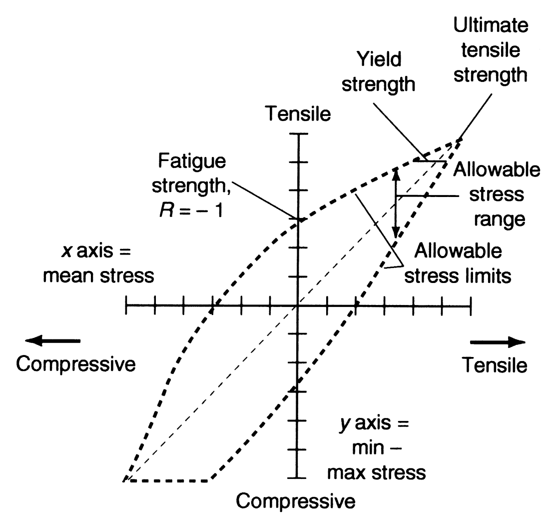


Fig. 7 Typical allowable-stress-range diagram. Source: Ref 9

mechanisms of fatigue failure, based on fracture surface examination, were found to be the same, but the single tooth testing showed higher levels of fatigue resistance than did the cantilever testing, a result attributed to the higher surface compressive stresses that were measured in the gear tooth specimens. More detailed information on testing of gears can be found in Chapter 15, "Mechanical Testing."

Stages of Fatigue and Fracture

Bending fatigue fractures of carburized steels consist of well-defined stages of crack initiation, stable crack propagation, and unstable crack propagation. The fracture sequence is strongly influenced by the gradients in strength, microstructure, and residual stress that develop in carburized steels. Figure 8 shows a series of SEM fractographs that characterize the typical fracture sequence of a direct-quenched carburized steel. The cantilever bend specimen from which the fractographs of Fig. 8 were taken was a 4320 steel carburized to a 1 mm (0.03937 in.) case depth at 927 °C (1700 °F), quenched from 850 °C (1560 °F) into oil at 65 °C (150 °F), and tempered at 150 °C (300 °F) for 1 h. The specimen was tested in bending fatigue with an *R* value of 0.1 (Ref 13). Figures 8(a) and (b) show a low-magnification overview of the initiation, stable propagation, and unstable fracture surfaces, and Fig. 8(c) shows the intergranular initiation and transgranular stable crack propagation zones of the fracture at a higher magnification.

Intergranular cracking at prior-austenite grain boundaries is an almost universal fracture mode in the high-carbon case of direct-quenched carburized steels (Ref 13). Not only do the fatigue cracks initiate by intergranular cracking, but also the unstable crack propagates largely by intergranular fracture until it reaches the lower-carbon portion of the case, where ductile fracture becomes the dominant fracture mode. In fact, sensitivity of the case microstructures to intergranular fracture makes possible the quantitative characterization of the size and shape of the stable fatigue crack, as shown in Fig. 8(b). The transition from the transgranular fracture of the stable crack to the largely intergranular fracture of the unstable fracture is identified by the dashed line.

A replica study of carburized specimens subjected to incrementally increasing stresses showed that surface intergranular cracks initiated when the applied stresses exceeded the endurance limits (Ref 13). Thus, it appears that in direct-quenched carburized specimens, intergranular cracks are initiated as soon as the applied surface bending stress reaches a level sufficient to exceed the surface compressive residual stress and the cohesive strength of the prior-austenite grain boundary structures. The surface intergranular cracks are shallow, typically approximately two to four austenite grains, and are arrested, perhaps because of a plastic zone smaller than the grain size at the tip

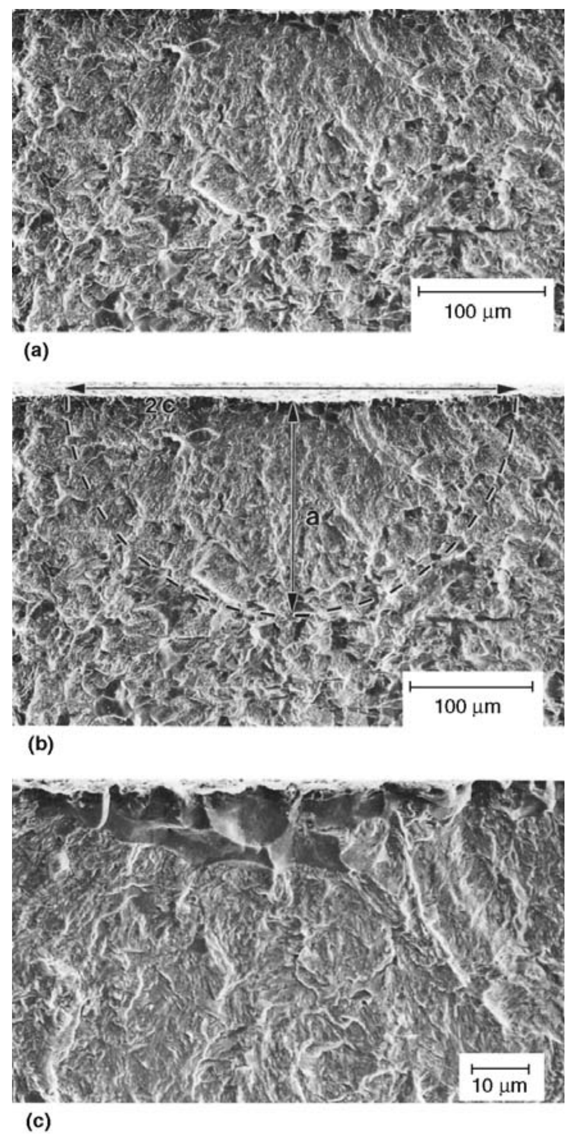


Fig. 8 Fatigue fracture in gas-carburized and modified 4320 steel. (a) Overview of initiation, stable crack propagation, and unstable crack propagation. (b) Same area as shown in (a), but with extent of stable crack indicated by dashed line. (c) Higher magnification of intergranular initiation and transgranular stable crack propagation areas. SEM Fractographs. Source: Ref 13

of the sharp intergranular cracks, and the fact that strain-induced transformation of retained austenite in the plastic zone ahead of the crack introduces compressive stresses (Ref 14, 15). The fatigue crack then propagates in a transgranular mode, and when the stable crack reaches critical size, as defined by the fracture toughness, unstable fracture occurs.

The initiation and stable crack zones of carburized steels are quite small and are often difficult to identify. Figure 9 based on the measurement of critical crack sizes in a number of direct-quenched carburized 4320 steels, shows that the size of the unstable cracks ranges from 0.170 to 0.230 mm (0.00669 to 0.00906 in.), and that the cracks, therefore, become unstable well within the high-carbon portion of the carburized specimens. The small critical crack sizes are consistent with the low fracture toughness of high-carbon steel low-temperature-tempered (LTT) microstructures susceptible to intergranular fracture (Ref 16). When the critical crack sizes and the stresses at which the cracks become unstable are used to calculate the fracture toughness of the case microstructures of carburized steels (Ref 13), the results show good agreement with the range of fracture toughness, 15 to 25 MPa$\sqrt{\text{m}}$, that has been measured from through-hardened specimens with high-carbon LTT martensitic microstructures (Ref 16). Table 7 shows the data used to calculate the various case fracture toughness values in gas-carburized 4320 steel and the fracture toughness values calculated according to three different fracture toughness equations (Ref 13):

$$K_{\text{IC}} = \frac{1.2\sigma_a\sqrt{a\pi}}{Q} \qquad \text{(Eq 1)}$$

$$K_{\text{IC}} = \frac{\left[1.2\sigma_a + 0.683\left(\dfrac{\partial\sigma}{\partial_x}\right)a\right]\sqrt{a\pi}}{Q} \qquad \text{(Eq 2)}$$

and

$$K_{\text{IC}} = \frac{M\sigma_a\sqrt{a\pi}}{Q} \qquad \text{(Eq 3)}$$

where

$$Q = \phi^2 - 0.212\left(\frac{\sigma_a}{\sigma_{ys}}\right)^2$$

with ϕ the aspect ratio of crack depth (a) and crack length (c) such that:

$$\phi^2 = 1 + 1.464\left(\frac{a}{c}\right)^{1.65}$$

The unstable crack that proceeds through the high- and medium-carbon martensitic portions of the case may be arrested when the sensitivity to intergranular fracture decreases at a case car-

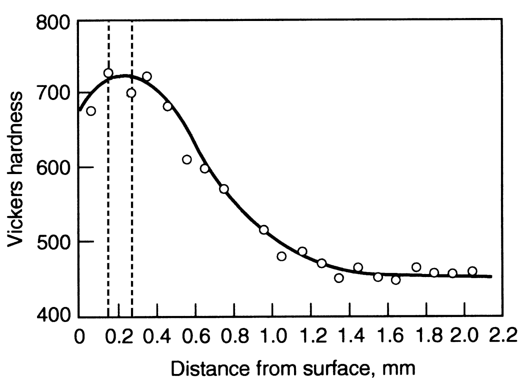


Fig. 9 Hardness vs. distance from the surface of direct-cooled gas-carburized SAE 4320 steel. Superimposed on the hardness profile is the range of critical depths (vertical dashed band) at which stable fatigue cracks became unstable in bending fatigue of similarly processed steels. Source: Ref 13

Table 7 Fracture toughness results for carburized SAE 4320 bending fatigue specimens

Max stress, MPa	Cycles to failure	Depth, a, μm	Width, c, μm	a/c	M	K_{Ic}, MPa$\sqrt{\text{m}}$, Eq 21	K_{Ic}, MPa$\sqrt{\text{m}}$, Eq 22	K_{Ic}, MPa$\sqrt{\text{m}}$, Eq 23
1370	6,400	175	388	0.45	0.87	30	29	22
	16,900	170	355	0.48	0.86	29	28	21
	15,300	200	210	0.95	0.78	18	18	12
1285	17,400	230	300	0.77	0.81	22	22	15
	18,700	230	295	0.78	0.81	22	22	15
1235	34,100	210	300	0.70	0.81	22	22	15
1160	21,700	230	295	0.78	0.81	19	19	13
	32,300	220	295	0.75	0.81	19	19	14

Equations used to determine the information in this table are defined in text. Source: Ref 13 bon content between 0.5 and 0.6% (Ref 17). The continued application of cyclic loading at this point then may cause a secondary stage of stable fatigue crack propagation, characterized by transgranular fracture, resolvable fatigue striations, and secondary cracking (Ref 12). This stage of low-cycle, high-strain fatigue is short and gives way to ductile overload fracture of the core.

Intergranular Fracture and Fatigue

Intergranular fracture at the prior-austenite grain boundaries of high-carbon case microstructures dominates bending fatigue crack initiation and unstable crack propagation of direct-quenched carburized steels. The intergranular cracking may be associated with other microstructural features, such as the surface oxides generated by gas carburizing, but it generally extends much deeper into a carburized case than the oxide layer. Several studies have documented bending fatigue crack initiation by intergranular fracture even in the absence of surface oxidation, where, for example, the oxidized surface has been removed by chemical or electropolishing (Ref 18) or no oxidation is present because the specimens were vacuum or plasma carburized (Ref 19).

Figure 10 shows an example of intergranular fatigue crack initiation in a direct-quenched specimen of gas-carburized type 8719 steel (Ref 20). There is a shallow zone of surface oxidation, about 10 μm in depth, but the intergranular cracking extends much deeper into the specimen. Figure 11 shows extensive intergranular cracking in the unstable crack propagation zone in the case of a direct-quenched, gas-carburized 4320 steel. Auger electron spectroscopy (Fig. 12) shows that such intergranular fracture surfaces have higher concentrations of phosphorus and carbon, in the form of cementite, than do transgranular fracture surfaces removed from prior-austenite grain boundaries (Ref 21, 22). Thus, the brittle intergranular fracture that occurs in stressed high-carbon case microstructures of carburized steels is associated with the combined presence of segregated phosphorus and cementite at prior-austenite grain boundaries. These grain boundary structures are present in as-quenched specimens and do not require tempering for cementite formation, as is typical in the intergranular mode of tempered martensite embrittlement in medium-carbon steels (Ref 23). This embrittlement, termed *quench embrittlement,* is found in quenched steels with carbon contents as low as 0.6% (Ref 24). There is evidence that phosphorus segregation stimulates the formation of the grain boundary cementite (Ref 22).

The higher the phosphorus content of a carburized steel, the lower its bending fatigue resistance and case fracture toughness. Figure 13 shows *S-N* curves for a series of gas-carburized and direct quenched modified 4320 steels with systematic variations in phosphorus content from 0.031 to 0.005% (Ref 22). Endurance limits and low-cycle fatigue resistance increase with decreasing phosphorus content, but little difference is noted between the performance of the 0.005 and 0.017% phosphorus specimens. All of the specimens, even those with the lowest phosphorus content, failed by intergranular initiation of fatigue cracks.

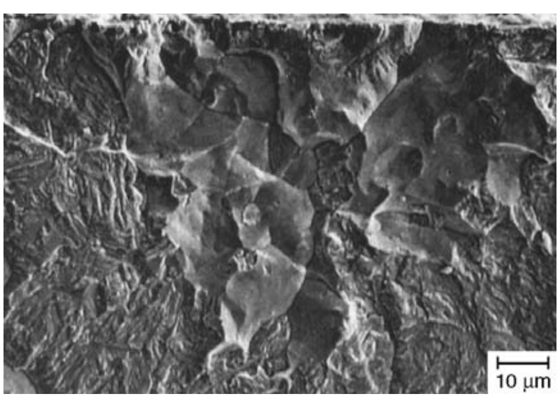


Fig. 10 Intergranular bending fatigue crack initiation at the surface of a gas-carburized and direct-cooled SAE 8719 steel specimen. Source: Ref 20

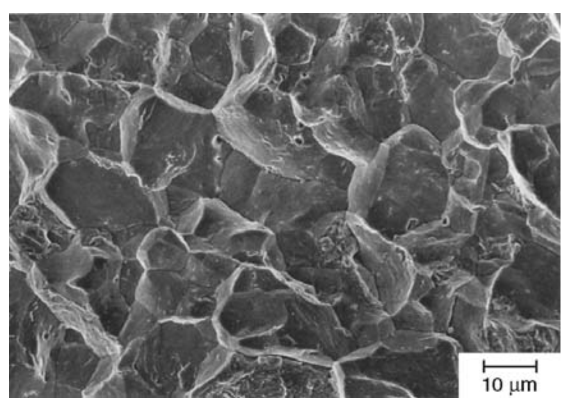


Fig. 11 Intergranular fracture in case unstable crack propagation zone in gas-carburized and direct-cooled SAE 4320 steel

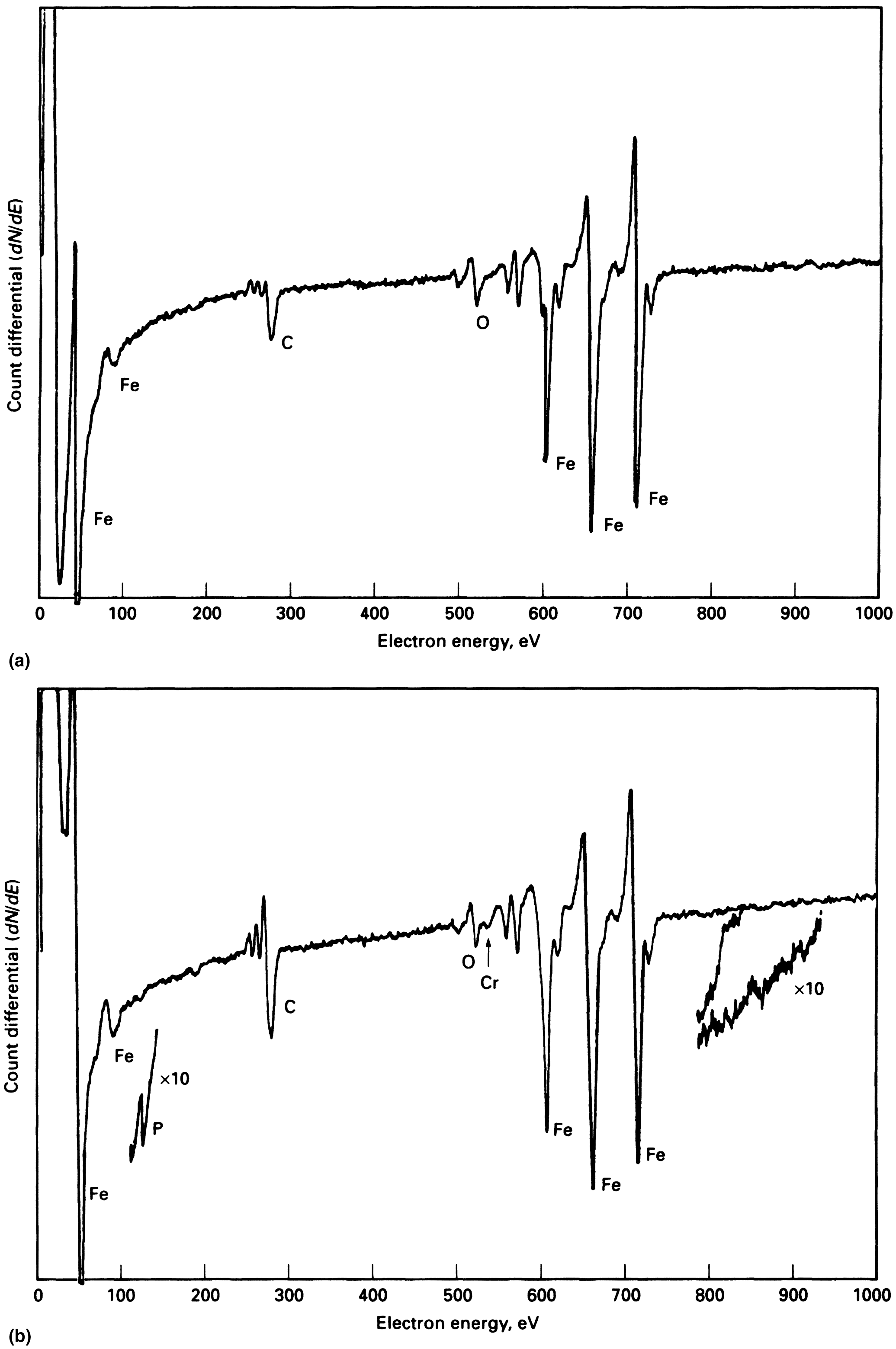


Fig. 12 Auger electron spectra from case fracture surfaces of carburized 8620 steel. (a) From transgranular fracture surface. (b) From intergranular fracture surface. No phosphorus peak is detectable in the spectrum produced from the transgranular fracture, and a small phosphorus peak, clearly shown by the 10× magnification, is produced from the intergranular fracture surface. These observations are consistent with other investigations that show that phosphrous segregates to austenite grain boundaries during austenitizing. Source: Ref 21

The bending endurance limits of gas-carburized specimens in which fatigue is initiated by intergranular fracture typically range between 1050 and 1260 MPa (152 and 183 ksi) (Ref 25, 26). This range is based on studies of cantilever bend specimens with good surface finish, rounded specimen corners, nominal amounts of surface oxidation, and loading at $R = 0.1$. Variations within this range may be due to variations in austenitic grain size, inclusion contents, retained austenite content, or residual stresses, as discussed subsequently. Nevertheless, the common mechanism of bending fatigue crack initiation of direct-quenched specimens is intergranular fracture at embrittled grain boundaries in a microstructure of LTT martensite and retained austenite.

Carburized steels with high nickel content do not appear to be as susceptible to intergranular cracking as steels with low nickel content (Ref 27). Also, major changes in the case microstructures of carburized steel, such as those produced by reheating, result in bending fatigue crack initiation sites other than embrittled prior-austenite grain boundaries. Microstructural conditions that produce fracture initiation other than by intergranular cracking are also described herein.

Inclusions and Fatigue

Inclusions—phases formed between metallic elements and nonmetallic elements such as sulfur and oxygen—are an important microstructural component of steels. Coarse or high densities of inclusions initiate fracture and lower the toughness of steels. As a result, modern steelmaking practices, which incorporate improved deoxidation, shrouding of liquid steel to prevent reoxidation, vacuum degassing, argon blowing, and desulfurization, are designed to substantially increase the "cleanliness" of steels by lowering the number and/or modifying the morphology of inclusions.

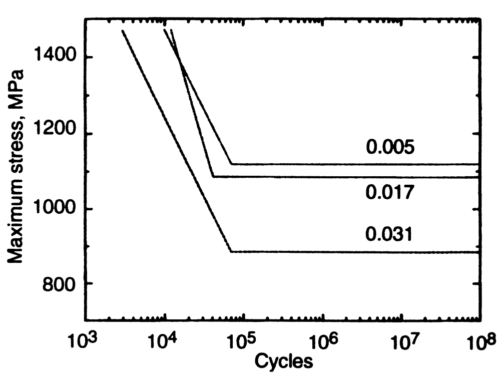


Fig. 13 Effect of phosphorus content on the bending fatigue of direct-quenched, gas-carburized modified 4320 steel with 0.005, 0.017, and 0.031 wt% phosphorus, as marked. Source: Ref 22

In carburized steels, the very high strength of the carburized case makes the plastic zone ahead of surface discontinuities (such as machining marks), flaws, or cracks very small, and therefore, in clean steels, distributed inclusions play a smaller role in fracture than, for example, uniformly distributed and closely spaced grain boundary embrittling structures or surface oxides. In other words, the high stresses in the plastic process zone have a much higher probability of acting on grain boundaries or surface oxides than on widely spaced inclusions.

Although inclusions often play a secondary role in the fatigue of gas-carburized steels, especially when there are other microstructural causes of fatigue crack initiation, they may be involved in the fatigue process in several ways. Inclusions in the carburized steel may either combine with other features that initiate fatigue cracks, or in the absence of such features, serve as the sole source of fatigue crack initiation.

An example of the first type of effect was demonstrated in a study that examined the effects of systematic variations in sulfur content on the bending fatigue resistance of a gas-carburized low alloy steel (Ref 20). Sulfur combines with manganese to form manganese sulfide (MnS) inclusions in steel. The MnS particles are plastic during hot rolling, and as a result, are elongated in the rolling direction. This elongation imparts an anisotropy to the mechanical properties of the steel, which makes the effect of inclusions a function of the orientation of the particles relative to the direction of the applied load.

S-N (stress vs. life) curves for specimens of a gas-carburized SAE 8219-type steel with three levels of sulfur are plotted in Fig. 14. The endurance limit decreases with increasing sulfur content. A number of the specimens with the higher sulfur contents showed runouts at 10 million cycles at stress levels higher than the endurance limits shown, but the specimens that failed at the lower stresses were used to establish the endurance limits. In these specimens, elongated MnS inclusion particles that happened to be close to the specimen surfaces were associated with the fatigue fracture initiation. Fatigue fracture initiation of the direct-quenched, gas-carburized specimens was still dominated by intergranular fracture, but if sulfides were present at the highly stressed surfaces of the bending fatigue specimens, they apparently provided an extra source of stress concentration and reduced fatigue performance.

An example of MnS particles associated with fatigue crack initiation in a carburized 8219-type steel is shown in Fig. 15. Although fatigue resistance is lowered somewhat by the presence of increased densities of MnS particles, the decrease may be outweighed by a gain in machinability associated with higher levels of sulfur.

Austenitic Grain Size and Fatigue

Effect of Fine Grain Size on Microstructures and Properties. Prior-austenite grain size of carburized steels correlates strongly with bending fatigue resistance. Generally, the finer the prior-austenite grain size, the better the fatigue performance. For example, Fig. 16 shows a direct relationship of bending fatigue endurance limit on prior-austenitic grain size, plotted as the inverse square root of the grain size, for several sets of carburized 4320 steels (Ref 28).

The refinement of austenite grain size has several effects on the case microstructure of carburized steels. A finer prior-austenite grain size produces a finer martensitic microstructure on quenching, and therefore, raises the strength of the carburized case. Increases in strength are beneficial to high-cycle fatigue resistance, as discussed previously in the section on bending fatigue testing.

Another very important consequence of fine austenitic grain size is the dilution of the grain boundary segregation of phosphorus. In fact, very fine austenitic grain sizes can eliminate the sensitivity of high-carbon case microstructures to intergranular fracture. As a result, other mechanisms of fatigue crack initiation replace intergranular cracking, generally to the benefit of fatigue performance. The high values of endurance limits shown for the very fine grain specimens in Fig. 16 were associated with fatigue crack nucleation at surface oxidation, not at embrittled prior-austenite grain boundaries. Although, as discussed below, surface oxides form on austenite grain boundaries and fatigue cracks may nucleate on the oxide-covered austenite boundaries, fine-grain specimens show no intergranular fracture below the oxidized surface layers.

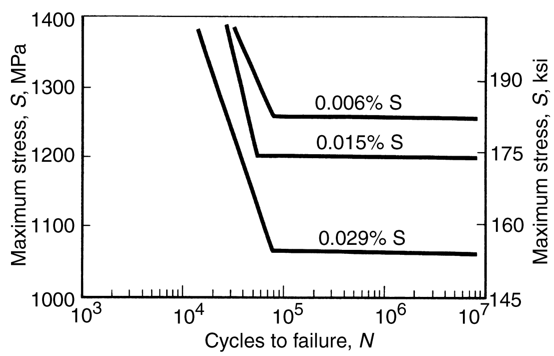


Fig. 14 *S-N* curves determined by bending fatigue of a gas-carburized SAE 8219-type steel containing 1.40 Mn, 0.61 Cr, 0.30 Ni, 0.20 Mo, and three levels of sulfur. Source: Ref 20

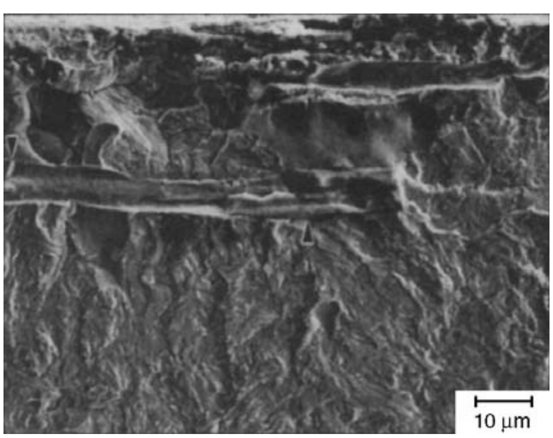


Fig. 15 Elongated manganese sulfide particles associated with bending fatigue crack initiation at the surface of a gas-carburized SAE 8219-type steel. SEM photomicrograph. Source: Ref 20

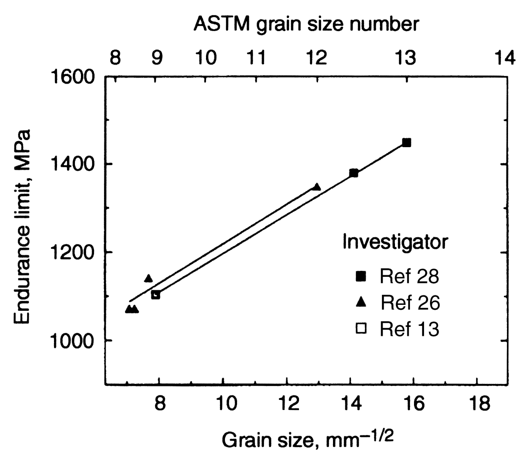


Fig. 16 Endurance limits as a function of prior-austenite grain size from various studies of bending fatigue of gas-carburized 4320 steels. Source: Ref 28

Reheat Treatments to Achieve Fine Grain Size. The most effective way to produce very fine grains in carburized steels is by slow cooling and reheating of carburized parts at temperatures below the A_{cm} where austenite and cementite are stable. The cementite particles effectively retard austenite grain growth and reduce the carbon content of the austenite. Specimens reheated to above the A_{cm} may show grain refinement, depending on the temperature of heating, but because all carbides are dissolved, grain size refinement is not as effective as in specimens heated below the A_{cm}. Figure 17 shows austenitic grain size as a function of distance from the carburized surface of gas-carburized 4320 steel specimens in the direct-quenched condition and after one and three reheating treatments (Ref 28). The reheat treatments very effectively reduce the near-surface case grain size where carbon content is the highest, and therefore, the greatest density of carbide particles is retained during intercritical reheating.

Intercritical-temperature reheat treatments of carburized steels produce very fine austenitic grain sizes and high endurance limits. Typically the endurance limits are above 1400 MPa (203 ksi) (Ref 28). However, the beneficial effects of the reheating on bending fatigue resistance are not due to grain size refinement alone. The reduced carbon content of the austenite when carbides are retained raises M_s temperatures and reduces the amount of retained austenite in the as-quenched case microstructures. Reduced levels of retained austenite raise the strength of case microstructures, and therefore, also may contribute significantly to the improved high-cycle fatigue performance of fine-grain, intercritically reheated carburized steels.

Surface Oxidation and Fatigue

The surface oxidation produced during gas carburizing may or may not significantly reduce bending fatigue resistance. The most severe effects of such oxidation are associated with a reduction in near-surface case hardenability, which results from the removal of chromium, manganese, and silicon from solution in the austenite by the oxide formation (Ref 29, 30). The reduced case hardenability can cause non-martensitic microstructures, such as ferrite, bainite, and pearlite, to form at the surface of the carburized steel. Not only is the surface hardness reduced, but the residual surface stresses may become less compressive or even tensile.

Figure 18 and 19 show the effects of surface oxidation with reduced hardenability on the bending fatigue and residual stresses of 8620 and 4615 gas-carburized specimens (Ref 31). The 4615 steel has higher hardenability by virtue of higher nickel and molybdenum contents and a lower sensitivity to surface oxidation by virtue of reduced manganese and chromium contents

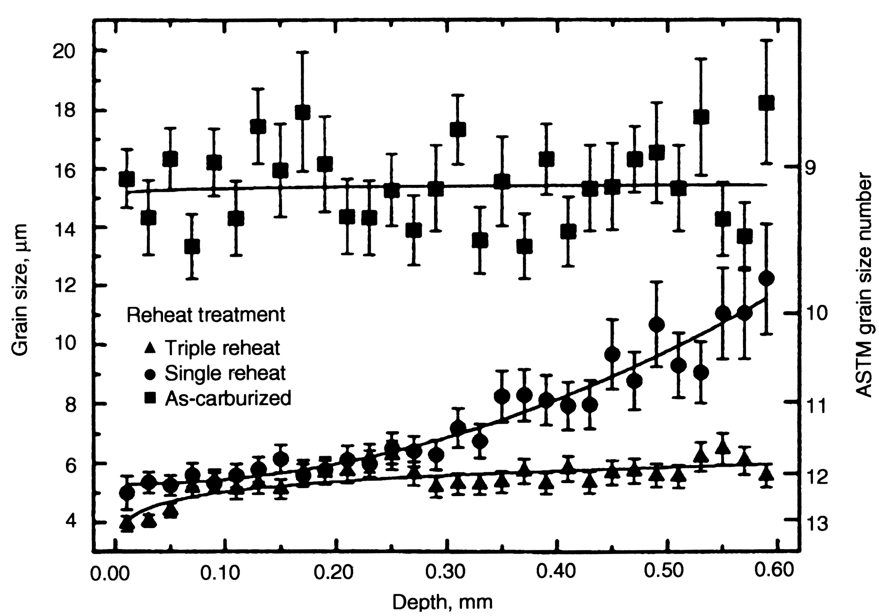


Fig. 17 Prior-austenite grain size as a function of depth from the surface of gas-carburized 4320 specimens in the as-carburized, direct-quenched condition and reheated conditions. Source: Ref 28

(Ref 31). As a result of the different chemistries, the 8620 steel formed pearlite in the near-surface regions of the case, while the microstructure of the 4615 steel, despite some oxidation, consisted only of plate martensite and retained austenite at the surface (Ref 31). These differences in microstructures due to surface oxidation and reduced case hardenability are consistent with the differences in bending fatigue performance and residual stresses shown between the two steels in Fig. 18 and 19. This study illustrates the importance of steel chemistry on controlling surface oxidation and the associated formation of nonmartensitic microstructures in gas-carburized steels. Another approach used to reduce surface oxide formation in steels with low hardenability is to use more severe quenching with higher cooling rates.

If the hardenability of a steel is sufficient to prevent the formation of nonmartensitic microstructures for a given gas carburizing and quenching schedule, surface oxidation has a much reduced effect on bending fatigue performance. In direct-quenched specimens, as discussed previously and demonstrated in Fig. 10, intergranular fracture to depths much deeper than the oxidized layers dominates fatigue crack initiation. However, when the conditions for intergranular crack initiation are minimized, as, for example, by reheating or shot peening, the surface oxide layers become a major location for bending fatigue crack initiation. Figure 20 shows crack initiation in the oxidized zone of a gas-carburized and reheated specimen of 4320 steel. The initiation is confined to the oxidized zone, and stable transgranular fatigue propagation proceeds directly below the oxidized zone with no evidence of intergranular fracture.

Retained Austenite and Fatigue

Retained austenite is one of the most important microstructural components in the case of carburized steels. The amounts of retained austenite vary widely, depending on carbon and alloy content, heat-treating conditions, and special processing steps such as shot peening and subzero cooling. Generally, the higher the carbon and alloy content, the lower the M_s temperature and the higher the retained austenite content in the microstructure. The low-temperature tempering applied to carburized steels, generally performed at temperatures below 200 °C (400 °F), is not high enough to cause the retained austenite to transform, and therefore, retained austenite remains an important component of the microstructure. At higher tempering temperatures, retained austenite transforms to cementite and ferrite with attendant decreases in hardness and strength as the martensitic microstructure coarsens.

The role that retained austenite plays in the bending fatigue performance of carburized steels has been difficult to identify because of the variable loading conditions that may be applied to carburized machine components and the complicating effects of other factors, such as

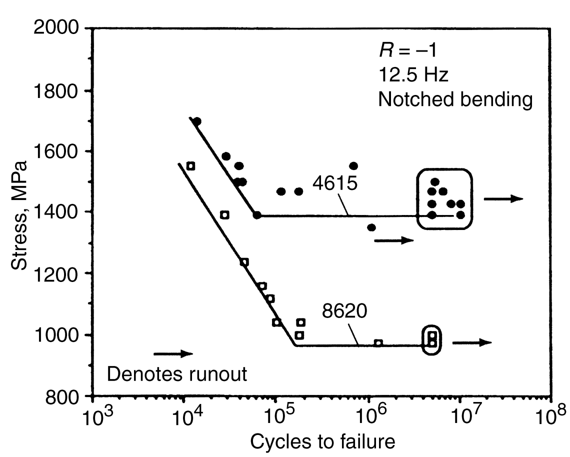


Fig. 18 *S-N* curves for direct-quenched gas-carburized 4615 and 8620 steels, notched 4-point bend specimens. Compositions of the steels are given in Table 1. Non-martensitic transformation products were present on the surfaces of the 8620 steel specimens and absent on the 4615 steel specimens. Source: Ref 31

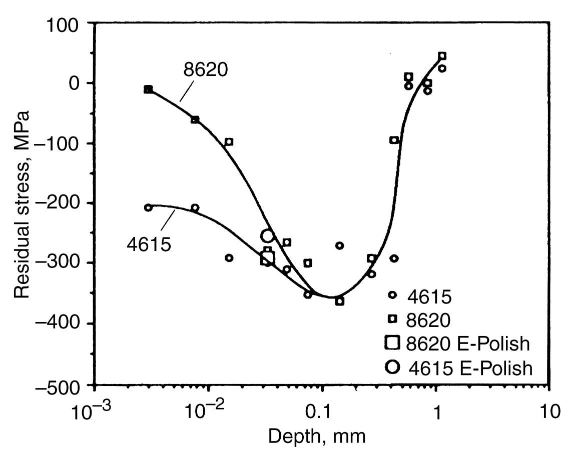


Fig. 19 Residual stress as a function of depth below the surface of the direct-quenched gas-carburized 4615 and 8620 steel specimens described in Fig. 18. E-Polish = electrolytic polish. Source: Ref 31

residual stresses, grain boundary embrittling structures, and surface oxidation. With respect to loading conditions, it appears that higher amounts of retained austenite are detrimental to high-cycle fatigue and reduce endurance limits (Ref 10, 14) while higher amounts of retained austenite are beneficial for low-cycle, high-strain fatigue (Ref 14, 32, 33).

Reduced retained austenite contents of LTT martensite/austenite composite microstructures increase elastic limits and yield strengths (Ref 34), and therefore, benefit stress-controlled, high-cycle fatigue. One of the approaches to reducing the retained austenite content in the case microstructures of carburized steels, is to reheat carburized specimens to temperatures below the A_{cm} and quench. Invariably such reheating and quenching significantly increases bending fatigue endurance limits compared to direct-quenched specimens of identically carburized specimens (Ref 28). The reheating not only reduces retained austenite but also refines the austenitic grain size, refines the martensitic structure, and reduces susceptibility to intergranular fracture—all features that are known to improve fatigue resistance. Therefore, improved high-cycle fatigue resistance of reheated and quenched specimens is related to a combination of microstructural changes, including low retained austenite contents.

The benefit of retained austenite to strain-controlled, low-cycle bending fatigue is related to the improved ductility and reduced strength and hardness that retained austenite contributes to a composite LTT martensite/austenite case microstructure. In addition, retained austenite, at sufficiently high applied strains and stresses, undergoes deformation-induced transformation to martensite (Ref 35). The volume expansion associated with the strain-induced formation of martensite creates compressive stresses (Ref 21) that lead to reduced rates of fatigue crack growth, accounting for the enhanced low-cycle fatigue performance that is observed in carburized steels with high amounts of retained austenite in the case (Ref 33).

Subzero Cooling and Fatigue

Cooling carburized steel below room temperature is a processing approach sometimes used to reduce the retained austenite content in the case regions of carburized steels. The transformation of austenite to martensite is driven by temperature changes, and the low M_s temperatures of high-carbon case regions of carburized alloy steels limit the temperature range between M_s and room temperature over which martensite forms. Therefore, the temperature range for martensite formation and the reduction of retained austenite is extended by cooling below room temperature. The cooling treatments are variously referred to as subzero cooling, refrigeration treatments, or deep cooling.

In addition to the effects of retained austenite on bending fatigue, as discussed, any deformation-induced transformation of retained austen-

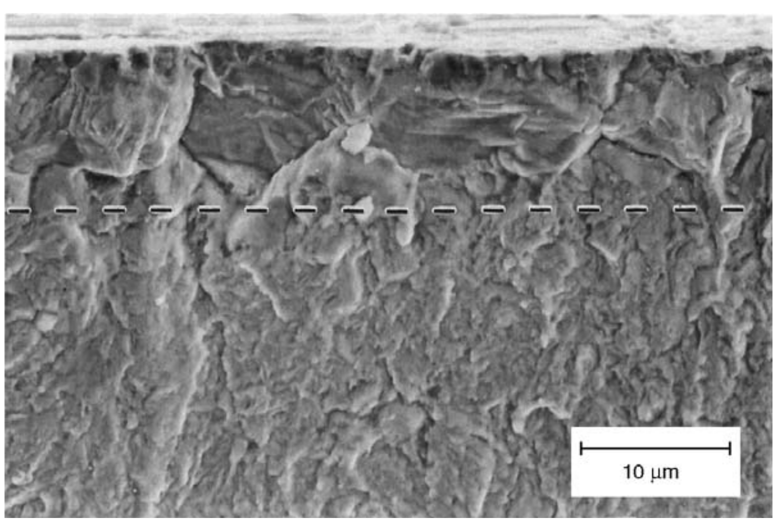


Fig. 20 Bending fatigue crack initiation in gas-carburized and reheated 4320 steel. The dashed line corresponds to maximum depth of surface oxidation, and all fracture below dashed line is transgranular. Source: Ref 28

ite during cyclic loading in service, because of the volume expansion that accompanies the transformation of austenite to martensite, may change the dimensions of a carburized component. Therefore, subzero cooling is one approach to reduce retained austenite in parts that require high precision and stable dimensions throughout their service life. However, several studies show that subzero cooling lowers the bending fatigue resistance of carburized steels. Nevertheless, high-quality, high-performance aircraft and helicopter gears are routinely subjected to subzero cooling without apparent detrimental effects (Ref 36). For example, a commonly used carburizing steel for aircraft gears is 9310, which contains about 3 wt% Ni (see Table 1). The high nickel content lowers the M_s temperature and increases the amount of austenite at room temperature. The austenite content can be reduced by subzero cooling, probably with adverse effects on localized residual stress. However, the latter adverse effect of subzero cooling may be offset by fine austenite grain size, and high nickel content may improve the fracture toughness and fatigue resistance of carburized steels (Ref 37) to a level where fatigue resistance is not adversely affected by subzero cooling.

In the low-alloy steels commonly used for carburizing, subzero cooling used to reduce retained austenite may reduce the bending fatigue resistance of carburized components. If refrigeration treatments are applied, parts should be tempered both before and after. Figure 21 shows an example of the detrimental effects of subzero cooling on the bending fatigue resistance of carburized specimens. The data were produced in an experimental study of vacuum-carburized specimens of 8620 and EX 24 steel that were deep cooled to –196 °C (–321 °F) in liquid nitrogen (Ref 38). The overall fatigue performance in this study was complicated by high retained austenite contents and coarse carbide particles at square specimen corners, but the detrimental effect of subzero cooling on bending fatigue performance is clearly demonstrated in Fig. 21.

The detrimental effect of subzero cooling on the bending fatigue of carburized specimens has been related to changes in residual stress by several investigations (Ref 39–41). The overall surface residual stresses become increasingly compressive, as measured from the martensite in the case and as expected from the constraint of the expansion that accompanies the transformation of austenite to martensite as temperature is decreased. However, the residual stresses in the austenite phase are measured to be tensile, especially at the surface of the carburized specimens. These tensile stresses would then be expected to lower the surface tensile stresses applied in bending to initiate fatigue cracks. Microcrack formation within martensite plates and at plate/austenite interfaces may be enhanced by the localized residual stresses induced by subzero cooling (Ref 41), but they could be minimized by maintaining a fine prior-austenite grain size and applying reheating treatments (Ref 42, 43).

Residual Stresses, Shot Peening, and Fatigue

Compressive residual stresses are formed in the case microstructures of carburized steels as a result of transformation and temperature gradients induced by quenching (Ref 44, 45). The magnitude and distribution of the residual stresses, therefore, are complex functions of the temperature gradients induced by quenching (Ref 46), which in turn are dependent on specimen size and geometry, the hardenability of the steel, the carbon gradient, and the case depth. The residual stresses as a function of case depth are routinely measured by x-ray diffraction, and considerable effort has been applied to modeling residual stress profiles in carburized steels as a function of cooling and hardenability (Ref 47, 48).

Figure 22 shows the range and pattern of compressive residual stresses typically formed in the case regions of direct-cooled carburized steels (Ref 49). The compressive residual stresses offset the adverse effects of factors such as quench embrittlement and intergranular fracture to

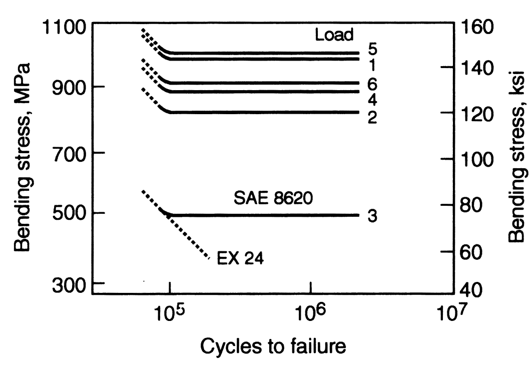


Fig. 21 *S-N* curves of vacuum-carburized 8620 and EX 24 (0.89% Mn, 0.24% Mo, 0.55% Cr) steels. The lower curves were obtained from specimens subzero cooled to –196 °C, and the upper curves were obtained from specimens not subjected to subzero cooling. Source: Ref 38

which high-carbon microstructures are susceptible (Ref 24), and they increase the fracture and fatigue resistance of direct-quenched parts to levels that provide good engineering performance. As discussed earlier, case residual stresses in carburized steels are adversely modified by subzero cooling, and they are positively modified (made locally more compressive) by the strain-induced transformation of austenite to martensite. Tempering lowers residual compressive stresses because of dimensional changes that accompany the recovery and coarsening of the martensitic microstructure during tempering (Ref 50).

The fatigue lives of carburized steel gears can be significantly improved by shot peening, which is a method of cold working in which compressive stresses are induced in the exposed surface layers of metallic parts by the impingement of a stream of shot. The shot, which may be composed of cast steel, cast iron, or glass beads, is directed at the metal surface at high velocity under controlled conditions. Details on the process can be found in the article "Shot Peening" in Volume 5 of the *ASM Handbook.*

The effect of shot peening process parameters on fatigue life of carburized helical gears has been investigated (Ref 51). Low-cycle gear-tooth bending fatigue lives were evaluated as a function of Almen intensity parameter. Almen intensity is an indirect method of monitoring the aggregate amount of energy transfer or residual compressive stresses imparted to the workpiece. The method uses a strip of SAE-AISI 1070 spring steel in the 44 to 50 HRC range. The strip is mounted to an Almen test block and exposed to the blast in identical fashion as the critical area of the workpiece, with the same parameters. After peening, the Almen strip is removed from the block and the amount of maximum deviation from flat is measured by an Almen gage. This measurement is called arc height. Details of the design and use of Almen gages and test strips are given in SAE standards J442, "Test Strip, Holder and Gage for Shot Peening," and J443, "Procedure for Using Shot Peening Test Strip." Additional information on the shot peening process can also be found in Chapter 9, "Carburizing." Refer to the section "Shot Peening of Carburized and Hardened Gears" and in particular Fig. 62 in Chapter 9 which illustrates the shot peening test setup.

Figure 23 shows the data obtained from the fatigue life versus Almen intensity study on carburized 4023 steel. The gears were peened to three Almen intensity "A" levels (0.305, 0.457, and 0.609 mm, or 0.012, 0.018, and 0.024 in.) using steel shot in the size of S280, and to one Almen intensity "C" level of 0.008 in. with S280 shot. As Fig. 23 clearly shows, the mean fatigue life of unpeened gears was about 9800 cycles while optimized shot peening procedures (24A Almen intensity) resulted in a mean fatigue life exceeding 33,000 cycles. The improved fatigue

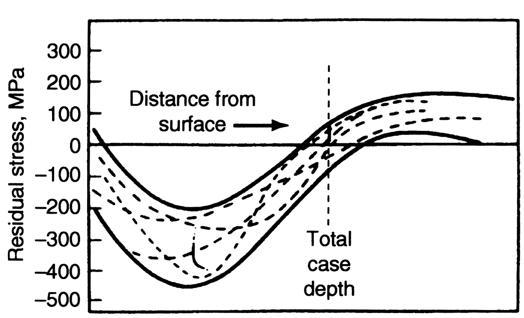


Fig. 22 Ranges and patterns of residual stresses as a function of depth for 70 carburized steels. Source: Ref 49

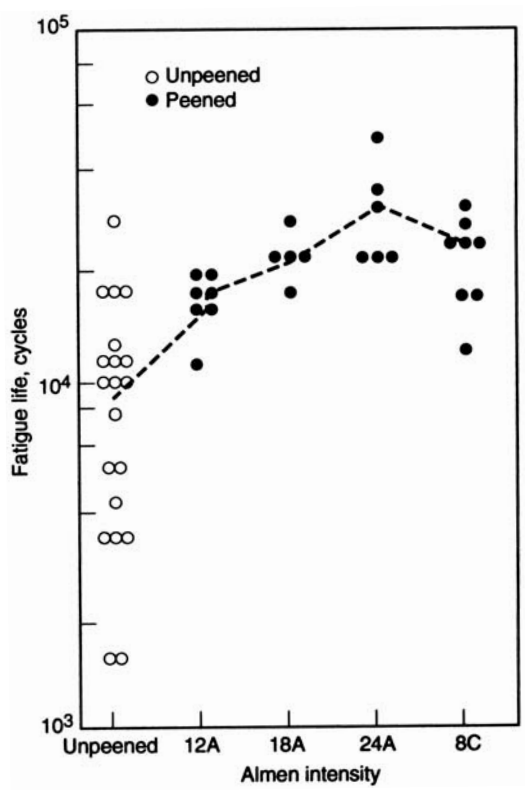


Fig. 23 Effect of a shot peening parameter (Almen intensity as described in text) on the low-cycle fatigue of a carburized helical gear made from 4023 steel. Source: Ref 51

life for 24A peened gears was attributed to the highest compressive residual stress incurred.

Other Properties of Interest for Carburized Steels

Although much of the recent research on properties of carburized steels has centered around bending fatigue (see previous section), there are other properties that effect the service life of carburized components. As will be described forthwith, these mechanical properties are strongly influenced by core and case microstructure, case depth, residual stresses, and alloy chemistry.

Rolling Contact Fatigue

Rolling contact fatigue is a surface-pitting-type failure commonly found in gears (see Chapter 13, "Gear Failure Modes and Analysis," for details). Rolling contact fatigue differs from classic structural fatigue (bending or torsional) in that it results from contact or Hertzian stress state. This localized stress state results when curved surfaces are in contact under a normal load. Generally, one surface moves over the other in a rolling motion. The contact geometry and the motion of the rolling elements produces an alternating subsurface shear stress. Subsurface plastic strain builds up with increasing cycles until a crack is generated. The crack then propagates until a pit is formed. Once surface pitting has initiated, the bearing becomes noisy and rough running. If allowed to continue, fracture of the rolling element and catastrophic failure occurs. Fractured races can result from fatigue spalling and high hoop stresses.

Extreme cases of spalling are associated with case crushing or cracking initiated at the case-core interface. If sliding is coupled with contact loading, surface pits develop. Very high contact loads cause microstructural changes within high-carbon martensite that are revealed by various types of etching (Ref 52–54). Generally retained austenite is regarded as a microstructural constituent that is beneficial for rolling-contact fatigue resistance (Ref 55, 56).

One of most effective means for improving the rolling contact fatigue life of carburized steels is by utilizing clean steel technology. Secondary metallurgical techniques in ladle refining and teeming practice have become the key elements in improved steelmaking practices. Important developments include degassing and deoxidation practices, improved temperature control, inert gas shrouding, improved refractories, and the use of bottom-poured or significantly improved tundish systems, and large cross-section continuous casters. Figure 24 demonstrates the improvements in rolling contact fatigue life of carburized steels as steelmaking practices have improved.

Wear Resistance (Ref 58)

Another important property afforded by the carburizing process is wear resistance resulting from the high hardness of the case. As with bend-

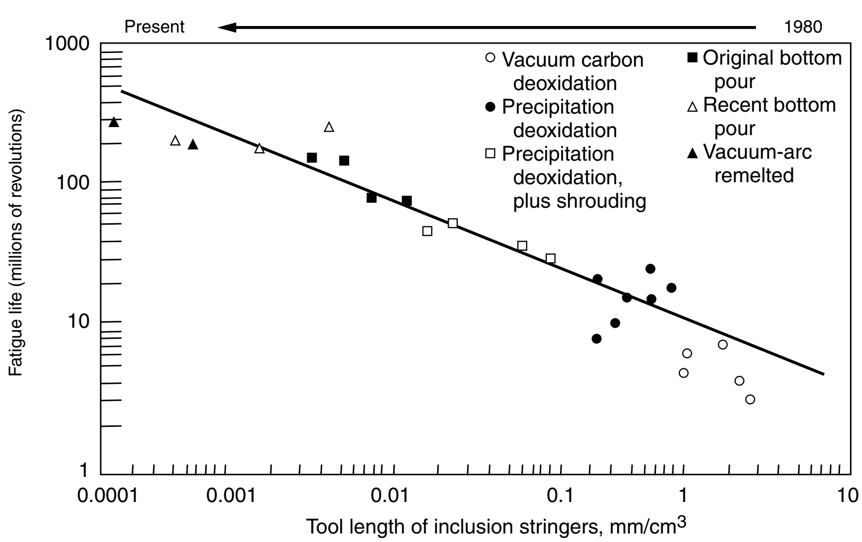


Fig. 24 Cleanness and rolling contact fatigue life improvements in carburized steels as steelmaking practices have changed. Source: Ref 57

ing fatigue, there are a number of microstructure/property relationships that must be considered for wear-resistant applications.

Microstructural Features. The independent variables available for controlling the microstructure/properties of carburized cases are those that define the carburizing alloy (composition, cleanliness) and those that define the carburizing process (time/temperature/carbon-potential carburizing history, time/temperature quenching history, time/temperature tempering history). These tools provide a considerable degree of control over these microstructural features:

- Martensite
  a. Carbon content of source austenite
  b. Plate size (austenite grain size)
  c. Strength
  d. Secondary hardening
- Primary carbides
  a. Size
  b. Volume fraction
- Retained austenite
  a. Volume fraction
  b. Carbon content
- Nonmetallic inclusions
  and these global features:
- Case depth
- Residual stress distribution

which determine the tribological properties of the case. The combination of properties that is best for each application must then be decided.

The necessary case depth and case hardness can be estimated from a Hertzian stress calculation, but other microstructural objectives can be specified only qualitatively. For many applications, the following "rules of thumb" apply:

- Sufficient case depth and case hardness must be provided to prevent indentation or case crushing under the anticipated contact loads. For gears loaded in "line contact," a minimum case hardness of 58 HRC frequently is specified. When high contact loads are accompanied by sliding, the near-surface hardness (to a depth of about 50 μm, or 2 mils) may have to be raised to prevent shearing of surface layers.
- The retained austenite content should be as high as possible, consistent with the requirements of the previous rule. The retained austenite content should be controlled by adjusting the case carbon content, not by subzero quenching after carburizing or by tempering at temperatures above 200 °C (390 °F).
- The tempering temperature chosen should be as low as possible, but above the surface temperatures anticipated in finishing operations and in service.
- The content of nonmetallic inclusions should be no higher than that needed for economical machining.
- Coarse primary carbides can be helpful in resisting abrasive wear. Fine primary carbides can permit more retained austenite at the same hardness level. Experiments should be conducted to verify any benefits presumed to be associated with primary carbides.

Hot Hardness

Retention of hardness at elevated temperatures (hot hardness) is vitally important in applications where high local temperature conditions can be encountered. Examples include helicopter gears, speed reduction gear sets, and turbine gearing. Figure 25 provides hot hardness data for several carburized steels. It is evident from these data that higher alloy content is needed to assure sufficient hardness at temperatures above 315 °C (600 °F) encountered in severe service.

Bending Strength and Bend Ductility

In service, carburized and hardened steels are subjected to bending loads and must be able to

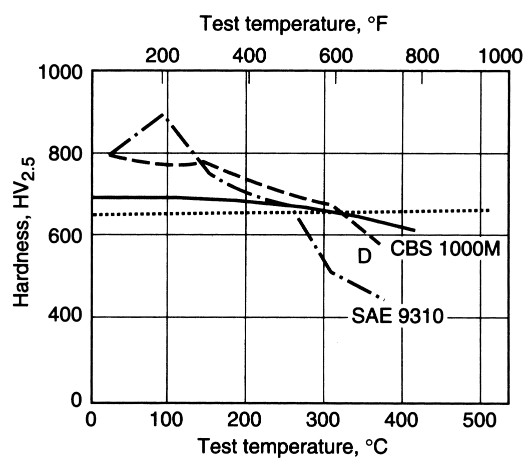


Fig. 25 Hot hardness of three carburized steels. The dashed line corresponds to a surface hardness of 58 HRC. Compositions for SAE 9310 and CBS 1000M are listed in Table 1. The nominal composition for steel D is 0.12% C, 0.5% Mn, 1.1% Si, 1.0% Cr, 2.0% Ni, 2.3% Mo, and 1.2% V. Source: Ref 59

resist design loads and overloads without fracture. A variety of laboratory tests have been performed to define the resistance of carburized and hardened steels to failure under bending loads, and to provide information on the contribution of alloys in resisting failure.

Bend ductility of several carburized and hardened steels over a range of test temperatures from room temperature to –195 °C (–320 °F) is given in Fig. 26. A comparison of the data for carburized SAE 4817 steel with those for SAE 4027 (both with about the same amount of retained austenite) shows that the steel alloyed with a substantial amount of nickel exhibited much greater ductility.

Razim (Ref 60) has observed that the static bend test is useful in evaluating the ability of a carburized and hardened surface zone to sustain plastic deformation without cracking. A suitable measurement can be the initial crack strength. His summary of such tests indicates that with low surface carbon contents of about 0.6% C, the initial crack strength increases with increasing core strength; while with high surface carbon contents (about 1.2% C), the initial crack strength becomes less dependent on core strength, but drops to a considerably lower level than the crack strength exhibited by the steels with 0.6% C at the surface.

Toughness

Toughness in carburized steels can be measured in a variety of ways. In recent years, attempts have been made to simulate service loads, single- as well as multiple-load applications, different rates of load application, and to include realistic details in the test specimen such as changes in notch severity or the use of a part such as a gear tooth as the specimen. Results of one such test, the impact fracture test, will be described in this section. Additional test results are reviewed in Ref 4.

Impact Fracture Strength. The specimen shown in Fig. 27 was used by Cameron and others (Ref 61) for single-blow impact tests as well as fatigue tests. This simulated gear specimen was loaded in a standard pendulum type testing machine, with the specimen held in a vertical position, similar to an Izod impact test. An instrumented tup permitted recording the energy absorbed by the specimen as a function of time during the test. For carburized-and-hardened steels, the investigators found that alloy content and core carbon content significantly influenced resistance to fracture under impact bending conditions. In Fig. 28 the chromium-molybdenum steels exhibited higher fracture strengths than the manganese-chromium steels, but the nickel-chromium-molybdenum steel, PS55, not only exhibited much higher fracture strength, but fracture strength did not decrease with increasing carbon content.

In another study by Smith and Diesburg (Ref 62), an attempt was made to define in more detail the effect of alloy interactions on fracture strength using the simulated gear-tooth specimen shown in Fig. 27. Figure 29 summarizes

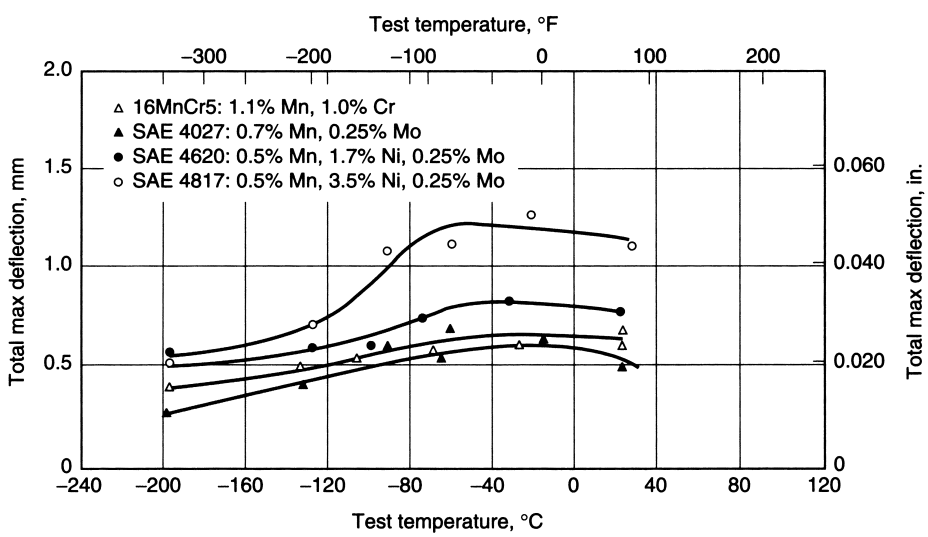


Fig. 26 Bend ductility transition curves for carburized and hardened steels. Nominal alloy contents of the steels are listed within the diagram. Source: Ref 59

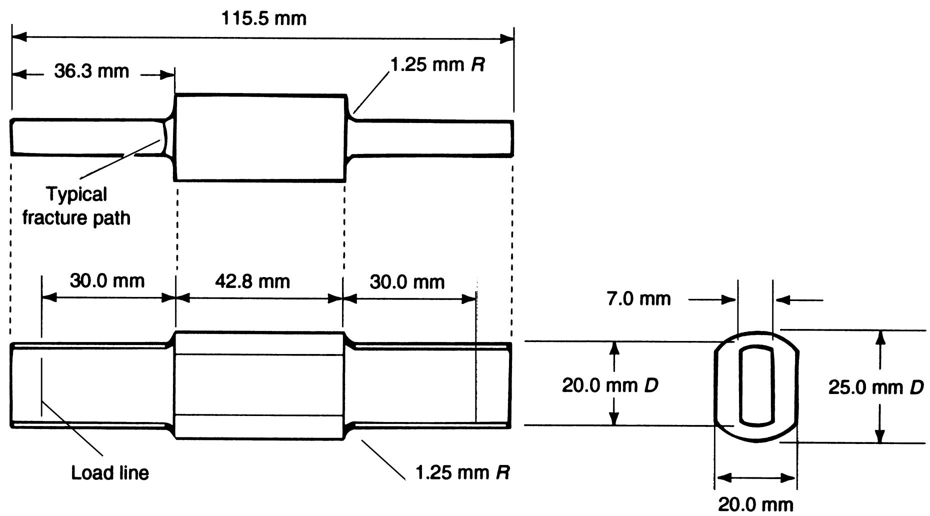
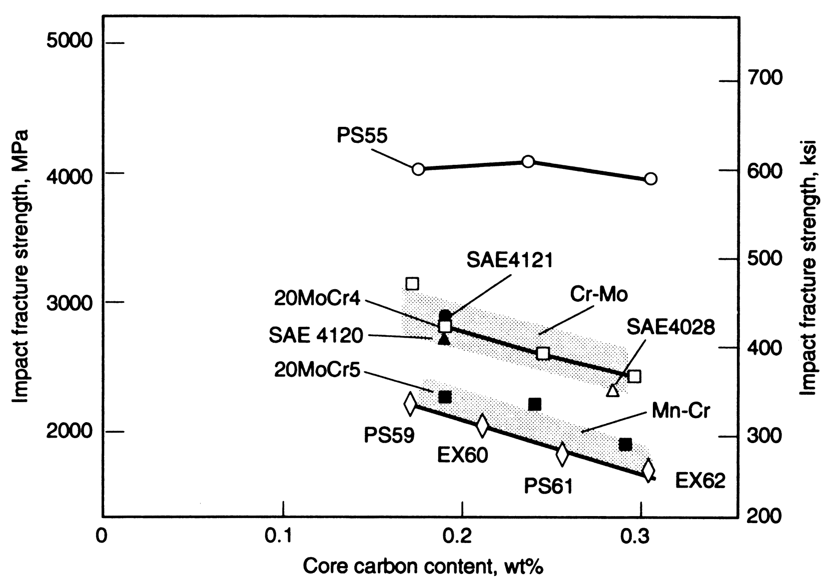


Fig. 27 Test specimen used by Cameron for fatigue, bend, and impact fracture tests. Typical results are shown in Fig. 28 and 29. Source: Ref 61

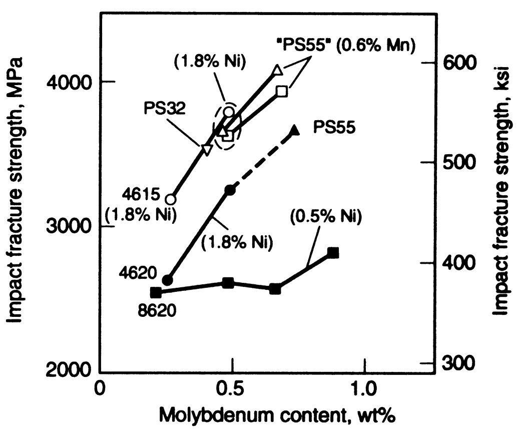


SAE (DIN) steel grade	C	Mn	Ni	Cr	Mo
	Composition, wt %				
PS55	0.17	0.87	1.82	0.55	0.72
4121	0.20	0.87	...	0.55	0.25
4028	0.28	0.80	...	...	0.25
4120	0.17	1.03	...	0.52	0.19
(20MoCr4)	0.19	0.84	...	0.40	0.50
(20MnCr5)	0.21	1.18	...	1.11	...
PS59	0.17	1.20	...	0.90	...
EX60	0.21	1.18	...	0.90	...
PS61	0.26	1.18	...	0.90	...
EX62	0.30	1.12	...	0.88	...

Fig. 28 Effect of core carbon content and alloy content on impact fracture strength of a series of steels, carburized at 925 °C (1700 °F), cooled to 840 °C (1550 °F), oil quenched, and tempered at 150 °C (340 °F). Source: Ref 61

the effect of molybdenum and nickel on fracture strength for several different groups of steels, each modified by adding molybdenum. Each group of steels was carburized to have similar case depths and surface carbon contents, then the steel specimens were cooled from the carburizing temperature to 845 °C (1550 °F), quenched in oil at 65 °C (150 °F), and then tempered at least 1 h at 170 °C (340 °F). It is evident that nickel in amounts greater than 0.5% is very effective in increasing fracture strength, especially when molybdenum content is greater than 0.25%. The investigators also compared vacuum-melted steels with air-melted steels and obtained higher fracture strengths with the vacuum-melted steels.

Fracture Toughness Data. Work by Diesburg (Ref 63) included a study of fracture toughness of several steels carburized to produce case depths (at 0.5% C) between 0.75 and 1.00 mm (0.030 and 0.040 in.) thick. Results shown in Fig. 30 indicate that the plane-strain fracture toughness (K_{Ic}) values for the higher nickel steels SAE PS55, PS32, 9310, and 4820 are quite similar, much higher than the values observed for the lower nickel SAE 8620 steel at all depths below the surface. Close to the carburized surface, only small differences could be observed among the steels tested.



SAE steel grade	Composition, wt %				
	C	Mn	Ni	Cr	Mo
8620	0.20	0.8	0.5	0.5	0.20
4615 (20)	0.15-0.20	0.5	1.8	...	0.25
PS32	0.20	0.8	0.9	0.5	0.5
PS55	0.17	0.8	1.8	0.5	0.7

Fig. 29 Impact fracture strength of carburized steels containing various combinations of molybdenum and nickel. Open data points are for vacuum-melted heats; solid data points are for air-melted heats. Compositions were those of standard steels to which molybdenum was added in varying amounts, as indicated in the accompanying table. Source: Ref 62

Improved Elevated-Temperature Performance (Ref 58)

Several secondary hardening carburizing alloys have been developed for applications that require resistance to elevated temperatures, such as helicopter gearing. (See the "Special alloys" listed in Table 1.) These alloys make use of the precipitation of copper and/or M_2C and MC carbides to provide resistance to softening for temperatures up to 550 °C (1020 °F). Because they contain substantial amounts of molybdenum and vanadium, these alloys resemble low-carbon versions of tool steels. Some of the alloys are difficult to carburize because of high silicon and chromium contents; preoxidation prior to carburizing is necessary to permit carbon penetration. Secondary hardening alloys could also be useful in ambient-temperature applications in which lubrication is marginal, because the heat generated by intermittent metal-to-metal contact would not readily soften the underlying metal.

One of the "Special Alloys" listed in Table 1 is Pyrowear 53 (UNS K71040) which is used for helicopter transmission drive gears. The higher copper content increases core impact strength, while the increased molybdenum provides improved elevated-temperature properties. The reduced chromium levels enhance carburizability, while the vanadium maintains a fine grain size.

When compared to conventional grades, e.g., SAE-AISI 9310, Pyrowear Alloy 53 exhibits superior temper resistance and high case hot hardness while maintaining high core impact strength and fracture toughness (126 to 132 MPa$\sqrt{m}$, or 115 to 120$\sqrt{in.}$). Table 8 lists core mechanical properties for this proprietary carburizing grade.

Other Ferrous Alloys for Gears

In addition to wrought steels, the following iron-base alloys are also used for gears:

- Cast carbon and alloy steels
- Gray and ductile cast irons
- P/M irons and steels
- Stainless steels

- Tool steels
- Maraging steels

Each will be discussed in subsequent sections.

Cast Steels

Steel castings are produced by pouring molten steel of the desired composition into a mold of the desired configuration and allowing the steel to solidify. The mold material may be silica, zircon, chromite sand, olivine sand, graphite, metal, or ceramic. The choice of mold material depends on the size, intricacy, dimensional accuracy of the casting, and cost. While the producible size, surface finish, and dimensional accuracy of castings vary widely with the type of mold, the properties of the cast steel are not affected significantly.

Steel castings can be made from any of the many types of carbon and alloy steel produced in wrought form. Those castings produced in any of the various types of molds and wrought steel of equivalent chemical composition respond similarly to heat treatment, have the same weldability, and have similar physical and mechanical properties. However, cast steels do not exhibit the effects of directionality on mechanical properties that are typical of wrought steels. This nondirectional characteristic of cast steel mechanical properties may be advantageous when service conditions involve multidirectional loading.

The cast steels used for gears are generally modifications of standard SAE-AISI designations. Common through-hardening cast steels include 1045, 4135, 4140, 8630, 8640, and 4340. Carburizing grades are usually 1020,

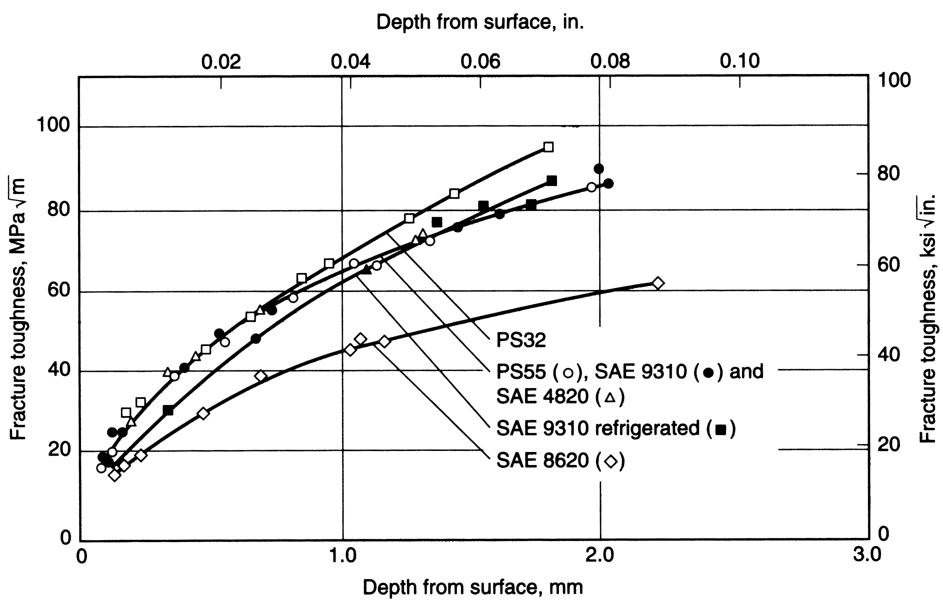


Fig. 30 Fracture toughness in carburized steels as a function of distance below the surface. The SAE PS55, 9310, and 8620 steels were commercial heats; the SAE PS32 and 4820 steels were laboratory heats. The PS32 and 4820 steels were quenched directly after carburizing at 925 °C (1700 °F) into 170 °C (340 °F) oil; other steels were cooled from 925 °C to 840 °C (1700 to 1550 °F) before quenching into 65 °C (150 °F) oil. Data are also shown for 9310 steel that was refrigerated after quenching and before tempering. Source: Ref 63

Table 8 Core mechanical properties of Pyrowear Alloy 53

Test temperature		Yield strength		Ultimate tensile strength		Elongation in 4D, %	Reduction of area, %	Charpy V-notch impact	
°C	°F	MPa	ksi	MPa	ksi			J	ft · lbf
–54	–65	1103	160	1331	193	18	63	53–56	39–41
Room temperature		965	140	1172	170	16	66.5	118–129	87–95
100	212	. . .	. . .	. . .	. . .	. . .	. . .	140–163	103–120
177	350	896	130	1207	175	12	46	155–157	114–116

Source: Carpenter Technology Corporation

8620, and 4320 types. The compositions of some cast steels are selected by the steel producer in order to achieve the specified properties. For example, the ASTM A 148 Grade 120-95 alloy steel rack segment shown in Fig. 31 must have a minimum tensile strength of 827 MPa (120 ksi) and a minimum yield strength of 655 MPa (95 ksi). Table 9 lists mechanical properties for various heat treated cast steels.

Cast Irons

The term "cast iron," like the term "steel," identifies a large family of ferrous alloys. Cast irons primarily are alloys of iron that contain more than 2% C and from 1 to 3% Si. Wide variations in properties can be achieved by varying the balance between carbon and silicon, by alloying with various metallic or nonmetallic elements, and by varying melting, casting, and heat treating practices. Two types of cast irons are used for gears: gray cast irons and ductile cast irons.

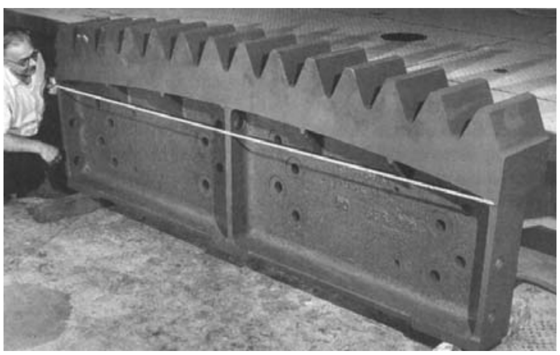


Fig. 31 Gear rack segment made from cast ASTM A 148 Grade 120-95 alloy steel

Gray iron refers to a broad class of ferrous casting alloys normally characterized by a microstructure of flake graphite in a ferrous matrix (Fig. 32). Gray irons are in essence iron-carbon-silicon alloys that usually contain 2.5 to 4% C, 1 to 3% Si, and additions of manganese, depending on the desired microstructure (as low as 0.1% Mn in ferritic gray irons and as high as 1.2% Mn in pearlitics). Sulfur and phosphorus are also present in small amounts as residual impurities.

Gray cast irons used for gears are classified in ASTM specification A 48 by their tensile strengths in ksi. As shown in Table 10, they range from class 20 (minimum tensile strength of 138 MPa or 20 ksi) to class 60 (minimum tensile strength of 414 MPa or 60 ksi). Generally, it can be assumed that the following properties of gray cast irons increase with increasing tensile strength from classes 20 to 60:

- All strengths, including strength at elevated temperature
- Ability to be machined to a fine finish
- Modulus of elasticity
- Wear resistance

On the other hand, the following properties decrease with increasing tensile strength, so that

Table 9 Properties of cast carbon and low-alloy steels

Class(a) and heat treatment		Tensile strength		Yield strength		Reduction in area %	Elongation, %	Hardness, HB	Endurance limit		Endurance ratio
		MPa	ksi	MPa	ksi				MPa	ksi	
Carbon steels											
60	A	434	63	241	35	54	30	131	207	30	0.48
65	N	469	68	262	38	48	28	131	207	30	0.44
70	N	517	75	290	42	45	27	143	241	35	0.47
80	NT	565	82	331	48	40	23	163	255	37	0.45
85	NT	621	90	379	55	38	20	179	269	39	0.43
100	QT	724	105	517	75	41	19	212	310	45	0.47
Alloy steels(b)											
65	NT	469	68	262	38	55	32	137	221	32	0.47
70	NT	510	74	303	44	50	28	143	241	35	0.47
80	NT	593	86	372	54	46	24	170	269	39	0.45
90	NT	655	95	441	64	44	20	192	290	42	0.44
105	NT	758	110	627	91	48	21	217	365	53	0.48
120	QT	883	128	772	112	38	16	262	427	62	0.48
150	QT	1089	158	979	142	30	13	311	510	74	0.47
175	QT	1234	179	1103	160	25	11	352	579	84	0.47
200	QT	1413	205	1172	170	21	8	401	607	88	0.43

(a) Class of steel based on tensile strength, in ksi. A = Annealed, N = Normalized, NT = Normalized and tempered, QT = Quenched and tempered. (b) Below 8% total alloy content. Source: Ref 64 low-strength irons often perform better than high-strength irons when these properties are important:

- Machinability
- Resistance to thermal shock
- Damping capacity
- Ability to be cast in thin sections

Gray cast iron has long been used as a gear material (Ref 65). Cast iron is low in cost and it can be cast easily into any desired shape for the rim, web, and hub of a gear. Cast iron machines easily.

Cast iron gears generally show good resistance to wear and are often less sensitive to lubrication inadequacies than are steel gears. Cast iron has good dampening qualities.

Cast-iron gear teeth have about three-quarters of the surface load-carrying capacity of steel gears of the same pitch diameter and face width. Their bending strength capacity is about one-third that of steel gears of the same normal diametral pitch. By making cast-iron gears somewhat coarser in pitch for the same pitch diameter, it is possible to design cast-iron gear sets about the same size as steel gear sets for a given application.

Gray cast iron has low impact strength and should not be used where severe shock loads occur.

Ductile cast iron, also known as nodular iron or spheroidal-graphite (SG) cast iron, is cast iron in which the graphite is present as tiny spheres (nodules) (see Fig. 33). In ductile iron, eutectic graphite separates from the molten iron during solidification in a manner similar to that in which eutectic graphite separates in gray cast iron. However, because of additives introduced in the molten iron before casting, the graphite grows as spheres, rather than as flakes of any of

Fig. 32 Flake graphite in a pearlitic matrix in a class 30 as-cast gray iron

Table 10 Minimum hardness and tensile strength requirements for gray cast irons

ASTM class number	Brinell hardness	Tensile strength MPa	Tensile strength ksi
20	155	140	20
30	180	205	30
35	205	240	35
40	220	275	40
50	250	345	50
60	285	415	60

Source: ASTM A 48

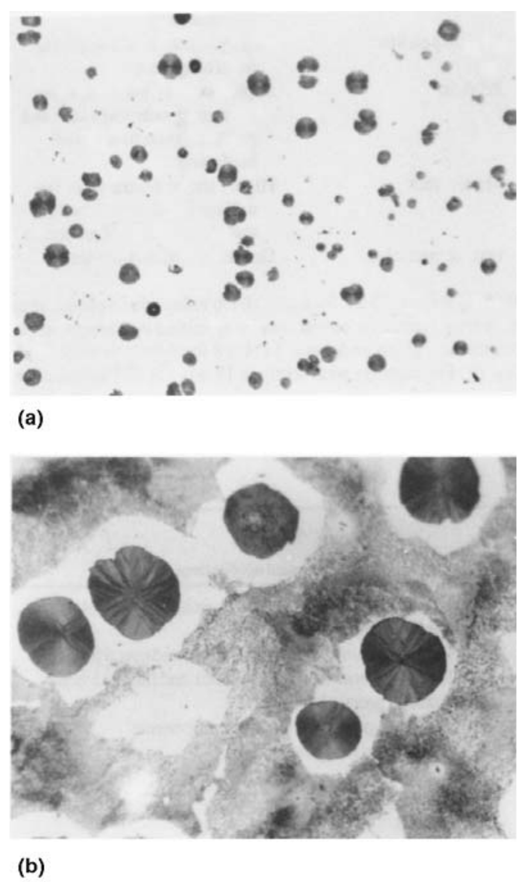


Fig. 33 Spheroidal graphite in an unetched ductile iron matrix shown at 75× (a) and in the etched (picral) condition shown at 300× (b). Etching reveals that the matrix consists of ferritic envelopes around the graphite nodules (bull's-eye structure) surrounded by a pearlitic matrix.

the forms characteristic of gray iron. Cast iron containing spheroidal graphite is much stronger and has higher elongation than gray iron. The flakes in gray cast iron act as stress risers and make the impact strength and fatigue strength low. The nodules in ductile iron exhibit less of a stress-riser effect. Ductile iron has good impact strength and enough ductility to have elongations in the range of 2 to 15%, depending on the class. The fatigue strength of ductile iron can approach that of steel of equal hardness.

Most of the specifications for standard grades of ductile iron used for gears are based on properties. That is, strength and/or hardness is specified for each grade of ductile iron, and composition is either loosely specified or made subordinate to mechanical properties. As shown in Table 11, the ASTM system for designating the grade of ductile iron incorporates the numbers indicating tensile strength in ksi, yield strength in ksi, and elongation in percent. For example, grade 80-55-06 has a tensile strength of 522 MPa or 80 ksi, a yield strength of 379 MPa or 55 ksi, and an elongation of 6%.

Ductile iron may be used as-cast or it may be heat treated to a wide range of strength and hardness levels. Common heat treatments include annealing, normalizing and tempering, quenching and tempering, and austempering (see discussion below). Figure 34 compares the strength and ductility of as-cast ductile iron with heat-treated ductile irons. The hardness levels in ductile irons can range from less than 160 HB to specified ranges greater than 300 HB.

If ductile iron is austenitized and quenched in a salt bath or a hot oil transformation bath at a temperature of 320 to 550 °C (610 to 1020 °F) and held at this temperature, it transforms to a structure containing mainly bainite with a minor proportion of austenite (Fig. 35). Irons that are transformed in this manner are called *austempered ductile irons.* Austempering generates a range of structures, depending on the time of transformation and the temperature of the transformation bath. The properties are characterized by very high strength, some ductility and toughness, and often an ability to work harden, which gives appreciably higher wear resistance than that of other ductile irons. As shown in Fig. 34 and Table 12, austempered ductile irons exhibit in excess of 5% elongation at tensile strengths exceeding 1000 MPa (145 ksi).

Most applications for austempered ductile iron gears are in transportation equipment: automobiles, trucks, and railroad and military vehicles. The same improved performance and cost savings are expected to make these materials attractive in equipment for other industries, such as mining, earthmoving, agriculture, construction, and machine tools.

P/M Irons and Steels

Both pressed-and-sintered and powder forged iron-base gears are produced for a variety of applications. Most P/M gears are made from iron-copper steels or iron-nickel steels. These steels can also undergo surface hardening treatments such as carburizing and carbonitriding. More detailed information on P/M materials for gears can be found in Chapter 7, "Powder Metallurgy."

Sintered Steels. The copper-bearing steels in the as-sintered condition have a tensile strength range of 170 to 570 MPa (25 to 82 ksi), an elongation of 2.0% or less, and an apparent hardness of 11 to 80 HRB. Heat-treated sintered copper steels have a tensile strength range of 450 to 720 MPa (65 to 105 ksi), an elongation less than 0.5%, and an apparent hardness of 58 to 60 HRC. The nickel steels in the as-sintered condition have a tensile strength range of 170 to 620 MPa (25 to 90 ksi), an elongation of 10% or less (average values are about 4%), and apparent hardness of 55 to 88 HRB. Heat-treated sintered nickel steels have a tensile strength range of 590 to 1340 MPa (85 to 195 ksi), and an elongation of less than 0.5%.

Table 11 Minimum mechanical property requirements for ductile cast irons

ASTM grade designation	Recommended heat treatment and microstructure	Brinell hardness range	Tensile strength		Yield strength		Elongation in 50 mm (2 in.), % min
			MPa	ksi	MPa	ksi	
60-40-18	Annealed ferritic	170 max.	415	60	275	40	18.0
65-45-12	As-cast or annealed ferritic-pearlitic	156–217	450	65	310	45	12.0
80-55-06	Normalized ferritic-pearlitic	187–255	550	80	380	55	6.0
100-70-03	Quench and tempered pearlitic	241–302	690	100	485	70	3.0
120-90-02	Quench and tempered martensitic	Range specified	830	120	620	90	2.0

Source: ASTM A 536

Powder forged nickel steels for gears are produced from 4600-series powders. The tensile strength for quenched-and-tempered powder forged 4600 materials can exceed 1500 MPa (220 ksi). Powder forged steels have densities approaching those of wrought steels.

Stainless Steels

Stainless steels are iron-base alloys that contain a minimum of approximately 11% Cr, the amount needed to prevent the formation of rust in unpolluted atmospheres (hence the designation stainless). Few stainless steels contain more than 30% Cr or less than 50% Fe. They achieve their stainless characteristics through the formation of an invisible and adherent chromium-rich oxide surface film. Unfortunately, the thin oxide layer on stainless steels is hard and brittle and breaks up easily under sliding loads, exposing the bare metal and thus promoting wear. Nevertheless, stainless steels gears are used in some applications where corrosion resistance is critical, for example, food processing machinery. Austenitic (types 303, 304, and 316), martensitic (type 440C), and precipitation-hardening (17-4PH and 17-7PH) stainless steels have been used as gear materials. Type 304 (18Cr-8Ni), which is one of the most widely used stainless steels, is used for gears in juice extractors and fishing reels.

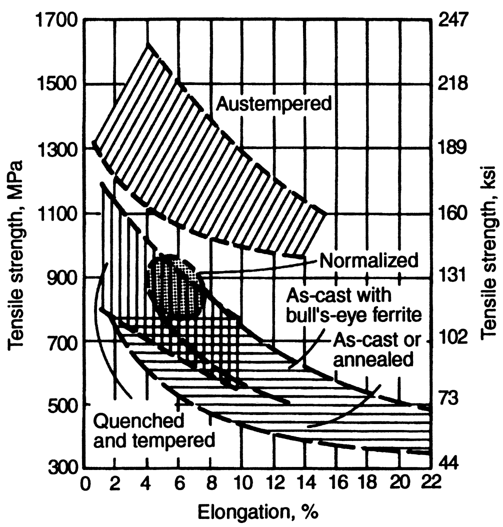


Fig. 34 Strength and ductility ranges of as-cast and heat-treated ductile irons

Fig. 35 Austempered ductile iron structure consisting of spheroidal graphite in a matrix of bainitic ferritic plates (dark) and interplate austenitic (white)

Tool Steels

Tool steels are normally used for machining and metal forming operations. The two types of tool steels that are occasionally used for gears are high-speed tool steels and hot-work tool steels. More detailed information on these materials can be found in the *ASM Specialty Handbook: Tool Materials.*

High-speed tool steels are so named primarily because of their ability to machine materials at high cutting speeds. They are complex iron-base alloys of carbon, chromium, vanadium, molybdenum, or tungsten, or combinations thereof, and in some cases substantial amounts of cobalt. The carbon and alloy contents are balanced at levels to give high attainable hardening response, high wear resistance, high resistance to the softening effect of heat, and good toughness.

Hot-work tool steels (group H steels) have been developed to withstand the combinations of heat, pressure, and abrasion associated with various metalworking operations such as punching, shearing, and other forming methods. Group H tool steels usually have medium carbon contents (0.35 to 0.45%) and chromium, tungsten, molybdenum, and vanadium contents of 6 to 25%.

Maraging Steels

Maraging steels are a special class of high-strength steels that differ from conventional steels in that they are hardened by a metallurgical reaction that does not involve carbon. Instead, these steels are strengthened by the precipitation of intermetallic compounds at temperatures of about 480 °C (900 °F). The term maraging is derived from martensite age hardening of a low-carbon, iron-nickel lath martensite matrix. Commercial maraging steels are designed to provide specific levels of yield strength from 1030 to 2420 MPa (150 to 350 ksi) with some having yield strengths as high as 3450 MPa (500 ksi). These steels typically have very high nickel, cobalt, and molybdenum contents and very low carbon contents. The use of maraging steels for gears is discussed in Chapter 8, "Through Hardening" (refer to the case history that describes the design and manufacture of a gear rack).

Nonferrous Alloys

Copper-base alloys account for much of the nonferrous gear materials used. Die cast aluminum-, zinc-, and magnesium-base alloys for gears are also produced but these materials have largely been replaced by plastics for low-load applications.

Bronzes

A family of four bronzes are widely used for power transmission gearing. Most of these are used in worm gears and are mated against steel gears. The four gear bronzes are:

- Tin bronzes
- Manganese bronzes
- Aluminum bronzes
- Silicon bronzes

Tin bronzes, also referred to as phosphor bronzes, are copper-tin or copper-tin-lead (leaded tin bronzes) that are deoxidized with phosphorus. These alloys are tough and have good corrosion resistance. They possess excellent wear resistance which permits their use in gears and worm wheels for severe wear applications. The most commonly used alloy in this group is C90700, or gear bronze, which contains 89% Cu and 11% Sn. Nominal compositions and properties of tin bronzes and leaded tin bronzes are listed in Table 13. Composition and mechanical property requirements for tin bronzes used for gears are covered in ASTM standard B 427.

Manganese Bronzes. This is the name given to a family of high-strength brass alloys (brasses are alloys which contain zinc as the principal alloying element). They are characterized by high strength and hardness and they are the toughest materials in the bronze family. They achieve mechanical properties through alloying without heat treatment. These bronzes have the same strength and ductility as annealed cast steels. Table 13 lists compositions and properties for cast manganese bronzes. Table 14 lists the composition and properties of a wrought manganese bronze containing 36.5% Zn. Cast and wrought manganese bronzes have good wear resistance but do not possess the same degree of corrosion resistance, wearability, or bearing quality as the tin bronzes or aluminum bronzes.

Aluminum bronzes are similar to the manganese bronzes in toughness, but are lighter in

Table 12 ASTM standard A 897-90 and A 897M-90 mechanical property requirements of austempered ductile iron

	Tensile (min)		Yield (min)			Impact(a)		
Grade	MPa	ksi	MPa	ksi	Elongation, %	J	ft · lbf	Hardness, HB(c)
125-80-10	. . .	125	. . .	80	10	. . .	75	269–321
850-550-10	850	. . .	550	. . .	10	100	. . .	269–321
150-100-7	. . .	150	. . .	100	7	. . .	60	302–363
1050-700-7	1050	. . .	700	. . .	7	80	. . .	302–363
175-125-4	. . .	175	. . .	125	4	. . .	45	341–444
1200-850-4	1200	. . .	850	. . .	4	60	. . .	341–444
200-155-1	. . .	200	. . .	155	1	. . .	25	388–477
1400-1100-1	1400	. . .	1100	. . .	1	35	. . .	388–477
230-185	. . .	230	. . .	185	(b)	. . .	(b)	444–555
1600-1300	1600	. . .	1300	. . .	(b)	(b)	. . .	444–555

(a) Unnotched Charpy bars tested at 72 ± 7 °F (22 ± 4 °C). The values in the table are a minimum for the average of the highest three test values of four tested samples. (b) Elongation and impact requirements are not specified. Although grades 200-155-1, 1400-1100-1, 230-185, and 1600-1300 are primarily used for gear and wear resistance applications, grades 200-155-1 and 1400-1100-1 have applications where some sacrifice in wear resistance is acceptable in order to provide a limited amount of ductility and toughness. (c) Hardness is not mandatory and is shown for information only.

weight and attain higher mechanical properties through heat treatment (refer to Table 13). As the strength of aluminum bronzes is increased, their ductility decreases. Bearing characteristics of aluminum bronzes are better than manganese bronzes but inferior to tin bronzes. Aluminum bronzes are available in both cast and wrought (Table 14) forms.

Silicon bronzes, also referred to as silicon brasses, are commonly used in lightly loaded gearing for electrical applications because of their low cost and nonmagnetic properties. As shown in Table 13, these alloys contain 14% Zn with silicon additions of 3 to 4%.

Other copper-base alloys used for gearing include yellow brass, an alloy containing

Table 13 Compositions and properties of cast gear bronzes

UNS designation	Nominal composition, %	Typical mechanical properties, as cast (heat treated): Tensile strength MPa	Tensile strength ksi	Yield strength MPa	Yield strength ksi	Elongation in 50 mm (2 in.), %	Brinell hardness 500 kg	Brinell hardness 3000 kg	Machinability rating(a)	Casting types(b)
Manganese bronzes										
C86100	67Cu, 21Zn, 3Fe, 5Al, 4Mn	655	95	345	50	20	...	180	30	C, I, P, S
C86200	64Cu, 26Zn, 3Fe, 4Al, 3Mn	655	95	331	48	20	...	180	30	C, T, D, I, P, S
C86300	63Cu, 25Zn, 3Fe, 6Al, 3Mn	793	115	572	83	15	...	225	8	C, T, I, P, S
C86400	59Cu, 1Pb, 40Zn	448	65	172	25	20	90	105	65	C, D, M, P, S
C86500	58Cu, 0.5Sn, 39.5Zn, 1Fe, 1Al	490	71	193	28	30	100	130	26	C, T, I, P, S
Silicon bronzes										
C87400	83Cu, 14Zn, 3Si	379	55	165	24	30	70	100	50	C, D, I, M, P, S
C87500	82Cu, 14Zn, 4Si	462	67	207	30	21	115	134	50	C, D, I, M, P, S
Tin bronzes										
C90700	89Cu, 11Sn	303 (379)	44 (55)	152 (207)	22 (30)	20 (16)	80 (102)	... .	20	C, T, I, M, S
C91600	88Cu, 10.5Sn, 1.5Ni	303 (414)	44 (60)	152 (221)	22 (32)	16 (16)	85 (106)	... ...	20	C, T, M, S
C91700	86.5Cu, 12Sn, 1.5Ni	303 (414)	44 (60)	152 (221)	22 (32)	16 (16)	85 (106)	... ...	20	C, T, I, M, S
Leaded tin bronzes										
C92500	87Cu, 11Sn, 1Pb, 1Ni	303	44	138	20	20	80	...	30	C, T, M, S
C92700	88Cu, 10Sn, 2Pb	290	42	145	21	20	77	...	45	C, T, S
C92900	84Cu, 10Sn, 2.5Pb, 3.5Ni	324 (324)	47 (47)	179 (179)	26 (26)	20 (20)	80 (80)	... ...	40	C, T, M, S
Aluminum bronzes										
C95200	88Cu, 3Fe, 9Al	552	80	186	27	35	...	125	50	C, T, M, P, S
C95300	89Cu, 1Fe, 10Al	517 (586)	75 (85)	186 (290)	27 (42)	25 (15)	... ...	140 (174)	55	C, T, M, P, S
C95400	85Cu, 4Fe, 11Al	586 (724)	85 (105)	241 (372)	35 (54)	18 (8)	... ...	170 (195)	60	C, T, M, P
C95500	81Cu, 4Ni, 4Fe, 11Al	689 (827)	100 (120)	303 (469)	44 (68)	12 (10)	... ...	192 (230)	50	C, T, M, P, S

(a) Free-cutting brass = 100. (b) C, centrifugal; T, continuous; D, die; I, investment; M, permanent mold; P, plaster; S, sand

Table 14 Compositions and properties of wrought gear bronzes

UNS designation	Nominal composition, %	Tensile strength MPa	Tensile strength ksi	Yield strength MPa	Yield strength ksi	Elongation in 50 mm (2 in.), % min	Hardness HB and HRB
C62300	87 Cu, 3 Fe, 10 Al	620	90	310	45	25	180 HB (1000 kgf)
C62400	86 Cu, 3 Fe, 11 Al	655	95	345	50	12	200 HB (3000 kgf)
C63000	82 Cu, 3 Fe, 10 Al, 5 Ni	620	90	310	45	17	100 HRB
C64200	91.2 Cu, 7 Al	640	93	415	60	26	90 HRB
C67300	60.5 Cu, 36.5 Zn, 1.2 Al, 2.8 Mn, 1 Si	485	70	275	40	25	70 HRB

Note: Typical mechanical properties vary with form, temper, and section size considerations nominally 65% Cu and 35% Zn, and copper alloys produced by P/M processing (refer to Table 3 in Chapter 7, "Powder Metallurgy").

Other Nonferrous Alloys

The following alloys are also used for gears:

- High-strength wrought aluminum alloys such as 2024 (Al-4.4Cu-1.5Mg-0.6Mn), 6061 (Al-1.0Mg-0.6Si-0.30Cu-0.20Cr), and 7075 (Al-5.6Zn-2.5Mg-1.6Cu-0.23Cr)
- Die cast aluminum-silicon alloys such as 360.0 and A360 (Al-9.5Si-0.5Mg), 383.0 (Al-10.5Si-2.5Cu), 384.0 (Al-11.2Si-3.8Cu), and 413.0 (Al-12Si)
- Die cast zinc-aluminum alloys such as ZA-8 (Zn-8Al-1Cu-0.02Mg), ZA-12 (Zn-11Al-1Cu-0.025Mg), and ZA-27 (Zn-27Al-2Cu-0.015Mg)
- Die cast magnesium-aluminum alloys such as AZ91A and AZ91B (Mg-9Al) and AM60A and AM60B (Mg-6Al)

Properties of these die cast alloys can be found in *Properties and Selection: Nonferrous Alloys and Special-Purpose Materials,* Vol 2, of the *ASM Handbook.*

Because of their high strength-to-weight ratio, titanium alloys, such as Ti-6Al-4V, have also been used for gears. The poor tribological properties of titanium alloys, however, limits their use to low-load applications. Wear-resistant coatings produced by ion implantation or diffusion treatments (oxidizing and nitriding) can increase the wear resistance of titanium alloys.

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REFERENCES

1. N.P. Chirons, Ed., *Gear Design and Application,* McGraw-Hill, 1967, p 269–274
2. Hardenability of Steels, *ASM Specialty Handbook: Carbon and Alloy Steels,* J.R. Davis, Ed., ASM International, 1996, p 138–166
3. R.E. Cohen, P.J. Haagensen, D.K. Matlock, and G. Krauss, "Assessment of Bending Fatigue Limits for Carburized Steel," Technical Paper 910140, SAE International, 1991
4. D.V. Doane, Carburized Steel—Update on a Mature Composite, *Carburizing: Processing and Performance,* G. Krauss, Ed., ASM International, 1989, p 169–190
5. H.O. Fuchs and R.I. Stephens, *Metal Fatigue in Engineering,* John Wiley & Sons, 1980
6. R.W. Landgraf and R.H. Richman, Fatigue Behavior of Carburized Steel, *Fatigue of Composite Materials,* STP 569, American Society for Testing and Materials (ASTM), 1975, p 130–144
7. H. Brugger and G. Krauss, Influence of Ductility on the Behavior of Carburizing Steel during Static and Dynamic Bend Testing, *Arch. Eisenhuttenwes.,* Vol 32, 1961, p 529–539
8. R.E. Cohen, D.K. Matlock, and G. Krauss, Specimen Edge Effects on Bending Fatigue of Carburized Steel, *J. Mater. Eng. Perform.,* Vol 1 (No. 5), 1992, p 695–703
9. D.H. Breen and E.M. Wene, Fatigue in Machines and Structures—Ground Vehicles, *Fatigue and Microstructure,* American Society for Metals, 1979, p 57–99
10. Z. Zurn and C. Razim, On the Fatigue Strength of Case Hardened Parts, *Carburizing: Processing and Performance,* G. Krauss, Ed., ASM International, 1989, p 239–248
11. M.B. Slane, R. Buenneke, C. Dunham, M. Semenek, M. Shea, and J. Tripp, "Gear Single Tooth Bending Fatigue," Technical Paper 821042, SAE International, 1982
12. D. Medlin, G. Krauss, D.K. Matlock, K. Burris, and M. Slane, "Comparison of Single Gear Tooth and Cantilever Beam Bend Fatigue Testing of Carburized Steel," Technical Paper 950212, SAE International, 1995
13. R.S. Hyde, R.E. Cohen, D.K. Matlock, and G. Krauss, "Bending Fatigue Crack Characterization and Fracture Toughness of Gas Carburized SAE 4320 Steel," Technical Paper 920534, SAE International, 1992
14. M.A. Zaccone, J.B. Kelley, and G. Krauss, Strain Hardening and Fatigue of Simulated

Case Microstructures in Carburized Steel, *Carburizing: Processing and Performance,* G. Krauss, Ed., ASM International, 1989, p 249–265
15. M.M. Shea, "Impact Properties of Selected Gear Steels," Technical Paper 780772, SAE International, 1978
16. G. Krauss, The Relationship of Microstructure to Fracture Morphology and Toughness of Hardened Hypereutectoid Steels, *Case Hardened Steels: Microstructure and Residual Stress Effects,* TMS/AIME, 1984, p 33–56
17. R.S. Hyde, "Quench Embrittlement and Intergranular Oxide Embrittlement: Effects on Bending Fatigue Initiation of Gas-Carburized Steel," Ph.D. dissertation, Colorado School of Mines, 1994
18. L. Magnusson and T. Ericsson, Initiation and Propagation of Fatigue Cracks in Carburized Steel, *Heat Treatment '79,* The Metals Society, London, 1979, p 202–206
19. J.L. Pacheco and G. Krauss, Microstructure and High Bending Fatigue Strength in Carburized Steel, *J. Heat Treat.,* Vol 7 (No. 2), 1989, p 77–86
20. K.A. Erven, D.K. Matlock, and G. Krauss, Effect of Sulfur on Bending Fatigue of Carburized Steel, *J. Heat Treat.,* Vol 9, 1991, p 27–35
21. G. Krauss, The Microstructure and Fracture of a Carburized Steel, *Metall. Trans. A,* Vol 9, 1978, p 1527–1535
22. R.S. Hyde, G. Krauss, and D.K. Matlock, Phosphorus and Carbon Segregation: Effects on Fatigue and Fracture of Gas-Carburized Modified 4320 Steel, *Metall. Trans. A,* Vol 25, 1994, p 1229–1240
23. G. Krauss, *Steels: Heat Treatment and Processing Principles,* ASM International, 1990
24. G. Krauss, Heat Treated Martensitic Steels: Microstructural Systems for Advanced Manufacture, *ISIJ Int.,* Vol 35 (No. 4), 1995, p 349–359
25. K.A. Erven, D.K. Matlock, and G. Krauss, Bending Fatigue and Microstructure of Gas-Carburized Alloy Steels, *Mat. Sci. Forum,* Vol 102–104, 1992, p 183–198
26. R.E. Cohen, G. Krauss, and D.K. Matlock, "Bending Fatigue Performance of Carburized 4320 Steel," Technical Paper 930963, SAE International, 1993
27. B. Thoden and J. Grosch, Crack Resistance of Carburized Steel under Bend Stress, *Carburizing: Processing and Performance,* G. Krauss, Ed., ASM International, 1989, p 303–310
28. R.S. Hyde, D.K. Matlock, and G. Krauss, "The Effect of Reheat Treatments on Fatigue and Fracture of Carburized Steel," Technical Paper 940788, SAE International, 1994
29. S. Gunnarson, Structure Anomalies in the Surface Zone of Gas-Carburized Case-Hardened Steel, *Met. Treat. Drop Forg.,* Vol 30, 1963, p 219–229
30. T. Naito, H. Ueda, and M. Kikuchi, Fatigue Behavior of Carburized Steel with Internal Oxides and Nonmartensitic Microstructures Near the Surface, *Metall. Trans. A,* Vol 15, 1984, p 1431–1436
31. W.E. Dowling, Jr., W.T. Donlon, W.B. Copple, and C.V. Darragh, Fatigue Behavior of Two Carburized Low Alloy Steels, *1995 Carburizing and Nitriding with Atmospheres,* J. Grosch, J. Morral, and M. Schneider, Ed., ASM International, 1995, p 55–60
32. C. Razim, Uber den Einfluss von Restaustenit auf des Festigkeitsverhalten Einsatzgeharteter Probenkorper bei Schwin-gender Bean-spruchung, *Harterei-Tech. Mitt.,* Vol 23, 1968, p 1–8
33. R.H. Richman and R.W. Landgraf, Some Effects of Retained Austenite on the Fatigue Resistance of Carburized Steel, *Metall. Trans. A,* Vol 6, 1975, p 955–964
34. M. Zaccone and G. Krauss, Elastic Limit and Microplastic Response of Hardened Steels, *Metall. Trans. A,* Vol 24, 1993, p 2263–2277
35. G.B. Olson, Transformation Plasticity and the Stability of Plastic Flow, *Deformation, Processing, and Structure,* G. Krauss, Ed., ASM International, 1984, p 391–424
36. Final Report, Advanced Rotorcraft Transmission Program, National Aeronautics and Space Administration, Cleveland, Ohio
37. R.J. Johnson, The Role of Nickel in Carburizing Steels, *Met. Eng. Quart.,* Vol 15 (No. 3), 1975, p 1–8
38. K.D. Jones and G. Krauss, Effects of High-Carbon Specimen Corners on Microstructure and Fatigue of Partial Pressure Carburized Steels, *Heat Treatment '79,* The Metals Society, London, 1979, p 188–193
39. C. Kim, D.E. Diesburg, and R.M. Buck, Influence of Sub-Zero and Shot-Peening Treatment on Impact and Fatigue Fracture

Properties of Case-Hardened Steels, *J. Heat Treat.,* Vol 2 (No. 1), 1981, p 43–53
40. M.A. Panhans and R.A. Fournelle, High Cycle Fatigue Resistance of AISI E9310 Carburized Steel with Two Different Levels of Surface Retained Austenite and Surface Residual Stress, *J. Heat Treat.,* Vol 2 (No. 1), 1981, p 55–61
41. J. Grosch and O. Schwarz, Retained Austenite and Residual Stress Distribution in Deep Cooled Carburized Microstructures, *1995 Carburizing and Nitriding with Atmospheres,* J. Grosch, J. Morral, and M. Schneider, Ed., ASM International, 1995, p 71–76
42. M.G. Mendiratta, J. Sasser, and G. Krauss, Effect of Dissolved Carbon on Microcracking in Martensite of an Fe-1.39 pct C Alloy, *Metall. Trans.,* Vol 3, 1972, p 351–353
43. R.P. Brobst and G. Krauss, The Effect of Austenite Grain Size on Microcracking in Martensite of an Fe-1.22C Alloy, *Metall. Trans.,* Vol 5, 1975, p 457–462
44. L.J. Ebert, The Role of Residual Stresses in the Mechanical Performance of Case Carburized Steel, *Metall. Trans. A,* Vol 9, 1978, p 1537–1551
45. D.P. Koistinen, The Distribution of Residual Stresses in Carburized Steels and Their Origin, *Trans. ASM,* Vol 50, 1938, p 227–241
46. B. Liscic, State of the Art in Quenching, *Quenching and Carburizing,* The Institute of Materials, London, 1993, p 1–32
47. A.K. Hellier, M.B. McGirr, S.H. Alger, and M. Stefulji, Computer Simulation of Residual Stresses during Quenching, *Quenching and Carburizing,* The Institute of Materials, London, 1993, p 127–138
48. G. Totten, Ed., *Quenching and Distortion Control,* ASM International, 1992
49. G. Parrish and G.S. Harper, *Production Gas Carburizing,* Pergamon Press Inc., 1985
50. G. Krauss, Microstructure, Residual Stresses and Fatigue of Carburized Steels, *Quenching and Carburizing,* The Institute of Materials, London, 1993, p 205–225
51. R.P. Garibay and N.S. Chang, Improved Fatigue Life of a Carburized Gear by Shot Peening Parameter Optimization, *Carburizing: Processing and Performance,* G. Krauss, Ed., ASM International, 1989, p 283–289
52. H. Swahn, P.C. Becker, and O. Vingsbo, Martensite Decay during Rolling Contact Fatigue in Ball Bearings, *Metall. Trans. A,* Vol 7, 1976, p 1099–1110
53. J.A. Martin, S.F. Borgese, and A.D. Eberhardt, Microstructural Alterations of Roller-Bearing Steel Undergoing Cyclic Stresses, *J. Basic Eng.,* 1966, p 555–567
54. V. Bhargava, G.T. Hahn, and C.A. Rubin, Rolling Contact Deformation, Etching Effects, and Failure of High Strength Bearing Steel, *Metall. Trans. A,* Vol 21, 1990, p 1921–1931
55. C.A. Stickels, Rolling Contact Fatigue Tests of 52100 Bearing Steel Using a Modified NASA Ball Test Rig, *Wear,* Vol 98, 1984, p 199–210
56. L. Kiessling, Rolling-Contact Fatigue of Carburized and Carbonitrided Steels, *Heat Treat. Met.,* Vol 7 (No. 4), 1980, p 97–101
57. C.V. Darragh, Engineered Gear Steels: A Review, *Gear Technology,* November/December, 2002, p 33–40
58. C.A. Stickels, Carburizing, *Friction, Lubrication, and Wear Technology,* Vol 18, *ASM Handbook,* ASM International, 1992, p 873–877
59. *Modern Carburized Nickel Alloy Steels, Reference Book Series 11,005,* Nickel Development Institute, 1989
60. C. Razim, "Some Facts and Considerations of Trends in Gear Steels for the Automotive Industry," *Alloys for the Eighties,* Climax Molybdenum Company, 1980, p 9
61. T.B. Cameron, D.E. Diesburg, and C. Kim, Fatigue and Overload Fracture of a Carburized Case, *J. Metals,* May 1983
62. Y.E. Smith and D.E. Diesburg, Fracture Resistance in Carburizing Steels, Parts I, II, III, *Met. Prog.,* May, June, July 1979
63. D.E. Diesburg, "High-Hardenability Carburizing Steels for Rock Bits," *Micon 78: Optimization of Processing, Properties, and Service Performance Through Microstructural Control,* STP 672, ASTM, 1979
64. *Steel Castings Handbook,* 6th ed., M. Blair and T.L. Stevens, Ed., Steel Founders' Society of America and ASM International, 1995, p 18–8
65. *Dudley's Gear Handbook,* 2nd ed., D.P. Townsend, Ed., McGraw-Hill, Inc., p 8–27

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CHAPTER 4

Plastics

PLASTIC GEARS are continuing to displace metal gears in applications ranging from automotive components to office automation equipment. In the past, plastic gears were not considered to be able to transmit power, were limited as to top operating speed, and were not considered able to transmit motion to a high degree of accuracy. However, improved materials with higher load capacities, advances in mold design and molding technology, and the development of reliable engineering data have led to the successful and increasing use of plastics for both motion-carrying gears and power transmission gears.

General Characteristics of Plastic Gears (Ref 1)

Advantages. Among the characteristics responsible for the large increase in plastic gear usage, the following are the most significant:

- Relative low cost (particularly for high-volume injection molded gears)
- Ease and speed of manufacture
- Wide range of configurations and complex shapes possible
- Elimination of machining and finishing operations
- Capability of fabrication with metal inserts and integral designs
- Lower density (light weight and low inertia)
- Ability to dampen moderate shock and impact
- Ability to operate with minimal or no lubrication
- Low coefficient of friction
- Smooth, quiet operation
- Lower critical tolerances than with metal gears, due in part to their greater resilience
- Resistance to corrosion

Limitations. There are also limitations of plastic gears relative to metal gears. These include:

- Maximum load-carrying capacity lower than metal gears
- Reduced ability to operate at elevated temperature. Operation is *generally* limited to less than 120 °C (250 °F). Cold temperature applications are also limited.
- Ambient temperature and temperatures at tooth contact surface must be limited
- Plastic gears cannot be molded to the same accuracy as high-precision machined/finished metal gears
- Plastic gears are subject to greater dimensional instabilities due to their greater coefficient of thermal expansion and moisture absorption
- Initial high mold costs in developing correct tooth form and dimensions
- Can be negatively affected by certain chemicals and even some lubricants
- Improper molding tools and process can produce residual internal stresses at the tooth roots resulting in overstressing and/or distortion with aging
- Cost of plastics are based on petrochemical pricing and thus are more volatile in comparison to metals

Classification of Plastics for Gear Applications

One very important classification scheme for plastics is based on their response to heat. There are two types of response possible, leading to the classification of plastics as thermoplastics or thermosets.

Thermoplastics are materials that repeatedly soften, or melt, when heated, and harden, or freeze, when cooled (Ref 2). Heating permits the intertwined molecular chains to slide relative to each other. At some higher temperature, the sliding is free enough for the materials to behave as a liquid and may be used to fill molds. The temperature at which this degree of softening occurs varies with the type and grade of plastic. Cooling restores the intermolecular bonds and the material behaves as a solid. However, these solid materials, to a varying degree, retain some aspects of a liquid in the form of viscoelastic behavior (see the section "Metals versus Plastics" in this chapter). Thermoplastics have greater toughness, or resistance to impact loads, than do thermosetting plastics. However, their upper temperature limit is only about 120 °C (250 °F). Above this temperature, thermoplastics lose about 50% of their rated room-temperature strength.

Thermosets are plastics that undergo chemical change during processing to become permanently infusible (Ref 2). If excessive heat is applied to a thermoset material after the chemical change has taken place, the plastic is degraded rather than melted. Before the processing, the molecular structure of the thermoset plastic is similar to that of a thermoplastic material. Heating permits the relative sliding of the molecules, the material takes on the properties of a liquid and can be used to fill molds. However, while still subjected to heat in the mold in a curing process, the intertwined molecules develop cross-links to form an irreversible network which prevents further relative sliding.

The resulting solid plastic behaves much more like an elastic material similar to metals. The viscoelastic behavior is much reduced from that of most thermoplastics. At the same time, these thermoset materials tend to be much less resistant to impact loading, and tend to be used only with some reinforcing medium in even moderate impact applications. On the other hand, thermosets tend to maintain their strength properties at much higher temperatures than most thermoplastics. Very little degradation of mechanical properties occurs at temperatures up to 230 °C (450 °F).

Metals versus Plastics

Designing with plastics is often a more complex task than designing with metals. Engineering plastics, which are viscoelastic, do not respond to mechanical stress in the linear, elastic manner for which most designers have developed an intuitive feel. In many applications, engineering plastics exhibit a more complex property mix than do metals. Because engineering plastics, as a family, are much younger than engineering metals, their database is not yet complete. In addition, their rapid evolution makes the material selection process more difficult.

A typical stress-strain curve for an engineering metal is shown in Fig. 1. The salient features of metal behavior are that the slope of the stress-strain curve is a constant up to the proportional limit and is known as the elastic modulus, and that the elastic modulus remains a constant over a wide range of strain rates and temperatures. A large number of design equations have been derived for use in the design of structural members such as beams, plates, and columns. All of these equations use the elastic modulus as a fundamental measure of the response of the material to stress.

However, if the designer attempts to use these equations for plastics part design using a modulus value taken from a plastics property specifications sheet, serious design errors will be made in many cases. The implicit assumption behind the design equations is that the material behaves in a linear, elastic manner. As already noted, this assumption is not true for plastics.

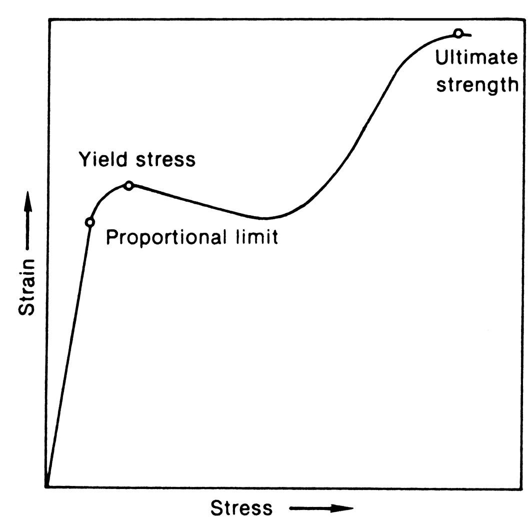


Fig. 1 Typical stress-strain curve for an engineering metal. Note high degree of linearity below the proportional limit.

When comparing the mechanical behavior of metals to that of plastics, as shown in Fig. 1 and 2, it can be seen that plastic, unlike metal, has no true proportional limit and that its stress-strain curve is not linear. Furthermore, the shape of the plastic stress-strain curve is greatly affected by the rate at which stress is applied, and stress-strain behavior is greatly affected by temperature.

Clearly, the mechanical response of plastic is more complex than that of metal. Because a plastic behaves in a manner that seems to combine elastic-solid and highly viscous liquid behaviors, it is said to be *viscoelastic.*

Plastic Gear Materials

This section will briefly review some of the commonly used plastic materials for gear applications (many more plastic formulations are anticipated for gears in the future). These materials can be used in the unmodified condition or modified with various additives to produce a wide variety of gear configurations including spur, worm, face, helical, bevel, internal, and elliptical gears, gear racks, and microgears (for computer memory devices). Selection of a material depends not only on its properties, but also on the manufacturing methods used, part shape and size, molding or machining characteristics, shrinkage rates, moisture absorption, and processing variables. Material suppliers and/or processors should always be consulted when choosing a plastic for a gear application.

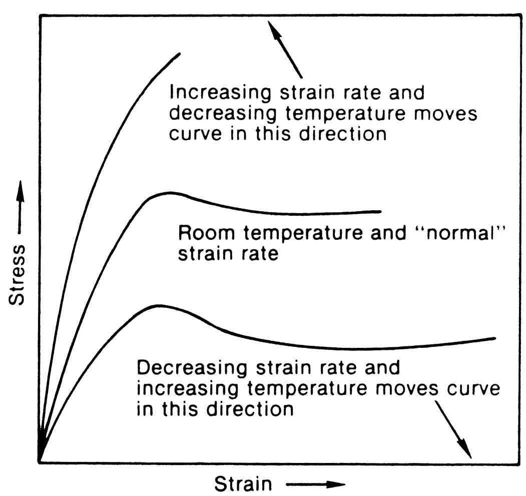


Fig. 2 Typical stress-strain curve for an engineering plastic. Note that there is no true proportional limit.

Additives

A wide variety of organic and inorganic materials are added to plastics either to reduce costs or property improvement. Some of these properties are (Ref 2):

- Strength
- Modulus (rigidity)
- Impact resistance
- Wear resistance
- Thermal conductivity
- Flame retardance
- Dimensional control
- Color
- Heat stability
- Noise reduction
- Oxidative stability
- Ultraviolet stability
- Lubricity
- Processability

Fillers are added to reduce costs, improve dimensional control (stability) during the molding operation, enhance conductivity, and improve heat resistance. Common mineral fillers include mica, talc, carbon powder, and glass beads. The amount of filler added can range from 5 to 40%. Organic fillers (<1%) are also sometimes used as a processing aide. Limitations of adding fillers include reduced impact resistance and increased tool wear.

Reinforcements are usually thought of as an added material whose purpose is to alter one or more of the mechanical properties of the basic material. Glass fibers (added in the range of 5 to >40%) are often used to alter both long- and short-term mechanical properties of the basic material resin. Heat and wear resistance are also improved. Limitations of adding glass fibers include increased distortion during molding, increased wear of the mating gear material, and increased tool wear.

Carbon fibers in the percentage range of 10 to >40% are added to enhance heat resistance, electrical properties, and wear resistance. Limitations of adding carbon fibers include higher costs, increased distortion during molding, and increased tool wear.

Kevlar (aramid) fibers in the percentage range of 5 to 20% are added to increase heat resistance, strength, and wear resistance and to reduce friction. The primary limitation of adding Kevlar fibers is the added cost. Some distortion may also occur during molding.

Lubricants. The following lubricants are added to plastics for the primary purpose of reducing friction and wear:

Lubricant	Typical percentage range added
Polytetrafluoroethylene (PTFE, or Teflon)	1–20%
Silicone	1–4%
Graphite	5–10%
Molybdenum disulfide (MoS_2)	2–5%

The primary limitation of adding PTFE is added cost. Graphite additions can reduce strength and impact resistance. There are no significant limiting factors for adding MoS_2.

Impact modifiers are additives that improve the impact strength or toughness of plastics with minimal impairment of other properties. They can be added in such large percentages (>50%) that they become a component in a blended material rather than an additive. To be effective, the added phase must be more impact resistant and have a lower modulus than the parent phase. Commonly used impact modifiers include acrylonitrile-butadiene-styrene (ABS), methacrylate-butadiene-styrene (MBS), acrylics, chlorinated polyethylene (CPE), and ethylene vinyl acetate (EVA).

Thermoplastic Gear Materials

Most of the plastics used for gear applications are thermoplastics. The two most commonly used thermoplastic formulations are the polyamides (nylons) and the acetals. These materials have been used in gears for more than 50 years in hundreds of diverse products including:

- Windshield wipers
- Automobile window lifts
- Automobile seat controls
- Speedometers
- Rotary pumps
- Appliances
- Electric garage door openers
- Small power tools
- Clocks
- Copy machines
- Laser printers

Nylons. This is a family of thermoplastic resins that were the first plastics used for gear applications. The most common grades used are Nylon 6, 6/6, and 6/12. These materials, which come in unmodified, toughened, and reinforced conditions, exhibit outstanding toughness and wear resistance, low coefficient of friction, and excellent electrical properties and chemical resistance. Properties of Nylons 6 and 6/6 are listed in Tables 1 (0.2% moisture content) and 2 (50% relative humidity). Some nylons are hygroscopic (absorb moisture) and this has a negative effect on their strengths (see Fig. 3 and compare Tables 1 and 2). In addition, the dimensional stability of these resins is poorer than other engineering plastics. Nylon gears can be manufactured by either machining or molding.

Acetal polymers have a lower water absorption rate than nylon and, therefore, are more stable after molding or machining. The acetals, which come in unmodified, toughened, reinforced, and internally lubricated conditions, are strong, have good resistance to creep and fatigue, have a low coefficient of friction, and are resistant to abrasion and chemicals. Properties of various acetal resins are listed in Table 3. The effect of temperature on the stress-strain curve for an unmodified acetal grade is shown in Fig. 4. The effect of internal lubricants on the sliding wear resistance of acetal resins is shown in Fig. 5. As this figure indicates, the PTFE-lubricated grades exhibited far greater wear resistance than the unmodified grades.

Other thermoplastics used for gears include:

- *Polycarbonates* are generally reinforced with glass fibers and lubricated with PTFE. Noted for their low-shrinkage characteristics, they are manufactured by either molding or machining.
- *Polyesters* are unmodified or reinforced with glass fibers. Applications for these molded resins are similar to the nylons and acetals. Polyesters can also be blended with elastomers for extra tough gears subject to extreme shock or when mesh noise reduction is required.
- *Polyurethanes* are unmodified, reinforced, or internally lubricated. These molded resins are noted for their flexibility, ability to absorb shock, and deaden sound.
- *Polyphenyline sulfides* are generally reinforced with glass fibers (up to 40%) with or without internal lubricants. These resins have higher elevated temperature strength than most other thermoplastics.
- *Liquid crystal polymer* resins contain mineral fillers or are reinforced with glass fibers. They are used for extremely thin or small

gears and exhibit both high-temperature resistance and chemical resistance and excellent dimensional stability.

Thermoset Gear Materials

Only a small number of thermoset plastics have been used for gears to any substantial degree. The properties of thermoplastics generally make them more suitable for gears. The different type of processing required for some thermosets also limits their use.

Polyimides, which are available commercially either as thermoplastic or thermoset resins, are used for high-temperature gear applications. Finish machined thermoset polyimide grades are unmodified or internally lubricated. Thermoplastic polyimide gears can be manufactured by either injection molding or machining. The thermoplastic grades are available in unmodified, glass reinforced (up to 65% glass fibers), toughened, and lubricated forms. Lubricants for thermoplastic polyimides include

Table 1 Property values for Nylons 6 and 6/6

Dry, as-molded, approximately 0.2% moisture content

			Nylon 6/6			
	Nylon 6				Toughened	
Property	Molding and extrusion compound	Glass fiber reinforced, 30–35%	Molding compound	Glass fiber reinforced, 30–33%	Unreinforced	Glass fiber reinforced, 33%
Mechanical						
Tensile strength at break, MPa (ksi)	. . .	165 (24)	94.5 (13.7)	193 (28)	50 (7.0)	140 (20.3)
Elongation at break, %	30–100	2.2–3.6	15–60	2.5–34	125	4–6
Tensile yield strength, MPa (ksi)	80.7 (11.7)	. . .	55 (8.00)	170 (25)	. . .	. . .
Compressive strength, rupture or yield, MPa (ksi)	90–110 (13–16)	131–165 (19–24)	86.2–103 (12.5–15.0)(a)	165–276 (24–40)	. . .	103 (15)
Flexural strength, rupture, or yield, MPa (ksi)	108 (15.7)	240 (35)	114–117 (16.5–17.0)	283 (41)	59 (8.5)	206 (29.9)
Tensile modulus, GPa (10^6 psi)	2.6 (0.38)	8.62–10.0 (1.25–1.45)	1.59–3.79 (0.23–0.55)	9.0 (1.3)	. . .	. . .
Flexural modulus at 23 °C (73 °F), GPa (10^6 psi)	2.7 (0.39)	9.65 (1.40)	2.8–3.1 (0.41–0.45)	8.96–10.0 (1.30–1.45)	1.65 (0.240)	7.58 (1.10)
Izod impact, 3.2 mm (1/8 in.) thick specimen, notched, J/m (ft · lbf/in.)	32–53 (0.6–1.0)	117–181 (2.2–3.4)	29–53 (0.55–1.0)	85–240 (1.6–4.5)	907 (17.0)	240 (4.5)
Rockwell hardness	R119	M93–96	R120	R101–119	R100	R107
Thermal						
Coefficient of linear thermal expansion, 10^{-6}/K	80–83	16–80	80	15–54	. . .	. . .
Deflection temperature under flexural load, °C (°F)						
At 1.82 MPa (0.264 ksi)	68–85 (155–185)	200–215 (392–420)	75–88 (167–190)	120–250 (252–490)	65 (150)	245 (470)
At 0.46 MPa (0.066 ksi)	185–190 (365–375)	215–220 (420–430)	230–246 (450–474)	125–260 (260–500)	. . .	260 (495)
Thermal conductivity, W/m · K (Btu · in./h · ft² · °F)	0.243 (1.69)	0.243–0.472 (1.69–3.27)	0.243 (1.69)	0.211–0.483 (1.46–3.35)	. . .	. . .
Physical						
Specify gravity	1.12–1.14	1.35–1.42	1.13–1.15	1.15–1.40	1.08	1.34
Water absorption, 3.2 mm (1/8 in.) thick specimen, 24 h, %	1.3–1.9	0.90–1.2	1.0–2.8	0.7–1.1	1.0	0.7
Saturation, %	8.5–10.0	6.4–7.0	8.5	5.5–6.5	. . .	. . .
Electrical						
Dielectric strength, 3.2 mm (1/8 in.) thick specimen, short time, MV/m (V/mil)	0.0157 (400)	15.7–17.7 (400–450)	0.0236 (600)	14.2–19.7 (360–500)	. . .	. . .

(a) Yield

graphite (15 to 40%), PTFE (10%) plus graphite (15%), and MoS_2 (up to 15%).

Phenolics are injection molded resins compounded with various mineral and glass fillers, glass fibers, and lubricants such as PTFE and graphite. Phenolics are used in applications requiring dimensional stability improved heat resistance.

Laminated Phenolic. Composite, or laminated thermosets, consist of a fibrous sheet material (paper, cotton fabric, nylon fabric, or glass fabric) that is impregnated or coated with a liquid phenolic resin binder and consolidated under high pressure and temperatures. By stacking and consolidating the unidirectional laminations (or plies) at different angles, the cut gear blanks/teeth will have equal strength in all directions.

Plastic Gear Manufacture

Plastic gears are manufactured by either machining or injection molding. Machining of plastic gears is performed by most of the same processes used in the machining of metal gears (see Chapter 5, "Machining, Grinding, and Finishing"). Machining may be selected over molding as the plastic gear manufacturing process for several reasons:

- The quantities may be too small to justify the tooling cost for molding
- The required accuracy or some special design feature (such as a very thick section) may be too difficult for molding
- The desired plastic material may not be suited to precision molding

Plastic materials to be machined into gears are available in a variety of forms, the most common of which are nonreinforced extruded circular rods or tubes. These rods are extruded by stock shape suppliers in diameters up to 200 mm (8 in.). Some of the properties of the material in extruded stock form can vary between the outer surface (or skin), which has formed through more rapid cooling, than the inner material (or core). Machining invariably leaves the gear teeth made from the core material, which generally has different strength, wear resistance, and chemical resistance. In addition, machining may expose voids in the stock. This is particularly true for fibrous materials, such as glass or carbon fiber, or hard granular materials, when used as additives in molded gears. These additives may be removed from the surface during machining to leave openings or surface roughness.

Injection Molding

Injection molding refers to a variety of processes that generally involve forcing or injecting a fluid plastic material into a closed mold. The process has been used to produce

Table 2 Property values for Nylons 6 and 6/6

Conditioned to 50% relative humidity

	Nylon 6		Nylon 6/6	
Property	Molding and extrusion compound	Glass fiber reinforced, 30–35%	Molding compound	Glass fiber reinforced, 30–33%
Tensile strength at break, MPa (ksi)	. . .	110 (16)	75 (11)	152 (22)
Elongation at break, %	300	. . .	150–300	5–7
Tensile yield strength, MPa (ksi)	51 (7.4)	. . .	45 (6.5)	. . .
Flexural strength, rupture or yield, MPa (ksi)	40 (5.8)	145 (21.0)	42 (6.1)	170 (25)
Tensile modulus, GPa (10^6 psi)	0.690 (0.100)	5.52 (0.800)	3.45 (0.500)	
Compressive modulus, GPa (10^6 psi)	1.70 (0.250)	. . .	. . .	. . .
Flexural modulus at 23 °C (73 °F), GPa (10^6 psi)	0.965 (0.140)	5.52–6.55 (0.800–0.950)	1.28 (0.185)	5.52 (0.800)
Izod impact, 3.2 mm ($^1/_8$ in.) thick specimen, notched, J/m (ft · lbf/in.)	160 (3.0)	197–292 (3.7–5.5)	45–112 (0.85–2.1)	138–159 (2.6–3.0)
Rockwell hardness	. . .	M78	. . .	. . .

plastic gears since the 1950s. American Gear Manufacturers Association (AGMA) quality levels ranging from 6 to 8 can be obtained (Ref 4). The process is particularly well suited for high-volume, automated production of thermoplastic gears. The development of manufacturing of injection molded gears requires the joint efforts of the gear design engineer, the mold producer, the molder, and the material supplier.

Injection-molding compounds are thermoplastic or thermosetting materials and their composites that are specifically formulated for

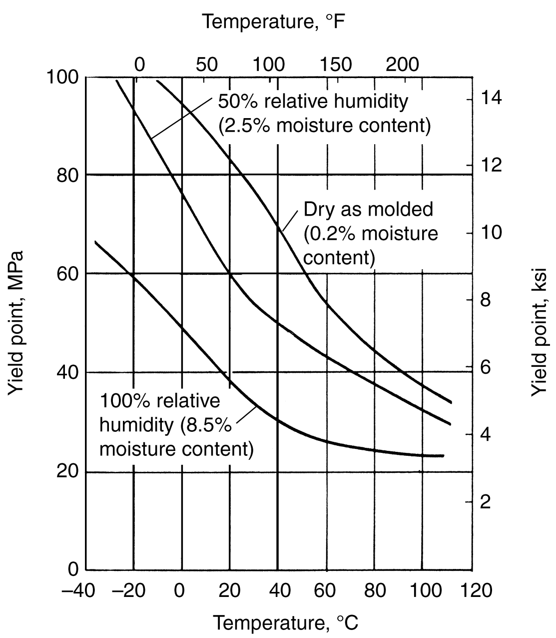


Fig. 3 Effect of temperature and moisture content of an unmodified nylon. Source: Ref 1

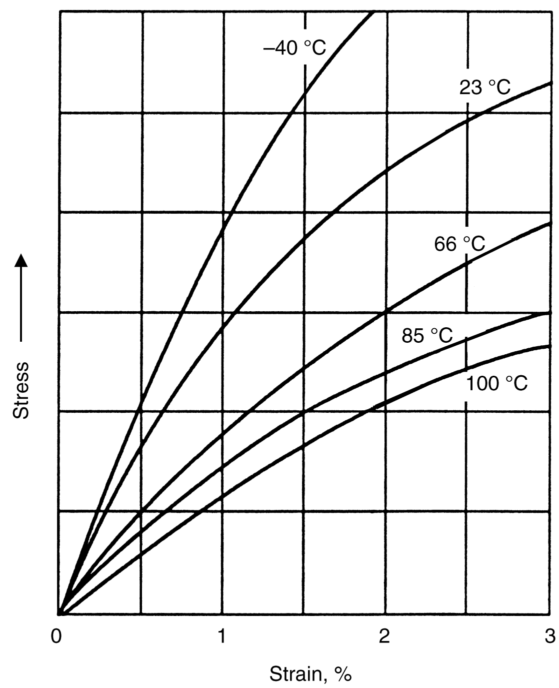


Fig. 4 Effect of temperature on the stress-strain behavior of an unmodified acetal. Source: Ref 2

Table 3 Mechanical properties of various Delrin acetal resins

Delrin is a registered trademark of DuPont Engineering Polymers

	Product description								
	Unmodified, general purpose			Enhanced crystallinity	Toughened		Glass reinforced	Lubricated PTFE	
Properties	100P	500P	900P	511P	100ST	500T	525GR	520MP	510MP
Tensile strength at yield MPa (ksi) ASTM D638	67 (9.7)	68 (9.9)	69 (10.0)	72 (10.4)	45 (6.5)	53 (7.7)	151(a) (21.9)(a)	54 (7.8)	61 (8.9)
Elongation at yield, % ASTM D638	23	15	11	11	35	15	NA(b)	12	10
Elongation at break, % ASTM D638	80	40	25	33	>150	75	3	14	13
Flexural modulus MPa (ksi) ASTM D790	2790 (410)	3100 (450)	3240 (470)	3300 (480)	1130 (160)	2250 (330)	8000 (1160)	3100 (450)	3160 (460)
Izod impact, notched J/m (ft-lb/in.) ASTM D256	120 (2.3)	75 (1.4)	69 (1.3)	73 (1.4)	NB	128 (2.4)	96 (1.8)	32 (0.6)	27 (0.5)
Izod impact, unnotched J/m (ft-lb/in.) ASTM D256	NB(c)	NB(c)	1630 (30.6)	NB(c)	NB(c)	NB(c)	1100 (20.6)	720 (13.5)	950 (17.8)

(a) Values are tensile strength at break since these materials do not yield. (b) Not applicable. (c) No break occurred. Material code: Delrin 100P, 500P, and 900P are unmodified acetals with high, medium, and low viscosities, respectively; Delrin 511P is a medium viscosity acetal; Delrin 100ST and 500ST have high and medium viscosities, respectively; Delrin 525GR is a medium viscosity, 25% glass-reinforced acetal; and Delrin 520MP and 510MP contain 20% and 10% PTFE, respectively.

the injection-molding process. This process requires materials capable of being fed into a molding machine, transported to accumulate pressure, injected through channels, and made to flow into a small opening in the mold. The process may cause major changes in both the physical and chemical properties of the molding compound. Because of their resistance to flow, neither high-molecular-weight resins nor long reinforcing fibers, or flakes, can be effectively manipulated through the molding process. Consequently, parts produced from molding compounds represent a compromise between optimal physical properties and the essential ability to flow under pressure. This compromise is offset by the ability to produce three-dimensional products with holes, ribs, and bosses, often without secondary operations or direct labor.

While the flow process has the potential to physically change the molding compound, these changes are not always negative. Improved homogeneity, better reinforcement wetting, and higher physical properties may be achieved if the response of the molding compound to the injection process conditions is well understood. Where this understanding is absent, reinforcements may be broken, and polymers may be either degraded or prematurely cured.

Injection Molding Machines. Although there are many variations in thermoplastic injection molding machines, the reciprocating-screw machine has become the most common type, and the following description of its operation illustrates the basic principles of all types. The functions performed by the machine include heating the plastic until it is able to flow readily under pressure, pressurizing this melt to inject it into a closed mold, holding the mold closed both during the injection and while the material is solidifying in the mold, and opening the mold to allow removal of the solid part.

As depicted in Fig. 6, the main components of the machine are the hopper, heated extruder barrel, reciprocating screw, nozzle, and mold clamp. The hopper feeds the unmelted thermoplastic, usually in pellet form, into the barrel. The hopper is often equipped with a desiccant-type drying system to remove moisture degra-

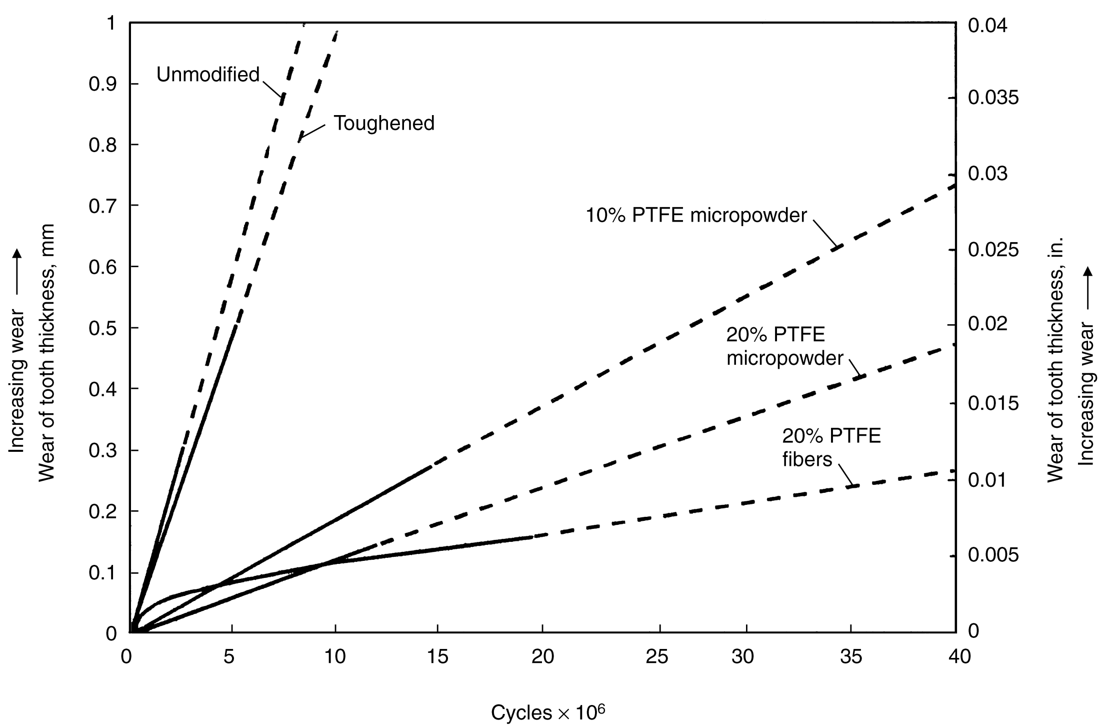


Fig. 5 Effect of internal lubricants on the sliding wear (acetal against steel) of 27 tooth, 2.12 module (12 pitch), involute thermoplastic spur gears. Initial contact stress = 42 MPa 9 (6 ksi). Pitchline velocity = 5.6 m/s (18.4 ft/s). Solid lines show median of measured wear range. Dotted lines are extrapolated data. Source: Dupont Engineering Polymers

dation. A magnet is placed in the hopper throat to remove any iron that has accidentally entered the feedstock.

In the heated extruder barrel, which contains the reciprocating screw, the thermoplastic is gradually melted by a combination of shear heating (caused by the mechanical working of the material as it is conveyed down the barrel) and heat conduction from the barrel. The thermoplastic is gradually conveyed by the rotating screw from the rear of the barrel to the front. The barrel is usually heated by band-type electrical resistance heaters fitted around its periphery. Because the amount of heat required varies with position along the barrel, the band heaters are controlled in several zones along the barrel. Some barrels are equipped about halfway down with a vent to remove gases from the melting feedstock. A vacuum is usually applied to the vent, and a specially designed screw is required to depressurize the melt in the region of the vent to keep it from extruding through the vent.

The reciprocating screw usually has three successive zones, each with specific functions: The feed zone of relatively deep screw flights conveys the unmelted plastic from the hopper throat into the heated barrel, where it begins to melt; the compression zone of decreasing flight depth, and thus volume, provides gas removal and melt densification, along with a material mixing action for better material and temperature uniformity; and the metering zone, generally of constant, shallow-flight depth, provides the final shear heating and mixing of the melt. After the melt passes through the metering section and a check valve at the end of the screw, it joins the melt pool in front of the screw. As the volume of melt in front of the screw increases, it forces the screw to the rear of the barrel against an adjustable "back pressure." This back pressure is applied hydraulically to the back end of the screw. Increasing the pressure increases the amount of mechanical working of the feedstock. Screw rotation and feedstock melting continue until a sufficient amount of melt is available in front of the screw to fill the mold. At this stage, the screw rotation is stopped, and the machine is ready for injection. The stages in the operation of a reciprocating screw are shown in Fig. 7.

The melted thermoplastic is injected into the mold through the nozzle, under high pressure (typically 70 to 205 MPa, or 10 to 30 ksi, depending on the mold-filling resistance). Injection occurs as the screw is hydraulically forced forward in the barrel. The hydraulic cylinder is located at the rear of the screw and barrel. A check valve at the tip of the screw keeps the melt from flowing back along the screw as the screw is pushed forward. In contemporary machines, the injection rate (determined by the

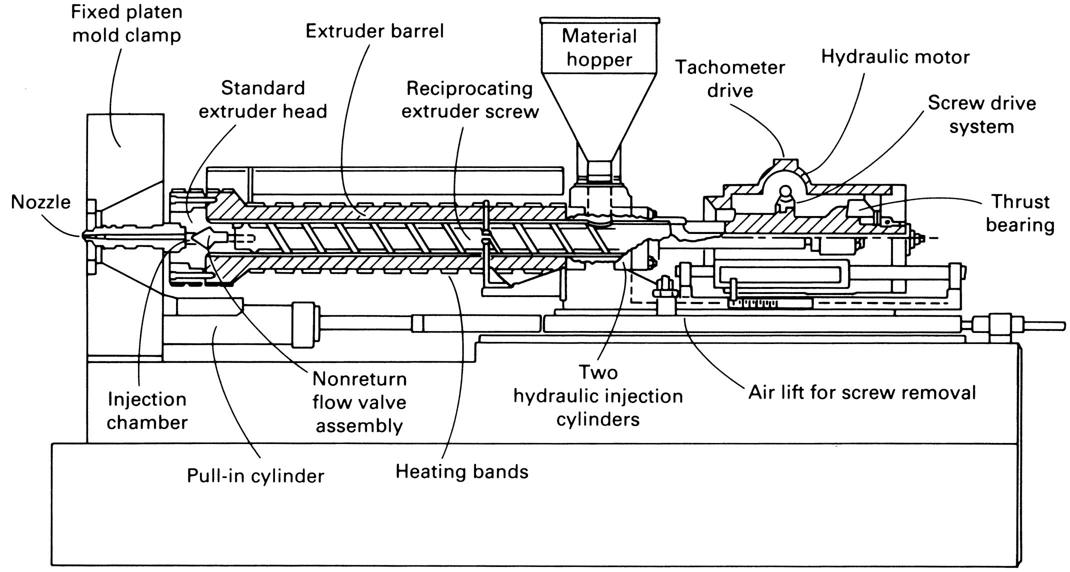


Fig. 6 The injection end of a reciprocating-screw injection molding machine

forward velocity of the screw) and injection pressure are closely controlled throughout the mold-filling stage.

The mold clamp, which holds the halves of the mold closed against the injection pressure of the melt, opens the mold to allow part removal after the thermoplastic has solidified. During the cooling and solidification period, the screw begins to rotate and melt new material for the next shot.

The clamping system consists of a fixed platen and a movable platen, each of which has half of the mold attached. The fixed platen (shown in Fig. 6) has a hole in its center to allow the injection cylinder nozzle to be placed in contact with the sprue bushing of the mold. A movable platen is moved along tie bars by either hydraulic or mechanical means, or a combination of both. The amount of clamping force the machine can apply, rated in tons, generally determines the size of the part that the machine can process. It is this clamping force that overcomes the injection pressure of the melt in the mold and keeps the mold halves together. The clamping force, or "tonnage," required is determined by the amount of injection pressure required to fill the mold and the projected area of the part (that is, the area perpendicular to the axis of the machine). If the machine has inadequate clamping force, the two halves of the tool will begin to separate during injection, causing the melt to squirt out, or "flash," at the mold parting line, potentially causing an incomplete mold-fill, or "short shot."

Contemporary molding machines have two or more closing speeds: a high-speed closing, requiring only a low force, followed by a slow-speed closing stage, which generates the high clamping force to close the mold firmly (stretching the tie bars) prior to injection. The clamp usually opens slowly at first, followed by a rapid traverse. These high-speed motions reduce the time of the overall process cycle.

Injection Mold Design. Injection molds, in their simplest form, consist of two halves, often called the core and cavity (Fig. 8). A hole or sprue bushing conducts the melt from the injection nozzle through the sprue and through a gate into the mold cavity.

Gates and Runners. Commonly, more than one gate is used to deliver material into the mold, and each gate is fed by a "runner" channel leading from the sprue to the gate. After the mold is

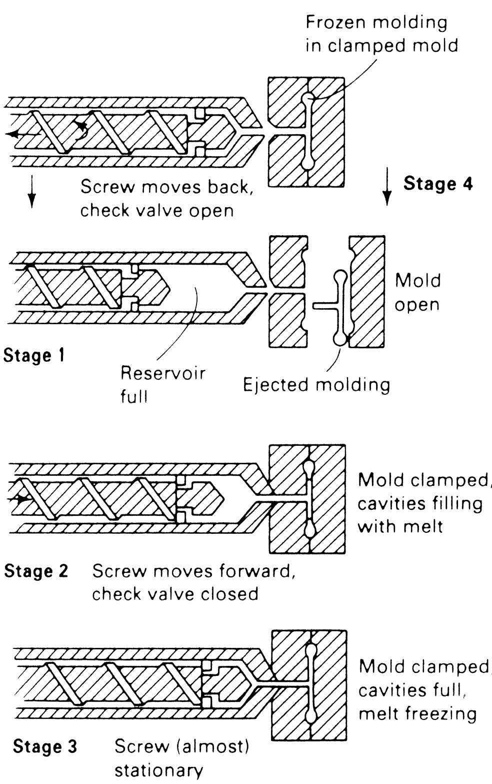


Fig. 7 Stages in the operation of a reciprocating-screw injection molding machine

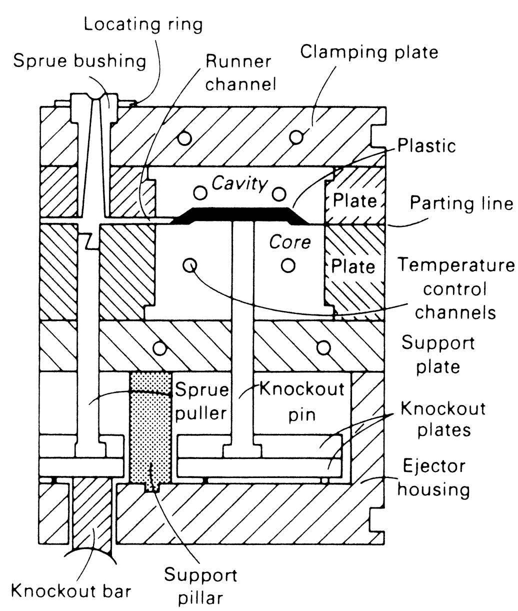


Fig. 8 Two-plate injection mold

filled and the melt has solidified, the material in the sprue and runners also solidifies and must be removed with the part before the next shot. The sprue and runner material are usually reground in a granulator machine and fed back, along with virgin material, into the injection cylinder for reuse. To reduce the amount of material that must be recycled, some molds are equipped with a hot runner manifold (Fig. 9), which keeps the material in the sprue and runners molten to become a part of the next shot.

With larger and more complex parts, it is common to use more than one gate for injecting material into the mold. The configuration of the runners and gates determines the way the mold is filled. This can change the amount of injection pressure required, the location and condition of "knit" lines, the orientation of material flow (which can affect mechanical properties), and the ability to vent air from the mold. Knit lines are surfaces along which the flow fronts meet and are generally weaker. Nonuniform mold filling, caused by poor gate locations, can result in overpacking the mold in the regions which fill first, causing residual stresses and possible warp-age in the part. Computer programs are now available to assist in mold design and the layout of runners and gates for effective mold filling at minimum pressure.

Mold Cooling. With thermoplastic injection molding, a method of cooling is usually incorporated into the mold to speed the solidification of the plastic. This usually consists of holes bored in each half of the mold through which a heat-exchange fluid, usually water, can circulate. The mold temperature is usually controlled above room temperature. The optimal mold temperature depends on the type of plastic being molded, but typical mold temperatures for the more common thermoplastics vary from 40 to 120 °C (100 to 250 °F).

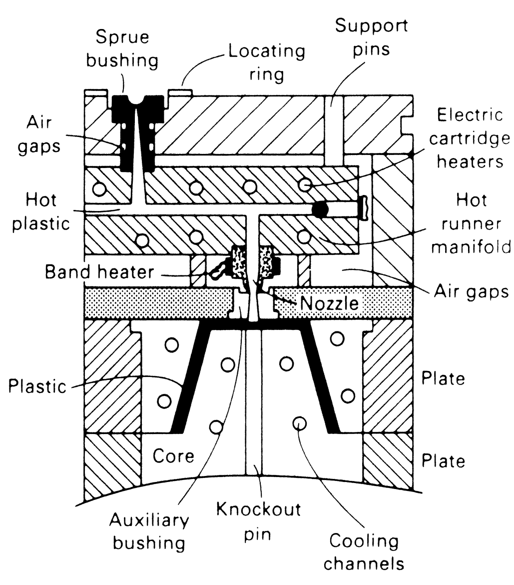


Fig. 9 Hot runner injection mold

Material Shrinkage. All thermoplastic solidification is accompanied by a volumetric shrinkage. For crystalline plastics, the shrinkage is associated with crystallization. For amorphous plastics, the shrinkage is generally less and is associated with the glass transition. In both cases, the amount of shrinkage depends on various processing parameters, including the mold temperature and rate of cooling of the melt. This shrinkage can continue for a period of time after the part is removed from the mold. It is important that the shrinkage be repeatable, so the mold can be appropriately sized. For parts requiring tight dimensional control or optimal mechanical properties, uniform cooling is essential. If the part does not solidify uniformly in the mold, residual stresses will occur as a result of differential shrinkage. Computer programs are available to assist in the optimal layout of cooling channels for uniform part cooling.

Part Removal. The mold is usually designed such that the part remains on the moving half of the mold when it is opened. Ejector pins are then actuated to separate the part from the mold. These ejector, or knockout, pins are activated either as a direct result of the moving platen (mechanical knockouts), or hydraulically.

The geometry of the mold must be such that the part can be readily removed after the material has solidified, sometimes by means of a mechanical part remover. This requires careful part design and selection of the "parting surface" across which the two halves of the mold separate. Neither half of the mold can have undercuts or "die locks" that would trap the part or keep it from being ejected. If the part cannot be designed without undercuts, then moving cores, slides, or lifters must be incorporated to eliminate the trapped condition when the mold is opened.

Mold Venting. Air must be removed as the mold fills with plastic. This is usually accomplished by grinding channels, or vents, into the parting line of the mold. These vents must be narrow enough to keep molten plastic from escaping, while allowing air to escape. They should be located adjacent to the last region of the mold to fill. Sometimes undersized ejector pins are also used to assist in venting the mold. Insufficient venting can result in an incompletely filled mold, or in cosmetic defects called diesel burns, which are caused by the heating of the air as it is compressed.

Mold Materials. For high-volume gear production applications, the mold cavity and core are usually machined from special mold-making steels. Steel is chosen for its wear resistance and durability.

ACKNOWLEDGMENT

A portion of this chapter was adapted from Injection Molding, *Engineered Materials Handbook Desk Edition,* ASM International, 1995, p 299–307

REFERENCES

1. C.E. Adams, *Plastics Gearing: Selection and Application,* Marcel Dekker, 1986
2. *Materials for Plastic Gears,* AGMA 920-A01, American Gear Manufacturers Association, 2001
3. M.I. Kohan, Polyamides (PA), *Engineering Plastics,* Vol 2, *Engineered Materials Handbook,* ASM International, 1988, p 124–127
4. D.P. Townsend, Ed., *Dudley's Gear Handbook,* Second Edition, McGraw Hill, Inc. 1991, p 17.2

Gear Materials, Properties, and Manufacture
J.R. Davis, editor, p89-127
DOI:10.1361/gmpm2005p089

CHAPTER 5

Machining, Grinding, and Finishing

METAL REMOVAL processes for gear manufacture can be grouped into two general categories:

- Rough machining (or gear cutting) operations such as milling, hobbing, and shaping which cut the tooth spaces into prepared gear blanks leaving the desired teeth
- Finishing (or high-precision machining) operations such as shaving, grinding, honing, and lapping which improve the accuracy and surface finish of previously prepared gear teeth

Gear Cutting. The methods of gear cutting can be divided broadly into form cutting and generating processes. The tooth profile in the former is obtained by using a form cutting tool. This may be a multiple-point cutter used in a milling machine or a broaching machine, or a single-point tool for use in a shaper. A variation of the form cutter method is based on transferring the profile from a templet. This method is used in the shaperlike cutting of large tooth forms and in cutting bevel gear teeth on a bevel gear planer. In all these processes, the workpiece is held stationary until a tooth is finished; the piece is then indexed for successive teeth.

In the generating process, the tooth profile is obtained by a tool that simulates one or more teeth of an imaginary generating gear. A relative rolling motion of the tool with the workpiece generates the tooth surface. This method is used in the hobbing, shaping, and milling processes for manufacturing spur and helical gears and in the face milling and face hobbing of bevel gears.

A wide variety of machines are used to cut gear teeth. As shown in Fig. 1, there are four more or less distinct ways to cut material from a gear blank so as to leave a toothed wheel after cutting. First, the cutting tool can be threaded and gashed. If so, it is a hob, and the method of cutting is called hobbing. Second, when the cutting tool is shaped like a pinion or a section of a rack, it will be used in a cutting method called shaping. Third, in the milling process, the cutting tool is a toothed disk with a gear tooth contour ground into the sides of the teeth. The fourth general method uses a tool (or a series of tools) that wraps around the gear and cuts all teeth at the same time. Methods of this type are termed broaching, punching, or shear cutting.

Finishing. As listed earlier, there are several methods available for the improving the quality of gears following the standard rough machining operation. Gear shaving is a chip forming finishing operation that removes small amounts of metal from the working surfaces of gear teeth. Its purpose is to correct errors in index, helix angle, tooth profile, and eccentricity. Shaving is carried out prior to gear heat treat-

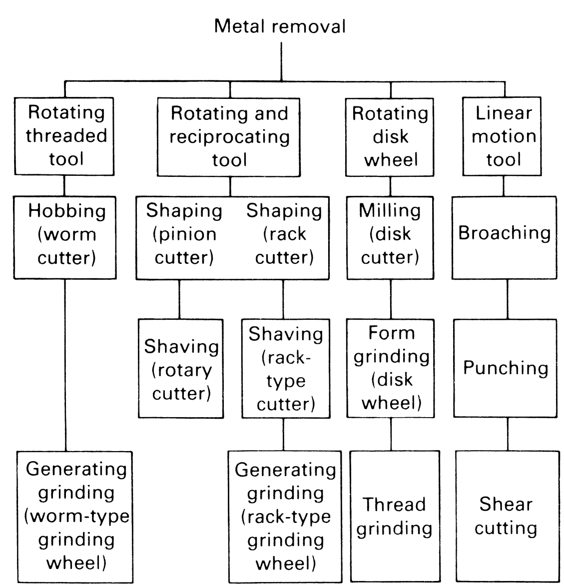


Fig. 1 Outline of methods of producing gear teeth

ment. Grinding is a technique of finish machining utilizing a rotating abrasive wheel. Grinding is an effective means of finishing heat-treated high-hardness steels (40 HRC and above). Gear honing is a particularly effective method of removing nicks and burrs from the active profiles of the teeth after heat treatment. Lapping is a low-speed low-pressure abrading operation used to refine the tooth surface and to reduce noise levels in gear sets. The mating gear and pinion are run together under a controlled light load while a mixture of abrasive compound and suitable vehicle are pumped onto the pair. As with grinding and honing, lapping is also carried out after gear heat treatment.

Machining Processes for Gears

Simple gear tooth configurations can be produced by basic processes such as milling, broaching, and form tooling. Complex gear tooth configurations require more sophisticated processes designed especially for the manufacture of gears.

Processes for Gears Other Than Bevel Gears

The methods used to cut the teeth of gears other than bevel gears are milling, broaching, shear cutting, hobbing, shaping, and rack cutting. In any method, a fixture must hold the gear blank in correct relation to the cutter, and the setup must be rigid.

Milling produces gear teeth by means of a form cutter. The usual practice is to mill one tooth space at a time. After each space is milled, the gear blank is indexed to the next cutting position.

Peripheral milling can be used for the roughing of teeth in spur and helical gears. Figure 2 shows teeth in a spur gear being cut by peripheral milling with a form cutter. End milling can also be used for cutting teeth in spur or helical gears and is often used for cutting coarse-pitch teeth in herringbone gears.

In practice, gear milling is usually confined to one-of-a-kind replacement gears, small-lot production, the roughing and finishing of coarse-pitch gears, and the finish milling of gears having special tooth forms. Although high-quality gears can be produced by milling, the accuracy of tooth spacing on older gear milling machines was limited by the inherent accuracy of the indexing mechanism. Most indexing techniques used on modern gear milling machines incorporate numerical control or computer numerical control, and the accuracy can rival that of hobbing machines.

Broaching. Both external and internal gear teeth, spur or helical, can be broached, but conventional broaching is usually confined to cutting teeth in internal gears. Figure 3 shows progressive broach steps in cutting an internal spur gear. The form of the space between broached gear teeth corresponds to the form of the broach teeth. The cutting action of any single broach tooth is similar to that of a single form tool. Each cross section of the broach has as many

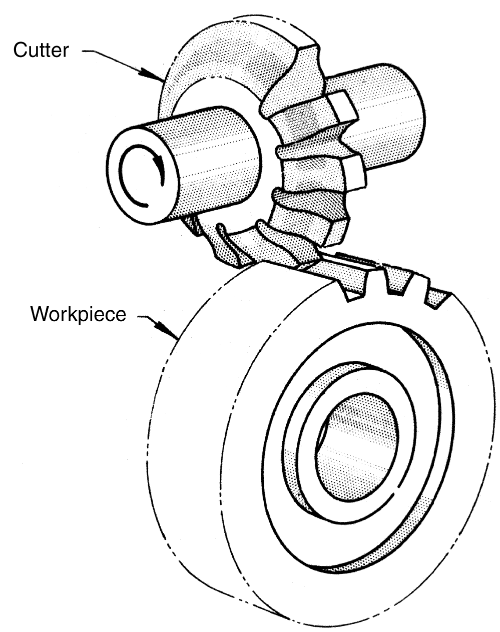


Fig. 2 Relation of cutter and workpiece when milling teeth in a spur gear

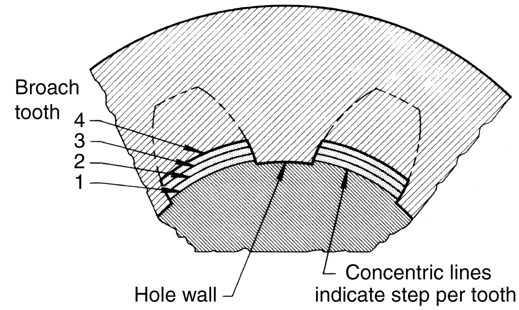


Fig. 3 Progressive action of broach teeth in cutting teeth of an internal spur gear

teeth as there are tooth spaces on the gear. The diameter of each section increases progressively to the major diameter that completes the tooth form on the workpiece. Broaching is fast and accurate, but the cost of tooling is high. Therefore, broaching of gear teeth is best suited to large production runs.

Shear cutting is a high-production method for producing teeth in external spur gears. The process is not applicable to helical gears. In shear cutting, as in broaching, all tooth spaces are cut simultaneously and progressively (Fig. 4). Cutting speeds in shear cutting are similar to those for broaching the same work metal. Machines are available for cutting gears up to 508 mm (20 in.) in diameter, with face width up to 152 mm (6 in.).

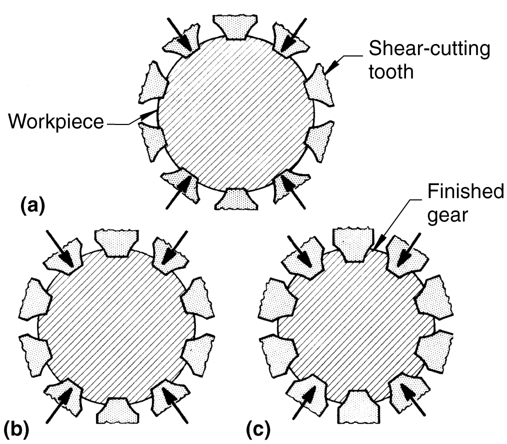


Fig. 4 Progressive action in shear cutting teeth of an external spur gear. Shear cutting operation proceeds from roughing (a) to intermediate (b) to finishing (c) operations.

The shear cutting head is mounted in a fixed position, and the gear blanks are pushed through the head. Cutting tools are fed radially into the head, a predetermined amount for each stroke, until the required depth of tooth space is reached. In shear cutting, some space is required for over-travel, although most workpieces with integral shoulders or flanges (such as cluster gears) do have enough clearance between sections to allow shear cutting to be used. Therefore, this process is best suited to large production runs.

Hobbing is a practical method for cutting teeth in spur gears, helical gears, worms, worm gears, and many special forms. Conventional hobbing machines are not applicable to cutting bevel and internal gears. Tooling costs for hobbing are lower than those for broaching or shear cutting. Therefore, hobbing is used in low-quantity production or even for a few pieces. On the other hand, hobbing is a fast and accurate method (compared to milling, for example) and is therefore suitable for medium and high production quantities.

Hobbing is a generating process in which both the cutting tool and the workpiece revolve in a constant relation as the hob is being fed across the face width of the gear blank. The hob is a fluted worm with form-relieved teeth that cut into the gear blank in succession, each in a slightly different position. Instead of being formed in one profile cut, as in milling, the gear teeth are generated progressively by a series of cuts (Fig. 5). The hobbing of a spur gear is shown in Fig. 6.

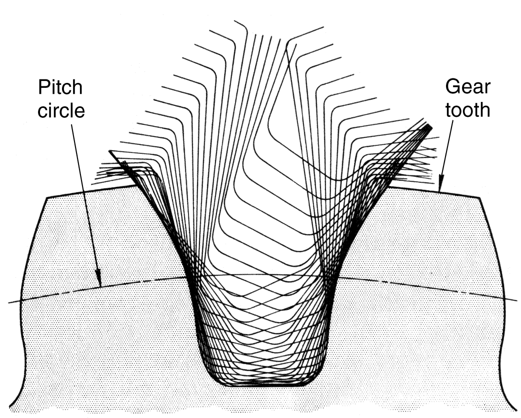


Fig. 5 Schematic of hobbing action. Gear tooth is generated progressively by hob teeth.

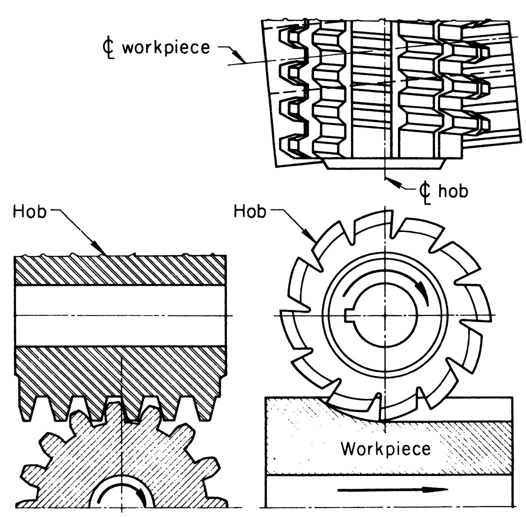


Fig. 6 Hobbing of a spur gear

Gear shaping is the most versatile of all gear cutting processes. Although shaping is most commonly used for cutting teeth in spur and helical gears, this process is also applicable to cutting herringbone teeth, internal gear teeth (or splines), chain sprockets, ratchets, elliptical gears, face gears, worm gears, and racks. Shaping cannot be used to cut teeth in bevel gears. Because tooling costs are relatively low, shaping is practical for any quantity of production. Workpiece design often prevents the use of milling cutters or hobs (notably, for cluster gears), and shaping is the most practical method for cutting the teeth.

Gear shaping is a generating process that uses a toothed disk cutter mounted on a spindle that moves in axial strokes as it rotates. The workpiece is carried on a second spindle. The workpiece spindle is synchronized with the cutter spindle and rotates as the tool cuts while it is being fed gradually into the work. The action between a shaping cutter and a gear blank is illustrated in Fig. 7. Shaping applied to the cutting of a worm (Fig. 8) involves no axial stroke of the cutter spindle.

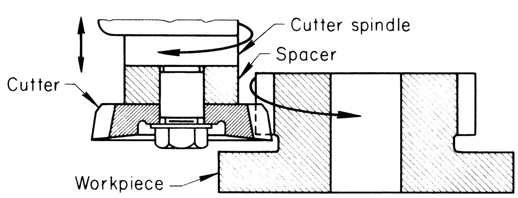


Fig. 7 Relation of cutter and workpiece in shaping gear teeth

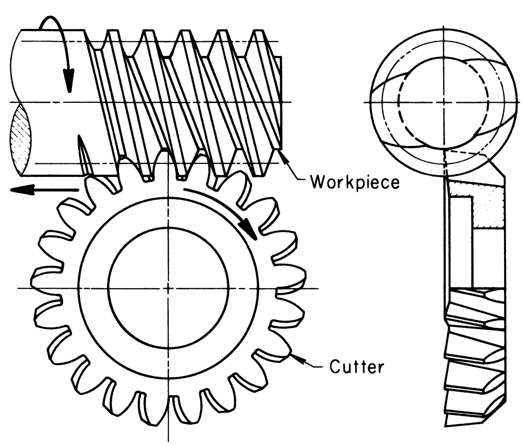


Fig. 8 Relation of cutter and workpiece in generating a worm by shaping

Rack cutting is done with a cutter in the form of a rack with straight teeth (usually three to five). This cutter reciprocates parallel to the gear axis when cutting spur gears and parallel to the helix angle when cutting helical gears. Metal is removed by a shaper-like stroke similar to the cutting action in gear shaping. In addition to the reciprocating action of the rack cutter, there is synchronized rotation of the gear blank with each stroke of the cutter, with a corresponding advance of the cutter as in the meshing of a gear and rack. By these combined actions, the true involute curve of the gear tooth is developed. Several gear cutting machines use this principle. Rack cutters are less expensive than hobs. Rack cutting is especially adapted to the cutting of large gears or gears of coarse pitch or both. Gears with diametral pitch as coarse as ⅜ are commonly cut by the rack method.

Processes for Bevel Gears

The machining of bevel gears is treated as a separate subject because most bevel gears are cut in special machines with special cutters. However, the action of these cutters bears a close resemblance to one or more of the basic processes—milling, broaching, or shaping. Both generating and nongenerating processes are discussed below.

Milling is not widely used for cutting bevel gears, because of the accuracy limits of indexing devices and because the operation is time consuming (as many as five cuts around the gear may be required for completing one gear). Straight bevel gears are sometimes roughed by milling and then finished by another method. This two-operation procedure is more common when the availability of special gear cutting machines is limited.

Template machining is a low-production, nongenerating method used to cut the tooth profiles of large bevel gears using a bevel gear planer (Fig. 9). Because the setup can be made with a minimum of tooling, template machining is useful when a wide variety of coarse-pitch gears are required. Template machining uses a simple, single-point cutting tool guided by a template several times as large as the gear tooth to be cut. Under these conditions, high accuracy in tooth forms is possible.

The necessary equipment is unique. The setup utilizes two templates; one for each side of the gear tooth. In theory, a pair of templates would be required for each gear ratio, but in practice a pair is designed for a small range of ratios. A set of 25 pairs of templates encompasses all 90° shaft angle ratios from 1:1 to 8:1 for either 14½ or 20° pressure angles. This system of templates is based on the use of equal-addendum tooth proportions for all ratios. The tooth bearing localization is produced by a slight motion of the tool arm as the cutting tool moves along the tooth. The length and position of the cut can be controlled by the machine operator.

To produce a finished gear, five or six cutting operations are required. The first is a roughing operation, made by feeding a slotting tool or a corrugated V-tool to full depth. If the first cut is made with a slotting tool, a second cut is required with a V-tool or a corrugated V-tool. Cuts are made with the template follower resting on a straight guide. After roughing, the templates are set up, and the teeth are finished by making two cuts on each side. Slotting tools are specified by point width and depth of cut; corrugated roughing tools, by point width, depth of cut, and pressure angle; finishing tools, by point width only. To set up a straight bevel gear planer for template machining, the operator need know only the tooth proportions of the gear to be cut, plus the template list and the index gear list furnished with the machine.

Formate cutting and Helixform cutting are nongenerating methods for cutting spiral bevel and hypoid gears. Nongenerating methods can be used for cutting the gear member of spiral bevel and hypoid pairs when the gear-to-pinion ratio is 2.5:1 or greater. The two principal methods are Formate cutting and Helixform cutting.

Formate cutting is applicable to both the roughing and single-cycle finishing of gears with pitch diameters up to 2540 mm (100 in.). Roughing and finishing are sometimes both done by one cutter in the same machine. More often, for greater efficiency, roughing and finishing operations are done in different machines.

The Formate method can be used with the two-cut roughing and finishing method, in which the gear is roughed to depth and then single-cycle finished, or with a one-cut roughing and finishing method known as completing. In the completing method, the cutter is plunged to approximately 0.25 mm (0.010 in.) of whole depth, the cutting speed is doubled, and the cutter is fed to whole depth, taking a light chip. This final portion of the cycle removes the built-up edge on the cutter and produces an acceptable surface finish. The completing method is used for both face milling and face hobbing.

In the Formate single-cycle finishing method, one tooth space is finish cut in one revolution of the cutter. Stock removal is accomplished by cutting blades mounted in a circular cutter that resembles a face milling cutter. Each blade in the cutter is slightly longer and wider than the preceding blade; thus, the cutting action is, in effect, that of a circular broach. A gap between the first and last cutting blades permits indexing of the workpiece as each tooth space is completed. The relative positions of cutter and gear during Formate single-cycle cutting are illustrated in Fig. 10.

Helixform cutting, another nongenerating method of cutting spiral bevel and hypoid gears, is generally similar to Formate cutting. However, there is one significant difference in the method and in the finishing operation for the gear member.

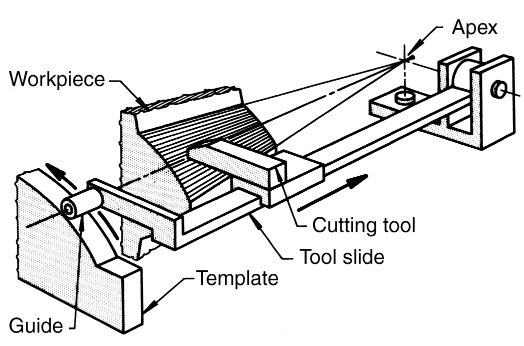


Fig. 9 Cutting teeth in a large straight bevel gear by template machining in a bevel gear planer

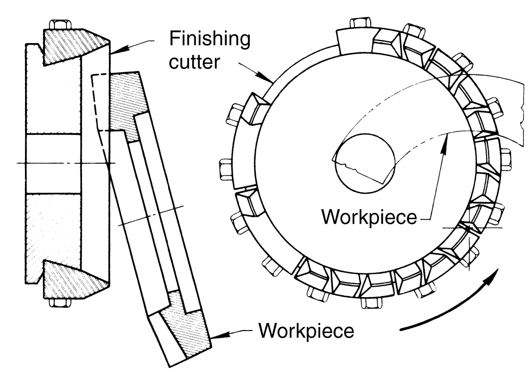


Fig. 10 Relative positions of cutter and workpiece in Formate single-cycle cutting

One turn of the Helixform cutter finishes both sides of a tooth space. The cutter has both rotational and reciprocating motion, and this combination makes the path of the cutter-blade tips tangent to the root plane of the gear. Because the cutting edge is a straight line, the surface cut by a Helixform gear cutter is a true helical surface. The principal advantage of Helixform cutting compared to Formate cutting is that the gear produced by Helixform is conjugate to the mating pinion, and the resulting contact pattern has little or no bias.

The Cyclex method is also a nongenerating method, and the result is the same as that for the Formate method. The Cyclex method was developed for the rough and finish cutting of gears in one operation and is particularly suitable when production quantities are not great enough to warrant separate Formate roughing and finishing machines for the gear member. Cyclex machines of the generator type can cut a wide range of gear sizes and can be used for cutting both gears and pinions.

In the Cyclex method, the gear is roughed and finished in one chucking from the solid blank. The finishing blades of the cutter are set below the roughing blades and do not contact the work during the roughing cycle. Several revolutions of the cutter may be necessary for roughing, the number required depending on the pitch of the gear. In the final revolution, after the last roughing blade has passed through the cut, the work is rapidly advanced, permitting the finishing blades to make contact and finish the tooth space to size. After the finishing blades have passed through the cut, the work is rapidly withdrawn and indexed, and the cycle is repeated until all teeth are completed.

Face mill cutting machines are used to finish cut teeth in spiral bevel, Zerol, and hypoid gears. Machines and cutters are available for cutting gears ranging from small instrument gears up to about 2540 mm (100 in.) in diameter.

The three types of face mill cutters are identified by the design of their cutting blades: integral, segmental, and inserted. All of these can be used for both roughing and finishing. Solid or integral-blade cutters are made from a single piece of tool steel and are usually less than 152 mm (6 in.) in diameter. They are used for fine-pitch gears. Segmental cutters are made up of sections, each having two or more blades. The segments are bolted to the cutter head around the periphery. Inserted-blade cutters are of two types. The first type has blades that are bolted to the slotted head. This cutter usually has parallels for changing diameter, and adjusting wedges for truing individual blades. The blades are sharpened in a radial or near-radial plane. The second type of inserted-blade cutter has blades that are clamped in the slot. The blades are sharpened by topping the blades down and repositioning them in the head.

For finishing, the three types of cutters can be furnished with all outside blades, all inside blades, or alternate outside and inside blades. Roughing cutters and completing cutters can have either alternate inside and outside blades or have end-cutting or bottom-cutting blades alternately spaced with inside and outside blades.

There are four basic cutting methods: completing, single-side, fixed-setting, and single-setting. In each case, the rotating cutting edges of a face mill cutter represent the imaginary gear surface.

The completing method is the generation of the part by a circular face mill or face hob cutter with alternate inside and outside blades that cut the tooth surfaces on both sides of a tooth space at the same time. With this method, each member is finished in one operation.

In the single-side method, the part is finished by a circular face mill cutter with alternate inside and outside blades that cut the tooth surface on each side of a tooth space in separate operations with independent machine settings.

In the fixed-setting method, the part is finished by two circular face mill cutters: one with inside blades only for cutting the convex side of the tooth, and the other with outside blades only for cutting the concave side. The two sides of the tooth are produced separately in two entirely different machine setups. For large production runs, a pair of machines is used. One machine is for cutting one side of the tooth, and the other is for the other side of the tooth.

The single-setting method is a variation of the completing method and is used when the available cutters have point widths too small for completing cutting. Both sides are cut with the same machine settings, and the blank is rotated on its axis to remove the amount of stock necessary to produce the correct tooth thickness. After the first cut, only one side of the cutter is cutting in the tooth slot. Figure 11 illustrates the action of a face mill when generating pinion teeth by the fixed-setting method (inside blades).

Face hob cutting is similar to face milling except that the indexing is superimposed on the generating cycle. Cutters are arranged with blade groupings. As each blade group passes through the cut, the work is being indexed one pitch (Fig. 12). Gears cut by this method are generally completed in one operation. Most face hob cutters are of the inserted-blade type. Integral and segmental systems are available.

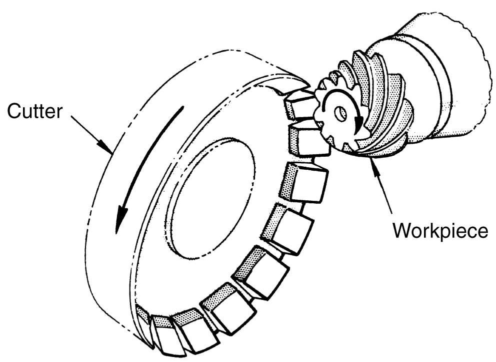


Fig. 11 Face mill cutter shown in position to generate a pinion by the fixed-setting method

Interlocking cutters, known also as completing generators, generate the teeth on straight bevel gears or pinions from a solid blank in one operation.

In this method, two interlocking disk-type cutters rotate on axes inclined to the face of the mounting cradle, and both cut in the same tooth space (Fig. 13). The cutting edges present a concave cutting surface that removes more metal at the ends of the teeth, giving localized tooth contact. The gear blank is held in a work spindle that rotates in timed relation with the cradle on which the cutters are mounted. A feed-cam cycle begins with the workhead and blank moving into position for rough cutting, without generating roll, and cutting proceeds until the cut is just short of full depth. After a rough generating roll, the work is fed in to full depth, and a fast up-roll finish generates the tooth sides. At the top of the roll, the work backs out, and the cradle and work spindle roll down again into

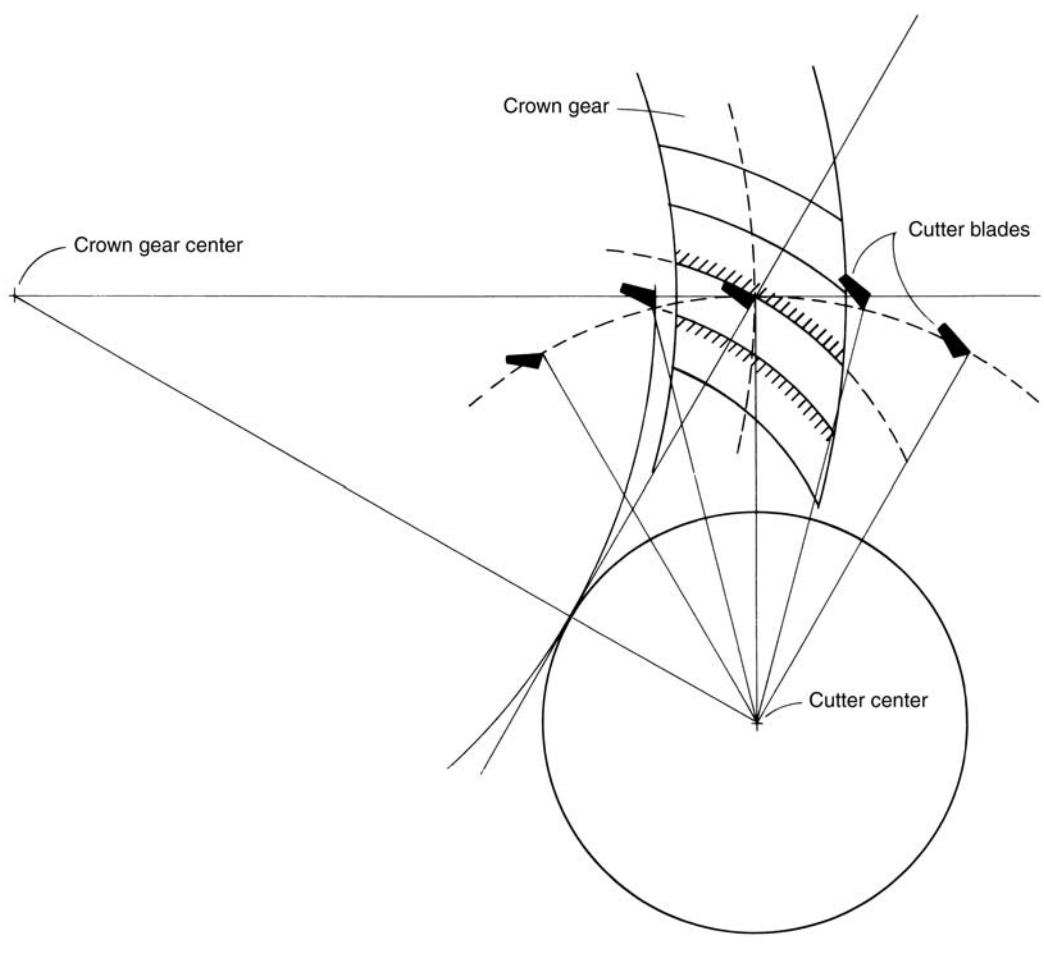


Fig. 12 Schematic of the face hob cutting method

roughing position. During this short down-roll, the blank is indexed.

Revacycle is a generating process used for cutting straight bevel gears up to about 255 mm (10 in.) pitch diameter in large production runs. This is the fastest method for producing straight bevel gears of commercial quality. Initial tooling cost is greater for the Revacycle process than for other processes for cutting straight bevel gears, but the high production rate results in the lowest cost for mass production.

Most gears produced by the Revacycle method are completed in one operation, using cutters 406, 457, 533, or 635 mm (16, 18, 21, or 25 in.) in diameter that rotate in a horizontal plane at a uniform speed (Fig. 14). The cutter blades, which extend radially outward from the cutter head, have concave edges that produce convex profiles on the gear teeth. During cutting, the work-piece is held motionless while the cutter is moved by means of a cam in a straight line across the face of the gear and parallel to its root line. This motion produces a straight tooth bottom while the desired tooth shape is being produced by the combined effect of the motion of the cutter and the shapes of the cutter blades. The cutter is actually a circular broach in that each successive cutting tooth is larger than the one that precedes it along the circumference of the cutter. The cutter makes only one revolution per tooth space.

Feed is obtained by making cutter blades progressively longer, rather than by moving the entire cutter into the work. The completing cutter contains three kinds of blades: roughing, semifinishing, and finishing. One revolution of the cutter completes each tooth space, and the work is indexed in the gap between the last finishing blade and the first roughing blade. For the small amount of Revacycle work that is too deep to be completed in one cut, separate roughing and finishing operations are used. Under these conditions, separate cutters and setups are required for each operation. The cutters and machine cycles are similar to those for completing cutters, except that the roughing cutters have no semifinishing or finishing blades. A second cutter has only semifinishing and finishing blades.

Two-tool generators are used for cutting straight bevel gears by means of two reciprocating tools that cut on opposite sides of a tooth (Fig. 15). Tooling cost is low for two-tool generators, but production rates are lower than those for other straight bevel generators, such as interlocking cutters and Revacycle machines. Two-tool generators are usually used when:

- The gears are beyond the practical size range (larger than about 254 mm, or 10 in., pitch diameter) of other types of generators
- Gears have integral hubs or flanges that project above the root line, thus preventing the use of other generators
- A small production quantity or a variety of gear sizes cannot be accommodated by other types of machines used for cutting straight bevel gears

Two-tool generators are used for both rough and finish cutting. When warranted by produc-

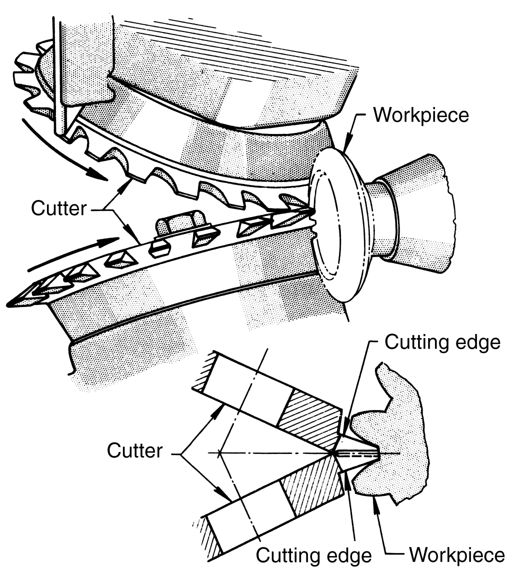


Fig. 13 Relation of interlocking cutters (completing generators) with the bevel gear being cut

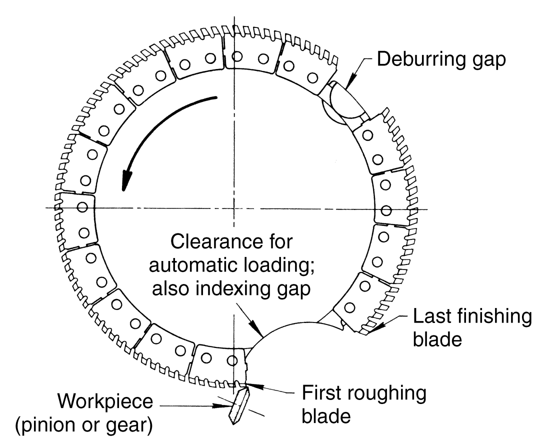


Fig. 14 Revacycle cutter in position to cut a bevel gear

tion quantities, roughing is done in separate machines, which are the same as the generators except that the machines used only for roughing have no generating roll. For small production quantities, both roughing and finishing cuts are made in the generators, the roughing cut being made without generating roll.

To make the machine setup for producing a gear that will operate at right angles to its mating gear, the operator must have the gear specifications and one calculated machine setting called the tooth angle (Fig. 15). The remaining setup data are taken from tables furnished with the machine. When the shaft angle of the two gears is not a right angle, the ratio of roll, as well as the data required for checking the roll, must be calculated.

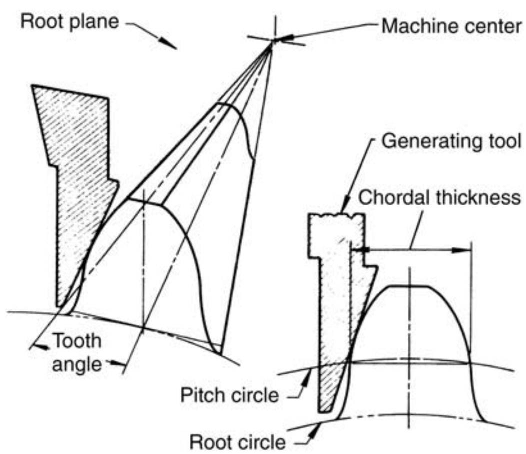


Fig. 15 Angle of straight bevel gear tooth and sections of tools used for two-tool generating

Most two-tool generators can produce straight bevel gears with teeth crowned lengthwise to localize tooth contact. Crowned teeth are produced by means of two angularly adjustable guides on the back of each slide. The guides ride on a pair of fixed rollers (Fig. 16). When the guides are in line with each other, the tool stroke is a straight line, and when they are out of line, the tool is stroked along a curved path. The amount of curvature is controlled by setting the two guides. A table with each machine lists the guide settings for making the tooth contact approximately one-half the face width. The machine settings can be varied to shorten or lengthen the tooth contact.

Planing generators are unique because they can cut both straight-tooth and curved-tooth bevel gears. However, the use of planing generators is ordinarily restricted to cutting gears about 889 mm (35 in.) in diameter or larger or to diametral pitch coarser than 1½. Standard machines can generate straight, Zerol, and spiral bevel gears. Special heads can be added to standard machines to permit the cutting of hypoid gears.

Tools have straight cutting edges and are mounted on a reciprocating slide that is carried on the face of the cradle and connected to a rotating crank by a connecting rod (Fig. 17). Tooth profiles are made by rolling the work with the generating gear. The lengthwise shape of the teeth is formed by a combination of three motions:

- Stroke of the tool
- Continuous, uniform rotation of the work

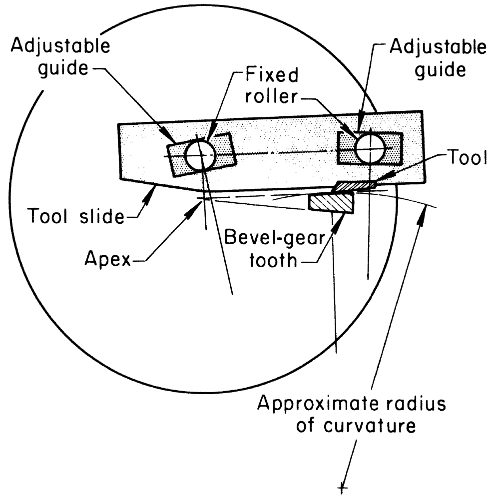


Fig. 16 Provision for crowning gear teeth by means of adjustable guides in two-tool generators

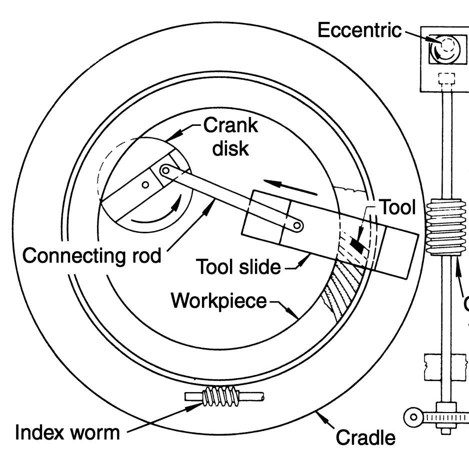


Fig. 17 Components of a planing generator

- An angular oscillation of the work produced by the eccentric shown in Fig. 17

The eccentric motion modifies the effect that the first two motions have on the shape of the teeth. The eccentric is timed for the correct tooth relief.

The spiral angle of the teeth is controlled by the angular offset of the tool slide from the angle of the cradle axis. Continuous rotation of the work is principally for indexing. In effect, the tool makes a cut on all teeth in succession in one generating position, and then the cradle and the work roll together a slight distance before another cut is taken on all teeth in the new generating position. Actually, however, rolling is continuous and occurs gradually until all teeth are completely generated in the last pass around the gear.

Several passes are required to complete a gear, the number depending on tooth depth and shape. Flat gear blanks are usually roughed without a roll, first using a corrugated tool and then using a single cut with a V-roughing tool. This is followed by at least two side-cutting operations on each side of the tooth, including generating cuts with roll. A similar sequence is used for cutting pinions except that roughing is done with roll.

Tools for use in planing generators are simple and inexpensive. Corrugated tools are furnished with a 14½° pressure angle, regardless of the pressure angle of the gear being cut, but point width and depth of cut must be specified when the tools are ordered. The operator must know the specifications for the gear being cut. A table of settings for specific requirements is supplied with the machine.

Selection of Machining Process

Each gear cutting process discussed in the preceding sections has a field of application to which it is best adapted. These fields overlap, however, so that many gears can be produced satisfactorily by two or more processes. In such cases, the availability of equipment often determines which machining process will be used.

The type of gear being machined (spur, helical, bevel, or other) is usually the major factor in the selection of machining process, although one or more of the following factors usually must be considered in the final choice of the method:

- Size of the gear
- Configuration of integral sections (flanges or other)
- Quantity requirements
- Accuracy requirements
- Gear-to-pinion ratio
- Cost

The following sections consider the type of gear as the major variable and discuss the machining methods best suited to specific conditions.

Machining of Spur Gears

Milling, shear cutting, hobbing, and shaping are the methods most commonly used for cutting teeth in spur gears.

Form milling, with the cutter ground to the desired shape of the tooth space (Fig. 2), is a simple means of cutting teeth in spur gears. Tooling cost is low, and the process requires only a conventional milling machine, a form cutter, and an indexing mechanism. Except for low-quality production, milling is seldom used for cutting spur gears. The main disadvantage in the form milling of spur gears is the lack of accuracy in tooth spacing, which depends on the accuracy of the indexing mechanism. In addition, form milling is much slower than shear cutting or hobbing.

One milling cutter is not universal for all numbers of teeth, as are hobs and shaper cutters. To produce theoretically correct gear teeth, the tooth form of the cutter must be designed for the exact number of teeth. However, if a small departure in tooth form is acceptable, cutters have been standardized for a range of teeth, the form being correct for the lowest number of teeth in that particular range. Thus, all teeth within the range are provided with sufficient tip relief. The same form is produced in all tooth spaces within that range. For reasonably accurate gear cutting, eight standard involute gear cutters are required to cut all sizes of gears of a given pitch:

Cutter No.	Gear tooth range
1	135 to rack
2	55–134
3	35–54
4	26–34
5	21–25
6	17–20
7	14–16
8	12–13

If a more accurate tooth form is desired for gears near the higher part of the ranges, seven half-number cutters can be obtained. A No. 1½ cutter, for example, will cut gears with 80 to 134 teeth; a No. 2½ cutter, 42 to 54 teeth; and a No. 3½ cutter, 30 to 34 teeth. For still greater accuracy, cutters of the proper shape for an exact number of teeth can be furnished by most tool manufacturers on short notice.

The tooth form of a single cutter is centered with the gear axis so that a symmetrical tooth space is produced. By the use of gang cutters, portions of adjacent tooth spaces can be rough machined simultaneously. Normally, a roughing and a finishing cutter are ganged, the finishing cutter being centered with the gear axis. Gang cutters and multiple-tooth cutters are specially designed for the specific application.

Shear cutting is faster and more accurate than milling for cutting teeth of almost any involute modification in spur gears. Total cutting time is often less than 1 min for gears up to about 152 mm (6 in.) in diameter. However, tooling cost is high, and shear cutting is therefore practical only for large-scale production.

Hobbing is the process most widely used for cutting teeth in spur gears, usually for one or more of the following reasons:

- High accuracy in a wide range of gear sizes
- Flexibility in quantity production
- Low cost
- Adaptability to work metals having higher-than-normal hardness

The shape of the workpiece sometimes limits the use of hobbing—for example, if the teeth to be cut are close to another portion of the workpiece having a diameter larger than the root diameter of the gear. The axial distance between the two sections must be large enough to allow for hob overtravel at the end of the cut. This over-travel is about one-half the hob diameter. The clearance required between the gear being cut and any flange or other projecting portion of the workpiece is, therefore, about one-half the hob diameter plus additional clearance to allow for the hob thread angle.

In many cases, it is necessary to cut teeth on heat-treated gear blanks to avoid the difficulties caused by distortion in heat treating. Hobbing is especially suitable for cutting gear teeth in hardened steel (sometimes as hard as 48 HRC). Although hob wear increases rapidly as workpiece hardness increases, normal practice should produce an acceptable number of parts per hob sharpening. Success in hobbing gears at high hardness depends greatly on maintaining minimum backlash in the machine and on rigid mounting of both the hob and the workpiece.

The ability to cut teeth in two or more identical spur gears in one setup can also justify use of the hobbing method. Inexpensive fixturing is often utilized for cutting two or more gears at one time when the ratio of face width to pitch diameter is small.

Shaping can produce high accuracy in cutting spur gears because shaping is a generating process. Although seldom as fast as hobbing, shaping is used for a wide range of production quantities. Many types of gears can be produced to requirements by either shaping or hobbing, and the availability of equipment determines which of the two processes is used. However, if the workpiece configuration cannot be hobbed, shaping is often the only practical method.

Cutting of teeth in cluster gears that must meet close tolerances is sometimes a problem because the method used must frequently be restricted to shaping and shaving. The same quality requirements cannot be met by shaping and shaving as by hobbing and grinding. When tolerances for cluster gears are closer than can be met by shaping and shaving, a corrective procedure sometimes must be employed. A cluster gear can be hobbed to greater precision by separating it into a two-piece assembly that is rigidly attached using threaded fasteners.

Machining of Helical Gears

Milling, hobbing, shaping, and rack cutting are methods most used for producing teeth in helical gears. Rack cutting is most often used for large gears. Identical machining methods are applicable to conventional helical and crossed-axes helical gears.

Milling is used less than any other method because of the difficulty in obtaining accuracy and productivity. However, for some low production requirements, milling is the most satisfactory method because the tooling cost is low. In low-volume production, milling is sometimes used for roughing only, and the gear is finished by hobbing or shaping.

The milling of helical gears usually requires cutters specially designed for the specific gear. In milling helical teeth, the cutter travels along the helix angle of the gear. At this setting, the cutter axis of rotation is in the normal plane through the center of the gear tooth space. Under these conditions, only one point on the finished profile is produced in this normal plane. All the others are produced in different planes. Therefore, the form of the cutter teeth is not reproduced in the gear. In addition to the setting angle, the diameter of the cutter affects the gear tooth form and must be considered in designing the cutter.

Hobbing is extensively used for generating the teeth of helical gears for any production volume. With the exception mentioned below, procedures for hobbing helical gears are the same as those for spur gears: When hobbing spur gears with a single-thread hob, the blank rotates one tooth space for each rotation of the hob, the rotation being synchronized by means of change gears. When hobbing helical gears, the rotation of the work is retarded or advanced, through the action of the machine differential, in relation to the rotation of the hob, and the feed is also held in definite relation to the work and the hob. The decision to advance or retard the workpiece rotation depends mainly on whether the hob is a right-hand or left-hand type or whether the helix angle is of right-hand or left-hand configuration. The amount by which the workpiece is retarded or advanced depends on the helix angle. In medium-to-high production, it is common to use fixtures that allow hobbing of two or more identical gears in one loading of the machine.

Gears with integral shanks can usually be hobbed without difficulty. The shanks can assist in fixturing and handling for loading and unloading. When warranted by high-volume production, hobbing can be done in automatic machines utilizing automatic unloading and loading.

Although hob life decreases as workpiece hardness increases, helical gears of hardness as high as 48 HRC are sometimes hobbed. When hardnesses of 48 HRC or lower can be tolerated, the sequence of rough hobbing, heat treating, and finish hobbing is likely to cost less than grinding after heat treatment.

Shaping is a practical process for generating teeth of helical gears having helix angles up to 45°. The only difference between shaping helical gears and spur gears is that the machines used for cutting helical gears must impart additional rotary motion to the cutter spindle as it reciprocates. The amount of rotation per stroke is controlled by a helix guide. The lead of the guide must be the same as the lead of the cutter.

Workpiece configuration is often the main factor in the selection of shaping as a process for cutting helical gears. For example, rotary cutters such as hobs cannot be used when the teeth being cut are too close to the flange.

Machining of Herringbone Gears

Milling, hobbing, and shaping are the methods most often used for cutting herringbone gears. Selection of method depends largely on whether the gear is designed with a gap between the two helixes or whether the herringbone is continuous.

Rotary cutters such as form milling cutters and hobs can be used to cut herringbone teeth only when there is a gap wide enough to permit cutter runout between the right-hand and left-hand helixes. Hobbing machines have been built that can cut herringbone teeth in gears up to 5590 mm (220 in.) in diameter.

End milling can also be used for machining teeth in herringbone gears, regardless of whether the gears have center slots. The end mills for cutting herringbone gears are used in special machines. Many large-diameter herringbone gears are cut by end milling.

Shaping is also a suitable method for cutting teeth on herringbone gears; those designed with a center slot as well as the continuous herringbone can be shaped. The type of shaper used for cutting herringbone gears is similar in principle to the type used for helical gears, except that for herringbone gears two cutters, one for each helix, are operated simultaneously. Both cutters reciprocate, one cutting in one direction to the center of the gear blank and the other cutting to the same point from the opposite direction when the motion is reversed. The cutters not only reciprocate, but also rotate. Both the gear blank and the cutters turn slowly, thus generating the teeth the same way as in a conventional shaper.

Machining of Internal Gears

Broaching, shear cutting, and shaping are the methods most frequently used for cutting internal parts. Milling is seldom used, except for some very large gears.

The broaching of internal spur gear teeth is restricted to workpieces having configurations that permit the broach to pass completely through the piece. The action of a broach in cutting internal gear teeth is shown in Fig. 3. The broaching of gear teeth, which is similar to other types of broaching, is discussed in the section "Broaching" in this chapter. Broaching is an extremely fast and accurate means of machining internal gear teeth, but tooling cost is high; therefore, broaching is practical only for high-volume production.

Shear cutting is applicable to internal gear teeth. The principle involved is essentially the same as that illustrated in Fig. 4 for cutting external spur gears, except that the cutting edges and direction of radial feed are reversed. Unlike broaching, shear cutting is not restricted to parts where the tool must pass completely through the piece. Shear cutting can be used with no more than a 3.2 mm (1/8 in.) relief groove between the end of the cut and a shoulder. Because tooling cost is high for shear cutting, the process is practical only for high-volume production.

Shaping is applicable to cutting internal gear teeth. Tooling cost is lower than that for broaching and shear cutting; therefore, shaping is applicable to low-volume production. Shaping is less restricted to specific configurations than broaching is, because gear teeth can be cut to within 3.2 mm (⅛ in.) of a shoulder; however, for adequate chip clearance and to avoid the danger of striking the shoulder, it is better to have ample clearance. Shaping is often the only practical method for cutting teeth in large internal gears because the cost of large broaches or shear cutting tools is prohibitive.

Machining of Worms

Several types of worms are used for power transmission, among them the double-enveloping type (Fig. 18), also known as the hour-glass worm because of its shape. Milling, hobbing, and shaping are used to machine the various types of worms.

Milling. For double-thread worms of low lead angle and commercial accuracy, a duplex cutter can be used. Each milling cutter is specially designed for cutting a specific worm. Another technique for machining worms utilizes the multiple-thread cutter. The cutter is set with its axis parallel to the work axis and is fed to depth. The work then makes one revolution for completion. The infeed can be made automatic, and because no indexing is required, this method is adaptable to volume production.

Hobbing produces the highest-grade worm at the lowest machining cost, but hobbing can be used only when production quantities are large enough to justify the tooling cost. Because of the large helix angle at which most worm

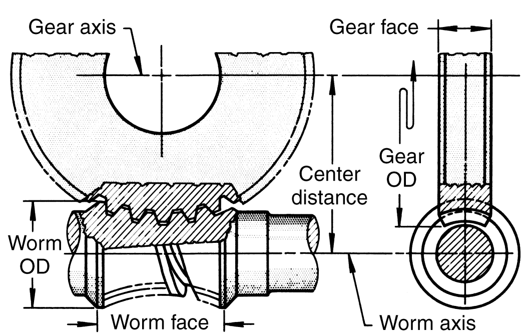


Fig. 18 Mating of worm gear (worm wheel) and worm in a double-enveloping worm gear set

hobs operate, the teeth at the entering end are chamfered to reduce the cutting load. Hobs for cutting worms are made to the same tolerance standards as those for cutting spur gears. When it is necessary to increase the hob diameter to provide more flutes, the tolerances are increased proportionately. The number of flutes in a worm hob is increased to improve surface finish, because the greater the number of flutes, the smaller the feed marks.

Shaping. Worms can also be generated by a shaper-cutter. In this technique, a helical gear cutter is used in a special machine similar to a hobber. Both the work and the cutter rotate, and the cutter is rolled axially along the worm, providing true generating action (Fig. 8).

Machining of Racks

Milling and shaping are used to cut teeth in spur and helical racks.

Milling can be used to produce teeth in both spur and helical racks. The milling cutter must have the exact tooth space form. Racks can be cut in any standard milling machine; requirements are essentially the same as those for a conventional milling operation, and the rack to be cut must be rigidly clamped. Either manual or automatic, indexing mechanisms are available for milling all sizes of racks in high-volume production.

The shaping of spur and helical racks involves the rolling action of the operating pitch circle of the generating shaper-cutter along the corresponding pitch line of the rack. In cutting racks on a gear shaper, the machine is equipped with a special fixture to hold the work. Several arrangements are used for imparting a transverse indexing movement to the member carrying the rack. One method employs a face gear secured on the work spindle that meshes with a pinion. The latter, by means of change gears, drives a lead screw, which operates the slide carrying the rack. Another method is to attach a pinion to the work spindle, which meshes with a master rack attached directly to the slide carrying the rack being cut. The first method is necessary when high ratios are involved in the drive; the second method needs only the regular work change gears.

Machining of Bevel Gears

Face mill cutting, face hob cutting, Formate cutting, Helixform cutting, the Cyclex method, interlocking cutters, Revacycle, two-tool generators, planing generators, and template machining are used to cut teeth in straight and spiral bevel gears. The fundamentals of these processes are discussed in the section "Processes for Bevel Gears" in this chapter. Choice of method depends mainly on the type of gear being cut (straight or spiral bevel), size, configuration, accuracy requirements, and production quantities.

Template machining and planing in a gear generator are more often used for cutting teeth in gears larger than 813 mm (32 in.) outside diameter.

Straight Bevel Gears. The two-tool generator is widely used for cutting straight bevel gears and is especially well adapted to the cutting of gears in a wide range of sizes (up to about 889 mm, or 35 in., outside diameter) in low-to-medium production quantities because tool cost is low. Two-tool generating is also adaptable to the cutting of gears that have protruding portions (such as front hubs) that preclude the use of some processes.

Interlocking cutters provide a means of completing straight bevel gears in one operation. The interlocking-cutter method is faster than two-tool generating, but more costly. Gears without front hubs and less than 406 mm (16 in.) in outside diameter are best adapted to machining with interlocking cutters.

The Revacycle process is the fastest method for cutting straight bevel gears. Gear teeth are often completed at the rate of 1.8 s per tooth. This method was primarily designed for cutting gears having up to 254 mm (10 in.) pitch diameter at 4:1 ratio with the pinion and having a maximum face width of 29 mm (1⅛ in.). In the Revacycle method, the cutter must have an uninterrupted path; therefore, the process cannot cut gears that have front hubs. Because of the high tooling cost, Revacycle cutting is economical only for high-volume production.

Cost Versus Quantity (Straight Bevel Gears). The two-tool generator, because it has the lowest tooling cost, is the most economical method for producing up to approximately 150 pairs of gears, at which point the two-tool generator and interlocking-cutter methods are equivalent. For large production runs, the two-tool generating method becomes prohibitively expensive. The Revacycle and interlocking-cutter methods are equivalent in cost at about 1200 pairs of gears, beyond which the Revacycle method is cheaper.

Spiral Bevel Gears. Spiral, Zerol, and hypoid bevel gears are cut in the same type of equipment and by the same general procedures. (Hypoid gears are by far the most numerous, being used in quantities exceeding those of spiral and Zerol gears combined.) Pinions are generally cut by some type of generator. Gears may or may not be cut in generators. When the gear-to-pinion ratio is greater than about 2.5:1, it is common practice to cut the gears without generating roll. Therefore, the Formate completing, Formate single-cycle, Helixform, and Cyclex methods are extensively used for cutting spiral bevel gears having a ratio of 2.5:1 or greater.

Nongenerated gears are less expensive than their generated counterparts, although there is a smaller difference in cost between the various methods for cutting spiral bevel gears (of less than 813 mm, or 32 in., in outside diameter) than between the various methods for cutting straight bevel gears.

Machining of Large Gears

There is no one dimension that defines a large or a small gear. As various sizes are reached, some methods of manufacture become impractical, and other methods must be used.

Herringbone Gears. The same conditions previously discussed in the section "Machining of Herringbone Gears" in this chapter also apply to the machining of large herringbone gear components.

Spur and Helical Gears. Milling, hobbing, and rack cutting are the methods most commonly used for cutting large spur and helical gears.

Milling is the least expensive and the least accurate of these three methods; therefore, the accuracy required in the gear will determine whether or not milling can be used.

Hobbing is more costly than milling, but produces more accurate gears. Hobs are available for cutting gears well over 2540 mm (100 in.) in outside diameter, provided the diametral pitch is finer than 1. Because of the size of the hob required and the limitations of hobbing machines, it is difficult to hob gears of 1 diametral pitch and coarser.

Once it has been decided that hobbing will be used, it must be determined whether a ground hob will be required or whether an unground hob will provide the required degree of accuracy. The two types vary greatly in cost. However, if a ground hob is selected, extreme care must be used in resharpening; if this is not done, the original accuracy will not be maintained, and errors may occur in the gear tooth form.

Rack Cutting. For machining large gears that have large teeth (coarser than 1 diametral pitch), rack cutting (Fig. 19) is usually the most practical method. Rack cutting may be less expensive than hobbing, even when teeth are finer than 1 diametral pitch.

Bevel Gears. Face milling and face hobbing, two-tool generating, and planing generating are the methods most commonly used for cutting large straight and spiral bevel gears.

Large face mill generators that can cut gears up to 2540 mm (100 in.) in outside diameter are the fastest and most accurate machines for cutting large bevel gears.

Two-tool generators offer a practical means for cutting straight bevel gears having diameters up to about 889 mm (35 in.), face widths up to 152 mm (6 in.), and teeth as coarse as 1 diametral pitch. Tooling cost for the two-tool generating method is also low.

Planing generators can be used for cutting straight bevel gears, but they are most widely used for cutting spiral bevel gears ranging from 889 to 1830 mm (35 to 72 in.) in outside diameter, with up to 254 mm (10 in.) face width and teeth as coarse as ¾ diametral pitch.

Shaving of Spur and Helical Gears

Gear shaving is a finishing operation that removes small amounts of metal from the flanks of gear teeth. It is not intended to salvage gears that have been carelessly cut, although it can correct small errors in tooth spacing, helix angle, tooth profile, and concentricity. Shaving improves the finish on tooth surfaces and can eliminate tooth-end load concentration, reduce gear noise, and increase load-carrying capacity. Shaving has been successfully used in finishing gears of diametral pitches from 180 to 2. Standard machines and cutters are available for shaving gears that range in size from 6.4 to 5590 mm (¼ to 220 in.) pitch diameter.

Leaving excessive stock for shaving will impair the final quality of the shaved gear. For maximum accuracy in the shaved gear and maximum cutter life, a minimum of stock should be allowed for removal by shaving; the amount depends largely on pitch. As little as 0.008 to 0.025 mm (0.0003 to 0.001 in.) of stock should be left on gears having diametral pitch as fine as 48; 0.08 to 0.13 mm (0.003 to 0.005 in.) is allowable for gears having diametral pitch of 2.

Operating Principles. The shaving operation is done with cutter and gear at crossed axes; helical cutters are used for spur gears, and vice versa. The action between gear and cutter is a combination of rolling and sliding. Vertical serrations in the cutter teeth take fine cuts from the profiles of the gear teeth.

During operation, the tip of the shaving cutter must not contact the root fillet, or uncontrolled, inaccurate involute profiles will result. For gears to be shaved, protuberance-type hobs that provide a small undercut at the flank of the tooth may be preferred. This type of hob avoids the initial tip loading of the shaving cutter.

Shaving Cutters. A typical rotary gear-shaving cutter is shown in Fig. 20(b). This cutter is serrated on the profile to form the cutting edges. The depth of the serrations governs total cutter life in terms of the number of sharpenings permitted. A shaving cutter is sharpened by regrinding the tooth profiles, thus reducing the tooth thickness. This causes a reduction in operating center distance for the same backlash and in turn changes the operating pressure angle. These changes are compensated for by a change in addendum after resharpening. Tolerance is an important consideration in original purchase and

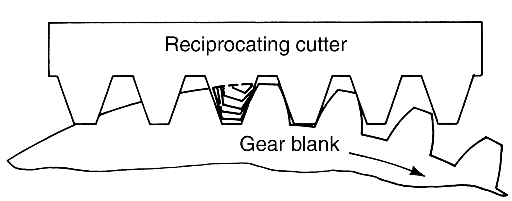


Fig. 19 Rack-type cutter generating the teeth of a spur gear

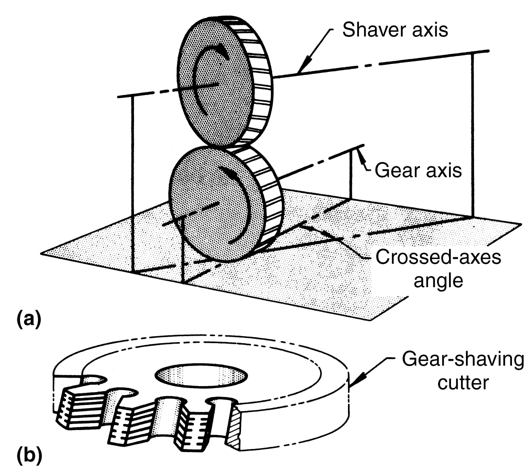


Fig. 20 Shaving of gears. (a) Work gear in mesh with shaving cutter. (b) Serrated gear-shaving cutter

resharpening. Shaving cutters are manufactured to standardized tolerances, not unlike those of master gears. For example, cumulative tooth spacing error can be held to 0.008 mm (0.0003 in.) and profile to 0.00064 mm (0.000025 in.). Because the engineering and facilities necessary to produce such accuracy are not available in most gear manufacturing plants, cutters are ordinarily returned to a tool manufacturer for resharpening.

Shaving Methods. Shaving is done by two basic methods: rack and rotary.

In rack shaving, the rack is reciprocated under the gear, and infeed takes place at the end of each stroke. Because racks longer than 508 mm (20 in.) are impractical, 152 mm (6 in.) is the maximum diameter of gear that can be shaved by the rack method.

Rotary Shaving. The several applications of rotary shaving include underpass, modified underpass, transverse, axial traverse, and angular traverse. Crown shaving can be incorporated in all of these modifications.

In rotary shaving, the cutter has the approximate form of a gear (Fig. 20). The size of gear that can be shaved is limited by the machine rather than by the cutter. Rotary shaving can be any of three types: underpass, modified underpass, and transverse.

Underpass shaving is used on cluster gears or gears with shoulders. To avoid interference with the adjacent gear or shoulder, the cross-axes angle is usually 4 to 6°. The face of the tool must be wider than the face of the shaved gear. Because underpass shaving is a one-cycle, short-stroke process, it is the fastest method of shaving. Disadvantages include relatively short tool life and light stock removal, thus requiring precise size control of the preshaved gear.

Modified underpass shaving is the most widely used method because it is a rapid one-cycle process. Tool cost is moderate because the cutter need be no wider than the gear and may be narrower. The high cross-axes angle of 30 to 60° promotes rapid stock removal and smoother surface finish.

Transverse shaving is the slowest shaving method because multiple passes are required. It is a method of handling gears much wider than the cutter; therefore, cutter cost is moderate for gears with wide faces.

Crown shaving is used to relieve load concentration at the ends of gear teeth caused by the misalignment of axes in operation. Crowning is a modification of the tooth profile in both the radial and axial planes. In the axial traverse method of shaping, crowning is done by rocking the worktable as it is reciprocated. In the higher-production angular traverse method, the cutter is modified to provide crowning. The amount of crown varies, but usually 0.0003 to 0.0005 mm/mm (0.0003 to 0.0005 in./in.) of face width is sufficient.

Speed and Feed. Although cutting speeds are always high, the optimum speed of rotation for gear shaving varies considerably with work metal hardness and composition. Speeds and feeds for several steels and hardness ranges are given in Table 1.

Cutter Material and Construction

High-speed tool steel is used almost exclusively as the material for cutting edges of gear cutting tools. The steels most widely used are the general-purpose grades such as M2 or M7. Grade M3 (higher in carbon and vanadium than general-purpose grades) is also used in many gear cutting applications and is often preferred to M2 and M7 for cutting quenched and tempered alloy steels. The more highly alloyed grades of high-speed tool steel such as T15 or M30 are recommended only for conditions where greater red hardness is necessary. Such conditions include hard work metal, inadequate supply of cutting fluid, or high cutting speeds. Cutters are made from these highly alloyed grades only when the general-purpose grades (or M3) have proved inadequate.

Carbide cutters are used to hand finish gears cut by the face mill and face hob methods when small quantities of high-quality parts are needed. For most applications, carbide cutters are not economical. However, one application in which they are a cost-effective tool is the Tangear generator method.

Tangear Generator. Single-point cemented carbide cutting tools are rotated in opposite directions on the peripheries of two cutter heads with horizontal and parallel axes on the Tangear generator as shown in Fig. 21. The workpiece, with its axis vertical, is rotated as it is fed horizontally (typically a short distance of 8 mm, or 5⁄16 in.) between the cutter heads. The motions are synchronized so that the cutters act as though in mesh with the gear and progressively generate the teeth and form cut the root fillets. The teeth are rough cut as the gear moves to the cutters and are finish cut on the back stroke. The

Table 1 Feeds and speeds for the shaving of carbon and low-alloy steel gears with high-speed steel tools

			Gear tooth size		Feed per revolution of gear(a)		Cutter pitch line speed		High-speed steel tool material	
Material	**Hardness, HB**	**Condition**	**Module**	**Diametral pitch**	**mm**	**in.**	**m/min**	**sfm**	**ISO**	**AISI**
Wrought free-machining carbon steels										
Low-carbon resulfurized	100–150	Hot rolled or annealed	25–6	1–4	0.3	0.012	185	610	S4, S2	M2, M7
			5–3	5–10	0.2	0.008				
1116 1119			2–1.5	11–19	0.12	0.005				
1117 1211			1 and finer	20 and finer	0.07	0.003				
1118 1212	150–200	Cold drawn	25–6	1–4	0.3	0.012	205	675	S4, S2	M2, M7
			5–3	5–10	0.2	0.008				
			2–1.5	11–19	0.12	0.005				
			1 and finer	20 and finer	0.07	0.003				
Medium-carbon resulfurized	175–225	Hot rolled, normalized, annealed, or cold drawn	25–6	1–4	0.3	0.012	150	500	S4, S2	M2, M7
			5–3	5–10	0.2	0.008				
1132 1144			2–1.5	11–19	0.12	0.005				
1137 1145			1 and finer	20 and finer	0.07	0.003				
1139 1146	325–375	Quenched and tempered	25–6	1–4	0.3	0.012	84	275	S4, S2	M2, M7
1140 1151			5–3	5–10	0.2	0.008				
1141			2–1.5	11–19	0.12	0.005				
			1 and finer	20 and finer	0.07	0.003				
Low-carbon leaded	100–150	Hot rolled, normalized, annealed, or cold drawn	25–6	1–4	0.3	0.012	215	700	S4, S2	M2, M7
12L13			5–3	5–10	0.2	0.008				
12L14			2–1.5	11–19	0.12	0.005				
12L15			1 and finer	20 and finer	0.07	0.003				
	200–250	Hot rolled, normalized, annealed, or cold drawn	25–6	1–4	0.3	0.012	185	600	S4, S2	M2, M7
			5–3	5–10	0.2	0.008				
			2–1.5	11–19	0.12	0.005				
			1 and finer	20 and finer	0.07	0.003				
Wrought carbon steels										
Low carbon	85–125	Hot rolled, normalized, annealed, or cold drawn	25–6	1–4	0.3	0.012	160	525	S4, S2	M2, M7
1005 1015 1023			5–3	5–10	0.2	0.008				
1006 1016 1025			2–1.5	11–19	0.12	0.005				
1008 1017 1026			1 and finer	20 and finer	0.07	0.003				
1009 1018 1029	225–275	Annealed or cold drawn	25–6	1–4	0.3	0.012	115	375	S4, S2	M2, M7
1010 1019 1513			5–3	5–10	0.2	0.008				
1011 1020 1518			2–1.5	11–19	0.12	0.005				
1012 1021 1522			1 and finer	20 and finer	0.07	0.003				
1013 1022										
Medium carbon	125–175	Hot rolled, normalized, annealed, or cold drawn	25–6	1–4	0.3	0.012	135	450	S4, S2	M2, M7
1030 1044 1526			5–3	5–10	0.2	0.008				
1033 1045 1527			2–1.5	11–19	0.12	0.005				
1035 1046 1536			1 and finer	20 and finer	0.07	0.003				
1037 1049 1541	325–375	Quenched and tempered	25–6	1–4	0.3	0.012	84	275	S5	M3
1038 1050 1547			5–3	5–10	0.2	0.008				
1039 1053 1548			2–1.5	11–19	0.12	0.005				
1040 1055 1551			1 and finer	20 and finer	0.07	0.003				
1042 1524 1552										
1043 1525										
Wrought free-machining alloy steels										
Medium-carbon resulfurized	150–200	Hot rolled, normalized, annealed, or cold drawn	25–6	1–4	0.3	0.012	150	500	S4, S2	M2, M7
			5–3	5–10	0.2	0.008				
4140 4145Se			2–1.5	11–19	0.12	0.005				
4140Se 4147Te			1 and finer	20 and finer	0.07	0.003				
4142Te 4150	325–375	Quenched and tempered	25–6	1–4	0.3	0.012	76	250	S5	M3
			5–3	5–10	0.2	0.008				
			2–1.5	11–19	0.12	0.005				
			1 and finer	20 and finer	0.07	0.003				

(continued)

(a) Feed recommendations apply to conventional (axial-transverse) gear shaving. Feeds should be increased 100% for gears shaved by the diagonal (angular-transverse) method. Source: Metcut Research Associates Inc.

Table 1 (continued)

Material	Hardness, HB	Condition	Gear tooth size: Module	Gear tooth size: Diametral pitch	Feed per revolution of gear(a): mm	Feed per revolution of gear(a): in.	Cutter pitch line speed: m/min	Cutter pitch line speed: sfm	High-speed steel tool material: ISO	High-speed steel tool material: AISI
Medium- and high-carbon leaded	150–200	Hot rolled, normalized, annealed, or cold drawn	25–6	1–4	0.3	0.012	160	525	S4, S2	M2, M7
			5–3	5–10	0.2	0.008				
			2–1.5	11–19	0.12	0.005				
41L30 41L50 86L20			1 and finer	20 and finer	0.07	0.003				
41L40 43L40 86L40	325–375	Quenched and tempered	25–6	1–4	0.3	0.012	84	275	S5	M3
41L45 51L32			5–3	5–10	0.2	0.008				
41L47 52L100			2–1.5	11–19	0.12	0.005				
			1 and finer	20 and finer	0.07	0.003				
Wrought alloy steels										
Low carbon	125–175	Hot rolled, annealed, or cold drawn	25–6	1–4	0.3	0.012	145	475	S4, S2	M2, M7
4012 4621 6118			5–3	5–10	0.2	0.008				
4023 4718 8115			2–1.5	11–19	0.12	0.005				
4024 4720 8617			1 and finer	20 and finer	0.07	0.003				
4118 4815 8620	325–375	Normalized or quenched and tempered	25–6	1–4	0.3	0.012	76	250	S5	M3
4320 4817 8622			5–3	5–10	0.2	0.008				
4419 4820 8822			2–1.5	11–19	0.12	0.005				
4422 5015 9310			1 and finer	20 and finer	0.07	0.003				
4615 5115 94B15										
4617 5120 94B17										
4620										
Medium carbon	175–225	Hot rolled, annealed, or cold drawn	25–6	1–4	0.3	0.012	120	400	S4, S2	M2, M7
1330 4427 6150			5–3	5–10	0.2	0.008				
1335 4626 81B45			2–1.5	11–19	0.12	0.005				
1340 50B40 8625			1 and finer	20 and finer	0.07	0.003				
1345 50B44 8627	325–375	Normalized or quenched and tempered	25–6	1–4	0.3	0.012	69	225	S5	M3
4027 5046 8630			5–3	5–10	0.2	0.008				
4028 50B46 8637			2–1.5	11–19	0.12	0.005				
4032 50B50 8640			1 and finer	20 and finer	0.07	0.003				
4037 5060 8642										
4042 50B60 8645										
4047 5130 86B45										
4130 5132 8650										
4135 5135 8655										
4137 5140 8660										
4140 5145 8740										
4142 5147 8742										
4145 5150 9254										
4147 5155 9255										
4150 5160 9260										
4161 51B60 94B30										
4340										

(a) Feed recommendations apply to conventional (axial-transverse) gear shaving. Feeds should be increased 100% for gears shaved by the diagonal (angular-transverse) method. Source: Metcut Research Associates Inc.

relationship set between workpiece and cutter head velocities determines the helix angle.

Conventional and crowned helical gears can be cut with diameters from 19 to 102 mm (¾ to 4 in.), face widths up to 25 mm (1 in.), and helix angles from 10 to 40°. Gears can be generated at the rate of about 600 per hour, six to ten times as fast as hobbing. The time required for changing cutting tools is about 30 min, but this needs to be done only every 30 to 50 h. Although fast, the process is applicable only for the large-quantity production of a limited variety of gears.

Construction. Most reciprocating tools such as those used in gear shaping and planing operations are made of solid high-speed tool steel. Rotary cutters (hobs and milling cutters) for spur and helical gears can be either solid or inserted blade. Economy is the governing factor; the two methods of construction are equally satisfactory in terms of producing acceptable gears. Cutters less than about 75 mm (3 in.) in diameter are invariably solid. As cutters increase in size, the practice of using high-speed tool steel cutting edges (blades) as inserts in alloy steel bodies is usually more economical. Inserts are normally held by mechanical fasteners.

Rotary cutters (face mill, rotary broach, and other) for bevel gears are usually designed to use inserts. In rotary cutters, feed rate is a direct function of the number of cutting faces. Therefore, it is desirable to have as many cutting edges as possible to increase the amount of metal removed during each revolution. However, tooth strength and chip clearance must be considered when selecting the number of blades (cutting edges). Despite the limitations imposed by chip clearance and tooth strength, gear cutters are often redesigned for greater efficiency.

The G-Trac gear generator is a special machine that uses large numbers of single-point high-speed tool steel cutting tools carried on an endless chain of tool blocks. One model for moderate quantities of gears has a single row of cutting tools (Fig. 22). As these are sped around the track, the work is fed into the cutters and rolled as though in mesh with a single tooth simulated by the cutters. Thus, one tooth space is generated. The work is then withdrawn, indexed to the next tooth space, fed in again, and so on, around the gear.

Another G-Trac model for large quantities has a number of rows of teeth (typically 14) around the chain of blocks. The work is fed into the cutters and revolved continuously as though in mesh with a simulated rack, until all tooth spaces are finished. On both models, spur and helical gears (up to 45° helix angle) as large as 356 mm (14 in.) in diameter can be cut and crowned and can be stacked to a height of 190 mm (7½ in.) times the cosine of the helix angle.

G-Trac generator cutter life is 6 to 175 times as long as that for hobs. Special machines are available for sharpening the cutters, and the cutters can also be sharpened all at one time in a special fixture on a standard surface grinder. A G-Trac machine has a 19 kW (25 hp) motor. It is six to ten times faster than hobbing, but the machine is expensive.

Speed and Feed

In addition to the type of gear being machined and the method of cutting, other factors that influence the choice of cutting speed are work metal composition and hardness, diametral pitch of the work gear, rigidity of the setup, tolerance and finish requirements, and cutting fluid used.

Hobbing Speed. Nominal speeds and feeds for gear hobbing that take into account several of the above variables are listed in Table 2. As carbon content, alloy content, and hardness of the steel workpiece increase, recommended cutter speed decreases; for any given work metal,

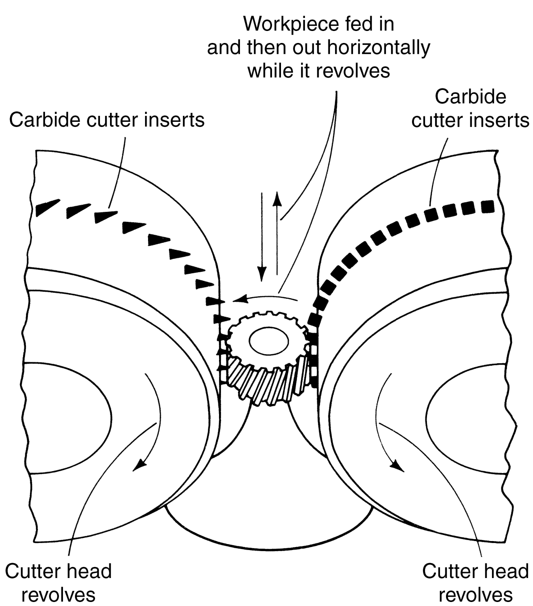


Fig. 21 Schematic of operation of the Tangear gear cutter

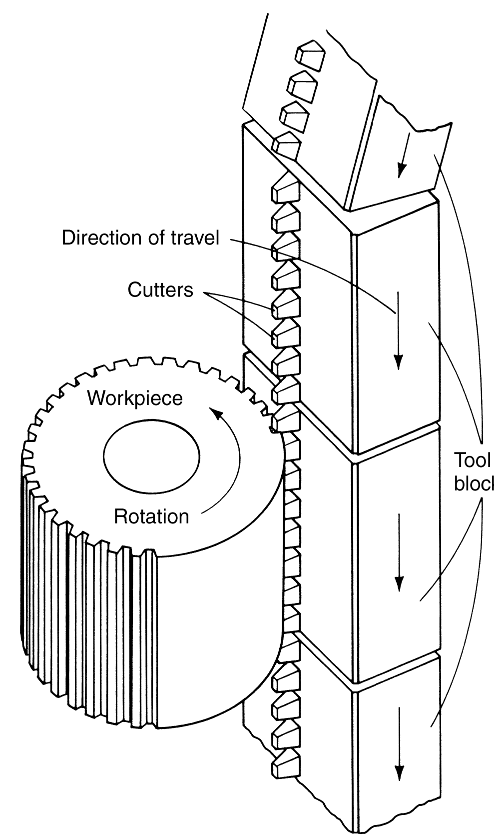


Fig. 22 Method of operation of a G-Trac gear generator

Table 2 Feeds and speeds for the hobbing of carbon and low-alloy steel gears with high-speed steel tools

For hobbing class 9 (per AGMA 390.03) or better gears, it may be necessary to reduce speeds and feeds by 50% and/or take two cuts. To meet finish requirements, it may be necessary to try both conventional and climb cutting.

			Gear tooth size			Feed per revolution of workpiece(a)		Hob speed		High-speed steel tool material	
Material	Hardness, HB	Condition	Module	Diametral pitch	Number of cuts	mm	in.	m/min	sfm	ISO	AISI
Wrought free-machining carbon steels											
Low-carbon	100–150	Hot rolled or	25–13	1–2	2	1.50	0.060	67	220	S4, S2	M2, M7
resulfurized		annealed	12–2.5	3–10	1	1.50	0.060	70	230		
1116			2–1.5	11–19	1	1.50	0.060	73	240		
1117			1–0.5	20–48	1	1.25	0.050	76	250		
1118			0.5 and finer	48 and finer	1	0.75	0.030	85	280		
1119	150–200	Cold drawn	25–13	1–2	2	1.50	0.060	70	230	S4, S2	M2, M7
1211			12–2.5	3–10	1	1.50	0.060	73	240		
1212			2–1.5	11–19	1	1.50	0.060	81	265		
			1–0.5	20–48	1	1.25	0.050	84	275		
			0.5 and finer	48 and finer	1	0.75	0.030	90	300		
Medium-carbon	175–225	Hot rolled,	25–13	1–2	2	1.50	0.060	52	170	S4, S2	M2, M7
resulfurized		normalized,	12–2.5	3–10	1	1.50	0.060	55	180		
1132 1144		annealed, or	2–1.5	11–19	1	1.50	0.060	58	190		
1137 1145		cold drawn	1–0.5	20–48	1	1.25	0.050	60	200		
1139 1146			0.5 and finer	48 and finer	1	0.75	0.030	67	220		
1140 1151	325–375	Quenched and	25–13	1–2	2	1.50	0.060	21	70	S5, S11	M3, M42
1141		tempered	12–2.5	3–10	1	1.50	0.060	27	90		
			2–1.5	11–19	1	1.50	0.060	34	110		
			1–0.5	20–48	1	1.25	0.050	37	120		
			0.5 and finer	48 and finer	1	0.75	0.030	40	130		
Low-carbon	100–150	Hot rolled,	25–13	1–2	2	1.65	0.065	78	255	S4, S2	M2, M7
leaded		normalized,	12–2.5	3–10	1	1.65	0.065	81	265		
12L13		annealed, or	2–1.5	11–19	1	1.50	0.060	90	295		
12L14		cold drawn	1–0.5	20–48	1	1.25	0.050	95	305		
12L15			0.5 and finer	48 and finer	1	0.75	0.030	105	340		
	200–250	Hot rolled,	25–13	1–2	2	1.50	0.060	50	165	S4, S2	M2, M7
		normalized,	12–2.5	3–10	1	1.50	0.060	58	190		
		annealed, or	2–1.5	11–19	1	1.50	0.060	69	225		
		cold drawn	1–0.5	20–48	1	1.25	0.050	73	240		
			0.5 and finer	48 and finer	1	0.75	0.030	84	275		
Wrought carbon steels											
Low carbon	85–125	Hot rolled,	25–13	1–2	2	1.80	0.070	62	205	S4, S2	M2, M7
1005 1016 1026		normalized,	12–2.5	3–10	1	1.80	0.070	75	245		
1006 1017 1029		annealed, or	2–1.5	11–19	1	1.50	0.060	81	265		
1008 1018 1513		cold drawn	1–0.5	20–48	1	1.25	0.050	85	280		
1009 1019 1518			0.5 and finer	48 and finer	1	0.75	0.030	88	290		
1010 1020 1522	225–275	Annealed or	25–13	1–2	2	1.50	0.060	37	120	S4, S2	M2, M7
1011 1021		cold drawn	12–2.5	3–10	1	1.50	0.060	49	160		
1012 1022			2–1.5	11–19	1	1.50	0.060	56	185		
1013 1023			1–0.5	20–48	1	1.25	0.050	59	195		
1015 1025			0.5 and finer	48 and finer	1	0.75	0.030	60	200		
Medium	125–175	Hot rolled,	25–13	1–2	2	1.80	0.070	50	165	S4, S2	M2, M7
carbon		normalized,	12–2.5	3–10	1	1.80	0.070	53	175		
1030 1044 1526		annealed, or	2–1.5	11–19	1	1.50	0.060	56	185		
1033 1045 1527		cold drawn	1–0.5	20–48	1	1.25	0.050	60	200		
1035 1046 1536			0.5 and finer	48 and finer	1	0.75	0.030	64	210		
1037 1049 1541	325–375	Quenched and	25–13	1–2	2	1.15	0.045	21	70	S5, S11	M3, M42
1038 1050 1547		tempered	12–2.5	3–10	1	1.15	0.045	24	80		
1039 1053 1548			2–1.5	11–19	1	0.65	0.025	30	100		
1040 1055 1551			1–0.5	20–48	1	0.65	0.025	32	105		
1042 1524 1552			0.5 and finer	48 and finer	1	0.65	0.025	37	120		
1043 1525											
Wrought free-machining alloy steels											
Medium-carbon	150–200	Hot rolled,	25–13	1–2	2	1.50	0.060	44	145	S4, S2	M2, M7
resulfurized		normalized,	12–2.5	3–10	1	1.50	0.060	47	155		
4140		annealed, or	2–1.5	11–19	1	1.50	0.060	50	165		
4140Se		cold drawn	1–0.5	20–48	1	1.25	0.050	55	180		
4142Te			0.5 and finer	48 and finer	1	0.75	0.030	60	200		
4145Se	325–375	Quenched and	25–13	1–2	2	1.15	0.045	20	65	S5, S11	M3, M42
4147Te		tempered	12–2.5	3–10	1	1.15	0.045	23	75		
4150			2–1.5	11–19	1	0.65	0.025	29	95		
			1–0.5	20–48	1	0.65	0.025	30	100		
			0.5 and finer	48 and finer	1	0.65	0.025	41	135		

(continued)

(a) Feeds are based on the largest standard recommended hob diameter. When using a smaller hob diameter, the feed must be reduced proportionally. Source: Metcut Research Associates Inc.

Table 2 (continued)

Material	Hardness, HB	Condition	Gear tooth size: Module	Gear tooth size: Diametral pitch	Number of cuts	Feed per revolution of workpiece(a): mm	Feed per revolution of workpiece(a): in.	Hob speed: m/min	Hob speed: sfm	High-speed steel tool material: ISO	High-speed steel tool material: AISI
Medium- and high-	150–200	Hot rolled,	25–13	1–2	2	1.50	0.060	52	170	S4, S2	M2, M7
carbon leaded		normalized,	12–2.5	3–10	1	1.50	0.060	55	180		
41L30 43L40		annealed, or	2–1.5	11–19	1	1.50	0.060	58	190		
41L40 51L32		cold drawn	1–0.5	20–48	1	1.25	0.050	60	200		
41L45 52L100			0.5 and finer	48 and finer	1	0.75	0.030	67	220		
41L47 86L20	325–375	Quenched and	25–13	1–2	2	1.15	0.045	21	70	S5, S11	M3, M42
41L50 86L4		tempered	12–2.5	3–10	1	1.15	0.045	27	90		
			2–1.5	11–19	1	1.15	0.045	34	110		
			1–0.5	20–48	1	1.15	0.045	37	120		
			0.5 and finer	48 and finer	1	0.75	0.030	40	130		
Wrought alloy steels											
Low carbon	125–175	Hot rolled,	25–13	1–2	2	1.50	0.060	49	160	S4, S2	M2, M7
4012 4621 6118		annealed, or	12–2.5	3–10	1	1.50	0.060	52	170		
4023 4718 8115		cold drawn	2–1.5	11–19	1	1.50	0.060	55	180		
4024 4720 8617			1–0.5	20–48	1	1.25	0.050	58	190		
4118 4815 8620			0.5 and finer	48 and finer	1	0.75	0.030	64	210		
4320 4817 8622	275–325	Normalized or	25–13	1–2	2	1.15	0.045	27	90	S4, S2	M2, M7
4419 4820 8822		quenched and	12–2.5	3–10	1	1.15	0.045	30	100		
4422 5015 9310		tempered	2–1.5	11–19	1	1.15	0.045	34	110		
4615 5115 94B15			1–0.5	20–48	1	1.15	0.045	37	120		
4617 5120 94B17			0.5 and finer	48 and finer	1	0.75	0.030	40	130		
4620											
Medium carbon	175–225	Hot rolled,	25–13	1–2	2	1.50	0.060	38	125	S4, S2	M2, M7
1330 4427 81B45		annealed, or	12–2.5	3–10	1	1.50	0.060	41	135		
1335 4626 8625		cold drawn	2–1.5	11–19	1	1.25	0.060	44	145		
1340 50B40 8627			1–0.5	20–48	1	1.25	0.050	49	160		
1345 50B44 8630			0.5 and finer	48 and finer	1	0.75	0.030	55	180		
4027 5046 8637	325–375	Normalized or	25–13	1–2	2	1.15	0.045	18	60	S5, S11	M3, M42
4028 50B46 8640		quenched and	12–2.5	3–10	1	1.15	0.045	20	65		
4032 50B50 8642		tempered	2–1.5	11–19	1	0.65	0.025	23	75		
4037 5060 8645			1–0.5	20–48	1	0.65	0.025	24	80		
4042 50B60 86B45			0.5 and finer	48 and finer	1	0.65	0.025	29	95		
4047 5130 8650											
4130 5132 8655											
4135 5135 8660											
4137 5140 8740											
4140 5145 8742											
4142 5147 9254											
4145 5150 9255											
4147 5155 9260											
4150 5160 94B30											
4161 51B60											
4340 6150											

(a) Feeds are based on the largest standard recommended hob diameter. When using a smaller hob diameter, the feed must be reduced proportionally. Source: Metcut Research Associates Inc.

cutting speed increases as diametral pitch becomes finer. For the numerical values given in Table 2. it is assumed that an additive-type cutting fluid will be used. Low cutting speeds prolong tool life and eliminate the difficulties involved in changing tools and matching cuts.

Shaping speed is influenced by variables similar to those that affect hobbing speeds, with the exception of diametral pitch. Nominal speeds and feeds for gear shaping under a variety of conditions are given in Table 3. Cutter speed varies with the composition and hardness of the work metal but not with diametral pitch. The speeds listed in Table 3 are generally conservative.

Speeds for Interlocking and Face Mill Cutters. Nominal speeds for cutting straight and spiral bevel gears with two common types of cutters are given in Table 4. As in shaping, speed is not varied with diametral pitch when other conditions are the same. For both the interlocking cutters and the face mill cutters, speeds are increased for finishing when roughing and finishing are done separately. The speeds used for cutting straight and spiral bevel gears with rotating cutters are close to those suggested in Table 4. When straight bevel gears are being cut with reciprocating cutters, much lower speeds are required.

Feeds. Nominal feeds for the hobbing of steel gears are included in Table 2. Values are given in millimeters per revolution (inches per revolution) of the gear being hobbed and are

Table 3 Feeds and speeds for the shaping of carbon and low-alloy steel gears with high-speed steel tools

Material	Hardness, HB	Condition	Number of cuts(a)	Gear tooth size: Module	Gear tooth size: Diametral pitch	Rotary feed per cutter stroke (102 mm, or 4 in., pitch diameter cutter)(a): mm	Rotary feed per cutter stroke (102 mm, or 4 in., pitch diameter cutter)(a): in.	Cutter speed: m/min	Cutter speed: sfm	High-speed steel tool material: ISO	High-speed steel tool material: AISI
Wrought free-machining carbon steels											
Low-carbon resulfurized	100–150	Hot rolled	4	25–6	1–4	0.55	0.022	26	85	S4, S2	M2, M7
1116 1119		or annealed	2	5.5–2.5	5–10	0.40	0.016				
1117 1211			2	2–1.5	11–19	0.28	0.011				
1118 1212			2	1–0.5	20–48	0.20	0.008				
	150–200	Cold drawn	4	25–6	1–4	0.55	0.022	27	90	S4, S2	M2, M7
			2	5.5–2.5	5–10	0.40	0.016				
			2	2–1.5	11–19	0.28	0.011				
			2	1–0.5	20–48	0.20	0.008				
Medium-carbon resulfurized	175–225	Hot rolled,	4	25–6	1–4	0.55	0.022	24	80	S4, S2	M2, M7
		normalized,	2	5.5–2.5	5–10	0.40	0.016				
1132 1144		annealed, or	2	2–1.5	11–19	0.28	0.011				
1137 1145		cold drawn	2	1–0.5	20–48	0.20	0.008				
1139 1146	325–375	Quenched and	4	25–6	1–4	0.30	0.012	12	40	S5, S11	M3, M42
1140 1151		tempered	2	5.5–2.5	5–10	0.25	0.010				
1141			2	2–1.5	11–19	0.20	0.008				
			2	1–0.5	20–48	0.20	0.008				
Low-carbon leaded	100–150	Hot rolled,	4	25–6	1–4	0.55	0.022	38	125	S4, S2	M2, M7
12L13		normalized,	2	5.5–2.5	5–10	0.40	0.016				
12L14		annealed, or	2	2–1.5	11–19	0.28	0.011				
12L15		cold drawn	2	1–0.5	20–48	0.20	0.008				
	200–250	Hot rolled,	4	25–6	1–4	0.55	0.022	27	90	S4, S2	M2, M7
		normalized,	2	5.5–2.5	5–10	0.40	0.016				
		annealed, or	2	2–1.5	11–19	0.28	0.011				
		cold drawn	2	1–0.5	20–48	0.20	0.008				
Wrought carbon steels											
Low carbon	85–125	Hot rolled,	4	25–6	1–4	0.45	0.018	27	90	S4, S2	M2, M7
1005 1015 1023		normalized,	2	5.5–2.5	5–10	0.30	0.012				
1006 1016 1025		annealed, or	2	2–1.5	11–19	0.25	0.010				
1008 1017 1026		cold drawn	2	1–0.5	20–48	0.20	0.008				
1009 1018 1029	225–275	Annealed or	4	25–6	1–4	0.40	0.016	18	60	S4, S2	M2, M7
1010 1019 1513		cold drawn	2	5.5–2.5	5–10	0.30	0.012				
1011 1020 1518			2	2–1.5	11–19	0.25	0.010				
1012 1021 1522			2	1–0.5	20–48	0.20	0.008				
1013 1022											
Medium carbon	125–175	Hot rolled,	4	25–6	1–4	0.45	0.018	23	75	S4, S2	M2, M7
1030 1044 1526		normalized,	2	5.5–2.5	5–10	0.30	0.012				
1033 1045 1527		annealed, or	2	2 –1.5	11–19	0.25	0.010				
1035 1046 1536		cold drawn	2	1–0.5	20–48	0.20	0.008				
1037 1049 1541	325–375	Quenched and	4	25–6	1–4	0.30	0.012	12	40	S5, S11	M3, M42
1038 1050 1547		tempered	2	5.5–2.5	5–10	0.25	0.010				
1039 1053 1548			2	2–1.5	11–19	0.20	0.008				
1040 1055 1551			2	1–0.5	20–48	0.20	0.008				
1042 1524 1552											
1043 1525											
Wrought free-machining alloy steels											
Medium-carbon resulfurized	150–200	Hot rolled,	4	25–6	1–4	0.55	0.022	24	80	S4, S2	M2, M7
		normalized,	2	5.5–2.5	5–10	0.45	0.018				
4140 4145Se		annealed, or	2	2–1.5	11–19	0.28	0.011				
4140Se 4147Te		cold drawn	2	1–0.5	20–48	0.20	0.008				
4142Te 4150	325–375	Quenched and	4	25–6	1–4	0.30	0.012	11	35	S5, S11	M3, M42
		tempered	2	5.5–2.5	5–10	0.25	0.010				
			2	2–1.5	11–19	0.20	0.008				
			2	1–0.5	20–48	0.20	0.008				
Medium- and high-carbon leaded	150–200	Hot rolled,	4	25–6	1–4	0.55	0.022	24	80	S4, S2	M2, M7
		normalized,	2	5.5–2.5	5–10	0.40	0.016				
41L30 41L50 52L100		annealed, or	2	2–1.5	11–19	0.28	0.011				
41L40 43L40 86L20		cold drawn	2	1–0.5	20–48	0.20	0.008				
41L45 51L32 86L40	325–375	Quenched and	4	25–6	1–4	0.30	0.012	11	35	S5, S11	M3, M42
41L47		tempered	2	5.5–2.5	5–10	0.25	0.010				
			2	2–1.5	11–19	0.20	0.008				
			2	1–0.5	20–48	0.20	0.008				

(continued)

(a) For cutting gears of class 9 (AGMA 390.03) or better, the rotary feed should be reduced, and the number of cuts should be increased. Source: Metcut Research Associates Inc.

Table 3 (continued)

Material	Hardness, HB	Condition	Number of cuts(a)	Gear tooth size: Module	Gear tooth size: Diametral pitch	Rotary feed per cutter stroke (102 mm, or 4 in., pitch diameter cutter)(a): mm	Rotary feed: in.	Cutter speed: m/min	Cutter speed: sfm	High-speed steel tool material: ISO	High-speed steel tool material: AISI
Wrought alloy steels											
Low carbon	125–175	Hot rolled,	4	25–6	1–4	0.45	0.018	21	70	S4, S2	M2, M7
4012 4621 6118		annealed, or	2	5.5–2.5	5–10	0.30	0.012				
4023 4718 8115		cold drawn	2	2–1.5	11–19	0.25	0.010				
4024 4720 8617			2	1–0.5	20–48	0.20	0.008				
4118 4815 8620	325–375	Normalized	4	25–6	1–4	0.30	0.012	12	40	S5, S11	M3, M42
4320 4817 8622		or quenched	2	5.5–2.5	5–10	0.25	0.010				
4419 4820 8822		and tempered	2	2–1.5	11–19	0.20	0.008				
4422 5015 9310			2	1–0.5	20–48	0.20	0.008				
4615 5115 94B15											
4617 5120 94B17											
4620											
Medium carbon	175–225	Hot rolled,	4	25–6	1–4	0.45	0.018	18	60	S4, S2	M2, M7
1330 4427 81B45		annealed, or	2	5.5–2.5	5–10	0.30	0.012				
1335 4626 8625		cold drawn	2	2–1.5	11–19	0.25	0.010				
1340 50B40 8627			2	1–0.5	20–48	0.20	0.008				
1345 50B44 8630	325–375	Normalized	4	25–6	1–4	0.30	0.012	11	35	S5, S11	M3, M42
4027 5046 8637		or quenched	2	5.5–2.5	5–10	0.25	0.010				
4028 50B46 8640		and tempered	2	2–1.5	11–19	0.20	0.008				
4032 50B50 8642			2	1–0.5	20–48	0.20	0.008				
4037 5060 8645											
4042 50B60 86B45											
4047 5130 8650											
4130 5132 8655											
4135 5135 8660											
4137 5140 8740											
4140 5145 8742											
4142 5147 9254											
4145 5150 9255											
4147 5155 9620											
4150 5160 94B30											
4161 51B60											
4340 6150											

(a) For cutting gears of class 9 (AGMA 390.03) or better, the rotary feed should be reduced, and the number of cuts should be increased. Source: Metcut Research Associates Inc.

based on the assumption that the gears will be ground after hobbing. Hobbing feeds should be decreased when gears are to be finished by shaving. As indicated in Table 2, hobbing feeds are sensitive to differences in diametral pitch, but feeds are not ordinarily changed for differences in the composition or hardness of the gear steel when diametral pitch is the same, except for the coarser-pitch heat-treated gears.

Nominal feeds for cutting gears by shaping are given in Table 3. These values are based on the assumption that the gears will be finished by grinding; a reduction in speed is recommended when gears are to be finished by shaving. Feeds for shaping vary with diametral pitch and vary comparatively little with work metal composition and hardness. Conditions for a specific operation may require feeds higher or lower than those given in Table 3.

Feeds for cutting straight and spiral bevel gears by rotary cutters are suggested in Table 4. The optimum feed is greatly influenced by diametral pitch; work metal composition and hardness have a minor influence. The same rates of feed are commonly used for both roughing and finishing, although the cutting edges of the teeth are altered for finishing, as indicated in Table 4.

Cutting Fluids

Cutting fluids are recommended for all gear cutting, although for some applications (notably the cutting of large gears) the use of a cutting fluid is impractical. The three cutting fluids most commonly used for gear cutting are:

- Mineral oil (without additives) having a viscosity of 140 to 220 SUS at 40 °C (100 °F)
- Cutting oil, sulfurized or chlorinated (or both), usually diluted with mineral oil to a viscosity of 180 to 220 SUS at 40 °C (100 °F)
- Motor oil (SAE 20 grade), either detergent or nondetergent type

Table 4 Gear cutting speeds for the rough and finish cutting of carbon and low-alloy steel straight and spiral bevel gears with high-speed steel tools

Material			Hardness, HB	Condition	Cutting speed(a) Roughing m/min	Roughing sfm	Finishing m/min	Finishing sfm	High-speed steel tool material ISO	AISI
Wrought free-machining carbon steels										
Low-carbon resulfurized			100–150	Hot rolled or annealed	38	125	76	250	S4, S2	M2, M7
1116	1118	1211	150–200	Cold drawn	43	140	84	275	S4, S2	M2, M7
1117	1119	1212								
Medium-carbon resulfurized			175–225	Hot rolled, normalized, annealed, or cold drawn	26	85	53	175	S4, S2	M2, M7
1132	1140	1145								
1137	1141	1146	325–375	Quenched and tempered	15	50	30	100	S5	M3
1139	1144	1151								
Low-carbon leaded			100–150	Hot rolled, normalized, annealed, or cold drawn	44	145	88	290	S4, S2	M2, M7
12L13	12L15									
12L14			200–250	Hot rolled, normalized, annealed, or cold drawn	34	110	69	225	S4, S2	M2, M7
Wrought carbon steels										
Low carbon			85–125	Hot rolled, normalized, annealed, or cold drawn	27	90	58	190	S4, S2	M2, M7
1005	1015	1023								
1006	1016	1025	225–275	Annealed or cold drawn	21	70	44	145	S4, S2	M2, M7
1008	1017	1026								
1009	1018	1029								
1010	1019	1513								
1011	1020	1518								
1012	1021	1522								
1013	1022									
Medium carbon			125–175	Hot rolled, normalized, annealed, or cold drawn	26	85	53	175	S4, S2	M2, M7
1030	1044	1526								
1033	1045	1527	325–375	Quenched and tempered	15	50	30	100	S5	M3
1035	1046	1536								
1037	1049	1541								
1038	1050	1547								
1039	1053	1548								
1040	1055	1551								
1042	1524	1552								
1043	1525									
Wrought free-machining alloy steels										
Medium-carbon resulfurized			150–200	Hot rolled, normalized, annealed, or cold drawn	26	85	53	175	S4, S2	M2, M7
4140	4142Te	4147Te								
4140Se	4145Se	4150	325–375	Quenched and tempered	14	45	29	95	S5	M3
Medium- and high-carbon leaded			150–200	Hot rolled, normalized, annealed, or cold drawn	26	85	53	175	S4, S2	M2, M7
41L30	41L50	52L100								
41L40	43L40	86L20								
41L45	51L32	86L40	325–375	Quenched and tempered	14	45	27	90	S5	M3
41L47										
Wrought alloy steels										
Low carbon			125–175	Hot rolled, annealed, or cold drawn	26	85	53	175	S4, S2	M2, M7
4012	4621	6118								
4023	4718	8115	325–375	Normalized or quenched and tempered	15	50	30	100	S5	M3
4024	4720	8617								
4118	4815	8620								
4320	4817	8622								
4419	4820	8822								
4422	5015	9310								
4615	5115	94B15								
4617	5120	94B17								
4620										
Medium carbon			175–225	Hot rolled, annealed, or cold drawn	20	65	41	135	S4, S2	M2, M7
1330	4047	4161								
1335	4130	4340	325–375	Normalized or quenched and tempered	15	50	30	100	S5	M3
1340	4135	4427								
1345	4137	4626								
4027	4140	50B40								
4028	4142	50B44								
4032	4145	5046								
4037	4147	50B46								
4042	4150	50B50								

(continued)

(a) Cutting speed recommendations are for use with alternate-tooth milling cutters or gear generators. For planer-type generators, use the recommended cutting speeds in Table 3. Source: Metcut Research Associates Inc.

Table 4 (continued)

Material			Hardness, HB	Condition	Cutting speed(a) Roughing m/min	Roughing sfm	Finishing m/min	Finishing sfm	High-speed steel tool material ISO	AISI
Medium carbon			175–225	Hot rolled, annealed,	20	65	41	135	S4, S2	M2, M7
5060	51B60	86B45		or cold drawn						
50B60	6150	8650	325–375	Normalized or	15	50	30	100	S5	M3
5130	81B45	8655		quenched and tempered						
5132	8625	8660								
5135	8627	8740								
5140	8630	8742								
5145	8637	9254								
5147	8640	9255								
5150	8642	9260								
5155	8645	94B30								
5160										

(a) Cutting speed recommendations are for use with alternate-tooth milling cutters or gear generators. For planer-type generators, use the recommended cutting speeds in Table 3. Source: Metcut Research Associates Inc.

Water-soluble oils, which are widely used in many metal cutting operations, are used to a lesser extent for gear cutting because the oils referred to above are more effective for producing the surface finishes desired on gear teeth and for prolonged cutter life. Nevertheless, soluble oils are sometimes used in gear cutting. One notable application for soluble oil (20 parts water to 1 part oil) is the finish broaching of fine-pitch gears.

The relative advantages and disadvantages of the three oils mentioned above are matters of opinion among different plants producing steel gears. Most data in speed and feed tables are based on the use of cutting oils with additives (the second of the cutting fluids listed above). However, in some plants, mineral oils (including motor oils) are used when records have proved that the use of these oils resulted in longer cutter life than when prepared cutting oils were used.

In some plants, additive cutting oils are used for roughing operations and mineral oils for finishing when the operations are done separately in different machines. An adequate supply (a flood) of fluid under slight pressure (about 35 kPa, or 5 psi) at the cutting area is extremely important—usually more important than the composition of the oil.

Comparison of Steels for Gear Cutting

End use is the main factor in the selection of steel for a specific gear. However, two or more steels will often serve equally well. Under these conditions, the cost of the steel and the cost of processing it are significant factors in making a final selection, and in all selections both these factors must be considered. When the cost of machining standard steels is compared with the cost of free-machining counterparts, the use of a free-machining grade will almost always result in lower machining costs. However, when changing from one standard grade of a given steel to its free-machining counter-part, the additional cost of the free-machining grade must be considered in the cost comparison. When changes in steel composition that result in lower machining costs require changes in the method of heat treating, the cost of heat treating must also be considered.

A change in microstructure, independent of steel composition, also has a significant effect on tool life or surface finish or both. Altering the heat treatment can produce a more machinable microstructure that results in increased cutter life.

Computer Numerical Control (CNC) Milling and Hobbing Machines

In practice, gear milling is usually confined to replacement gears or to the small-lot production, roughing, and finishing of coarse-pitch gears. The milling of low-production, special tooth forms is also done. Milling is often used as the first operation prior to finish hobbing or shaping, especially in large diametral pitch spur, helical, and straight bevel gears. The gear is then finished to final size by another method selected by the manufacturer.

Special high-power (56 to 75 kW, or 75 to 100 hp) CNC machines, called gashers (Fig.

23), are used for the roughing of gears from 6 to ⅝ diametral pitch. These machines employ strong, rigid machine components and are extremely powerful. They operate at peak efficiency with large inserted carbide cutters. The ability to remove large amounts of metal at substantial feed rates makes this process quite economical for coarse pitches. The gears can be finished on the same machines with high-speed tool steel formed cutters. The accuracy of index on these machines is more than adequate to produce tooth spacing requirements.

The advent of the CNC-controlled index mechanism of the gasher allows the manufacture of large, coarse-pitch sprockets for roller chains of all sizes. The face capacity of these machines permits the stacking of many plate sprockets at one time with good quality and reliable production.

The power of these machines and their ability for two-axis interpolation make coarse-tooth straight bevel gears a natural application. The time savings can be significant; two-tool generators can be used to finish the gashed bevel blank.

Computer numerical control gear hobbing machines are available in 305, 508, 914, 1220, and 1625 mm (12, 20, 36, 48, and 64 in.) capacities. Figure 24 shows a CNC machine hobbing a gear.

Grinding of Gears

Gear teeth can be produced entirely by grinding, entirely by cutting, or by first cutting and then grinding to the required dimensions. Usually, gear grinding removes only several hundredths of a millimeter of metal from precut

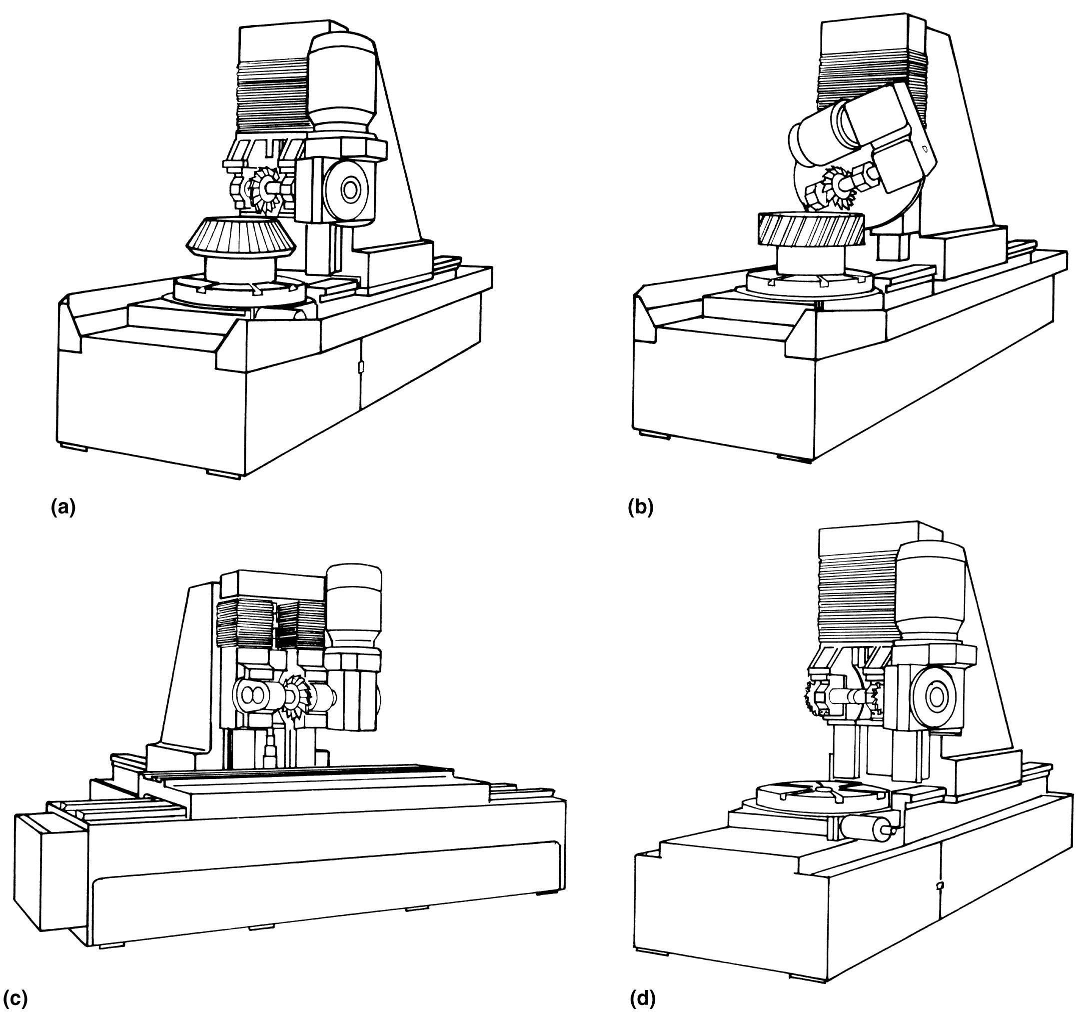


Fig. 23 Four types of gashers. (a) Duo-axis. (b) Helical machine. (c) Rack machine. (d) Slitting head

gears to make accurate teeth for critical applications. Teeth made entirely by grinding are usually only those of fine pitch, for which the total amount of metal removed is small. The grinding of fine-pitch gear teeth from uncut blanks may be less costly than the two-step procedure if there is not much metal to be removed.

The two basic methods for the grinding of gear teeth are form grinding (nongenerating) and generation grinding. Many varieties of machines have been built specifically to grind gears and pinions.

Advantages and Disadvantages. A gear is ground when it is so hard that it cannot be finished by other methods or when the required accuracy is greater than can be obtained by other methods. If the hardness of the gear does not exceed 40 HRC and if its size is within the capacity of the machine, shaving will produce teeth of an accuracy close to that produced by grinding. Gears made of steel harder than 40 HRC are finished by grinding when dimensional accuracy is required or when correction beyond that feasible by honing is required.

The major disadvantage of grinding gears is the cost. It is usually more expensive to grind gears than to cut them because material is removed in small increments in grinding. Ground gears are usually subjected to more inspection than cut gears. Magnetic particle inspection of the finished teeth detects fine cracks, and macroetching with dilute nitic acid detects grinding burn. Both inspection procedures are advised in gear grinding.

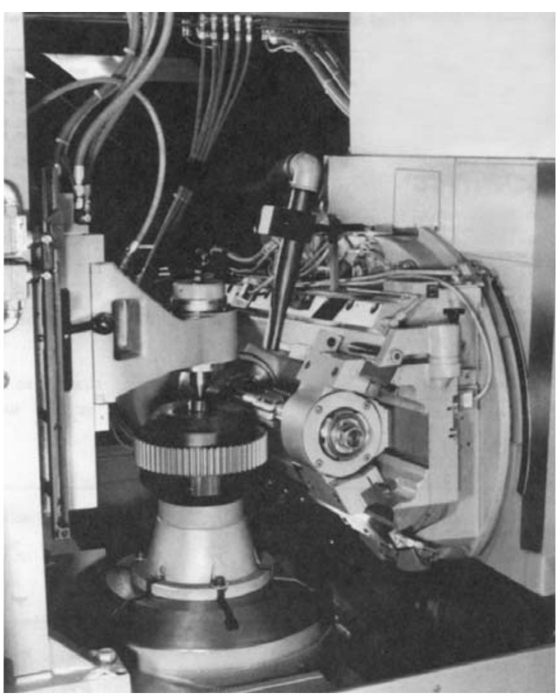


Fig. 24 CNC gear hobbing machine producing a gear

A carbide network in carburized steels, or the presence of sufficient retained austenite or of hydrogen in any hardened steel, can increase the susceptibility of the steel to grinding cracks. Therefore, it is necessary to keep these conditions under control. Reducing the heat generated in grinding (by decreasing the rate of stock removal or by the use of a grinding fluid) can help to eliminate cracking and burning of the ground surface.

Stock Allowance. When workpieces are prepared for grinding by cutting, the least amount of stock possible should be left to be removed by grinding. This stock should be evenly distributed to reduce the wheel dressings and operating adjustments needed to maintain wheel contour when using conventional (aluminum oxide or silicon carbide) dressable wheels. With cubic boron nitride (CBN) wheels, where stock is removed in one pass, evenly distributed stock results in more uniform grinding pressure.

In general, it is not practical to remove more than 0.15 mm (0.006 in.) of stock from each face of the teeth of spur gears up to 178 mm (7 in.) in diameter or 0.20 mm (0.008 in.) from gears between 178 and 254 mm (7 and 10 in.) in diameter. These amounts of stock can usually be removed in three to five passes with conventional dressable wheels or in one pass with CBN wheels. However, the number of passes depends on number of teeth, pressure angle, and required finish. Sometimes nine or ten passes are needed with conventional wheels.

The recommended stock allowance for grinding straight bevel teeth is 0.08 mm (0.003 in.) at the root of the tooth and 0.08 to 0.13 mm (0.003 to 0.005 in.) on the face of each tooth. Grinding time depends on the amount of stock to be removed. For example, removing 0.076 to 0.089 mm (0.0030 to 0.0035 in.) of stock from a gear of 4 diametral pitch and 32 teeth takes 30 s per tooth; a gear of 10 diametral pitch and 32 teeth takes about 20 s per tooth. Figure 25, which shows the cross sections of gear teeth of various diametral pitches, can be used to gage the relative sizes of the 4 and 10 diametral pitch gears.

Grinding stock allowance per tooth side for spiral bevel and hypoid gears should be 0.10 to 0.13 mm (0.004 to 0.005 in.) for gears coarser than 10 diametral pitch and 0.08 to 0.10 mm

(0.003 to 0.004 in.) for gears of 10 to 20 diametral pitch. Stock allowance at the root of the tooth should be about 0.08 mm (0.003 in.) for gears up to 20 diametral pitch.

Multiple-Piece Loads. Because gear grinding is an expensive operation, it is important to control the amount of metal to be removed by grinding, to reduce runout and spacing errors, and to try to grind multiple-piece loads. Multiple-start hobs can cause enough spacing error to make extra grinding passes necessary. Therefore, when a hobbed gear is to be ground, it is better to use a single-thread hob. The multiple-start hob will not save enough to pay for the extra grinding passes. Single-thread hobs are available with nonground class-C tooth forms.

Wheel Specifications. Aluminum oxide wheels are preferred for the grinding of teeth in through-hardened and case-hardened steel gears. For nitrided gears, silicon carbide wheels are sometimes used. Grit size, which depends mainly on the type of grinding and the pitch of the gear, ranges from 46 to 100, although somewhat finer grits can be used for form grinding. A grit size of 60 is generally recommended for diametral pitch in the range of 1 to 10; 80 for the range of 8 to 12; and 100 for gears of pitch finer than 14. With crush-formed wheels, grit sizes as small as 400 are used in grinding gears of diametral pitch as fine as 200. For grade (hardness), a range of H to M is generally recommended.

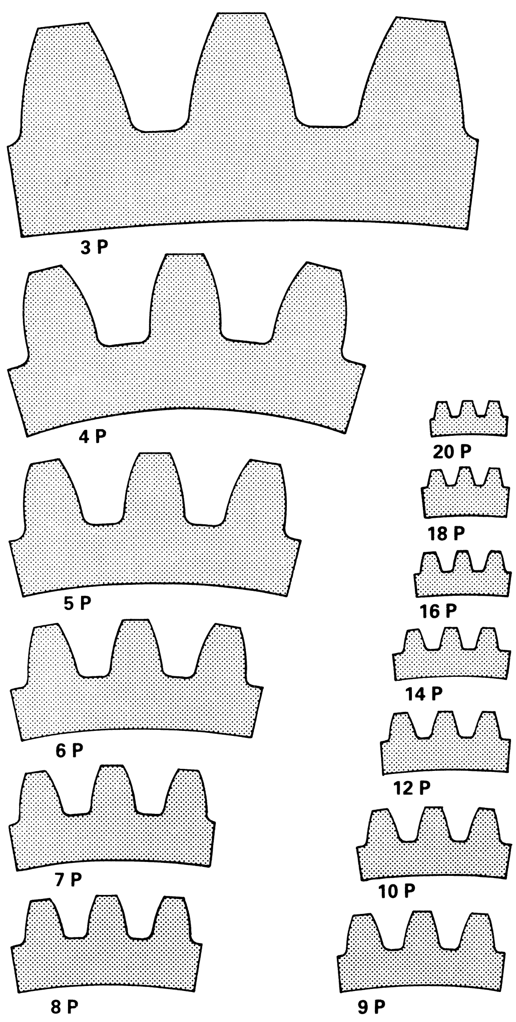


Fig. 25 Gear teeth of various diametral pitch

Vitrified bond wheels are used because of their rigidity and ability to be diamond dressed with a minimum of diamond wear. Wheel rigidity is particularly important for dish-shaped wheels of the large sizes used in generation grinding. Resinoid bond cup wheels are satisfactory for generating bevel gear teeth.

Vitrified bond wheels are used for forming. They are generally crush trued. The vitrified bond is friable and is rigid enough to support the force of crush truing without deflection.

Steel-core wheels plated with CBN are commonly used because their open grain texture allows for slow traverse across the tooth surface and rapid stock removal without burning. Surfaces are generally finished in one pass. Because of the hardness of the material, several thousand parts can be ground before wheel contours need to be replated. More detailed information on grinding wheel specifications and factors in selection can be found in Ref 1 and 2.

Speed. Most gears ground with conventional wheels are ground with wheel surface speeds of 1700 to 2000 m/min (5500 to 6500 sfm). When very small amounts of stock are being removed, speeds higher than 2000 m/min (6500 sfm) can be used. When grinding gears with cup-shaped or dish-shaped wheels, slower speeds (140 m/min, or 450 sfm, or less) are common. Wheel speeds up to 3000 m/min (10,000 sfm) are used with CBN wheels.

Infeed. When using conventional wheels, pitch of the gear being ground is the main factor influencing optimum infeed. For example, in the plunge grinding of case-hardened or hardened-and-tempered steels (50 to 62 HRC), an infeed of 0.038 mm/pass (0.0015 in./pass) is recommended for gears having diametral pitches of 1 to 4. Infeed is reduced to 0.030 mm/pass (0.0012 in./pass) for pitches of 5 to 8 and to 0.013 mm/pass (0.005 in./pass) for pitches finer than 8. With CBN wheels, 0.13 to

0.25 mm (0.005 to 0.010 in.) of stock is removed in one pass.

Correcting for Distortion in Heat Treating

Most steel gears are heat treated. Because some diametral change occurs during heat treatment, gear teeth are cut prior to heat treating and ground to finish dimensions after heat treating. The magnitude of the dimensional change depends on the type of heat treatment and the shape of the part. Distortion is greatest when the entire part is heated and quenched. Induction hardening and flame hardening cause small changes because only the tooth area of the part is heated and quenched.

Usually, nitrided gears do not need to be ground to correct for distortion. When gears are properly nitrided, dimensional change is so small that any correction required can be accomplished by honing or lapping.

When the entire gear is heat treated, dimensional change is less in gears of nearly uniform cross section than in those of intricate design or large variation in cross section. For example, short, stubby pinions are least likely to lose accuracy because of unequal cooling rates during quenching, while thin-web gears with heavy-section teeth or heavy rims change the most.

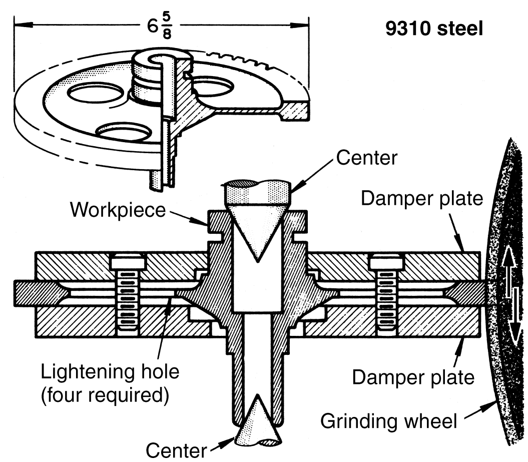
The use of a heat-treating procedure that allows maximum dimensional control, such as placing the heated part in a fixture for quenching, helps reduce the amount of grinding and sometimes has eliminated the need for grinding. However, some dimensional changes occur during heat treatment regardless of special techniques. The following example is typical of a gear that required grinding even though it had been quenched in a die to control distortion.

Example 1: Grinding Die-Quenched Gears. The gear teeth for the part illustrated in Fig. 26 were cut by hobbing, after which they were carburized and quenched in a die to minimize distortion. However, this practice did not prevent distortion beyond tolerance, and lightening holes caused a scalloping of the pitch line by approximately 0.10 mm (0.004 in.) total indicator reading. Therefore, grinding was necessary. No grinding burns or cracks could be accepted.

Conditions of the grinding operation are given in the table with Fig. 26. Damper plates prevented excessive vibration of the gear during grinding, thus reducing the likelihood of both burns and cracks.

Form Grinding

Usually, form grinding consists of passing a formed wheel through a tooth space to grind to root depth the left side of one tooth and the right side of the next tooth at the same time (see modifications in Fig. 27). When straight teeth are ground, as in spur gears, the workpiece is held in a fixed radial position during a pass. For helical gears, however, the workpiece must be



Gear details	
Type	Involute spur
Number of teeth	63
Diametral pitch	12
Pressure angle	20°
Tolerance	
Involute error, mm (in.)	0.005 (0.0002)
Tooth-to-tooth spacing, mm (in.)	Within 0.005 (0.0002)
Spacing between any three adjacent teeth, mm (in.)	Within 0.013 (0.0005)
Finish required, involute sides and root fillet, μm (μin.)	0.80 (32) max
Hardness, HRC	58–63
Grinding wheel	
Classification	A-180-H9-V
Size, mm (in.)	349 (13¾) OD
Form	Diamond dressed
Operating conditions	
Machine	Gear grinding, rack type
Speed, at 1850 rev/min, m/min (sfm)	2030 (6650)
Infeed per pass	
Roughing, mm (in.)	0.030 (0.0012)
Finishing, mm (in.)	0.020 (0.0008)
Axial feed	
Roughing, mm (in.)	0.89 (0.035)
Finishing, mm (in.)	0.41 (0.016)
Grinding fluid	Sulfurized grinding oil
Time per piece, min	12

Fig. 26 Setup for grinding a thin-web gear. Damper plates prevent excessive vibration. Dimensions in figure given in inches

rotated so that the helix will be followed as the wheel passes through the tooth space. A lead bar having the same lead as the helix controls rotation during grinding.

When the wheel pass through a tooth space is completed, the workpiece is indexed to the next position, and the procedure is repeated. The indexing mechanism usually consists of an index plate with the same number of spaces at the workpiece. Control of tooth spacing errors depends primarily on the accuracy of this plate.

Spur gears are ground with a single straight wheel with a periphery that has been formed to produce the space between teeth (Fig. 27a), a single wheel with several ribs that grind more than one tooth space during each wheel pass (Fig. 27b), or two single-ribbed wheels (Fig. 27c). Helical gears are ground with a formed, single straight wheel. Bevel gears are form ground with a cup-shaped wheel. Form grinding can also be used to finish grind precut threads of worms.

Wheels for form grinding are usually less than 152 mm (6 in.) in diameter; wheel thickness depends on the size of the teeth to be ground. Wheels of larger diameter can be used, but the smaller wheels are more common because they need less clearance in finishing a gear that is close to a larger gear or to a shoulder. In addition, smaller wheels are better for grinding internal gears.

Standard gear grinding machines are available for form grinding spur gears ranging from a minimum root diameter of 19 mm (¾ in.) to an outside diameter of 914 mm (36 in.). Special machines have been built to grind spur gears 2.4 m (8 ft) in diameter. Machines are available that are capable of form grinding external helical gears up to 457 mm (18 in.) in diameter.

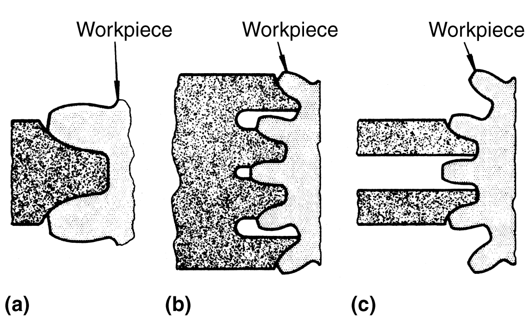


Fig. 27 Relations of wheel and workpiece in the form grinding of spur gears (top views). (a) Single-ribbed grinding wheel. (b) Multiribbed grinding wheel. (c) Two single-ribbed wheels, also known as a straddle wheel

Generation Grinding

There are four methods of generating gear teeth by grinding. Each method is identified by the type of wheel used: straight, cup-shaped, dish-shaped, and rack-tooth worm wheels.

Straight Wheels. In this method, a straight wheel beveled on both sides reciprocates across the periphery of the workpiece as the workpiece rolls under it in a direction perpendicular to its reciprocating motion. The reciprocating action of the wheel is similar to that of a reciprocating gear cutter. This grinding method can be used to generate the teeth of spur gears.

Cup-Shaped Wheels. In this method, the gear tooth is rolled against a cup-shaped wheel, the sides of which are beveled to an angle equal to the pressure angle required on the teeth. During grinding, the wheel moves in a straight line along the length of the tooth. The sides of the wheel simultaneously grind adjacent sides of two teeth. A number of mechanical methods are used to control the generating motion and the indexing of the workpiece. Electronic controls are used on newer machines.

Teeth can be generated by grinding with a cup-shaped wheel on spur and helical gears up to 457 mm (18 in.) in pitch diameter. However, when helical gears are being ground, the generating motion must match the pitch in the plane of rotation of the gear to be ground. With this grinding method, it is impractical to generate tip and root reliefs.

Dish-Shaped Wheels. This method uses two dish-shaped grinding wheels, which may be inclined 15 or 20° or may be parallel in a vertical position (0°), as shown in Table 5.

The generating motion is sometimes controlled by steel bands that are fastened to a pitch block of the same diameter as the pitch circle of the workpiece minus the thickness of the rolling bands. The shape of the tip and root of the teeth can be modified by using a cam to change the normal generating motion.

Usually an index plate similar to that used in form grinding indexes the workpiece if a pitch block is used. Large machines use worm gearing for indexing. Gears are ground dry, and machines incorporate a device depending on a feeler diamond to compensate for wheel wear. Machines are available for grinding spur and helical gears ranging from 25 to 3610 mm (1 to 142 in.) in diameter, with diametral pitch 2 to 17 by this method.

When the wheels are in the 15 or 20° position, as in illustration (a) in Table 5 their active faces rotate in planes forming an acute angle. In this position, the wheels simulate the tooth flanks of a rack that the workpiece engages during the rolling-generating motion (Fig. 28). The inclination of the grinding wheels to the vertical in the pitch plane generally equals the pressure angle of the gear to be ground (usually 15 or 20°). Under these conditions, the generating circle coincides with the pitch circle of the gear. This method produced a criss-cross pattern of grooves (lay of surface roughness) and has been widely used because machine settings are simple and the desired pressure angle is accurately and simply set by adjusting the inclination angles of the wheels.

The 15 or 20° wheel position is not suitable for making longitudinal modifications. Although the two simultaneous points of contact between the grinding wheels and the respective tooth flanks move along the tooth-form involutes during each transverse rolling-generating motion, the locations of these points relative to each other change constantly.

For helical gears, this differential position applies not only along the involutes but also to the position in the longitudinal tooth direction. Therefore, it would be impractical to try to coordinate modification impulses of the grinding wheel and the generating or feed motion to make longitudinal modifications.

In a newer method, the angle between the two wheels is 0°, as shown in illustration (b) in Table 5. The active surfaces of the wheels are parallel and face each other. Because the grinding pressure angle is 0° and corresponds to the pressure angle on the base circle, the generating motion is obtained by rolling on the base circle.

The advantages of grinding gears in the 0° position over the 15 or 20° position are:

- Greater production without sacrifice in quality
- Ability to grind more exacting flank profiles and to make longitudinal modifications

The main disadvantage is the lack of uniform transition between the flank and the tooth root, which is objectionable in some gearing applications.

Rack-Tooth Worm Wheels. The wheel used in this method of generating gear teeth is crush trued into the shape of a conventional rack-tooth worm or a CBN-plated rack-tooth worm. Grinding a gear by this method is illustrated in Fig. 29.

The workpiece is mounted vertically on an arbor or in a fixture that fits between the centers of the machine. For grinding helical gears, the workholding slide of the machine is set at an angle equal to the helix angle required on the gear. The workpiece is rotated by a motor that is synchronized with the wheel motor.

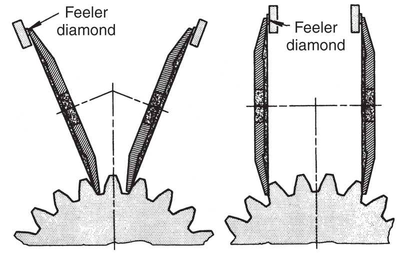
The gear can be ground with a small portion of the length of the grinding worm wheel. Generation takes place as a result of rotating the worm wheel and gear in mesh as the wheel is traversed axially across the gear face.

A soft, conventional aluminum oxide or silicon carbide wheel is used to prevent burning the hardened teeth, but this results in greater wheel breakdown and loss of correct tooth profile. To continue using a fresh, unbroken part of the wheel, the work is fed in a direction parallel to the axis of the worm using the setup shown in Fig. 30.

Table 5 Grinding of gears with dish-shaped wheels in the 0° position

Details for seven different spur and helical gears that had been carburized and hardened.

Parameter	Gear details						
Diametral pitch	10	5	4	3	5	3	2.75
Module	2.5	5	6	8	5	8	9.5
Number of teeth	23	44	80	80	120	100	379
Face width, mm (in.)	55.9 (2.2)	61.0 (2.4)	119 (4.7)	119 (4.7)	300 (11.8)	180 (7.1)	419 (16.5)
Tip diameter, mm (in.)	71.1 (2.8)	234 (9.2)	503 (19.8)	653 (25.7)	630 (24.8)	828 (32.6)	3350 (132)
Pressure angle, degree	15	20	20	20	20	15	15
Helix angle, degree	26	20	10	10	15	0	7
Grinding allowance per flank, mm (in.)	0.15 (0.006)	0.18 (0.007)	0.20 (0.008)	0.23 (0.009)	0.18 (0.007)	0.23 (0.009)	0.25 (0.010)



(a) 15 or 20° position (b) 0° position

Success in gear grinding with a conventional grinding worm wheel depends greatly on careful dressing of the wheel to the required contour and on synchronization of the workpiece rotation with the rotation of the worm wheel. Special diamond wheel dressers are required. The accuracy of the diamond and of the dressing operation determine to a great extent the quality of the gear teeth produced.

Wheels are about 50 mm (2 in.) wide, and they can be rough formed by crushing and can be finished to the proper pressure angle by diamond dressing. Some machines use wheels that have been finish dressed by crushing only. Because a considerable amount of time is required to crush true a new wheel, it is advisable to stock wheels for each pitch and pressure angle required. In diamond dressing, two diamonds are needed to dress both sides of the thread at the same time. Gear tooth design can be easily modified by lapping the tip of the dressing diamond.

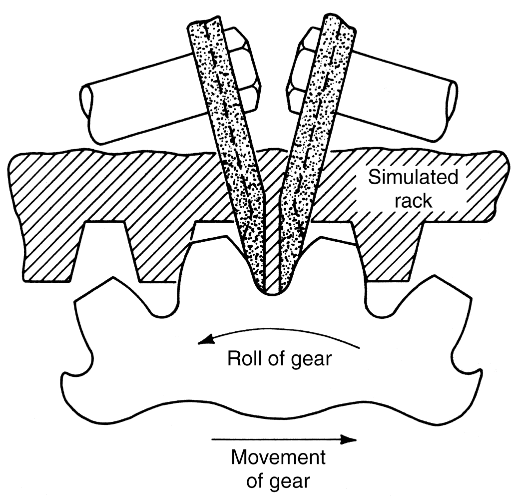


Fig. 28 Schematic of dish-shaped wheels simulating the motion of a rack. The gear being machined rolls and moves. Wheels may or may not be reciprocated.

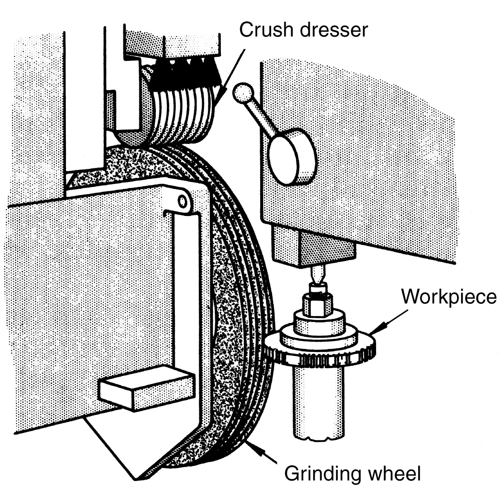


Fig. 29 Generation grinding with a rack-tooth worm wheel

Burning the hardened steel is much less of a problem when using CBN-plated wheels because of the open texture of the wheel surface. The hardness of the CBN material and the stiffness of the steel core allow for several thousand pieces to be ground before the worm needs replating and returning. Modifications of the gear tooth design are manufactured into the core before it is plated.

Spur gears and helical gears up to 508 mm (20 in.) in diameter can be ground in machines using a threaded wheel. Although fine-pitch gears can be ground from a solid gear blank, gear teeth with diametral pitch of 24 or coarser should be cut prior to grinding.

The advantages of gear grinding with a threaded wheel are accuracy, speed, and ease of incorporating changes in tooth design. Machines are also available that crown the gear teeth during grinding. Because mechanical grinders are complicated and sensitive, a considerable amount of operator skill is required. However, with electronic CNC machines, crowning is a standard feature. Gear grinding with a threaded wheel is six to thirty times as fast as the other methods discussed in this section.

Grinding of Bevel Gears

When accuracy is required, grinding is the most economical method for finishing teeth as coarse as 1.5 diametral pitch on straight bevel, spiral bevel, and hypoid gears. With good grinding practice, bevel and hypoid gears can be finished with a tooth-to-tooth spacing accurate

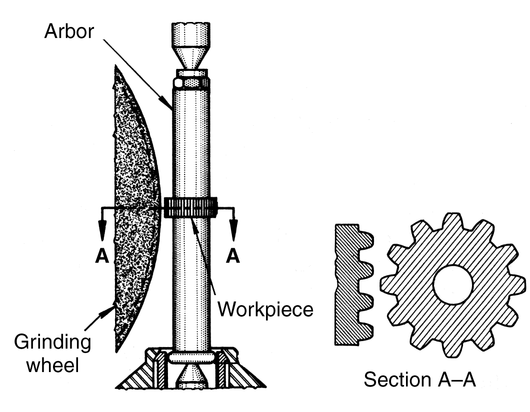


Fig. 30 Setup used to grind an aircraft pinion by the worm wheel method

within 0.005 mm (0.0002 in.), eccentricity with bore or shaft within 0.0064 mm (0.00025 in.), as little backlash as 0.025 to 0.051 mm (0.001 to 0.002 in.), and a surface finish of 0.38 to 0.76 μm (15 to 30 μin.). In general, any bevel or hypoid gear that is cut with a circular cutter can be ground with a wheel of similar shape and grinding action, except nongenerated gears that are cut with a face mill.

Nongenerated (Formed) Gears. The gear teeth are ground by two methods. The first method involves a cup-shaped wheel that simultaneously grinds the facing sides and root of adjacent teeth. There is no generating motion, and the wheel produces a curved tooth whose sides have a straight profile. Three or four revolutions of the workpiece are required for the complete finish grinding of a bevel gear using conventional dressable wheels. A predetermined amount of stock is removed with each revolution of the workpiece. With CBN-plated wheels, the entire finishing stock is removed in one revolution of the workpiece. The grinding wheel makes line contact with the tooth being ground. This method is applicable for finishing hypoid and spiral gears and for special spiral gears that have a 0° mean spiral angle.

The second method is called the Waguri method. The grinding wheel is circular and has an average point width of 0.25 to 0.50 mm (0.010 to 0.020 in.) less than the slot width of the workpiece. The wheel axis follows a circular path with a diameter equal to the difference between its point width and its slot width. Thus, the desired surfaces are ground with line contact (Fig. 31). Conventional dressable wheels can be used with several revolutions of the workpiece, or CBN-plated wheels with one revolution of the workpiece.

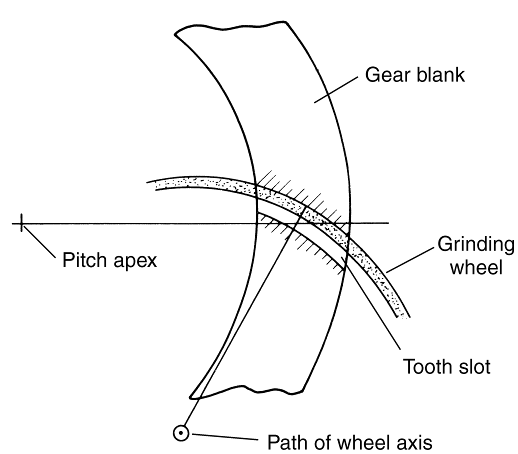


Fig. 31 Waguri grinding method for nongenerated spiral bevel and hypoid gears

An A-46-J-V wheel is the conventional wheel commonly used to grind gears ranging from 2 to 10 diametral pitch. Wheel speed ranges from 1200 to 1400 m/min (3800 to 4500 sfm). The wheel is dressed automatically at predetermined intervals between grinds.

Cubic boron nitride-plated wheels cover the same range of diametral pitch. Wheel speeds range from 1400 to 2000 m/min (4500 to 6500 sfm).

Generated Gears. Spiral bevel and hypoid gears are ground with a cup-shaped wheel. The conventional grinding wheel is dressed to produce a smooth blend of flank and root profile. The core of the CBN wheel is machined to produce a smooth blend of the flank and tooth profile. The curved profile of the tooth is generated by relative rolling motion between the grinding wheel and the workpiece.

Vitrified bond and resinoid bond aluminum oxide cup-shaped wheels are the conventional wheels used for spiral bevel and hypoid gears. Typical wheel classifications are A-60-J-V and A-60-M-B. Wheel speed ranges from 1200 to 1400 m/min (3800 to 4500 sfm). Cubic boron nitride-plated wheels are also used.

Generating gear teeth with a cup-shaped wheel produces spiral bevel gears with highly accurate tooth spacing, concentricity, and profile shape. The following is an example of a typical application of this grinding method.

Example 2: Grinding Spiral Bevel Gears With a Cup-Shaped Wheel. The spiral bevel gear illustrated in Fig. 32 was carburized, hardened, and tempered before being ground by the generation method with a cup-shaped wheel. An automatic wet-type 3.7 kW (5 hp) machine designed for grinding spiral bevel, Zerol, and hypoid gears was used for this application. Additional manufacturing information is given in the table with Fig. 32.

Grinding Fluids

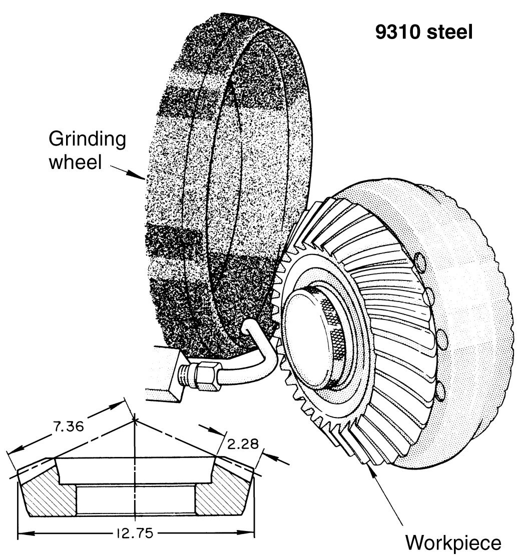
The use of a grinding fluid is generally recommended for the grinding of steel gears. A grinding fluid prolongs wheel life between dressings, flushes away chips, and improves gear tooth finish. Flooding the work surface minimizes the possibility of burning the surfaces of gear teeth. The complete absence of grinding burn is essential for high-quality gears. Virtually all the grinding fluids mentioned in the section "Cutting Fluids" in this chapter have been used successfully for the grinding of gears.

In many critical gear-grinding applications, mineral-base sulfochlorinated or sulfurized oils are used. Plain soluble-oil emulsions have proved satisfactory for many gear grinding applications. Chemical solutions have also been satisfactory.

Surface Finish

Grinding produces a finish unlike that produced by any other process. Whether this finish is more desirable than some other finish depends on the service requirements of the gear. In some applications, a gear with a ground finish operates more quietly, even though grinding has not improved the dimensional accuracy of the product. The effect of a ground finish, in comparison with other finishes, on the lubrication of gears has not been precisely evaluated.

The surface finish of ground gears usually ranges from 0.38 to 0.80 μm (15 to 32 μin.). Under conditions of unusually good control, a surface finish of 0.25 μm (10 μin.) or better can be produced. Surface finish sometimes becomes the major consideration in determining whether a gear will be ground. Grinding can have the dual purpose of correcting dimensional change from heat treating and of producing a fine surface finish.



Gear details	
Type	Spiral bevel (octoid)
Number of teeth	51
Diametral pitch	4
Helix angle	30°
Whole depth, mm (in.)	12.07 (0.475)
Tolerance, tooth-to-tooth, mm (in.)	0.008 (0.0003)
Surface finish, μm (μin.)	0.63–0.76 (25–30)
Hardness, HRC	58–60
Grinding wheel	
Type	Cup(a)
Classification	A-54-J8-V
Size, mm (in.)	305 (12) OD
Operating conditions	
Wheel speed, at 1200 rev/min, m/min (sfm)	1150 (3770)
Feed	3.2 s/tooth(b)
Grinding fluid	Straight mineral oil (no active sulfur)
Production rate, pieces/h	1.7(c)

(a) Straight-angled grinding area. (b) 12 passes per tooth. (c) At 100% efficiency

Fig. 32 Grinding of a spiral bevel gear. Dimensions in figure given in inches

Honing of Gears

Honing is a low-velocity abrading process that uses bonded abrasive sticks to remove stock from metallic and nonmetallic surfaces. As one of the last operations performed on the surface of a part, honing generates functional characteristics specified for a surface, such as geometric accuracy, dimensional accuracy, and surface features (roughness, lay pattern, and integrity). It also reduces or corrects geometric errors resulting from previous operations.

The most common application of honing is on internal cylindrical surfaces (Fig. 33a). However, honing is also used to generate functional characteristics on external cylindrical surfaces, flat surfaces, truncated spherical surfaces, and toroidal surfaces (both internal and external). A characteristic common to all these shapes is that they can be generated by a simple combination of motions.

Gear-tooth honing is designed to improve geometric accuracy and surface conditions of a hardened gear. The teeth of hardened gears are honed to remove nicks and burrs, to improve finish, and to make minor corrections in tooth shape. Gear teeth are honed on high-speed machines specially designed for the process (Fig. 34). The honing tool is like a gear driving the workpiece at high speed (up to 30 m/min, or 100 sfm) while oscillating so that the teeth slide axially against the workpiece.

Spur gears and internal or external helical gears ranging in diametral pitch from 24 to 2.5, in outside diameter from 19 to 673 mm (¾ to

26½ in.), and up to 75 mm (3 in.) in face width have been honed on these machines. Finishes of 0.75 μm (30 μin.) are easily achieved, and finishes of 0.075 to 0.10 μm (3 to 4 μin.) are possible. Both taper and crown honing can be done.

Tools used in honing gear teeth are of two types, a helical gear shape tool made of abrasive impregnated plastic, and a metal helical gear with a bonded abrasive coating that is renewable. The plastic tool, which is discarded at the end of its useful life, is widely used. The metal tool is used mainly for applications in which plastic tools would be likely to break; also, it is used primarily for fine-pitch gears.

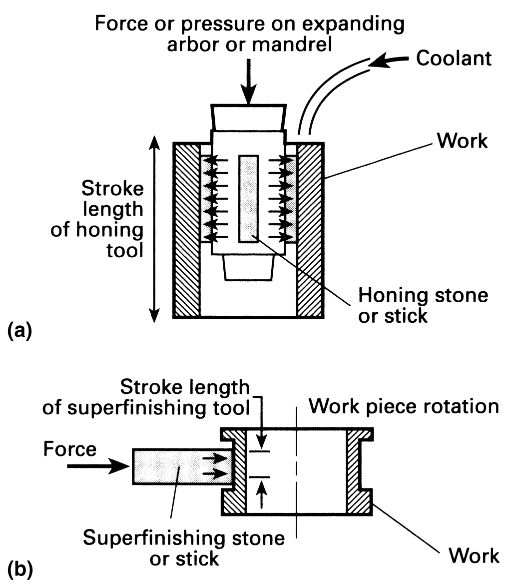


Fig. 33 (a) Schematic representation of honing. (b) Schematic representation of superfinishing

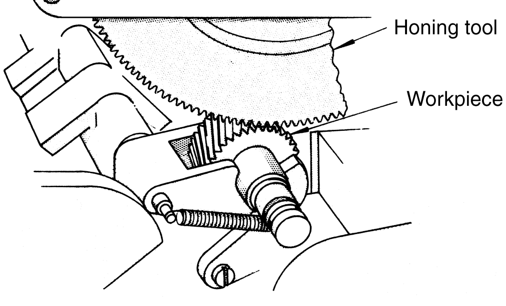


Fig. 34 Honing of helical gear teeth

Plastic tools are supplied with abrasives of 60-grit to 500-grit size. Size of abrasive, gear pitch, and desired finish are usually related as:

		Finish	
Grift size	Gear pitch	μm	μin.
60	≤16	0.75–0.89	30–35
100	16–20	0.63–0.75	25–30
180	>20	0.38–0.50	15–20
280	>20	0.25–0.30	10–12
500	>20	0.075–0.10	3–4

Honing tools do not load up, and a plastic honing tool can wear until its teeth break. Stock removal of 0.025 to 0.050 mm (0.001 to 0.002 in.) measured over pins is the recommended maximum.

Methods. The two methods used to hone gear teeth are the zero-backlash method and the constant-pressure method. In the zero-backlash method, which is used for gears made to commercial tolerances, the tool head is locked so that the distance between the center of the work gear and the center of the honing tool is fixed throughout the honing cycle. In the constant-pressure method, which is used for gears produced to dimensions outside commercial tolerance ranges, the tool and the work gear are kept in pressure-controlled tight mesh.

Applicability. The use of honing for removing nicks and burrs from hardened gears can result in a considerable cost saving in comparison to the usual method. In the usual method, the gears are tested against master specimens on sound test machines. Nicks indicated are searched for and removed using a hand grinder. The gear is then retested to make certain the nick has been removed. When honing is used, all of these various tests and procedures can be eliminated.

Some shape correction can be achieved in the removal of 0.050 mm (0.002 in.) of stock by honing. A helical gear 127 mm (5 in.) in diameter may show lead correction of 0.010 mm (0.0004 in.), involute profile correction of 0.0075 mm (0.0003 in.), and eccentricity correction of 0.010 mm (0.0004 in.).

The advisability of using honing for salvaging hardened gears hinges on cost considerations. As the error in tooth shape increases, honing time increases and tool life decreases. On the other hand, if the gears represent a large investment in production time and material, honing may be the most economical method.

Because honing is not designed for heavy stock removal or tooth correction, it cannot be substituted for grinding or shaving of gears. Rotary shaving usually leaves gear teeth smooth within 0.25 to 1.00 μm (10 to 40 μin.).

Microhoning is a low-velocity abrading process very similar to honing. However, unlike conventional honing, microhoning focuses primarily on the improvement of surface finish and much less on correction of geometric errors (Fig. 33b). As a result, the pressures and amplitude of oscillation applied during microhoning are extremely small. This process is also referred to as *microsurfacing, microstoning,* and *superfinishing.*

Lapping of Gears

Lapping is the process of finishing work materials by applying a loose abrasive slurry between a work material and a closely fitting surface, called a *lapping plate.* When loose abrasive is used to machine the work material, it may slide, roll, become embedded, or do all three, depending on the shape of the abrasive grain and the composition of the backup surface.

Gear lapping corrects the minute errors in involute profile, helix angle, tooth spacing, and concentricity created in the forming or cutting or in the heat treatment of the gears. The lapping can be done by running a set of gears in mesh or by running one gear with a gear-shaped master lapping tool.

Gear lapping is most often applied to sets of hardened gears that are required to run silently in service. Gear lapping is strictly a mating process and is not intended for stock removal. Two gears that have been matched by lapping should be operated as a set, and they should be replaced as a set, rather than singly.

Gears are lapped in special machines, which can be arranged for manual, semiautomatic, or automatic operation. In semiautomatic operation, loading and unloading are manual; in automatic operation, loading and unloading are done automatically in accordance with a programmed cycle.

Angular, spur, and helical gears can be lapped, but the process is mainly applied to spiral bevel gears and hypoid gears. A typical setup used for lapping hypoid gears is shown in Fig. 35. Gears of up to 915 mm (36 in.) pitch diameter can be lapped in semiautomatic or automatic operation. Manual lapping, in special equipment, is employed for gears of approximately 2540 mm (100 in.) pitch diameter down to the smallest gear that can be manufactured. Production lapping machines can be adjusted to lap gears with shaft angles of 0 to 180°.

Lapping Media. Optimum grit size varies with different types and sizes of gears. A 280-grit abrasive is used for spiral bevel gears; a finer abrasive (about 400-grit) is more suitable for hypoid gears because the sliding action is greater. As a rule, coarser grit is used for gears having a coarse pitch, and finer grits are used for gears having a fine pitch. When compound is brushed on, as in manual operation, a paste-type vehicle is used. However, in semiautomatic or automatic lapping, the abrasive should be mixed with a thin oil (such as mineral seal oil) so that it can be pumped to the workpieces (Fig. 35).

Processing Techniques. It is important to roll all mating gears together before lapping to detect nicks and burrs, which can be removed by a small portable hand grinder before the gears are lapped. This preliminary rolling also inspects tooth contact, which should be in the same location for each set of gears and is especially important in automatic lapping.

During the lapping operation, the pinion (smaller gear) is used as the driver, and the larger gear is the driven member. The driven spindle is also used for applying the necessary tooth-contact load by adjusting to a slight drag.

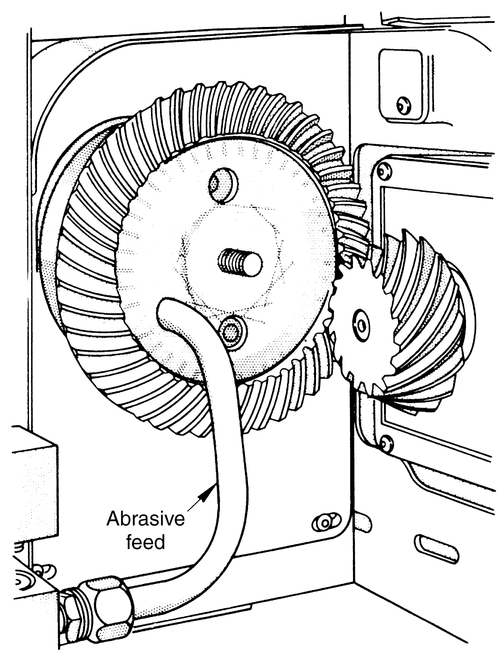


Fig. 35 Setup for the lapping of hypoid gears

Running cycles as short as 15 s at about 76 m/min (250 sfm) can frequently produce desired results. However, longer time cycles may be necessary, depending on initial gear-tooth finish and service requirements.

Low noise level is the criterion of successful gear lapping. Because gear lapping is strictly a mating process and no stock removal is intended, measurements are not made as in most other lapping processes. Minor corrections in tooth bearing shape and position can be obtained, however. Lapping does improve the finish of gear teeth, but improved finish is seldom the purpose of gear lapping.

Superfinishing

Superfinishing processes are those that can achieve a surface roughness average (R_a) of ≤0.10 μm (4 μin.) (Ref 3). As shown in Fig. 36, superfinishing can be achieved under carefully controlled conditions by grinding, honing/microhoning, lapping, polishing /electropolishing, or combinations of finishing processes (for example, grinding, honing, and lapping). In the past, such processing was rarely carried out on gears because of the time and expense involved. Recently, however, there have been a number of studies that show that the reduction of surface roughness improves the lubricating condition of

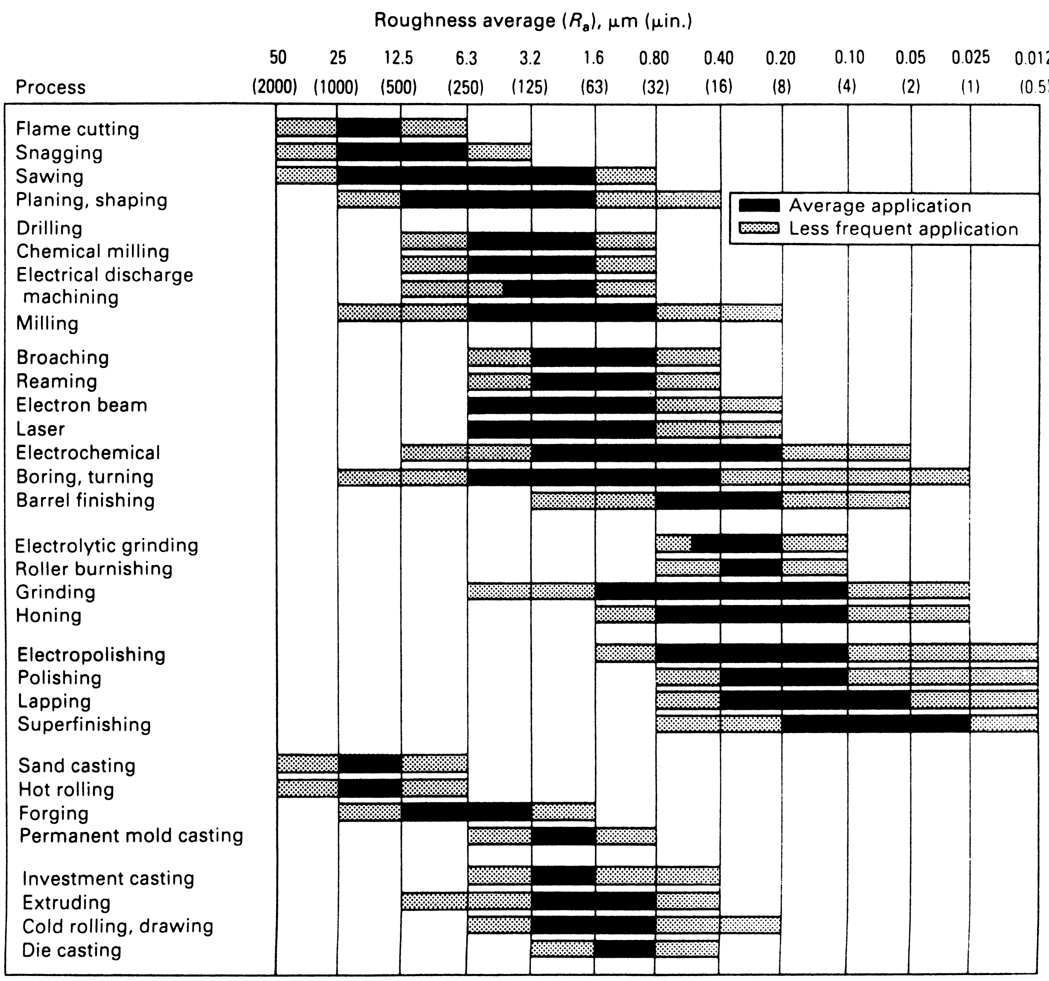


Fig. 36 Surface roughness values produced by common manufacturing/production methods. The ranges shown are typical of the processes listed. Higher or lower values can be obtained under special conditions. Source: Ref 3

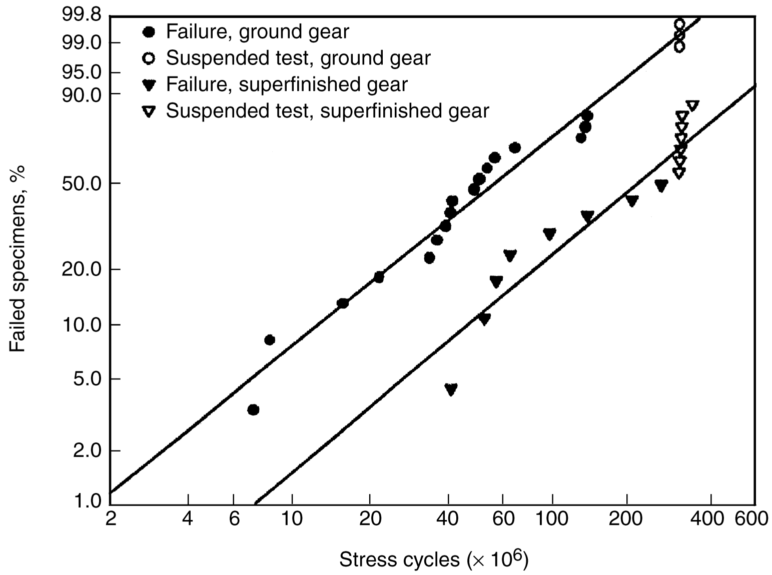


Fig. 37 Weibull plot of surface fatigue test results for both carburized and ground and carburized, ground, and superfinished gears. Source: Ref 4

gears and offers the possibility of increasing their surface fatigue lives (Ref 4).

In order to quantify the effect of surface roughness on fatigue life, a series of tests were carried out on two sets of 28-tooth, 8-pitch gears made from consumable-electrode vacuum-melted (aerospace quality) AISI 9310 steel (Ref 4). Both sets of gears were case carburized and ground. One set of gears was superfinished by placing them in a vibrating bath consisting of water, detergent, abrasive powder, and small pieces of zinc for several hours (details of the superfinishing treatment can be found in Ref 5). Upon removal from the bath the gears had highly polished, mirrorlike surfaces. The superfinishing treatment removed about 2 to 3 μm (79 to 118 μin.) of material from the tooth surfaces. The effects of the superfinishing on the quality of the gear tooth surfaces were determined from metrology, profilometry, and interferometric microscope inspections. The superfinishing reduced the roughness average by about a factor of 5. The mean R_a value before superfinishing was 0.380 μm (15.0 μin.). After superfinishing the mean R_a value was lowered to 0.070 μm (2.8 μin.). The gears were then fatigue tested at a Hertzian contact stress of 1.71 GPa (248 ksi) for 300 million cycles or until surface failure occurred on any one tooth. As shown in Fig. 37, the lives of the superfinished gears were about four times longer than those of the conventionally ground gears.

ACKNOWLEDGMENTS

This chapter was adapted from:

- T.J. Krenzer and J.W. Coniglio, Gear Manufacture, *Machining,* Vol 16, *Metals Handbook,* ASM International, 1989, p 330–355
- Honing, *Machining,* Vol 16, *Metals Handbook,* ASM International, 1989, p 472–491
- P. Lynah, Lapping, *Machining,* Vol 16, *Metals Handbook,* ASM International, 1989, p 492–505

REFERENCES

1. W.N. Ault, Grinding Equipment and Processes, *Machining,* Vol 16, *Metals Handbook,* ASM International, 1989, p 430–452
2. K. Subramanian, Superabrasives, *Machining,* Vol 16, *Metals Handbook,* ASM International, 1989, p 453–471
3. M. Field, J.F. Kahles, and W.P. Koster, Surface Finish and Surface Integrity, *Machining,* Vol 16, *Metals Handbook,* 9th ed., ASM International, 1989, p 19–36

4. T.L. Krantz, M.P. Alanou, H.P. Evans, and R.W. Snidle, Surface Fatigue Lives of Case-Carburized Gears With an Improved Surface Finish, *J. Tribology* (Trans. ASME), October 2001, Vol 123, p 709–716
5. R.W. Snidle, H.P. Evans, M.P. Alanou, The Effect of Superfinishing on Gear Tooth Profile, Report AD-327916, June 1997, available from the Defense Technical Information Center (DTIC), the National Technical Information Service (NTIS), or the NASA Scientific and Technical Information Center (http://www.grc.nasa.gov)

SELECTED REFERENCES

- *Dudley's Gear Handbook: The Design, Manufacture, and Application of Gears,* 2nd ed., D.P. Townsend, Ed., McGraw-Hill Inc., 1991
- D.W. Dudley, *Handbook of Practical Gear Design,* McGraw-Hill Book Company, 1984

Gear Materials, Properties, and Manufacture
J.R. Davis, editor, p129-137
DOI:10.1361/gmpm2005p129

CHAPTER 6

Casting, Forming, and Forging

GEAR MANUFACTURE depends on machinery available, design specifications or requirements, cost of production, and type of material from which the gear is to be made. There are many methods for manufacturing gears including:

- Metal removal processes (hobbing, shaping, milling, shaving, grinding, honing, and lapping) as discussed in Chapter 5
- Various casting processes for both production of gear blanks and near-net shape gears
- Stamping and fine blanking
- Cold drawing and extrusion
- Powder metallurgy (P/M) processing as discussed in Chapter 7
- Injection molding (applicable to plastic gear materials as discussed in Chapter 4)
- Gear rolling
- Forging for production of gear blanks and precision-forged near- and net-shape gears

Most of the processes listed are suited to gears for low wear requirements, low power transmission, and relatively low accuracy of transmitted motion (Ref 1). When the application involves higher values of one or more of these characteristics, forged or cut/machined gears are used. Table 1 lists the tolerances in terms of American Gear Manufacturers Association (AGMA) quality numbers for various gear manufacturing processes (Ref 2).

Casting

Although the casting process is used most often to make blanks for gears which will have cut teeth (Fig. 1 and 2), there are several variations of the casting process that are used to make toothed gears with little or no machining. For example, cast tooth internal gears (Fig. 3) are produced in several sizes up to 1633 kg (3600 lb). They are heat treated to strength levels of 689 MPa (100 ksi) and machining is not required on these gears. In circumstances where machining is necessary, the machining expense is reduced by casting close to final shape.

Another example of a cast tooth gear is the pinion gear produced from cast high-manganese (Hadfield) steel for an electric mining shovel shown in Fig. 4. It was not necessary to machine the gear teeth.

Specific Casting Processes. Most casting processes have been used to produce gear blanks or cast tooth gears including sand casting, shell molding, permanent mold casting, centrifugal casting, investment casting, and die

Table 1 Recommended tolerances in terms of AGMA quality numbers for various gear manufacturing processes

	Highest quality number	
Process	Normal	With extra care
Sand mold casting	1	3
Plaster mold casting	3	5
Permanent mold casting	3	5
Investment casting	4	6
Die casting	5	8
Injection and compression molding	4	8
Powder metallurgy	6	9
Stamping	6	9
Extrusion	4	6
Cold drawing	6	9
Milling	6	9
Hobbing	8	13
Shaping	8	13
Broaching	7	12
Grinding	10	15
Shaving	9	14
Honing	10	15
Lapping	10	15

Source: Ref 2

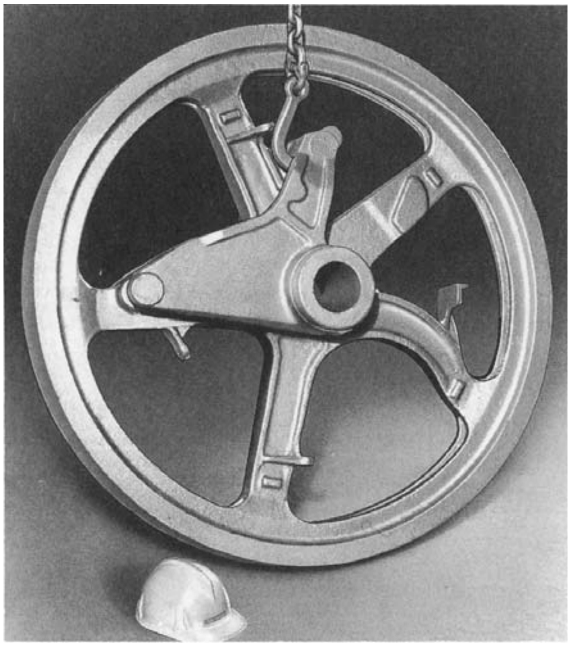


Fig. 1 Cast steel gear blank. Weight: 478 kg (1053 lb). Source: Ref 3

casting. Cut gears have also been produced from continuously cast bars. Some of the commonly employed processes will be briefly reviewed below. More detailed information on these processes can be found in *Casting,* Volume 15 of the *ASM Handbook.*

Sand casting is used primarily to produce gear blanks. In recent times there has been only very limited use of gears with teeth made by sand casting (Ref 4). In some instances gears for farm machinery, stokers, and some hand-operated devices have used cast teeth. The draft on the pattern and the distortion on cooling make it difficult to obtain much accuracy in cast iron or cast steel gear teeth. Table 1 shows that sand cast gears have the lowest AGMA quality levels of the major gear manufacturing methods.

The shell molding process is particularly suited to castings for which:

- The greater dimensional accuracy attainable with shell molding (as compared with conventional green sand molding) can reduce

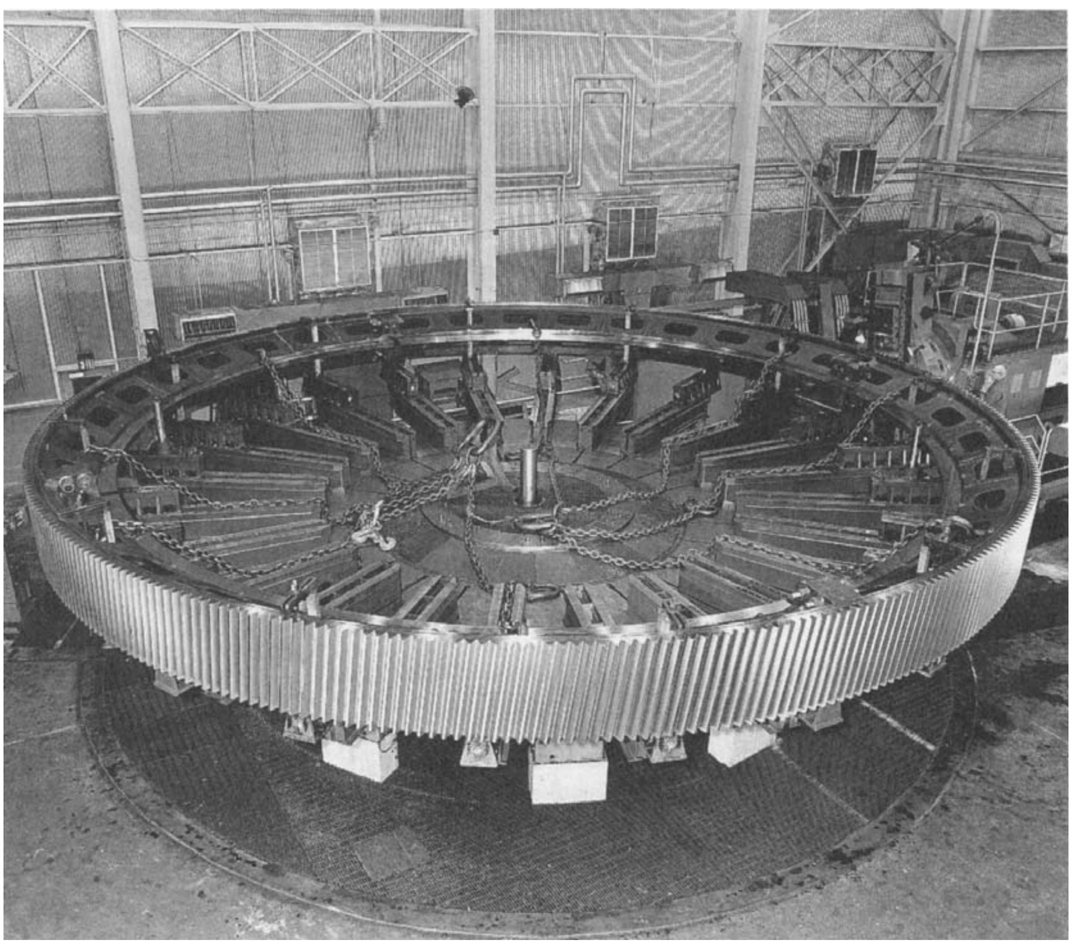


Fig. 2 Large machined cast steel gear. Source: Ref 3

the amount of machining required for completion of the part
- As-cast dimensions may not be critical, but smooth surfaces (smoother than can be obtained by sand casting) are the primary objective. An example of a cast tooth bevel gear with an excellent surface finish produced by shell molding is shown in Fig. 5.

The investment casting process has also had limited use in gear manufacture. Its most apparent value lies in the making of accurate gear teeth out of materials which are so hard that teeth cannot be readily produced by machining (Ref 4). The process can be used with a variety of steels, bronzes, and aluminum alloys. With machinable materials, the process is still useful if the gear is integral with some complicated shape that is very difficult to produce by machining.

Large quantities of small, low-cost gears are made by the cold chamber die casting process (die cast gears are usually under 150 mm (6 in.) in diameter and from 10 to 48 diametral pitch, DP). Complicated gear shapes which would be quite costly to machine can be made quickly and at low cost by the die casting process. The main disadvantage of the process is that the low-melting point metals suitable for die casting—aluminum, zinc, and copper—do not have high enough hardness for high load-carrying capacity.

Many different types of gears can be die cast, such as spur, helical worm, clusters, and bevel gears. Applications for these types of gears include toys, washing machines, small appliances, hand tools, cameras, business machines, and similar equipment.

Forming

Stamping and Fine Blanking. Stamping is a metalworking technique that has been compared to using a cookie cutter. In this process, a sheet of metal is placed between the top and bottom portions of a die; the upper die is pressed into the lower section and "removes" or cuts the gear from the sheet. This is a low-cost, very efficient method for producing lightweight gears for no-load to medium-duty applications. Stamping is restricted by the thickness of the

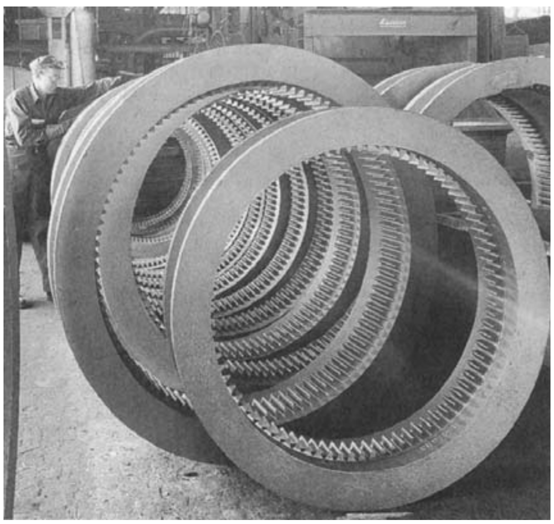


Fig. 3 Cast tooth internal gears. Weights up to 1633 kg (3600 lb). Source: Ref 3

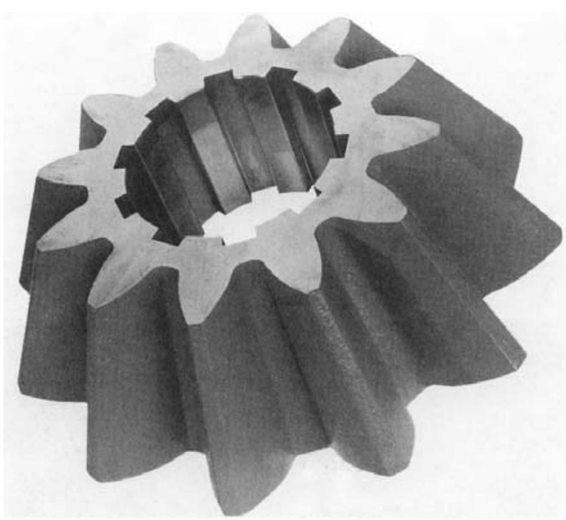


Fig. 4 Cast tooth pinion gear for an electric mining shovel. Weight: 212 kg (468 lb). Source: Ref 3

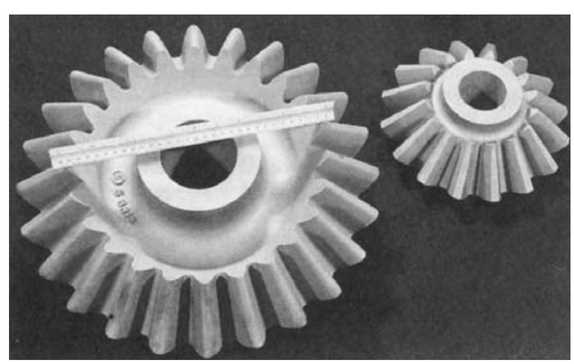


Fig. 5 Cast tooth bevel gear produced by the shell molding process to obtain excellent surfaces and close tolerances. Source: Ref 3

workpiece and is used primarily for spur gears and other thin, flat forms (Ref 5). Stamped gears range in size from 20 through 120 DP and 0.25 to 3 mm (0.010 to 0.125 in.) thick (Ref 6). As the pitch becomes finer, the materials specification must become thinner. Table 2 shows recommended stock thicknesses for various pitches that are commonly used and require no special care in die maintenance. As shown in Table 1, tolerances for stamped gears are good and AGMA quality class 9 can be achieved with extra care.

A wide range of materials can be processed by stamping, including all the low and medium carbon steels, brasses, and some aluminum alloys. Nonmetallic materials can also be stamped. Gears manufactured by this process are used in toys, clock and timer mechanisms, watches, small appliances such as mixers, blenders, toasters, and can openers, as well as larger appliances such as washers and dryers.

Fine blanking (also known as fine-edge blanking) is actually more akin to cold extrusion than to a cutting operation such as stamping. The process takes metal from a sheet like stamping but differs from it in that it uses two dies and forms the workpiece by pressing it into the desired shape. The metal is extruded into the die cavities to form the desired shape. Also unlike stamping, fine blanking offers the designer a limited three-dimensional capability and can thus be used to create bevels, multiple gear sets, and other complex forms (Ref 5). Fine blanked gears can be found in a wide range of applications including the automotive, appliance, office equipment, hydraulic, and medical equipment industries.

Cold Drawing and Extrusion (Ref 6). This process requires the least tool expenditure for mass production of spur gear toothed gear elements and is extremely versatile in that almost any tooth form that can be desired can be produced. As the name implies, a bar is pulled (drawn) or pushed (extruded) through a series of several dies, the last having the final shape of the desired tooth form. As the material is run through these dies it is actually squeezed into the shape of the die. Since the material is displaced by pressure, the outside surface is work hardened and quite smooth.

The bars that are "blanks" for this process are usually 3 to 3.7 m (10 to 12 ft) in length. After passing through the dies, they are known as pinion rods, and often are put into screw machines that finish the individual gears. Experience has shown that it is more economical to slice a segment off an extruded bar than to cut an individual gear. In some cases it would be impossible to produce the desired shape of pinion in any other way. Pinion rods from 16 to 100 DP can be obtained, but as the pitch becomes finer, it becomes more difficult to obtain the close tolerances that are sometimes desired on fine-pitch pinions. Any material that has good drawing properties, such as high-carbon steels, brass, bronze, aluminum, and stainless steel, may be used for the drawn pinion rod.

Gears and pinions manufactured by this process have a large variety of applications and have been used on watches, electric clocks, spring-wound clocks, typewriters, carburetors, magnetos, small motors, switch apparatus, taximeters, cameras, slot machines, all types of mechanical toys, and many other parts for machinery of all kinds.

Gear Rolling (Ref 1). Spur and helical gears, like splines, are roll formed. Millions of high-quality gears are produced annually by this process; many of the gears in automobile transmissions are made this way. As indicated in Fig. 6, the process is basically the same as that by which screw threads are roll formed, except that in most cases the teeth cannot be formed in a single rotation of the forming rolls; the rolls are gradually fed inward during several revolutions.

Because of the metal flow that occurs, the top lands of roll-formed teeth are not smooth and perfect in shape; a depressed line between two slight protrusions can often be seen. However, because the top land plays no part in gear tooth action, if there is sufficient clearance in the mating gear, this causes no difficulty. Where desired, a light turning cut is used to provide a smooth top land and correct addendum diameter.

Rolling produces gears 50 times as fast as gear cutting and with surfaces as smooth as 0.10 μm (4 μin.). Not only does rolling usually need

Table 2 Recommended stock thicknesses for stamped gears

	Thickness	
Pitch range	mm	in.
20–36	0.51–1.98	0.020–0.078
38–60	0.38–1.57	0.015–0.062
62–72	0.25–1.02	0.010–0.040
74–90	0.25–0.89	0.010–0.035
92–120	0.25–0.64	0.010–0.025

Source: Ref 6

no finish operation, but rolling refines the microstructure of the workpiece.

Production setup usually takes only a set of rolling dies and the proper fixture to equip the rolling machine. By either the infeed (plunge) method or the throughfeed method, the rolling dies drive the workpiece between them, forming the teeth by pressure.

Limits. Spur gears can be rolled if they have 18 teeth or more. Fewer teeth cause the work to roll poorly. Helical gears can be rolled with fewer teeth if the helix angle is great enough.

It is usually impractical to roll teeth with pressure angle less than 20°. Lower angles have wide flats at root and crest that need more pressure in rolling. Lower angles also hinder metal flow. Although 0.13 mm (0.005 in.) radius fillets can be rolled, 0.25 mm (0.010 in.) is a better minimum. For greater accuracy, gear blanks are ground before rolling. Chamfers should be 30° or less.

Steels for gear rolling should not have more than 0.13% S and preferably no lead. Blanks should not be harder than 28 HRC.

Forging

Forging has long been used in the manufacture of gears. This is particularly true for the production of gear blanks which would subsequently be cut/machined into the final desired configuration. Gear blanks have been produced by open-die forging, closed-die forging (Fig. 7), and hot upset forging. During the past thirty-five years there has been considerable research and development aimed at producing near-net or net-shape gears by precision forging. Today precision forged gears requiring little or no finish machining are commonly used in the automotive, truck, off-highway, aerospace, railroad, agriculture, and material handling industries, as well as the energy and mining fields.

High-Energy Rate Forging

One of the first forging processes for manufacturing near- or net-shape gears was the high-energy rate forging process which is a closed-die hot or cold forging process in which the work metal is deformed at unusually high velocities. Ideally, the final configuration of the forging is developed in one blow, or at most, a few blows. Velocity of the ram, rather than its mass, generates the major forging force.

It is possible to produce gears with a contoured grain flow that follows the configuration of the teeth using high-energy-rate forging. In the case of spur gears, this is achieved by pancaking to cause lateral flow of the metal in a die containing the desired tooth configuration at its periphery. Contoured grain increases the load-bearing capacity without increasing the tooth size. In addition, the process minimizes the machining required to produce the finished gear. Although spur gears are the easiest to forge, helical and spiral-bevel gears can also be

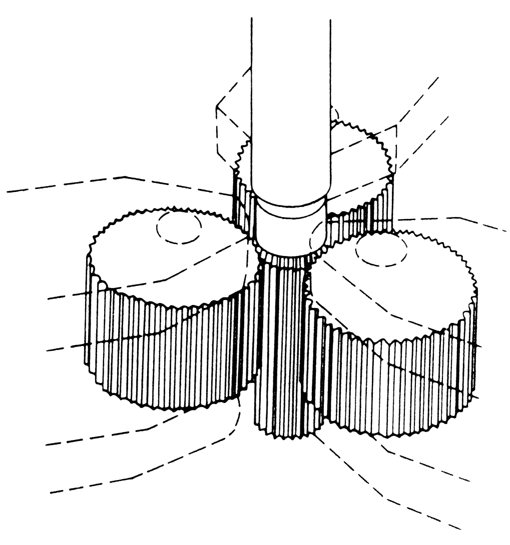


Fig. 6 Method for forming gear teeth and splines by cold forming. Source: Ref 1

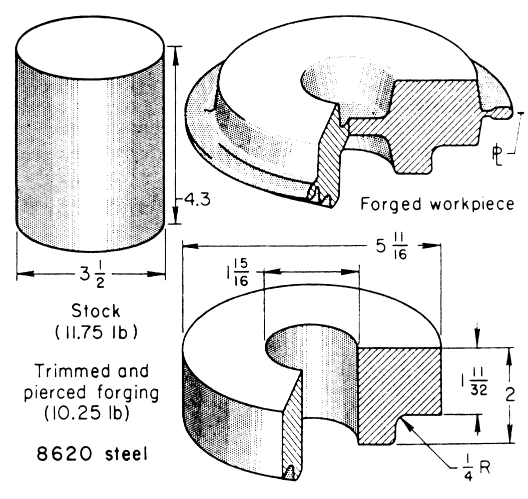


Fig. 7 Gear blank that was closed-die forged in four hammer blows from pancaked stock (not shown), and then trimmed and pierced in one press stroke. Dimensions are given in inches.

forged if their configurations permit ejection of the gear from the die cavity. Gears have been forged from low-alloy steel, brass, aluminum alloys, stainless steel, titanium, and some of the heat-resistant alloys.

Gears with a DP of 5 to 20 are commonly forged with little or no machining allowance. The die life decreases significantly when forging finer-pitch gears.

The forging of 5 DP gears with an involute tolerance of 0.013 mm (0.0005 in.) and total composite error of 0.08 mm (0.003 in.) has been reported. These gears were forged with a tooth-to-tooth spacing deviation of about 0.025 mm (0.001 in.) and a total accumulated deviation of 0.089 mm (0.0035 in.). Over-the-pins dimensions were held to ±0.05 mm (0.002 in.) on these gears, and the total composite error was about 0.20 mm (0.008 in.).

Holding gear dimensions to extremely close tolerances may eliminate finish machining, but the savings may be exceeded by higher die making/maintenance costs. Consequently, most forged gears have an allowance for machining.

A surface finish of 0.5 to 1.5 μm (20 to 60 μin.) on gear teeth is practical. However, even with a 0.5 μm (20 μin.) finish, local imperfections can increase the average to 1.5 μm (60 μin.) or greater. Therefore, it would be difficult to maintain a good surface finish on gear teeth without grinding.

Typical Gear Forgings. The 4.5 kg (10 lb) gear shown in Fig. 8 was forged from 8620 steel billet 75 mm (3 in.) in diameter by 124 mm (4.9 in.) in length. An energy level of 353,000 J (260,000 ft · lbf) was needed to forge the gear in one blow at 1230 °C (2250 °F). The web on the gear was forged to final thickness; the teeth were forged with 0.51 mm (0.020 in.) of stock for finish machining.

The die inserts originally used to forge this gear were made of H11 or H13 tool steel. This steel typically softened after producing 20 gears because of its temperature rising above the 565 °C (1050 °F) tempering temperature of H13 steel. The use of Alloy 718 (UNS N07718) was found to improve the die insert life.

The automotive flywheel shown in Fig. 9, 272.49 mm (10.728 in.) in diameter over the teeth and weighing 11 kg (24 lb), was forged from a machined blank cast from class 40 gray iron (generally considered unforgeable). The machined preform, a section of which is shown in Fig. 9, was heated to 955 °C (1750 °F) and forged at an energy level of 271,000 J (200,000 ft · lbf). This part had the smallest tolerance specification. The diameter over the teeth and the thickness of the body had a tolerance of +0.00 mm, –0.18 mm (+0.000 in., –0.007 in.). The largest tolerance on the part was ±1.02 mm (±0.040 in.) on the diameter of a recess. Tolerances on the other recesses were ±0.18 mm (±0.007 in.) and +0.48 mm, –0.00 mm (+0.019 in., –0.000 in.). This gear was forged to the finished dimensions.

Various gears with teeth as an integral part have been forged. These have ranged in outside diameter from 64 to 267 mm (2.5 to 10.5 in.) and in weight from 0.54 to 11 kg (1.2 to 24 lb). Most have been made with 0.13 to 0.51 mm (0.005 to 0.020 in.) of stock on the flank of each tooth for finish hobbing and grinding. Gears forged with integral teeth normally have longer

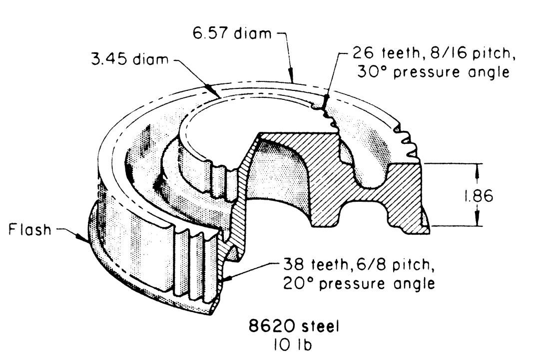


Fig. 8 Near-net shape cluster gear made by high-energy rate forging. Dimensions are given in inches.

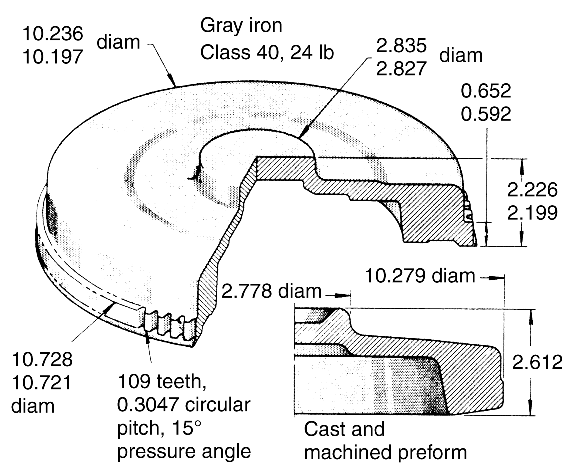


Fig. 9 Near-net shape automotive flywheel made by high-energy rate forging. Dimensions are given in inches.

fatigue and wear life than those made from a conventionally forged blank on which the teeth are hobbed, shaped, or milled.

Precision Forging

The term "precision forging" does not specify a distinct forging process but rather describes a philosophical approach to forging. The goal of this approach is to produce a net shape, or at least a near-net shape, in the as-forged condition.

The term net indicates that no subsequent machining or finishing of a forged surface is required. Thus, a net shape forging requires no further work on any of the forged surfaces, although secondary operations may be required to produce minor holes, threads, and other such details. A near-net shape forging can be either one in which some but not all of the surfaces are net or one in which the surfaces require only minimal machining or finishing. Precision forging is sometimes described as close-tolerance forging to emphasize the goal of achieving, solely through the hot forging operation, the dimensional and surface finish tolerances required in the finished part.

In recent years, computer-aided design and manufacturing (CAD/CAM) techniques have been applied to various forging processes (Ref 7). This computerized approach is applicable to precision hot forging of spiral bevel, spur, and helical gears in conventional presses in that it allows the die designer to examine the effects of various process variables (loads, stresses, and temperature) on the die design.

Precision hot-forged gears have the same advantages over cut gears as other molded gears (cast, P/M processed, injection molded) in that there is little or no material lost (Fig. 10). This is a cost savings from the standpoint of both the cost of the material itself and, more importantly, the cost of machining. In addition, precision forged gears also have the advantage over cut gears of increased load-carrying capacity. This added strength in the form of increased fatigue strength is due to the difference in grain flow between gears cut from bar stock and forged gears. The grain flow in cut gears is determined by the hot rolled orientation of the bar stock and has no relationship to gear tooth contour. On forged specimens, the grain flow follows the tooth contour in every gear tooth. Figure 11 compares the fatigue properties of cut and forged gears (Ref 8).

Near-Net Shape Quality Gears. The majority of forged gears produced today are near-net shape configurations. Gear teeth are forged with an envelope of material (stock allowance) around the tooth profile. This envelope is subse-

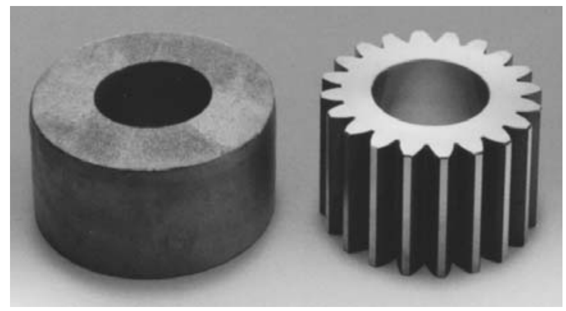


Fig. 10 Material/weight savings using the near-net shape forging process. The large spur gear weighs 25 kg (55 lb) as a blank (left side). As a forged tooth gear (right side) with 1 mm (0.04 in.) of stock allowance on the tooth profile for finish machining, it weighs 17 kg (37 lb). Source: Presrite Corporation

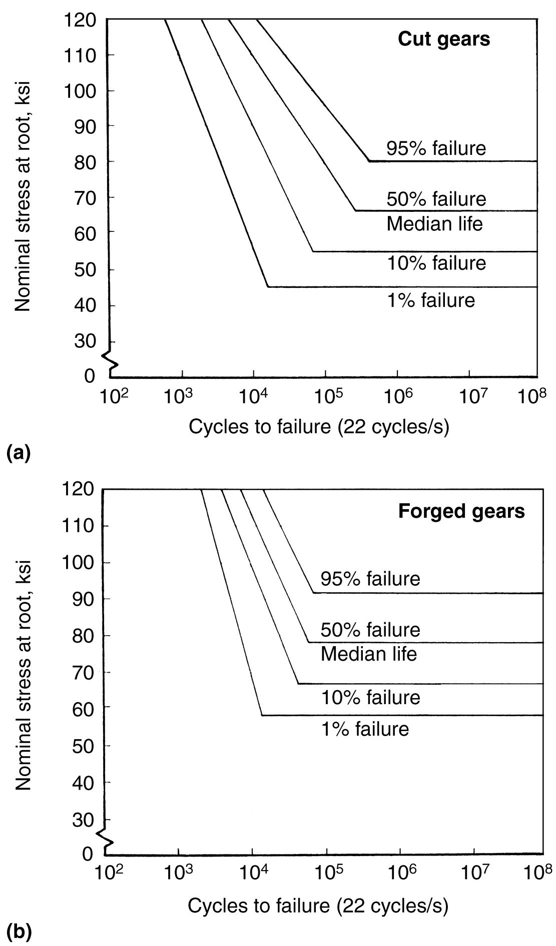


Fig. 11 Fatigue data for (a) cut gears and (b) near-net shape forged gears. Source: Ref 8

quently removed by the forging house or the customer purchasing the forged gears.

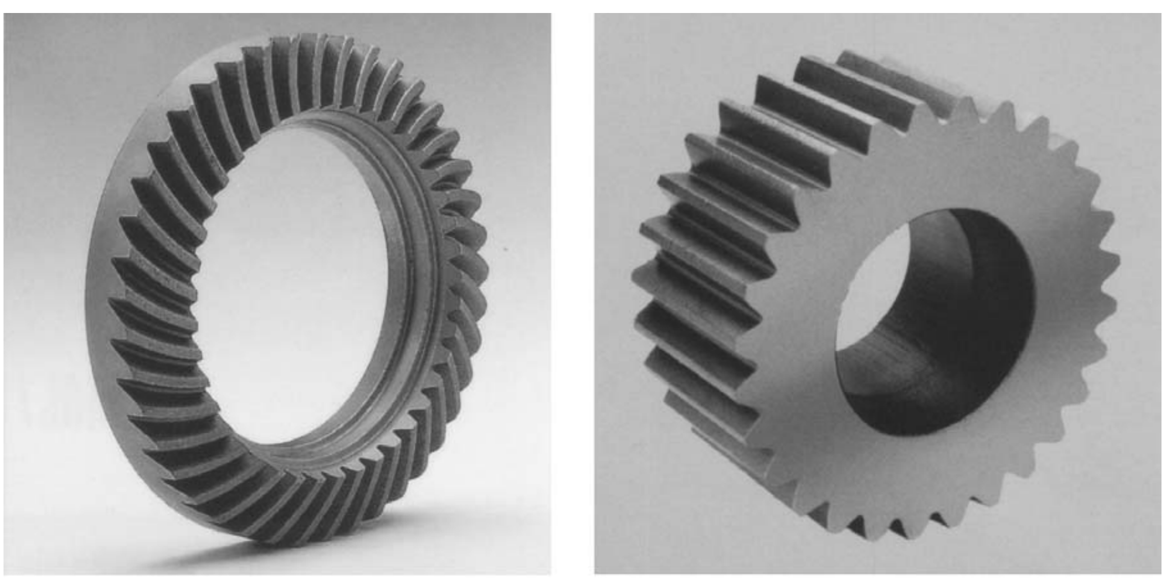
The manufacturing process begins with steel bar stock, usually turned and polished to improve the surface, and cut to the exact weight. The exact weight is critical because the amount of steel must completely fill the die to produce the complete gear profile. Prior to forging, billets are heated between 925 and 1230 °C (1700 and 2250 °F) in an electrical induction furnace that is controlled by an optical pyrometer to ±14 °C (25 °F).

In a single stroke, standard mechanical forging presses, ranging from 14,235 to 53,375 kN (1600 to 6000 tonf), form near-net shape gears with the complete allowable stock allowance. The purpose of this first operation, which forms a "pancake," is to break the scale off of the billet and size the outside diameter to just under the size of the root diameter in the gear die. Next, an operator positions the billet into the finish die. After forging, a hydraulic knockout system immediately extracts the gear from the finish die.

After the raw forged gear is hydraulically ejected from the die, it is placed in a trimming nest where the hole is punched. It is then allowed to cool to ambient temperature, which usually takes up to 24 h. Once cooled, it is ready for finish machining.

Near-net shape gears can be produced using any carburizing or induction hardening steel in five basic configurations: spiral bevel, helical, straight bevel, spur gears with a 1 mm (0.04 in.) stock allowance, and spur gears with a 0.1 to 0.3 mm (0.004 to 0.012 in.) stock allowance. The near-net shape gears can be produced in diameters up to 425 mm (17 in.) with stock allowances ranging from 0.1 to 1.5 mm (0.004 to 0.06 in.). The specifications for various gear configurations include:

- Spiral bevel gears can be produced up to 425 mm (17 in.) in diameter, with 0.5 mm (0.02 in.) minimum stock per flank. A maximum spiral angle of 35° and a pitch range of less than 7 DP can be achieved.
- Straight bevel gears have configurations/properties similar to spiral bevel gears
- Helical gears can be produced up to 250 mm (10 in.) in diameter and up to 40 kg (90 lb) in weight, with a 0.5 mm (0.02 in.) minimum stock per flank. A maximum helix angle of 25° and a pitch range of 4 to 12 DP can be achieved.
- Spur gears with a 1 mm (0.04 in.) stock allowance can be produced up to 400 mm (16 in.) in diameter and up to 135 kg (300 lb) in weight, with a 1 mm (0.04 in.) minimum



(a) (b)

Fig. 12 Examples of near-net shape forged gears. (a) Spiral bevel gear with a 0.5 mm (0.02 in.) stock allowance developed for use on gears with a DP less than 7. (b) Coarse-pitch (less than 5 DP) spur gear with a stock allowance of 1 to 2 mm (0.04 to 0.80 in.). Source: Presrite Corporation stock allowance per flank. A pitch range of less than 5 DP can be achieved. This type of gear requires a finish process of hobbing, hobbing and shaving, or hobbing and grinding, or skiving.

- Spur gears with a 0.1 to 0.3 mm (0.004 to 0.012 in.) stock allowance can be produced up to 250 mm (10 in.) in diameter and up to 150 mm (6 in.) maximum face width, with a 0.1 to 0.3 mm (0.004 to 0.012 in.) stock allowance per flank. A pitch range of 4 to 12 DP can be achieved. This type of gear requires a finish process of grinding or skiving. A net root is possible.

Figure 12 shows examples of precision forged near-net shape gears.

REFERENCES

1. T.T. Krenzer and J.W. Coniglio, Gear Manufacture, *Machining,* Vol 16, *ASM Handbook,* ASM International, 1989, p 330–355
2. J.G. Bralla, *Handbook of Product Design for Manufacturing,* Vol 16, McGraw-Hill Book Company, 1986, p 355
3. M. Blair and T.L. Stevens, Ed., *Steel Castings Handbook,* Sixth Edition, Steel Founders' Society and ASM International, 1995, p 2–34
4. D.W. Dudley, Gear-Manufacturing Methods, *Handbook of Practical Gear Design,* McGraw-Hill Book Company, p 5.86
5. C. Cooper, Alternative Gear Manufacturing, *Gear Technol.,* July/Aug 1998, p 9–16
6. D.P. Townsend, Ed., Gears Made by Dies, *Dudley's Gear Handbook: The Design, Manufacture, and Application of Gears,* Second Edition, McGraw Hill Book Company, 1992, p 17.1–17.21
7. D.J. Kuhlmann and P.S. Raghupathi, Manufacturing of Forged and Extruded Gears, *Gear Technol.,* July/Aug 1990, p 36–45
8. T. Russell and L. Danis, Precision Flow Forged Gears, *Gear Manufacture and Performance,* American Society for Metals, 1974, p 229–239

Gear Materials, Properties, and Manufacture
J.R. Davis, editor, p139-153
DOI:10.1361/gmpm2005p139

CHAPTER 7

Powder Metallurgy

POWDER METALLURGY (P/M) is a flexible metalworking process for the production of gears. The P/M process is capable of producing close tolerance gears with strengths to 1240 MPa (180 ksi) at economical prices in higher volume quantities. Spur, helical, bevel, face, spur-helical, and helical-helical gears are produced by P/M techniques. The process is particularly attractive when the gear contains depressions, through holes, varying levels, or projections.

The first application of P/M gears was for oil pumps in 1937/1938 (Ref 1). Gear pumps constitute a common type of application. Depending on application stress levels, different materials and density levels are used. For example, iron-carbon or iron-carbon-copper materials with densities of 6.0 to 6.8 g/cm^3 are used in gear pumps for low-pressure applications such as engine lubrication and automatic transmissions. For higher stress applications up to 20 MPa (3 ksi), alloy steels at the minimum density of 7.1 g/cm^3 are used. Powder metallurgy material requirements for gear pump applications at various stress levels are given in Table 1.

With the considerable progress in P/M technology over the years, the quality and application of P/M gears has increased. Progress in production, compaction, and sintering has resulted in new techniques such as infiltration, powder forging, surface rolling, and several finishing treatments. These developments have enabled the P/M parts to compete successfully with wrought metal counterparts.

Figure 1 summarizes the various P/M manufacturing methods used in gear production. The conventional process consisting of compaction (pressing) and sintering is the most commonly employed method for producing gears. Selection of a process method depends on factors such as dimensional accuracy. The American Gear Manufacturers Association (AGMA) quality classes 5 to 7 can be obtained through conventional press-and-sinter operations. When AGMA quality levels above 7 are required, secondary operations such as coining, burnishing, and grinding typically are employed. See the section “Gear Tolerances” in this chapter for additional information.

Performance of any P/M part is influenced primarily by density. Theoretical density of steel is around 7.85 g/cm^3, and Table 2 summarizes density levels obtained from various P/M processes for steel. All of these P/M methods are employed for gear manufacture, including

Table 1 Material requirements for P/M pump gear applications

	Material	
Application	Designation(a)	Density, g/cm^3
Output pressure less than 0.69 MPa (100 psi) and light service	FC-0208 type; as-sintered	5.8–6.2
Output pressure less than 6.9 MPa (1000 psi) but subject to high vibration and heavy service; hardened by heat treating	FC-0208 or FN-0106	6.4–6.8
Medium high pressure 6.9–8.6 MPa (1000–1250 psi) for short intervals; hardened by heat treating	FC-0208 type	6.2–6.5
General-purpose hydraulic pumps up to 10.3 MPa (1500 psi) continuous service	FN-0106 type; heat treated	6.8–7.1
General-purpose hydraulic pumps up to 17.2 MPa (2500 psi) continuous service	AISI 4630 type; heat treated	7.2–7.6

(a) FC-0208 is a copper infiltrated steel. FN-0106 is a nickel steel. Source: Ref 1 various surface treatment such as carbonitriding, shot peening, and steam oxide treatments (Ref 2).

A wide variety of base metals are available in powder form, such as brass, bronze, iron, low-alloy steels, and stainless steel. The selection is governed by required mechanical properties, physical properties, service conditions, and cost. Table 3 lists commonly used P/M materials and broad applications of standard grades. Customized mixing and blending of elemental powders also provides a variety of possibilities for the development of alloy compositions formulated for specific mechanical and physical properties.

Capabilities and Limitations

Powder metal gears are used in various applications for appliances, office machines, machinery, machine tools, and automotive products. The P/M process offers various advantages

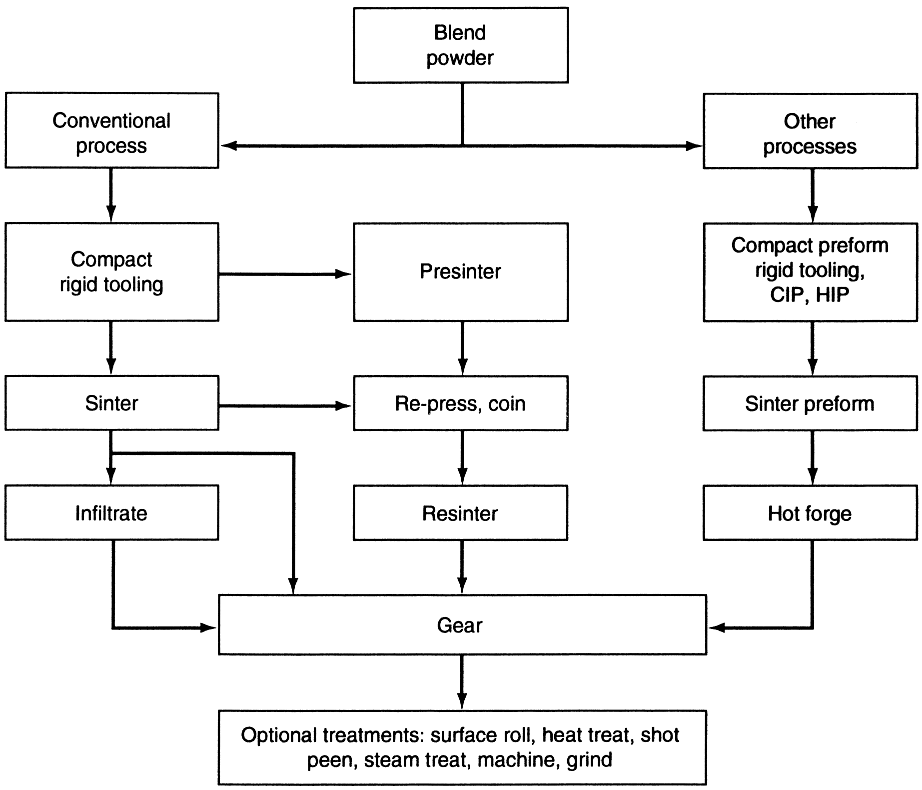


Fig. 1 P/M gear production process. CIP, cold isostatic pressing; HIP, hot isostatic pressing

Table 2 Density range for P/M steel consolidation methods

P/M method	Density range, g/cm^3	Notes
Pressed-and-sintered steel	6.9–7.1	Conventional powder metallurgy
Double processing (press, presinter, restrike, full sinter, heat treat)	7.2–7.4	Restrike (or repress) tooling has tolerances closer to finish dimensions than the tooling for the first strike
Warm compaction (single pressed)	7.2–7.5	Specially formulated powders to obtain higher green strength at lower compaction pressure
Warm compaction (double pressed)	7.4–7.7	...
Cold densification (press, sinter, cold form, heat treat)	7.6–7.8	Cold forming improves fatigue and wear resistance
Powder forging (press, sinter, forge, heat treat)	7.6–7.8+	Limited to spur, bevel, and face gears although trials with helical gears have been done (Ref 2)
Powder injection molding	7.8+	...
Copper infiltrated steel	Near full density	Copper infiltration also improves machinability
Roll densification	7.6–7.8	Increases localized densification for wear and fatigue resistance

over traditional gear manufacturing for various applications.

In traditional methods, gears are manufactured from blanks obtained by machining of castings, forgings, and rolled bar stock. These blanks are machined by various methods such as milling, hobbing, and shaping. For precision and high-speed applications, the gear teeth are finished by secondary operations such as shaving and grinding. Gear production by these methods has the following disadvantages:

- Processing times are long, particularly with difficult-to-machine alloys.
- Material utilization is very low.
- Machined gear teeth have an unfavorable grain flow pattern with flow lines intersecting the gear tooth profile.

In comparison to the traditional gear manufacture process, the P/M process offers several advantages, particularly the elimination of machining and scrap losses for the manufacture of various gear types such as helical, bevel (both straight and spiral), rack, face, internal and external spur gears, and also compound gears. Internal configurations (splines, keys, keyways) also can be formed simultaneously with the gear profile during P/M manufacture, eliminating subsequent machining operations. Thus, P/M gear manufacturing allows efficient utilization of materials for moderate- to high-volume production. The minimum quantity required to make the P/M process economically competitive with machining depends on the size and complexity of the part and its accuracy and other property requirements. While P/M processing is best suited and most widely known for high-volume production quantities, there are numerous applications where small quantities (a thousand parts) can still offer cost benefits over traditional manufacturing methods.

General process advantages of P/M gear manufacture include:

- Economy in mass production
- Repeatability and uniformity of part features and dimensions
- Production of multilevel gears
- Close control of density or, conversely, porosity to suit a particular application
- Reduction or elimination of secondary operations
- Improved surface finishes on gear teeth by reducing or eliminating the machining marks (or "scoring") that can be imparted by conventional methods
- Self-lubricating ability from impregnation of oils and lubricants
- Noise reduction from the sound dampening qualities of the pore structure of a P/M component, high surface finish and tooth form consistency also can have an effect on overall reduction of noise in comparison to cut gear sets of comparable quality
- Weight reduction by incorporating weight saving or lightening holes (or similar shapes) in the initial tool design

Limitations of the P/M process for gear manufacturing include (Ref 2):

- Not economical in low- and medium-quantity production
- Not normally suitable for production of worn gears, herringbone gears, and helical gears with helix angle exceeding 35° because of various pressing and tooling considerations
- In view of the press capacities required for compaction, the sizes of gears are limited
- The gear thickness or face width is restricted by the requirement that the depth of the die cavity and the stroke of the press must be at least 2 to 2½ times the gear thickness

Table 3 Commonly used metal powder grades

MPIF designation	Chemistry	Applications
F-0000	Pure Fe	Low strength at high density, soft magnetic properties
F-0005	Fe + 0.5% C	Moderate strength, medium-carbon steel
FC-0208	Fe + 2% Cu + 0.8% C	Higher-strength structural components
FN-0205	Fe + 2% Ni + 0.5% C	High strength (heat treated), good impact resistance
FX-2008	Fe + 20% Cu + 0.8% C	Copper-infiltrated steel, high strength, machinable
SS-316L	316 stainless steel	Good toughness, corrosion resistance
SS-410	410 stainless steel	Good hardenability, abrasion resistance
CZ-2002	Brass	Good toughness, elongation, corrosion resistance
CT-1000	Bronze	Structural and bearing applications
CNZ-1818	Nickel silver	Decorative, tough, ductile, corrosion resistance

MPIF: Metal Powder Industries Federation

Gear Forms

The size of P/M gears range from as small as a few millimeters to more than 300 mm in diameter. For all practical purposes the face width of P/M gears is limited to less than 75 mm (3 in.). The commonly used compaction presses have insufficient press stroke to consolidate the powder to greater than this length (face width).

The following types of gears can be produced by the P/M process:

- Spur gears
- Bevel gears
- Face gears
- Hypoid gears
- Helical gears of helix angle not exceeding 35°

Combination gears or multiple gears in which two or more gear forms are combined into one component are a specialty of the P/M process.

Some examples of P/M gear design are shown in Fig. 2 to 5. True involute forms (Fig. 3a) that are expensive to make by other methods can be easily made by the P/M process. The P/M process also is particularly advantageous for gear designs that have irregular curves, radial projections, keyways, or recesses, because these features are easily obtainable without requiring secondary machining operations. Full tooth face width for compound gears and sufficient fillet radii at the root diameter for tip clearance can be attained without allowances for cutter clearance.

Spur gears are the most common type of P/M gear in use today. The P/M process is capable of producing spur gears of many tooth forms and modifications. If an electrodischarge machining (EDM) electrode can be ground or cut, the mating form can be duplicated in P/M dies.

Compared to wrought metal gears, the face width or thickness of a spur gear is restricted in powder metallurgy in order that the density and mechanical properties are maintained fairly uniform throughout the section. Fine pitch (module <0.75 mm) spur gear teeth present powder filling problems, which are more serious if the face width is large.

Helical gears are also commonly made by the P/M process. However, in contrast to metal-cutting processes, the helix angle of P/M gears

Fig. 2 Examples of powder metallurgy gear forms. Courtesy of Carbon City Products

is limited to approximately 35°. As with spur gears, many tooth forms and modifications are possible.

Helical gears have not been produced by powder forging, but recent production tests have been performed on a unique tooling method for powder forging of helical gears. One production trial involved the powder forging of a helical gear with a 31° helix angle and a normal pressure angle of 15° for an automotive transmission part (Ref 4). Experimental work later carried out by the German Research Society showed the superiority of the finished product in terms of toughness, tooth root fatigue, and contact fatigue (Ref 5). During powder forging of a helical gear, the die is the only helical part of the tool.

Bevel and face gears are readily made by the P/M process, but differ somewhat from spur and helical gears in mechanical property flexibility. Due to the nature of the compaction process, sections such as face gear teeth cannot be made to high densities without resorting to copper infiltration or repressing. Most bevel and face gear forms can be made providing they are not undercut, but the maximum mechanical properties attainable will be somewhat less than those that can be achieved for spur and helical gears.

All face tooth forms must be such that the green compact does not lock into the punch. For this reason undercut gear forms cannot be produced. Under the action of the punch, the crown area of face gear teeth also normally receives less compaction than the root zone. This results in an undesirable density variation, which can be offset by copper infiltration or repressing.

Another method to offset density variations is by putting a nonfunctional gear form with its teeth in the offset position so that a tooth gap comes below a tooth crown; this allows full punch pressure to be exerted on the tooth crowns of the functional gear, enabling a better density distribution (Ref 3).

Combination Gears. Powder metallurgy gear manufacture allows the production of combination gears or multilevel (two-to-four level) gears resulting in a single part with two or more gear forms. All the gear forms can be positioned close to one another, thus making the part very compact. This helps in saving space and handling and assembly costs. Combination gears can be produced by either of the two methods:

- Compacting the part in the combined form
- Compacting separately, assembling them in green state and sintering

Gear Tolerances

Tolerances of P/M gears are determined by the compaction tooling, compaction process, sintering process, and the P/M alloy. Tooth form and dimensions remain relatively stable throughout the process. The density variation in the compact should not exceed 0.2 g/cm^3, otherwise, dimensional control will be difficult during sintering.

Powder metallurgy gears are produced typically in AGMA 5 to 7 quality levels. Higher AGMA classification is possible, depending on the size of the compacted gear and the additional secondary operations. Higher AGMA classification increases the cost of manufacturing due to the secondary (machining) or restrike operations that are added to meet the tighter tolerances. In general, traditional press-and-sinter operations

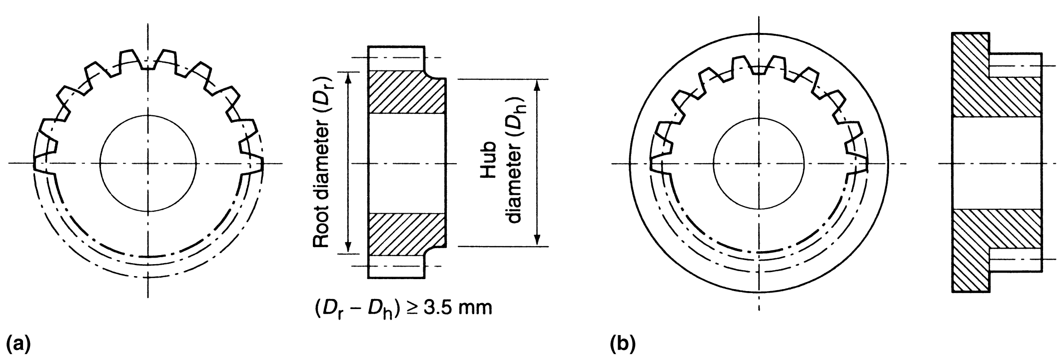


Fig. 3 Features produced in powder metallurgy gears. (a) True involute forms. (b) Gear teeth straight up to a flange in P/M gear. Source: Ref 3

are economical for achieving AGMA quality levels 5 to 7. It is not advisable to produce higher quality levels through only compaction and sintering, as closer tolerances and tool wear will impact costs. Where higher accuracy is required, secondary operations such as sizing, shaving, burnishing, and grinding are required. Sizing (repressing) is the most common method because it improves mechanical properties through cold working.

Because dimensional variation is a function of part geometry, material, density, and subsequent thermal operations, dimensional tolerances may vary by application.

Powder metallurgy gear tolerances are determined by the compaction tooling, compaction process, sintering techniques, and the alloy. The tolerances can be divided into functional tolerances that directly affect gear function and nonfunctional elements that pertain to portions of the gear that are more properly termed general machine design considerations. Nonfunctional elements include such items as hub design dimensions, fastening designs such as keyways, set screw holes, inside diameters for press fits, and other special body configurations.

Functional gear tolerances are controlled by the compaction tooling, the processing methods, and gear material. Because most P/M gears are pressed in carbide dies, the tooth form and dimensions remain relatively stable and constant throughout the process. Concentricity of the gear outside diameter and pitch diameter to the bore are also controlled by the tooling. If the pressing core rod or punches are eccentric to the die, the parts will be eccentric. Once the part is pressed, the material characteristics and sintering process determine the dimensional tolerance capability of the outside diameter, pitch diameter, test radius, and bore diameter.

Because of the number of variables within the P/M process it is not possible to state absolute dimensional tolerance capabilities. However minimum tolerances that one should expect for ferrous-base P/M gears without secondary operations are listed in Table 4 for spur and helical gears and in Table 5 for bevel gears. Typical tolerances for single press and sinter are listed in Table 6.

Gear Design and Tooling

Properly designed gears and compaction tooling are the key factors in the economical production of P/M gears with consistent quality.

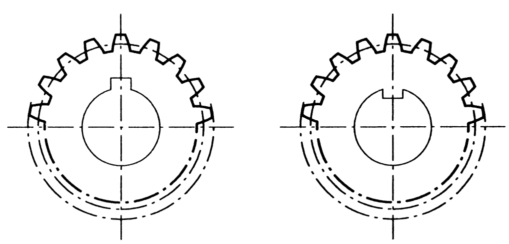


Fig. 4 Keyways and integral keys. Source: Ref 3

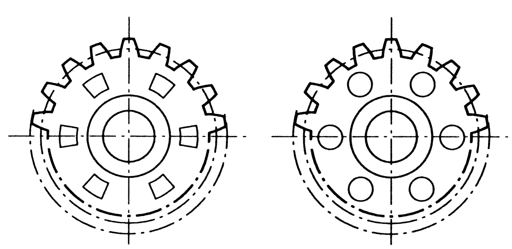


Fig. 5 Weight reducing holes on P/M gear. Source: Ref 3

Table 4 Minimum tolerance capabilities for spur and helical P/M gears

Tooth-to-tooth composite tolerance	AGMA class 6
Total composite tolerance	AGMA class 6
Test radius	0.05 mm + 0.002 × diameter
Over-pin measurement	0.10 mm + 0.002 × diameter
Lead error	0.001 mm/mm
Perpendicularity—face to bore	0.05 mm + 0.001 × diameter
Parallelism	0.025 mm + 0.001 × diameter
Outside diameter	0.10 mm + 0.002 × diameter
Profile tolerance	0.008 mm

Tolerances with single press and sinter without secondary operations. Source: Ref 3

Table 5 Minimum tolerance capabilities for bevel P/M gears

Tooth-to-tooth composite tolerance	AGMA class 6
Total composite tolerance	AGMA class 6
Outside diameter	0.10 mm + 0.002 × diameter
Perpendicularity—face to bore	0.05 mm + 0.001 × diameter

Tolerances with single press and sinter without secondary operations. Source: Ref 3

Blended metal powder is usually compacted at room temperature, at pressures between 275 to 690 MPa (40 to 100 ksi) of projected surface area. The tooling for a single-level gear includes a die, an upper punch, and a lower punch. If the particular component to be formed requires a bore or other inside-diameter configuration, a core rod is also part of the tooling. Multilevel compound gears or other complex structural parts may have two or more upper and/or lower punches (Fig. 6).

To form the gear, the die is filled with powder at a ratio of approximately two times the parts thickness; for example, a spur gear that has a thickness of 19 mm (0.75 in.) will be compacted to form a column of 38 mm (1.50 in.) thick. During the compaction cycle of the more commonly used type of press, the upper punch enters the die while the lower punch, which in the fill position seals off the bottom of the die, remains stationary. As the upper punch travels downward, the compressive forces cause the die assembly to move downward in relationship to the lower punch, resulting in the same effect as if the lower punch were moving upward during the compaction stroke. The rate of movement and pressure of the upper punch and the motion of the die are relatively equal to ensure uniform density within the compacted preform. After completion of the compaction cycle, the upper punch retracts from the die, and the lower punch initiates an upward motion, ejecting the preform from the dies.

The shape of the compacted preform is determined by the shape of the tooling and the axial motion of the compaction press. Two main factors influence part design: the flow behavior and characteristics of the metal powders and the movement of the tools within the pressing cycle. Metal powders do not flow hydraulically, and the allowance for friction between the powder particles themselves and with the moving tool members must be factored in the final P/M part design. The pressing action from both top and bottom largely governs the shape, length, and dimensional details of the preform. The clearance tolerance of the moving tool members, relative to the inside diameter or bore, contributes to a larger degree of eccentricity than with machined gears. Details, or features that increase the number of tool members compound the eccentricity. Concentricity can be improved with the addition of secondary operations, such as grinding the bore in relationship to the pitch diameter.

The compacted preform is sintered typically at temperatures of approximately 1120 °C (2050 °F) to obtain tensile properties around 125 to 620 MPa (18 to 90 ksi), depending on the material composition. The addition of heat treatment can increase ultimate tensile properties of a compacted-and-sintered steel component to 1240 MPa (180 ksi).

Table 6 Typical tolerances of single pressed and sintered gears

See Tables 4 and 5 for specific gear forms.

Feature	Tolerance (without secondary operations) mm	in.
Length	±0.125	±0.005
Flatness on end	0.05	0.002
Concentricity	±0.075	±0.003(a)
Inside diameter	±0.050	±0.002(a)
Profile tolerance	±0.0075	±0.0003
Tooth-to-tooth error	±0.025	±0.001
Circular tooth error	±0.05	±0.002
Parallelism on ends	±0.04	±0.0015
Lead error	±0.025	±0.001
Total composite error	±0.08	±0.0032(b)
Outside diameter tolerance with pitch diameter up to:		
25 mm (1 in.)	0.075	0.003(c)
50 mm (2 in.)	0.1	0.004(c)
75 mm (3 in.)	0.125	0.005(c)
100 mm (4 in.)	0.15	0.006(c)

(a) Tolerance depends on feature size. (b) Tolerance depends on gear size and number of levels. (c) Total tolerance on outside diameter and measurement over wires

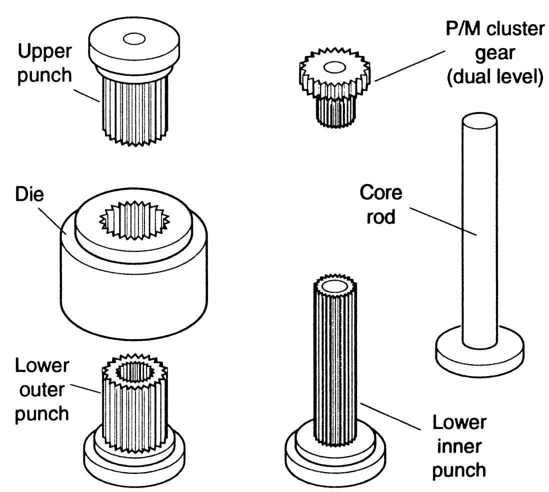


Fig. 6 Compaction tooling for compound gear. Source: Ref 6

Design Guidelines

Design guidelines for P/M gears include the following considerations:

- Ejection from the die
- Die filling during compaction
- Clearance
- Thickness changes
- Right-angle intersections
- Edge detail

Ejection from the Die. Some features must be eliminated from the final part (or added through secondary machining operations) when they inhibit ejection of the preform from the die. For example, undercuts, reverse angles, details at right angles to the direction of pressing (e.g., holes, grooves), threads, diamond knurls and reentrant angles would interfere with smooth ejection.

Die Filling During Compaction. Gear design should allow for the movement of the metal powders throughout the tool members during the compaction cycle. Metal powders do not flow hydraulically. Therefore, extremely thin-walled sections, very narrow grooves, and deep counter-bores should be avoided because the metal powders may not completely fill these details within the tool members.

Tool life can be improved by avoiding narrow deep grooves, very sharp edges, complete spherical profiles, and knife-thin tool thicknesses. Often design simplification of these features allows for a more robust or "practical" set of compaction tools without adding machining operations.

Clearance. Hubs or bosses for gears, sprockets, or cams can be readily produced, but the designs should ensure the maximum permissible material between the outside diameter of the hub and the root diameter of gear or sprocket features (Fig. 7) (Ref 6). Another consideration for gear design is to maintain sufficient clearance between the inside diameter and the root diameter, which can range from 0.9 mm (0.035 in.) on small pinions to 7.5 mm (0.30 in.) for more demanding applications. Design consideration should be given to maximize the clearance between the gear root diameter and compound gears, pinions, or features (i.e., cams, etc.) that are incorporated parallel to the gear teeth.

Thickness Changes. The configuration of the gear should limit major changes in section thickness (Fig. 8) for compound gear geometries such as gear-pinion and spur-face combinations, which are well within the capability of the P/M process. Uniform density and high strength is best achieved by limiting the number of section thickness changes (levels) that are designed into the preform. The number of levels a part may have is determined by the specific type of compaction press and/or the design of the compaction tooling.

Right-Angle Intersections. The configuration of the gear should minimize right-angle intersections. Radii should be incorporated at right-angle intersection of section thickness changes. These radii improve the integrity of the preform. Gear engagement should be designed to occur above the radius. Often raised surface features can be added to the compacted preform to assist in this engagement.

Edge Detail. The configuration of the gear should incorporate edge detail (i.e., chamfers on top and bottom of the gear teeth and on outside and inside diameters) (Fig. 9). This edge detail of chamfer has two main purposes. First, it increases the density of the teeth. Higher tooth density results in improved mechanical properties, particularly strength. It also reduces the adverse effects of burrs. The chamfer detail will in most cases keep the burr within the overall

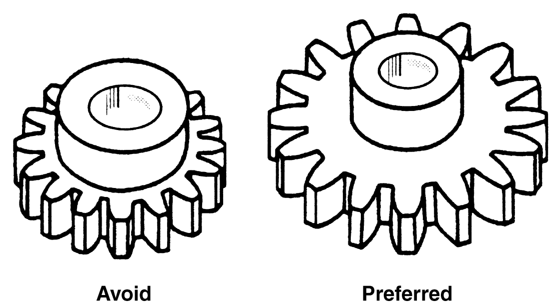


Fig. 7 Hub feature. Source: Ref 6

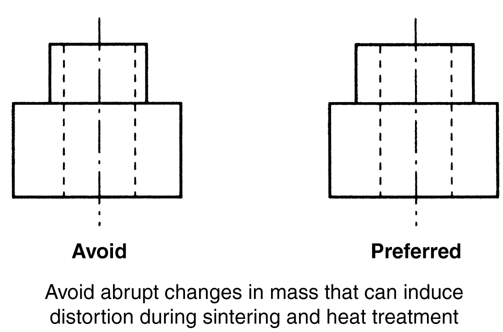


Fig. 8 Preferred design for section thickness changes. Source: Ref 6

thickness (width) of the gear. These burrs are a result of the fit clearance of the tool members and, if necessary, can be removed by a vibratory finishing (tumbling) operation.

Gear Performance

In general terms, gear teeth loading is akin to normal loading of two parallel cylinders with a combined sliding and rolling. In rolling contact, two elements roll in contact with each other without slippage. Both elements rotate at the same velocity. Gear teeth have sliding contact over most of their surfaces, with rolling contact occurring only at the pitch line. The amount of sliding varies with the distance from the pitch line. When the rolling and sliding are in the same direction, less damage occurs than with negative slide (Ref 7).

Gears can be damaged by several mechanisms such as bending fatigue or by surface fatigue mechanisms such as pitting or spalling. The origin of pitting occurs where the combination of pressure and negative sliding are the highest. On spur gears, this is usually at the lowest point of single-tooth contact on the dedendum of the drive gear (Ref 8).

As with any other material the choice of a P/M alloy is governed by the magnitude and nature of the transmitted load, the speed, the life requirements, the environment, the type of lubrication, and the gear and assembly precision. Allowable loads on contact surfaces of gears are limited by the occurrence of pitting, in most cases, provided they are not run at very high speeds to cause scoring (Fig. 10). Even at light loads, gear teeth experience relatively high concentrated stresses because contact areas are very small. In general, Hertz equations are used to design for contact stresses, and these equations consider compressive loads on stationary contact surfaces.

Spalling damage of wrought and P/M alloys has similar appearance but different crack propagation mechanisms. In wrought alloys, most of the time is spent in the crack initiation phase (under high-cycle conditions) with relatively quick crack propagation once the crack is big enough. In P/M alloys, the porosity blunts crack growth to the surface. Until the crack can propagate to the surface, a band of subsurface cracking continues. The depth of the band is probably related to the depth of the maximum shear stress.

Selecting Gear Materials

Required strength is the first factor when selecting a gear material. In spite of the approximations involved, one can generally arrive at the correct material by using the methods outlined in the AGMA standards for rating the strength of spur, helical, and straight bevel gear teeth (Ref 9–11). A good approximation of compressive load carrying ability can be found by using the methods outlined in the AGMA standards for surface durability of spur, helical, and bevel gear teeth (Ref 12–14). In addition, the allowable contact stress S_{ac}, the elastic modulus, and Poisson's ratio of the P/M material must be determined.

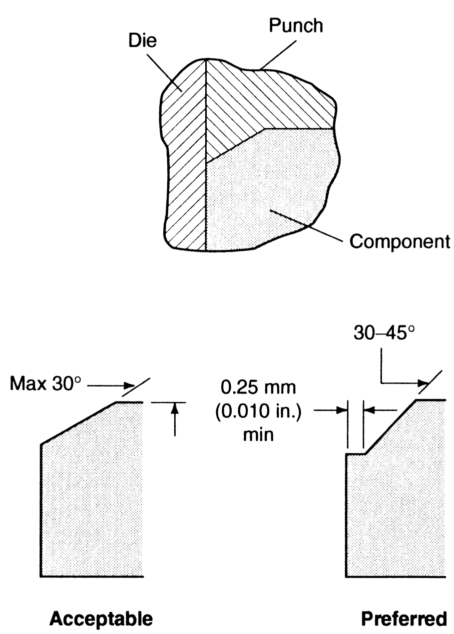


Fig. 9 Edge radii design features. Source: Ref 6

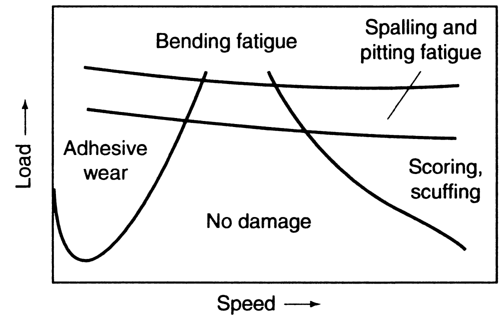


Fig. 10 Schematic of gear damage mechanisms. Source: Ref 15

Allowable Contact Stress. Tensile strength, elastic modulus, and Poisson's ratio for most ferrous P/M materials is given in MPIF standard 35. In the absence of data for allowable contact stress of ferrous P/M materials, an approximation of S_{ac} is 70 MPa (10 ksi) below the ultimate tensile strength (S_{ac} = UTS = 10 ksi).

Allowable bending stress (S_{ab}) for a P/M material can be approximated in the absence of data at 0.3 of the ultimate tensile strength of ferrous-base P/M materials (S_{ab} = 0.3 UTS). The only caution one should be aware of is that tensile data may be typical properties. The designer should consult the supplier or specifications for minimum property data.

Impact and Fatigue Strength. Most P/M gears are limited in performance by their impact and fatigue strength as opposed to their compressive yield strength and surface durability. Surface treatments such as shot peening, material cleanliness, and other factors influence contact fatigue. Detailed information on P/M gear fatigue testing and evaluation is discussed further in Ref 15 and 16. Additional information on fatigue is also contained in Chapter 13, "Gear Failure Modes and Analysis" and Chapter 14, "Fatigue and Life Prediction."

Quality Control and Inspection

Inspection is one of the most important stages in gear production. Inspection and testing is done on dimensional specifications, mechanical properties, and surface durability (hardness). The information required to describe a gear for the P/M process and subsequent inspection methods are essentially the same as for a machine-cut gear. The part drawing should include:

- Number of teeth
- Diametral pitch (normal diametral pitch for a helical gear)
- Pressure angle (normal pressure angle for a helical gear)
- Tooth form for spur, helical, and straight bevel gears
- Helix angle and rotation (right or left hand) for a helical gear
- Pitch, face, root, and back angles for straight bevel gears

To assist in the qualification (i.e., inspection) the drawing should also include:

- Outside diameter (as a reference dimension for straight bevel gears)
- Root diameter
- Pitch apex to back face
- Total composite tolerance
- Tooth-to-tooth tolerance (not appropriate for straight bevel gears)
- Test radius

Measurement over wires (MOW) and measurement between wires (MBW) (for internal gears) are convenient measurable criteria that should also be included as drawing specifications. Master gear information (specification and/or data) should be included on the drawing. Tip radius detail for spur gears is also necessary. It is advisable that right-angle (rolling) fixtures be developed to check mesh and backlash. Often several ID (inside diameter) pins are required to confirm mesh, after/during critical processing operations (minimally, compaction, sintering, and heat treatment). Typically, these inspection fixtures are specified and purchase at the time the tooling is manufactured.

Gear drawing specifications are outlined in ANSI/AGMA 6008-A98, "Specifications for Powder Metallurgy Gears." This standard defines the minimum detailed information to be included in the P/M gear specifications submitted by the gear purchaser to the gear producer. In addition to gear drawing specifications, this information also covers gear tooth geometry data and gear material specifications.

In production, it is common practice to check the face width, outside diameter, over-pin measurement, total composite error, tooth-to-tooth error, perpendicularity of face-to-bore, parallelism, and inside diameter of the gears. Also the tooth strength and macroscopic hardness are usually checked. The sampling plan and frequency of measurement varies and depends on the criticality of the application.

Gear inspection terms include:

- *Runout (radial):* the total variation of the radial distance of the gear teeth from the center or bore
- *Runout (axial):* the total variation of gear teeth along its axis, measured from a reference plane perpendicular to its axis. Also referred to as "wobble" or face runout
- *Lead:* the axial advance of a gear tooth in one revolution (360°)
- *Lead variation:* the condition in a gear where some teeth vary in lead, plus or minus from the average lead

- *Total composite error:* the total variation of the center distance in one complete revolution
- *Tooth-to-tooth variation:* the variation of the center distance in a section of its circumference (the section of the circumference is an angle of 360°/*N* where *N* is the number of gear teeth)

Mechanical Testing. Mechanical properties are also measured, and in many cases bench life and impact tests or field tests are made to check the gear functionality. Because tensile strength of a gear cannot be measured directly, it is appropriate to establish a minimum tooth breakage strength that can be used as a production acceptance criteria from either sample gears made from a die or gears cut from a P/M slug.

One quick and accurate means of measuring tooth strength of a spur gear is shown in Fig. 11. By mounting the gear between fixed supports and applying a vertical load, one can measure a tooth breakage strength. By testing a number of gears that perform successfully in the particular application, it is possible to calculate statistical limits for the tooth breakage strength that can be used as a future material strength acceptance. Helical and bevel gears are usually evaluated in torque-type tests and the same type of statistical analysis applied. The use of statistics is particularly expedient in gear strength testing because proper sampling allows process control to a normal statistical distribution (i.e., a $\pm 3\sigma$) minimum acceptance criteria ensures 99.85% confidence of adequate strength.

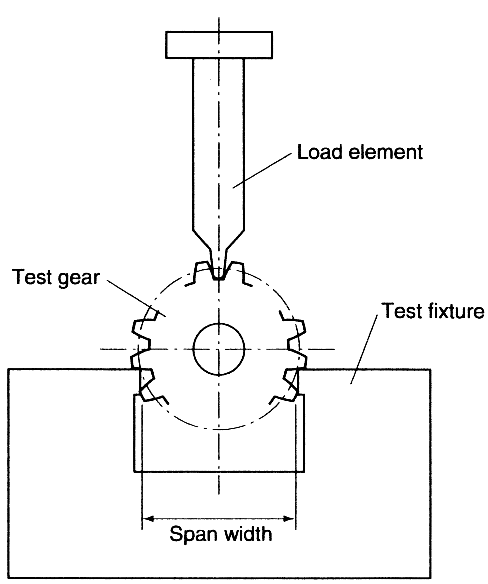


Fig. 11 Spur gear tooth strength test. Source: Ref 17

Surface durability is a specified and measured minimum Rockwell hardness because this macroscopic hardness relates to the compressive yield strength of the materials. The minimum Rockwell hardness depends on the material and can be established either from samples or from the supplier's historical information.

In cases where scoring may be a problem, in particular with heat treated gears, it is appropriate to specify and measure a microscopic hardness. This is accomplished with a diamond indenter, and the microscopic or individual particle hardness of the material is measured. The particle hardness specification is generally determined from historical data for the producer's material.

In contrast to most metals, P/M materials have both a macro- and microhardness as a result of porosity within the structure. Because the P/M structure consists of individual hard particles bonded together with interdispersed voids, the conventional Rockwell indention registers a composite hardness. Because sliding and surface contact involves microscopic contact (individual asperities) the particle or microhardness of the P/M material influences the score resistance of the P/M material more so than the macro Rockwell hardness. As a generality, when a P/M gear is run against a wrought gear, the P/M particle hardness should be no harder than the hardness of the wrought gear to prevent wear of the mating gear.

Examples of P/M Gears

Powder metallurgy gears are used in a wide variety of application areas including business machines, appliances, farm, lawn, and garden equipment, automobiles, trucks, and military vehicles. The examples that follow illustrate the versatility of the P/M process for gear manufacture.

Example 1: Drive Gear for Mailhandling Machine. A flat-belt drive gear weighing 490 g (17.3 oz), which is compacted to a density of 7.45 g/cm^3, is shown in Fig. 12. This type of gear is used in high-speed mailhandling and inserting machines. This part, consisting of electrolytic iron preblended with 0.35% C, replaced a two-piece gear assembly made of American Iron and Steel Institute (AISI) 1030 steel, thus resulting in a total cost savings of 66%.

The powder was first compacted on a 5340 kN (600 ton) press to a density of 6.8 to 6.9 g/cm^3 at a compaction pressure of 414 MPa (30 tsi). It was then presintered and repressed to a density of 7.45 g/cm^3 at 965 MPa (70 tsi). After repressing, the part was sintered at 1120 °C (2050 °F) for 30 min.

As shown in Fig. 12, considerable machining (turning) of the hub was required to generate the final shape. In use, this gear rotates and a magnetic clutch rides on the machined hub. When the operator trips the cycle control on the machine, the clutch closes around the small hub diameter, and the load is transferred to the P/M gear. Pressing of a large hub ensures a uniform density in the cross-sectional area, but requires additional machining of the P/M blank.

After machining, the hub area is carbonitrided to a case depth of 0.127 mm (0.005 in.). Case hardness is 700 DPH (minimum), while core hardness is 550 DPH. To maintain a 0.025 mm (0.001 in.) diameter tolerance between the hub and the magnetic clutch, the hub diameter is ground to final dimensions. Typical properties for this P/M part are:

Tensile strength, MPa (ksi)	414–586 (60–85)
Yield strength, MPa (ksi)	365–414 (53–60)
Elongation, %	1–2
Transverse-rupture strength, MPa (ksi)	930 (135)

Example 2: Transfer Gears for Postage Meter Counting Applications. The high-precision, high-strength gears shown in Fig. 13 are made of 4600 alloy powder prealloyed with 0.25% Mn, 1.7 to 1.9% Ni, and 0.5 to 0.6% Mo and subsequently preblended with 0.75% Cu and 0.35 to 0.45% C. These gears are used in postage meters for counting applications.

The powder was compacted in a floating die press system to a density of 6.85 to 7.0 g/cm^3. Sintering was conducted in an endothermic atmosphere at 1120 °C (2050 °F). After sintering, the parts were carbonitrided to provide a case hardness of 700 DPH for a 0.127 mm (0.005 in.) case depth and a core hardness of 400 DPH. After case hardening, the parts were deburred and oil impregnated.

The original method for manufacturing these gears consisted of a two-piece copper-brazed and gear-hobbed assembly of AISI 1030 steel. The larger (bottom) gear was first turned as a disk with a center hole. Gear teeth were then hobbed.

The smaller (top) gear was also turned and gear hobbed. The two parts were then press fitted and copper brazed. At this point, inspection was required to reject and scrap any gears that shifted during brazing, thus losing the critical tooth-to-tooth relationship required between the small-diameter gear and the large-diameter gear. The rejection rate due to assembly errors ran as high as 50%.

Figure 13(a) shows the single-piece transfer gear manufactured using P/M techniques. The flange section was made thick enough so that the hub section could be machined as a secondary operation. The normal timing of the floating die press mechanism forced powder to transfer from the small gear teeth into the flange section, thus resulting in low density in the small (top) gear teeth. Consequently, tooling was modified to produce the gear configurations shown in Fig. 13(b), which required no machining of the gear

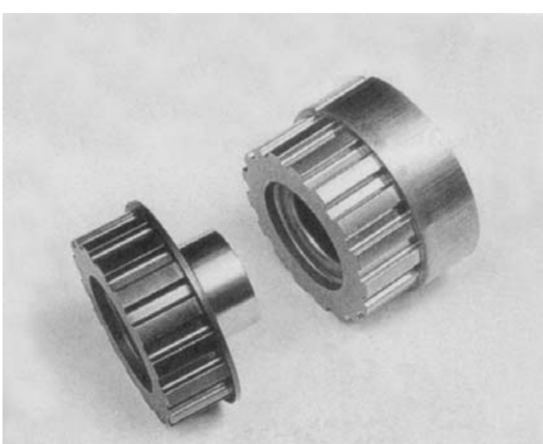
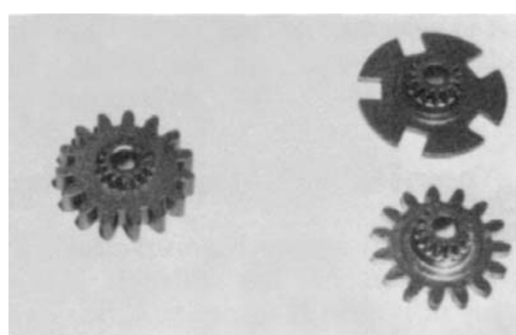


Fig. 12 Iron-carbon P/M drive gear. (Left) Finished machined, hardened, ground part. The hub diameter is machined oversized (+0.254 mm, or +0.010 in.) to allow for final grinding. (Right) P/M blank as-pressed



(a) (b)

Fig. 13 P/M transfer gear made of high-strength low-alloy steel. (a) Original P/M processing technique, which required machining of flange section. (b) Modified P/M technique, which required no additional machining teeth located on the hub section. Typical finished P/M properties are:

Ultimate tensile strength	586–793 MPa (85–115 ksi)
Yield strength	552–655 MPa (80–95 ksi)
Elongation	<1%
Transverse-rupture strength	1172–1379 MPa (170–200 ksi)

Example 3: Eccentric Gear. Figure 14 shows an eccentric gear for a washing machine that is made of a blend of 2% copper, 0.5% graphite, and the balance iron, which is compacted, sintered, sized, and machined. An intermediate level of graphite was chosen to produce an as-sintered part that could be sized readily to ensure an accurate gear tooth profile. Lower carbon content also improves machinability. Density of this part is 6.5 g/cm^3.

This gear is made from atomized powder to obtain optimum compressibility. Because this gear is a two-level part, green strength is critical to eliminate cracking at juncture. Press motion and tool design also are critical in preventing hub cracking.

Sintering is done in an endothermic atmosphere for 30 min at 1105 °C (2025 °F). Dew point is monitored to prevent carburization of the surface layers so as not to affect sizing or machining. After sintering, the part is deburred in a steel medium to facilitate sizing. Prior to sizing, the part is dipped in soluble sizing wax, which also acts as a long-term storage and rust protection medium.

Sizing enhances the tooth form and pitch diameter tolerances. After sizing, the center hole is bored for concentricity, and the outside of the hub is machined to correct any ovality. This part replaced a class 30 gray iron casting that was machined from a rough blank.

Example 4: Fourth Reduction Gears. Lawn and garden tractors require gears that can withstand extremely high wear and heavy loading. For example, the fourth reduction gear shown in Fig. 15 is a drive gear in a six-speed transmission. A crush load of 4000 kg (8800 lb) per tooth is specified. High tooth density is required of the P/M part. To provide this feature, repressing the part to a density of 7.3 to 7.5 g/cm^3 would seem to be the solution. However, the part is too large to be compacted on a 4.45 MN (500 ton) press. To overcome this limitation, the part is compacted to three densities: 6.4 to 6.6 g/cm^3 in the hub, 6.6 to 6.8 g/cm^3 in the inner flange, and 6.9 to 7.0 g/cm^3 in the tooth area.

After presintering, the gear is repressed only in the tooth area to a density of 7.3 to 7.5 g/cm^3. Carbon content is kept low to improve compressibility during repressing, after which the teeth are carburized.

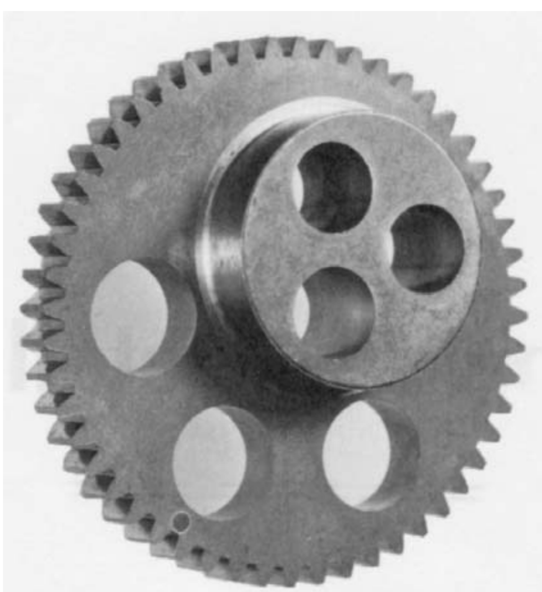
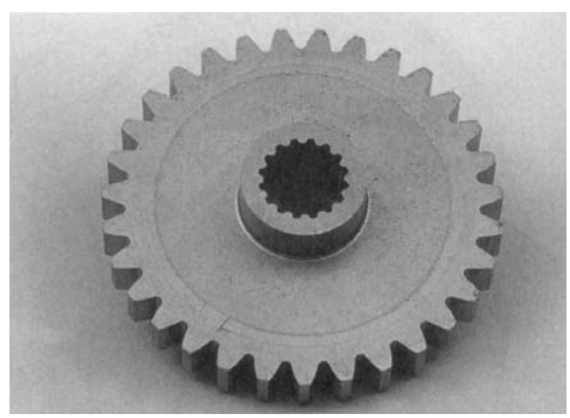


Fig. 14 Eccentric gear for a washing machine made of iron alloy powder containing 2% copper and 0.5% graphite. Center hole inside diameter: approximately 19 mm (0.76 in.); eccentric outside diameter: approximately 57 mm (2.28 in.); total thickness: approximately 28.50 mm (1.14 in.); eccentric thickness: approximately 15.90 mm (0.63 in.); pitch diameter: approximately 115 mm (4.59 in.)



MPIF material	FN-0205
Carbon, %	0.5
Nickel, %	2
Molybdenum, %	0.5
Density, g/cm^3	
Tooth	7.3–7.5
Inner flange	6.6–6.8
Hub	6.4–6.6
Tooth apparent hardness, HRC	50–56
Tooth crush strength, kN (ton)	40 (4.5)
Finished weight, kg (lb)	1.042 (2.300)

Fig. 15 Fourth reduction gear

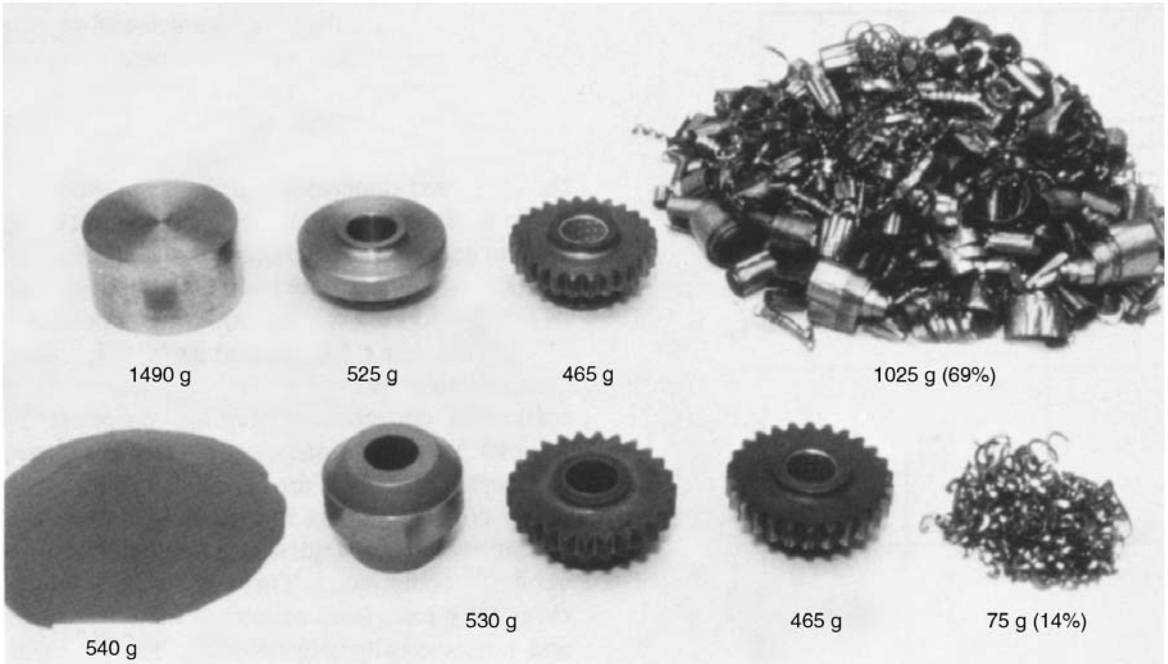


Fig. 16 Comparison of material used for a conventionally forged reverse idler gear (top) and the equivalent powder forged part (bottom). Material yield in conventional forging is 31%; that for powder forging is 86%. 1 lb = 453.6 g.

Example 5: Powder Forged Internal Ring Gear. Powder forging is a process in which unsintered, presintered, or sintered powder metal preforms are hot formed in confined dies. One of the main economic benefits of powder forging is the reduced amount of machining required, as illustrated in Fig. 16.

The powder forged internal ring gear shown in Fig. 17 is used in automatic transmissions for trucks with a maximum gross vehicle weight of 22,700 kg (50,000 lb). The gear transmits 1355 N · m (1000 ft · lbf) of torque through the gear and spline teeth.

Originally, the gear was produced by forging an AISI 5140M tubing blank. The conventionally forged blank required rough machining, gear tooth shaping, spline machining, core heat treating, carburizing, and deburring. The only secondary operations required on the powder forged part are surface grinding, hard turning, shot blasting, and vibratory tumbling.

The powder forged 4618 ring gear is produced to a minimum density of 7.82 g/cm^3. The part is selectively carburized using a proprietary process and quench hardened. Minimum surface hardness is 57 HRC (2070 MPa, or 300 ksi, ultimate tensile strength), while the core hardness is 25 HRC (825 MPa, or 120 ksi, ultimate tensile strength). The internal gear teeth are produced to AGMA Class 7 tolerances.

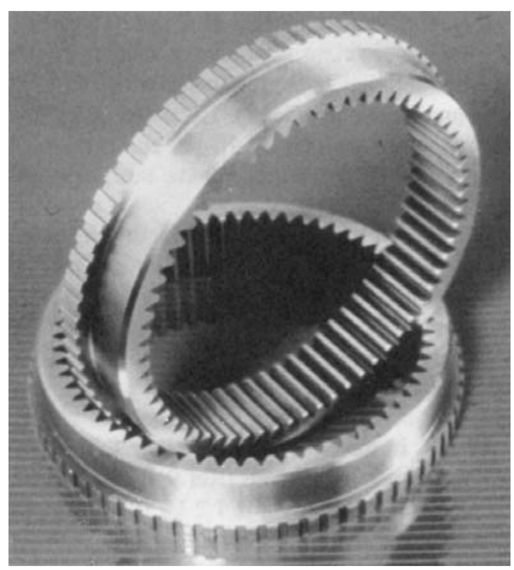


Fig. 17 Powder forged internal ring gear used in automatic transmission for trucks of up to 22,700 kg (50,000 lb) gross vehicle weight. Courtesy of Precision Forged Products Division, Borg Warner Corp.

ACKNOWLEDGMENTS

This chapter was adapted from:

- G.T. Shturtz, Powder Metallurgy Gears, *Powder Metal Technologies and Applications,* Vol 7, *ASM Handbook,* ASM International, 1998, p 1058–1064
- Appendix 3: Examples of Powder Metallurgy Parts, *Powder Metal Technologies and Applications,* Vol 7, *ASM Handbook,* ASM International, 1998, p 1101–1108
- W.B. James, M.J. McDermott, and R.A. Powell, Powder Forged Steel, *Powder Metal Technologies and Applications,* Vol 7, *ASM Handbook*, ASM International, 1998, p 803–827

REFERENCES

1. P.W. Lee, Press and Sintered Parts and Their Applications, *Powder Metallurgy Applications Advantages and Limitations,* Erhard Klar, Ed., American Society for Metals, 1983, p 82
2. S.D.K. Saheb and K. Gopinath, Powder Metallurgy Gears—A Brief Review, *Powder Metall. Sci. Technol.,* Vol 1 (No. 4), July 1990, p 43–66
3. S.D.K. Saheb and K. Gopinath, Tooling for Powder Metallurgy Gears, *Powder Metall. Sci. Technol.,* Vol 2 (No. 3), April 1991, p 25–42
4. *Met. Powder Rep.,* Vol 43, 1988, p 536
5. W. Konig, G. Rober, K. Vossen, and M. Stromgren, Powder Forging of Helical Gears for Car Manual Gear Boxes—Concept and Properties, *Met. Powder Rep.,* Vol 45 (No. 4), 1990, p 269–273
6. *Powder Metallurgy Design Manual,* Metal Powder Industries Federation, 1994, p 193
7. G. Parrish and G.S. Harper, *Production Gas Carburizing,* Pergamon, 1985, p 20
8. *Gear Design Manufacturing and Inspection Manual,* SAE No. 841083, Society of Automotive Engineers, 1990, p 46
9. "AGMA Standard for Rating the Strength of Spur Gear Teeth," AGMA 220.02, American Gear Manufacturers' Association, 1966
10. "AGMA Standard for Rating the Strength of Helical and Herringbone Gear Teeth," AGMA 221.02, American Gear Manufacturers' Association, 1965
11. "AGMA Standard for Rating the Strength of Straight Bevel and Zero Bevel Gear Teeth, AGMA 222.02, American Gear Manufacturers' Association, 1964
12. "AGMA Standard for Surface Durability of Spur Gear Teeth," AGMA 210.02, American Gear Manufacturers' Association, 1965
13. "AGMA Standard for Surface Durability of Helical and Herringbone Gear Teeth," AGMA 211.02, American Gear Manufacturers' Association, 1969
14. "AGMA Standard for Surface Durability Formulas for Straight Bevel and Zero Bevel Gear Teeth," AGMA 212.02, American Gear Manufacturers' Association, 1964
15. T. Prucher and H. Sanderow, Surface Fatigue of P/M Alloys, *Characterization, Testing, and Quality Control,* Vol 2, *Advances in Powder Metallurgy and Particulate Materials,* Metal Powder Industries Federation, 1994, p 99–111
16. R. Gnanamoorthy and K. Gopinath, Surface Durability Studies on As-Sintered Low Alloy Steel Gears, *Advances in Powder Metallurgy and Particulate Materials—1992,* Vol 6, Metal Powder Industries Federation, 1992, p 237–250
17. W. Smith, Ferrous Based Powder Metallurgy Gears, *Gear Manufacture and Performance,* American Society for Metals, 1974, p 257–269

SELECTED REFERENCES

- AGMA & MPIF Develop Standards, Information Sheet for Powder Metal Gears, *Gear Technol., J. Gear Manuf.,* Sept/Oct 1996
- R. Moderow, Gear Inspection and Measurement, *Gear Technol., J. Gear Manuf.,* July/Aug 1992
- MPIF Standard 35, *Material Standards for P/M Structural Parts,* Metal Powder Industries Federation, 1997
- *Powder Metallurgy Design Manual,* 2nd ed., Metal Powder Industries Federation, 1995
- *Powder Metallurgy Design Solutions,* Metal Powder Industries Federation, 1993
- H. Sanderow, Powder Metal Gear Design and Inspection, *Gear Technol., J. Gear Manuf.,* Sept/Oct 1996
- G. Shturtz, The Beginner's Guide to Powder Metal Gears, *Gear Technol., J. Gear Manuf.,* Sept/Oct 1995

Gear Materials, Properties, and Manufacture
J.R. Davis, editor, p155-162
DOI:10.1361/gmpm2005p155

CHAPTER 8

Through Hardening

THE THROUGH-HARDENING PROCESS is generally used for gears that do not require high surface hardness. Typical gear tooth hardness following through hardening ranges from 32 to 48 HRC. Most steels used for through-hardened gears have medium carbon content (0.3 to 0.6%) and a relatively low alloy content (up to 3%). The purpose of alloying is to increase hardenability. The higher the hardenability, the deeper is the through hardening of gear teeth. Since strength increases directly with hardness, high hardenability is essential for through hardening steels. High hardenability, again, has some adverse effect on material ductility and impact resistance. The other drawback of through-hardened gears is lower allowable contact stresses than those of surface-hardened gears. This tends to increase the size of through-hardened gears for the same torque capacity compared with those with hardened surfaces.

In through hardening, gears are first heated to a required temperature and then cooled either in the furnace or quenched in air, gas, or liquid. The process may be used before or after the gear teeth are cut. If applied before cutting the teeth, the hardness usually is governed and limited by the most feasible machining process. Since these gear teeth are cut after heat treatment, no further finishing operation is needed. On the other hand, gears that are designed for hardness above the machining limit are first cut to semi-finish dimensions and then through hardened. In case of some minor heat treat distortion, a finishing operation such as lapping or grinding is very often used to improve the quality of these gears—American Gear Manufacturers Association (AGMA) class 10 and above. For quality up to class 9, gears are finished cut at least one AGMA class above the requirement prior to heat treatment.

Four different methods of heat treatment are primarily used for through-hardened gears. In ascending order of achievable hardness, these methods are annealing, normalizing and annealing, normalizing and tempering, and quenching and tempering. Sometimes, hardnesses of through-hardened gears are specified and measured in other scales besides Rockwell, such as Vickers and Brinell. Table 1 shows an approximate relationship among the various commonly used hardness scales.

Through-Hardening Processes

Annealing refers to any heating and cooling operation that is usually applied to induce softening. There are two types of annealing—full and process. In full annealing, the steel is heated usually to approximately 38 °C (100 °F) above the upper critical temperature and held for the desired length of time, followed by very slow cooling in the furnace. The purposes of full annealing are to:

- Soften the steel and improve ductility and machinability
- Relieve internal stresses caused by previous treatment and improve dimensional stability
- Refine the grain structure

In process annealing, the steel is heated to a temperature below or close to the lower critical temperature followed by the desired rate of cooling. The purpose here is to soften the steel partially and to release the internal stresses. In this treatment, grain refinement by phase transformation is not accomplished as it is in full annealing. Process annealing uses temperatures between 550 and 650 °C (1020 and 1200 °F).

Gears with hardness up to 34 HRC are fully annealed by heating to 800 to 900 °C (1475 to

1650 °F) and then furnace cooled to a prescribed temperature, generally below 315 °C (600 °F). Typical hardnesses obtained after full annealing gears of different materials are shown in Table 2.

Normalizing and Annealing. In general, the term *normalizing* refers to the heating of steel to approximately 38 °C (100 °F) above the upper critical temperature, followed by cooling in still air. The normalizing and annealing process is used, either singularly or in a combination, as a grain structure homogenizing for alloy steel gears. The process also is used to reduce metallurgical nonuniformity such as segregated alloy microstructures from previous mechanical working. A hypoeutectoid steel consisting of a structure of ferrite and coarse pearlite may be made easier to machine if the ferrite and cementite are more finely distributed. A very soft steel has a tendency to tear in machining; therefore, some increase in hardening obtained by normalizing leads to a more brittle chip and thus improves machinability. Some through-hardened gears may just require hardness obtained with normalizing and annealing.

Normalizing and Tempering. Normalizing consists of heating gears to 870 to 980 °C (1600 to 1800 °F) and then furnace cooling in still or circulated air. This process results in higher hardness than annealing, with hardness being a function of the grade of steel and gear tooth size. However, normalizing does not increase hardness significantly more than annealing does, regardless of tooth size for plain carbon steels containing up to 0.4% carbon (but it definitely helps to ensure homogeneous microstructure of steels). After normalizing, alloy steel gears are tempered at 540 to 680 °C (1000 to 1250 °F) for uniform hardness and dimensional stability.

Table 1 Approximate relation between various hardness-test scales

	Rockwell								
Brinell(a)	C	A	30-N	15-N	B	30-T	15-T	Vickers pyramid	Tukon (Knoop)
...	70	86.5	86.0	94.0	...	...	...	1076	...
...	65	84.0	82.0	92.0	...	...	...	820	840
...	63	83.0	80.0	91.5	...	...	...	763	790
614	60	81.0	77.5	90.0	...	...	...	695	725
587	58	80.0	75.5	89.3	...	...	...	655	680
547	55	78.5	73.0	88.0	...	...	...	598	620
522	54	77.5	71.0	87.0	...	...	...	562	580
484	50	76.0	68.5	85.5	...	...	...	513	530
460	48	74.5	66.5	84.5	...	...	...	485	500
426	45	73.0	64.0	83.0	...	...	...	446	460
393	52	71.5	61.5	81.5	...	...	...	413	425
352	38	69.5	57.5	79.5	...	...	...	373	390
301	33	67.0	53.0	76.5	...	...	...	323	355
250	24	62.5	45.0	71.5	...	...	...	257	...
230	20	60.5	41.5	69.5	...	...	...	236	...
200	...	...	...	...	93	78.0	91.0	210	...
180	...	...	...	...	89	75.5	89.5	189	...
150	...	...	...	...	80	70.0	86.5	158	...
100	...	...	...	...	56	54.0	79.0	105	...
80	...	...	...	...	47	47.7	75.7	...	...
70	...	...	...	...	34	38.5	71.5	...	...

(a) Load, 3000 kgf; diam, 10 mm (0.4 in.)

Table 2 Typical Brinell hardness ranges of gears after through hardening

	Brinell hardness (HB)				Brinell hardness (HB)		
Material	Annealed; normalized and annealed	Normalized and tempered	Maximum Brinell hardness, quench and tempered	Material	Annealed; normalized and annealed	Normalized and tempered	Maximum Brinell hardness, quench and tempered
4130	155–200	170–215	350	4145	195–240	285–330	450
8630	155–200	170–215	350	4150	195–240	285–330	450
4140	185–230	260–300	425	4340	210–255	300–340	480
4142	185–230	260–300	425	HP 9-4-30	200–240	(a)	520
8640	185–230	260–300	425	Maraging steel	200–240	(a)	485

(a) Process generally is not used with these types of materials.

Quench and Temper. The quench and temper process involves heating the gears to form austenite at 800 to 900 °C (1475 to 1650 °F), followed by quenching in a suitable media such as oil. The rapid cooling causes the gears to become harder and stronger by the formation of martensite. Hardened gears then are tempered at a temperature, generally below 690 °C (1275 °F), to achieve the desired mechanical properties.

Tempering lowers both the hardness and strength of quenched steels but improves materials properties such as ductility, toughness, and impact resistance. The tempering temperature must be carefully selected based on the specified hardness range, the quenched hardness of the part, and the material. Normally, the optimum tempering temperature is the highest temperature possible while maintaining the specified hardness range. It is to be remembered that hardness after tempering varies inversely with the tempering temperature used. After tempering, parts usually are air cooled at room temperature.

Some steels can become brittle and unsuitable for service if tempered in the temperature range of 430 to 650 °C (800 to 1200 °F). This phenomenon is called temper brittleness and generally is considered to be caused by segregation of alloying elements or precipitation of compounds at ferrite and austenite grain boundaries. If the gear materials under consideration must be tempered in this range, investigation to determine their susceptibility to temper brittleness is needed. Molybdenum content of 0.25 to 0.50% has been shown to eliminate temper brittleness in most steels. (Note: Temper brittleness should not be confused with the tempering embrittlement phenomenon that sometimes results from tempering at a lower temperature range, such as 260 to 320 °C, or 500 to 600 °F.)

The major factors of the quench and temper process that influence hardness and material strength are:

- Material chemistry and hardenability
- Quench severity
- Section size
- Time at temper temperature

Of the four commonly used through-hardening processes, the quench and temper method is the most commonly used. This is particularly true when:

- The hardness and mechanical properties required for a given application cannot be achieved by any of the other three processes.
- It is necessary to develop mechanical properties (core properties) in gears that will not be altered by any subsequent heat treatment such as nitriding or induction hardening.

Typical hardness ranges achieved for different materials after through hardening by different processes are listed in Table 2.

Through-Hardened Gear Design

After finalizing a design, a gear designer needs to specify the following information on a through-hardened gear drawing. This information will help to minimize confusion for all involved with gear manufacturing and material procurement:

- Grade of steel with Aerospace Material Specification (AMS), if applicable
- AMS specification for material cleanliness, if required
- Hardnesses on tooth surface and at the core
- AGMA gear quality level

Each hardness callout should have at least a range of 4 points in HRC scale or 40 points in Brinell hardness (HB). Also, specify a tempering temperature range on the drawing. This allows gear manufacturing engineers to select a particular tempering temperature for a specified hardness.

Hardness Measurement

The hardness of through-hardened gears generally is measured either on the gear tooth end face or rim section. This is the hardness that is used for gear rating purposes. Sometimes, achieving specified hardness on the tooth end face may not necessarily assure the desired hardness at the roots of teeth because of the grade of steel, tooth size, and heat treat practice. If gear tooth root hardness is critical to a design, then it should be specified and measured on a sample (coupon) processed with the gears. However, needless increase of material cost by selecting a higher grade of steel should be avoided.

Distortion of Through-Hardened Gears

All steel gears experience distortion during a heat treat process. It is a physical phenomenon and cannot be eliminated from any heat treat operation, although distortion of through-hardened gears is not as severe as in other processes such as carburizing and nitriding. Still, through-hardened gears, particularly quench and tempered class, experience enough distortion that will eventually lower the quality level of gears after heat treatment. This necessitates a finishing operation for higher quality. In general, some materials expand after a through-hardening operation while others contract. This requires a suitable stock allowance to be provided on teeth for finish machining before heat treatment of gears that are likely to distort. The allowance needs to include expansion or contraction of material and also distortion of tooth geometry. For materials with predictable and uniform distortion, gears could be cut to include the distortion so that no finishing operation is required, possibly up to AGMA class 10 gear tooth quality. The majority of steels listed in Table 2 expand during the through-hardening process, whereas a few materials, such as maraging steel, are found to contract. The amount of expansion or contraction depends on alloy content, quality of steel, and configuration of gears. In this regard, knowledge of distortion characteristics is helpful in optimizing the manufacturing process of gears. When gears are made from a material without any previous heat treat distortion data, an experimental investigation is beneficial to establish the distortion characteristics of the material. With such data, cost-effective manufacturing methods can be established. An investigation of this nature carried out by an aerospace company to determine the heat treat distortion characteristics of a through-hardened gear rack for an aerospace application is discussed at the end of this chapter. Table 3 shows a comparative distortion rating of some preferred through-hardening materials for gears.

Applications

Of the four different through-hardening processes described, those gears hardened by quenching and tempering have some limited use in power transmission applications. The other three processes are only employed to either improve machinability or to enhance homogeneous grain structure of the gear steel. Use of through-hardened gears is limited because of the low surface hardness that results in low gear pitting life and low power density gearbox compared with the one made with case-hardened gears. Also, for similar torque capacity, through-hardened gears are larger with higher pitch line velocity. This increases dynamic problems substantially in a gearbox. However, in a bending strength limited design, through-hardened gears sometimes are successfully used, particularly for large gears (over 508 mm, or 20 in., outer diameter) that normally exhibit high distortion if a case hardening process is used. An example for such an application is the internal ring gear of an epicyclic gearbox. These gears are usually designed with hardness in the range of 32 to 34 HRC that can be finish cut after hardening, thus eliminating costly finishing operations. Through-hardened gears also are found to be effective in applications susceptible to gear scuffing. It is claimed that profile conformance of through-hardened gears, because of their low surface hardness, reduces sliding friction and thereby helps to increase scuffing resistance.

Overall, through-hardened gears are used in gearboxes that require large gears that cannot be economically case hardened, such as large marine propulsion gears and railway power transmission gears.

Case History: Design and Manufacture of a Rack

As explained in the section "Distortion of Through-Hardened Gears," all steel gears distort after any type of heat treat process. Carburizing imparts the highest distortion, while through hardening imparts the least distortion. Even then, distorted gears require a finishing operation for higher tooth quality. Sometimes,

Table 3 Distortion ratings of through-hardened gears

Material	AMS specification	AMS quality	Hardness (HRC)	Distortion rating
AISI 4340	6414	2300	48/50	Good(a)
Maraging 250	6520	2300	49/52	Predictable and good(a)
Maraging 300	6521/6514	2300	52/56	Predictable and good(a)
AISI 4140	6382	2300	48/50	Good(a)

(a) Within one AGMA class of gear quality.

for through-hardened gears, the knowledge of distortion characteristics may be included in the design of gear cutting tools such that gears after heat treatment meet the desired quality. Such a case history is presented here.

The project was to develop a low-cost, high-bending strength (minimum of 250 ksi, or 1720 MPa, ultimate tensile strength) corrosion-resistant rack. For this application, the quality required was rack teeth of AGMA class 9. To minimize manufacturing cost, it was decided not to consider any post-heat-treat finishing operation. To meet these criteria, selection of a proper material and a process was vital, for which the following investigation was carried out. The dimensions and configuration of the rack are shown in Fig. 1.

Material Selection

Table 4 lists the chemical compositions of various materials considered in this case history. The positive and negative attributes of each and the associated heat treat process considered before rack material selection are described subsequently.

Option 1: Use of Quench-Hardening Steels. The following steels were considered:

- AISI 4340
- 300M

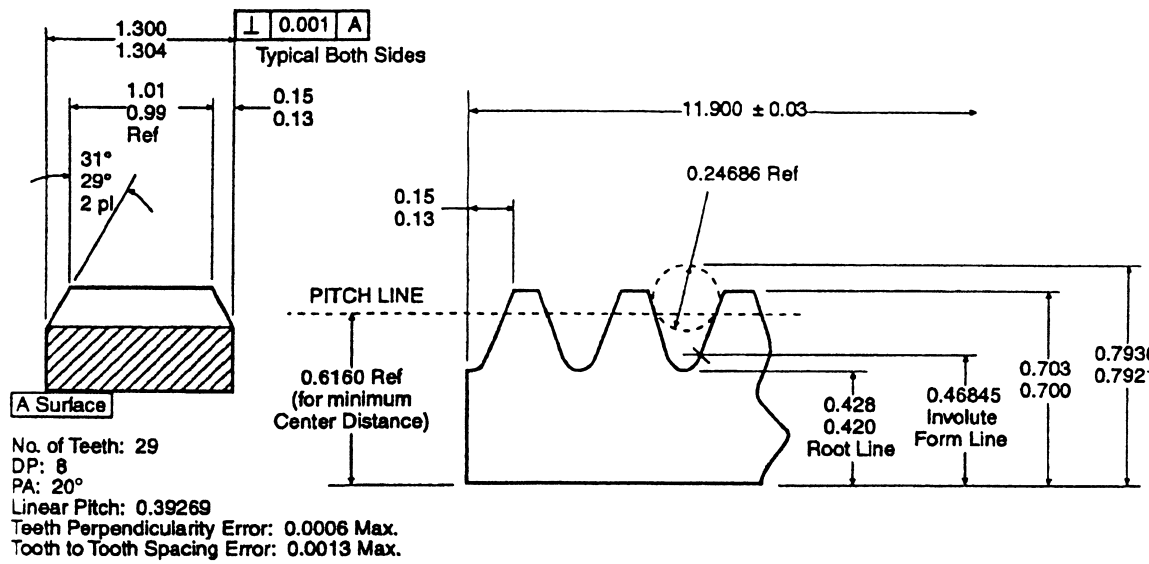


Fig. 1 Rack dimensions (given in inches) for preliminary tests. DP, diametral pitch; PA, pressure angle

Table 4 Chemical compositions of steels considered for racks

Material	C	Mn	Si	P	S	Ni	Cr	Mo	V	Co	W	Cu
Quench-hardening steels												
AISI 4340	...	0.60/0.80	0.20/0.35	...	...	1.65/2.00	0.70/0.90	0.20/0.30	...	...	...	...
HP 9-4-30	0.30	0.20	0.01	0.005	0.0007	7.50	1.00	1.00	0.08	4.0		
H-11	0.35	...	...	...	...	...	5.00	2.50	0.40	...	...	...
300M	0.42	0.68	1.61	0.005	0.006	1.77	0.85	0.40	0.09	...	...	...
Precipitation-hardening steels												
17-4 PH	...	...	...	...	...	4.00	16.50	...	...	...	...	4.00
13-8 Mo	...	...	...	...	...	8.00	12.80	2.30	...	...	...	...
Age-hardening steels												
Maraging C-250	0.026	0.10	0.11	...	...	18.5	...	4.30	...	7.0	...	...

- HP 9-4-30
- H-11

An excellent survey was made from published literature to determine the various properties of each, with the following results and conclusion:

Results. High heat treat distortion:

- Grinding of rack is necessary after heat treatment to attain the required accuracy of teeth
- High cost of material
- Poor corrosion resistance; additional process needed to make racks corrosion resistant

Conclusion. None of these materials was found suitable for the application.

Option 2: Use of Precipitation Hardening Steels. Steels considered:

- 17-4 PH
- 13-8 Mo

Results were as follows:

- Attainable mechanical properties were not at specified strength level
- Heat treat distortion was not predictable
- Sensitive to grind burns
- Problems with alloy segregation for any post-heat treat finishing, in section size needed

Conclusion. Materials were not suitable.

Option 3: Use of an Age-Hardening Steel (Maraging C-250). This maraging steel has high nickel (18% or more), very low carbon (under 0.03%), and is capable of developing very high tensile and yield strengths by means of an aging process. It is sold in the martensitic state, which, because of the low carbon, is soft enough to be readily machinable. Heating to approximately 480 °C (900 °F) for aging, and cooling in the furnace, causes a change in material microstructure that increases the hardness up to 52 HRC. This meets the required tensile strength.

Results were as follows:

- Published literature indicated distortion of maraging C-250 steel is predictable.
- The material is available as forged, as well as in bar form, to AMS 6412. Sheet or plate stock available to AMS 6420 was not acceptable due to nonuniform distribution of mechanical properties.

Conclusion. Maraging C-250 forgings met the design requirements and were selected for this application.

Process Selection

Heat Treat Distortion of Racks Made of C-250. Although published literature indicated distortion of maraging C-250 material is predictable, it was still necessary to find out how much the distortion would be for a rack tooth used in this application. To determine these characteristics, a preliminary investigation was undertaken with racks made from readily available C-250 of shorter lengths (305 mm, or 12 in., long); longer lengths were not commercially available at the time of this investigation. To expedite the program further, a standard shaper cutter was used to cut the teeth. The racks were then heat treated.

Heat Treatment of Racks. Some preliminary experiments were conducted to select a suitable heat treat furnace. The following furnaces were considered:

- Partial vacuum furnace
- Full vacuum furnace
- Air furnace

Heat treatment in the air furnace was not acceptable due to:

- Oxidation of racks
- Scale removal resulted in size change

Both full and partial vacuum furnaces produced oxidation-free racks.

Conclusion. Full vacuum furnace that ensures oxidation-free parts was selected to determine heat distortion.

Twelve racks were selected for heat treatment. All critical dimensions of the racks were inspected and recorded before heat treatment.

The heat treat procedure consisted of the following steps:

- Vapor degrease racks
- Wipe racks with a cleaning chemical such as acetone
- Select any two racks
- Hold the racks together back to back with nickel-plated bolts—processed horizontally on a flat base in the furnace
- Heat treat racks along with one tensile test bar in each production lot, at 480 ± 6 °C (900 ± 10 °F) for 4.5 h at this temperature
- Furnace cool racks

Inspection consisted of:

- Critical rack dimensions after heat treatment
- Mechanical properties of material such as hardness and tensile strength

Results. From the experimental results and inspection, the following conclusions were made:

- Contraction rate of maraging steel was between 0.0005 and 0.0007 mm/mm (in./in.) and found to be linear and consistent in each lot
- Parts remained flat after the hardening process
- Pitch and accumulative pitch errors were within acceptable limit
- Teeth perpendicularity (lead errors) were between 0.013 and 0.025 mm (0.0005 and 0.001 in.)
- Pitch dimensions measured over a pin were held to 0.038 to 0.076 mm (0.0015 to 0.003 in.)

Recommendations included:

- Design of rack to include 0.076 mm (0.003 in.) tolerance for pitch dimension over the pin
- Tooth perpendicularity (lead) error to 0.025 mm (0.001 in.)
- A shaper cutter to be developed to include 0.0006 mm/mm (in./in.) contraction rate of rack tooth geometry with the expectation that this might eliminate finish processing of racks after heat treatment

New Shaper Cutter. With the proposed contraction rate, a shaper cutter was designed and manufactured by a cutter manufacturing company. Figure 2 shows the dimensions of this special cutter. An investigation was then carried out with full-length racks.

Manufacturing method for full-length racks consisted of the following steps:

- Forge blanks
- Solution anneal to 35 HRC
- Machine (mill and drill), leaving ground stock on sliding surfaces only
- Shape rack teeth to final dimensions—rough, semifinish, and finish
- Straighten to 0.16 mm/m (0.002 in./ft)
- Inspect
- Heat treat (age) to 50 HRC—two pieces bolted back to back
- Clean
- Grind sliding surfaces locating from the pitch line of rack teeth
- Inspect all rack dimensions before and after heat treatment

Test results are shown in Tables 5 and 6. Analysis of the results indicates:

- Part remained flat after heat treatment within 0.25 mm (0.010 in.)
- Pitch and accumulative pitch errors were within the specified tolerance
- Tooth perpendicularity (lead) error was within 0.025 mm (0.001 in.)
- Measurements of pitch line over the pin were within the new tolerance

Conclusions. Racks met the required quality level (AGMA class 9). In this case, the modified cutter and superior heat treat facilities made it

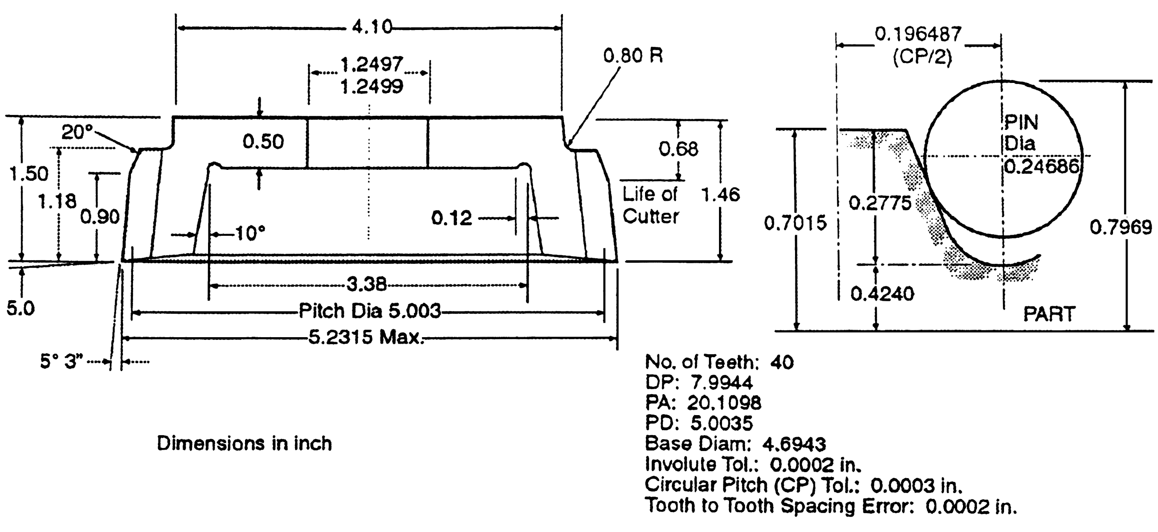


Fig. 2 Modified shaper cutter dimensions

Table 5 Experimental results: full-sized racks dimensions before heat treat

		Rack 1		Rack 2	
Item	Drawing dimension	Left flank	Right flank	Left flank	Right flank
Linear pitch, mm (in.)	9.9746 (0.3927)	9.9720–10 (0.3927–0.3936)	9.977–9.9949 (0.3936–0.3927)	9.9771–9.9949 (0.3928–0.3935)	9.9771–9.9924 (0.3928–0.3934)
Spacing error, mm (in.)	0.0330 (0.0013) max	0.0178 (0.0007) max	0.0229 (0.0009) max	0.0203 (0.0008) max	0.0178 (0.0007) max
Accumulated spacing error, mm (in.)	0.0762 (0.003) max, over any 12 teeth	0.1778 (0.007) max	0.1778 (0.007) max	0.0152 (0.006) max	0.0152 (0.006) max
Tooth perpendicularity (lead error), mm (in.)	0.0127 (0.0005) max	0–0.0533 (0–0.0021)	0–0.0533 (0–0.0021)	0–0.0406 (0–0.0016)	0–0.0406 (0–0.0016)
Dimension over pin, mm (in.)	20.1193–20.1574 (0.7921–0.7936)	20.2946–20.7518 (0.799–0.817)	20.2946–20.7518 (0.799–0.817)	20.2692–20.5994 (0.798–0.811)	20.2692–20.5994 (0.798–0.811)
Surface finish, rms	63	≅45	≅45	≅45	≅45

rms, root mean square

Table 6 Experimental results: full-sized racks dimensions after heat treat and grinding back face

		Rack 1		Rack 2	
Item	Drawing dimension	Left flank	Right flank	Left flank	Right flank
Linear pitch, mm (in.)	9.9746 (0.3927)	9.9670–9.9873 (0.3924–0.3932)	9.9720–9.990 (0.3926–0.3933)	9.9593–9.9873 (0.3921–0.3932)	9.9670–9.9924 (0.3924–0.3934)
Spacing error, mm (in.)	0.0330 (0.0013) max	0.0152 (0.0006) max	0.0152 (0.0006) max	0.0152 (0.0006) max	0.0178 (0.0007) max
Accumulated spacing error, mm (in.)	0.0762 (0.003) max, over any 12 teeth	0.1778 (0.007) max	0.1778 (0.007) max	0.1143 (0.0045) max	0.1143 (0.0045) max
Tooth perpendicularity (lead error), mm (in.)	0.0127 (0.0005) max	0–0.0533 (0–0.0021)	0–0.0533 (0–0.0021)	0.0025–0.0279 (0.0001–0.0011)	0.0025–0.0279 (0.0001–0.0011)
Dimension over pin, mm (in.)	20.1193–20.1574 (0.7921–0.7936)	20.1016–20.1422 (0.7914–0.7930)	20.1016–20.1422 (0.7914–0.7930)	20.1270–20.1955 (0.7924–0.7951)	20.1270–20.1955 (0.7924–0.7951)
Surface finish, rms	63	≅45	≅45	≅45	≅45

rms: root mean square

possible to manufacture the racks to AGMA class 9 without any subsequent finishing operation such as grinding.

Experiments of this nature are definitely useful in determining the distortion of heat treated gears and planning for subsequent finishing operation, if needed. When the quantity of gears to be produced is limited, and the allocated production development time is short, there may not be many choices other than to finish the gear after heat treatment. For low distortion, honing is useful, while grinding is necessary for large distortion. However, grinding after through hardening is not recommended. Some manufacturers, instead of honing a gear and the mating pinion individually, lap the gear and pinion together with a slurry of fine abrasive compound in the mesh until the desired quality is obtained.

In general, gears designed and manufactured to AGMA class 7 and below do not require any such process development due to the fact that the hobbing or shaping process can produce gears to AGMA class 8 and above. It is expected that gears so produced will meet AGMA class 7 after heat treatment.

ACKNOWLEDGMENT

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SELECTED REFERENCES

- C.E. Bates, G.E. Totten, and R.L. Brennan, Quenching of Steel, *Heat Treating,* Vol 4, *ASM Handbook,* ASM International, 1991, p 67–120
- B.L. Bramfitt and A.K. Hingwe, Annealing of Steel, *Heat Treating,* Vol 4, *ASM Handbook,* ASM International, 1991, p 42–55
- T. Ruglic, Normalizing of Steel, *Heat Treating,* Vol 4, *ASM Handbook,* ASM International, 1991, p 35–41
- M. Schmidt and K. Rohrbach, Heat Treating of Maraging Steels, *Heat Treating,* Vol 4, *ASM Handbook,* ASM International, 1991, p 219–228
- M. Wisti and M. Hingwe, Tempering of Steel, *Heat Treating,* Vol 4, *ASM Handbook,* ASM International, 1991, p 121–136

Gear Materials, Properties, and Manufacture
J.R. Davis, editor, p163-226
DOI:10.1361/gmpm2005p163

CHAPTER 9

Carburizing

CARBURIZING is a process in which austenitized ferrous metal is brought into contact with an environment of sufficient carbon potential to cause absorption of carbon at the surface and, by diffusion, create a carbon concentration gradient between the surface and interior of the metal. The depth of penetration of carbon is dependent on temperature, time at temperature, and the composition of the carburizing agent. As a rough approximation, a carburized depth of approximately 0.76 to 1.3 mm (0.030 to 0.050 in.) on a 6 diametral pitch (DP) gear tooth can be obtained in about 4 h at 930 °C (1700 °F) with a carburizing agent, which may be solid, liquid, or gas.

The primary objective of carburizing and hardening gears is to secure a hard case and a relatively soft but tough core. For this process, low-carbon steels (up to a maximum of approximately 0.30% carbon), either with or without alloying elements (nickel, chromium, manganese, molybdenum), normally are used. After case carburizing, the gear teeth will have high carbon at the surface graduating into the low-carbon core.

Sometimes to prevent through hardening of the tooth tip, carbon penetration through tip of tooth needs to be controlled. This is accomplished by plating or spraying the outside diameter of gear before cutting the teeth with some material that prevents the passage of the carburizing agent. However, the most widely used method is copper plating. Several proprietary solutions and pastes, which are quite effective in preventing carburization, also are available.

There are five general methods of carburizing, depending on the form of the carburizing medium. These methods are:

- Solid or pack carburizing, employing solid carburizing material
- Liquid carburizing, employing fused baths of carburizing salts
- Gas or atmosphere carburizing, employing suitable hydrocarbon gases
- Vacuum carburizing, employing a rough vacuum and a partial pressure of hydrocarbon gas
- Plasma or ion carburizing, employing both vacuum and glow-discharge technology (carbon-bearing ions are introduced to the surface for subsequent diffusion below the surface)

Today, gas (atmosphere) carburizing is considered the *de facto* standard by which all other surface hardening techniques are measured and will be the emphasis of this chapter. However, more recently introduced carburizing methods such as vacuum carburizing are rapidly gaining acceptance in the gear industry. The final section of this chapter reviews vacuum carburizing and compares the attributes of conventional gas carburizing and vacuum carburizing. Figure 1 compares the use (market breakdown) of various carburizing methods.

Gas Carburizing: Processing and Equipment

As shown in Fig. 1(a), it is estimated that 90% of gear carburizing is performed in a carbonaceous gas atmosphere. In this type of carburizing, carbon is induced into the ferrous base material heated in the gaseous atmosphere with a carbon potential that allows the surface to absorb carbon. The most commonly used medium is endothermic (commonly known as "endo") gas produced by reacting natural gas (mainly methane, CH_4) with air (1:2.5 to 2.7

ratio) over a heated catalyst. The important chemical reactions that take place can be expressed as:

$$2CH_4 + O_2 \rightarrow 2CO + 4H_2 \qquad \text{(Eq 1)}$$

$$2CO \rightarrow C + CO_2 \qquad \text{(Eq 2)}$$

Varying the ratio of methane to air alters the composition of endo and the chemical reactions slightly.

Free carbon resulting from chemical reaction is then dissolved in the austenite that is formed when gears are heated above 720 °C (1330 °F) and precipitates as iron carbide (Fe_3C).

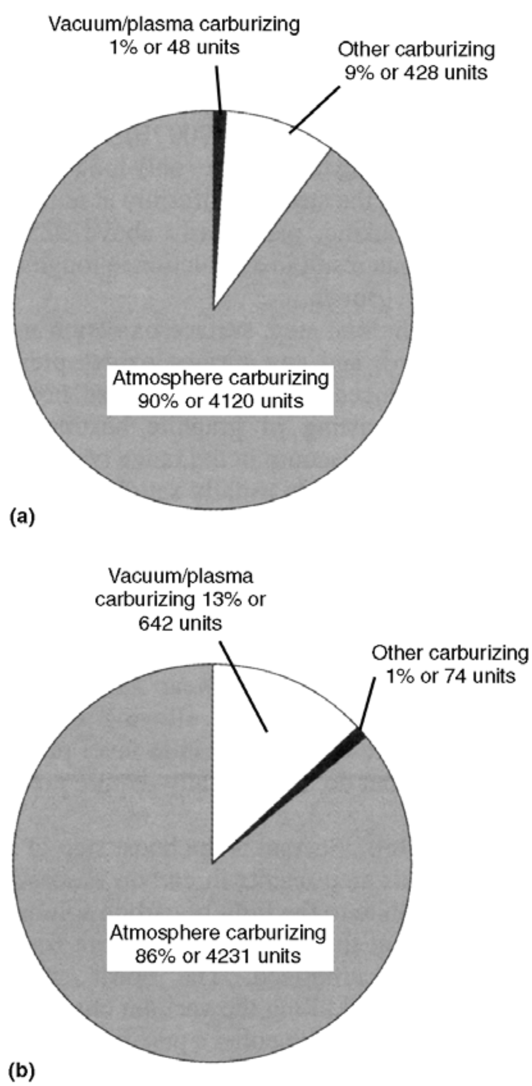

Fig. 1 North American carburizing market. (a) Market in 2000. (b) Anticipated market in 2010. Source: Ref 1

Recently, nitrogen-methanol has been used for supplying the carburizing atmosphere. This type of system offers a number of benefits over the conventional endo gas generator. Here, N_2 plays the most important function to keep air out of the furnace and prevents the gears from being oxidized. Typically, N_2 is more than 90% of the nitrogen-methanol atmosphere. Carburizing in nitrogen-methanol systems also ensures accurate carbon potential for improved carburized case properties. Because of higher cost with nitrogen methanol system, most of the gears are still carburized in endo gas.

Carburizing Temperature

The penetration of carbon into the steel depends on the carburizing temperature, the time at temperature, and the carburizing agent. Since the solubility of carbon is greatest above the Ac_3 temperature, carburization takes place most readily above this temperature. Furthermore, the higher the temperature is, the greater the rate of carbon penetration will be, since the rate of diffusion is greater. It is thus customary to select a temperature approximately 55 °C (100 °F) above the Ac_3 point. Again, the time at the carburizing temperature is the most influential factor in the control of the depth of carbon penetration as illustrated in Fig. 2.

Temperatures as low as 790 °C (1450 °F) and as high as 985 °C (1800 °F) have been used for carburizing gears, although it should be kept in mind that the life of a furnace deteriorates rapidly above 955 °C (1750 °F). With the desired amount of carbon absorbed into the tooth sur-

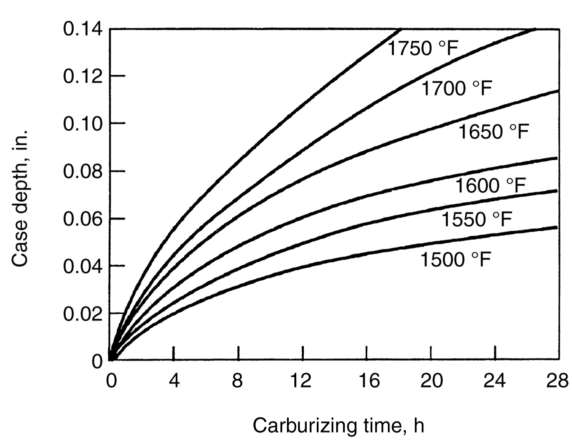

Fig. 2 Depth of carbon penetration for different times and different temperatures in gas carburizing a gear steel

face, gears are quenched in a suitable medium (generally oil) to obtain the required case hardness. Quenching may be performed either directly from the carburizing temperature or from a somewhat lower temperature. In some instances, parts after carburizing are completely cooled to room temperature, reheated to the austenitizing temperatures, and then quenched. As already mentioned, the depth of case is dependent on time and temperature selected during carburizing. The following equation generally satisfies the relationship between the case depth and carburizing time:

$$d = \varphi\sqrt{t}\ldots \qquad \text{(Eq 3)}$$

where d is the total case depth in inches, t is the carburizing time in hours at temperature with saturated austenite at the surface, and φ is the proportionality factor of material that varies with the carburizing temperature.

For low carbon and alloy steels, the value of φ is found to be approximately 0.025 for gas carburizing at approximately 930 °C (1700 °F). Thus, Eq 3 can be rewritten for most alloy steels as:

$$d = 0.025\sqrt{t}\ldots \qquad \text{(Eq 4)}$$

Another relationship used to determine d is:

$$d = \frac{31.6\sqrt{t}}{10\left(\dfrac{6700}{T + 460}\right)} \qquad \text{(Eq 5)}$$

where T is the carburizing temperature in °F. Both Eq (4) and (5) are used in industry and provide satisfactory results.

Besides time and temperature, the quality of case also depends on the type of carburizing furnace and equipment used. A proper selection is thus essential for successful carburizing and hardening.

Furnaces and Equipment for Gas Carburizing

There are three basic types of furnaces used for gas carburizing: atmospheric, vacuum, and fluidized bed. Each of these has its own advantages and disadvantages, although a great majority of gears are gas carburized in atmospheric furnaces because gas carburizing seems to offer acceptable control of case depth, surface carbon content, and diffusion of carbon into steel. The gas mixture may be adjusted to provide either a carburizing or neutral atmosphere, making it possible to diffuse the carbon in the case without the further addition of surface carbon.

Atmospheric Furnaces. Two types of atmospheric furnaces are used: batch and continuous. The fundamental difference between these, aside from size, is the method by which the work is handled. With a batch furnace, as the name implies, the workload is charged and discharged as a single unit or batch. With continuous furnaces, the load is fed on a continuous basis at the charged end and is received at the discharge end after processing. In a batch furnace, changes in composition of carburizing agent take place in both environment and workpieces until equilibrium is reached. In continuous furnaces, the heating chamber is divided into zones. These zones may be separated by internal doors. The atmospheres in each zone can be controlled to different carbon potentials. Coupled with different temperatures in each zone, atmosphere control provides the main means of controlling the carburizing and diffusion portion of the carburizing cycle. The chief advantage of a batch furnace is its adaptability to a variety of cycles. Each batch usually consists of several individual gears. Very large gears may be carburized one at a time; small gears may be loaded several hundred to a batch. One of the major disadvantages of this furnace is that gears are transferred from ambient temperature into a furnace usually operating at the carburizing temperature, causing thermal shock that may lead to uncontrolled distortion.

Vacuum and Fluidized Bed Furnaces. Besides atmospheric-type furnaces, carburizing also may be performed in vacuum furnaces or fluidized bed furnaces. In general, heat transfer characteristics of these furnaces are superior to atmospheric-type furnaces, assuring better case properties and uniformity of case. But the operating cost of these furnaces is higher than atmospheric furnaces. Hence, such furnaces have more limited use.

Carburizing in a vacuum furnace is a relatively new process. Any carburizing done in an atmospheric furnace also can be accomplished in a vacuum furnace. However, vacuum carburizing offers some significant benefits in time, cost, and quality. A major reduction of time is in the heating-up phase of the process, where the parts are loaded into a cold furnace; parts and furnace are then heated up together. Furthermore, the process allows a stepped heat-up mode, which has also been found beneficial. As opposed to conventional carburizing in atmospheric furnaces, where the carburizing medium is adjusted to generate a near eutectoid surface composition, vacuum carburizing is based on a supersaturated carbon reaction.

Selecting the most suitable alloy for vacuum carburizing is very important for its success. Correct quench rates, commensurate with the alloy selection, offer a less drastic quench mode for fully hardened case and core, which atmospheric carburizing and quenching does not allow. This also results in reduced distortion of parts. The major advantage of vacuum carburizing is that it offers better control of case depth even at the root fillet of the gear tooth.

Carburizing in a fluidized bed furnace is very similar to vacuum carburizing, in that supersaturation through direct reaction with the carburizing media (natural gas, methane or propane) can take place at the face to control the surface carbon level, together with the controlled case depth.

Quenching and Hardening

After carburizing, gears are quenched in a cooling medium for hardening. Quenching develops a martensitic or a bainitic case with core microstructures other than a mixture of proeutectoid ferrite and pearlite. Thus, the selection of a proper quenchant is of utmost importance, and the cooling rate, ideally, should be just fast enough to produce the desired core structure but not so fast that the case cracks or that an undue amount of austenite is retained. For industrial and automotive gears, however, quenching conditions often are chosen solely on the basis of developing required surface hardness, especially in applications where the core properties are known to have little or no effect on product performance.

Depending on part size and shape, and on transformation characteristics of the steel, gears may be quenched in water, oil, or any proprietary fluids. Most often oil is used because it is a suitable quenchant for most carburizing grades of steel, especially for relatively fine-pitch gears. Small DP gears may require a more drastic quench, particularly if densely packed, and often are susceptible to quench cracks. Regardless of the type of quenchant used, good circulation within the quench bath is extremely important to promote uniform cooling of gears.

Sometimes, for some materials, the desired properties of the case can be developed without resorting to a liquid quench, in which case air cooling, furnace cooling, or gas cooling may be appropriate. Any of these three cooling media can be used when parts are to be reheated for hardening. If intermediate operations such as straightening or machining are needed prior to hardening, furnace or gas cooling is preferred. Both are done under a protective atmosphere that keeps the gears clean and free of oxide scale and prevents decarburization of the surface.

Direct Quenching

Most gas-carburized gears are quenched directly after carburizing. Furnace temperature usually is reduced to normal austenitizing temperature (approximately 790 °C, or 1450 °F) prior to quenching. In certain cases, quenching directly from the carburizing temperature also is acceptable provided this does not induce thermal cracks in gears. On the other hand, sometimes gears made of some high-alloy steels (alloy content above 5%), are first cooled in air to room temperature after carburizing and then reheated and quenched for low distortion.

Nevertheless, direct quenching has gained wider acceptance, primarily because of economy and simplicity of the procedure. To minimize carbide network in the case, carburizing above the A_{cm} temperature is suggested. Direct quenching reduces the amount of energy used for heating and eliminates or avoids some of the equipment and operating expense of the hardening operation. Labor costs are reduced, and nicks and other part damage are minimized because the parts are handled less frequently. The use of fine-grain (ASTM 5 and above) steels, which exhibit a relatively uniform response to heat treatment, and the development of equipment and techniques for improved confidence in carbon control have led to wider acceptance of direct quenching as a means of hardening carburized gears made from a great variety of steels.

The fixtures required for adequately supporting and separating individual gears during carburizing also promote uniform direction and velocity of the quenchant movement relative to each part on the fixture. This leads to more consistent metallurgical and dimensional quality. Furthermore, a single direct-quench operation minimizes distortion by bringing about crystallographic phase changes during only one heating and one cooling cycle. Each such phase change results in volume change of grain microstructure and increase in internal stress that may produce substantial dimensional change of a gear.

Both horizontal-batch and pusher-type continuous furnaces are well suited for direct quenching. Continuous furnaces sometimes are designed to remove one part at a time from the reduced-temperature section of the furnace for press quenching with a fixture to control distortion.

The degree of distortion in some gears, for example, varies with case depth and the amount of retained austenite in the case, as well as with alloy and process variables. Good control of carbon-gradient shape, case depth, and surface carbon content are essential for direct quenching of dimensionally sensitive gears. In case the temperature of gears is reduced prior to quenching to minimize thermal shock, carbon content near the surface must be held to below saturation; otherwise, carbides will precipitate. A grain-boundary network of carbides in the case is usually considered to be detrimental to gear life, although slow cooling after carburizing and then reheating before quench is one way to avoid or minimize the development of a carbide network.

In the case of severe quench sometimes required to obtain high core hardness, the shape of the gear section being quenched is of great importance since a combination of thick and thin sections (for example, annulus of epicyclic gearbox) may lead to cracking. Cracking results due to a difference in the rate of cooling of thick and thin sections. The transformation of thicker sections will take place when the thin sections are at a lower temperature. The expansion in the thick sections on transformation will set up very high stress concentrations, which may cause warpage or cracking. If proper core hardness cannot be achieved for a certain gear tooth size, an alternate material needs to be investigated.

Reheating of Carburized Gears and Quenching

Originally, only high-alloy steels were carburized, because of the relatively unsophisticated steel-producing techniques required for alloy additions to yield a steel with uniform response to heat treatment. At that time, carburizing was done only by pack carburizing, which has a different set of material-handling and carbon-control techniques than does gas carburizing. The original carburizing grade steels, which were high in nickel, required a reheat operation after pack carburizing to produce a uniform microstructure in the case. Reheating was also the only effective method of reducing the surface carbon content below saturation. Later, although low-alloy steels were introduced for gears, pack carburizing or crude gas-carburizing techniques still required gear reheating to control surface carbon and microstructure. Today, the modern carburizing equipment is capable of producing the desired microstructure in the case of both high- and low-alloy steels. Even then, certain types of gear steels are still reheated before quenching to ensure the quality of case microstructure and low distortion.

Gears requiring individual quenching in a fixture sometimes are reheated as a practical means of confining the tedious one-at-a-time hardening operation to a few simple hardening furnaces, while a larger, more-expensive continuous carburizing furnace is permitted to operate at its maximum capacity for direct quench. Sometimes, the total cost including labor and fuel can be lower for a carburize, cool, and reheat procedure than for direct quenching. Also, this technique of carburizing, followed by slow cooling, machining certain areas to remove the case, and then hardening the entire gear sometimes is used when selected areas must be free of a carburized hardened case.

Furthermore, reheating of gears occasionally is specified for "grain refinement." However, there is considerable disagreement over the advantage of a reheated microstructure over a direct-quenched microstructure, and whether the former is preferred because of tradition, or because of a real need, is unknown. The amount of retained austenite is usually lower, or at least is less visible in microstructures of reheated gears. Again, the effect of retained austenite on gear performance is controversial, but clearly it is not always detrimental. Also, if reheating is used, there is always the danger of greater thermal distortion. The choice between direct quenching, and reheating and quenching should often be decided on the basis of a specific application. Reheating generally is recommended only after high-temperature carburizing (above 930 °C, or 1700 °F).

Surface Hardness Variations after Quenching

Variation in the surface hardness of gears within a lot is a problem that is often encountered when many small gears are heat treated in the same basket. This variation is due to the parts being too densely loaded, especially at the center of the load. This restricts the flow of quenchant in such a manner that gears near the basket's perimeter may attain full surface hardness while those in the center do not. If it is not possible to space out the gears for economic reasons, at least divider screens should be inserted to "layer" the load.

Another possible cause of gear surface hardness variation in a lot is insufficient quench bath agitation. The obvious solution is to speed up the circulation system of the quench tank in order to move more quenchant through the load faster. Approximately 230 to 260 L/min (60 to 70 gal/min) of gear is considered an ideal rate. This establishes the rate of cooling for ideal martensitic transformation of most gear steels.

Tempering of Carburized and Quenched Gears

Tempering is a process of reheating quench-hardened gears to a temperature below the transformation range of steel and holding at this temperature to reduce thermal stresses induced during quenching and improve dimensional stability. Normal tempering temperature for carburized and quenched gears varies between 115 and 175 °C (240 and 350 °F). The surface hardness of quenched gears decreases as the tempering temperature increases as shown in Fig. 3. In addition, tempering temperature has a significant effect on core hardness, as illustrated in Fig. 4. Furthermore, higher tempering temperatures reduce both case hardness and case depth. In applications where gears are required to maintain high compressive and bending strengths at an elevated temperature, carburizing steels that are least affected by tempering temperature are preferred.

To enhance the effect of tempering, it should follow soon after the quench but not until the gears can be comfortably touched with bare hands. Tempering too early can cause serious problems by interrupting the martensitic transformation. To the other extreme, too long a delay before tempering might create a major distortion problem and even cracking of the gears.

Tempering is a necessary finishing treatment after hardening. However, it also involves heating and cooling. This may again generate new stresses in the gears being processed. Fortunately, the influence of these new stresses on geometric shapes of gears is very small due to low temperature levels involved. Nevertheless, uniform heating and cooling is advisable during tempering to keep distortion-causing stresses at a minimum.

Some controversy still exists concerning the value of tempering carburized and quenched gears. For critical applications, experience has proved that tempering is definitely beneficial. Carburized and hardened gears used in aerospace applications invariably need to be tem-

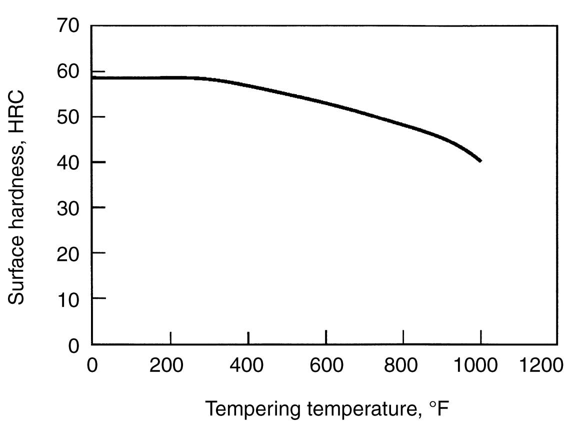

Fig. 3 Variation of tooth surface hardness with tempering temperature of carburized and hardened AISI 8620H gears

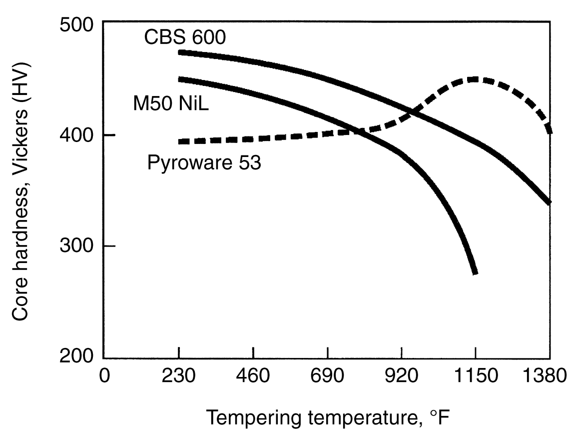

Fig. 4 Influence of tempering temperature on core hardness of some high alloy steels

pered. The reasoning is that tempering is not harmful and provides some benefit to resist cracking or chipping of gears under edge loading. However, in thousands of other less critical applications, it is difficult or sometimes impossible to prove the need of a tempering operation for carburized and hardened gears.

Recarburizing

Occasionally after carburizing and hardening, gears in a certain lot are found to have lower surface carbon and case depth even when all the furnace carburizing parameters are kept the same. To salvage these under-carburized gears by recarburizing sometimes creates a number of potential problems. For example:

- Every time a gear is heated, there is more distortion.
- If a hardened gear is charged into a hot furnace, it might crack.
- If the carburized case depth is shallow, carburizing the second time will increase case carbon content.

This also can result in excessive retained austenite or an undesirable carbide network. Thus, all scenarios are to be considered before recarburizing gears.

Cold Treatment

The presence of retained austenite in a heat treated case can be the source of dimensional instability, excessive residual stress, or cracking, all of which may cause service problems with a carburized gear. Retained austenite is more prominent at high surface carbon and alloy contents, and in cases where gears have been direct quenched from high carburizing temperatures rather than cooled, reheated, and quenched. High-carbon and high-alloy contents in steels depress the temperatures at which martensitic transformation begins and ends. In some instances, the transformation temperature may be well below room temperature, which favors the retention of large amounts of austenite.

One way of reducing the amount of retained austenite in the case microstructure is to cold treat a gear following quenching. Cold treatment is basically a continuation of quenching, during which retained austenite in the case is transformed to martensite. The percentage of transformation is related to temperature rather than to time at a temperature—lower temperatures yield higher levels of transformation. Multiple treatments produce diminishing improvement. Almost all of the significant transformation is achieved by the first cold treatment.

The specific amount of martensitic transformation achieved by a given subzero treatment is extremely difficult to predict. The degree of reluctance to transform at a given temperature is influenced by:

- The amount of retained austenite at the start of cold treatment
- The elapsed time between quenching and cold treating
- Any intermediate thermal treatment, such as tempering
- The general level of residual compressive stress in the part
- Any cold working of the material, such as straightening of long slender pinions after carburizing and quenching

Temperatures in the range –75 to –100 °C (–100 to –150 °F) are routinely used in cold treating. Equipment for cold treatment up to –75 °C (–100 °F) can be as simple as dry ice mixed with kerosene, trichloroethylene, or alcohol in a bucket. Temperatures down to –100 °C (–150 °F) is reached with relatively simple mechanical refrigeration. Liquid nitrogen can be used for chilling to any temperature down to –195 °C (–320 °F) but is seldom used.

Selection of Materials for Carburized Gears

There is a wide variety of carburizing grade materials that offer different mechanical properties (Table 1). Heat treat data for some of these steels are given in Table 2. In general, a material that can attain a tooth surface hardness around 60 HRC and a core hardness between 32 and 48 HRC after carburizing and hardening is selected. The choice determines the surface case and core hardnesses. A low tooth surface hardness reduces the pitting life of gears. On the other hand, low core hardness reduces bending fatigue life. As already discussed, the surface hardness of a gear tooth is strictly dependent on the percentage of carbon at the surface, whereas the core hardness is related to the hardenability of the material and is determined by the alloying elements in the steel. It is thus essential to have a clear understanding of the terms *hardness* and *hardenability* of gear materials. Metallurgically, these two terms are quite different from one another as discussed subsequently.

Hardness and Hardenability

Hardness is a surface property, whereas hardenability of steels refers to depth and hardness distribution induced by quenching. Steels with low hardenability can be hardened to a relatively shallow depth. In these types of steels, austenite-martensite transformation takes place in the area close to the surface only. The center of heat treated section remains soft or transforms to a structure softer than martensite, such as pearlite. Steels with high hardenability develop a much deeper martensitic structure when similarly treated.

As explained previously, the higher the surface carbon is, the higher the surface hardness will be until surface carbon reaches around 0.6%. To increase hardenability of steels the chemical compositions of steels need to be altered to slow down martensitic transformation. This is achieved by alloying the steels. All alloying elements, except cobalt, increase hardenability of steels. The common method to determine hardenability is by the Jominy, or end-quench, test where a test bar of the steel, 25 mm (1 in.) in diameter by 100 mm (4 in.) long, normalized and machined to remove the decarburized surface, is heated to the hardening temperature for 30 minutes. It is then quickly transferred to a fixture that holds the bar in a vertical position, and a jet of water under controlled conditions is directed immediately against the bottom end only (Fig. 5a). The end of the bar is thus cooled very

Table 1 Chemical compositions of frequently used carburizing grade gear steels

	Composition, wt%										
Material	**C**	**Mn**	**P**	**S**	**Si**	**Ni**	**Cr**	**Mo**	**V**	**W**	**Co**
AISI grades											
3310	0.08–0.13	0.45–0.60	0.025 max.	0.025 max.	0.20–0.35	3.25–3.75	1.40–1.75	. . .	. . .	. . .	. . .
3310H	0.07–0.13	0.30–0.70	0.035 max.	0.040 max.	0.20–0.35	3.20–3.80	1.30–1.80	. . .	. . .	. . .	. . .
4023	0.20–0.25	0.70–0.90	0.035 max.	0.040 max.	. . .	. . .	. . .	0.20–0.30	. . .	. . .	. . .
4027	0.25–0.30	0.70–0.90	0.035 max.	0.040 max.	. . .	. . .	. . .	0.20–0.30	. . .	. . .	. . .
4118	0.18–0.23	0.70–0.90	0.035 max.	0.040 max.	0.20–0.35	. . .	0.40–0.60	0.08–0.15	. . .	. . .	. . .
4118H	0.17–0.23	0.60–1.00	0.035 max.	0.040 max.	0.20–0.35	. . .	0.30–0.70	0.08–0.15	. . .	. . .	. . .
4140	0.38–0.43	0.75–1.00	0.035 max.	0.040 max.	0.20–0.35	. . .	0.80–1.10	0.15–0.25	. . .	. . .	. . .
4320	0.17–0.22	0.45–0.64	0.035 max.	0.040 max.	0.20–0.35	1.65–2.00	0.40–0.60	0.20–0.30	. . .	. . .	. . .
4330M	0.30–0.33	0.96	0.008 max.	0.006	0.30	1.82	0.88	0.44	0.09	. . .	. . .
4620	0.17–0.22	0.45–0.65	0.035 max.	0.040 max.	0.20–0.35	1.65–2.00	. . .	0.20–0.30	. . .	. . .	. . .
4815	0.13–0.18	0.40–0.60	0.035 max.	0.040 max.	0.20–0.35	3.25–3.75	. . .	0.20–0.30	. . .	. . .	. . .
4820	0.18–0.23	0.50–0.70	0.035 max.	0.040 max.	0.20–0.35	3.25–3.75	. . .	0.20–0.30	. . .	. . .	. . .
8617	0.15–0.20	0.70–0.90	0.035 max.	0.040 max.	0.20–0.35	0.40–0.70	0.40–0.60	0.15–0.25	. . .	. . .	. . .
8617H	0.14–0.20	0.60–0.95	0.035 max.	0.040 max.	0.20–0.35	0.35–0.75	0.35–0.65	0.15–0.25	. . .	. . .	. . .
8620	0.18–0.23	0.70–0.90	0.035 max.	0.040 max.	0.20–0.35	0.40–0.70	0.40–0.60	0.15–0.25	. . .	. . .	. . .
8620H	0.17–0.23	0.60–0.95	0.035 max.	0.040 max.	0.20–0.35	0.35–0.75	0.35–0.65	0.15–0.25	. . .	. . .	. . .
8720	0.18–0.23	0.70–0.90	0.035 max.	0.040 max.	0.20–0.35	0.40–0.70	0.40–0.60	0.20–0.30	. . .	. . .	. . .
8822H	0.19–0.25	0.70–1.05	0.035 max.	0.040 max.	0.20–0.35	0.35–0.75	0.35–0.65	0.30–0.40	. . .	. . .	. . .
9310	0.08–0.13	0.45–0.65	0.025 max.	0.025 max.	0.20–0.35	3.00–3.50	1.00–1.40	0.08–0.15	. . .	. . .	. . .
Special quality grades											
HP 9-4-30	0.30	0.20	0.005	0.007	0.01	7.50	1.00	1.00	0.08	. . .	4.0
Pyrowear	0.10	0.37	. . .	. . .	. . .	2.15	1.05	3.30	0.12	. . .	. . .
17CrNiMo6	0.14–0.19	0.40–0.80	0.125 max.	0.005–0.020	0.15–0.35	1.40–1.70	1.50–1.80	0.25–0.35	0.06 max.	. . .	. . .
VASCO X2M	0.15	0.30	0.15 max.	0.01 max.	0.90	. . .	5.0	1.40	0.45	. . .	1.35
M50NiL	0.15	0.25	. . .	. . .	. . .	3.5	3.5	4.0	1.0	. . .	. . .
EX 30	0.13–0.18	0.70–0.90	0.04	0.04	0.20–0.35	0.70–1.00	0.45–0.65	0.45–0.60	. . .	. . .	. . .
EX 55	0.15–0.20	0.20–1.00	0.04	0.04	0.20–0.35	1.65–2.00	0.45–0.65	0.45–0.80	. . .	. . .	. . .

Table 2 Heat treating data for some gear steels

AISI No.	Normalizing temperature		Annealing temperature		Hardening temperature		Carburizing temperature		Reheat temperature		M_s temperature(a)	
	°C	°F	°C	°F	°C	°F	°C	°F	°C	°F	°C	°F
3310	900–950	1650–1750	860	1575	775–800	1425–1475	900–930	1650–1700	790–815	1450–1500	350	655
3140	815–930	1500–1700	790–840	1450–1550	815–840	1500–1550	. . .		. . .		310	590
4028	870–930	1600–1700	830–860	1525–1575	. . .		870–930	1600–1700	790–815	1450–1500	400	750
4047	840–950	1550–1750	830–860	1525–1575	800–840	1475–1550	. . .		. . .		. . .	. . .
4130	870–930	1600–1700	790–840	1450–1550	840–900	1550–1650	. . .		. . .		365	685
4140	870–930	1600–1700	790–840	1450–1550	830–885	1525–1625	. . .		. . .		315	595
4320	870–930	1600–1700	860	1575	. . .		900–930	1650–1700	775–800	1425–1475	380	720
3440	870–930	1600–1700	590–660	1100–1225	800–830	1475–1525	. . .		. . .		285	545
4620	930–980	1700–1800	860	1575	. . .		900–930	1650–1700	800–830	1475–1525	290	555
4640	870–930	1600–1700	790–840	1450–1550	790–840	1450–1550	. . .		. . .		320	605
4820	900–950	1650–1750	875	1575	. . .		900–930	1650–1700	790–815	1450–1500	365	685
5145	870–930	1600–1700	790–840	1450–1550	800–830	1475–1525	. . .		. . .		. . .	. . .
6120	930–980	1700–1800	870	1600	. . .		930	1700	775–800	1425–1475	400	760
6150	900–950	1650–1750	840–900	1550–1650	840–900	1550–1650	. . .		. . .		285	545
8620	870–930	1600–1700	860	1575	. . .		930	1700	775–840	1425–1550	395	745
9310	900–950	1650–1750	860	1575	. . .		900–930	1650–1700	775–840	1425–1550	345	650
EX 24	900–950	1650–1750	870	1600	870	1600	900–930	1650–1700	815–840	1500–1550	445	830
EX 29	900–950	1650–1750	870	1600	870	1600	900–930	1650–1700	815–840	1500–1550	445	830
EX 30	900–950	1650–1750	840	1550	870	1600	900–930	1650–1700	815–840	1500–1550	445	830
EX 55	900–950	1650–1750	830	1525	870	1600	900–930	1650–1700	815–840	1500–1550	420	790

(a) M_s is the temperature at which martensite forms.

quickly while cross sections remote from the end are cooled more slowly. The rate of cooling is dependent on the distance from the quenched end. After the cooling is completed, two diametrically opposite flats approximately 6 mm (¼ in.) wide are ground along the length of the bar and Rockwell hardness measurements are made at intervals of 1.6 mm (1⁄16 in.), on one or both the flat surfaces so prepared. The relationship of hardness to distance from the quenched end is an indication of the hardenability of steel (Fig. 5b and 6).

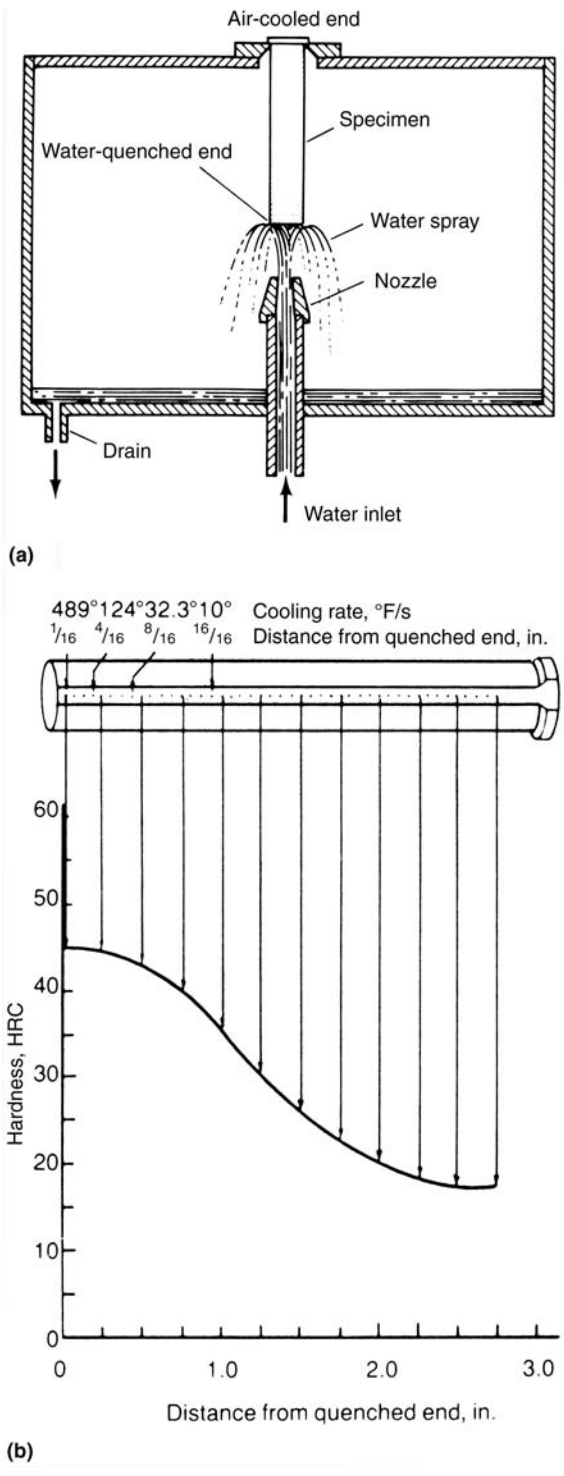

Fig. 5 Jominy end-quench apparatus (a) and method for presenting end-quench hardenability data (b). Source: Ref 2

Carbon content in alloy steels also plays an important role in developing hardenability. Figures 7 and 8 show hardenability curves for some steels with different content of carbon. The degree of hardness at the quenched end (surface) depends primarily on the carbon content, but the hardness at any point away from this end depends on the alloy content in the steel as well as carbon. Deep-hardening steels produce flatter hardenability curves (Fig. 6). Commercially available "H-band" steels assure high hardenability. Producers of such steels usually indicate minimum and maximum hardnesses that are expected at any depth from the quenched end of the bar.

Effect of Common Alloying Elements on Hardness and Hardenability. There are five fundamental factors that influence hardenability of steel:

- Mean composition of the austenite
- Homogeneity of the austenite
- Grain size of the austenite
- Nonmetallic inclusions in the austenite
- Undissolved carbides and nitrides in the austenite

It has been found that the effect of dissolved elements that combine with carbon in preference to dissolving in ferrite have the greatest influence in increasing hardenability if they are dissolved in the austenite before quenching. A carbide-forming element that is not dissolved in the austenite has no effect on hardenability except that as a carbide it may restrict grain growth, thus reducing the hardenability of the steel. Undissolved carbides reduce both the alloy and carbon content of the austenite. Since undissolved carbides restrict grain growth, in some instances, quenching from higher temperature that increases grain size promotes deeper hardening. In summary, the factors that increase hardenability are:

- Dissolved elements in austenite (except cobalt)
- Coarse grains of austenite
- Homogeneity of austenite

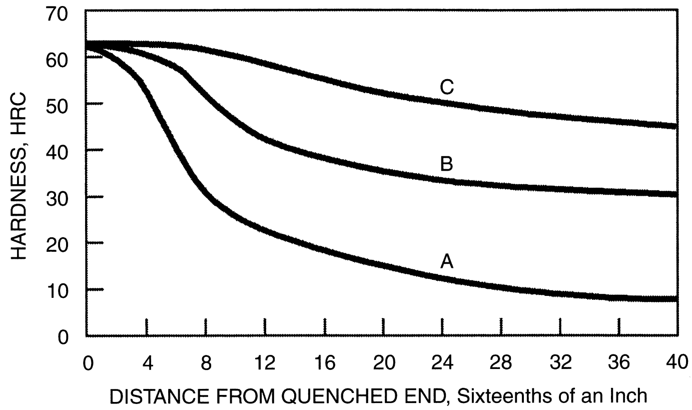

Fig. 6 Hardenability curves for different steels with the same carbon content. A, shallow hardening; B, intermediate hardening; C, deep hardening

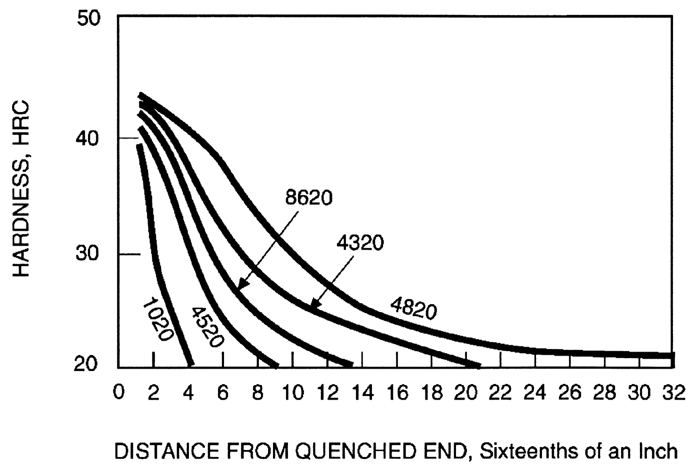

Fig. 7 Comparative hardenability of 0.20% C AISI alloy steels

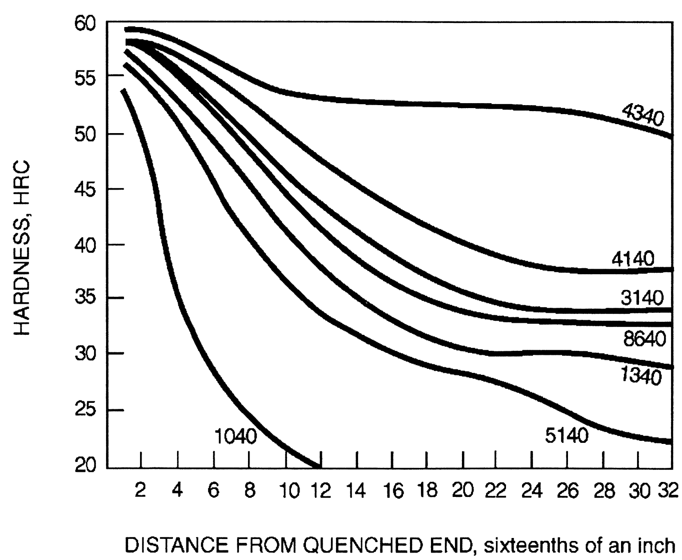

Fig. 8 Comparative hardenability of 0.40% C AISI alloy steels

The factors that reduce hardenability are:

- Fine grains of austenite
- Undissolved nonmetallic inclusions, carbides, or nitrides

Of the various alloying elements used in commonly used gear materials, the following are considered to effect tooth surface and core hardnesses significantly:

- *Nickel (Ni):* The principal advantage lies in higher tensile strength that can be obtained without appreciable decrease of elongation and reduction area. Nickel also lowers the critical temperatures, and hence, lower heat treat temperature can be used. It decreases the critical cooling rate; therefore, a less rapid quench is required to obtain hardness equal to that of plain carbon steel. Nickel increases hardenability and fatigue strength of steels.
- *Chromium (Cr):* It is essentially a hardening element and frequently used with other elements such as nickel to improve strength and wear resistance and hardenability. Chromium has, however, the disadvantage of being temper brittle, and hence, precautions must be taken when tempering in the range above 540 °C (1000 °F).
- *Molybdenum (Mo):* The pronounced effects of molybdenum when added in relatively small amounts (0.15 to 0.3%) are (a) greater ductility and toughness, (b) reduced temper brittleness, (c) improved creep resistance at high temperatures, and (d) greater hardenability when present with chromium.
- *Vanadium (V):* When present with nickel, chromium, and molybdenum, vanadium (a) improves fatigue resistance, (b) provides fine grain structure, (c) reduces grain-growth tendencies, and (d) Improves hardenability only when quenched from higher temperature
- *Tungsten (W):* Tungsten forms a hard, stable carbide that imparts wear and abrasive resistance. In the dissolved form, tungsten increases hardenability. In certain combinations with chromium and vanadium, tungsten decreases the tendency to form cracks in case-core boundary, and distortion during heat treatment.
- *Cobalt (Co):* Cobalt improves high temperature strength characteristics and corrosion resistance. In addition, it imparts excellent wear resistance. But, the hardenability of steel is reduced with cobalt over 0.4%.

Selection of Gear Steel by Hardenability. Since there is a variety of steels with standard and nonstandard analyses, the problem of selecting just the right steel for a certain application can become quite confusing if chemistry comparisons alone are used. The establishment of end-quench hardenability curves demonstrates how alloys slow down the reaction rates of steels. Also, mass has significant influence on hardenability. The rate of heat transfer is not uniform for gears with varied cross sections. Larger gears (heavier mass) experience more nonuniform heat transfer. There is danger of cracking large gears if they are quenched drastically enough to harden completely. A milder quench, on the other hand, may not develop microstructures with the required strength. In extreme cases, the part may be so large that even with considerable alloy content, it is not practical to use a fast enough quench to develop full hardness. In these cases, a gear designer needs to compromise by designing with low allowable stresses to get by with the properties that can be obtained in the steel after heat treatment. Figure 9 shows end-quench hardenability curves for several kinds of gear steels frequently used in industrial and aerospace applications.

Table 3 presents core hardness of some carburized steels of different diameter sections.

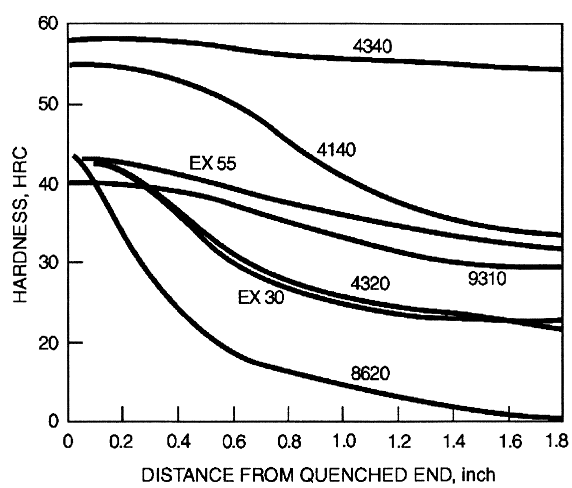

Fig. 9 Some typical Jominy curves showing end-quench hardenability. Source: Ref 3

This table also shows how the tensile properties of the specimens vary with size.

Carbon Content and Case Properties

A carburized gear tooth may be regarded as a composite structure consisting of a low carbon in the core and a high carbon of the same steel composition at the surface.

Increasing the carbon content of the case (within limits) increases wear resistance and resistance to contact fatigue. As the carbon level increases above 0.8%, there is a tendency to retain austenite and to produce carbide networks in the case. Heavy carbide networks can produce brittle cases, leading to tooth end chipping. It has been shown that a level of 15 to 20% retained austenite is desirable both for sliding wear resistance and resistance to pitting fatigue. Excessive retained austenite causes soft cases and lowers surface hardness and should be avoided. Moderate quantities of retained austenite transform and work harden under the contact load. A carbon level in the range 0.9 to 1.0% (for low-alloy steels) at the surface generally gives the optimum case properties to resist contact fatigue and surface wear. Similar case properties in high-alloy steels may be obtained with lower carbon level.

Case Depth of Tooth. Case depth is an important parameter of carburized and hardened gears and plays a significant role in determining their pitting fatigue life. The case must be sufficiently deep to resist case crushing by the applied load on the gear. In general, case depth of a carburized tooth is a function of diametral pitch. The bigger the tooth, the more case is needed to carry the loads that will be imposed on the tooth. For each size of tooth there is an optimum case depth. Too much case makes a tooth brittle with the tendency to shatter off the top of a tooth. However, too thin a case reduces tooth resistance to pitting. A high-hardenability steel with strong core structure may not need a deep case as with a lower hardenability steel. This makes it difficult to calculate an optimum case depth with complete certainty for all types of steels. A further practical point is that case depths vary according to the heat treatment cycle and equipment used. For example, in a batch-type furnace where a load is made up of a large number of densely packed gears there will be a variation in case depth throughout the load. This is due to variations in temperature and carburizing atmosphere circulation. Using a lower

Table 3 Effect of section size on core hardness and tensile properties of carburized steels

Steel type, AISI	Heat treatment: Quenching temperature of oil, °C	°F	Tempering temperature, °C	°F	Bar diam, mm	in.	Core hardness, HB	Tensile properties: Tensile strength, MPa	ksi	Yield strength, MPa	ksi	Elongation, %	Reduction of area, %
4320	815	1500	150	300	13	½	415	1055	153	1124	163	12	46
					25	1	302	738	107	738	107	17	51
					50	2	255	593	86	593	86	22	56
					102	4	248	517	75	517	75	24	57
4620	815	1500	150	300	13	½	255	876	127	621	90	20	60
					25	1	197	676	98	462	67	26	70
					50	2	192	662	96	448	65	27	70
					102	4	170	586	85	359	52	30	69
4820	800	1475	150	300	13	½	401	1441	209	1193	173	14	54
	800	1475			25	1	352	1172	170	869	126	15	51
	800	1475			50	2	277	938	136	641	93	20	56
	800	1475			102	4	241	820	119	558	81	23	59
	925	1700			152	6	226	765	111	517	75	22	62
8620	845	1550	150	300	13	½	388	1379	200	1082	157	13	49
	845	1550			25	1	255	876	127	579	84	21	53
	845	1550			50	2	235	807	117	503	73	23	58
	845	1550			102	4	207	676	98	400	58	24	58
	925	1700			152	6	200	655	95	400	58	24	64
9310	790	1450	150	300	13	½	363	1234	179	986	143	16	59
					25	1	321	1096	159	848	123	16	58
					50	2	293	1000	145	745	108	18	67
					102	4	277	938	136	655	95	19	62

temperature for carburizing, much better control over case depth is possible. At a temperature of 900 °C (1650 °F), it is not unrealistic to expect a case depth tolerance of 0.08 to 0.15 mm (0.003 to 0.006 in.).

Measurement of Case Depth. In the gear industry, two terms are frequently used to determine the quality of a carburized case—total case depth and effective case depth. Thus, an accurate and repeatable method for measuring these case depths on a gear tooth is essential not only to control the reliability of carburizing process but also for evaluation of gear performance. The total case depth is the perpendicular distance from the surface of a carburized and hardened tooth to a point inside the tooth at which difference in chemical and mechanical properties of the case and core can no longer be distinguished. Effective case depth, on the other hand, is the perpendicular distance form the surface to the farthest point inside the tooth at which a hardness of 50 HRC is measured. To measure case depths, a mechanical method is considered to be one of the most useful and accurate of all the methods available. In this method, for gears in critical applications, a cross section of a representative tooth from a pie-shaped test coupon prepared from the same "heat" of steel is preferred for reliability. However, for vacuum melt steels with cleanliness of AMS 2300 or 2304, the test coupon does not necessarily have to be made from the same heat of steel to ensure a similar reliability of case microstructure. Nevertheless, considerable care should be exercised in preparing such a specimen. The tooth needs to be sectioned perpendicular to the hardened surface. Cutting or grinding that would affect the original hardness is to be avoided. Polishing of the specimen is quite important. This polishing should be done finely enough so that hardness impressions are unaffected. A microhardness tester is recommended for case-depth measurements.

In noncritical gears, case depths are determined by step grinding (0.13 mm, or 0.005 in., step) a test coupon of rectangular cross section as illustrated in Fig. 10. Here, the hardness readings are taken on steps that are of known distances below the carburized surface, and a standard Rockwell hardness tester may be used. At a particular step, if the hardness is found to be greater than 50 HRC and less than 50 HRC on the next deeper step, the effective case depth is taken to be between the two depths. Figure 11 illustrates typical hardness versus case depth readings taken at pitch line and root fillet of a carburized and hardened gear tooth. The effective case depth at root fillet area is always somewhat lower than at the pitch line. In a gear, the effective case depth at pitch line is of importance for gear pitting life, and hence, the value at this line is regarded as the effective case depth of a gear tooth. Case depth in root fillet contributes to higher bending fatigue life.

Effective Case Depth of High-Alloy Steel Gears. Hardness versus case depth relationship as depicted in Fig. 11 applies to low- to medium-alloy steels such as AISI 8620, 4320, and so on. In high-alloy steels such as HP 9-4-30 and AISI 4330M, the core hardness after carburizing and

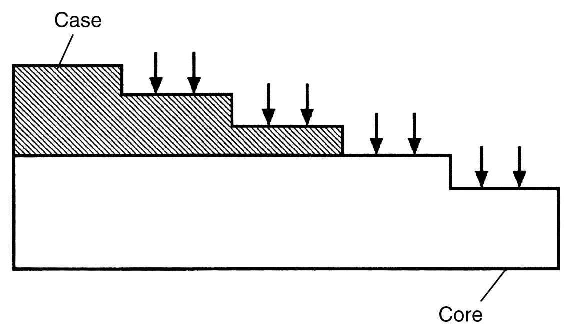

Fig. 10 Step-ground specimen for hardness-traverse method of measuring depth of medium and heavy cases. Arrows indicate location of hardness-indenter impression

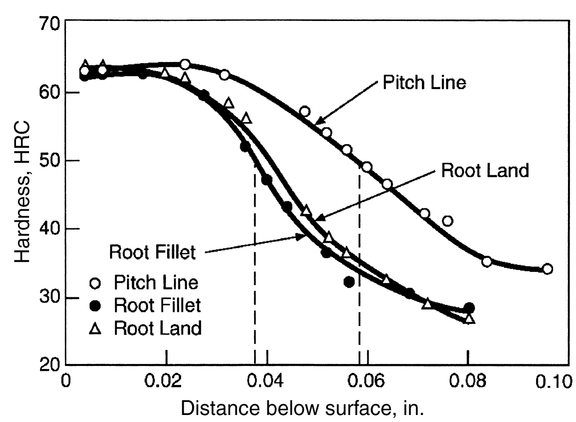

Fig. 11 Variation of hardness with distance below the surface for a carburized and hardened gear made of 8620H steel

hardening may be as high as 52 HRC. For this type of steel, effective case depth cannot be determined as already outlined. Typical hardness versus case depth for a high-alloy and high core hardness steel (HP 9-4-30) is shown in Fig. 12. For such steels, 55 HRC instead of 50 HRC is considered a better approximation of effective case depth and is successfully used in various aerospace applications.

Now the question is, how much case is needed on a gear tooth to prevent case failure due to Hertzian contact stress that causes pitting? In general, high case depths adversely affect the quality of case and, hence, the gear life. So, the determination of proper case depth on a gear tooth is quite important and is based on its capacity to resist contact stress-induced pitting.

Determination of Case Depth-Shear Stress Theory. Of the several parameters used in optimizing a gear design, the case depth, surface hardness, and core hardness of tooth are of significant importance. These three parameters are normally selected on the basis of applied load to a gear and its required life under the service conditions. Research carried out shows that a proper combination of case depth, surface hardness, and core hardness provides the maximum gear life. Analytically, these parameters are determined as outlined subsequently.

In transmitting torque, a gear tooth is subjected to at least two types of major stresses: contact and bending. These stresses cause tooth failure due to metal fatigue.

Gear tooth failure due to contact stress, commonly known as pitting, occurs when small pits initiated by fatigue cracks are formed on or below the tooth surface. These pits usually emanate at the highest point of single-tooth contact (HPSTC) for pinion and at the lowest point of single-tooth contact (LPSTC) for the mating gear (Fig. 13). The occurrence of pits is also found to be influenced by other factors such as surface quality (surface finish, microstructure of case, etc.), surface hardness, lubricating conditions, and operating temperature. Empirically, to resist crack-initiated pit formation, a gear designer considers maximum contact stress to act along the axis of load at HPSTC or LPSTC. There are two current theories that prevail to explain formation of cracks on any of these axes. The most widely held theory for well-lubricated gears is based on shear stress induced cracks below the surface due to contact load. The second one is based on cracks initiated at the surface, which happens only when the coefficient of friction goes below 0.1. According to shear stress induced pit formation below the surface, the maximum shear stress occurs on the axis of maximum contact load at HPSTC for pinion and LPSTC for gear. Because of the presence of sliding at these points, it is not known at what angle to the load axis the plane of maximum shear stress lies. That the maximum shear stress occurs at 45° to the load axis below the surface is true for pure rolling condition. In gear mesh, pure rolling exists only at the pitch point. Above and below this point there is sliding. This alters the location and magnitude of shear stress and presents difficulty in determining the point of maximum shear stress. To overcome this difficulty, the peak value of shear

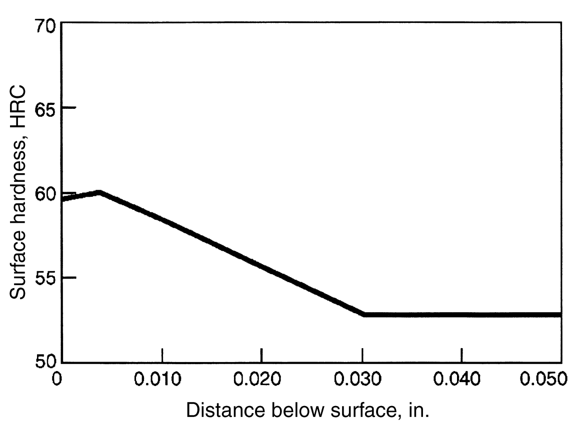

Fig. 12 Variation of hardness versus case depth in gears made of HP 9-4-30 steel

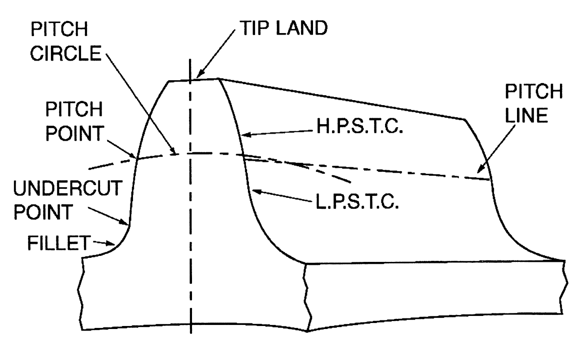

Fig. 13 Gear tooth profile

stress is assumed to occur below the pitch point and is quite valid for a well-lubricated gear mesh (coefficient of friction, $\mu \approx 0.1$, Fig. 14). Under these conditions, the value of the maximum shear stress is found to be 0.140 (0.304) P_0 at a depth of 20 (0.786)b, where P_0 is the maximum contact load in kg (lb) and b is the half width in mm (in.) of contact ellipse between two meshing teeth (Fig. 15). The shear stress so induced causes plastic flow in the outer core while the harder case still behaves elastically. This makes a Poisson's ratio for the core to deviate from 0.3 to approximately 0.5 while that of the case remains near 0.3. Thus, the core will tend to contract more than the case, inducing transverse stress, which, in turn, produces triaxial tensile stress in the core. This in combination with the discontinuity in the compressive hoop stress at the case-core interface causes cracks to form.

Cracks formed by this mechanism at the case-core boundary are resisted by the toughness of

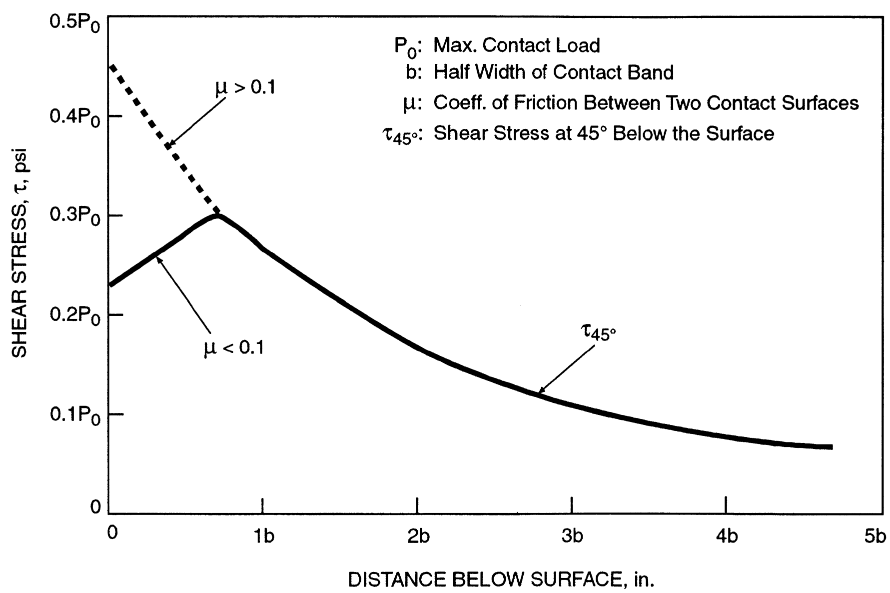

Fig. 14 Shear stress distribution along the load axis of two parallel cylinders under contract pressure

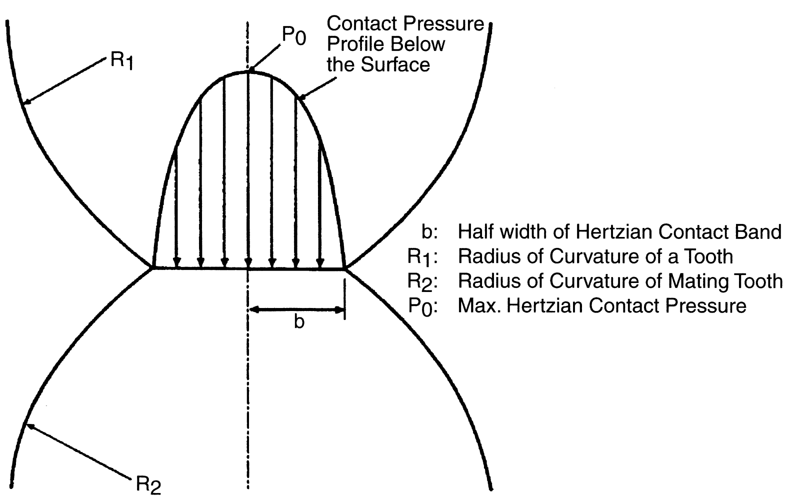

Fig. 15 Contact stress profile between two meshing gear teeth

core for propagation in the direction of the core. Facing such resistance, cracks then follow the path of maximum tensile stress that exists in the case and finally find their way out to the surface, resulting in a pitlike formation. Progressive pitting of this nature destroys the tooth profile, and the gears run rough with increased vibration levels until the teeth break off.

To prevent destructive pitting of this nature, a gear design engineer needs to make sure that the maximum shear stress lies well within the case so as not to cause any plastic flow of material in the case-core boundary. This does not mean that one should specify case depth more than what is needed. Deep cases have some adverse effects on the properties of case such as excessive retained austenite, free carbides, internal oxidation, and so on. Many of these may be responsible for modifying the residual stress distribution in the quenched case in a manner not quite beneficial to the gear life.

The theory as explained is quite useful and seems to satisfy the design requirements of gears made from most low- and high-alloy steels. But it fails to explain crack formation in gears made of some high-alloy steels with high core hardness. In these gears, pits are formed at or just below the tooth surface due to some induced tensile stresses in the case, and the gears do not seem to require high case depth as in low core hardness steels. This is possible only if the coefficient of friction becomes less than 0.1. The magnitude of maximum shear stress under this condition is approximately 0.25 P_0. Hence, case depth requirement may be reduced in comparison to shear stress induced pit below the surface theory. Experimental investigations so far carried out show this theory to be true for gears made of materials such as HP 9-4-30 with core hardness over 50 HRC. Further tests are needed for its wider acceptance with high core hardness steels. Until then, the shear stress below the surface theory will be used to determine case depth at pitch diameter of carburized gears. Now, the question is how much case depth is needed to resist bending fatigue failure of a tooth.

Gear tooth failure due to cyclic bending stress occurs when the bending stress at the root fillet surface exceeds the allowable bending fatigue strength, which is dependent on the core hardness. Higher core hardness results in higher allowable bending fatigue strength. But there is an optimum range of core hardness for maximum bending fatigue life of gears. On the other hand, case depth at the root fillet does not seem to have any direct relationship with the bending fatigue life, although it has been found that case depth indirectly influences bending fatigue characteristics through altering the magnitude of surface residual stress. In general, the case at the root fillet develops high compressive surface residual stress that is considered beneficial for improved bending fatigue life.

Total Case Depth. The shear stress theory, as explained, is used to determine the point of maximum shear stress below the surface at the pitch line (Fig. 16). Knowing this point, the total case depth (δ_c) may be calculated for gears with different core hardness from the following relationship:

$$\delta_c = k\delta_\tau \ldots \qquad \text{(Eq 6)}$$

where δ_τ is the depth at the point of maximum shear stress below the surface, and k is a constant that depends on core hardness of gear. For a safe design, its value may be taken as: 2 for core hardness up to 48 HRC, and 1.2 for core hardness above 48 HRC. Total case depth obtained with Eq 6 has been applied successfully to industrial, automotive, and aerospace gears.

The total case depth obtained by considering k = 1.2 works well for some high-alloy steel gears such as HP 9-4-30 with core hardness above 48 HRC. The case depth thus obtained does not deteriorate pitting or bending fatigue life or induce any failure due to case crushing. Furthermore, this seems to improve bending fatigue life, particularly when carburizing is done in vacuum furnaces. This is partly due to uniform case depth along the tooth profile including root fillet that is achieved with vacuum furnace carburizing. Carburizing in atmospheric furnaces cannot assure such consistency.

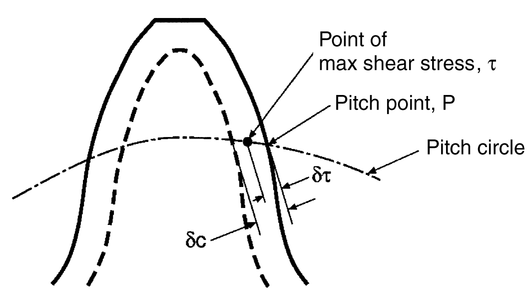

Fig. 16 Estimation of total case depth

Recommended Total Case Depth for Low-Alloy Steel Gears. As discussed, the case depth required on a carburized tooth is primarily a function of its diametral pitch. The bigger the tooth, the deeper the case needed to carry the loads. Also, for each size of tooth there is an optimum case depth. Too much case makes the tooth brittle with a tendency for the tip of the tooth to shatter. Too thin a case, on the other hand, reduces tooth strength and resistance to pitting. Table 4 shows the general practice on case depth for different DP gear tooth based on maximum shear stress theory.

Carburizing of gears with acceptable case microstructure below 1 DP is extremely difficult because of tooth size. This is the reason, carburized and hardened gears below 1 DP are rarely used in industrial applications. If surface hardening is needed for such gears, an alternate hardening process such as induction hardening is recommended.

For extremely critical gears, it is advisable to hold case depth toward the maximum as shown in Table 4 and the minimum limit raised. For example, case depth on a critical 10 DP gear might be held to 0.635 to 0.90 mm (0.025 to 0.035 in.) case depth instead of 0.508 to 0.90 mm (0.020 to 0.035 in.). The case depth so specified is the total case depth as noted in an etched specimen. The effective case (h_{ec}) for surface durability is taken to be about 75% of the total case or may be estimated by the following equation:

$$h_{ec} = \frac{S_c \cdot d \cdot \sin \phi_t}{7.0 \times 10^6 \cdot \cos\psi_b} \left(\frac{m_G}{m_G + 1}\right) \text{ in.} \qquad \text{(Eq 7a)}$$

where S_c is the maximum contact stress psi, in the region of 10^6 to 10^7 cycles; d is the pinion pitch diameter (in.); ϕ_t is the pressure angle; Ψ_b is the base helix angle; and m_G is the tooth ratio.

In metric units:

$$h_{ec} = \frac{S_c \cdot d_m \cdot \sin \phi_t}{48{,}250 \cos\psi_b} \left(\frac{m_G}{m_G + 1}\right) \text{ mm} \qquad \text{(Eq 7b)}$$

where S_c is the maximum contact stress, N/mm^2; and d_m is the pinion diameter, mm.

Recommended case and core hardness at pitch line for different applications are given in Table 5.

Case Depth at Tooth Tips. Under normal conditions, gear tooth tips do not experience any load. In case of misaligned gears, the tips may be subjected to high contact load due to which they may fail, particularly if the tips are through hardened during carburizing. To avoid such failures, case depth in the region of tooth tips, h_{em}, should be between 0.3 and 0.4/DP in inches or 0.3 and 0.4 × module mm.

Recommended case depths at tooth tips are given in Table 6.

Grain Size and Case Depth. Besides time and temperature, case depth also depends to a certain extent on grain size of steel. Coarse-grain steels (ASTM 4) develop a deeper case, whereas fine-grain steels (ASTM 10 to 12) produce a shallow case. Grain size between ASTM 5 and 7 is recommended for proper case depth.

The grain size usually is reported as a number that is calculated from the relation:

$$n = 2^{N-1} \qquad \text{(Eq 8)}$$

Table 4 Recommended case depths at pitch line

Diametral pitch (DP)	Case depth, mm	Case depth, in.
20	0.25–0.46	0.010–0.018
16	0.30–0.58	0.012–0.023
10	0.50–0.90	0.020–0.035
8	0.64–1.02	0.025–0.040
6	0.76–1.27	0.030–0.050
4	1.02–1.52	0.040–0.060
2	1.78–2.54	0.070–0.100
1	2.29–3.30	0.090–0.130

Table 5 Recommended case and core hardnesses at pitch line of low-alloy steel gears

Application	Surface hardness, HRC	Core hardness, HRC
General-purpose industrial gearing	55 min	28–32
High-capacity industrial gearing	58 min	32–38
Aircraft gearing	60 min	38–48

Table 6 Recommended case depth at tooth tip

Diametral pitch (DP)	Maximum case depth, mm	Maximum case depth, in.
20	0.71	0.028
16	0.86	0.034
12	1.17	0.046
10	1.40	0.055
8	1.75	0.069
6	2.34	0.092
4	3.18	0.125
2	3.56	0.140

where n is the number of grains per square inch at a magnification of 100×, and N is the grain size number commonly called the ASTM grain size.

Table 7 shows ASTM grain size and grains per square inch.

Core Hardness of Gear Teeth

Core hardness of teeth, as already discussed, is an important parameter that determines the strength of a gear tooth. Currently, it is a common practice in the gear industry to accept hardness at the center of a tooth on form diameter as the core hardness (Fig. 17). This consideration has some merit. First, it allows a conservative estimate of allowable bending fatigue strength of a tooth because the hardness at the point of maximum bending stress that occurs near to the root surface is higher than at the center of the tooth. Hence, the actual bending strength of a tooth is always higher than what is considered to establish its bending fatigue life. Second, it identifies a definite spot in a tooth to measure hardness.

Table 7 ASTM grain size

ASTM No.	Mean No. of grains/in.^{2}	ASTM No.	Mean No. of grains/in.^{2}
1	1	7	64
2	2	8	128
3	4	9	256
4	8	10	512
5	16	11	1024
6	32	12	2048

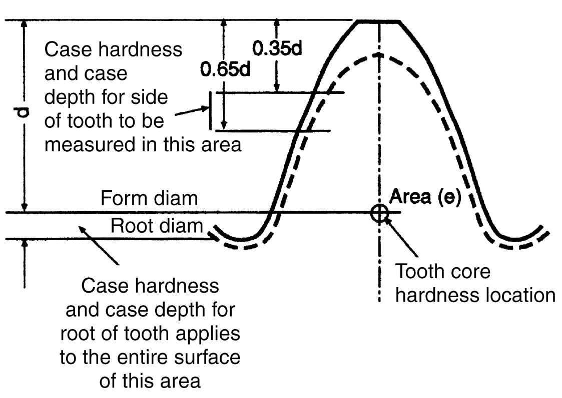

Fig. 17 Measurement locations of case and core hardnesses

The major drawback of this practice is the hardness at this location is always low for low-hardenability steels. Hence, the allowable strengths are low. This makes the gears made of such steels large, and eventually, the gearbox becomes large. In reality, the locations of maximum stresses that cause failures are not at the center of the tooth. Recent investigations show maximum tensile stress due to bending of a tooth that causes failure occurs just below the case at root, and also the stress that causes case crushing is located in the case-core interface area on the pitch diameter. Hardness in these areas is of importance because this is where failures originate, and they are substantially higher than at the center of tooth. Hence, it is logical to determine bending or case crushing failures on the basis of these strengths. Such considerations would be beneficial in optimizing gear designs that are bending-strength limited.

The main problem with the new approach lies in precisely identifying the locations for proper core hardness measurements. To avoid complexity in preparing a sample and subsequent discrepancy in measurements, the following locations are considered realistic as illustrated in Fig. 18:

- C_1 *for bending strength:* Between root and center of tooth
- C_2 *for case crush support strength:* Between pitch point and center of tooth

Core Hardness versus Maximum Bending Strength. Metallurgically, for a carburized and

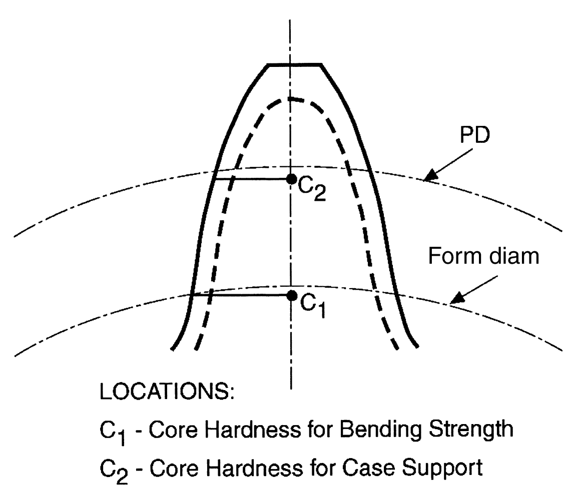

Fig. 18 Measurement locations of importance for core hardnesses

hardened gear tooth, higher core hardness results in higher bending fatigue strength. But unfortunately, this relationship is not fully true for gears. Research shows maximum bending fatigue strength of a gear tooth for low- and high-alloy steels is achieved with core hardness in the range of 34 to 46 HRC. It is easy to understand the reason for lower bending fatigue strength for core hardness below this range. Why bending fatigue strength for such alloys does not increase above this range is difficult to explain. For very high-alloy (above 9%) steels such as HP 9-4-30, maximum bending strength is achieved with core hardness between 46 and 50 HRC. It is believed at core hardness above 52 HRC, an unsatisfactory tensile residual stress is occasionally observed. Figure 19 illustrates the relationship of core hardness with bending fatigue strength of carburized and hardened gears.

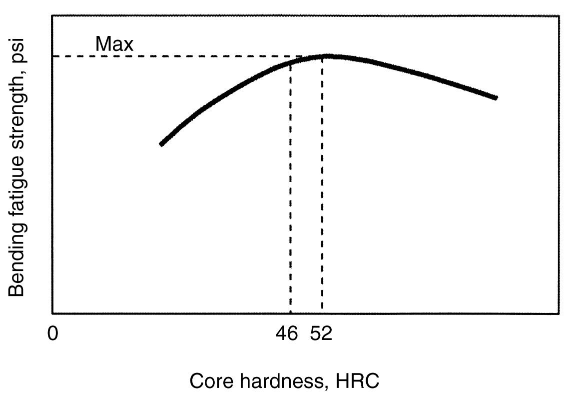

Fig. 19 Core hardness vs. bending fatigue strength of gear tooth

Table 8 Core hardness of tooth for some gear materials

Diametral pitch (DP)	Core hardness, HB			
	AISI 4620	AISI 8620	AISI 4320	AISI 9310
0.75	180–230	180–250	215–270	245–340
1	180–245	180–260	225–300	260–360
1.25	180–260	230–290	240–325	280–370
1.5	200–280	230–310	255–343	285–380
1.75	200–280	235–320	260–350	290–380
2	225–310	240–335	270–370	300–380
2.5	235–340	250–360	280–390	310–390
3	235–340	260–390	300–400	310–390
3.5	235–340	270–395	315–410	315–390
4	255–375	290–400	315–410	315–390
5	290–395	320–420	335–420	315–390
6	310–405	320–420	340–430	315–390
7	310–405	335–430	340–430	315–390
8	310–405	335–430	340–430	315–390
10	350–425	370–440	370–440	315–390

Core hardness ranges are based on min and max hardenability curves for H grades and quench temperatures between 80 and 95 °C (175 and 200 °F).

Table 8 shows achievable core hardness range (HB) versus DP of tooth for some commonly used carburizing grade gear materials.

Surface Oxidation Effect. Oxidation at the surface should be avoided during carburizing. The most undesirable effect of surface oxidation is a loss of hardenability because O_2 reacts preferentially with the alloying elements. It is less of a problem for steels with high-alloy content.

Grain Size and Core Hardness. Sometimes, full core hardness is not achieved in gears made from a "heat" of very fine grain steel (ASTM 10 to 12) instead of normally used ASTM 5 to 7. It is not unusual to see approximately one point HRC hardness gain or loss accompanied with each grain size change in gear tooth size from 3 to 6 DP. In general, core hardness drops as grain size becomes finer and increases when it coarsens. Similarly, bending fatigue strength increases with increase of austenitic grain size as illustrated in Fig. 20. Pitting fatigue strength, on the other hand, increases with finer grain size. It is to be noted that the temperature to which a steel is heated has a marked influence upon the austenitic grain size. It is necessary, therefore, when determining the austenitic grain size of carburized steels, to make the determination under conditions that are similar to the heat treating procedure.

Effect of Carburizing Processes on Surface Carbon. Surface hardness and hardness below the surface are dependent on carbon gra-

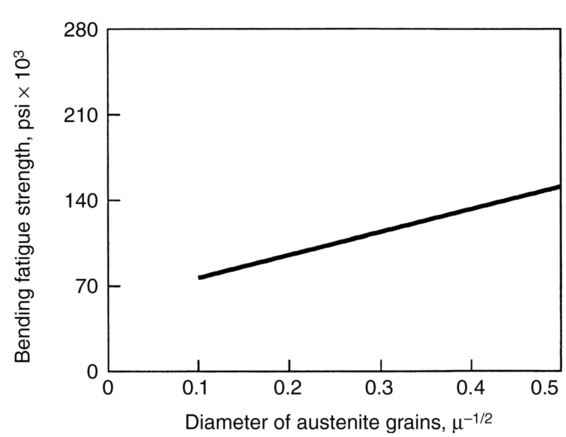

Fig. 20 Austenitic grain size and bending fatigue strength of a typical gear steel

dient. For higher surface hardness without a large carbide network, 0.80% carbon is preferred on the surface. For low carbon potential (below 0.70%), some decarburization may take place at the surface of steel during carburizing by any of the processes such as endothermic, vacuum, fluidized bed, and vacuum pulsation. Decarburization results in lower surface hardness after quenching. This is why it is important to maintain proper carbon potential during carburizing. Also, the penetration of carbon into the surface depends on the carbon gradient, which is found to vary with the carburizing process selected. Figure 21 illustrates various carbon gradients for gears made of HP 9-4-30 with different carburizing processes. It shows that carburizing in a vacuum pulsating-type furnace offers the highest carbon on the surface for the same carbon potential. Figures 22 and 23 illustrate typical carbon profiles for AISI 9310 and 8620 steels for different carbon potentials in an endothermic carburizing furnace.

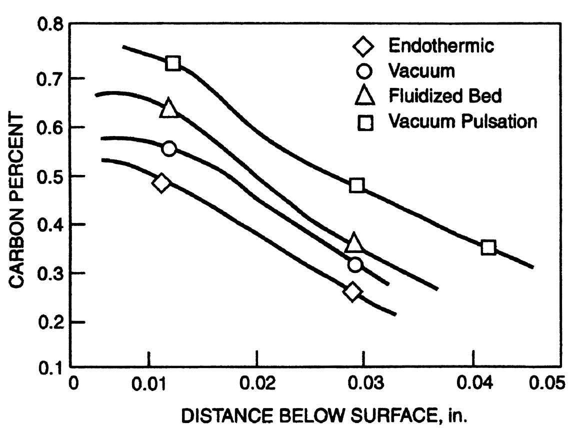

Fig. 21 Carbon gradients by different carburizing processes. Material: HP 9-4-30

Microstructure of Carburized Cases

The microstructure of a carburized gear tooth changes from the surface to the core. Depending on carbon content and post carburizing heat treatment, the microstructure at the surface will be pearlitic or martensitic, with or without grain-boundary carbides. Toward the interior, there is a gradual transition in microstructure to transformation products that are characteristics of lower carbon contents until the core is reached. In some steels, there is only a transition from high-carbon martensite to low-carbon martensite, but in other steels, there is a transition from pearlite with proeutectoid cementite through eutectoid pearlite to a predominantly ferritic structure. Such a gradation in microstructure is useful to measure case depth by microscopic examination.

In planning carburizing of a gear, it is thus necessary to consider both the carburizing cycle and the post carburizing heat treatment. The proper control of heat treatment develops the required properties in the core and at the same time, develops the required tooth surface hardness and hardness gradient in the case. Case properties depend strongly on carbon content and carbon gradient. It is only through adjustment of carbon content and carbon gradient that

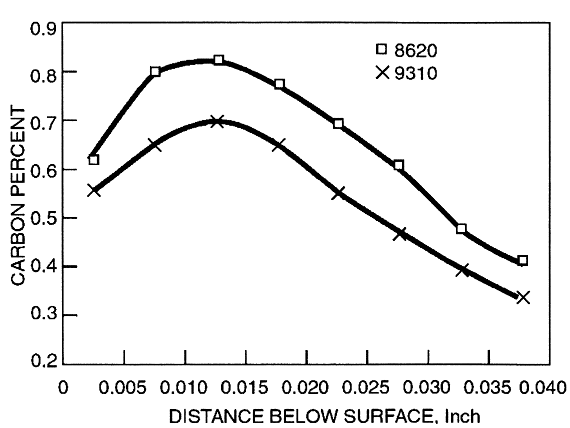

Fig. 22 Carbon profiles for two different AISI steels (carburized with carbon potential of 0.7%)

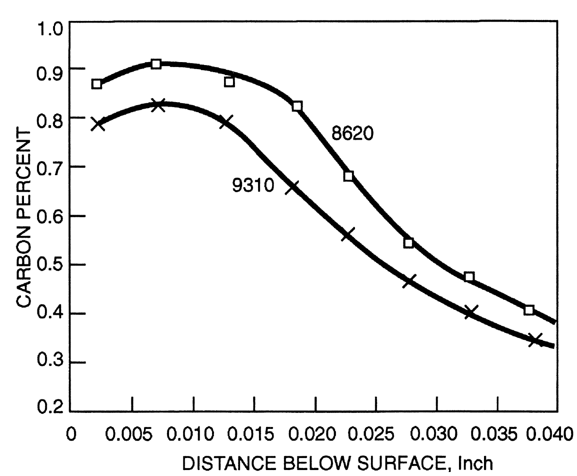

Fig. 23 Carbon profiles for the same two steels shown in Fig. 22 (carburized with carbon potential of 1.0%)

the heat treatment can be made to develop good case properties.

The microstructures of case and core also are influenced by the heat treatment after carburizing. Because of the generally low carbon content in the core, its structure may consist of low-carbon martensite to mostly ferrite. On the other hand, the case structure, because of higher carbon content, may contain martensite, bainite, or pearlite, depending on the rate of cooling from the austenitizing temperature. Some proeutectoid ferrite may appear near the case-core interface, especially if the cooling rate is slow. In some alloy steels, retained austenite may appear near the surface, particularly if the surface carbon content and the austenitizing temperature are high or the cooling rate is too fast. Steels containing nickel are especially susceptible to such austenite retention. Microstructure requirements of case and core generally vary with the class of gears. Table 9 shows recommended microstructures for different classes of gears. Metallographic standards for carburized gear steel structures are shown in Fig. 24 and 25. More detailed information on the microstructures of carburized steels can be found in Ref 4 and 5.

Problems Associated with Carburizing

During carburizing, gear tooth tips experience higher carbon absorption than at the roots. The reason for this is the convergence of carburizing

Table 9 Hardness and microstructure requirements in case and core for different classes of carburized and hardened gears

			Hardness		Microstructure	
Gear class(a)	Process	Material	Case surface (Knoop 500 g)	Core HRC	Area of part	Requirement
A	Carburize and harden	Carburizing grade	720 min on tooth surfaces 710 min at root fillet areas	34–44	Case	High-carbon refined tempered martensite. Retained austenite 10% max. Continuous carbide network or cracks are not acceptable. Scattered carbides are acceptable provided the max carbide particle size does not exceed 0.005 mm (0.0002 in.) in any direction. Transformation products such as bainite, pearlite, proeutectoid ferrite, or cementite not permitted in excess of the amount. No white martensite (untempered) permitted.
					Core	Low carbon (tempered) martensite. No blocky ferrite, pearlite, or bainite. Ferrite patches not to exceed 1.6 mm ($^{1}/_{16}$ in.) in width or length as measured at 250× magnification. Excessive banding not permitted.
B	Carburize and harden	Carburizing grade	690 min (all areas)	30–44	Case	High-carbon tempered martensite. Retained austenite 20% max. No continuous carbide network is acceptable. Scattered carbides are acceptable provided the maximum carbide particle size does not exceed 0.010 mm (0.0004 in.) in any direction. Surface oxidation not to exceed 0.013 mm (0.0005 in.). Transformation products not permitted in excess of the amount shown. No white martensite permitted.
					Core	Essentially low-carbon martensite with some transformation products permissible. Ferrite patches up to 3.18 mm ($^{1}/_{8}$ in.) wide and length permissible as measured at 250×. Excessive banding not permitted.
C	Carburize and harden	Carburizing grade	630 min (all areas)	28–45	Case and core	Defects such as laps and cracks are not permitted. Retained austenite, 30% max. Case depth shall meet drawing requirements. Excessive inclusions that may affect the function of the part shall be cause for rejection.

(a) A, critical applications where a gear failure may result in loss of life; B, not as critical as A but still requires high reliability; C, industrial application gas flow at the tips, whereas divergence exists at the root area as illustrated in Fig. 26. As a result, the case thickness does not run parallel to the tooth profile. The thickness is deeper at the tip and shallower at the root. Consequently, this difference results in the slope variation of hardness versus case depth relationship taken at different points on a gear tooth as depicted in Fig. 27. The difference in case depth becomes more critical as the gear tooth geometry also changes. For example, with an increase in pressure angle, the case thickness at the tip increases significantly, making it almost through hardened. Figure 28 shows this condition. Such tooth tips are very brittle and subject to chipping in case of any misalignment of gears.

Copper Plating of Gear to Control Case Depth at Tooth Tip. To overcome the problem of high case depth at tooth tips, very frequently gear blanks are plated with copper before cutting the teeth. After the teeth are cut, plating is still left on the tooth land, which acts as a buffer during carburizing and reduces case depth at and near the tips of the gear teeth. Results obtained so far show that copper plating helps the reduction of case near the tips of coarse pitch (up to 10 DP) and small pressure angle (up to 20°) gear teeth where an appreciable top land exists. In finer-pitch (20 DP and higher) and higher-pressure angle gears (above 20°), the tooth land is relatively small, and copper plating offers only limited case depth con-

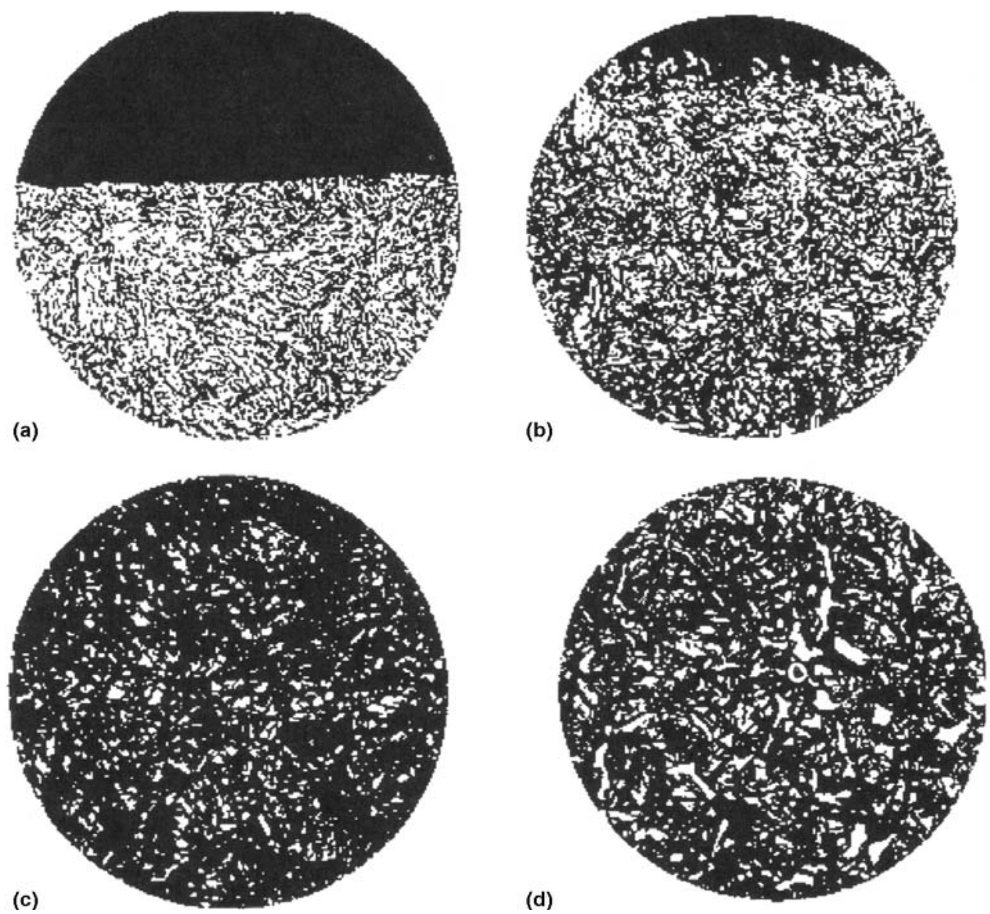

Fig. 24 Metallographic standard for carburized, hardened, and tempered cases in grades A, B, and C gears. (a) Desired high carbon, carburized, hardened, and tempered martensitic case. No retained austenite. No surface oxidation. Acceptable for grade A. (b) Case structure 10% retained austenite and small amounts of transformation products. Maximum austenite permissible for grade A. (c) Case structure with 20% retained austenite and some transformation products. Maximum permissible for grade B. (d) Case structure with over 30% retained austenite and a considerable amount of transformation products. Continuous grain boundary carbides, massive carbides. Not acceptable for grade A or B

trol at the tips. On the basis of recent experimental work, the following recommendations (Table 10) are put forward with regard to copper plating of gears before carburizing.

In industries where in-process inspection is critical, a note on the gear drawing is helpful (e.g., "peeling of plating on the edges of teeth resulting from gear cutting should not be the cause of rejection of plated gears"). Also, stripping of plating after carburizing is not necessary provided a small amount of peeled copper plating is acceptable in the gearbox lubricating oil system.

Tip and Edge Radii. For good heat treatment and performance of gears, tooth tips and edges are to be rounded as suggested in Table 11.

Retained Austenite and Its Effect on Gear Performance. After carburizing and hardening, it is possible that some retained austenite may exist near the surface of the gear teeth. Steels containing nickel are especially susceptible to such austenite retention. The retained austenite is not generally considered harmful to gear life when present in the amount not exceeding 15 to 20% by volume. In fact, retained austenite present between 15 to 20% by volume seems to increase bending fatigue resistance of gear teeth (Fig. 29). On the other hand, retained austenite in the martensitic microstructure of the case lowers the surface hardness, which is not at all desirable for contact fatigue life. Also, a high percentage of retained austenite (above 20% by volume) is found to be detrimental dur-

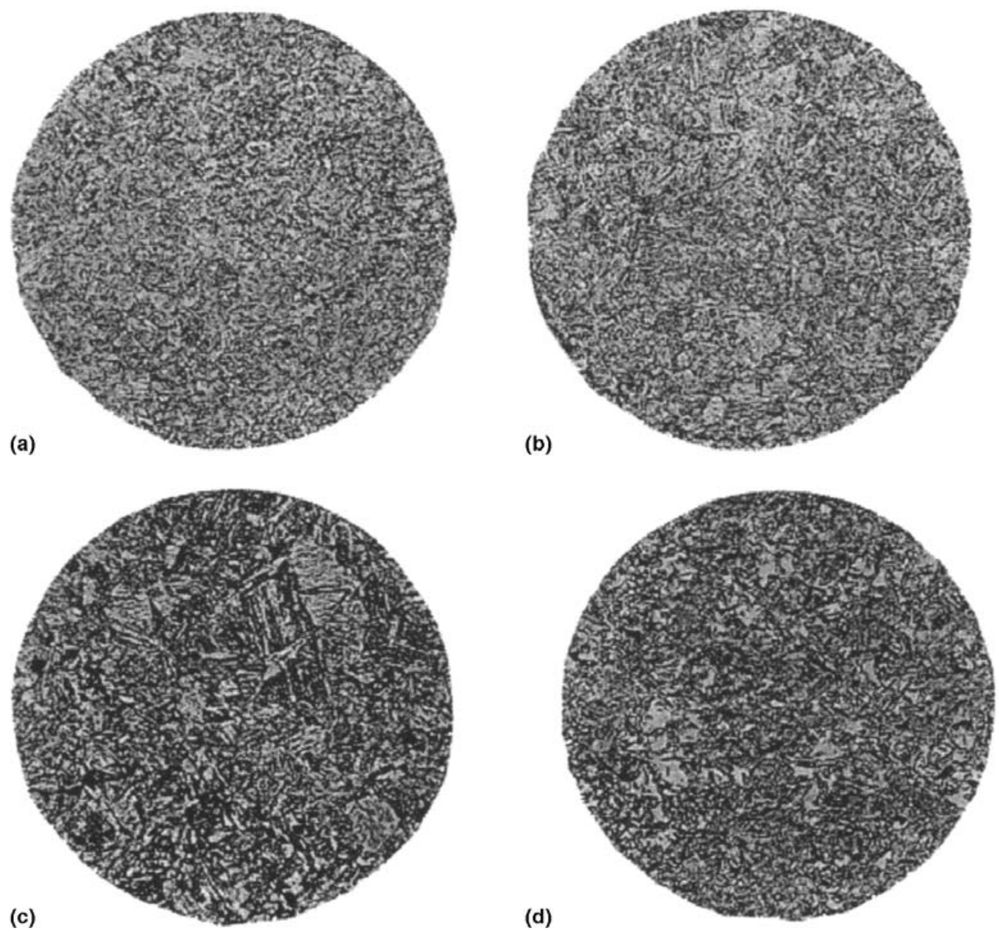

Fig. 25 Metallographic standards for carburized, hardened, and tempered core structure. (a) Desired low-carbon, tempered martensite, free from ferrite patches and with some transformation products. Acceptable for grade A. (b) Low-carbon, tempered martensite with maximum allowable transformation products. Acceptable for grade A. (c) Low-carbon, tempered martensite with maximum allowable transformation products. Acceptable for grade B. (d) Tempered martensite with block ferrite patches. Not acceptable for grades A and B

ing the service life of gears where the volume accompanying austenite-martensite transformation causes dimensional change in gear tooth geometry. Furthermore, martensite formed in this manner is untempered and brittle and may accelerate crack formation in the case. Hence, it is essential to control the amount of retained austenite for maximum service life of gears. Recent research indicates that finely dispersed, retained austenite in the amount of up to 15% is not detrimental to the contact fatigue (pitting) life of gears. Retained austenite above 20% may cause "grind burn," discussed later in this chapter, particularly if the gears are ground on wet gear grinding machines with vitrified aluminum oxide wheels.

A number of variables during carburizing affect retained austenite in gears. These variables include carbon potential, quench temperature, cooling rate, and so on. Holding the carbon potential between 0.7 and 0.85% and using quench oil temperature below 95 °C (200 °F) and fast cooling rate, the retained austenite can be significantly reduced. Another way of reducing the amount of retained austenite in the case microstructure is to cold treat the gears following quenching.

The specific amount of transformation achieved by a given subzero treatment is extremely difficult to predict. The degree of reluctance to transform at a given temperature is influenced by:

- The amount of retained austenite at the start of cold treatment

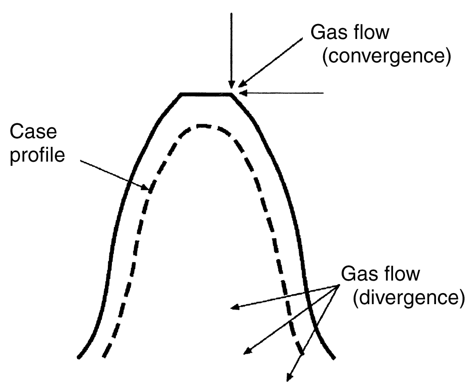

Fig. 26 Gas flow over a gear tooth during carburizing

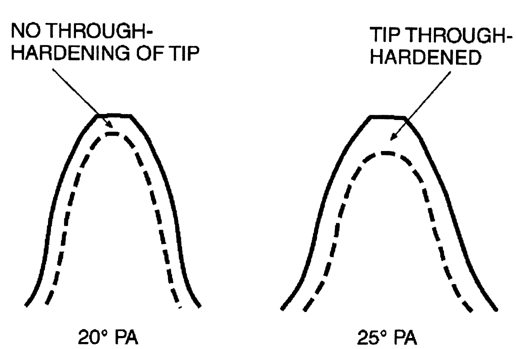

Fig. 28 Case depth profile vs. tooth pressure angle. Dashed line indicates case depth profile

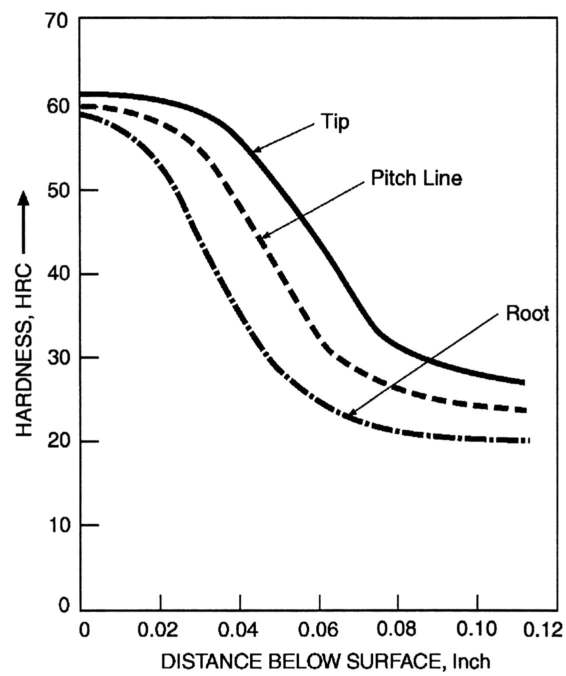

Fig. 27 Hardness profiles at different locations of a tooth

Table 10 Copper plating of gears

Diametral pitch (DP)	Pressure angle (PA)	Copper plating
15 and higher	20° and above	Plating not recommended
10–14	20° and above	May use plating on selective basis
10 and lower	20° and below	Use plating

Table 11 Recommended tip and edge radii of teeth

	Tip and edge radii	
Diametral pitch (DP)	mm	in.
1–4	0.51–0.76	0.020–0.030
5–8	0.38–0.64	0.015–0.025
10–12	0.25–0.51	0.010–0.020
16–20	0.13–0.25	0.005–0.010

- The elapsed time between quenching and cold treatment
- Any intermediate thermal treatment such as tempering
- The general level of residual compressive stress in the parts
- Any cold working of material such as straightening

Tempering of gears at 150 to 175 °C (300 to 350 °F) prior to cold treating is a common practice. It decreases the tendency to form subsurface microcracks. Also, tempering tends to stabilize retained austenite. After tempering, a slightly lower temperature is required to attain the same degree of transformation upon cold treating. Holding gears at room temperature for some time probably has a stronger stabilizing effect than tempering immediately after quenching. Because a volume increase accompanies such transformation, higher levels of compressive residual stress after quenching tend to retard transformation until lower temperatures are reached. Temperatures in the range of –75 to –100 °C (–100 to –150 °F) are routinely used in cold treating as already discussed.

To make sure the carburized case has the proper amount of retained austenite, it is necessary to have a proper measurement technique. Currently, there is no exact method to determine the percentage of retained austenite in a carburized case. Of the various methods available, metallographic examination and x-ray diffraction are frequently used. In metallographic examination, austenite is not attacked by nital, and therefore, it appears white when the nital-etched specimen is examined (Fig. 30). Metallographic method, however, is not a reliable method of estimating the percentage of retained austenite. X-ray diffraction technique is considered to be more reliable. Even this method fails to positively identify the percentage when the retained austenite is below 10%.

Heat Treat Distortion of Carburized and Hardened Gears

Distortion is always a problem in all heat treating processes, and it is most severe in carburizing and hardening. Thus, its reduction or elimination is a very important factor in the manufacture of high-quality gears. In gears, two types of distortion occur. One is a body distortion, which, for gears, is measured in terms of out-of-roundness, out-of-flatness, or run-out dimensions. The second is the distortion in gear tooth geometry. Body distortion also influences tooth geometry distortion a great deal. In gen-

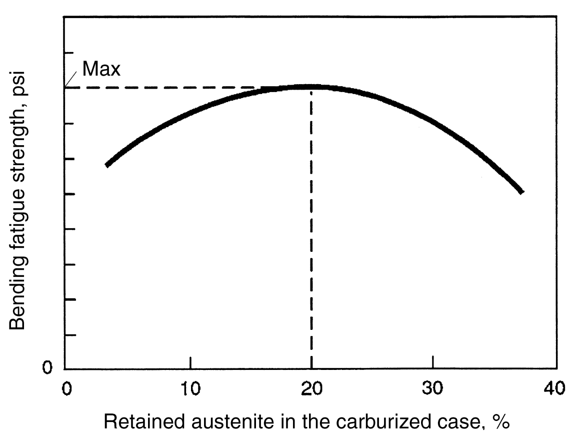

Fig. 29 Influence of retained austenite on bending fatigue strength

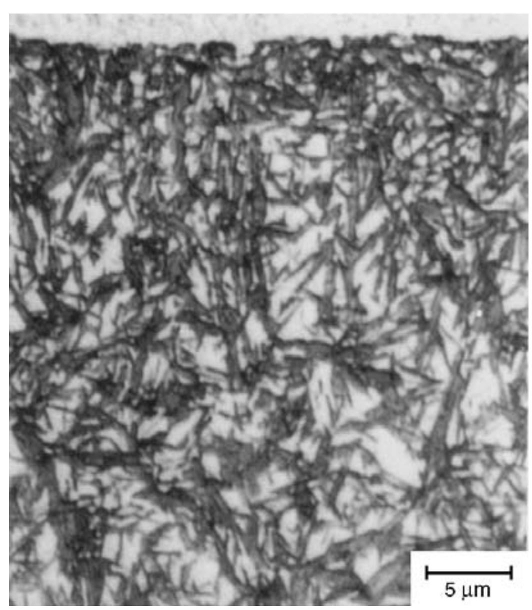

Fig. 30 Typical near-surface case microstructure of direct-quenched carburized steel. Martensite plates etch dark and retained austenite appears white. Gas-carburized AISI 8719 steel (1.06% Mn, 0.52% Cr, 0.5% Ni, 0.17% Mo). Light micrograph, nital etch

eral, distorted gears need a finishing operation after carburizing and hardening to improve their quality.

The mechanism of heat treat distortion is quite complex for any component and more so for gears. It is expected that the following brief discussion on this phenomenon will be helpful to manufacture high quality gears at an optimum cost.

Mechanics of Heat Treat Distortion

After carburizing, gears are quenched either in oil, water, or gas to increase tooth surface hardness. As the gear body cools during quenching, two types of internal stresses are induced: thermal and grain microstructure transformation.

Thermal Stress. Thermal stress is induced due to exterior surfaces of the gear cooling more rapidly and contracting than the inner ones, thus developing a temperature gradient. In alloy steels, the various alloying elements with different densities result in different specific volumes that cause additional thermal stresses. These internal stresses may cause extensive deformation as the gear cools down to room temperature. In general, the more drastic the quench is, the greater the thermal stress causing gear distortion.

Transformation Stress. During the carburizing process, as a gear is heated up to the first critical temperature, its volume increases. This volume then decreases as ferrite transforms into austenite. Once the austenitic transformation is complete, volume increases again with higher temperature. However, the coefficient of volumetric expansion is different for austenite than for ferrite. After the austenitic transformation, as the temperature is lowered, thermal contraction takes place until the temperature falls to quenching temperature. As the temperature drops during quenching, austenite is transformed to martensite and the volume increases again. This transformation goes on until the temperature drops to room temperature. Even at this temperature, all martensitic steels contain some austenite, the amount increasing with the amount of alloying elements dissolved during austenitization. The larger the quantity of retained austenite in the steel after hardening is, the smaller the increase in volume. Also, the volume of austenite increases with the amount of dissolved carbon.

Furthermore, all gear steels are basically anisotropic, which means the volume change occurring during the heat treat process will not be the same in the direction of rolling from which gear blanks are made as in the direction of right angles (90°) to it. The volume change in gears also is affected by the configuration and geometry of gear blanks (with or without web). All such volumetric changes induce additional stresses in a gear body that cause distortion.

Material and Heat Treat Process Factors

Besides thermal and transformation stresses, the factors discussed below contribute significantly to the distortion of carburized and hardened gears.

Quality of steel plays an important role in the process of distortion. The better the quality of the steel is, the lower the distortion. For example, alloying segregation in any gear material may cause variations in hardness in a given cross section, which, in turn, initiates a differential response during machining operations. The result is various degrees of cold work with unpredictable dimensional change during heat treatment. Another material property is grain structure. A uniform grain structure is desirable for predictable distortion. Furthermore, the nonmetallic inclusions in the material need to be minimized, preferably by selecting vacuum-melted steels. Low inclusions make the mechanical properties of vacuum-melted steels far superior to the air-melted variety. In the following section, the primary advantages and limitations of vacuum-melted steels are discussed in detail. Also, it is important that gears of the same material heat code provided by steel mills are processed together to maintain uniform heat treat response.

Material cleanliness is an important criterion that needs to be considered for acceptable quality of gear materials. Also, material cleanliness plays an important role that determines the consistency of heat treat response. The cleaner the material (low nonmetallic inclusions) is, the more predictable the heat treat distortion of gears. The degree of cleanliness usually is controlled by an Aerospace Material Specification (AMS). Of the various applicable specifications, the following are frequently applied to gear steels:

- AMS 2300: "Steel Cleanliness, Premium Aircraft Quality Magnetic Particle Inspection Procedure"

- AMS 2301: "Steel Cleanliness, Aircraft Quality Magnetic Particle Inspection Procedure"
- AMS 2304: "Steel Cleanliness, Special Quality Magnetic Particle Inspection Procedure"

As the titles indicate, these specifications also call for strict magnetic particle inspection standards to control surface integrity. AMS 2300 specifies the highest cleanliness standard and can only be achieved in vacuum arc remelt (VAR) steels. On the other hand, AMS 2301 quality steels are generally achieved by air melting, whereas AMS 2304 quality may be attained either with VAR or air melting under strict control of nonmetallic inclusions.

Sometimes, AMS 2300 quality steels are not readily available because of strict control of inclusions. Thus, it may create some unacceptable bottlenecks in manufacturing. Also, in-house inspection of such materials presents quite a bit of difficulty for gear manufacturers not equipped with proper measuring equipment. Furthermore, these materials are not easily machinable due to lack of sufficient sulphur, which causes cutting speeds and feeds to be slowed down considerably. Even then, high stresses may be induced in the gears during machining. For predictable heat treat distortion, it is thus a good practice to stress relieve gears made of these materials before any heat treatment. In addition, experimental results show that material cleanliness affects some important mechanical properties such as allowable contact fatigue stress and bending fatigue stress in gears.

Studies on the effect of steel cleanliness on gear performance led to the development of cleanliness standards for three different materials by the American Gear Manufacturers Association (AGMA):

- Grade 3 materials call for AMS 2300. Gears made of these materials are used in critical applications. Allowable contact fatigue stress is 1900 MPa (275 ksi).
- Grade 2 materials call for AMS 2301. Gears used in these applications are not as critical as those made of grade 3 materials but have been successfully used in high-speed applications up to 185 m/s (36,000 ft/min). These materials are easy to machine and procure. Cost/pound is significantly lower than grade 3 materials. Allowable contact fatigue stress is 1550 MPa (225 ksi).
- Grade 1 materials do not specify any special cleanliness quality and nonmetallic inclusions may vary widely. A great percentage of industrial-type gears are made with these materials. Allowable contact fatigue stress for such steels is 1240 MPa (180 ksi). A major disadvantage with this grade of materials is significant uncontrolled heat treat distortion of gears.

Although there is no separate AGMA grade for materials that meet AMS 2304, their mechanical properties are far superior to AMS 2301 quality materials. Allowable contact fatigue stress may be considered to be similar to grade 3 materials. Also, heat treat response of these materials is as predictable as AMS 2300 materials. Besides, AMS 2304 materials are easily available and their cost/pound is less than AMS 2300 materials. All of these features sometimes make AMS 2304 material quite attractive even for some critical applications.

The Hardenability of Steel. Steels with higher hardenability, in general, experience more heat treat distortion. On the other hand, lower hardenability steels exhibit low distortion but may not meet the design requirements.

Alloy Elements. The higher the alloy content, the larger the heat treat distortion will be.

Method of Quenching. Direct quenching results in lower distortion. However, distortion may increase when gears are cooled after carburizing and then reheated before quenching (double heating cycle and related thermal stress).

Flow of Quench Media. Whether oil or gas, an insufficient and turbulent quenchant flow increases distortion because of slow heat transfer between the gears and quench media.

Temperature of Quench Media. Low quench media temperature increases thermal shock and gear distortion. High quench temperature reduces distortion but may not produce acceptable case properties.

Speed of Quench Media. Low-speed oil media (16 s and lower) causes higher distortion. Higher than 18 s is suitable for low distortion but may not produce the desired case microstructure.

Gear Blank Design (Solid or Webbed Construction). Solid construction seems to have smaller distortion than the webbed configuration. However, larger gears are usually webbed in order to reduce weight and often have asymmetric sections as well. Distortion of this type of gear blank could be large. Again, webbing is needed for uniform heat treatment tooth along the face width. Without web, the center of tooth does not have the same metallurgical properties as the end of a tooth as depicted in Fig. 31.

Vacuum-Melted versus Air-Melted Alloy Steels

Because of high strength, ductility, and toughness, alloy steels are frequently used for gears, particularly in aerospace and other critical applications. These steels are available in either vacuum- or air-melted condition. Vacuum-melted steels have fewer nonmetallic inclusions, are much cleaner than the air-melted variety, and offer several advantages as discussed previously. Even then, gear engineers are reluctant to use these types of steel due to higher material cost per pound without any consideration for the total manufacturing cost. Vacuum-melted steels offer optimum designs in several applications because of low heat distortion that significantly reduces gear grinding time. Also, the rejection rate of gears after carburizing for improper case microstructure is very low. It is advisable, therefore, to consider all pros and cons before selecting a proper gear material. To enable design engineers to decide on the type of material, it is considered helpful to summarize the major benefits of vacuum-melted steels:

- Lower unwanted gas content
- Fewer nonmetallic inclusions
- Less center porosity and segregation
- Increased transverse ductility
- Increased fatigue properties

Typical average gas contents (wt%) of two widely used steels before and after consumable electrode vacuum melting, as determined by the vacuum fusion technique, are shown in Table 12. It can be seen that the gas contents of AISI 4340 and 9310 steels have been reduced to extremely low levels by vacuum melting. The low hydrogen content in steels reduces the susceptibility to flaking, while the low oxygen content reduces the amount of oxygen available for formation of nonmetallic inclusions.

There are two methods available for comparing the cleanliness of vacuum-melted and air-melted steels. One is the magnetic particle test that has been extensively used in the aircraft industry for rating steels such as AISI 4340 and 9310. The other method for rating cleanliness is thorough microexamination, whereby the number, sizes, and types of nonmetallics are counted.

The magnetic particle inspection of vacuum-melted steel requires that indications of inclu-

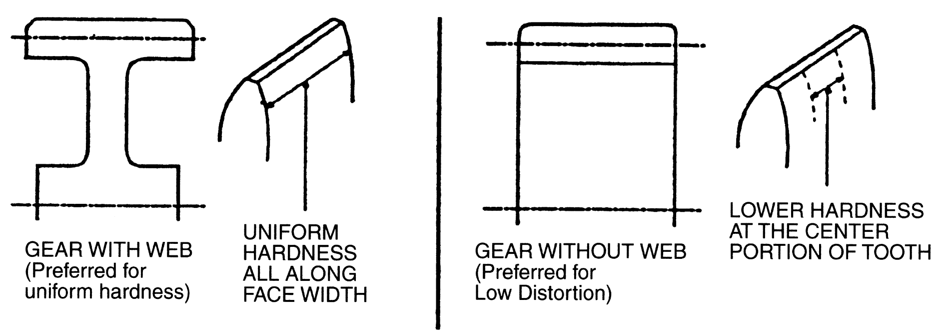

Fig. 31 Effect of gear blank web on uniformity of tooth hardness

Table 12 Gas contents of two gear steels before and after consumable electrode vacuum melting

	Gas content, wt%					
	Before vacuum melting			After vacuum melting		
Alloy	H_2	O_2	N_2	H_2	O_2	N_2
AISI 4340	0.00014	0.0025	0.010	0.00009	0.0004	0.0053
AISI 9310	0.00019	0.0035	0.011	0.00008	0.0006	0.0072

sions as small as 0.40 mm (1/64 in.) long be counted, whereas, in testing air-melted steel, it is required only to count indications down to 1.6 mm (1/16 in.) in length. A comparison of vacuum-melted, and air-melted heats of AISI 4340 and 9310 steels was made where indications down to 0.40 mm (1/64 in.) in length were counted in the steel produced by both vacuum and air melting. In every case, the vacuum-melted steels were 5 to 10 times cleaner than the air-melted products. The Jern Konntoret (JK) chart (Fig. 32) described in ASTM E 45 is one method for rating cleanliness of steels. Row 1 in this figure shows the least amount of inclusions while row 5 has the highest amount of inclusions.

Improvements in fatigue resistance are directly related to the cleanliness of steel. It has been shown that large nonmetallic inclusions lower the fatigue strength of a metal. In vacuum-melted steels, there are no large nonmetallic inclusions. This increases their fatigue strengths. The transverse ductility of steels produced in a vacuum also shows improvement, especially at the ultra-high-strength level. The improvement in transverse ductility of AISI 4340 (AMS 6485) and 300M steels after vacuum melting is denoted by higher elongation as shown in Table 13. These materials are extremely brittle at the 2070 MPa (300 ksi) strength level in the air-melted condition; however, after consumable electrode vacuum melting, they have sufficient transverse ductility to be fabricated into aircraft landing gears and missile components. The higher, more uniform quality of the consumable-electrode vacuum-melted steels makes them popular for aerospace applications where fatigue life (over 10 billion cycles) and high reliability are frequently a requirement.

Cost of Material. Although air-melted steel costs are approximately 50% less, the use of VAR steels permits cost savings in several areas:

- Magnetic particle test rejections are reduced to almost zero. Each finished gear that is

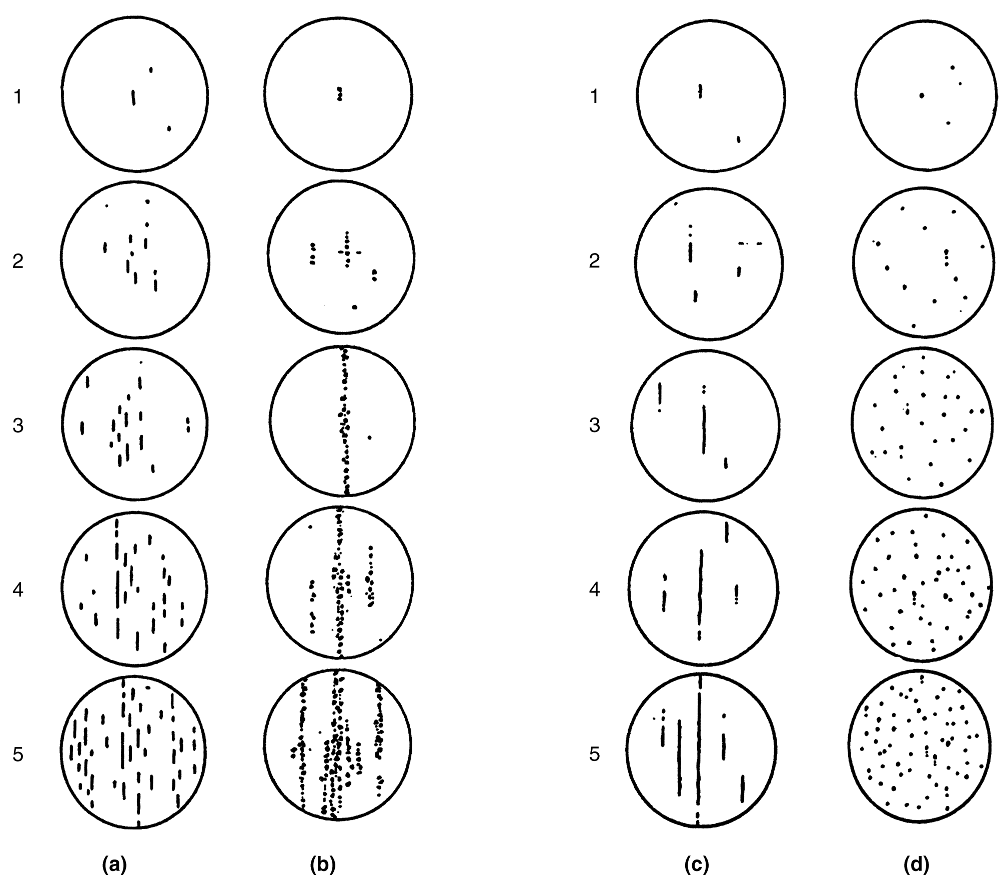

Fig. 32 Jern Konntoret (JK) chart for determining the inclusion in steel. (a) Sulfide-type inclusion. (b) Alumina-type inclusion. (c) Silica-type inclusion. (d) Globular-type inclusion

scrapped due to mag-particle inspection represents a loss of 10 to 30 times raw material cost.

- VAR material would eliminate the need for forged blanks in gear sizes under 130 mm (5 in.) outside diameter. Forging dies and forgings are a significant cost item in gears.
- VAR material can be procured with about three times closer control on chemical composition. Heat treat practice, a high source of gear rejections, can be much more easily perfected by having large quantities of steel that are almost exactly the same composition and hardenability.
- Problems of "banding" (grains not properly diffused in certain areas of a forging or rolled bar) and alloy segregation can be eliminated. In gear blanks made of air-melted steels, such a condition may exist. Banding is the result of improper solidification of steel in an ingot and may cause reduced core hardness in the range of 1 to 2 points in HRC scale. Minor banding is generally eliminated during subsequent heat treatment, particularly normalizing and austenitizing as diffusion occurs during these processes. Severe banding that occasionally exists in air-melted steels destroys material microstructure, and the hardness may be reduced as high as 10 HRC points, which are detrimental to gear life.
- Early failure of a gear due to undetected nonmetallic inclusions is minimized.
- VAR material gears carry more load than air-melted steels because there is only about $\frac{1}{20}$ the amount of macro and micro impurities present. Allowable gear rating could be raised when VAR material is matched up with a skilled and disciplined heat treat practice. For example, a 1200 hp gearbox (gears made of air-melted steel) may be upgraded to transmit 1500 hp with the same center distance if gears are made of VAR steels, provided other components such as bearings and shafts in the gearbox can carry the additional load.

Measurement of Gear Distortions

The distortion associated with gears is usually characterized by the following changes in gear geometry:

- *Profile change:* Due to greater growth near the root of gear teeth than at the tips giving the appearance of collapsed tips
- *Lead change:* A condition, experienced quite often, with gears of wide face width resulting in a change of tooth helix angle (spur gear tooth: 0° helix angle)
- *Size change:* Determined by measurements of pitch diameter (PD) before and after heat treatment. Change in dimension over or between pins bears a direct relationship with the change in PD of gears.
- PD runout
- *Change in blank dimension:* Flatness of rim and hub

Some Recommendations to Minimize Distortion

Steel Cleanliness. Material needs to be as clean as possible (AMS 2300 or 2304), preferably vacuum melted and of homogeneous com-

Table 13 Comparison of transverse mechanical properties between air-melted and consumable-electrode, vacuum-melted steels

Alloy	Type of melt	Tempering temperature		Tensile strength		Yield strength		Elongation, %	Reduction of area, %
		°C	°F	MPa	ksi	MPa	ksi		
AISI 4340	Air	230	450	1944	282	1586	230	6.0	14
		480	900	1379	200	1193	173	8.0	16
		538	1000	1241	180	1103	160	10.0	22
	Vacuum	230	450	1931	280	1634	237	6.5	17
		480	900	1379	200	1207	175	9.0	20
		530	1000	1241	180	1103	160	10.5	24
300M	Air	315	600	1958	284	1620	235	5.0	11
		425	800	1758	255	1538	223	7.0	14
		538	1000	1586	230	1482	215	9.0	22
	Vacuum	260	500	2020	293	1620	235	7.0	25
		425	800	1758	255	1551	225	10.0	34
		538	1000	1586	230	1482	215	11.0	35

position—alloy segregation or banding is to be avoided. AMS 2300 is highly recommended for cleanliness. If difficulty is experienced in procuring material under this specification, then AMS 2304 may be considered.

Lot Verification. The material in each lot of gears shall conform to the specification on the drawing. A lot shall consist of gears of one part number, manufactured from one heat of steel, and should not be mixed with another heat of steel during heat treatment. Each part must be identified with the lot number.

Heat Verification. Verification of conformance of heat for the steel shall be performed prior to manufacture of gears. The verification shall include chemical analysis, hardenability check, magnetic particle inspection rating, microinclusion rating, and macroetch examination, as applicable.

Grain Flow. When a forging is required for an application, the forge die and upset ratio shall be determined by examination of a sample forging prior to production. Evaluation for approval shall be made on a section cut through the centerline, or longitudinal axis of the sample forging. After approval, the forge die and upset ratio shall not be changed. Figure 33 shows some acceptable and unacceptable grain flow patterns in a gear forging. Grain flow parallel to gear axis is not acceptable.

Proper normalizing of material is essential prior to machining with subsequent microstructure analysis for verification. It is an important operation for uniformity of grain structure. It is carried out by heating gears to the austenitic region and then cooling in "still air." The normalizing temperature should be selected to slightly exceed the carburizing temperature, so that grain size changes will not again occur during carburizing. Normally, carburizing is performed between 870 and 930 °C (1600 and 1700 °F) for which an optimum normalizing temperature would be between 940 and 950 °C (1725 and 1750 °F).

Materials after normalizing with hardness above acceptable machining range (HRC 34) may be annealed at a lower temperature to obtain proper hardness. Grain size between ASTM 5 and 7 (Fig. 34) after normalizing is quite effective for low distortion. Test coupons must accompany parts.

Normalizing refines grain structure, whereas annealing reduces hardness. Gears may be normalized before or after rough machining. An adequate normalizing procedure is as follows:

1. Load the part into a furnace that is at or below 870 °C (1600 °F), if part is of uniform section size. Limit the furnace tem-

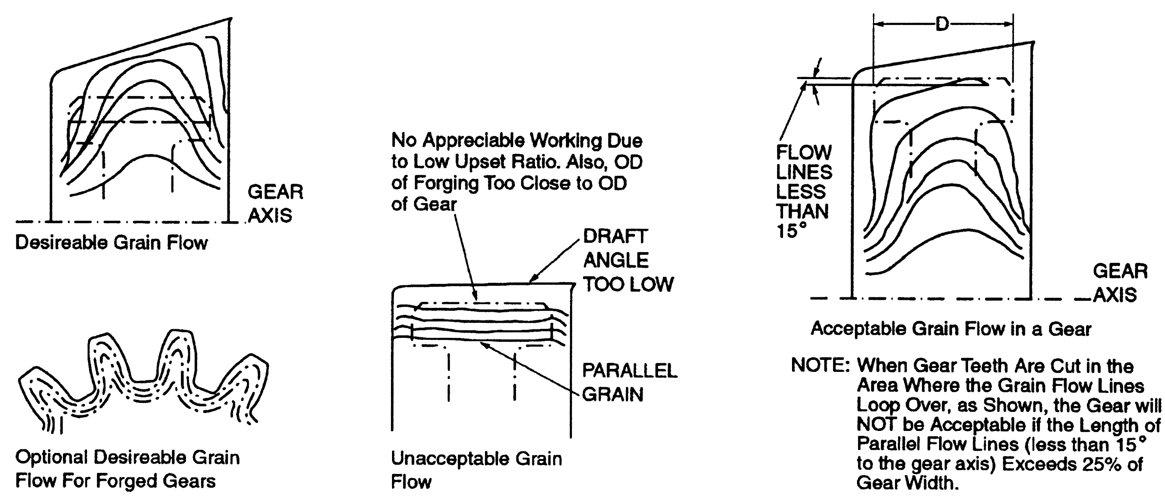

Fig. 33 Acceptable and unacceptable grain flow in a gear forging

perature to 540 °C (1000 °F) maximum, when loading a part with more than one section thickness (to avoid cracking).
2. Furnace temperature rise shall not exceed 170 °C (300 °F) per hour.
3. Normalizing temperature will be between 940 and 950 °C (1725 and 1750 °F)—slightly above the carburizing temperature planned, not by more than 30 °C (50 °F).
4. Hold at temperature for ½ h of maximum section thickness or 2 h, whichever is greater.
5. Remove from furnace and allow to cool in still air. Do not accelerate or inhibit air movement.
6. Stress relieve before gear cutting. Stress relief is also suggested after rough gear cutting for gears below 5 DP.
7. Uniform carburizing atmosphere: The entire load needs to be at a narrow temperature band (930 ± 8 °C, or 1700 ± 15 °F) to ensure uniform result. The vacuum furnaces are generally capable of maintaining temperature within ±6 °C (±10 °F) in the work zone.
8. High, uniform laminar flow (225 to 300 L/min, or 60 to 80 gal/min, for each 0.45 kg, or 1 lb, of gears) of quench oil, preferably an oil that allows quenching at approximately 120 °C (250 °F) for acceptable hardness profile of gear teeth; gas quench if carburized in a vacuum furnace.
9. Speed of oil: approximately 18 s
10. Press quench: This and other means of quenching in fixture such as plug, and cold die are the most effective methods used to keep the gears flat and minimum PD runout.

Load arrangement in the furnace has a significant effect on the outcome of directly quenched gears. Small, lightweight, more intricate gears should be loosely hung on rods mounted on special carburizing fixtures or laid flat on their faces. Larger and heavier gears should either rest vertically on their teeth with overhead wiring for added stability or lie flat on the faces.

When arranging a load for heat treatment in an in-and-out furnace, it may be necessary to experiment in order to find the best method suitable for the gears in question. In one load arrangement (Fig. 35), 510 mm (20 in.) outside diameter (OD), 98 tooth spur gears made from forging

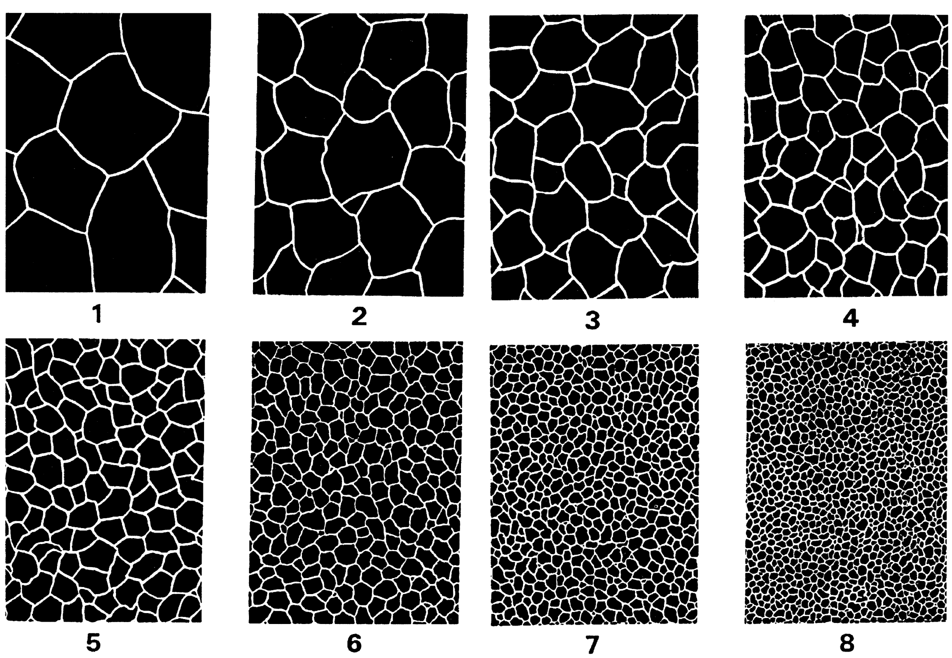

Fig. 34 Standard ASTM grain sizes of steel

(AISI 8620) with 13 mm (½ in.) face width (gears numbered 1 through 5 and 14) distorted excessively, while those numbered 6 through 13 did not. During the production run, instruction was given not to place gears in the positions with excessive distortion. This improved overall quality of gears after heat treatment.

In addition to these factors, control of the carburizing process itself is considered vital for effective carburizing with minimum distortion. There are many ways in which a carburizing furnace and its controls can malfunction. For instance, failure of a thermocouple can cause the process to go out of control; soot buildup in the furnace can interfere with carburizing reactions. An operator can detect these and many other relatively common occurrences and take corrective action before there is an adverse effect on parts in the furnace. The following is a partial check list of items to be considered in setting up the furnace for production:

- *Temperature distribution:* Hot or cold spots in the furnace are not acceptable. Thermocouples need to be properly installed.

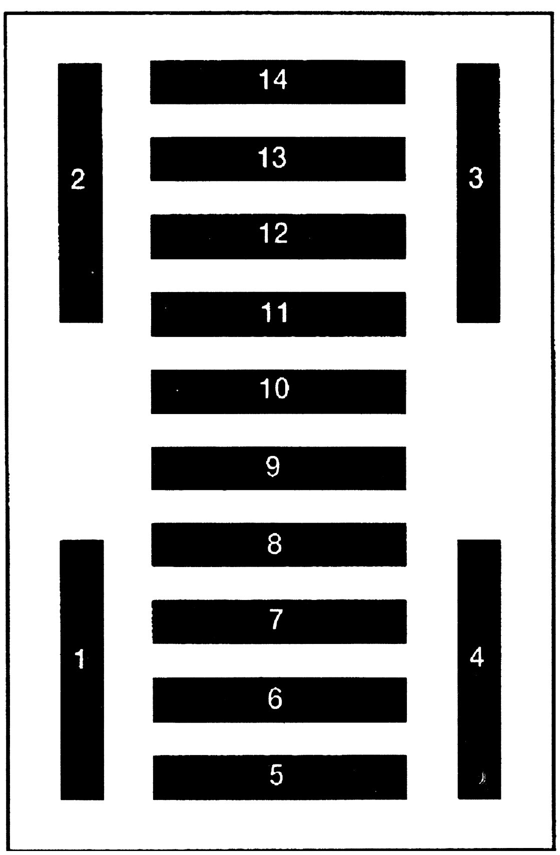

Fig. 35 Arrangement of 510 mm (20 in.) OD spur gears in a carburizing furnace

- *Variation of carbon concentration:* Test bars for carbon determination should be placed at varying locations on trays with gears to be carburized. Results of preliminary tests are to be analyzed statistically to ascertain the process capability of the furnace.
- *Carburizing atmosphere:* For shallow cases, most of the enriching gas is to be delivered into the furnace close to the discharge end. For deep cases, it is to be nearer to the charging end, keeping in mind that in either instance, the enriching gas should enter at the point where the work load has reached the carburizing temperature.
- *Dew point control:* It should be ascertained whether the last furnace zone (diffusion zone) can develop the required dew point, with some latitude for adjustment. A small, measured quantity of air may be introduced to increase the dew point and lower the carbon potential.

Preheating of Gears

It is advisable to preheat gears prior to loading into a carburizing furnace. This process generally is done in a draw furnace. Preheating helps to control distortion that results from introducing gears from room temperature directly to the austenitizing temperature.

Distortion Characteristics of Some Gear Materials

The distortion characteristics of carburizing grade gear materials depend on a large number of variables. Research work on distortion of high-strength steels suggests the degree of distortion depends not only on the steel composition, but also on the type of heat treat process. For example, step quenching, or so-called ausbay quenching, significantly reduces distortion by as much as a factor of three compared with normal quenching procedure. Furthermore, heat treating the steel to a bainitic structure compared with martensitic structure reduces the degree of distortion. Table 14 shows comparative distortion ratings of some commonly used carburizing grade gear steels in industrial and aerospace applications.

An analysis of data presented in Table 14 shows gear materials that exhibit good mechanical properties with acceptable distortion generally cost more than materials with higher distortion. But the finishing cost of gears made of high distortion materials is significantly higher. Thus, it is important to consider all aspects before selecting a proper gear material. From mechanical properties, manufacturing, and cost points of view, AISI 9310 is considered to be one of the best materials currently available. This material is available both in vacuum- and air-melted conditions. Vast field data of successful applications are also available. However, AISI 8620 steels are extensively used for industrial gears because of their low cost. It is found that gears up to 410 mm (16 in.) OD made of these materials could maintain AGMA class 9 gear quality after carburizing and hardening.

Improvement in Gear Design to Control Heat Treat Distortion

Gear blank design has a profound effect on distortion. To minimize distortion, blank design should address not only uniform metallurgical properties of teeth along the face width but also distortion. For example, gear blanks as depicted in Fig. 36 are found to distort more than blanks shown in Fig. 37 under similar heat treat condi-

Table 14 Distortion ratings of common gear materials

Material	AMS specification	AMS quality	Quench media	Hardness, HRC		Distortion rating(a)
				Case	Core	
AISI 8620	6276, 6277	2304	Oil	58–60	28–32	Fair
AISI 9310/9315	6265(b), 6414(c)	2300/2304	Oil	58–62	34–42	Fair
AISI 4320	6299	2300/2304	Oil	58–60	36–40	Fair
AISI 4330 M(e)	6411, 6429	2300	Oil	58–62	42–50	Poor
AISI 4130	6370	2300/2304	Oil	58–62	34–42	Poor
HP 9-4-30(e)	6526	2300	Oil	58–62	48–52	Good
M50 Nil	6490D	2300	Oil	58–62	35–45	Good
VASCO X2M	N/A(d)	2300	Oil	58–62	38–44	Good
CBS 600	6255	2300	Oil	58–62	34–42	Good
AISI 4620	6294	2300/2304	Oil	58–62	34–40	Poor
Pyrowear 53	6308	2300	Oil	59–64	36–42	Good

(a) Distortion ratings: poor, degradation of AGMA quality by 2 to 3 classes; fair, degradation of AGMA quality by 1 to 2 classes; good, degradation of AGMA quality by 0 to 1 class. (b) AMS 6265, consumable electro-vacuum-melted steel. (c) AMS 6414, consumable electro-remelted steel. (d) N/A, not available. (e) Effective case depth determination is difficult due to high core hardness of these materials.

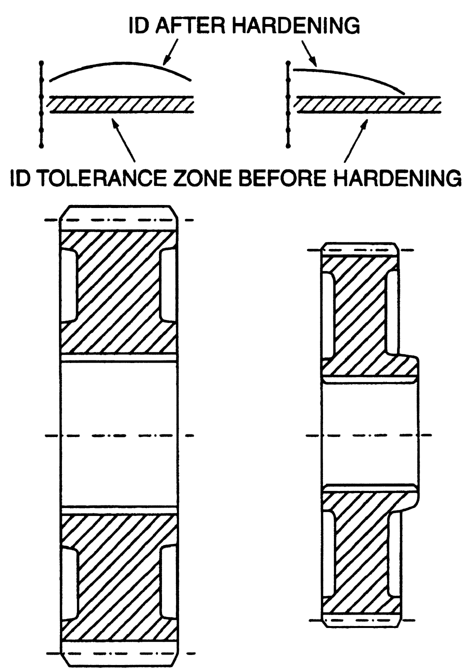

Fig. 36 Standard gear blanks for high distortion

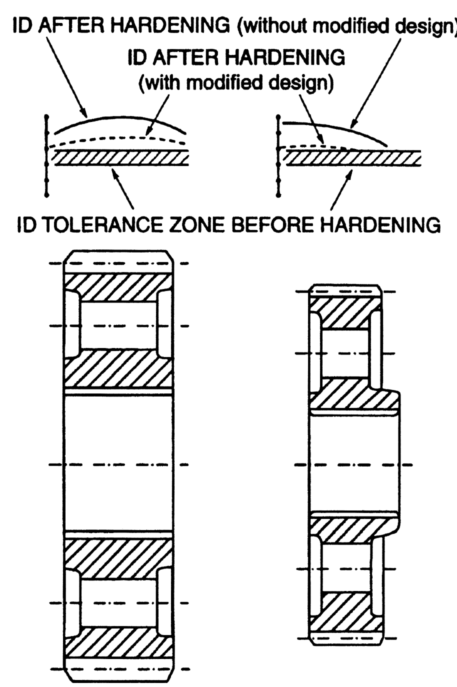

Fig. 37 Improved gear blanks for low distortion

tions. Holes and reduced mass are beneficial for good heat treatment and low distortion. Pinion teeth with blank ends as illustrated in Fig. 38(a) seem to have large distortion of lead. Such distortion can be minimized with a small properly designed relief at the end of the teeth as shown in Fig. 38(b).

Carburizing of Long Slender Pinions. In this type of pinion, Fig. 39(a), after carburizing and hardening, in addition to distortion of tooth geometry, the shaft bows. Such a bow can be minimized by press or die quenching the pinions. Furthermore, if the design of the pinion is such that the teeth are located at the center of shaft, they will experience the maximum bow as illustrated in Fig. 39(b). Because the teeth need to be ground for higher quality (AGMA class 10 and above) with respect to the centerline of bearing journals, which are at the minimum bow position, grinding may remove most of the case from the teeth. To ensure minimum case removal from teeth after grinding, pinion shafts of this type are sometimes straightened under a press before grinding of the journals and teeth with proper control. An uncontrolled straightening operation could damage the parts for the following reasons:

- Stress induction during straightening may initiate cracks in the hard case.
- After straightening, the part can be in an unstable condition and is likely to return, at least partially, to its unstraightened shape when put into service, causing improper contact between meshing teeth that may deteriorate gear life.

Because straightening is such a potentially damaging and expensive operation, everything practical should be done to eliminate the need for it. If it is still necessary, a reasonably satisfactory solution to the problem is to heat the parts to just below tempering temperature and straighten, followed by air cooling to room temperature. All parts shall then be magnetic particle or dye penetrant inspected for cracks.

Dumb-Bell Shape Pinion. The design as illustrated in Fig. 40 presents a high degree of difficulty in controlling heat treat distortion after carburizing and hardening. For acceptable distortion, the ratio between largest and smallest outside diameters in the shaft is recommended to be less than 1.5:1. Pinion configuration of this type should be avoided, if possible.

Blank Design for Uniform Hardness of Wide Face Width Gears. Of the various parameters that control the heat treat quality of carburized and hardened gear teeth, the rate of heat transfer between the teeth and quench media plays an important role. In general, besides material hardenability, fast heat transfer results in acceptable surface hardness, effective case depth, and core hardness of gear teeth. In achieving these properties, configuration of the

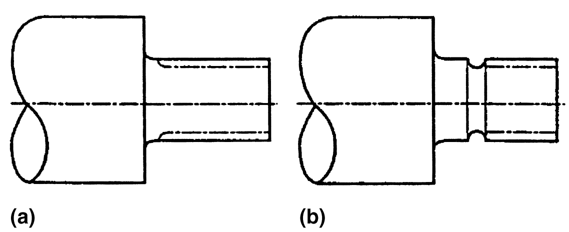

Fig. 38 Improved pinion design for low distortion. (a) Original design (blind end teeth). (b) Improved design (open end teeth)

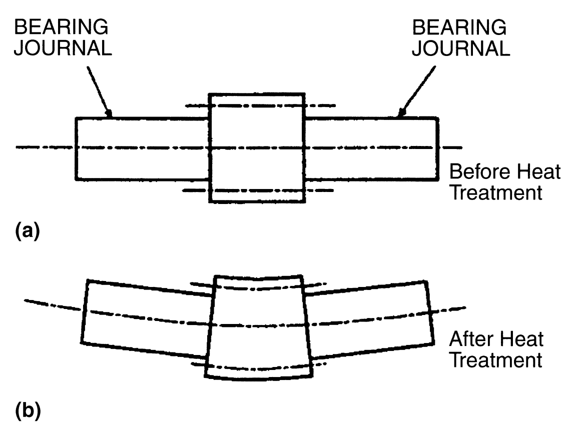

Fig. 39 Distortion of slender pinion. (a) Before heat treatment. (b) After heat treatment

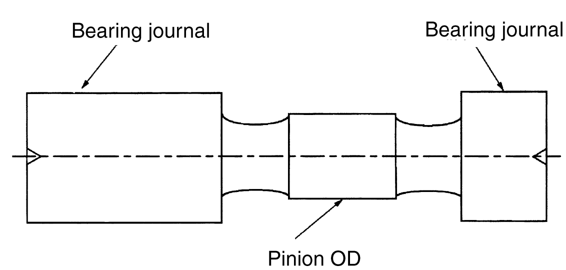

Fig. 40 Pinion blank dumb-bell-type configuration

gear blank (with or without web), size of gear, face width of tooth, and DP also contribute significantly. Design of a gear blank with or without web depends largely on the form of material used—bar or forging. Bars are used for small gears, whereas large alloy steel gear blanks (above 250 mm, or 10 in., diameter), in particular for high production, are made primarily from forgings with web. Occasionally, for small production quantities, large gear blanks with web are machined out of "pancake" forgings. In any case, when large gears with or without web are carburized and hardened, hardnesses at the surface and core, and case depth are found to be lower at the center region "a" of tooth face width (Fig. 41a) than at the outer regions "b." Figure 41(b) illustrates an approximate difference in surface hardness versus case depth gradient at "a" and "b". On the other hand, such a difference is not greatly pronounced in small gears (less than 250 mm, or 10 in., diameter) because of their smaller mass. Nevertheless, with high *l/d* (face width/pitch diameter) ratio over 1.5:1, lower case depth and hardness result at the center region of teeth in all gears, large or small, the variations being quite distinct in large gears.

The cause of lower effective case thickness and core hardness is attributed mainly to differential heat transfer rate during quenching—slowest at the center region. Large mass of the web below the center region acts as a barrier to fast heat transfer rate required for high hardness and uniform case depth along the face width. Without web (solid blank) the differential heat transfer extends from the outer ends to the center region of teeth that results in higher hardness and case depth at the ends gradually decreasing toward the center.

Although small variations in hardness and case thickness along the face width may not be of any concern for gears in industrial applications, these can play an important role in ensuring the reliability of carburized and hardened gears in critical applications. An improvement to the blank design is recommended to ensure both uniform hardness and case depth along the face width of gears.

This can be partly achieved by designing a blank with a small fillet radius at the web that is acceptable from a stress concentration point of view. Also, heat transfer from the middle section of the tooth could be improved by drilling small holes in the tooth spaces as illustrated in Fig. 42. Small fillet radius and holes in tooth spaces not only improve uniformity of case and core hardnesses along the face width of tooth

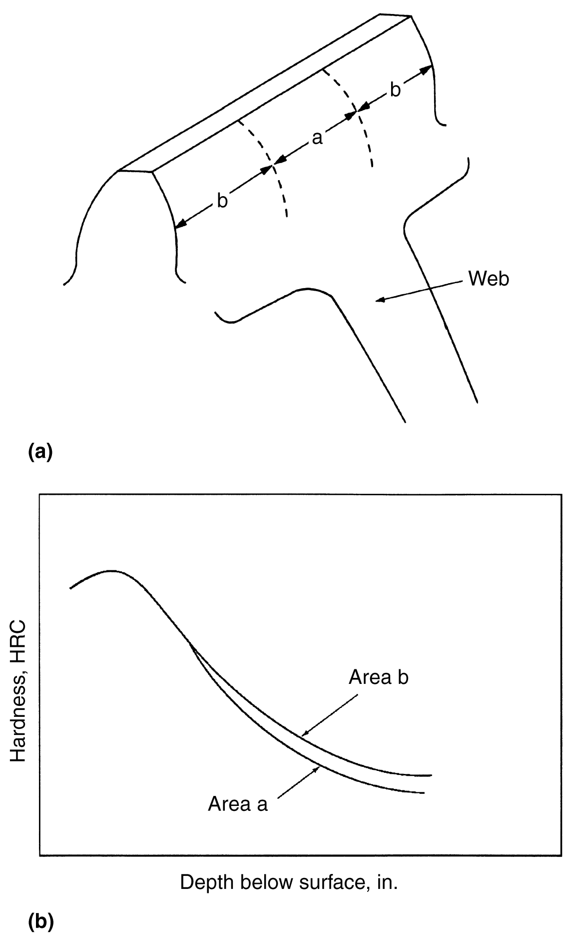

Fig. 41 Effect of blank design on hardness. (a) Typical webbed gear tooth. (b) Surface hardness vs. case depth at different locations of a tooth along the face width

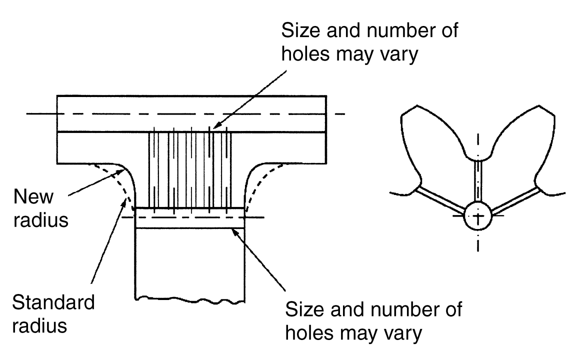

Fig. 42 Drilling of blank for uniform core hardness

but also provide better lubrication, especially in high-speed gearing with pitch line velocity above 50 m/s (10,000 ft/min).

Pinion Shaft with Threads. The threads on a pinion shaft as depicted in Fig. 43 grow and distort similar to pinion teeth after carburizing and hardening to the extent that may require the threads to be ground. To eliminate such a costly operation, it is advisable not to carburize the threads, which could be achieved either by copper plating or covering with noncarburizing-type solution before carburizing. Any small distortion during heat treat process may be removed by thread chasing, a fairly inexpensive operation compared with grinding.

Heat Treat Cracks at Tooth Edges. In helical gears, unchamfered tooth edges, which, in particular, form acute angles with the face width (Fig. 44), are frequently found to develop microcracks after carburizing and heat treatment. These types of cracks are more pronounced on the side that enters the quench media first (Fig. 45). This is due to high thermal shock and stress—the cooler the quench media, the higher the shock.

Experimental investigations showed that cracks emanating from the tooth edges could be avoided by a proper chamfer (Fig. 46) that helps to eliminate the sharpness of the acute angle edges. In chamfering the tooth edges, it is important to note that the amount of chamfer may not be equal on the acute and obtuse angle sides—smaller on the edge with the acute angle. Because cracks are found primarily on the edge with acute angle, chamfering on this edge is important. The following chamfer dimensions are found to be adequate and recommended for avoiding cracks from tooth edges of helical gears:

- *4 DP and lower:* 1 mm (0.04 in.) × 45°
- *Above 4 DP:* 0.75 mm (0.03 in.) × 45°

Also, quenching of gears hanging from fixturing rods (Fig. 47) helps to reduce cracks at tooth edges. Certainly, this method of quenching is limited to smaller gears with low mass; large

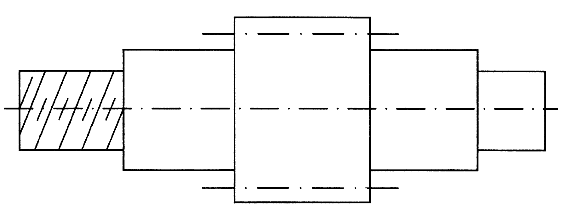

Fig. 43 Pinion design with threads in the shaft

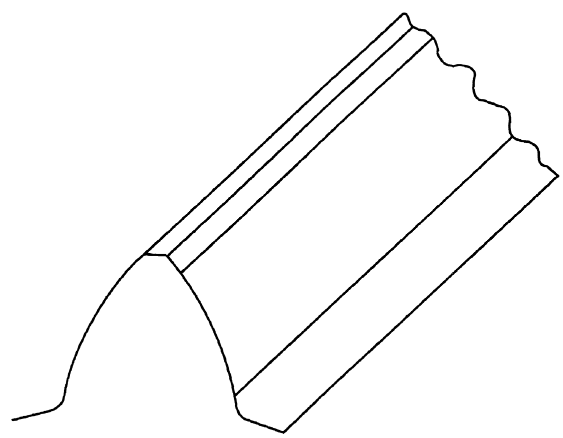

Fig. 44 Helical tooth without chamfer on tooth edges

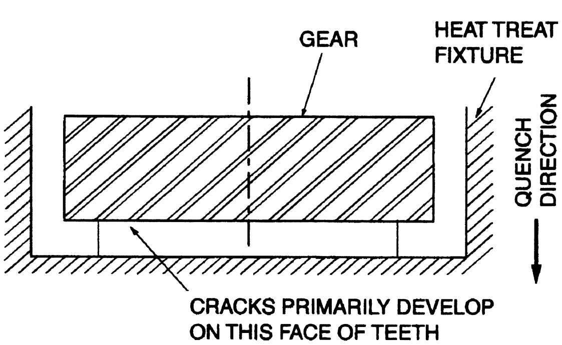

Fig. 45 Gear quenched in flat position

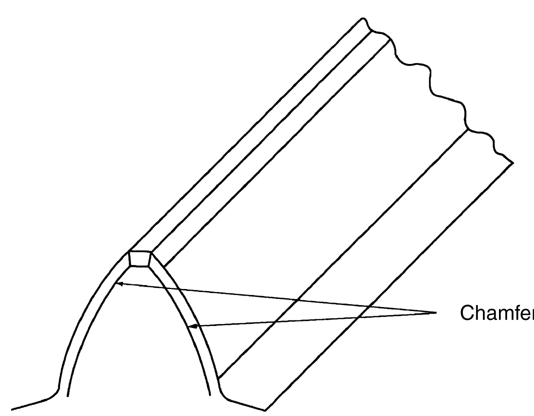

Fig. 46 Helical tooth with chamfer on tooth edge

gears with heavy mass are not suitable because of higher distortion (PD runout). Cracks of this nature are not common with spur gears because spur gear tooth edges do not form any acute angles.

Grinding Stock Allowance on Tooth Flanks to Compensate for Distortion

For gears that do not meet the quality requirements after heat treatment and grinding is required, additional stock is provided on tooth flank. For a ground root, stock is also needed on the root. Grind stock is determined on the basis of cleaning of all the surfaces of teeth considering distortion and growth of gears after carburizing and hardening. The grind stock allowance on gear teeth should not deemphasize the processing control required for minimum distortion. In fact, the control is just as necessary for gears that will be ground as it is important for gears that are not ground after heat treat. The reason is that if the distortion is high and corrected by grinding, it may result in uneven case depth or sometimes no case at all on some teeth. Furthermore, the addition of extra stock increases grinding time and cost. So, for good quality gears at optimum cost, the distortion during heat treat needs to be controlled. Distortion control will not only allow minimum removal of stock during grinding but also ensures "sound" gears from the metallurgical viewpoint.

In a well-controlled carburizing and hardening process, distortions could be held to within one AGMA class below the pre-heat-treat quality for gears made of commonly used materials such as AISI 4320H and 8620H. Such a process allows the establishment of a minimum grind stock on a tooth surface and the maintenance of the minimum surface hardness required after grinding. This satisfies the design requirement for minimum gear pitting life in most applications. Unfortunately, the state-of-the-art carburizing and quenching technology is unable to assure low gear distortion consistently in a production mode of operation. Sometimes, distortions within the same batch of gears are found to vary widely. Hence, it is very likely to find some gears after heat treatment with distortion far more than others. Grinding of such gears for minimum stock removal necessitates inspection of every gear in a batch for distortion severity. This increases the cost of gears. Furthermore, tooth surface hardness and effective case depth may not be acceptable after grinding.

Besides grinding, surface hardness reduction of carburized and hardened teeth is also influenced by the slope of hardness versus case depth gradient of gear steel. Each carburizing grade steel has its own unique slope and is primarily controlled by its alloying elements. The steeper the slope is, the greater the hardness reduction on tooth surface after grinding will be. In addition, grinding machine setup plays an important role in the equal amount of stock removal from each flank of the gear teeth and its associated hardness reduction.

To establish an ideal grind stock, detailed distortion characteristics and growth data of gears are beneficial. For gears without any preproduction or historical data on heat treat distortion, general knowledge on heat treat response of the gear material helps to establish effective grind stock.

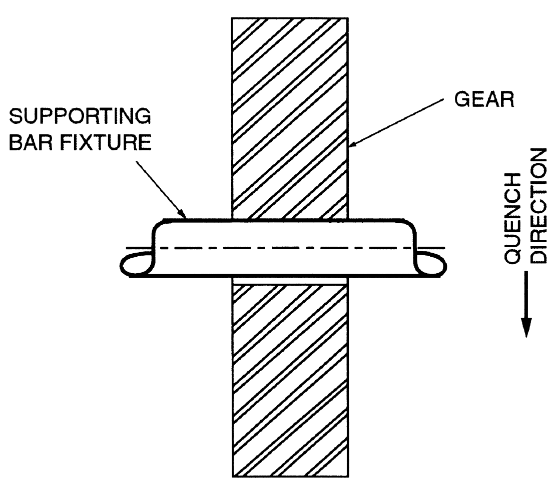

Fig. 47 Gear quenched hanging from bar

Grinding of Distorted Gears

With the grind stock so determined either by a preproduction investigation or from data of similar gears, grinding of carburized and hardened gears is performed by properly positioning the grinding wheel in the tooth space. In modern gear grinding machines with advanced measurement systems, a probe measures the helix (lead) deviation on a predetermined number of right and left flanks of teeth at different locations that include teeth at maximum PD runout. The measured values not only contain the effect of all helix deviations and PD runouts, but also the effect of profile and cumulative pitch variations. The computer in the machine control calculates the optimum angular position of the gear before grinding starts. This method ensures that the grinding wheel will be positioned at the center between two flanks, allowing equal stock removal from each flank starting with teeth at the maximum PD runout. The method works well as long as the distortions are comparatively small and their variations are distributed within a narrow band—one AGMA class below the pre-heat-treat quality.

For gears made of high-alloy steels such as AISI 9310 and AISI 8620 that exhibit uncontrolled distortion after carburizing and hardening, the selection of an effective grind stock is quite difficult. Tests carried out show gears made of these materials have larger variations of PD runouts and spacing of teeth. Furthermore, the distortion seems to vary in a random manner. Hence, considerable difficulty is experienced for equal stock removal from all the flanks of distorted teeth even with modern grinding machines. This leads to more-than-planned stock removal from some teeth flanks during grinding as illustrated in Fig. 48.

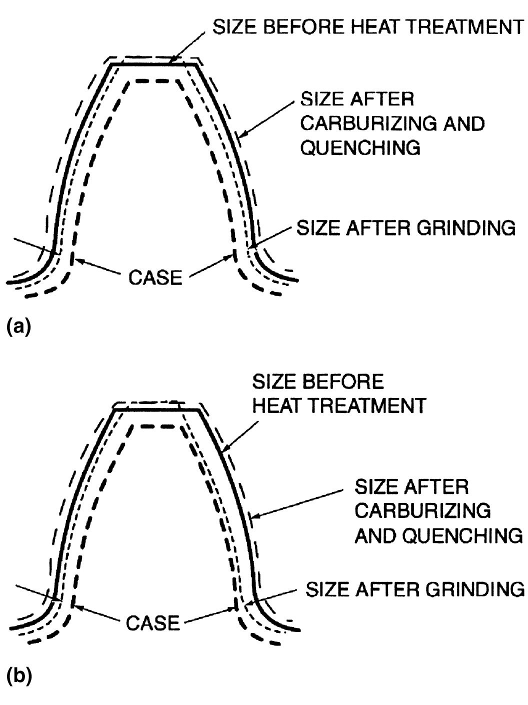

Fig. 48 Schematic of materials ground from a carburized and hardened gear tooth. (a) Tooth with no distortion; equal stock removal from both flanks. (b) Tooth with distortion; more stock removal from one flank

Actual Stock Removal and Tooth Surface Hardness

The more stock that is removed from the surface of a carburized and hardened tooth, the lower the tooth surface hardness will be. The degree of hardness reduction is dependent also on the slope of the hardness gradient of the tooth. Figures 49 and 50 illustrate hardness gradients of two low-alloy steels, the first one for 17CrNiMo6 and the other for AISI 8620H after carburizing and hardening. In comparison with 17CrNiMo6, the hardness gradient of 8620H exhibits a lower hardness at the tooth surface followed by a gradual increase in hardness with the maximum at a depth of 0.05 to 0.08 mm (0.002 to 0.003 in.) below the surface. This is believed to be due to higher than normal per-

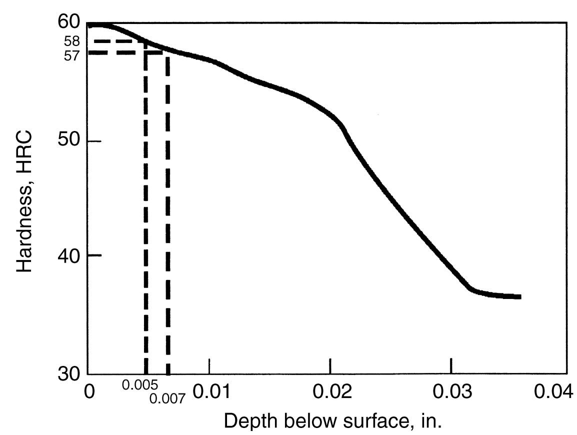

Fig. 49 Hardness gradient of a carburized and hardened gear tooth. Material: 17CrNiMo6

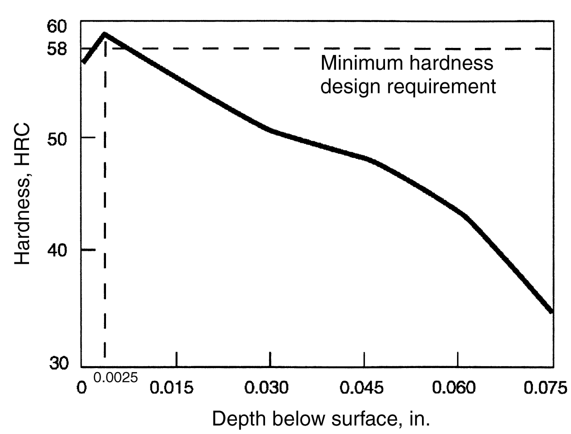

Fig. 50 Hardness gradient of a carburized and hardened gear tooth. Material: AISI 8620H

centage (about 20%) of retained austenite at the surface of these gears after quenching. On the other hand, the percentage of retained austenite after cold treatment is much better controlled in most of the high-alloy steels. Consequently, these materials do not exhibit the characteristic of some low-alloy steels. This is depicted in Fig. 51 for AISI 9310H.

It is clear from the hardness gradients of various alloy steels that stock removal over 0.13 mm (0.005 in.) (normally allowed for grinding) from the flanks of distorted carburized and hardened gear teeth may result in lower than the minimum surface hardness required for the expected pitting life. Because of the steeper slope, a usual characteristic of low alloy steels (Fig. 50), the surface hardness reduction of gear teeth made of these materials is generally higher than those made of high-alloy steels. Again, distortion associated with low-alloy steel gears is comparatively less, and hence, these warrant smaller stock removal during grinding. Thus, the actual tooth surface hardness reduction after grinding is almost of the same magnitude for both low- and high-alloy steel gears. Sometimes, carburizing with boost-diffuse cycle produces a flatter hardness gradient. Because it is generally cost prohibitive, this type of carburizing is very seldom used. This allows the consideration of similar amounts of grind stock for all gear materials. Furthermore, grind stock required on each tooth flank for both low- and high-alloy steel gears with controlled heat treat distortion is found not to exceed an average of 0.13 mm (0.005 in.). Removal of stock to this depth does not seem to reduce tooth surface hardness below the minimum considered in design with carburized and hardened gears.

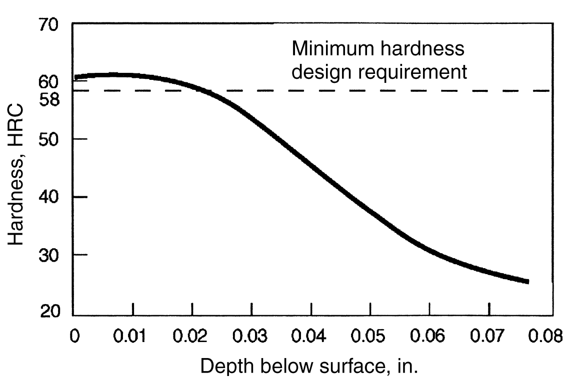

Fig. 51 Hardness gradient of a carburized and hardened gear tooth. Material: AISI 9310H

In a production-type carburizing and hardening operation, which is frequently associated with uncontrolled distortion, stock over 0.13 mm (0.005 in.) is very often removed from teeth located at maximum distortion. Such stock removal in some gears may result in surface hardness reduction as high as two points on the HRC scale, particularly for materials with a steep hardness gradient slope. As an example, in gears made of 17CrNiMo6, the hardness reduction for an additional 0.05 mm (0.002 in.) stock removal over planned 0.10 mm (0.005 in.) is approximately one point on the HRC scale. This is a significant hardness reduction below the minimum considered during design. Lower surface hardness will definitely reduce pitting life. The new pitting life may be calculated from the equation derived from the pitting life versus contact stress relationship.

New Pitting Life

The equation for pitting life in terms of stress cycles (L_1) of a gear at any power level can be estimated as:

$$L_1 = 10^7\left(\frac{S_{ac_1}}{S_c}\right)^{\alpha} \qquad \text{(Eq 9)}$$

where S_{ac_1} is the allowable contact stress for the gear material at minimum hardness, S_c is the actual contact stress due to applied load, and α is the slope of S-N curve for gear durability = 17.62 (AGMA standard 2001-B88). Similarly, the new pitting life (L_2) for a lower allowable contact stress (S_{ac_2}) due to reduced tooth surface hardness of the same gear and at the same power level is:

$$L_2 = 10^7\left(\frac{S_{ac_2}}{S_c}\right)^{\alpha} \qquad \text{(Eq 10)}$$

Equations 8 and 9 yield:

$$L_2 = L_1\left(\frac{S_{ac_2}}{S_{ac_1}}\right)^{\alpha} \qquad \text{(Eq 11)}$$

From Eq 11, the pitting life of a gear can be calculated at any allowable contact stress that corresponds to a specific tooth hardness. For example, if due to additional stock removal (0.05 mm, or 0.002 in., over the allowed 0.10 mm, or 0.005 in.), the surface hardness of teeth reduces from 58 to 57 HRC (Fig. 49), the allowable contact stress (S_{ac_2}) for AGMA grade 1 material (Fig. 52) becomes:

$$S_{ac_2} = 26{,}000 + 327 \text{ HB}$$

Brinell hardness number (HB) of tooth surface $\simeq$ 1475 MPa (214 ksi)

The allowable contact stress (S_{ac_1}) at minimum hardness (58 HRC) per design is 1500 MPa (218 ksi). Using Eq 11 the new pitting life is:

$$L_2 = 0.72\, L_1 \qquad \text{(Eq 12)}$$

This is a significant reduction of gear pitting life. Hence, for true pitting life of carburized, hardened, and ground gears, it is essential to consider heat treat distortion. Certainly, this necessitates the availability of heat treat distortion data for all newly designed gears—a difficult task to accomplish. One possible solution to the problem is to develop distortion-derating factors for various gear materials from known heat treat data and apply them to derate pitting life.

Distortion Derating Factor

The distortion derating factor (DDF) is defined as the ratio of actual pitting life to the required pitting life of gears. For realistic values of distortion derating factors, it requires comprehensive knowledge on composite distortion characteristics that include different gear materials, configuration and size of gears, gear diametral pitch and helix angle, heat treat process and equipment, and hardness gradient of each of these materials. Such a task is beyond the scope of this discussion. Table 15 lists derating factors for some commonly used materials based on gear distortion characteristics. These are based on an additional 0.05 mm (0.002 in.) stock removal above 0.10 mm (0.005 in.) normally allocated. Similar derating factors for other gear materials can be established from their composite gear distortion data.

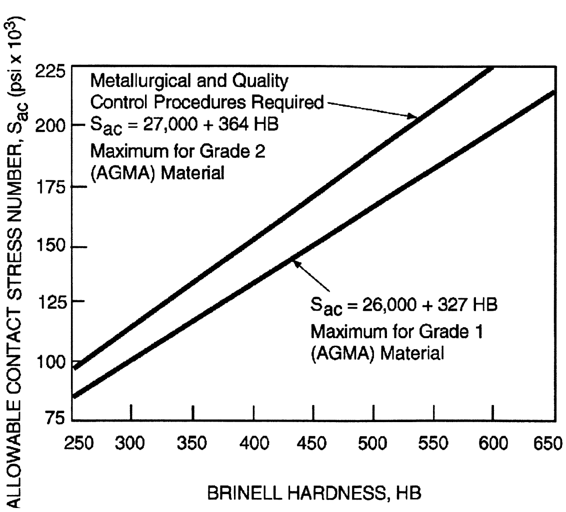

Fig. 52 Allowable contact stress number for carburized and hardened steel

Sometimes, to avoid the task of developing such distortion derating factors, an extra grind stock with additional case depth is suggested. The extra stock no doubt allows grinding of all the teeth to the required quality and geometry but does not help to maintain the surface hardness equally on all the teeth surfaces. Also, deeper case creates some serious heat treat problems for gears. First, it will increase carburizing cycle time and, hence, the cost. Secondly, it does not help to increase case hardness. Finally, a deeper case may cause some severe adverse effects such as:

- Through hardening of teeth, particularly for fine-pitch gears and gears with large pressure angle, above 20°
- Higher surface carbon—a potential source for inducing grind burn
- Additional grinding time—higher cost
- Larger step at the intersection of root radius and tooth profile for unground roots—higher stress concentration
- Increased percentage of retained austenite—lower surface hardness
- Increased level of undesirable carbide network in the case microstructure—mechanical properties of teeth may be affected

Considering all of these potential problems, a deeper case than what is needed is not recommended.

Additional Effects of Grinding Carburized and Hardened Gears

Carburized and hardened gears are generally ground to improve distorted tooth geometry caused by heat treatment. To maintain the required case depth and tooth surface hardness after grinding, a certain amount of grind stock is provided to pre-heat treat tooth flanks, and the carburizing process is adjusted for case depth that includes grind stock. Besides improving the tooth geometry and maintaining an acceptable case depth and surface hardness, grinding has a profound effect on the tooth surface integrity such as grind burn and removal of residual surface compressive stress. Both grind burn on tooth surface and surface compressive stress reduction detrimentally affect the fatigue life of ground carburized and hardened gears. In this section, some discussion on the causes of and remedy for grind burn is presented.

Gear Grind Burns. Gears and, as a matter of fact, any surface that is carburized, hardened, and subsequently ground, are subjected to retempering of localized areas popularly known as grind burn. The reasons for such burns on ground surface are generally attributed to improper carburizing and grinding operation. A large number of researchers have endeavored to identify the causes of the grind burn phenomenon, but none of them has come up with any useful solution to the problem. Nevertheless, the results of all these investigations indicate that a large number of carburizing process and grinding parameters influence grind burn. Carburizing factors that cause grind burn are:

- High surface carbon—0.9% and above
- High surface hardness—61 HRC and higher
- High retained austenite—above 20%

Grinding parameters that can contribute to grind burn are:

- High stock removal rate during grinding
- Sudden increase in stock removal from a tooth surface due to nonuniform heat treat distortion
- Too fine grit of abrasives on vitrified grinding wheel
- High grinding wheel hardness
- Unbalance in grinding wheel
- Infrequent dressing of grinding wheel resulting in glazing of wheel
- Improper type of coolant
- Inadequate flow rate and direction of coolant flow
- Presence of grind sludge in coolant due to malfunction of gear grinding machine coolant filters
- High coolant temperature
- Unstable machine

Any one or a combination of these factors can contribute to grind burn on a gear tooth surface that is ground, possibly by wet grinding that uses vitrified aluminum oxide wheel and cutting fluid. Gears are less vulnerable to such defect when ground in machines that operate dry. Unfortunately, such dry grinding machines are no longer used due to their long grind cycle time. Wet grinding machines recently developed for cubic boron nitride (CBN) wheels are found to be quite effective in ensuring grind-burn-free gear teeth. In any case, improperly carburized gears ground by an improperly controlled grinding process can result in grind burn on tooth surfaces.

Grind burn lowers surface hardness on the burnt areas usually in the range of 2 to 3 points on the HRC scale. A burnt tooth, particularly at the point of maximum contact stress, lowers the contact fatigue life of the gear. Furthermore, a burnt tooth surface is frequently accompanied by microcracks. Although these cracks are very

Table 15 Distortion derating factors for true pitting life of carburized and ground gears

		Hardness versus case depth gradient		Distortion severity		
Gear material	High or low alloy	Hardness drop less than half a point HRC for every 0.05 mm (0.002 in.). Stock overallocated (0.127 mm, or 0.005 in.)	Hardness drop more than half a point HRC for every 0.05 mm (0.002 in.). Stock overallocated (0.127 mm, or 0.005 in.)	Low (within 1 AGMA class below pre-heat-treat quality)	High (over 1 AGMA class below pre-heat-treat quality)	Distortion derating factor (DDF)
Air melt						
4320	High		X	X		0.80
8620H	Low		X		X	0.80
9310H	High	X			X	0.80
17CrNiMo6	Low		X		X	0.75
Vacuum melt						
4320	High		X	X		0.85
4330M	High	X			X	0.80
9310H	High	X		X		0.85
HP 9-4-30	High	X		X		0.90

minute, they still can affect the fatigue life of a gear subjected to high cyclic load or that operating near a resonant condition. To ensure grind-burn free teeth, it is thus essential to inspect gears after grinding.

There are two different methods available in the gear industry to inspect gear burns—one is destructive and the other nondestructive. The destructive method provides a positive identification of burnt surfaces and is based on the microhardness reading of the surface below the burnt area. Such a method is not quite practical, whereas a properly controlled nondestructive method currently available can provide sufficient evidence of grind burn. The most common one is the nital, or temper etch, method.

Gear Grind Burn Identification. A good deal of controversy exists over the accuracy of the temper etch method results. Generally, nital etch inspectors are specially trained to distinguish between burnt and regular tooth surfaces, but sometimes they fail, as was the case with one gear manufacturing company discussed in this section. At one point during normal manufacturing, this company was experiencing continual rejection of carburized and ground gears due to grind burns identified with nital etch. Even after implementing all necessary controls during carburizing and subsequent grinding, the problem of grind burn showed up from time to time. An external metallurgical inspection company was then employed to investigate whether the gears were definitely burnt or not. Figure 53 shows photographs of two such apparently burnt and rejected gears after nital etch. The gears were made of AISI 8625H, carburized and hardened, and the teeth ground on a wet grinding machine. Being unable to draw a definite conclusion, the company employed destructive microscopic examination of sections taken at the burnt areas. Surprisingly, the results did not reveal any rehardening or retempering of these areas at 1000× magnification. It was then concluded that the visible burn indications after nital etching were caused by differential skin etch reflection. A subsequent successful full-load test of the gears confirmed this conclusion. This is why each gear manufacturing company needs to develop its own criteria of acceptance and rejection of grind burns.

Inspection and Limited Acceptance Criteria of Grind Burns. Because it is extremely difficult to grind gears except by CBN grinders without any grind burn in a production type environment, a discussion on inspection and its limited acceptance is considered helpful to establish optimum carburizing and heat treat conditions.

Gears shall be inspected visually for evidence of grind burns following the etching procedure under a light source of 200 footcandles (ftc) minimum at the surface being inspected. Surfaces not burnt will be uniformly gray to light brown in color, depending on the alloy used. Surfaces burnt will appear dark gray to black in color. These could be either overtempered or rehardened areas. Overtempered (burnt) area is due to localized overheating during grinding and will appear dark to black in color. This results in a lower surface hardness. A rehardened (burnt) area is due to transformation of retained austenite to untempered martensite and will appear as a white or light gray spot surrounded by black overheated area. Rehardened areas are usually associated with grind cracks.

The hardness of overtempered areas may be measured with a Rockwell superficial hardness tester using the 15 N scale in accordance with

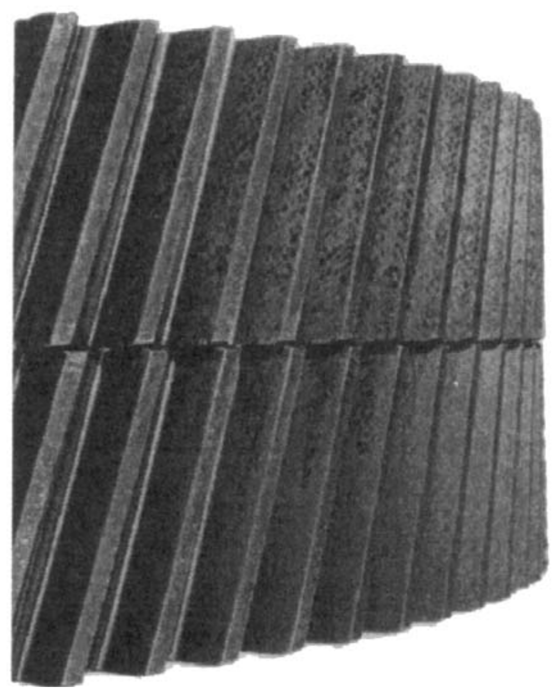

Fig. 53 Experimental gears after nital etch for surface damage. Grinding patterns were observed after surface nital etch on both gears. Microscopic examination of sections removed from the gears did not reveal microstructural changes in the case hardened matrix in either gear.

ASTM E 18. For comparison, similar readings may be taken on adjacent areas of grind burn.

Acceptance Criteria. Basically, there shall be no rehardened and overtempered areas on any ground teeth of fracture-critical gears. Limited overtempering may be acceptable on non-fracture-critical gears. A number of aircraft and non-aircraft industries have been consulted to develop the acceptance or rejection criteria for grind burn. Refer to Fig. 54 for nomenclature of a gear tooth.

Overtempering of the active profile of a gear tooth is not acceptable. Limited overtempering below the form diameter and above the tooth root fillet area (transition zone) may be acceptable if the total burnt area does not extend more than the area designated in Fig. 55. Overtempering is also acceptable on the end faces of teeth as long as it complies with the figure and applies to gears that might experience tip and edge loading. Figure 56 shows acceptable overtempering of the active profile of gear tooth near the edges. This is applicable to gears that do not experience edge loading even with misalignment between meshing teeth designed with proper crowning on tooth lead. Figure 57 depicts acceptable overtempering of the active profile near the edges and the tip of a gear tooth that does not experience either edge or tip loading.

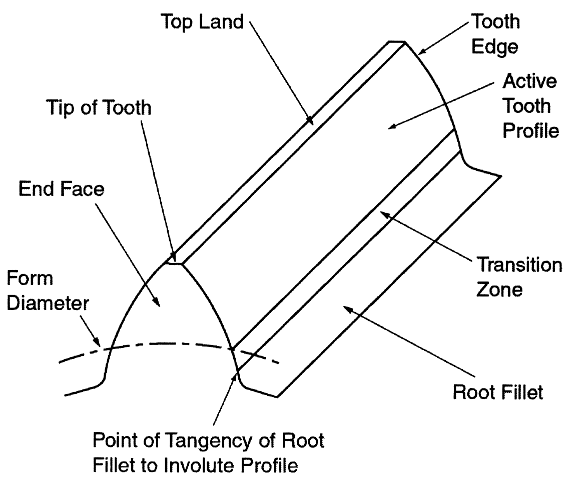

Fig. 54 Gear tooth nomenclature discussed in the section "Inspection and Limited Acceptance Criteria of Grind Burns"

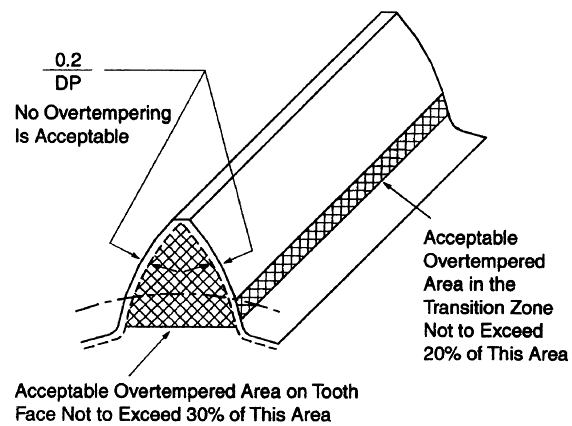

Fig. 55 Overtempering of gear tooth

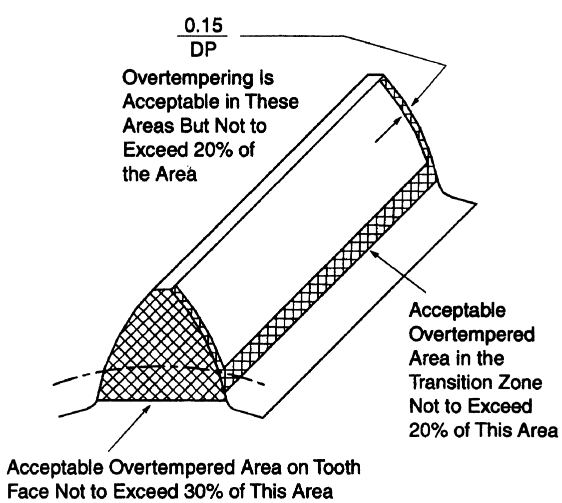

Fig. 56 Acceptable overtempering of active profile gear tooth near the edges

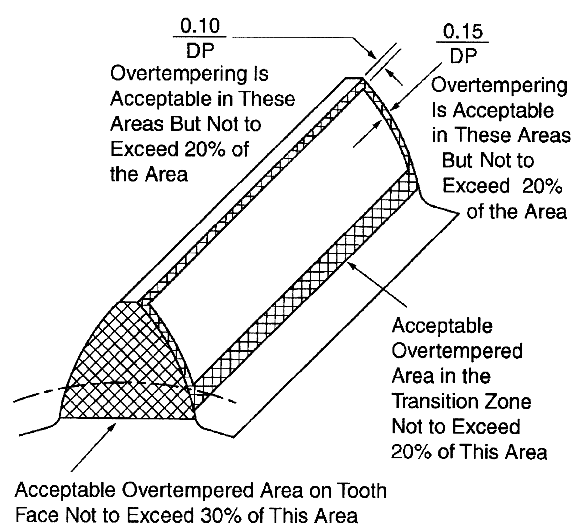

Fig. 57 Acceptable overtempering of active profile gear tooth near the edges and tip of a gear tooth that does not experience either edge or tip loading

Shot Peening of Carburized and Hardened Gears

In general, all carburized and hardened gears are either ground or honed. Gears with low distortion and low tooth quality requirement (AGMA class 9 or below) are normally honed. Conversely, gears with high distortion and high quality requirement (AGMA class 10 and above) are finish ground. Honing does not remove much stock—less than 0.01 (0.0005 in.) from a tooth surface. Thus, the hardness profile of tooth remains approximately the same as obtained after carburizing and hardening. This keeps the compressive stress induced at the surface during carburizing intact. However, in ground gears this is not the case. Very often stock up to 0.13 mm (0.005 in.) stock or more is removed from a tooth surface that reduces the surface hardness as well as removes the induced compressive stress during carburizing. These dual effects lower pitting life of ground gears significantly.

To improve pitting life of ground gears, tooth surfaces (ground and unground part of the profile as indicated in Fig. 58) are frequently shot peened after grinding to induce compressive residual stress at and below the surface. Thus, some knowledge of the proper shot peening process that enhances the performance of carburized, hardened, and ground gears is considered beneficial.

Shot peening is basically a mechanical process for improving fatigue strength of a part and has been in practice for a long time. In medieval times, a knight's armor was cold worked to final shape and hardness by hammering with a rounded-edged hammer. The repeated hammering improved armor fatigue life. Today, hammering is replaced with shot peening, which consists of bombarding a surface with small spherical metallic balls at a high velocity. Each shot acts as a tiny peenhammer, making a small dent on the surface of metal and stretching the surface radially as it hits (Fig. 59). The impact of the shot causes a plastic flow of the surface fibers extending to a depth depending upon degree of impact of the shot and the physical properties of the surface being peened. The effect of shot peening is similar to cold working to distinguish it from metal flow at high temperatures. Furthermore, it does not affect case thickness or case properties of ground, carburized, and hardened gears.

Experimental work indicates the surface compressive stress after shot peening to be several times greater than the tensile stress in the

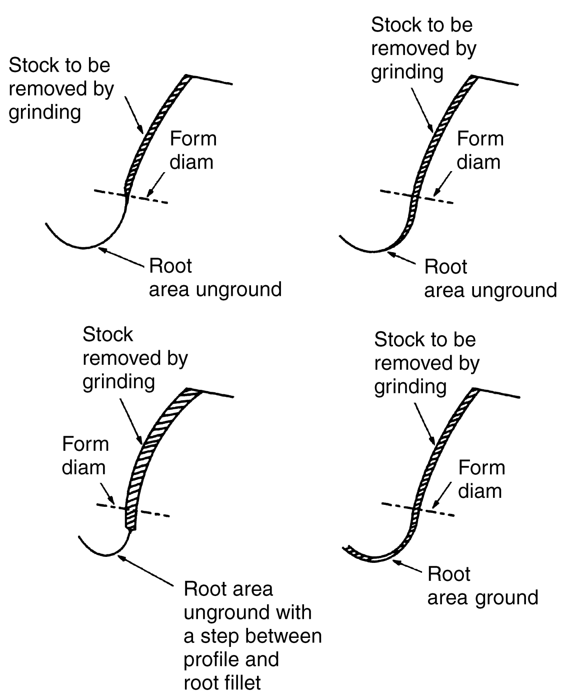

Fig. 58 Different configuration of tooth surfaces after grinding

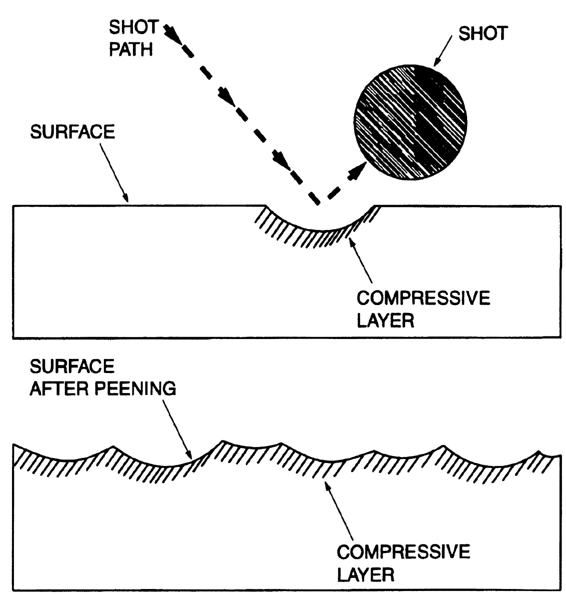

Fig. 59 Principle of shot peening

interior of the section. Figure 60 shows a typical compressive stress profile developed on a gear tooth fillet that was shot peened after grinding.

Because all fatigue failures originate with cracks, compressive stresses induced on the gear tooth fillet surface provide considerable increase in gear fatigue life because cracks do not initiate or propagate in compressively stressed zones. This makes the bending fatigue life of properly shot-peened gears 1.5 to 2 times the life of unpeened gears as illustrated in Fig. 61. In some gears, shot peening is limited to root fillet alone, which makes it necessary to mask the rest of the gear tooth surface. The masking needs to be removed after peening and this increases the cost of gear manufacture. Active tooth profiles used to be masked because it was believed shot peening deteriorated the tooth profile and surface, and thereby reduced the pitting life of the gear. Thus, gear engineers were not very enthusiastic about shot peening the entire tooth surface until recent research showed a considerable increase in pitting fatigue life of gears besides increased bending fatigue life when the active tooth profile also was peened. Such an improvement was achieved with a precisely controlled and monitored shot peening process. Now that shot peening is widely accepted to enhance gear life, the following are some of the guidelines for the process.

Type of Shot. From the standpoint of economy of operation, steel shots are preferred over cast iron shots. For good peening, the shots must be hard in the range of 42 to 50 HRC. Also, the shots need to be approximately round and uniformly sized at all times. As the shots break up, broken particles must be removed because sharp corners of broken or uneven shot may produce harmful effects that lead to a lower fatigue life. Shot supply is to be monitored in the machine so that no more than 20% of the particles, by weight, pass through the screen size specified for the shots, which come in different sizes such as S230, S170 (number indicates the diameter of the shot: 0.023 in. for S230, and 0.017 in. for S170).

Shot Size. The basic rule for shot size is that its diameter should not be greater than one-half as large as the root fillet radius of gear tooth. Shot size also should be correlated to the intensity of peening required, but without violating this rule. For low intensity, small shot should be specified and for high intensity, large shot. However, the peening machine, nozzles, air pressure, and volume also dictate, to a large extent, the most appropriate shot size. In general, the smallest shot size that will produce the required intensity should be used. The smaller the shot is, the faster the coverage rate and the shorter the peening time will be.

Intensity. The depth of the compressive layer on tooth surface is proportional to the shot intensity used. Thus, calibration of the impact energy from shot peening is essential for a controlled process. The energy of the shot stream is found to be a function of the shot size, material, hardness, velocity, and impingement angle. Specifying all of these variables would be difficult and impractical. The job is simplified by a method of measuring and duplicating shot-peening intensity on standard steel control strips

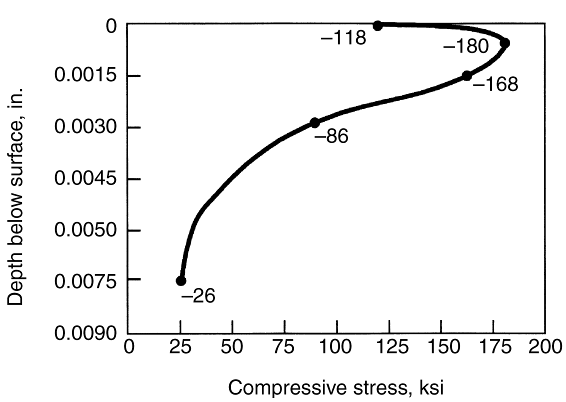

Fig. 60 Stress profile of carburized gear tooth root ground and shot peened

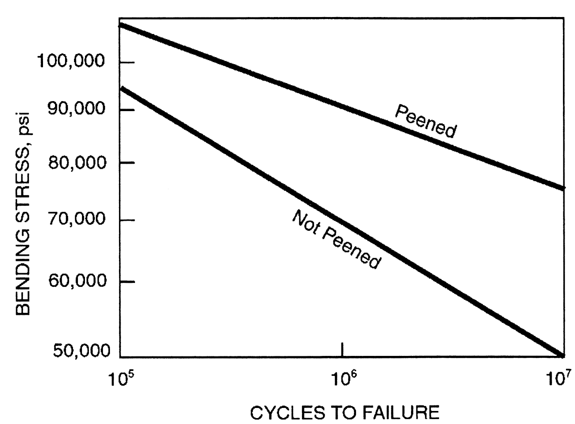

Fig. 61 Fatigue strength chart of carburized gears. Courtesy of Metal Improvement Company, Inc. (*Shot Peening Applications*)

developed many years ago by J.O. Almen of General Motors Research Laboratories. In his method, a flat strip of cold-rolled spring steel (AISI 1070) with a hardness of 44 to 50 HRC is clamped to a solid steel block and exposed to a shot stream for a given period of time (Fig. 62). Upon removal from the block, the residual compressive stress and surface plastic deformation produced by the peening impacts will cause the strip to curve, concave on the peened surface (Fig. 62). The height of this curvature serves as a measure of peening intensity.

There are three standard strips used to provide for different ranges of intensities: "A" strip (1.3 mm, or 0.051 in. thick), "C" strip (2.4 mm, or 0.094 in., thick), and "N" strip (0.8 mm, or 0.031 in., thick). The approximate relationship between the A, N, and C strips is: 3N = A = 0.3C. The A strip is the one most commonly used. If the arc height of the A strip is less than 0.10 mm (0.004 in.), the N strip should be used. For A strip readings greater than 0.51 mm (>0.020 in.), the C strip should be used. A reading of 10 A intensity means 0.25 mm (0.010 in.) arc height on the A strip.

Coverage. Amount of coverage is an essential element of a peening operation. Full, or 100%, coverage is defined as the uniform and complete dimpling of tooth surface as determined visually by a magnifying glass or the peen scan process. Most carburized and hardened gears require 200 to 300% coverage (double and triple peening process). Peening of gears beyond this range will not produce much improvement. Saturation or full coverage is defined as that point when doubling the peening time results in a 10% or less increase in height of Almen strips.

Angle of Shot Impingement. In case the shots do not impinge the surfaces at a proper angle, sometimes tensile stresses are introduced on tooth surfaces. For fine DP (above 20 DP) gears, the angle is particularly important to ensure that the shots impinge the root fillet as well as the profile of the tooth.

Whenever possible, microprocessor-controlled shot peening is recommended for pre-

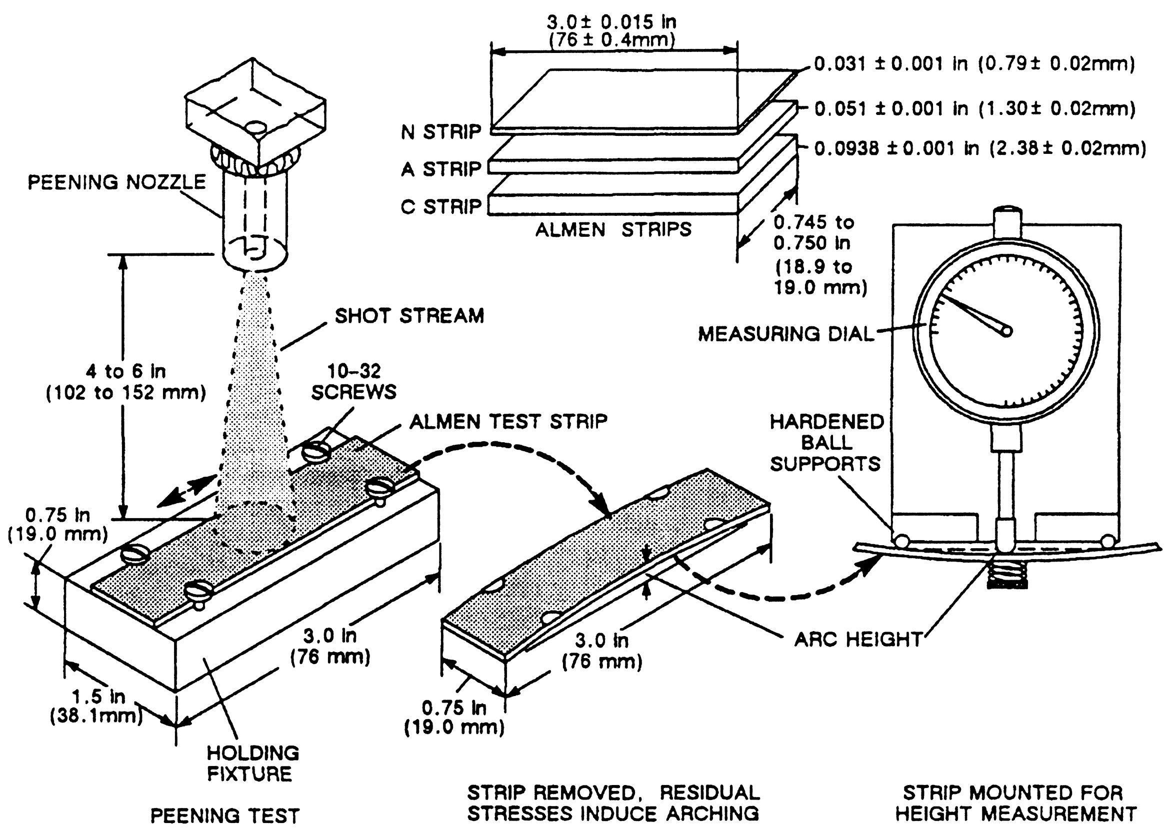

Fig. 62 Shot peening intensity control. Source: Ref 6

dictability, reproducibility, and verifiability of the process. This allows optimum level of each variable to be maintained, thereby improving the quality of shot peening.

Residual Stress. Compressive stress at and below the surface after shot peening of gears is shown in Fig. 63. The figure definitely shows the benefits of shot peening carburized, hardened, and ground gears.

Some guidelines of shot peening parameters for different DP carburized and hardened gears are given in Table 16.

Tooth Surface Profile. A typical gear tooth surface profile after shot peening is shown in Fig. 64. It clearly shows there is not much deterioration of the surface after shot peening.

Shot Peening Specification. In specifying shot peening requirements on a gear, the following parameters identified on a drawing help to minimize confusion:

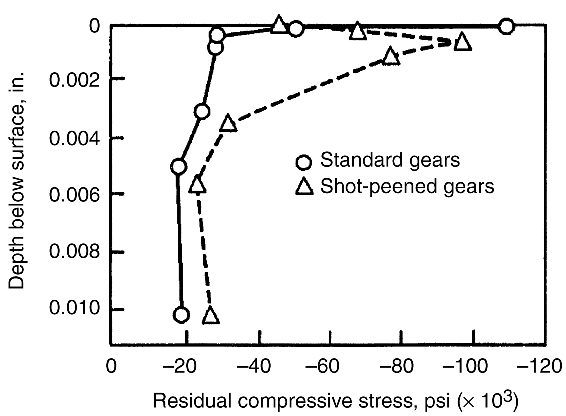

Fig. 63 Residual compressive stress below surface for standard and shot-peened gears

- Area to be shot peened
- Areas to be masked
- Optional areas that can be shot peened or masked
- Shot size, hardness, and material
- Intensity
- Coverage
- Equipment: microprocessor controlled or standard

Some typical shot peening parameters for carburized and hardened gears (AISI 9310 material) are:

Parameter	Value
Shot size	70
Shot type	Cast steel
Shot hardness	45 to 50 HRC
Intensity	0.178 to 0.229 mm
(height of Almen strip; type A)	(0.007 to 0.009 in.)
Coverage (profile and root)	200%

Applications and Costs

Applications. The vast majority of gears used in industrial applications today are carburized and hardened. It is because the process offers the highest power density in a gearbox through optimum gear design. High surface hardness, high case strength, favorable compressive residual stress in the hardened case, and suitable core properties, based on appropriate grade of steel, result in the highest gear rating. Furthermore, carburized gears offer superior heat resistance compared with other case hardening processes. These gears are also capable of withstanding high shock load. In general, smaller gears are carburized. Nonetheless, gears

Table 16 Recommended shot peening parameters

Diametral pitch (DP)	Shot peening intensity(a)	Shot diam mm	Shot diam in. × 10^{-4}	Shot type (HRC 42–50)	Coverage(b), %
20	0.006A	0.178	70	Cast steel	200
16	0.008A	0.178	70	Cast steel	200
10	0.010A	0.279	110	Cast steel	200
7	0.014A	0.432	170	Cast steel	200
5	0.018A	0.584	230	Cast steel	200
4	0.021A	0.711	280	Cast steel	200
3	0.007C	0.838	330	Cast steel	200
2	0.010C	1.168	460	Cast steel	200

(a) Almen strips: A, 1.3 ± 0.025 mm (0.051 ± 0.001 in.); C, 239 ± 0.025 mm (0.094 ± 0.001 in.). (b) 200% indicates a double shot-peening process.

up to a 2030 mm (80 in.) diameter have been successfully carburized.

Today, gas carburizing is extensively used to carburize gears because a great deal of improvement has been made in this technology since the days of pack or liquid carburizing. This is carried out mostly in air furnaces, although partial and full vacuum furnaces and fluidized-bed furnaces with improved controls also are used for high quality of carburized and hardened case.

The major disadvantage of carburizing and hardening is high heat treat distortion; although, recent advancements in carburizing furnace technology and equipment, press quenching, selection of proper steel, and control of manufacturing process have contributed greatly in controlling and minimizing this distortion. Low distortion helps to reduce gear finishing cost.

Costs. Tables 17 to 20 show how carburizing cost varies with case depth, tolerance, and any special features.

Vacuum Carburizing

Vacuum carburizing, also referred to as low-pressure carburizing, is a non-equilibrium, boost-diffusion-type carburizing process in which the steel being processed is austenitized in a rough vacuum, carburized in a partial pressure of hydrocarbon gas, diffused in a rough vacuum, and then quenched in either oil or gas. Compared to conventional atmosphere (gas) carburizing, vacuum carburizing offers excellent uniformity and repeatability because of the high degree of process control possible with vacuum furnaces; improved mechanical properties due to the lack of intergranular oxidation; and potentially reduced cycle times, particularly when the higher process temperatures possible with vacuum furnaces are used. As a result of these benefits, the vacuum process is being used increasingly to carburize gears in the automotive, off-highway, heavy truck, and aerospace industries (Ref 7).

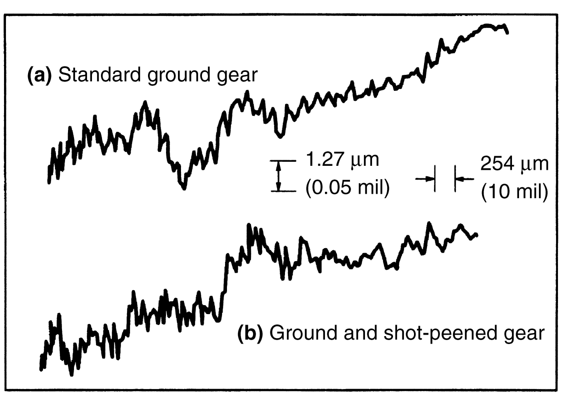

Fig. 64 Tooth surface finish of standard ground and shot-peened gears

Table 17 Case depth versus cost

Case depth requirement, mm	Case depth requirement, in.	Cost factor	Rejection, %	Scrap, %	Manufacturing difficulty index(a)
0.25	0.010	1.0	5	3	8
0.51	0.020	1.1	3	2	4
0.76	0.030	1.2	3	2	4
1.02	0.040	1.3	4	2	6
1.27	0.050	1.4	6	3	8
1.52	0.060	1.6	8	4	9
1.78	0.070	1.8	8	4	10

The control of carbon concentration and case depth is more difficult in long carburizing cycles. Also, very low case presents higher manufacturing difficulty as does excessive case. (a) Higher number indicates more difficulty.

Table 18 Case depth tolerance versus cost

Case depth tolerance(a), mm	Case depth tolerance(a), in.	Cost factor	Rejection, %	Scrap, %	Manufacturing difficulty index(a)
±0.05	±0.002	2.0	20	10	10
±0.08	±0.003	1.4	15	8	9
±0.10	±0.004	1.2	4	2	7
±0.13	±0.005	1.0	3	2	5
±0.18	±0.007	0.8	0	0	4
±0.25	±0.010	0.7	0	0	2

(a) Case depth tolerance must be greater than the carburizing process control capability to allow for various stock removal requirements. (b) Higher number indicates more difficulty.

Table 19 Special features versus cost

Carburizing features	Cost factor	Rejection, %	Scrap, %	Manufacturing index(a)
Single case	1.0	3	1.5	5
Dual case	2.5	25	12.0	10

(a) Higher number indicates more difficulty.

Table 20 Surface coverage versus cost

Surface coverage	Cost factor	Rejection, %	Scrap, %	Manufacturing index(a)
All over	1.0	4	2	5
Partial	3.5	4	2	10

The cost of masking and copper plating for selective carburizing is significant. (a) Higher number indicates more difficulty.

Process Overview

Vacuum carburizing of steel is typically a four-step process:

1. Heat and soak step at carburizing temperature to ensure temperature uniformity throughout steel
2. Boost step to increase carbon content of austenite
3. Diffusion step to provide gradual case-core transition
4. Quenching step. This may be carried out by direct quenching in oil or in a high-pressure gas quenching system.

Heat and Soak Step. The first step is to heat the steel being carburized to the desired carburizing temperature, typically in the range of 845 to 1040 °C (1550 to 1900 °F), and to soak at the carburizing temperature only long enough to ensure that the steel is uniformly at temperature. Oversoaking, particularly above 925 °C (1700 °F), can result in a reduction in toughness due to grain growth.

During the first step, surface oxidation must be prevented, and any surface oxides present must be reduced. In a graphite-lined heating chamber consisting of graphite heating elements, a rough vacuum in the range of 13 to 40 Pa (0.1 to 0.3 torr) is usually satisfactory. In a ceramic-lined heating chamber with silicon carbide heating elements, a partial pressure of approximately 40 to 67 Pa (0.3 to 0.5 torr) of hydrogen is effective. Steels with a high chromium content (M-50 NiL, X-2 Modified), a high silicon content (Pyrowear Alloy 53), or other high-oxygen affinity alloying elements usually require a higher vacuum level prior to carburizing but do not normally require preoxidizing.

Boost Step. Second is the boost step of the process. This step results in carbon absorption by the austenite to the limit of carbon solubility in austenite at the process temperature for the steel being carburized. The boost step is achieved by backfilling the vacuum chamber to a partial pressure with either a pure hydrocarbon gas (for example, propane or acetylene) or a mixture of hydrocarbon gases. Ammonia can be added if nitrogen alloying of the case is desired. An inert gas such as nitrogen can also be added to the gas or gas mixture.

Carbon transfer occurs by dissociation of the hydrocarbon gas on the surface of the steel, with direct absorption of the carbon by the austenite and hydrogen gas being liberated. The reaction with propane is:

$$C_3H_8 + 3Fe = 3Fe(C) + 4H_2 \qquad \text{(Eq 13)}$$

At typical carburizing temperatures, this reaction proceeds rapidly from left to right of the equation. Because such reactions are difficult to measure in situ, they cannot be used to control carbon potential when vacuum carburizing. Because there is no oxygen present, the oxygen-base methods of carbon-potential control used in conventional atmosphere carburizing cannot be used either. However, at least one furnace manufacturer has designed a system for vacuum carburizing that measures and controls the carbon potential of the carburizing gas.

A minimum partial pressure of hydrocarbon gas is required to ensure rapid carburizing of the austenite. The minimum partial pressure required varies with the carburizing temperature, the carburizing gas composition, and the furnace construction. Above the minimum partial pressure, the partial pressure of carburizing gas used has no relationship to the carburizing potential of the atmosphere. Typical partial pressures vary between 1.3 and 6.6 kPa (10 and 50 torr) in furnaces of graphite construction and 13 and 25 kPa (100 and 200 torr) in furnaces of ceramic construction. Partial pressures in excess of 40 kPa (300 torr) are not normally recommended because of the excessive carbon deposition within the furnace that accompanies higher partial pressures.

Diffusion Step. Third in the process is the diffusion step. If a steel were hardened with the carbon gradient resulting from the boost step only, particularly if no means of carbon-potential control were employed during the boost step, an undesirable microstructure adjacent to the carburized surface and an extremely abrupt case-core interface would result. The diffusion step enables the diffusion of carbon inward from the carburized surface, resulting in a lower surface carbon content (relative to the limit of carbon solubility in austenite at the carburizing temperature) and a more gradual case-core transition. The diffusion step is usually performed in a rough vacuum of 67 to 135 kPa (0.5 to 1.0 torr) at the same temperature used for carburizing. If carbon-potential control was used during the boost step, the diffusion segment might be shortened or eliminated.

Oil Quenching Step. The fourth step of the process is quenching. If a reheat step is not going to be employed, and/or no further machining is required, the steel is directly quenched in oil, usually under a partial pressure of nitrogen.

When vacuum carburizing is performed at a higher temperature than is normally used with conventional atmosphere carburizing, cooling to a lower temperature and stabilizing at that temperature prior to quenching is usually required. Alternatively, if a reheat step is going to be employed for grain refinement, and/or further machining is required, the steel is gas quenched from the diffusion temperature to room temperature, usually under a partial pressure of nitrogen.

Reheating usually consists of austenitizing in the 790 to 845 °C (1450 to 1550 °F) range followed by oil quenching. When aircraft-quality gearing or bearings are being processed, reheating is usually preceded by a subcritical anneal. A diagram of temperature and pressure versus time for a typical vacuum carburizing cycle with a reheat cycle is shown in Fig. 65.

High-Pressure Gas Quenching Step. Increasingly, vacuum carburizing is being carried out in conjunction with high-pressure gas quenching in 20 bar (2,000 kPa, or 300 psi) nitrogen or nitrogen-helium mixtures. High-pressure gas quenching is ideally suited for light loads, thin sections, and moderate-to-highly alloyed steels. An advantage of gas quenching compared to liquid quenchants (oil, water, and aqueous polymers) is that quenching with gas proceeds more uniformly, minimizing residual stresses and distortion. In addition to improved quench uniformity, gas quenching is a "clean" process, eliminating the need for a vapor degreasing step often used for oil quenching processes. Potential fire hazards and disposal problems are also eliminated.

Furnace Design

Vacuum carburizing is usually performed in a furnace specifically designed for this application. Units have been developed to operate in cell manufacturing operations found in commercial heat treating shops or just-in-time manufacturing plants. The standard line of furnaces consists of both one- and two-zone models. The two-zone model has one chamber for heating and the other for quenching.

The furnace can be of either graphite construction (graphite insulation and heating elements) or ceramic construction (refractory board insulation and silicon carbide heating elements). Graphite construction permits higher operating temperatures useful for a multipurpose furnace, whereas ceramic construction is well suited for vacuum carburizing, because it can be safely operated in air at process temperatures for die quenching or for facilitating soot removal. Figure 66 shows a typical continuous ceramic construction vacuum carburizing furnace. Figure 67 shows a typical batch-graphite construction vacuum furnace with carburizing capability.

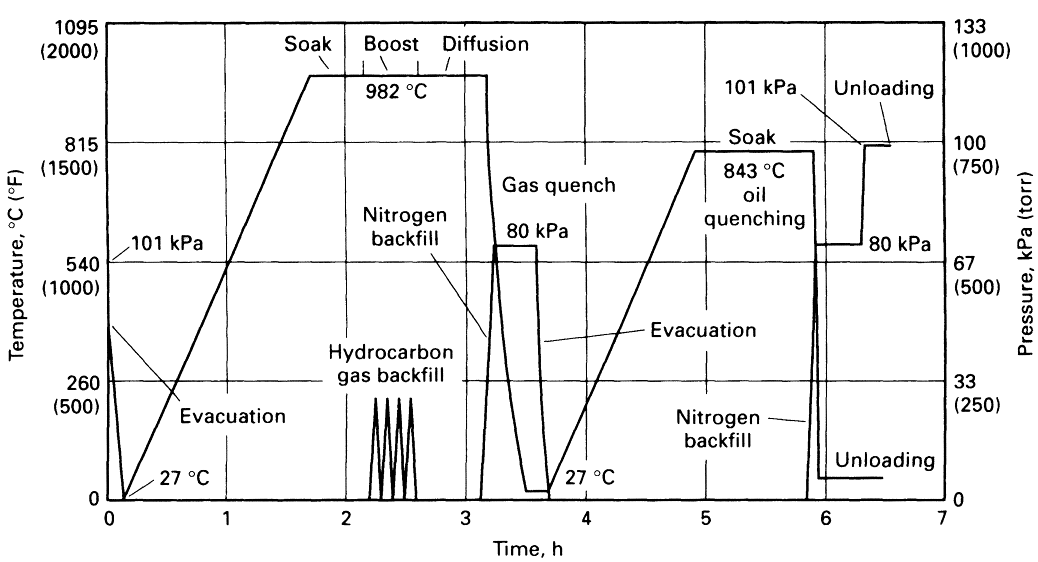

Fig. 65 Plot of temperature and pressure versus time for a typical vacuum carburizing process with a reheat cycle

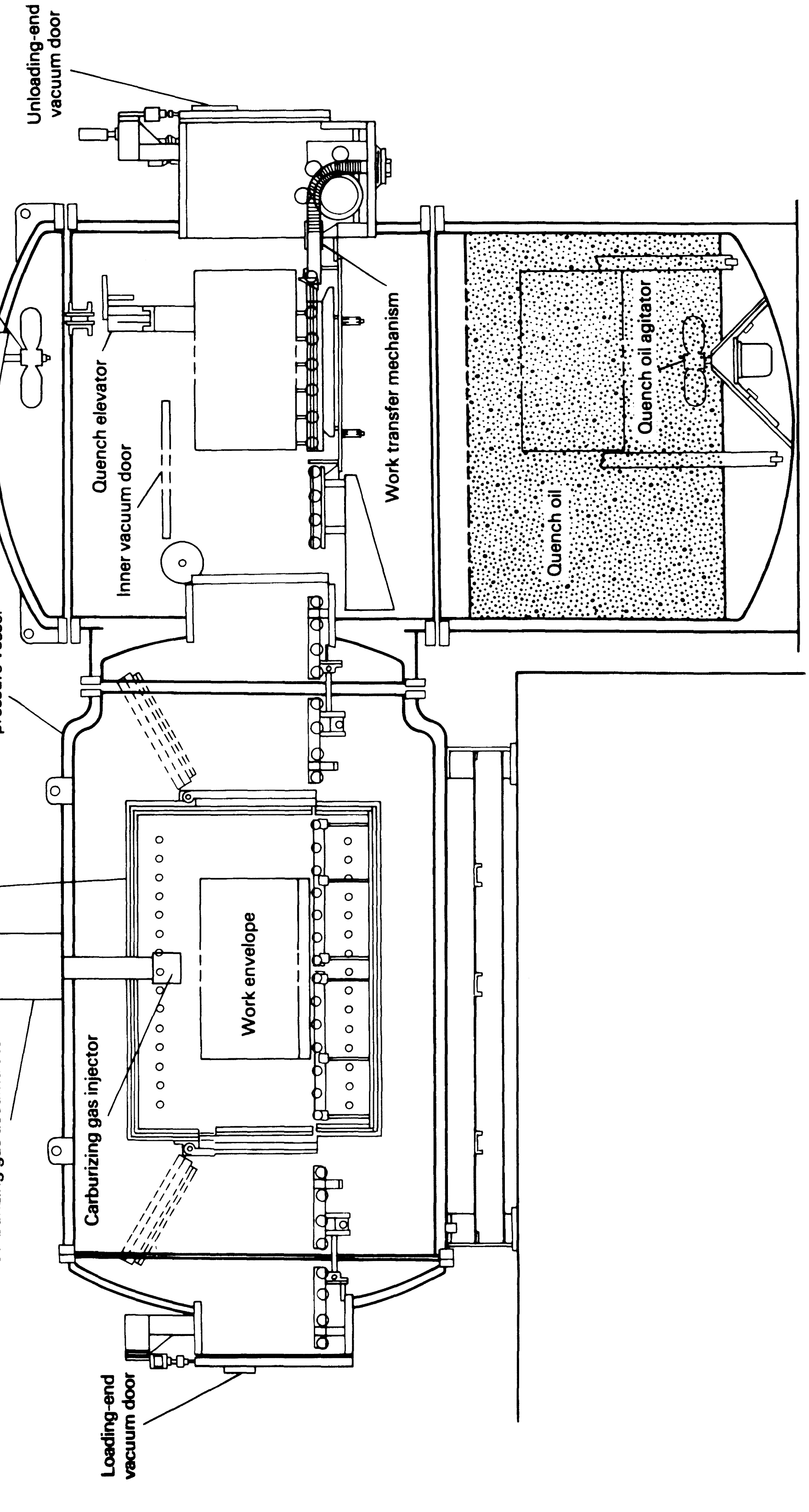

Fig. 66 A continuous ceramic vacuum carburizing furnace

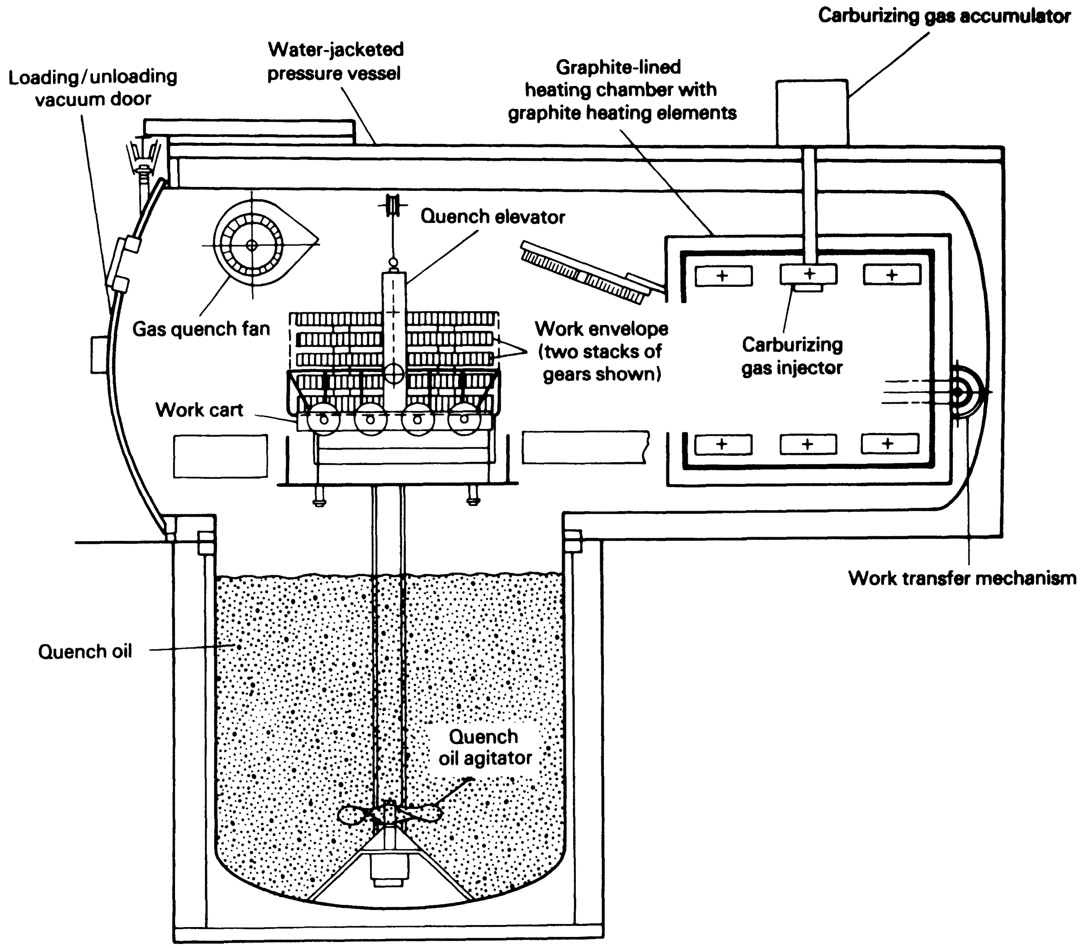

Fig. 67 A batch-graphite integral oil quench vacuum furnace with vacuum carburizing capability

High-Temperature Vacuum Carburizing

The reduction in carburizing time associated with a higher carburizing temperature has long been appreciated. However, typical atmosphere furnace construction generally restricts the maximum carburizing temperature to approximately 955 °C (1750 °F). The higher temperature capability of vacuum furnaces, as compared to typical atmosphere furnaces, permits the use of higher carburizing temperatures with correspondingly reduced cycle times.

High-temperature vacuum carburizing can significantly reduce the overall cycle time required to obtain effective case depths in excess of 0.9 to 1.0 mm (0.035 to 0.040 in.). For obtaining smaller case depths, high-temperature vacuum carburizing does not offer any advantages, because a grain-refining step is required, and the boost times tend to be too short for acceptable uniformity. Table 1 compares the time required to obtain 0.9 mm (0.035 in.) and 1.3 mm (0.050 in.) effective case depths via vacuum carburizing at both 900 °C (1650 °F) and 1040 °C (1900 °F) for an AISI 8620 steel. As is apparent from Table 21, significant reductions in the total cycle time can be obtained by using high-temperature vacuum carburizing.

Metallurgists not familiar with high-temperature vacuum carburizing are often concerned that although reduced cycle times can be obtained by high-temperature carburizing, a degraded microstructure with reduced mechanical properties results. There is no evidence that any reduction in either monotonic or cyclic mechanical properties results from high-temperature vacuum carburizing, provided that the process is properly specified and controlled. One aircraft-quality gearing user has performed extensive work in the area of high-temperature vacuum carburizing and concluded that there is no loss of properties when either AISI 9310 or X-2 Modified are high-temperature vacuum carburized to aircraft-quality gearing process specifications.

There is also concern that the high process temperatures involved result in excessive distortion and size change. Although it is true that some geometries are sensitive to the ultimate process temperature used, distortion can be minimized by using proper preheating/heating techniques, minimizing times at temperature, proper fixturing, and quenching techniques that are only severe enough to result in the desired microstructure and do not develop excessive nonuniform stresses within the part. Gas pressure quenching helps alleviate residual stresses, particularly with the new moderate- to high-alloy grades of carburizing steels being developed. As far as any dimensional change greater than normal is concerned, the uniformity and repeatability of the vacuum carburizing process, even at elevated temperatures, allows for dimensional change during manufacturing planning.

Comparison of Atmosphere and Vacuum Carburizing

As indicated in Fig. 1, atmosphere or gas carburizing remains the most popular carburizing method, because it represents a good compromise between cost and performance. In recent years, improvements in the reliability of the vacuum carburizing process have allowed its benefits to be realized, and a number of papers have been published that compare the benefits of atmosphere and vacuum carburizing (Ref 1 and 8–10). This section reviews the various advantages and disadvantages associated with these carburizing methods and presents the relative technical merits of each process (Table 22). The section that follows describes a case study that compares the properties of a low-alloy gear steel processed by both atmosphere and vacuum carburizing.

Atmosphere Carburizing Characteristics (Ref 1). Atmosphere carburizing is an empirically based, time-proven process in which a carbon-rich atmosphere surrounding a workload is used to chemically react with the surface of the parts to allow an adequate quantity of carbon to be absorbed at the surface and diffuse into the material.

Advantages of atmosphere carburizing include:

- The lowest initial capital equipment investment cost
- Adequate process control; that is, all of the process variables are understood, and reliable control devices are available to provide a measure of process repeatability.
- Capability of high-volume output using a wide variety of equipment styles, types, and workload sizes. Furnace types include box, pit, mechanized box (integral- and sealed-quench furnaces), pusher, conveyor (mesh belt and cast link belt), shaker hearth, rotary hearth, rotary drum (rotary retort), and car-bottom.
- Full automation capability, with recipe and part-number control of heat treat cycles
- Well-understood process problems allowing troubleshooting based on an established theoretical and empirical knowledge base

Disadvantages of atmosphere carburizing include:

- The need to "condition" equipment if idled or shut down prior to processing work

Table 21 Comparison of time required to obtain a 0.9 mm (0.035 in.) and 1.3 mm (0.050 in.) effective case depth in an AISI 8620 steel at carburizing temperatures of 900 °C (1650 °F) and 1040 °C (1900 °F)

Effective depth		Carburizing temperature		Time, min								
mm	in.	°C	°F	Heating to carburizing temperature	Soaking prior to carburizing	Boost	Diffusion	Gas quench to 540 °C (1000 °F)	Reheat to 845 °C (1550 °F)	Soak at 845 °C (1550 °F)	Oil quench	Total
0.9	0.035	900	1650	78	45	101	83	(a)	(a)	(a)	15	>322
		1040	1900	90	30	15	23	20	22	60	15	275
1.3	0.050	900	1650	78	45	206	169	(a)	(a)	(a)	15	>513
		1040	1900	90	30	31	46	20	22	60	15	314

(a) Not available

- A requirement of knowledge through empirically gained experience to achieve repeatable results. This is due to a wide variability in the type of equipment, its operation, maintenance, and constantly changing process conditions.
- The need for large material allowances for postprocessing operations due to accuracy and finish requirements. Case depths typically are specified in wide ranges (e.g., 0.75 to 1.25 mm, or 0.030 to 0.050 in.) to compensate for cycle-induced variability.
- Case depth quality issues; the best part of the case often is lost due to the amount of stock removal required.
- The need to constantly monitor environmental pollution issues, including air quality (for potentially hazardous gases such as CO and NO_x), water quality (for contamination concerns such as oil, minerals, etc.), waste disposal (quench oils), and safety issues (e.g., fire from combustible gases and quench oils, hot contact surfaces, and pinch points)

Vacuum Carburizing Characteristics (Ref 1). Vacuum carburizing is a proven method of pure carburizing and pure diffusion in which carbon penetrates into the surface of the steel being processed without interference from external influences, such as gas chemistry or surface contaminants.

Advantages of vacuum carburizing include:

- Easy integration into manufacturing. The process is clean, safe, simple to operate, and easy to maintain. Also, working conditions are excellent (that is, there are no open flames, heat, and pollution).
- Full automation capability using recipe or part-number control of heat treating cycles
- Capability of higher temperatures and flexible cycles, due to the type of equipment and the nature of the process
- Precise process control achieved using computer simulations, which allow adjustments to established cycles
- Consumption of energy by the equipment and process only when needed, due to the nature of the vacuum operation

Disadvantages of vacuum carburizing include:

- Higher initial capital equipment cost than atmosphere carburizing equipment

Table 22 Comparison of atmosphere and vacuum carburizing technologies

Criteria	Atmosphere carburizing	Vacuum carburizing
Temperature range, °C (°F)	790–980 (1450–1800)	790–1100 (1450–2000)
Case uniformity, mm (in.)(a)	±0.25 (±0.010)	±0.05 (±0.002)
Carbon-transfer control	Yes	Limited to control of time and temperature
Load density, kg/m^3 (lb/ft^3)(b)	45–70 (100–150)	22.5–45 (50–150)
Carburizing time, min	*x* minutes	*x* minutes minus 10–20%
Carbonitriding(c)	NH_3 additions	NH_3 additions
Microstructure	Acceptable (in most cases)	Optimal (in most cases)
Internal oxidation, mm (in.)	0.0076–0.0127 (0.0003–0.0005) common	None
Carbides	Suppression difficult	Suppression possible
Dealloying	Yes(d)	None
Decarburization	Possible	None
Hydrogen pickup	Yes (at high temperature)	Slight (internal porosity diffusion)
Furnace conditioning	Required (4 h typical)	None
Shell temperature, °C, or °F	Warm (typically >65, or 150)	Cold (typically ≪65, or 150)
Environmental impact	CO/NO_x emissions	Slight or none
Energy consumption	Low (~30%)	Lower (<30%)
Gas consumption	High (*x* cfh)	Low (1/3–1/6 *x* cfh)
Integration with cellular manufacturing	Difficult	Easy
Investment cost	Average	High

(a) Atmosphere and vacuum carburizing processes typically are different with respect to when gas additions are introduced, and this has the greatest impact on case uniformity. In atmosphere carburizing, enriching gas typically is added after the furnace set point is reestablished. Depending on the mass and configuration of the workload, a large temperature differential can exist between different locations within the workload. In the case of vacuum carburizing, a soak or stabilization period is built into the cycle to allow the workload to reach carburizing temperature prior to gas additions. (b) Loading density in vacuum carburizing equipment often is limited, due to the use of high-gas-pressure quenching chambers. (c) Techniques for vacuum carbonitriding using low pressure, <25 mbar (20 torr), are still being developed. (d) Due to oxidation at the part surface. Source: Ref 1

- Empirical process control, which requires processing loads to determine optimal settings or to fine tune simulator
- Formation of soot and tar, which occur due to the type, pressure, and quantity of hydrocarbon gas introduced

It is important to note that research during recent years has succeeded in finding combinations of pressure, gas type (e.g., acetylene), and flow parameters to minimize soot and tar formation as a concern in the vacuum carburizing process.

Case Study: Property Comparisons of a Gear Steel Processed by Atmosphere and Vacuum Carburizing (Ref 8)

Case Study Overview. The purpose of the study described in this section was to investigate whether vacuum carburizing could be used to improve the fatigue life of steels used for off-road vehicle transmission gearing. Fatigue is a major cause of gear failure, where the primary failure modes are gear tooth root bending and tooth pitting. Test samples were:

- Atmosphere carburized and oil quenched
- Vacuum carburized and oil quenched
- Vacuum carburized and high-pressure gas quenched

The effects of postheat grinding and shot peening were also examined.

The methods used to compare the vacuum and atmosphere carburizing processes in this study were x-ray diffraction (XRD) and microhardness testing.

Coupons of AISI 8620 low-alloy steel were heat treated using the different carburizing methods and subjected to identical post-heat treat grinding and shot peening operations.

X-ray diffraction was selected as an evaluation tool, because it can be used to measure residual stresses. Residual stresses are additive with applied stress, which makes their level an important factor in fatigue-critical components such as gears. Residual compressive stresses are desirable, because they oppose the applied, repetitive, and undesirable tensile stresses that cause fatigue failure. For gears, the areas of most concern are the flanks, which are subjected to contact loads that could cause pitting fatigue, and the roots, which experience tensile bending fatigue loads.

The greater the magnitude and depth of residual compressive stress, the greater the ability to improve fatigue properties. To enhance resistance to fatigue crack initiation, it is particularly important to have a higher compressive stress level at the outer surface. Also note that a deeper layer of compressive stress provides resistance to fatigue crack growth for a longer time than a shallower layer.

Carburizing Process Basics. Carburizing of a metal surface is a function of both the rate of carbon absorption into the steel and the diffusion of carbon away from the surface and into the interior of the part. Once a high concentration of carbon has developed at the surface during what is commonly called the boost stage, the process normally introduces a diffuse stage, where solid-state diffusion occurs over time. This step results in a change in the carbon concentration gradient between the carbon-rich surface and the interior of the steel. The result is a reduction of carbon concentration at the surface of the part, accompanied by an increase in the depth of carbon absorption.

The carburization process also induces desirable residual compressive stresses through the case-hardened layer. This stress state results from the delayed transformation and volume expansion of the carbon-enriched surface of the steel.

In atmosphere carburizing, parts are heated to austenitizing temperature in a neutral or carrier gas atmosphere that contains approximately 40% H_2, 40% N_2, and 20% CO. Small percentages of carbon dioxide (CO_2, up to 1.5%), water vapor (H_2O, up to 1%), and methane (CH_4, up to 0.5%), along with trace amounts of oxygen (O_2), also are present. The carburizing process also requires the addition of a hydrocarbon enriching gas, usually natural gas.

Of the 180 chemical equations that describe the reactions occurring during atmosphere carburizing, one of the most important is the water-gas reaction:

$$CO + H_2O = CO_2 + H_2 \qquad \text{(Eq 14)}$$

Control of the atmosphere carburizing process is done by looking at the CO/CO_2 and H_2O/H_2 ratios of this equation using instruments such as dewpoint analyzers, infrared analyzers, and oxygen (carbon) probes.

In atmospheres containing CO and H_2, carbon transfer is dominated by the CO adsorption (ad) and the oxygen desorption reactions:

$$CO \rightarrow CO_{ad} \rightarrow [C] + O_{ad} \quad \text{(Eq 15)}$$

$$O_{ad} + H_2 \rightarrow H_2O \quad \text{(Eq 16)}$$

These two reactions yield an alternate form of the water-gas reaction:

$$CO + H_2 = [C] + H_2O \quad \text{(Eq 17)}$$

Thus, the transfer of carbon in atmospheres containing CO and H_2 is connected with a transfer of oxygen, giving rise to an oxidation effect in steel containing oxide-forming alloying elements such as silicon, chromium, and manganese. This phenomenon is known as internal or intergranular oxidation of steel (Ref 4).

Atmosphere Carburized and Oil Quenched Hardness Profile. Figure 68 shows Knoop (500 gf) microhardness profiles for an atmosphere carburized and oil quenched AISI 8620 steel gear.

Atmosphere carburizing to a depth of 0.36 mm (0.014 in.) produced a hardness of 58 HRC at both the gear tooth pitch line and root. From this depth, the hardness values quickly diverge. The effective case depth (at 50 HRC) is 0.76 mm (0.030 in.) in the root and 1.33 mm (0.0525 in.) at the pitch diameter. These values are typical of the vast majority of carburized gears currently in service.

For resistance to bending fatigue, it is desirable to achieve a deeper case in the root. This produces a deeper level of high-hardness, high-strength material with the benefit of residual compressive stress.

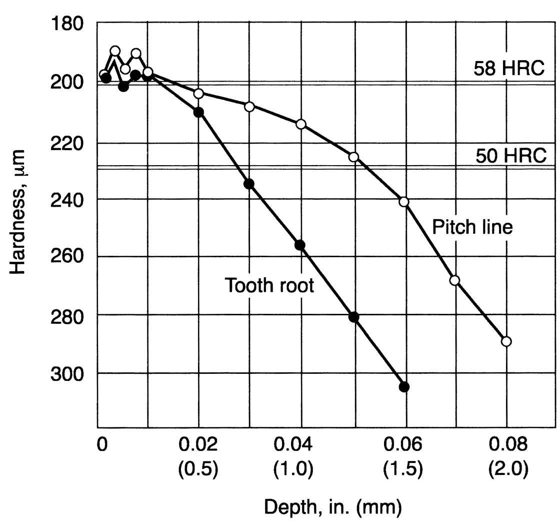

Fig. 68 Microhardness profiles at pitch line and tooth root for an atmosphere carburized and oil quenched AISI 8620 gear. For resistance to bending fatigue, it is desirable to achieve a deeper case in the root.

Vacuum carburizing, by comparison, does not use a carrier gas atmosphere but instead uses vacuum pumps to remove the atmosphere from the chamber before the process begins. For carburizing to take place in a vacuum furnace, all that is needed is a small, controlled addition of a hydrocarbon gas.

Unlike atmosphere carburizing, the breakdown of hydrocarbons in vacuum carburizing is via nonequilibrium reactions. This means that the carbon content at the surface of the steel is very rapidly raised to the saturation level of carbon in austenite. By repeating the boost and diffuse steps, any desired carbon profile and case depth can be achieved.

Today, vacuum carburizing is best performed using low-pressure techniques under 20 torr (25 mbar) and typically at temperatures between 790 and 1040 °C (1455 and 1900 °F). Hydrocarbon gases currently being used for vacuum carburizing are acetylene (C_2H_2), propane (C_3H_8), and, to a lesser degree, ethylene (C_2H_4). Methane (CH_4) is not used, because it is nearly nonreactive at these low pressures, unless the temperature is at or above 1040 °C (1900 °F).

Carbon is delivered to the steel surface in vacuum carburizing via reactions such as these:

$$C_2H_2 \rightarrow 2C + H_2 \quad \text{(Eq 18)}$$

$$C_3H_8 \rightarrow CH_4 + C_2H_4 \rightarrow C + 2CH_4 \quad \text{(Eq 19)}$$

$$C_2H_4 \rightarrow C + CH_4 \quad \text{(Eq 20)}$$

In the past, propane has been the primary hydrocarbon gas used for vacuum carburizing; however, propane dissociation occurs before the gas comes in contact with the surface of the steel, thus producing free carbon or soot. This uncontrolled soot formation results in poor carbon transfer to the part and loss of up-time productivity due to the need for additional heat treat equipment maintenance.

Development work done in the past few years has demonstrated that acetylene is a good performing gas for vacuum carburizing. This is because the chemistry of acetylene (Eq 18) is vastly different from that of propane or ethylene (Eq 19 and 20). Dissociation of acetylene delivers two carbon atoms to the one produced by dissociation of either propane or ethylene and avoids formation of nonreactive methane.

Control of the vacuum carburizing process is on a time basis. Carbon transfer rates are a function of temperature, gas pressure, and gas flow rate. Simulation programs have been written to determine the boost and diffuse times of the cycle.

Vacuum Carburized and Oil Quenched Hardness Profile. Figure 69 shows hardness profiles for a vacuum carburized and oil quenched AISI 8620 steel gear.

The overall case depth of maximum hardness for the vacuum carburized part is noticeably deeper than that of the atmosphere carburized part in Fig. 68. The vacuum carburized case depth of approximately 0.81 mm (0.032 in.) at 58 HRC is more than double that obtained with atmosphere carburizing, while the effective case depths (depth at 50 HRC) are similar. Also note the much greater consistency in root and pitch line hardnesses through a depth of 0.81 mm (0.032 in.) for vacuum carburizing versus atmosphere carburizing (Fig. 69 versus Fig. 68).

Vacuum Carburized and Gas Quenched Hardness Profile. The hardness profiles shown in Fig. 70 are for an AISI 8620 steel gear that has been vacuum carburized and then high-pressure gas quenched (HPGQ) in 20 bar (2,000 kPa, or 300 psi) nitrogen.

A comparison of Fig. 68 and 69 shows that use of HPGQ instead of oil quenching in vacuum carburizing results in a more uniform case depth between gear pitch line and root. The absence of a vapor layer in gas quenching results in a more uniform cooling rate along the gear tooth and root profile.

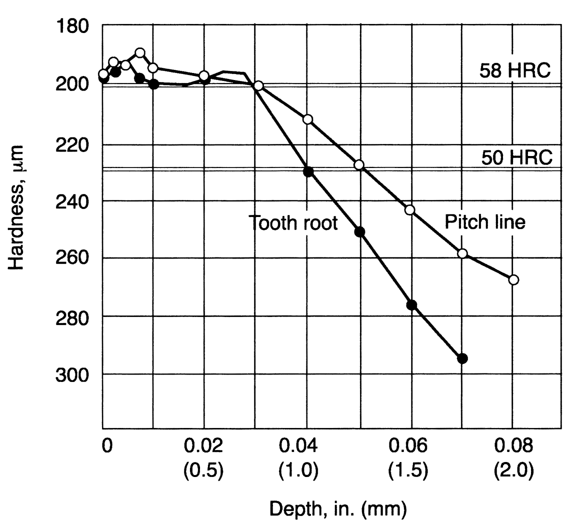

Fig. 69 Microhardness profiles at pitch line and tooth root for a vacuum carburized and oil quenched AISI 8620 gear. The overall case depth of maximum hardness is deeper than that of the atmosphere carburized part in Fig. 68

The Test Procedure. The following procedure was used to properly evaluate the effect of different heat treatments and post-heat treatment processes on residual stress in coupons of AISI 8620 low-alloy gear steel:

- Five coupons from the same heat lot of AISI 8620 were cut to size: 76 by 19 by 13 mm, ±0.05 mm (3.00 by 0.75 by 0.505 in., ±0.002 in.). The coupons were stamped, and a separate manufacturing process was defined for each (Table 23).
- Coupons were sent out for heat treatment—vacuum or atmosphere carburizing—according to the parameters in Table 24. Required surface hardness: 59 to 61 HRC. Vacuum carburized coupons were nitrogen gas quenched, while atmosphere carburized coupons were oil quenched.
- Heat treated coupons were ground to 12.7 ± 0.013 mm (0.5000 ± 0.0005 in.), removing

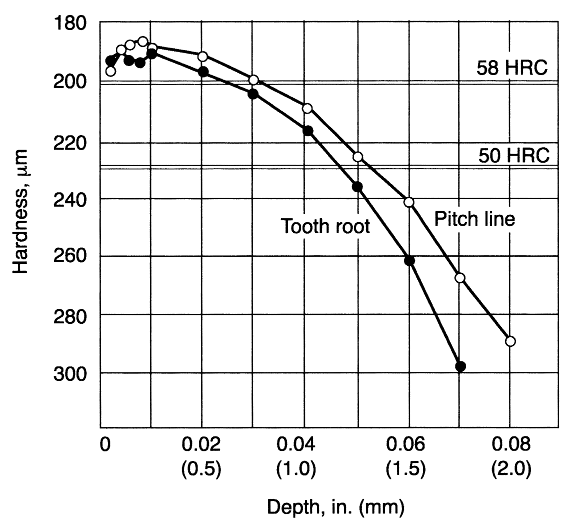

Fig. 70 Microhardness profiles at pitch line and tooth root for a vacuum carburized and high-pressure gas quenched AISI 8620 steel gear. Use of gas quenching instead of oil quenching (Fig. 69) results in a more uniform case depth between pitch line and root.

Table 23 Test coupon manufacturing processes

Coupon	Identification	Process
1	EX2470	Vacuum carburize (VC)
2	EX2470-1	Vacuum carburize and shot peen (VC and SP)
3	EX2470-2	Atmosphere carburize (AC)
4	EX2470-3	Atmosphere carburize and shot peen (AC and SP)
5	EX2470-4	Vacuum carburize and dual shot peen (VC and DSP)

no more than 0.15 mm (0.006 in.) from the non-stamped side where XRD was to take place.

- Three of the five coupons were sent out for shot peening.
- All five coupons were sent out for XRD on the nonstamped side.

The Benefits of Shot Peening. The primary purpose of shot peening gears is to enhance their fatigue life by inducing a high residual compressive stress at the surface of the tooth roots. Shot peening is most effective for parts subject to high-cycle fatigue loading.

A basic explanation is provided by the graph in Fig. 71, a typical stress-number of cycles (*S*-*N*) curve. It plots (tensile) stress, *S,* versus the number of load cycles, *N.* It is important to note that the vertical scale is linear, whereas the horizontal scale is logarithmic. This means that as tensile stress is reduced, fatigue life improves exponentially. An ~35% reduction of stress from 760 MPa (110 ksi) to 485 MPa (70 ksi) results in an improvement in fatigue life from 40,000 cycles to 160,000 cycles (400%). Additional reductions in tensile stress result in significantly more fatigue enhancement. At 415 MPa (60 ksi), for example, the anticipated fatigue life is ~400,000 cycles.

The residual compressive stresses produced by shot peening counteract applied tensile stresses. The compressive stresses are induced by impacts of small, spherical media (shot). The impact of each individual shot stretches the surface enough to yield it in tension. Because the surface cannot fully restore itself due to the mechanical yielding that has taken place, it is left in a permanent compressed state.

Shot peening results in a residual compressive stress at the surface—where most fatigue cracks initiate—that is ~55 to 60% that of the material ultimate tensile strength. For carburized gears, the surface compression is typically 1170 to 1725 MPa (170 to 250 ksi), which results in a significant improvement in fatigue properties.

The grinding process is applied to components so often and in so many forms (automatic, manual, with and without coolant) that it is often overlooked from a residual stress standpoint. However, its influence should not be discounted, especially when dealing with fatigue-critical parts.

Table 24 Test parameters for atmosphere and vacuum carburized coupons

Parameter	Atmosphere	Vacuum
Temperature, °C (°F)	940 (1725)	940 (1725)
Boost time, min	300	32
Diffusion time, min	120	314
Hardening temperature, °C (°F)	845 (1550)	845 (1550)
Quenching method	Oil at 60 °C (140 °F)	Nitrogen gas at 20 bar
Tempering temperature, °C (°F)	175 (350)	175 (350)
Tempering time, h	2	2

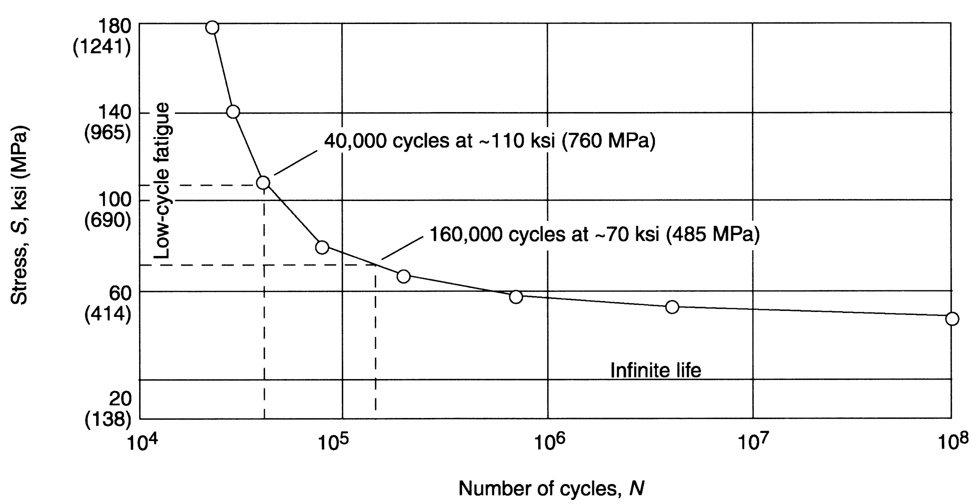

Fig. 71 Typical *S-N* curve, or plot of (tensile) stress, *S,* vs. number of load cycles, *N*. The primary purpose of shot peening gears is to enhance their fatigue life by inducing a high residual compressive stress at the surface of the tooth roots. Shot peening is most effective for parts subject to high-cycle fatigue loading (>10^4 to 10^5 cycles).

During grinding, residual tensile stress may be created from generation of excessive, localized heat. The localized surface area being ground heats from friction and attempts to expand but cannot, because it is surrounded by cooler, stronger metal. If the temperature generated from grinding is high enough, however, the metal yields in compression due to the resistance to its expansion and reduced mechanical properties at elevated temperature. On cooling, the yielded material attempts to contract. The surrounding material resists this contraction, thus creating residual tensile stress. Because heat is the major cause of residual tensile stress from grinding, the importance of coolant for controlling these stresses is paramount.

X-Ray Diffraction Residual Stress Measurements. X-ray diffraction was used to measure the residual stresses at surface and subsurface locations. The technique measures strain by measuring changes in atomic distances. It is a direct, self-calibrating method that measures tensile, compressive, and neutral strains equally well. Strains are converted to stresses by multiplying by elastic constants appropriate for the alloy and atomic planes measured.

For this study, chromium Kα radiation was used to diffract the (211) planes at approximately 156° 2θ. The area measured was nominally 4 mm (0.16 in.) in diameter. Because only a few atomic layers are measured, the technique is considered a surface analysis technique. The subsurface measurements were made by electrochemically removing small amounts of material. These subsurface measurements were subsequently corrected for stress gradient and layer removal effects using standard analytical calculations.

Comparing the Processes. Hardness profiles for vacuum carburized (coupon 1) and atmosphere carburized (coupon 3) AISI 8620 steel coupons are compared in Fig. 72. A major advantage of vacuum carburizing over atmosphere carburizing is a deeper case of high hardness. Hardness values for the two carburizing processes are given in Table 25.

Effect of Peening. Residual stress distributions in the three carburized, ground, and shot peened coupons—coupons 2, 4, and 5—are plotted in Fig. 73.

From a fatigue standpoint, the solid layer of compression demonstrated for all three coupons implies excellent resistance to initiation and growth of fatigue cracks. The tensile stress required for a fatigue crack to develop must first overcome compressive stresses that are ~1035 MPa (150 ksi) at the surface and ~1515 MPa (220 ksi) at 0.05 mm (0.002 in.) below the surface. A tensile stress of 1035 MPa (150 ksi) produces a net stress of 0 MPa (0 ksi) at the surface when added to the residual compressive stress.

Coupons 2 (vacuum carburized and shot peened) and 4 (atmosphere carburized and shot peened) were shot peened at an Almen intensity of 14 to 16. (Almen intensity is a measure of the energy of the shot stream.) The steel shot had a hardness of 55 to 62 HRC and a nominal diameter of 0.58 mm (0.023 in.).

The residual stress curves in Fig. 73 have shapes typical of shot peened material. All three have a similar maximum compressive stress of ~1515 MPa (220 ksi). This value is ~55 to 60% that of the steel ultimate tensile strength at the surface. Because all three coupons were hard-

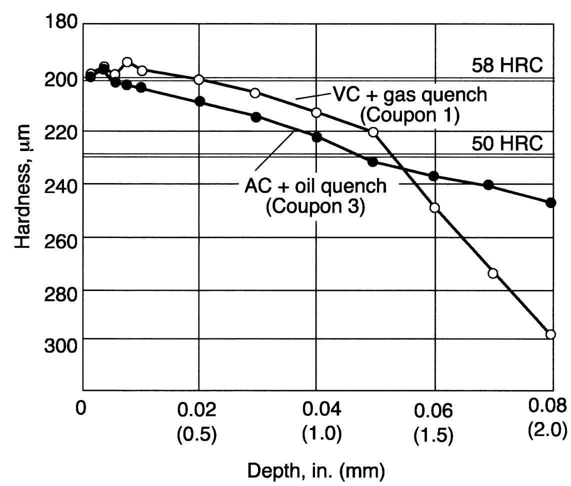

Fig. 72 Microhardness profiles for vacuum carburized and gas quenched (coupon 1) and atmosphere carburized and oil quenched (coupon 3) AISI 8620 steel coupons. A major advantage of vacuum carburizing is a deeper case of high hardness. VC, vacuum carburized; AC, atmosphere carburized

Table 25 Comparison of atmosphere and vacuum carburizing results

Property	Atmosphere	Vacuum
Depth to 58 HRC, mm (in.)	0.20 (0.008)	0.58 (0.023)
Surface hardness before grinding, HRC	59	60
Surface hardness after removal of 0.1 mm (0.004 in.) stock by grinding, HRC	58	62

ened to 59 to 62 HRC, they also had similar tensile strengths (at the surface).

The depth of the compressive stress layer is a function of the Almen intensity. It can be increased by increasing shot size and/or velocity. The depth is the location where the residual stress versus depth curves would cross the neutral axis (into tension) if the positively sloped lines were extended. A greater depth of compression is desired, because this layer is what resists fatigue crack growth. Coupons 2 and 4 were shot peened to the same intensity, so that the depth of their compressive stress layers is also essentially the same at ~0.18 to 0.20 mm (0.007 to 0.008 in.).

Dual Peening. The trade-off to increasing shot peening intensity is that there is additional cold work and material displacement at the point of shot impact. This generally results in less compression right at the surface (depth = 0) and a more aggressive surface finish.

Dual peening is performed to make up for the reduced compression resulting from high-intensity peening. The technique consists of shot peening the same surface twice—peening at a higher intensity is followed by peening at a lower intensity, usually with smaller media. The second peening reduces the degree of cold work at the surface, improving the surface finish, which, in turn, makes the surface more compressed.

Coupon 5 was dual peened. The process specified: MI-230H shot at 18 to 20 Almen followed by MI-110H shot at 8 to 10 Almen. The residual stress curve for this coupon (Fig. 73) is ~0.025 to 0.05 mm (0.001 to 0.002 in.) deeper than the curves for the coupons single shot peened at 14 to 16 Almen using MI-230H shot.

This would be expected for a carburized gear steel. The surface stress of coupon 5 (at depth = 0) is the same as that of the other two shot peened coupons. What most likely occurred is that it was less compressed after the first peening step. When the second was performed, the surface became even more compressed, to the ~930 MPa (135 ksi) level shown in Fig. 73. Therefore, coupon 5 would be expected to have the best fatigue performance of the three, because it has the most compressive stress throughout its depth. This is particularly evident between 0.08 and 0.20 mm (0.003 and 0.008 in.) below the surface. At 0.10 mm (0.004 in.) below the surface, for example, there is still 1380 MPa (200 ksi) of compression for coupon 5, compared with 1170 MPa (170 ksi) for coupon 4 and 1000 MPa (145 ksi) for coupon 2.

Effect of Grinding. Testing of coupons that had been ground gave unexpected results that

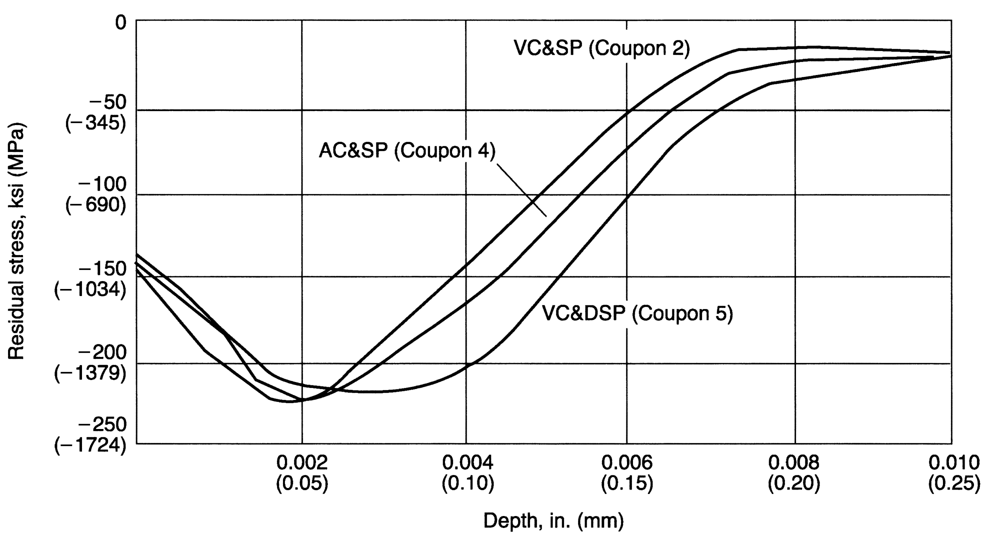

Fig. 73 Residual stress distributions in the three carburized, ground, and shot peened coupons (coupons 2, 4, and 5) of AISI 8620. Each contains a solid layer of compression that implies excellent resistance to initiation and growth of fatigue cracks. Note that the residual stress curve for dual-peened coupon 5 is ~0.025 to 0.05 mm (0.001 to 0.002 in.) deeper than those for the single-shot-peened coupons. See Table 23 for definition of process abbreviations for coupons.

required further investigation. All coupons were ground at the same time. Grinding was performed using a wheel that had coolant flow. The operator was instructed to remove no more than 0.025 mm (0.001 in.) of stock per pass, for a total of 0.10 mm (0.004 in.) of material removed from each coupon. This appeared to be acceptable grinding practice, and little thought was given to the technique prior to testing.

X-ray diffraction measurements indicated that tensile stresses existed on the surface of the vacuum carburized coupon (coupon 1) as high as 255 MPa (37 ksi) at 0.013 mm (0.0005 in.) below the surface. At a depth of ~0.10 mm (0.004 in.), the values crossed the neutral axis into compression. These results were immediately questioned. However, retesting at several locations using XRD verified that the original values were correct.

The explanation lies in the fact that additional heat was generated when grinding vacuum carburized coupon 1. The coupon was 1 HRC point harder at the surface and 4 HRC points higher after 0.10 mm (0.004 in.) of stock removal. These values are higher than those for the atmosphere carburized specimen (coupon 3). Additional heat from an increase in friction resulted in the generation of residual tensile stresses on the vacuum carburized coupon.

This is an excellent example of why it is important to carefully evaluate the amount of heat generated when grinding fatigue-critical parts. It also demonstrates that XRD is an effective tool for determining the residual stress state of components before they enter service.

Test Results Reviewed. This study compared atmosphere and vacuum carburizing of AISI 8620 gear steel and evaluated the influence of the subsequent manufacturing operations of shot peening and grinding. The primary goal of the study was to determine which carburizing process was more suitable for heavy-duty transmission gears. Gears are subject to both sliding and rolling-contact stresses on their flanks in addition to bending stresses in tooth roots. To meet these demanding performance criteria, the steel gears ideally would be hardened for strength and contact properties and have residual compressive surface stresses for bending fatigue resistance.

It was concluded that vacuum carburizing is superior to atmosphere carburizing for heat treating heavy-duty transmission gears and enjoys the following advantages:

- *Higher hardness (XRD coupons):* Surface (before grinding), 1 HRC point higher (60 versus 59 HRC); subsurface (after 0.10 mm, or 0.004 in., stock removal), 4 HRC points higher (62 versus 58 HRC)
- *Greater depth of high, ≥58 HRC, hardness (XRD coupons):* 58 HRC depths were vacuum, 0.58 mm (0.023 in.); atmosphere, 0.20 mm (0.008 in.)
- *Greater depth of high, ≥58 HRC, hardness (pitch line and root of actual gears):* 58 HRC depths were vacuum, 0.81 mm (0.032 in.); atmosphere, 0.38 mm (0.015 in.)
- *Deeper effective case in tooth root (actual gears):* Vacuum, 1.0 mm (0.040 in.); atmosphere, 0.699 mm (0.0275 in.)
- *Higher surface residual compression (XRD of coupons without shot peening or grinding):* Vacuum, 135 MPa (19.6 ksi); atmosphere, 98 MPa (14.2 ksi)
- *Improved consistency between the case layer at the pitch line of the gear flank and gear roots (actual gears):* Vacuum, 0.28 mm (0.011 in.) variation; atmosphere, 0.648 mm (0.0255 in.) variation

Shot Peening Results Reviewed. Both the vacuum carburized and atmosphere carburized surfaces responded equally to shot peening:

- Maximum compressive stress: ~1515 MPa (220 ksi)
- Compressive layer depth: ~0.18 to 0.20 mm (0.007 to 0.008 in.)

Dual shot peening at first a higher and then a lower intensity resulted in a greater depth of compression by ~0.025 to 0.05 mm (0.001 to 0.002 in.). The surface stress of the dual-peened coupon was very similar, at ~930 MPa (135 ksi), to that of the conventionally shot peened coupons. The higher-intensity first peen would have produced a less compressed surface, but the second, lower-intensity peen would have restored compressive stress to the ~930 MPa (135 ksi) level. The dual-peened coupon should have significantly better high-cycle fatigue properties than the single-peened coupons.

Fatigue. In terms of fatigue performance, the additional 34.5 MPa (5 ksi) of compression measured for the vacuum carburized coupon (not shot peened or ground) should yield significant increases in gear life under high-cycle fatigue loading, compared with that for the atmosphere carburized coupon.

ACKNOWLEDGMENTS

This chapter was adapted from:

- J.R. Davis, Ed., Vacuum and Plasma Carburizing, *Surface Hardening of Steels: Understanding the Basics,* ASM International, 2002, p 91–114
- A.K. Rakhit, Carburizing and Hardening Gears, *Heat Treatment of Gears: A Practical Guide for Engineers,* ASM International, 2000, p 33–132

REFERENCES

1. D.H. Herring, Pros and Cons of Atmosphere and Vacuum Carburizing, *Ind. Heat.,* Jan 2002, p 45–48
2. J.R. Davis, Ed., Hardenability of Steels, *ASM Specialty Handbook: Carbon and Alloy Steels,* ASM International, 1996, p 138–166
3. D.W. Dudley, *Handbook of Practical Gear Design,* McGraw-Hill Book Company, 1984, p 4.13
4. J.R. Davis, Ed., Gas Carburizing, *Surface Hardening of Steels: Understanding the Basics,* ASM International, 2002, p 17–89
5. G.F. Vander Voort, Ed., Metallography and Microstructures of Case-Hardening Steel, *Metallography and Microstructures,* Vol 9, *ASM Handbook,* ASM International, 2004, p 627–643
6. *Gear Materials and Heat Treatment Manual,* ANSI/AGMA 2004-B89, American Gear Manufacturers Association, 1989, p 48
7. F.J. Otto and D.H. Herring, Gear Heat Treatment, Part II, *Heat. Treat. Prog.,* July/Aug 2002, p 27–31
8. G.D. Lindell, D.H. Herring, D.J. Breuer, and B.S. Matlock, Atmosphere versus Vacuum Carburizing, *Heat Treat. Prog.,* Nov 2001, p 33–41
9. W.J. Titus, Considerations when Choosing a Carburizing Process: Traditional Batch Atmosphere Carburizing versus Vacuum and Ion Carburizing, *Ind. Heat.,* Sept 1987
10. B. Edenhofer, Overview of Advances in Atmosphere and Vacuum Heat Treatments, *Heat Treat. Met.,* Vol 4, 1998, and Vol 1, 1999

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J.R. Davis, editor, p227-244
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CHAPTER 10

Nitriding

NITRIDING is a surface hardening heat treatment that introduces nitrogen into the surface of steel at a temperature generally in the range of 500 to 575 °C (930 to 1065 °F) while it is in the ferritic condition. Nitriding, therefore, is similar to carburizing in that surface composition is altered, but different in that the nitrogen is added into ferrite instead of austenite. The fact that nitriding does not involve heating into the austenite phase field and a subsequent quench to form martensite means that nitriding can be accomplished with a minimum of distortion and excellent dimensional control.

Although nitriding can be carried out using a gaseous, liquid, or solid medium, gas nitriding using ammonia (NH_3) as the nitrogen-carrying species is the most commonly employed process and will be the emphasis of this chapter. Steels that are nitrided are generally medium-carbon steels that contain strong nitride-forming elements such as aluminum, chromium, molybdenum, tungsten, and vanadium. Prior to nitriding, the steels are austenitized, quenched, and tempered. Tempering is performed at temperatures between 540 and 750 °C (1000 and 1380 °F), a range above which the nitriding is performed. Tempering above the nitriding temperature provides a core structure that is stable during nitriding.

Nitriding produces a wear- and fatigue-resistant surface on gear teeth and is frequently used in applications where gears are not subjected to high shock loads or contact stress above 1340 MPa (195 ksi) for American Gear Manufacturers Association (AGMA) grade 3, and 1190 MPa (172 ksi) for grade 2 materials. It is particularly useful for gears that need to maintain their surface hardness at elevated temperature. Gears used in industrial, automotive, and aerospace applications are commonly nitrided.

The Gas Nitriding Process

Gears to be nitrided are placed in an airtight container or oven and an atmosphere of NH_3 is supplied continuously while the temperature is raised and held between 500 and 575 °C (930 and 1065 °F) to produce the best combination of surface hardness and case depth. At this temperature, NH_3 breaks down into atomic nitrogen and hydrogen according to the following reaction:

$$2NH_3 \leftrightarrow 2N + 3H_2 \quad \text{(Eq 1)}$$

The atomic nitrogen slowly penetrates into the steel surface and combines with the base metal and the alloying elements such as aluminum, chromium, vanadium, and molybdenum that might be present in the steel selected to form hard nitrides of such elements. Of these, aluminum nitrides offer the highest surface hardness, whereas chromium, vanadium, and molybdenum nitrides result in a deeper and tougher case. Because nitriding takes place at a temperature well below the critical temperature of steel, no molecular change in the grain structure is expected. Therefore, properly nitrided gears exhibit very little distortion and it is a common practice to finish machine gears prior to nitriding. Ideally, the only work required after nitriding is stripping of the mask used for selective nitriding. The masking is generally done with fine-grained copper or nickel plating of thickness not less than 0.025 mm (0.001 in.). Gas nitriding is carried out with gears that are already heat treated, quenched, and tempered for core hardness. This provides a strong, tough core with a hard wear-resisting case, usually harder than obtained by carburizing. Steels selected have carbon content somewhat higher than is employed for carburizing grades to provide support for the hard, brittle case. The chemical compositions of nitriding steels commonly used in gears are shown in Table 1.

To improve the quality of nitriding, some pre-nitriding requirements are important, including:

- Gears are to be free from decarburization. Nitriding a decarburized steel causes excessive growth and the case becomes very brittle and susceptible to cracking and spalling.
- Normalizing and subcritical annealing should occur prior to core hardening.
- Gears are to be core hardened and tempered.
- Nitriding temperature is to be at least 28 °C (50 °F) below the tempering temperature of core-hardened gears.

Gas Nitriding Parameters

Case Depth in Nitriding. Nitrided gears do not require as much case depth as required in carburized gears. This is due to the fact that nitriding is used basically to increase the wear life of gears under a moderate load. The depth of case and its properties are greatly dependent on the concentration and type of nitride-forming elements in the steel. In general, the higher the alloy content, the higher the case hardness. However, higher alloying elements retard the N_2 diffusion rate, which slows the case depth development. Thus, nitriding requires longer cycle times to achieve a given case depth than that required for carburizing. It is also to be noted that higher case depth does not increase the contact fatigue life of nitrided gears in the same ratio as it does in carburized gears. This is due to the fact that hardness drops quickly below the surfaces of nitrided case, whereas, in a carburized case, the drop in hardness is very small. Recommended case depths on different diametral pitch (DP) gear teeth are given in Table 2.

Nitriding Cycle Time. Nitriding is a slow process and it takes hours to develop useful case depths on tooth surfaces. Figure 1 shows some typical cycle times for nitriding versus case depth relationship for commonly used materials, such as the nitriding steels listed in Table 1.

The effective case depth of a nitrided gear tooth should be the depth below the tooth surface at which the minimum hardness of 50 HRC is met for aluminum and high-chromium steels. On the other hand, low-chromium steels do not develop surface hardnesses above 50 HRC. For these steels, effective case depth at 40 HRC works very well (Ref 1). In the case of some gear materials with core hardness approaching 40 HRC, a higher hardness, such as 45 HRC, may be considered for an effective case depth measurement.

Effective case depth is determined by a properly calibrated microhardness tester using a 500 g load and Knoop indenter. The microhardness impressions should be spaced so that they are not disturbed by adjacent impressions. There should be positive assurance that the surface being tested is normal to the indenter. Surfaces to be microhardness tested should be properly polished (scratch-free) and lightly etched in 4% nital solution. When microhardness tests indicate unacceptable results, two additional sets of readings should be obtained—if possible, one on each side of the original set. On small gears, the additional sets may be obtained on adjacent teeth. The three sets of readings should then be averaged. The hardness values are then plotted against their depth from the surface, and a curve is drawn connecting the points. Determine the depth at which the minimum hardness level of 50 HRC crosses the curve.

Total case depth should be determined metallographically using a suitable etch procedure. On an etched tooth section two identifiable bands appear, dark and light. The depth of the dark band is considered to be the total case depth.

Surface Hardness. Because case depth of nitrided gears is generally low, surface hardness measurements are recommended to be taken in Rockwell 15-N scale. Some typical surface

Table 1 Chemical compositions of nitriding steels commonly used in gears

Steel	C	Mn	Si	Cr	Al	Mo	Ni
Nitralloy 135	0.35	0.50	0.30	1.20	1.00	0.20	. . .
Nitralloy 135M	0.41	0.55	0.30	1.60	1.00	0.35	. . .
Nitralloy N	0.23	0.55	0.30	1.15	1.00	0.25	3
AISI 4340	0.40	0.70	0.30	0.80	. . .	0.25	1
AISI 4140	0.40	0.90	0.30	0.95	. . .	0.20	. . .
31CrMoV9	0.30	0.55	0.30	2.50	. . .	0.20	. . .

hardnesses that can be attained after nitriding different alloy steel gears are:

Material	Minimum hardness, Rockwell 15-N
Nitralloy 135M	92.5
Nitralloy N	92.5
Nitralloy EZ	92.5
AISI 4140, 4330, 4340	85.5

Both Nitralloy 135M and Nitralloy N are outstanding materials for gears. Some typical core properties of these two steels before and after nitriding are shown in Table 3.

Table 2 Recommended case depths for nitrided alloy steel gears

Diametral pitch (DP) of tooth	Case depth	
	mm	in.
20	0.127–0.254	0.005–0.010
16	0.203–0.330	0.008–0.013
10	0.305–0.457	0.012–0.018
8	0.356–0.508	0.014–0.020
6	0.406–0.559	0.016–0.022
4	0.508–0.711	0.020–0.028

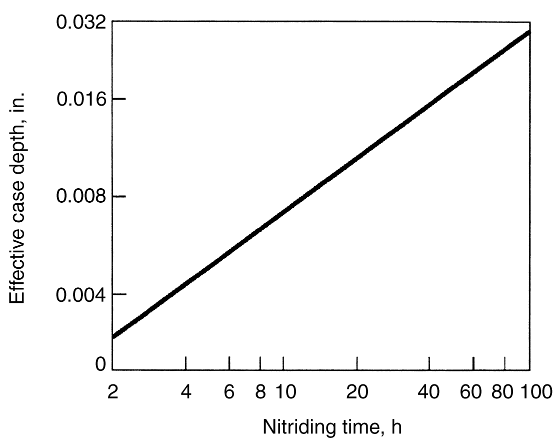

Fig. 1 Nominal time for different nitrided case depths

Core hardness. As with carburized gears, core hardness of nitrided steels is measured in the center of a tooth. See the discussion of "Core Hardness of Gear Teeth" in Chapter 9, "Carburizing," for details (see, in particular, Fig. 17 in Chapter 9).

Recommended Tip and Edge Radii of Teeth. To avoid chipping tooth edges in case of any misalignment of gears during service, it is important to remove sharp corners of gear teeth before nitriding. This is achieved by rounding off the edges. Suggested tip and edge radii are given in Table 4.

White Layer in Nitrided Gears

The microstructure of nitrided steels is comprised of three distinct regions: the compound layer, the diffusion zone, and the core (Fig. 2). The compound layer remains unetched in nital (3% nitric acid in ethyl alcohol) and appears white under the microscope. For this reason, the compound layer is called the "white layer" and consists of a mixture of gamma prime and epsilon iron nitrides. Below the compound nitride layer is the diffusion zone containing precipitated alloy nitrides. From the mechanical strength viewpoint, the white layer is very hard and brittle. The brittleness of this layer is detrimental to gear life, particularly when gears experience misalignment during service. The thickness of the white layer may vary from one material to the other. It is also found that the process used during nitriding has a significant effect on the thickness of this white layer. For example, single-stage gas nitriding may result in a white layer of 0.025 to 0.076 mm (0.001 to 0.003 in.) on materials normally used for gears. On the other hand, it is possible to hold this thickness to 0.0127 mm (0.0005 in.) or less with a two-stage gas nitriding process, also known as diffusion or Floe process. This double-stage

Table 3 Properties of Nitralloy 135M and Nitralloy N

Alloy	Tensile strength		Yield point		Elongation in 2 in., %	Reduction of area, %	Brinell hardness at core
	MPa	ksi	MPa	ksi			
Nitralloy 135 M							
Before nitriding	950	138	830	120	26	60	320
After nitriding	950	138	760	110	4	17	310
Nitralloy N							
Before nitriding	910	132	790	114	2	59	277
After nitriding	1310	190	1240	180	6–15	43	415

process uses two nitriding cycles, the first similar to the single-stage process (except for duration). In the first stage, gears are nitrided at a 15 to 30% dissociation rate of NH_3 for 4 to 12 h at a temperature of approximately 505 to 560 °C (940 to 1040 °F). The longer second stage takes place at a temperature of 525 to 565 °C (975 to 1050 °F) with a dissociation rate of 80 to 85%. To get this high dissociation rate of NH_3, an external dissociator is used. Some typical case hardnesses and depths achieved with double-stage nitriding cycles are shown in Table 5. More detailed information on single- and double-stage nitriding can be found in Ref 2.

Table 4 Tip and edge radii

Diametral pitch (DP)	Maximum edge radius		Maximum tip radius	
	mm	in.	mm	in.
2–5	1.524	0.060	0.381	0.015
6–10	1.270	0.050	0.254	0.010
12–20	0.762	0.030	0.127	0.005

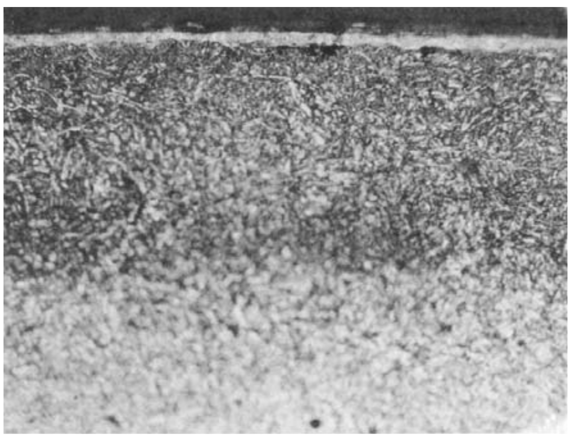

Fig. 2 Typical nitrided case structure showing the white layer (top), the diffusion zone, and the core below the diffusion zone

Any presence of white layer (above 0.025 mm, or 0.001 in.) on the tooth surface is considered very detrimental to the fatigue life of nitrided gears. To improve fatigue life of such gears, as much of this white layer as possible must be removed after nitriding. The most commonly used processes for the removal of this layer are honing, acid pickling, and fine sandblasting. Of these, sandblasting is the most expeditious method, but it destroys the surface finish and profile of gear teeth. Honing is a much superior process, but it is extremely slow. Frequently, a combination of these processes is used to remove the white layer and ensure the quality of the tooth surface and optimize cost. For gear quality (AGMA class 10 and above), honing is recommended.

In case the gears need to be finish-machined to reduce white-layer thickness, it is important to make sure there is only minimal removal of the case. Removal of the case is detrimental to the life of nitrided gears, due to the fact that the hardness of a nitrided surface drops very rapidly along the depth of the case, as shown in Fig. 3. This figure also shows the differences in hard-

Table 5 Double-stage gas nitriding cycles

Steel	Cycle	Effective case depth at 50 HRC		Maximum white layer thickness		Minimum hardness at surface, Rockwell 15-N	Core hardness, HRC
		mm	in.	mm	in.		
Nitralloy 135M	10 h at 525 °C (975 °F), 28% dissociation, 50 h at 550 °C (1025 °F), 84% dissociation	0.457	0.018	0.0127	0.0005	91–92	32–38
Nitralloy N	10 h at 525 °C (975 °F), 28% dissociation, 50 h at 525 °C (975 °F), 84% dissociation	0.356	0.014	0.0127	0.0005	91–92	38–44
AISI 4140	10 h at 525 °C (975 °F), 28% dissociation, 50 h at 525 °C (975 °F), 84% dissociation	0.635	0.025	0.0178	0.0007	85–87	32–38
AISI 4340	10 h at 525 °C (975 °F), 28% dissociation, 50 h at 525 °C (975 °F), 84% dissociation	0.635	0.025	0.0178	0.0007	84.5–86	32–38
31CrMoV9	10 h at 525 °C (975 °F), 28% dissociation, 50 h at 525 °C (975 °F), 84% dissociation	0.559	0.022	0.0178	0.0007	89.3–91	27–33

ness versus case depth gradient of a nitrided surface compared to a carburized and hardened surface. The difference in these two gradients narrows as DP of gears increases. For 20 DP and higher, to avoid removal of any case, complete removal of white layer from a nitrided case is not advocated. Recent investigation on this subject also shows that a white layer on the order of 0.0127 mm (0.0005 in.) or below does not affect the fatigue properties of single-stage nitrided gears, particularly gears that are not heavily loaded (contact stress of 1030 MPa, or 150 ksi, or below). In this region of contact stress, the elastic deformation of contact surface between two mating teeth is usually negligible, and, consequently, tensile stress that develops at the interface of the contact zone is also negligible. Furthermore, the maximum shear stress below the surface at the load center is also relatively small for gears that are not heavily loaded. Hence, the failure of such gears is not due to pitting fatigue but rather to simple wear. To enhance wear life of nitrided gears it is thus necessary to have good surface finish (0.61 μm, or 24 μin., or better) and high surface hardness (60 HRC and higher) on gear teeth. While surface finish depends primarily on the gear cutting and finishing processes, the surface hardness depends on the alloying elements in the gear material. As discussed earlier, aluminum alloy in nitriding type steels (Nitralloy 135M) helps to achieve higher surface hardness, whereas alloying elements such as chromium (AISI 4330) help to obtain higher case depth. With good process controls, Nitralloy 135M provides surface hardness of 60 to 62 HRC and is considered very suitable for gears that are not very heavily loaded. These gears, when nitrided with a diffusion cycle (two-stage process), are not expected to have more than 0.0127 mm (0.0005 in.) of white layer on tooth surfaces.

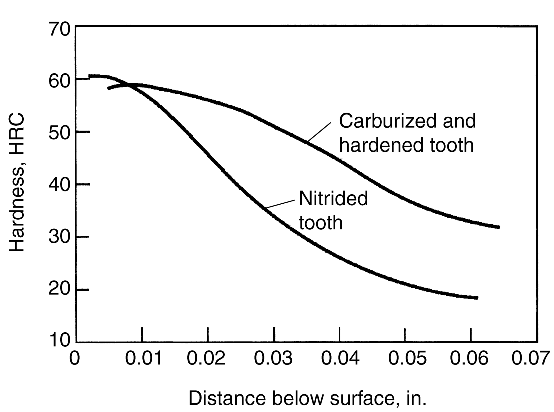

Fig. 3 Comparison of hardness gradients for a carburized tooth and a nitrided tooth

Now, the question is whether a white layer in the range of 0.0127 mm (0.0005 in.) is acceptable for highly loaded gears (contact stress of approximately 1340 MPa, or 195 ksi, for AGMA grade 3 materials). This, of course, depends on the mechanical properties of the white layer and the magnitude of contact stress that is induced by the applied load. In analyzing the properties of the white layer, an interesting characteristic is revealed. It is found that the white layer produced in a two-stage process is somewhat softer and more ductile than the white layer produced in the single-stage process. This allows the white layer of a two-stage process to withstand some elastic deformation. Results obtained so far indicate such a white layer of 0.0127 mm (0.0005 in.) does not detrimentally affect the fatigue life of heavily loaded nitrided gears. The failure in these applications is predominantly controlled by resistance to case crushing at the case-core boundary. To resist this type of crushing, a deeper case with high core hardness is desirable. Materials such as Nitralloy N and AISI 4340 offer these characteristics.

General Recommendations for Nitrided Gears

Several guidelines should be followed when nitriding gears:

- Use a two-stage nitriding process wherever possible.
- Use Nitralloy 135M or a similar material with aluminum alloying addition for gears that are not very heavily loaded. Allow a maximum 0.0127 mm (0.0005 in.) of white layer on tooth surface. A minimum core hardness of 30 HRC is recommended.
- For highly loaded gears where the mode of failure is primarily due to case crushing, select steels with chromium, such as Nitralloy N and AISI 4340. Core-harden tooth to a minimum of 35 HRC.
- All nitrided gear teeth should have proper tip relief on their profile to avoid tip loading that may occur due to tooth deflection or misalignment of gears. No sharp corners are allowed on tooth tips.
- Honing or grinding (not to exceed 0.025 mm, or 0.001 in., stock removal) is accept-

able to remove the white layer, but only if such removal is essential.

- For satisfactory nitriding, liquid-abrasive blast (200–1200 grit size) surfaces prior to nitriding. Burnished or polished surfaces do not nitride satisfactorily.
- No shot peening is necessary for nitrided gears.

Microstructure of Nitrided Cases and Cores

As with carburizing, the microstructure of a nitrided gear tooth changes from the surface to the core. Preferred microstructures for various grades of nitrided gears are given in Table 6. Metallographic standards for core structures and nitrided case are illustrated in Fig. 4 and 5.

Overload and Fatigue Damage of Nitrided Gears

One major drawback of nitrided gears is the reduction of fatigue life subsequent to any momentary overload. The most popular theory of adequately representing the fatigue life of a mechanical element, such as gear teeth subjected to a cumulative fatigue damage, was developed by Palmgren and Miner. This theory is popularly known as Miner's Rule and is now widely used to calculate both bending and pitting fatigue life of gears (Ref 3). According to this theory, failure occurs when the following is satisfied:

$$\sum_i D_i = \sum_i \frac{n_i}{N_i} = 1 \qquad \text{(Eq 2)}$$

where D_i is the damage done to a part due to a certain stress level, n_i is the number of cycles the part experienced at this stress level, and N_i is the number of cycles to failure at that stress level.

Figure 6 shows a typical endurance (*S*-*N*) curve of a gear steel. For gears carburized and hardened, the *S*-*N* curve for original and damaged material due to momentary overload remains more or less the same. But unfortunately, this is not the case with gears that are nitrided. Tests show that the nitrided gears do not have the same overstressing capability as the carburized ones. In fact, the damaged endurance limit of momentarily overloaded nitrided gears seems to be quite a bit below than that of virgin material,

Table 6 Recommended material, attainable hardness, and case microstructure for different applications of nitrided gears

Material	Hardness: Cased surface min, Knoop 500 gram	Hardness: Uncased surface, HRC	Hardness: Core, HRC	Requirement
Grade A, gas nitrided (double stage)				
Nitralloy 135	754	32–38	32–38	Case: Maximum white layer of 0.0127 mm (0.0005 in.) permitted on working surfaces of gear and spline teeth, other surfaces 0.0203 mm (0.0008 in.) max. No soft spots, microcracks, or heavy nitrogen penetration along grain boundaries. No evidence of improper heat treatment in prior processing, excessive ammonia, or excessive temperature. Heavy continuous iron nitride network at tooth tip may be cause for rejection. Core: The core structure should be essentially tempered martensite. Ferrite patches and transformation products (bainite and pearlite) should not exceed 1.586 mm (1/16 in.) in length and width as measured at 250×.
Nitralloy N	732	38–44	38–44	
AISI 4140	540	32–38	32–38	
AISI 4340	540	32–38	32–38	
Grade B, gas nitrided (double stage)				
AISI 4130	500	28–36	23–36	Case: Maximum white layer of 0.0203 mm (0.0008 in.). Tempered martensite with nitride needles. No evidence of decarburization or improper heat treatment. Complete grain boundary network is acceptable. Core: Tempered martensite. Ferrite patches and transformation products should not exceed 3.175 mm (1/8 in.) in width and length as measured at 250×.
AISI 4140	500	28–36	28–36	
AISI 4340	500	28–36	28–36	
Nitralloy 135	754	30–38	30–38	
Nitralloy N	732	38–44	38–44	
Grade C, liquid salt and gas nitrided (single stage)				
AISI 4340	466	28–36	28–36	Case: Compound layer 0.025 mm (0.001 in.) max. Diffusion zone per drawing requirements. Microstructure per AMS 2755
AISI 4140	466	28–36	28–36	
AISI 4130	466	28–36	28–36	
Nitralloy 135	754	28–36	28–36	

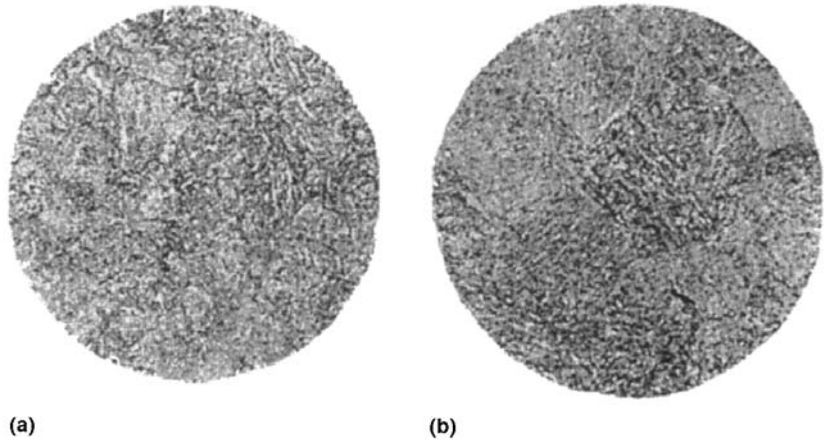

Fig. 4 Metallographic standards for nitrided core structure. (a) Desirable tempered martensite core structure for grades A and B. (b) Maximum acceptable large-grained tempered martensite core structure for grades A and B

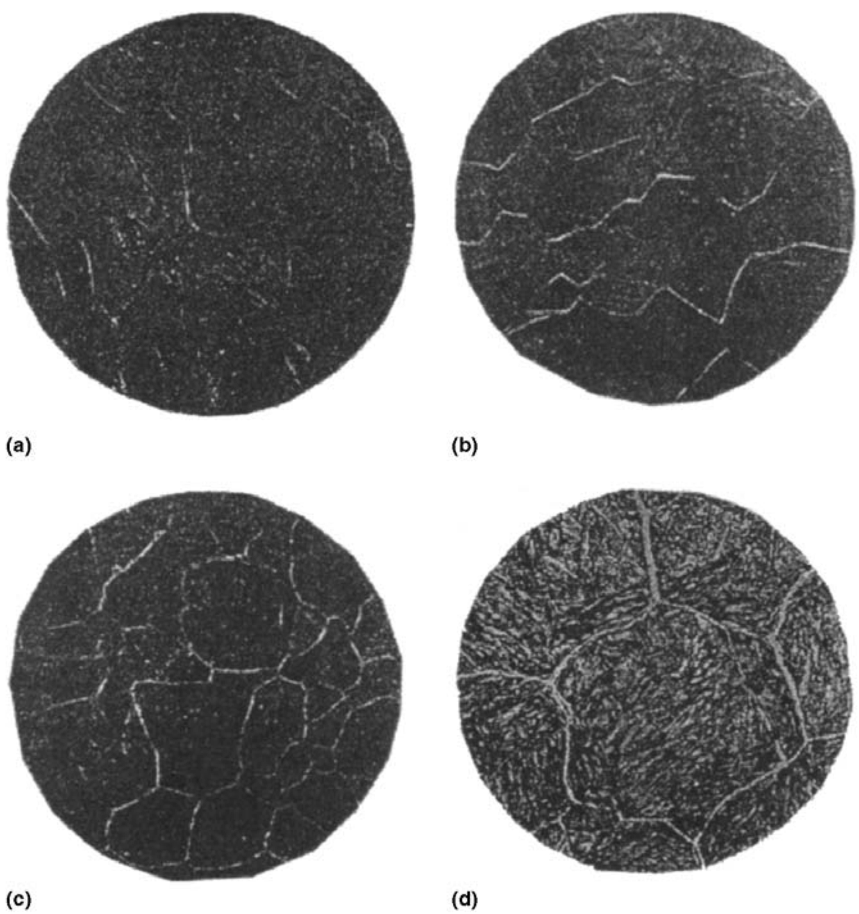

Fig. 5 Metallographic standards for nitrided case structure. (a) Desired nitrided case showing small amount of grain boundary nitride; acceptable for grade A. Dark field illumination. (b) Nitride case with some continuous grain boundary nitrides; maximum acceptable for grade A. Dark field illumination. (c) Nitride case with an increase in continuous grain boundary nitride; maximum acceptable for grades A and B tooth tip. Dark field illumination. (d) Nitride case with complete grain boundary nitrides; not acceptable for grades A or B. Dark field illumination

as depicted in Fig. 7. Therefore, it is advisable to consider this phenomenon while determining the fatigue life of nitrided gears that may be subjected to occasional overload. However, that the life of nitrided gears may also deteriorate when they are subjected to a rated load that varies widely in a very short time period is relatively unknown. A case study on the failure of such gears under wide fluctuating load is presented next.

Case History: Fatigue Failure of Nitrided Gears

As discussed in the preceeding section, case carburized and hardened gears are known to operate satisfactorily both under steady and overload conditions. Nitrided gears, on the other hand, seem to perform well only under steady load. Shock loads detrimentally affect the life of nitrided gears. Thus, nitrided gears are rarely used in applications that experience occasional shock loads. However, that the life of nitrided gears may also deteriorate when subjected to a rated load that varies widely in a very short time period is relatively unknown. Although, such load spectrums are not very common, sometimes these are observed in gas turbogenerator applications. An analysis of a gas turbogenerator load during startup shows that load fluctuations occur whenever the fuel-control system malfunctions. In this case study, the failure of nitrided gears used in a gearbox subjected to a load with wide fluctuations is presented.

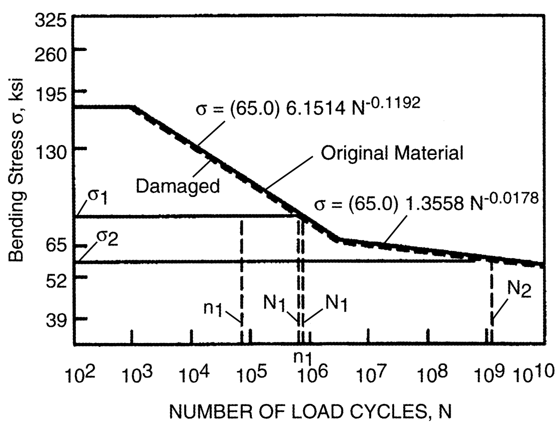

Fig. 6 Bending-fatigue life of original and damaged carburized and hardened gears

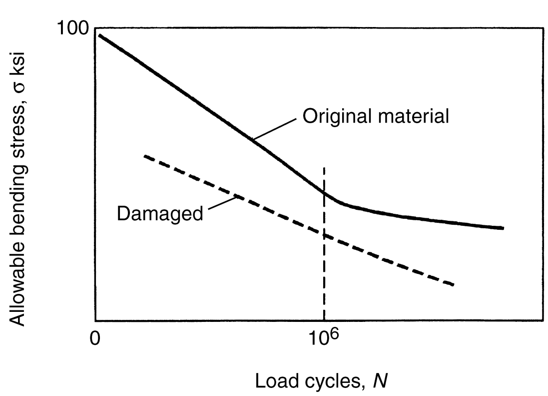

Fig. 7 Bending-fatigue life of original and damaged nitrided gears

Failed Gearbox. The gearbox is of star epicyclic configuration and is used as a speed reducer in between a 10.4 megawatt (MW) gas turbine and an electric generator. The general characteristics of the gearbox are:

- *Horsepower rating:* 16,500 hp
- *Input speed:* 8,625 rpm
- *Output speed:* 1,500 rpm

The sun pinion and planet gears are nitrided, whereas the ring gear is just through-hardened. The design life of the gears is a minimum of 100,000 h.

Failure Incident. According to a field representative, gearbox failure occurred after 3 h of operation following a scheduled online turbine water wash cycle. In general, all turbines go through a water wash cycle after a few thousand hours of operation. During such a cycle for this turbine, the generator load is brought down to approximately the 8 MW level and then water is injected through the compressor section of the turbine. Following wash, the load is raised to the maximum continuous level of 10.4 MW. After the last wash cycle, high vibration levels of the gearbox were reported just before the gearbox failed. Unfortunately, no data were taken. The gearbox had a total of approximately 16,000 h of service before the failure.

Following this incident, to ensure the failure was not caused by misalignment of equipment or any malfunction of components, such as bearings, couplings, or a shaft, a thorough inspection was conducted. The equipment alignment of the package was found to be within the allowable tolerance. Also, no visible damage of any component was noticed. This led the inspection toward the internal components of the gearbox.

Visual inspection of the disassembled gearbox revealed severe mechanical damage to all gears. The failure was so catastrophic that one complete tooth fell off one of the planet gears and was found at the bottom of the housing. Figure 8 shows the damaged planet gear, and Fig. 9 shows the severed tooth.

Tooth debris was noticed in various gear meshes. No abnormal distress was found on planet sleeve bearings, although the thrust faces of output shaft sleeve bearings were completely wiped out. Also, sealing surfaces of the output shaft and labyrinth seal were found to be burnt.

To determine if there were any discrepancies with the quality of gear material and heat treatment, detailed metallurgical evaluation was carried out with a section from the failed tooth.

Metallurgical Evaluation. Examination of the fragmented tooth revealed some well-defined beach marks (Fig. 10) suggesting a typical fatigue failure. The tooth segment was also analyzed by an energy-dispersive x-ray spectroscope (EDS) and examined with a scanning electron microscope (SEM). The primary fracture surface was relatively flat and showed well-defined crack growth marks, as illustrated in Fig. 11. Features of transgranular fatigue propa-

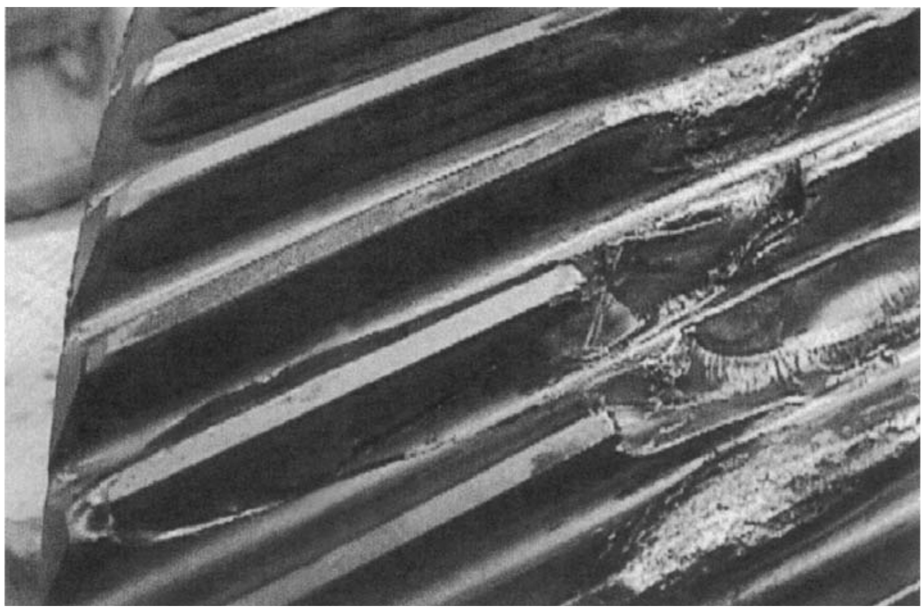

Fig. 8 Overall view of one damaged planet gear

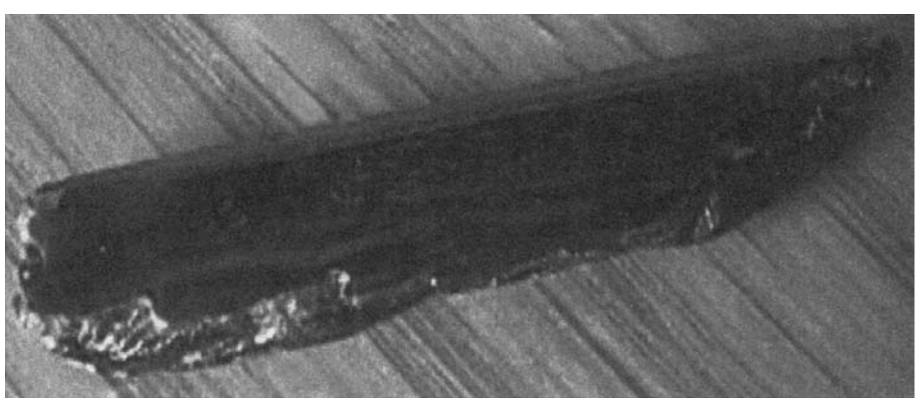

Fig. 9 View of dislocated tooth

gation were identified on the fracture surfaces (Fig. 12), confirming cracking of the tooth due to a bending-fatigue mechanism. Material certification shows gears were made from European nitriding grade steel (EN-4B), and the chemical composition seems to meet the requirements as shown below:

- C: 0.2–0.28
- Si: 0.10–0.35

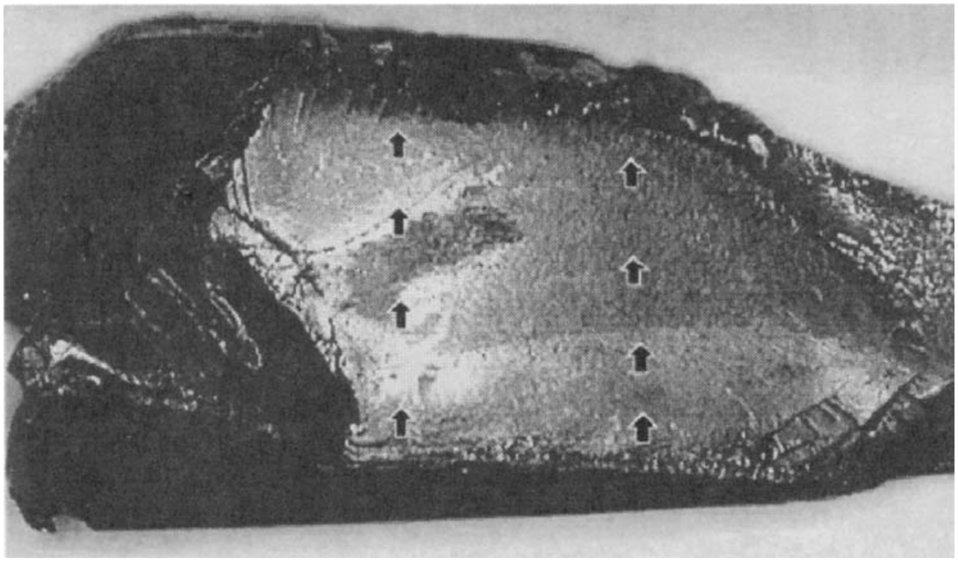

Fig. 10 Detail view of the received fragment from the primary failed tooth. Arrows indicate the crack propagation direction. 4×

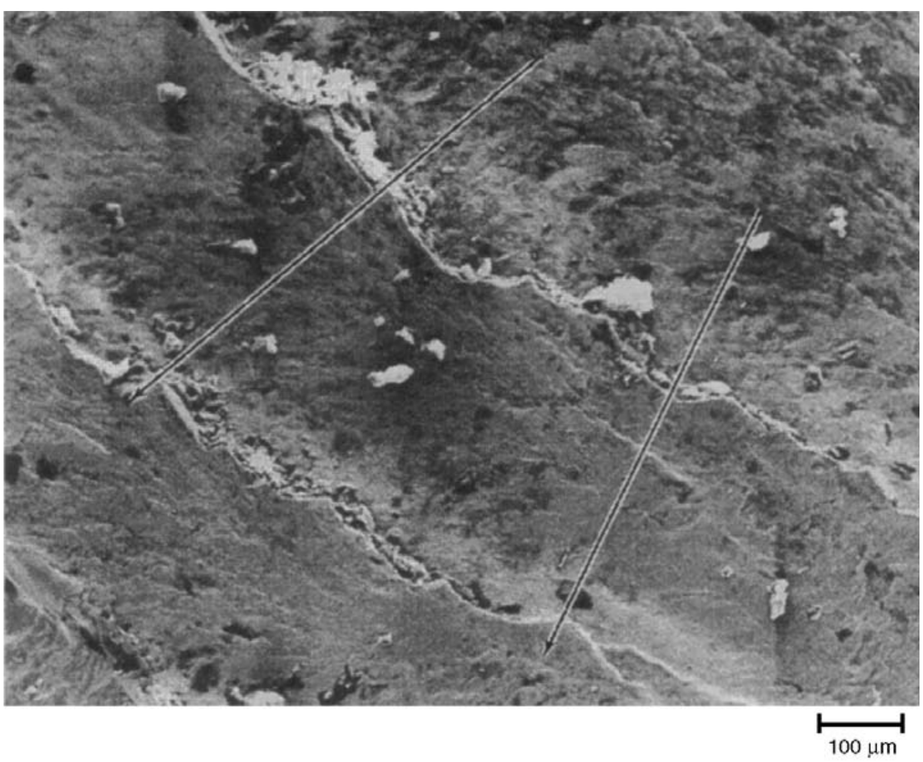

Fig. 11 Scanning electron microscope view of the fracture surfaces of the gear fragment. Arrows indicate the crack propagation direction. 100×

- Mn: 0.45 max
- Cr: 3.00–3.50
- Mo: 0.45–0.65
- S: 0.025 max
- P: 0.025 max

The case-hardness profile and the effective case depth of teeth were determined by a microhardness tester and are shown in Fig. 13. The effective case depth was found to be 0.381 mm (0.015 in.). This satisfies the design requirements for these gears. The core hardness of tooth varied from 25.8 to 27.5 HRC. The desired minimum core hardness is 28 HRC.

Failure Analysis. In a mature design of a gearbox as in this case, the failure is believed to be due to the influence of one or more of the following factors:

- Defective gear material or forging
- Improper heat treatment of gears
- Sudden overload beyond design consideration
- High fluctuating load

In this analysis, failure initiation due to inadequate lubrication is ruled out because no evidence, such as discoloration of tooth or any burn marks, was observed on teeth surfaces. Also, vibration was not believed to be the cause because no abnormal vibration levels were reported while the generator set was running following the turbine wash cycle. Thus, it is justified to assume that gearbox health was reasonably satisfactory for some time before the failure occurred.

In regard to material, the chemical analysis shows an acceptable alloy composition. Also, there were no adverse reports on forging quality, such as banding or grain flow. Therefore, material quality did not seem to play any significant role in the failure mechanism.

Hardness and case depth profile, along with the case microstructure, indicate there was not much discrepancy during heat treatment. Lower core hardness of the planet gears has some influence on the reduction of bending fatigue life but not to the extent as in this case. Because there were no records of any sudden overload, it is believed the major cause of failure was load fluctuation.

Analysis of the generator load spectrum (Fig. 14) subsequent to an online engine wash cycle shows quite a bit of load fluctuation—from 4 to 12 MW in a short interval of time (1.5 cycles per second for approximately 25 seconds). Field reports indicate this type of loading occurred many times since the generator set was commissioned. Nobody paid any attention to this

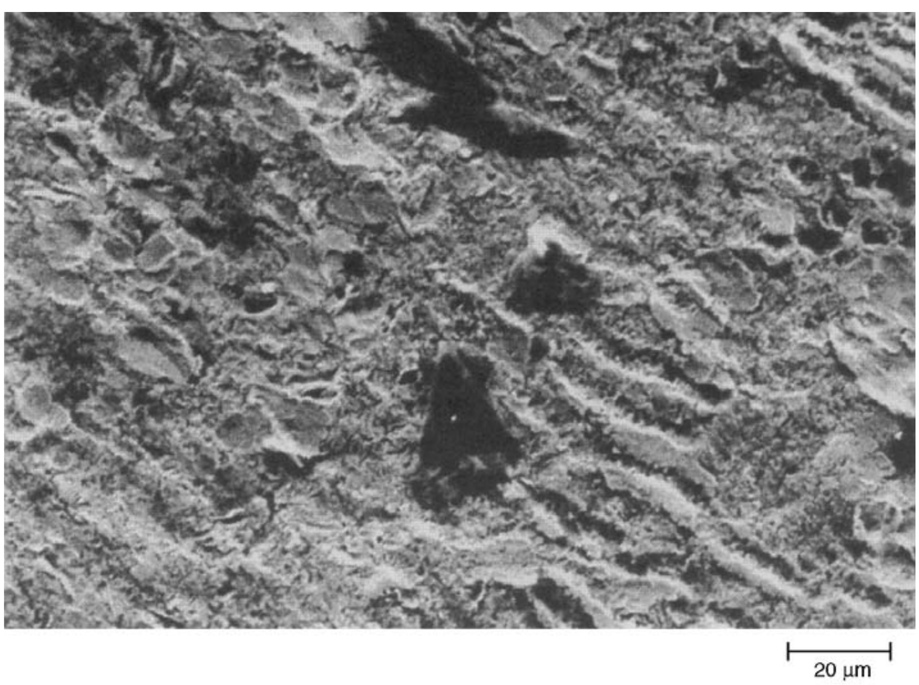

Fig. 12 Scanning electron microscope view of the fracture surfaces of the gear fragment showing transgranular fatigue propagation. 500×

loading pattern because it did not represent a real overload condition for the gearbox designed with an overload capacity of four times the nominal rating. That a high fluctuating nominal load could be as damaging as overload to the nitrided gears was simply unknown. Although reduced allowable bending fatigue strength was considered in designing the planet gears for reverse bending, it did not take into account any possible change in the slope of the *S-N* curve (damage line) due to the high magnitude of a fluctuating load.

It has been shown that the slope of *S-N* curves for both bending and pitting shifts downward when nitrided gears are subjected to any overload, as illustrated in Fig. 7. Conversely, the slope of *S-N* curves for carburized and hardened gear steels changes very little, as shown in Fig. 6. Therefore, it is certain that the allowable fatigue strengths for both sun and planet gears that were nitrided had been reduced due to high fluctuating load. With reduced allowable fatigue strength, one of the planet gears that might have been metallurgically weakest of all failed first. It is believed the fragments of the failed tooth then went into mesh, causing the premature failure of the gearbox.

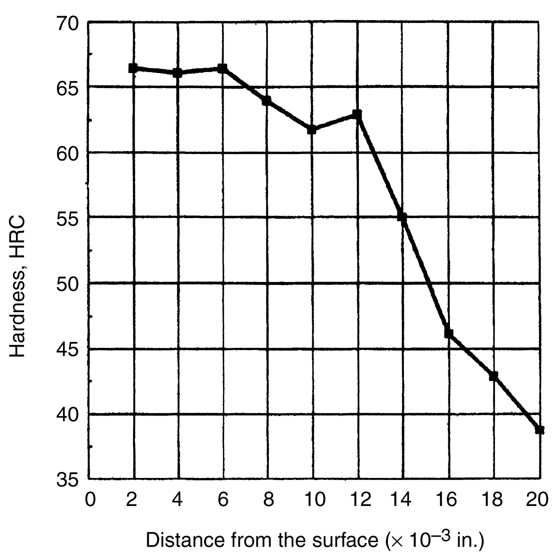

Fig. 13 Gear tooth microhardness profile

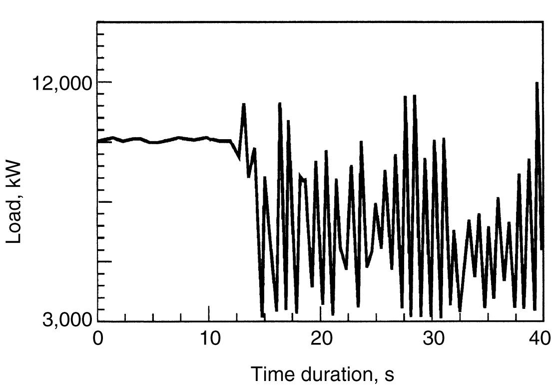

Fig. 14 Fluctuation of generator load after online engine washing

Discussion. The fluctuating load following a turbine water wash cycle was due to some instability in the fuel-control system. This sort of instability and, hence, load fluctuation may be avoided by replacing the online wash of the turbine with a stationary wash procedure. Such an arrangement will definitely improve the life of such gearboxes.

Conclusions. The fatigue life of nitrided gears is reduced not only under overload but may also be reduced under highly fluctuating nominal load. The failure due to overload or fluctuating nominal load definitely accelerates with the level of fatigue already induced by previous stress cycles in the nitrided gears. In this case, it is believed that 16,000 h of turbine operation contributed to some extent in the failure of gears. Therefore, it is advisable to consider lower allowable fatigue strengths when designing nitrided gears that are subjected to overload or occasional highly fluctuating load. Also, a cleaner material that meets AMS 2300 or 2304 is recommended in such applications for reliable and high-fatigue strength.

Bending-Fatigue Life of Nitrided Gears

To increase bending-fatigue life it is beneficial to have compressive stress at the surface of the tooth root fillet, the location of maximum bending stress. The mechanism for obtaining the compressive stress on a nitrided surface is to restrict the growth of case by the core during nitriding process. For small DP gear teeth with the section size much greater than the hardened case, the expansion in the nitrided layer cannot alter the dimensions of the tooth. This action simply tends to stretch the core and, in so doing, places the surface in compression. On the other hand, in large DP (smaller tooth section) gears, the case growth cannot be restricted by a thinner section of the tooth and results in greater growth, which generates excessive tensile stress in the case. This lowers the residual compressive stress in the case and thereby reduces the fatigue life. This is why small DP nitrided gears show higher bending fatigue life than large DP nitrided gears.

Distortion in Nitriding

As with any other heat-treating process, nitriding causes gears to experience distortion, although its severity is less in comparison to that with other processes because this process is carried out at a relatively low temperature, approximately 510 °C (950 °F). Because of the low temperature, no phase transformation of steel, the major cause of distortion, takes place. The small dimensional changes that occur are due to heating and cooling mechanisms and the gear blank configuration. Sometimes, higher distortion is observed due to high case depth requirements in some gears that require many hours of nitriding. In general, gas-nitrided gears with a case depth less than 0.254 mm (0.010 in.) do maintain the same AGMA quality level after nitriding. Gears with case deeper than 0.254 mm (0.010 in.) experience some distortion resulting in lower quality level. Furthermore, pitch diameter (PD) and DP of a gear tooth are also important parameters that influence distortion. Large PD with a small DP tooth distorts more than a small PD with a large DP tooth. Distortion is also influenced by the material selected. Some materials distort more than others. In general, distortion of gears after nitriding is low. Occasionally, the distortion becomes high due to large size and configuration of gears (e.g., internal gears in epicyclic arrangements). Higher distortion is also noticed in gears with selective nitriding (e.g., nitriding of teeth only). Table 7 presents general distortion ratings of various steels after single-stage gas nitriding.

Although some distortion is always present in any type of nitriding, the process has a unique advantage: reproducibility of the distortion in batch after batch of gears. Gears of identical geometry and similar metallurgical quality distort exactly the same way. This means compensation for expected distortions can be made during tooth cutting prior to nitriding.

Gas Nitriding Applications and Costs

Applications. While carburizing is the most effective surface-hardening method, nitriding excels when gear tooth geometry and tolerances before heat treating need to be maintained without any finishing operation, such as grinding after heat treatment. Although, through-hardening is capable of maintaining close tooth dimensional tolerances, the process cannot provide sufficient wear and pitting resistance. This is why nitriding is an alternative to carburizing especially for lightly loaded gears.

The major disadvantage of nitrided gears is their inability to resist shock load due to inherent brittleness of the case. Also, nitrided gears do not perform well in applications with possible misalignment during which the highly brittle nitrogen oxides on tooth edges break off and may go into the gear mesh.

The quality of nitrided gear teeth is not as good as carburized, hardened, and ground gears. Grinding to improve tooth geometry is not recommended for nitrided gears because this may detrimentally affect their load carrying capability if more than 0.025 mm (0.001 in.) stock is removed.

Costs. Higher case depth requires a longer nitriding cycle with an increase in cost. Steels with chromium need shorter nitriding cycles than steels with aluminum for the same case depth. Cost factors for one nitriding steel, Nitralloy 135M, are given in Table 8.

Wider case depth tolerance is preferred for reduced costs. Table 9 gives case depth tolerances and costs for Nitralloy 135M.

Partial coverage increases cost and should be avoided if possible. As shown in the table given below, the cost of masking and copper plating

Table 7 Distortion ratings of common nitriding gear materials

Material	AMS specification	AMS quality	Hardness, HRC		Distortion rating
			Case	Core	
Nitralloy 135M	6471	2300	60–65	32–36	Good
Nitralloy EZ	6475	2300	60–65	34–38	Good
Nitralloy N	6475	2300	60–65	40–44	Good
AISI 4140	6349	2300	50–55	25–32	Good
AISI 4340	6414	2300	48–53	28–34	Good
Maraging 300	6419	2300	57–59	52–54	Excellent

for selective nitriding contribute to a high cost factor:

Surface coverage	Cost factor
Complete	1.0
Partial	3.5

Controlled Nitriding (Ref 4, 5)

Controlled nitriding is a further development of the traditional gas nitriding in which all of the process parameters are computer controlled. The key to the success of the process lies in its ability to effectively control the concentration of nitrogen in the surface layer of the treated parts. By controlling the activity of nitrogen in the gas atmosphere, it is possible to control the activity of nascent nitrogen, a factor that determines the nitriding capability of the atmosphere. The technology involves measuring and adjusting what is termed the nitriding potential, N_p, which is expressed by the formula:

$$N_p = \frac{P_{NH_3}}{(P_{H_2})^{1/2}} \qquad \text{(Eq 3)}$$

where P_{NH_3} is the partial pressure of ammonia, and P_{H_2} is the partial pressure of hydrogen in the furnace atmosphere.

It should be noted that the term *potential* is in no way equivalent to what is meant by carbon potential in carburizing atmospheres. It is, rather, the reaction constant for the dissociation of ammonia. Its control allows the formation of predictable case structures and depths for a range of steels, including some tool and stainless grades. Higher N_p values produce higher surface concentrations of nitrogen and steeper concentration gradients. Lower potentials allow the development of nitrided cases without any brittle-compound (white) layer in high-alloy steels. This can be of particular advantage in certain applications, such as aerospace gears subject to higher contact stresses. In these applications, the brittleness and/or unpredictability of the white layer produced by conventional nitriding is detrimental.

Table 8 Case depth and cost for Nitralloy 135M

Case depth		
mm	in.	Cost factor
0.254	0.010	1.0
0.004	0.012	1.0
0.356	0.014	1.4
0.406	0.016	1.6
0.457	0.018	2.0

Table 9 Case depth tolerances and cost for Nitralloy 135M

Case depth tolerance		
mm	in.	Cost factor
±0.051	±0.002	1.4
±0.076	±0.003	1.0
±0.102	±0.004	0.8
±0.127	±0.005	0.7

Note: Similar cost factor is applicable to all other nitriding materials.

The controlled nitriding process differs from conventional gas nitriding in that it does not use pure ammonia. Instead, other gas components—including nitrogen, previously dissociated ammonia, hydrogen, argon, and oxygen—are added in a controlled way to conventional ammonia. The effect of these additions, with the exception of argon, is to reduce the nitriding potential of the atmosphere; argon has no effect on N_p. Controlled gas nitriding expands on the advantages of nitriding, such as corrosion resistance and low distortion, and expands the range of metals suitable for this surface-hardening process.

How the Process Works. This system makes it possible to adjust the availability of atomic nitrogen at the gas-metal interface and to precisely control the properties by changing the composition of the gas mix. This is done not only by selecting the correct atmosphere composition for a given material and process temperature but also by maintaining the preset nitriding potential value for each stage of the process.

Temperature and gas-flow controls, linked by a computer connected to a nitriding potential-sensing device, allow the programming of the complete process cycle to produce predictable and reproducible surface-layer characteristics for a given material. The control system can store a number of process recipes that may be selected for specified materials and applications. When the operator selects a process from the software's "library," the time, temperature, atmosphere composition, and nitriding potential for each stage of the cycle are automatically controlled.

The computer program also controls functions such as opening and closing of valves, switching electric motors on and off, and maintaining optimal pressure inside the retort. The operator may manually select less critical factors, such as printout frequency and the temperature at which the process is to stop after cooling.

Process Advantages. The control of the nitrogen concentration in the surface zone by maintaining a preset nitriding potential of the atmosphere offers several advantages over traditional gas nitriding and other competitive surface treatment methods:

- Improved ductility and toughness of the compound layer, reducing the risk of cracking or spalling
- Improved surface roughness values
- Improved microhardness profiles (Fig. 15)
- Cost-effectiveness through optimized cycle times
- Often, even lower temperatures than conventional nitriding, thus further minimizing distortion
- A broader range of treatable steel grades
- Enhanced corrosion resistance through the introduction of additional compounds into the surface layer

In addition, by eliminating the white layer where it is not wanted, the process expands on the traditional advantage of nitriding not needing mechanical finishing operations. It also limits the surface roughness that accompanies nitriding by appropriately structuring the nitriding stages. The process is ecologically more "friendly" than conventional gas nitriding, because less ammonia is used overall.

Ion Nitriding

Ion, or plasma, nitriding is a vacuum process that takes place in the plasma of high current glow in a vessel. The workpiece, in this case, a gear, forms the cathode, whereas the vessel wall is the anode. The vessel is evacuated prior to nitriding. Gas containing nitrogen is then introduced, and the treatment pressure in the vessel is set to between 0.1 to 10 torr (0.13 to 13.3×10^2 Pa). At this point, electric voltage is switched on and a glow discharge takes place; the nitrogen ions thus produced strike the surface of the cathode with high kinetic energy emitting heat that results in a sputtering of the cathode, which atomizes the cathode (gear) surface material. The temperature inside the vessel may vary from 350 to 580 °C (660 to 1080 °F). The atomized ions then combine with nitrogen ions in the plasma to form iron nitride, which then is deposited as an even iron nitride layer on the cathode. In turn, the iron nitrides are partially broken down on the surface of the cathode, whereby the nitrogen diffuses into the gear material.

As the process continues, the iron from the iron nitrides is partially dusted off in the plasma in front of the workpiece surface. Based on this continual dusting process, the surface of gears being treated is kept active longer. The advantage is that the nitride produced does not immediately act as a diffusion blocker, as in the case of conventional gas nitriding. The case produced by plasma nitriding is thus thicker and free of pores because of the constant ion bombardment. Also, in comparison with the case produced in gas or liquid nitriding process, ion-nitrided cases are more ductile and more resistant to wear.

In ion nitriding, as in other nitriding processes, the nitrided layer has a diffusion zone

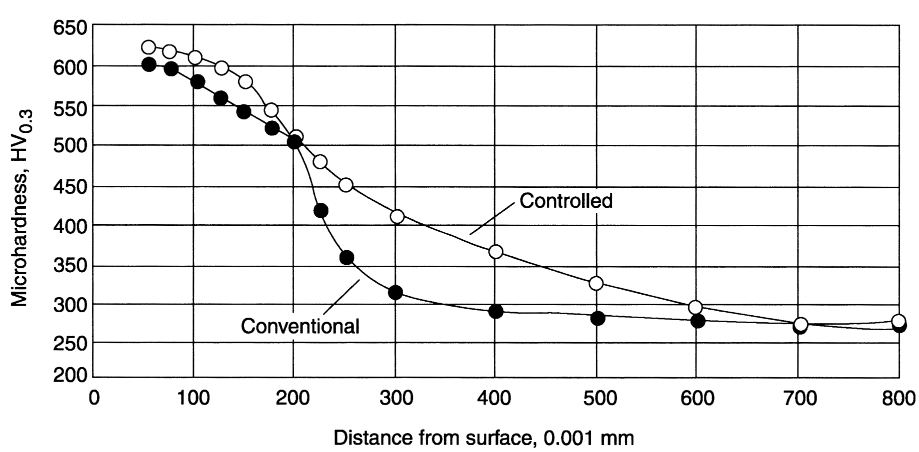

Fig. 15 Microhardness profiles of EN 19 chromium-molybdenum steel produced by controlled and conventional gas nitriding processes. Source: Ref 5

and a compound layer. In the diffusion zone, nitrogen diffuses in steel according to classical principles producing a hardened zone by precipitation and solid solution hardening. This zone is detected as a dark etching region below the surface, varying in hardness from the surface to the core. This region is measured as case and may be as deep as 0.9 mm (0.035 in.), depending on the type of steel, nitriding cycle time, and temperature. On the other hand, the compound layer above the diffusion zone known as "white layer" is essentially composed of pure nitrides of iron and is very brittle. In ion nitriding, the white layer is usually below 0.0127 mm (0.0005 in.). It is possible to reduce the thickness of this layer further by controlling the ratio of nitrogen in the nitrogen and hydrogen gas mixture during ion nitriding.

Basically, any type of ferrous gear materials can be ion nitrided. Heat treatable steels are particularly suitable. Table 10 shows the results of ion nitriding for some typical gear materials.

To maintain the properties as listed in Table 10 after ion nitriding, it is essential that there are no traces of rust, paint, or grease on gears because these would prevent nitrogen from being absorbed into the tooth surfaces. Also, it is advisable to have lower arithmetic surface roughness average (R_a) value of these surfaces before nitriding because surface roughness increases in most materials after nitriding due to nitrogen absorption into the surface. The longer the process is, the greater the increase of roughness, although the level of roughness can be returned to its original state by honing or lapping at an additional cost.

Honing or lapping also may be used to remove the white layer. If grinding (not to exceed 0.025 mm, or 0.001 in., stock removal) is used instead, the part shall be stress relieved at 160 to 280 °C (325 to 540 °F) after grinding. Removal of white layer by chemical means is not advisable. Allowable white layer should not exceed 0.0127 mm (0.0005 in.) and shall be of single-phase Fe_4N (gamma prime iron) composition.

For selective ion nitriding, masking by mechanical means such as plate covers, which act as barriers between the glow of discharge and the part surfaces, is advisable. Masking with copper plating causes sputtering and is not advisable. As an alternate to masking, parts may be nitrided all over first and then the surfaces ground where case is not desired or required.

Dimensional growth is minimal during ion nitriding. The growth is dependent on the quantity of nitrogen deposited. High-alloy steels take up more nitrogen than those with a low-alloy content. Hence, there is more growth of high-alloy steel gears. In case of insufficiently annealed and normalized steels, the annealing effect of ion nitriding may cause disintegration of the stable retained austenite, leading to a disproportional growth and distortion. Also, ion nitriding can cause a decrease in volume in gears made of martensitic steels. As in other heat treat processes, dimensional change of gears is better determined by a preproduction run. In the absence of such data, a growth of 0.025 mm per 0.127 mm (0.001 in. per 0.005 in.) diffusion depth may be considered for most gear steels.

In ion nitriding, the discharge parameters of voltage and current determine the supply of active ions, and that nitriding potential of the low-pressure atmosphere in the chamber is essentially independent of the temperature of the charge, unlike conventional gas nitriding. This capability results in a number of unique advantages; for example, materials that would lose their core strength under conventional nitriding conditions can be ion nitrided because the prerequisite for maintaining the core strength is that nitriding temperature be below the temper temperature. This also reduces distortion considerably.

Ion-Nitriding Time and Case Depth. For the same processing time, case depth attained

Table 10 Results after ion nitriding of common gear materials

Steel group	Core hardness, HRC	Nitriding temperature, °C (°F)	Surface hardness, 15-N scale	Total case depth, mm (in.)	Thickness of white layer, mm × 10^{-4} (in. × 10^{-4})	White layer composition
AISI 9310	28–32	520–550 (970–1020)	89.0	0.305–0.711 (0.012–0.028)	38.10–80.01 (1.50–3.15)	Fe_4N
AISI 4130	28–36	510–550 (950–1020)	89.0	0.203–0.660 (0.008–0.026)	38.10–80.01 (1.50–3.15)	Fe_4N
AISI 4140	34–38	510–550 (950–1020)	89.0	0.203–0.610 (0.008–0.024)	38.10–80.01 (1.50–3.15)	Fe_4N or none
AISI 4340	38–42	510–550 (950–1020)	89.0	0.254–0.635 (0.010–0.025)	38.10–80.01 (1.50–3.15)	Fe_4N or none
Nitralloy 135M	28–32	510–550 (950–1020)	92.0	0.203–0.508 (0.008–0.020)	20.32–100.33 (0.8–3.95)	Fe_4N
Nitralloy N	25–32	510–550 (950–1020)	92.0	0.203–0.508 (0.008–0.020)	25–100.33 (1–3.95)	Fe_4N

with ion nitriding is higher than gas nitriding. Figure 16 shows the results of case depth versus treatment time for AISI 4140 steel ion nitrided at 510 °C (950 °F) and gas nitrided at 525 °C (975 °F). Similar results also are achieved with Nitralloy 135M steel. As in gas nitriding, higher case depth is obtained with AISI 4140 steel than with Nitralloy 135M. But Nitralloy 135M produces higher surface hardness. Similar to gas nitriding, time for ion nitriding varies with case depths and material. Figure 17 depicts time required for different case depths and two gear materials.

Hardness and Case Depth. Case-hardened samples of AISI 4140 and 4340 steels produced by ion-nitriding process showed encouraging results as tabulated in Table 11. The major shortcoming of earlier ion-nitriding processes was the large variation in case depth from area to area of teeth. However, the current improved process offers fairly uniform case depths. Some typical microhardness traverses for gear tooth made of AISI 4140 and 4340 steels are illustrated in Fig. 18 and 19, respectively.

Improved Case Property. Ion-nitrided surfaces, especially with high-alloy steels, show

Table 11 Summary of metallographic measurements of ion nitrided specimens

Measurement	4140 Sample	4340 Sample	Requirement
Case depth, mm (in.)	Over 0.889 (over 0.035)	0.508 (0.020)	0.51 (0.02)
Surface hardness, HK	690	604	540 min.
Core hardness, HRC	36	35	32–38
Grain size (ASTM)	No. 7	No. 6	No. 5 or finer
White layer depth, mm (in.)	0.0025 (0.0001)	0.0051 (0.0002)	None

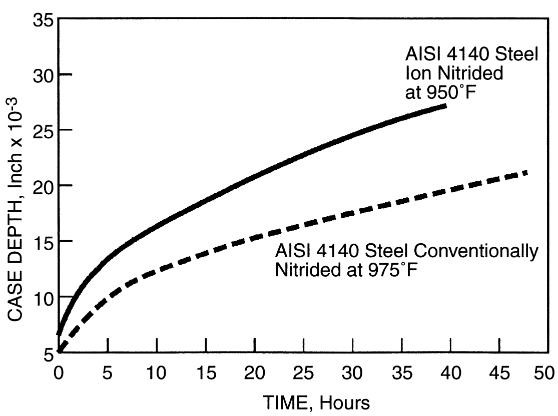

Fig. 16 Comparison of case depth vs. process time for ion and conventional nitriding

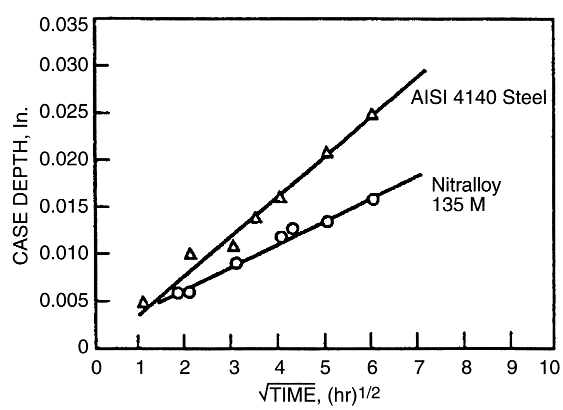

Fig. 17 Case depth vs. square root of ion-nitriding time for two materials

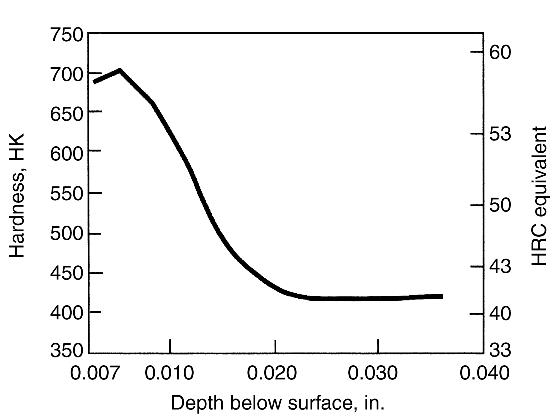

Fig. 18 Microhardness traverse of AISI 4140 steel gear sample

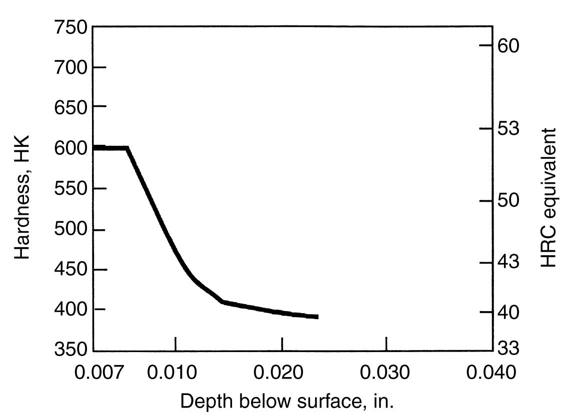

Fig. 19 Microhardness traverse of AISI 4340 steel gear sample

Table 12 Distortion ratings of some ion-nitriding materials

Material	AMS specification	AMS quality	Nitriding temperature, °C (°F)	Hardness, HRC		Distortion rating
				Case	Core	
AISI 9310/9315	6265	2300/2304	520–550 (970–1020)	50/58	28/32	Good
AISI 4340	6414	2300/2304	510–550 (950–1020)	54/58	38/42	Good
Nitralloy 135M	6471	2300/2304	510–550 (950–1020)	60/62	28/32	Good
Nitralloy N	6475	2300/2304	510–550 (950–1020)	60/62	26/32	Good

comparatively good ductility. This characteristic is due to the ability to closely control the amount and type of white layer. High ductility results in high fatigue properties.

Distortion. Because gears are ion nitrided under vacuum, only the stresses originating from the thermal and preliminary treatments lead to distortion. For this reason, to minimize distortion, gears are stress-free annealed before ion nitriding. The temperature of the stress relief is maintained at least 25 °C (75 °F) over the ion-nitriding temperature. Also, slower cooling after the stress relief is recommended. A second stress relief is sometimes beneficial after rough machining. If done properly, the distortion is expected to be minimal. Should the core hardness be diminished through stress relieving to a value too low, an alternate material should be used. Table 12 presents comparative distortion ratings of some ion-nitriding materials.

ACKNOWLEDGMENTS

This chapter was adapted from:

- A.K. Rakhit, Nitriding Gears, *Heat Treatment of Gears: A Practical Guide for Engineers,* ASM International, 2000, p 133–158
- A.K. Rakhit, Modern Nitriding Processes, *Heat Treatment of Gears: A Practical Guide for Engineers,* ASM International, 2000, p 159–169
- J.R. Davis, Ed., Nitriding, *Surface Hardening of Steels: Understanding the Basics,* ASM International, 2002, p 141–194

REFERENCES

1. D.W. Dudley, Gear Materials, *Handbook of Practical Gear Design,* McGraw-Hill Book Company, 1984, p 4.1–4.61
2. Gas Nitriding, *Heat Treating,* Vol 4, *ASM Handbook,* ASM International, 1991, p 387–409
3. R. Errichello and D.R. McVittie, Predicting Gear Life with Miner's Rule, *Power Transmission Design,* Vol 31 (No. 3), March 1989, p 33–38
4. G.J. Tymowski, W.K. Liliental, and C.D. Morawski, Typical Nitriding Faults and Their Prevention Through the Controlled Nitriding Process, *Ind. Heat.,* Jan 1995, p 39–44
5. G.J. Tymowski, W.K. Liliental, and C.D. Morawski, Take the Guesswork Out of Nitriding, *Adv. Mater. Process.,* Dec 1994, p 52–54

SELECTED REFERENCES

- Plasma (Ion) Nitriding, *Heat Treating,* Vol 4, *ASM Handbook,* ASM International, 1991, p 420–424
- D. Pye, Nitriding Techniques and Methods, *Steel Heat Treatment Handbook,* G.E. Totten and M.A.H. Howes, Ed., Marcel Dekker 1997, p 721–764
- D. Pye, *Practical Nitriding and Ferritic Nitrocarburizing,* ASM International, 2003

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CHAPTER 11

Carbonitriding

CARBONITRIDING is a modified form of gas carburizing, rather than a form of nitriding. The modification consists of introducing ammonia into the gas carburizing atmosphere to add nitrogen to the carburized case as it is being produced. Nascent nitrogen forms at the work surface by the dissociation of ammonia in the furnace atmosphere; the nitrogen diffuses into the steel simultaneously with carbon. Typically, carbonitriding is carried out at a lower temperature and for a shorter time than is gas carburizing, producing a shallower case than is usual in production carburizing.

Carbonitriding is performed in a closed furnace chamber with an atmosphere enriched with gaseous compound of carbon and nitrogen. Many types of gas are used. Common practice is to use an endothermic gas, such as natural gas, as a carrier for the ammonia and hydrocarbons. The process primarily imparts a hard, wear-resistant case. A carbonitrided case has better hardenability than a carburized case. Also, the presence of N_2 in the case, as in nitriding, increases its hardness. Most carbonitriding is done between 770 and 890 °C (1425 and 1625 °F) for gears to be liquid quenched and between 650 and 790 °C (1200 and 1450 °F) for gears not requiring liquid quench.

The addition of nitrogen has four important effects:

- It inhibits the diffusion of carbon, which favors production of a shallow case.
- It enhances hardenability, which favors attainment of a very hard case.
- It makes it possible to form martensite in plain carbon and low-alloy steels that initially have low hardenability.
- Nitrides are formed, increasing the surface hardness further.

Nitrogen, similar to carbon, manganese, and nickel, is an austenite stabilizer, so retained austenite is a concern after quenching. Lowering the ammonia percentage reduces the amount of retained austenite and should be done if decreases in hardness or wear resistance cannot be tolerated. Another consequence of too-high nitrogen is the formation of voids or porosity. In general, the nitrogen content at the surface should be no greater than 0.40% (Fig. 1).

The nitrogen in the carbonitrided case also increases the resistance of steel to softening at slightly elevated temperatures. The greater the nitrogen content, the greater the resistance to softening. This is why higher tempering temperatures—up to 225 °C (440 °F)—are often

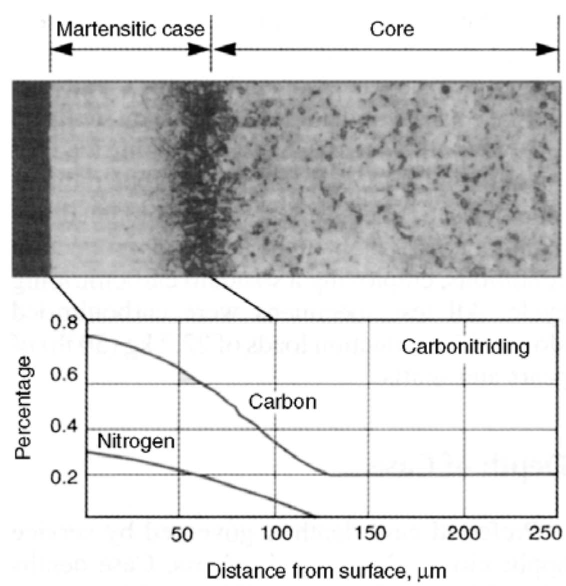

Fig. 1 Surface layers produced by carbonitriding of steel at 850 °C (1560 °F), where carbon predominates in the formation of a martensitic layer. Source: Ref 1

used on carbonitrided parts. The resistance to softening manifests itself in improved wear resistance. Carbonitrided gears, for example, often exhibit better wear properties than carburized gears.

Applicable Steels and Applications

Gear steels commonly carbonitrided include carbon steels (1018, 1022, 1117) and alloy steels in the 4100, 4300, 5100, and 8600 series, with carbon contents up to approximately 0.25%. Also, many alloy steels in these same series with a carbon range of 0.30 to 0.50% are carbonitrided to case depths up to approximately 0.3 mm (0.01 in.) when a combination of a reasonably tough, through-hardened core and a hard, long-wearing surface is required (for example, transmission gears). Steels such as 4140, 5130, 5140, 8640, and 4340 for applications such as heavy-duty gearing are treated by this method at 845 °C (1550 °F). High-density powder metallurgy steels are also often carbonitrided for improved wear resistance and toughness.

Applications. Carbonitriding is limited to shallower cases for fine pitch gearing used primarily in aerospace applications. Deep case depths require prohibitive time cycles. The carbonitrided case has better wear and temper resistance than a carburized case. For many applications, carbonitriding of low-alloy steels provides case properties equivalent to those obtained in gas carburized high-alloy steels. But the core often has low hardness. This is why the process is generally applied to gears of low-duty cycle.

Depth of Case

Preferred case depth is governed by service application and by core hardness. Case depths of 0.025 to 0.075 mm (0.001 to 0.003 in.) are commonly applied to thin parts that require wear resistance under light loads. Case depths up to 0.75 mm (0.030 in.) may be applied to parts for resisting high compressive loads. Case depths of 0.63 to 0.75 mm (0.025 to 0.030 in.) may be applied to gears that are subjected to high tensile or compressive stresses caused by torsional, bending, or contact loads.

Medium-carbon steels with core hardnesses of 40 to 45 HRC normally require less case depth than steels with core hardnesses of 20 HRC or below. Low-alloy steels with medium-carbon content, such as those used in automotive transmission gears, are often assigned minimum case depths of 0.2 mm (0.008 in.).

Measurements of the case depths of carbonitrided parts may refer to effective case depth or total case depth, as with reporting case depths for carburized parts. For very thin cases, usually only the total case depth is specified. In general, it is easy to distinguish case and core microstructures in a carbonitrided piece, particularly when the case is thin and is produced at a low carbonitriding temperature; more difficulty is encountered in distinguishing case and core when high temperatures, deep cases, and medium-carbon or high-carbon steels are involved. Whether or not the core has a martensitic structure is also a contributing factor in case-depth measurements.

Hardenability of Case

One major advantage of carbonitriding is that the nitrogen absorbed during processing lowers the critical cooling rate of the steel. That is, the hardenability of the case is significantly greater when nitrogen is added by carbonitriding than when the same steel is only carburized (Fig. 2). This permits the use of steels on which uniform case hardness ordinarily could not be obtained if they were only carburized and quenched. Where core properties are not important, carbonitriding permits the use of low-carbon steels, which cost less and may have better machinability or formability.

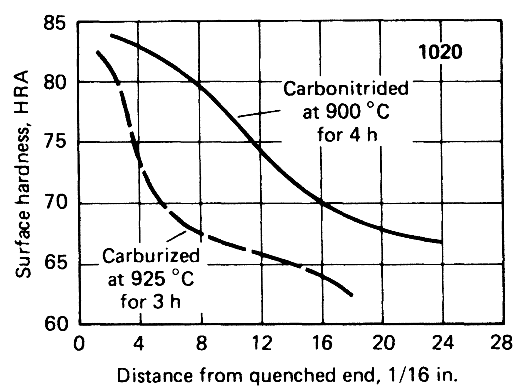

Fig. 2 End-quench hardenability curve for 1020 steel carbonitrided at 900 °C (1650 °F) compared with curve for the same steel carburized at 925 °C (1700 °F). Hardness was measured along the surface of the as-quenched hardenability specimen. Ammonia and methane contents of the inlet carbonitriding atmosphere were 5%; balance, carrier gas.

Table 1 Effect of tempering on Charpy V-notch impact strength of carbonitrided 1041 steel

Specimens were carbonitrided at 845 °C (1550 °F) for 3 h in an atmosphere containing 7% ammonia, and were oil quenched from the carbonitriding temperature. Specimens were copper plated before machining of V-notch to permit exposure of the notch to the carbonitriding atmosphere.

	Tempering temperature		Impact strength		Hardness, HRC(a) Distance below surface, mm (in.)								
Test	°C	°F	J	ft · lb	Surface(b)	Core	0.075 (0.003)	0.15 (0.006)	0.25 (0.01)	0.38 (0.015)	0.64 (0.025)	1.0 (0.04)	1.4 (0.055)
1	AQ	AQ	1.4	1	60	53	63	64	64	63	61	61	58
2	370	700	2, 2	1.5, 1.5	47	46	57	57	55	54	49	50	50
3	425	800	29, 29	21.5, 21.5	42.5	43	57	57	56	55	49	47	47
4	480	900	69, 60	51, 44	38	38	54	54	52	50	42	38	38
5(c)	480	900	47, 52	35, 38	...	...	...	...	...	...	...	...	...
6	540	1000	78, 81	57.5, 60	35	32	49	50	50	47	36	33	32

AQ: as-quenched. (a) Converted from Vickers hardness. (b) Surface hardness is less than hardness at 0.075 mm below the surface because of retained austenite. (c) Tested at −18 °C (0 °F); all other tests at room temperature

Quenching and Tempering

Quenching. Carbonitrided steels may be quenched in water, oil, or gas depending on the allowable distortion, metallurgical requirements (such as case and core hardness), and type of furnace. For the alloy gear steels discussed earlier, oil quenching is most commonly employed. Quenching oil temperatures may vary from approximately 40 to 105 °C (100 to 220 °F). Special high-flash-point oils may be used at the higher temperatures to minimize distortion; sometimes, molten salt is used for the same reason. In the normal range of oil temperature (approximately 50 to 70 °C, or 120 to 160 °F), a mineral oil with a minimum flash point of 170 °C (335 °F) and a viscosity of 21×10^{-6} m^2/s at 38 °C (21 centistokes, or 100 Say-bolt universal seconds, or SUS, at 100 °F) is commonly used. Special oils containing additives for increasing the quenching rate also may be used. To maintain maximum quenching effectiveness, quenching oils should have a low capacity for dissolving water. Quenching oils that dissolve even small amounts of water may lose effectiveness in 3 to 6 months; those that shed water completely may be used significantly longer.

Tempering. Because tempering a carbonitrided case at 425 °C (795 °F) and above results in a marked increase in notch toughness (see Table 1), parts that are to be subjected to repeated shock loading are invariably tempered to avoid impact and impact fatigue failures. Most carbonitrided gears are tempered at 190 to 205 °C (375 to 400°F) to reduce surface brittleness and yet maintain a minimum case hardness of 58 HRC. Alloy steel parts that are to be surface ground are tempered to minimize grinding cracks.

Distortion

Distortion in carbonitrided gears is far less than that of carburized gears because of lower process temperatures and shorter time cycles. Nevertheless, it is more than any nitriding process. But one major advantage of carbonitriding is that the hardenability of the case is significantly greater than carburizing or nitriding process. This permits in many cases to finish grind teeth (0.076 to 0.102 mm, or 0.003 to 0.004 in., stock removal) without any significant loss of hardness.

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This chapter was adapted from Carbonitriding, *Surface Hardening of Steels: Understanding the Basics,* J.R. Davis, Ed., ASM International, 2002, p 127–139

REFERENCE

1. D.H. Herring, Comparing Carbonitriding and Nitrocarburizing, *Heat Treat. Prog.,* April/May 2002, p 17–19

SELECTED REFERENCES

- R. Davies and C.G. Smith, A Practical Study of the Carbonitriding Process, *Met. Prog.,* Vol 114 (No. 4), 1978, p 40–53
- J. Dossett, Carbonitriding, *Heat Treating,* Vol 4, *ASM Handbook,* ASM International, 1991, p 376–386

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CHAPTER 12

Induction and Flame Hardening

INDUCTION AND FLAME HARDENING are methods of hardening the surfaces of components, usually in selected areas, by the short-time application of high-intensity heating followed by quenching. The heating and hardening effects are localized, and the depth of hardening is controllable. Unlike thermochemical case hardening treatments (carburizing, nitriding, and carbonitriding) applied to steels, induction and flame hardening do not promote chemical enrichment of the surface with carbon or nitrogen but rely on the presence of an adequate carbon content already in the material to achieve the hardness level required. The properties of the core remain unaffected and depend on material composition and prior heat treatment.

Induction and flame hardening have been used successfully on most gear types (spur, helical, bevel, herringbone, etc.). These processes are used when gear teeth require high hardness, but size or configuration does not lend itself to carburizing and quenching the entire part. Both processes may also be used when the maximum contact and bending strength achieved by carburizing is not required. Induction and flame hardening are also used in place of the more costly nitriding process which cannot economically generate some of the deeper cases required. The characteristics of induction and flame hardening methods are compared in Table 1.

Induction Hardening

In the induction hardening process, rapid heating is generated by electromagnetic induction when a high-frequency current is passed through a coil surrounding a gear. The depth to which the heated zone extends depends on the frequency of the current and on the duration of the heating cycle. Because of an electrical phenomenon called skin effect, the depth of the heated area is inversely proportional to the frequency used. It means the finer the pitch of gear tooth is, the higher the frequency of current needed will be. The time required to heat the surface layers to above the material transformation range is surprisingly brief, a matter of a few seconds. Selective heating and, therefore, hardening, is accomplished by suitable design of the coils or inductor blocks. At the end of the heating cycle, the steel usually is quenched by water jets passing through the inductor coils. Precise methods for controlling the operation, such as the rate of energy input, duration of heating, and rate of cooling, are thus necessary. These features are incorporated in induction hardening

Table 1 Comparison of flame- and induction-hardening processes

Characteristics	Flame	Induction
Equipment	Oxyfuel torch, special head quench system	Power supply, inductor, quench system
Applicable material	Ferrous alloys, carbon steels, alloy steels, cast irons	Same
Speed of heating	Few seconds to few minutes	1–10 s
Depth of hardening	1.2–6.2 mm (0.050–0.250 in.)	0.4–1.5 mm (0.015–0.060 in.); 0.1 mm (0.004 in.) for impulse
Processing	One part at a time	Same
Part size	No limit	Must fit in coil
Tempering	Required	Same
Can be automated	Yes	Yes
Operator skills	Significant skill required	Little skill required after setup
Control of process	Attention required	Very precise
Operator comfort	Hot, eye protection required	Can be done in suit
Cost		
Equipment	Low	High
Per piece	Best for large work	Best for small work

equipment, which usually is operated entirely automatically.

Induction hardening employs a wide variety of inductors ranging from coiled copper tubing to forms machined from solid copper combined with laminated materials to achieve the required induced electrical currents. Coarser pitch teeth (below 20 diametral pitch, DP) generally require inductors powered by medium-frequency motor generator sets or solid-state units. Finer-pitch gearing uses encircling coils with power provided by high-frequency vacuum tube units. Wide-faced gearing is heated by scanning-type equipment, while more limited areas can be heated by stationary inductors. Parts are rotated when encircling coils are used. Induction heating depth and pattern are controlled by frequency, power density, shape of the inductor, workpiece geometry, and workpiece area being heated.

Quenching after induction heating can be integral with the heat source by use of a separate following spray or by using an immersion quench tank. Oil, water, or polymer solutions can be used, in addition to air, depending on hardenability of the steel and hardening requirements.

Applicable Materials

A wide variety of materials can be induction hardened, including (cast and wrought) carbon and alloy steels, martensitic stainless steels, and ductile, malleable, and gray cast irons. Generally, steels with carbon content of approximately 0.35 to 0.50% are suitable for induction hardening. Alloy steels with more than 0.5% carbon are susceptible to cracking. The higher the alloy content with high carbon is, the greater the tendency to crack will be. Some of the common gear materials that offer acceptable case and core properties after induction hardening are AISI 1040, 1050, 4140, 4340, and 5150 steels.

Selection of the material condition (hot rolled, cold rolled) can affect the magnitude and repeatability of induction hardening. Hot-rolled materials exhibit more dimensional change and variation than cold drawn because of densification of material from cold working. A quench and tempered material condition before heat treatment, however, provides the best hardening response and most repeatable distortion.

Preheat Treatment

For more consistent results, it is recommended that coarser pitch gears of leaner alloy steels receive a quench and temper pretreatment, for example, 4140 steel with teeth coarser than 4 DP.

Both carbon and alloy steels with normalized or annealed structures can be induction hardened. These structures do, however, require longer heating cycles and a more severe quench, which increase the chance of cracking. The annealed structure alone is the least receptive to induction hardening.

Hardening Patterns

There are two basic methods of induction hardening gears: spin hardening and tooth-to-tooth, or contour hardening. Figure 1 shows variations of these processes and the resultant hardening patterns. In spin hardening that uses a circular inductor, the teeth are hardened from the tips downward. While such a pattern may be acceptable for splines and some gearing, heavily loaded gears need a hardness pattern that is more like a carburized case. This is achieved with contour hardening.

The spin or induction coil method is generally limited to gears of approximately 5 DP and finer. The maximum diameter and face width of gears capable of being hardened by this method are determined by the area of gear outside diameter and kW capacity of the equipment. Long slender parts can be induction hardened with lower kW capacity equipment by having coils scan the length of part while it is rotating in the coil.

Contour hardening can be applied to almost any tooth size with appropriate supporting equipment and kW capacity. However, for gears of approximately 16 DP and finer, this method does not produce satisfactory results. In

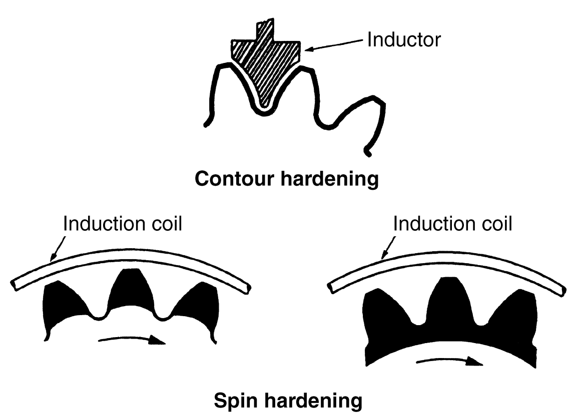


Fig. 1 Variations in hardening patterns obtainable on gear teeth by induction hardening

such cases, an induction coil method is recommended.

Heating with Induction

Accurate heating to the proper surface temperature is a critical step. Inductor design, heat input, and cycle time must be closely controlled. Underheating results in less than specified hardness and case depth. Overheating can result in cracking. For effective heating, recommended current frequencies used for different DP of gears are:

Diametral pitch (DP)	Frequency, kHz
20	500–1000
10	300–500
8	300–500
6	10–500
4	6–10
2	6–10

Quenching and Tempering

Quenching. Heat must be removed quickly and uniformly to obtain desired surface hardness. The quenchant should be such that it produces acceptable as-quenched hardness, yet minimize cracking. Quenchants used are water, soluble oil, polymer, oil, and air.

Parts heated in an induction coil usually are quenched in an integral quench ring (Fig. 2) or in an agitated quench media. Contour hardening is equipped with an integral quench following the inductor, or the gear may be submerged in a quench media.

Tempering is performed only when specified. However, judgment should be exercised before omitting tempering. It is a good practice to temper after quenching to increase toughness and reduce residual stress and crack susceptibility. Tempering should be done for sufficient time to ensure hardened teeth reach the specified tempering temperature.

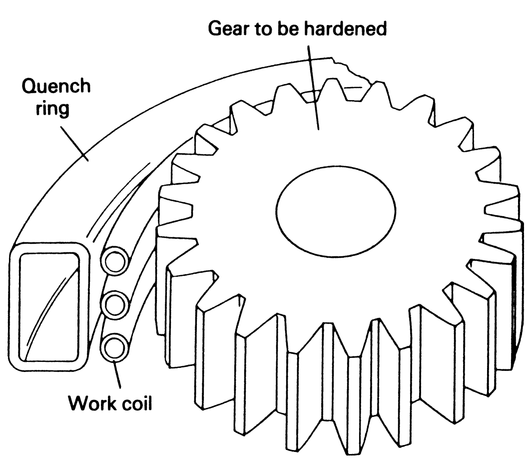


Fig. 2 Quench ring and solenoid coil surrounding a gear to be induction hardened.

Surface Hardness and Case Depth

Frequency and power density of electrical power and its time duration govern the depth of heating, which eventually controls the surface hardness and case depth that can be achieved after induction hardening. Surface hardness is primarily a function of carbon content. It also depends on alloy content, heating time, mass of the gear, and quenching considerations. Hardness achieved is generally between 53 and 55 HRC. The core hardness is developed by quenching and tempering prior to induction hardening. As already discussed, high frequency of current can control heat to a shallow depth on the tooth surfaces, whereas low frequencies produce greater depth of heat penetration. For example, a case depth of 0.254 mm (0.010 in.) can be produced with a frequency between 100 kHz and 1 MHz.

For larger case depths, frequencies between 3 and 25 kHz are used. Table 2 shows approximate case depths that are normally achieved with the induction hardening process.

In an induction-hardened tooth that requires high bending strength, it is necessary to get a reasonable hardness depth at the root fillet. Table 3 shows how much is needed for different size gear teeth.

Table 2 Current frequency versus case depth for induction hardening

Frequency, kHz	Approximate case depth, mm (in.)
3	3.81 (0.150)
10	1.52–2 (0.060–0.080)
500	0.51–1 (0.02–0.04)
1000	0.25–0.51 (0.01–0.02)

Table 3 Recommended depth of hardness in root region for induction-hardened teeth

Diametral pitch (DP)	Hardness depth, mm (in.)	Frequency, kHz
20	0.25–1 (0.010–0.040)	500–1000
16	0.38–1.52 (0.015–0.060)	500–1000
10	0.51–2.54 (0.020–0.100)	300–500
8	0.76–3.18 (0.030–0.125)	300–500
6	1.14–3.81 (0.045–0.150)	100–300
4	1.52–4.45 (0.060–0.175)	6–10

Depth reading taken at the center of root fillet.

Even though the correct hardness depth is obtained in the root region, it still is difficult to obtain high bending strength with induction-hardened gears made of any alloy steel. Therefore, a proper material selection is critical. Another drawback of induction hardening is residual tensile stress. With a clever development program, it is usually possible to work on an induction-hardening cycle so that the timing of the heating, the delay before quench, and the quenching are such that gears are free from dangerous residual tensile stresses.

Effective Case Depth. Effective case depth for induction-hardened gears normally is defined as the distance below the surface at the 0.5 tooth height where hardness drops 10 HRC points below the surface (Fig. 3). In case a tooth is through hardened (finer DP gears), effective case depth does not apply. When root is to be hardened, depth of case at the root may be separately specified.

Core Hardness of Tooth and Fatigue Strength. For every gear material, there is an optimum core hardness for maximum fatigue strength, provided the tooth is not through hardened. As the size of tooth decreases (higher DP), teeth tend to become through hardened. In case of induction-hardened teeth, through hardening takes place for tooth size above 10 DP, meaning allowable fatigue strength for induction-hardened gears above 10 DP needs to be reduced.

Induction Hardening Problems

It is quite difficult to obtain uniform case depth on a gear tooth with induction hardening. Typical case profiles of induction-hardened teeth are shown in Fig. 4. Furthermore, it also is difficult to obtain a reasonable depth of hardness at the center of root fillet. This results in lower bending strength compared with a carburized and hardened gear tooth. Furthermore, the maximum attainable surface hardness with induction hardening is about 55 HRC, which limits the durability of gears.

Another problem with an induction-hardened gear is residual stress in the case/core interface. In this region, there are high residual stresses due to drastic differences of the case and core microstructures and the fact that the transition occurs in a very short distance, as illustrated in Fig. 5. This figure also shows the differences in case hardness gradients for induction-hardened and carburized gear teeth. If the induction hardening process is not properly controlled, the case/core interface area could be susceptible to cracking.

In any case, considering all the merits and demerits, induction hardening is a viable gear

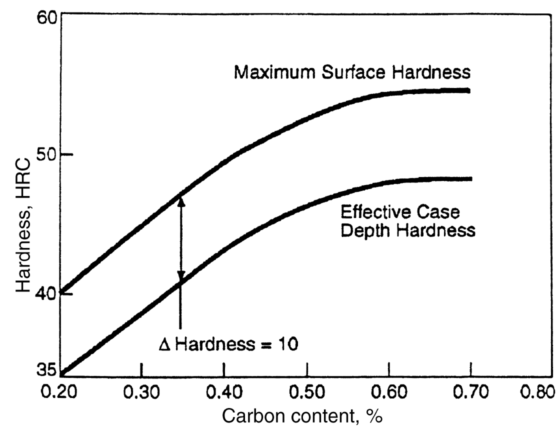


Fig. 3 Recommended maximum surface hardness and effective case depth hardness vs. carbon percent for induction-hardened gears

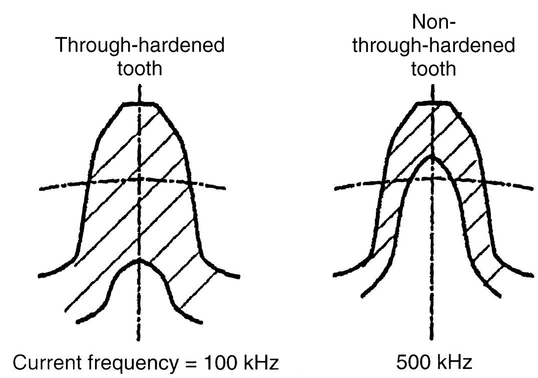


Fig. 4 Case depth profiles at different current frequency

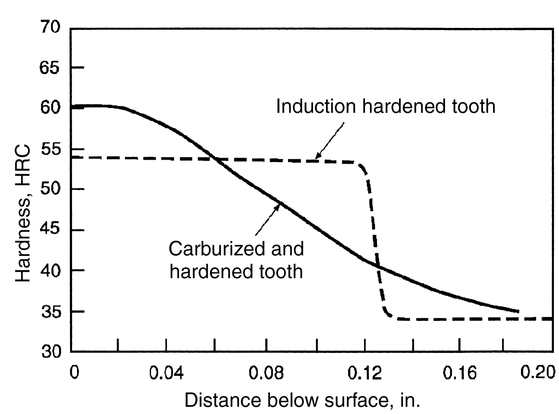


Fig. 5 Comparison of hardness gradients—induction-hardened and carburized gears

heat treat process. For successful use of the process, it is necessary to go through several developmental steps similar to other heat treat processes.

Heat Treat Distortion. As with other heat treat processes, there is some distortion of gears after induction hardening. Hot-rolled materials exhibit more dimensional change and variation than cold-drawn materials that are subjected to densification due to cold working. A quench and tempered material condition or pre-heat treatment, however, provides the best hardening response and most repeatable distortion. Table 4 shows comparative ratings of some common induction-hardening type materials. In general, after induction hardening, the quality level of gears does not go down by more than one AGMA quality level. Thus, in most applications, induction-hardened gears do not require any post-heat-treat finishing except for high-speed applications (pitch line velocity above 50.8 m/s, or 10,000 ft/min); mating gears sometimes are lapped together.

Dual-Frequency Process

The dual-frequency process uses two different frequencies for heating: high and low. At first, the gear is heated with a relatively low-frequency source (3–10 kHz), providing the energy required to preheat the mass of the gear teeth. This step is followed immediately by heating with a high-frequency source, which ranges from 100 to 270 kHz, depending on the DP of the gear. The high-frequency source rapidly heats the entire tooth surface to the hardening temperature. The gear is then quenched for the desired hardness.

The process is usually computer controlled. Because it is fast, surfaces remain clean and free from carbon depletion and scale, and the core material retains its original properties. The process puts only the necessary amount of heat into the part (much less than single-frequency induction); hence, case depth and gear quality level can be met precisely.

Residual (compressive) stress levels in a gear tooth induction hardened by the dual-frequency method are considerably higher than those in the single-frequency induction method. Figure 6 shows comparative stress levels between single-frequency induction, dual-frequency induction systems, and carburized and hardened gear tooth.

Distortion. Because the amount of heat applied by the dual-frequency process is considerably less than single frequency, heat treat distortion is significantly lower. In a test with gears (8 DP; 54 teeth; 29.21 mm, or 1.15 in., face width) made from AISI 5150, results showed distortion on any gear geometry dimension to be less than ±0.0102 mm (±0.0004 in.).

Materials. A variety of materials can be used. Materials that have been successfully contour hardened by the dual-frequency method include AISI 1050, 4140, 4340, 4150, and 5150.

Applications. Dual-frequency process is superior to regular induction hardening and is particularly useful for higher root hardness and close control of case depth.

Flame Hardening

Flame hardening involves the direct impingement of oxyfuel gas flames from suitably designed and positioned burners onto the surface area to be hardened, followed by quenching. The result is a hard surface layer of martensite over a softer interior core. There is no change in composition, and therefore, the flame-hardened steel must have adequate carbon content for the desired hardness. The rate of heating for this process is rapid, although not as fast as with induction heating.

Equipment. Flame-hardening equipment may be a single torch with a specially designed

Table 4 Distortion ratings of some induction-hardening materials

Material (AISI)	AMS specification	AMS quality	Hardness, HRC		Distortion rating(a)
			Case	Core	
1040	. . .	2300/2304	54	30	Good
1050	. . .	2300/2304	54	30	Good
4140	6382	2300/2304	55	40	Good
4340	6414	2300/2304	55	40	Poor
5140	. . .	2300/2304	55	40	Poor

Note: Alloy steels with 0.5% carbon and above are susceptible to cracking. (a) Compared with carburizing and hardening

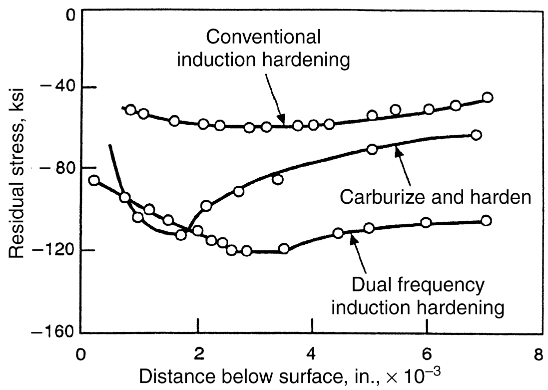


Fig. 6 Residual compressive stress distribution—induction hardening and carburizing

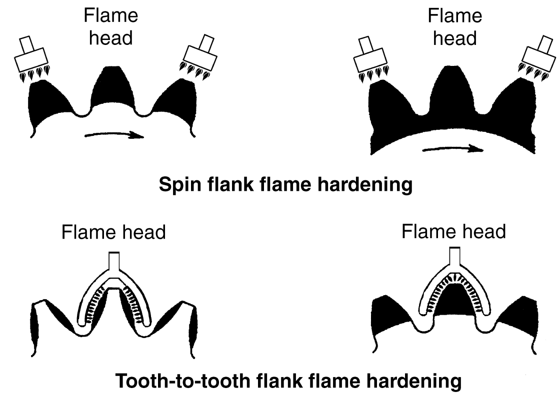


Fig. 7 Variations in hardening patterns obtainable on gear teeth flanks by flame hardening

head or an elaborate apparatus that automatically indexes, heats, and quenches parts. Gears that are so large that conventional furnace treatments are not practical or economical are flame hardened. The quality requirement for such gears is generally below AGMA class 7. It should be noted that high operator skill is necessary when flame hardening is carried out manually using a single torch.

Table 5 Response of steels to flame hardening

	Typical hardness, HRC, as affected by quenchant	
Material	**Air(a)**	**Water(b)**
Plain carbon steels		
1025–1035	. . .	33–50
1040–1050	. . .	55–60
1055–1075	50–60	60–63
1080–1095	55–62	62–65
1125–1137	. . .	45–55
1138–1144	45–55	55–62
1146–1151	50–55	58–64
Carburized grades of plain carbon steels(c)		
1010–1020	50–60	62–65
1108–1120	50–60	62–65
Alloy steels		
1340–1345	45–55	55–62
3140–3145	50–60	60–64
3350	55–60	63–65
4063	55–60	63–65
4130–4135	. . .	55–60
4140–4145	52–56	55–60
4147–4150	58–62	62–65
4337–4340	53–57	60–63
4347	56–60	62–65
4640	52–56	60–63
52100	55–60	62–64
6150	. . .	55–60
8630–8640	48–53	58–62
8642–8660	55–63	62–64
Carburized grades of alloy steels(c)		
3310	55–60	63–65
4615–4620	58–62	64–66
8615–8620	. . .	62–65

(a) To obtain the hardness results indicated, those areas not directly heated must be kept relatively cool during the heating process. (b) Thin sections are susceptible to cracking when quenched with water. (c) Hardness values of carburized cases containing 0.90 to 1.10% C

Gases. Three basic gases are used for flame hardening: acetylene, MAPP (methylacetylene propadiene), and propane. These gases are each mixed with air in particular ratios and are burned under pressure to generate the flame which the burner directs on the workpiece.

Applicable Materials. Any type of hardenable steel can be flame hardened. For best results, the carbon content should be at least 0.35%, the usual range being 0.40 to 0.50% as shown in Table 5.

Quenching. Plain carbon steels usually are quenched by a water spray, whereas the rate of cooling of alloy steels may be varied from a rapid water quench to a slow air cool, depending on the composition of the steel. Typical hardnesses obtained by quenching in air or water are given in Table 5.

Applications. The general application of flame hardening is to the tooth flanks only. This can be accomplished by spin flame hardening or tooth-to-tooth flame hardening (Fig. 7). Induction contour hardening is preferred over flame hardening when root hardness and closer control of case depth is required. Although the spin flame hardening process is capable of hardening the whole tooth and also below the root, reduced core ductility and increased distortion result.

ACKNOWLEDGMENT

Portions of this chapter were adapted from A.K. Rakhit, Induction Hardening Gears, *Heat*

Treatment of Gears: A Practical Guide for Engineers, ASM International, 2000, p 175–184

SELECTED REFERENCES

- R.E. Haimbaugh, *Practical Induction Heat Treating,* ASM International, 2001
- P.A. Hassell and N.V. Ross, Induction Heat Treating of Steel, *Heat Treating,* Vol 4, *ASM Handbook,* ASM International, 1991, p 164–202
- T. Ruglic, Flame Hardening, *Heat Treating,* Vol 4, *ASM Handbook,* ASM International, 1991, p 268–285
- S.L. Semiatin and D.E. Stutz, *Induction Heat Treatment of Steel,* American Society for Metals, 1986
- S. Zinn and S.L. Semiatin, *Elements of Induction Heating: Design, Control, and Applications,* ASM International, 1988

Gear Materials, Properties, and Manufacture
J.R. Davis, editor, p257-291
DOI:10.1361/gmpm2005p257

CHAPTER 13

Gear Failure Modes and Analysis

GEARS can fail in many different ways, and except for an increase in noise level and vibration, there is often no indication of difficulty until total failure occurs. In general, each type of failures leaves characteristic clues on gear teeth, and detailed examination often yields enough information to establish the cause of failure. Despite the variety of ways in which gears fail, service failures of gears are relatively rare (Ref 1).

Classification of Gear Failure Modes

Several failure analysists have classified gear failure modes in different ways. As shown in Table 1, the American Gear Manufacturers Association (AGMA) has classified 36 modes of gear failure under the broad categories of wear, scuffing, plastic deformation, contact fatigue, cracking, fracture, and bending fatigue. Each of these failure modes are described in the AGMA gear failure nomenclature publication (Ref 2).

Errichello (Ref 3) has classified gear failures into two broad groups:

- Nonlubrication-related failures
- Lubrication-related failures

As shown in Table 2 in Chapter 2, "Gear Tribology and Lubrication," nonlubricated-related failures include both overload and bending fatigue types of failure. Lubricated-related failures include Hertzian fatigue (pitting), wear, and scuffing.

In Ref 4, gear failure modes were broken down into two groups:

- Failure modes on gear tooth flanks, including pitting, scuffing, and wear
- Failure modes on gear root fillets, including bending fatigue and impact

Alban (Ref 1) has classified gear failures into four groups: fatigue, impact, wear, and stress rupture. Table 2 lists these failure modes in order of decreasing frequency (the primary cause of failure is fatigue, the second cause is impact, and so on). In a composite analysis of more than 1500 studies, the three most common failure modes, which together account for more than half the failures studied, are tooth-bending fatigue, tooth-bending impact, and abrasive tooth wear (Ref 1).

Fatigue Failures

Fatigue failure results from cracking under repeated stresses much lower then the ultimate tensile strength. This type of failure:

- Ordinarily depends upon the number of repetitions of a given stress range rather than the total time under load
- Does not occur below some stress amplitude called the fatigue limit
- Is greatly encouraged by notches, grooves, surface discontinuities, and subsurface imperfections that will decrease the stress amplitude that can be withstood for a fixed number of stress cycles
- Is increased significantly by increasing the average tensile stress of the loading cycle

There are three stages within a fatigue failure that must be studied closely: the origin of the fracture, the progression under successive cycles of loading, and final rupture of the part when the spreading crack has sufficiently weakened the section. Most attention is devoted to the first stage to answer the question, Why did it start at this point? The second stage is observed to determine the direction of the progression. It appears that a fatigue crack will follow the path of least resistance through the metal. The final fracturing may be by shear or tension, but in either case, the size of the final-fracture area can be used to calculate the apparent magnitude of stress that had been applied to the part.

Bending Fatigue

The origins of bending fatigue failures typically are imperfections in the surface of the root fillet (e.g., tooling "witness" marks) or nonmetallic inclusions near the surface. Cracks slowly propagate around the origin until they reach the critical size for the case material at the prevailing stress level. For hardened high-carbon material typically used for gears, when the crack reaches this critical size, it "pops" through the case (i.e., the entire case fractures). At this point, the rigidity of the tooth is reduced (compliance increases), and dynamic load increases significantly. This produces a readily detectable increase in noise and vibration and represents failure.

As stated earlier, bending fatigue is the most common mode of fatigue failure in gearing. There are many variations of this mode of failure, but the variations can be better understood by first presenting the classical failure.

The classical bending fatigue failure is that which occurs and progresses in the area designed to receive the maximum bending stress. Figure 1 shows such an example. There are five conditions that make Fig. 1 a perfect tooth-bending fatigue failure:

- The origin is at the surface of the root radius of the loaded (concave) side of the tooth

Table 1 Gear failure nomenclature recommended by the American Gear Manufacturers Association (AGMA)

Category	Failure mode
Wear	Adhesion
	Mild
	Moderate
	Severe
	Abrasion
	Mild
	Moderate
	Severe
	Polishing
	Corrosion
	Fretting corrosion
	Scaling
	Cavitation
	Erosion
	Electrical discharge
	Rippling(a)
Scuffing	Mild scuffing
	Moderate scuffing
	Severe scuffing
Plastic deformation	Indentation
	Cold flow
	Hot flow
	Rolling
	Tooth hammer
	Rippling(a)
	Ridging
	Burr
	Root fillet yielding
	Tip-to-root interference
Contact fatigue	Pitting (macropitting)
	Initial
	Progressive
	Flake
	Spall
	Micropitting
	Subcase fatigue
Cracking	Hardening cracks
	Grinding cracks
	Rim and web cracks
	Case/core separation
	Fatigue cracks
Fracture	Brittle fracture
	Ductile fracture
	Mixed mode fracture
	Tooth shear
	Fracture after plastic deformation
Bending fatigue	Low-cycle fatigue
	High-cycle fatigue
	Root fillet cracks
	Profile cracks
	Tooth end cracks

(a) Rippling is generally associated with plastic deformation, but it may also be a form of wear. Source: Ref 2

Fig. 1 Spiral bevel pinion showing classic tooth-bending fatigue. The origin is at midlength of the root radius on the concave (loaded) side. Original magnification at 0.4×

Table 2 Gear failure modes proposed by Alban in Ref 1

Failure mode	Type of failure
Fatigue	Tooth bending, surface contact (pitting or spalling), rolling contact, thermal fatigue
Impact	Tooth bending, tooth shear, tooth chipping, case crushing, torsional shear
Wear	Abrasive, adhesive
Stress rupture	Internal, external

- The origin is at the midpoint between the ends of the tooth where the normal load is expected to be
- One tooth failed first, and the fracture progressed slowly toward the zero-stress point at the root, which shifted during the progression to a point under the opposite root radius and then proceeded outward to that radius
- As the fracture progressed, the tooth deflected at each cycle until the load was picked up simultaneously by the top corner of the next tooth, which (because it was now overloaded) soon started a tooth-bending fatigue failure in the same area. The fracture of the second tooth appears to be of more recent origin than the first fracture
- The material and metallurgical characteristics are within specifications

If any of the above five conditions is changed, there will be a variation of bending fatigue away from the classical example

The rudiments of the classical bending fatigue can be studied using photoelastic models. Once the crack at the root radius is initiated and is progressing toward the zero-stress point (Fig. 2), the point moves away from the leading edge of the crack and shifts laterally until it reaches a position under the opposite root. At that time, the shortest remaining distance is outward to the root, and the point terminates there. In the meantime, as the failing tooth is being deflected, the top of the adjacent tooth has picked up the load. The load on the first tooth has now been relieved, allowing a slower progression due to lower cyclic stress. It is now only a matter of time until the next tooth initiates a crack in the same position. Variations from the classical bending fatigue appear for many reasons.

Bending fatigue of a spur-gear tooth with the origin along the surface of the root radius of the loaded side but at a point one-third the distance from the open end (Fig. 3) can be considered a classical failure in that the maximum crown of the teeth had been placed intentionally at the area where the failure occurred; therefore, the maximum applied loads were at the same area. This, then, is the area that was designed for the point of initiation of normal bending fatigue. The progression was directly to the bore, which was the shortest direction of least resistance.

Bending fatigue of a spur-gear tooth meeting all but one of the conditions for a classical failure is shown in Fig. 4. The origin was at one end of the tooth and not at the midlength. There could have been several reasons for this point of initiation; in this instance, a severe shock load had twisted the parts until a momentary overload was applied at the end of the tooth, causing a crack to form. The crack became a high stress-concentration point from which fatigue progression continued.

An example of bending fatigue that did not conform to classical conditions is shown in Fig. 5. Crack initiation was at a nonmetallic inclusion at the case/core interface; it did not originate at the surface. Subsequent progression was

Fig. 2 Photoelastic study of mating teeth indicates the shift of the zero-stress point during crack propagation until final fracture reaches the opposite root radius. At the same time, deflection of the tooth allowed the adjacent tooth to pick up the load

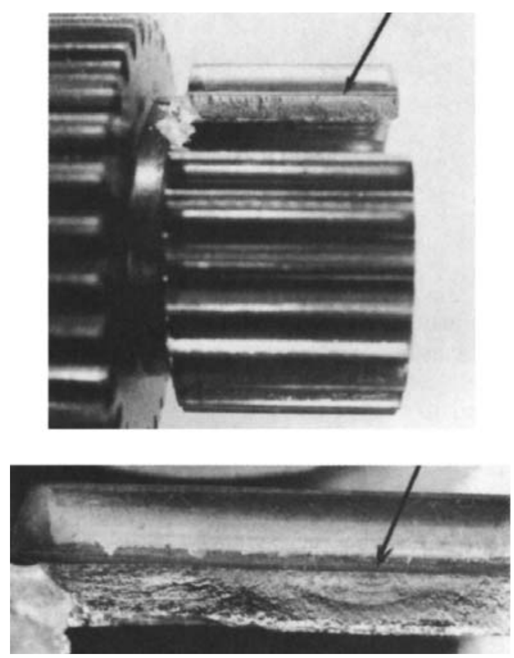


Fig. 3 Spur tooth pinion at 0.5× (top) and 1.5× (bottom). Tooth-bending fatigue originating at the root radius (arrows), loaded side, one-third the distance from the open end. Progression was to the bore

through the case to the surface as well as through the core toward the zero-stress point.

Another example of bending fatigue that did not conform is shown in Fig. 6. The failure was at the root radius and at the midlength, but the origin was not at the surface. The origin was at the apex of a tapped bolt hole that had been drilled from the back face of the gear and had terminated less than 6.4 mm (0.25 in.) from the root radius. Subsequent progression of the fatigue crack was to the surface at the root radius as well as into the core toward the zero-stress point.

Fig. 4 Spur pinion failure. Tooth-bending fatigue with origin at root radius of loaded side at one end of the tooth. Original magnification at 0.6×

Additional examples of gear failures due to bending fatigue fracture can be found in Fig. 7 to 10.

Bending Fatigue Properties. Detailed information on the bending fatigue characteristics of gear steels can be found in Chapter 3, "Ferrous and Nonferrous Alloys," and Chapter 9, "Carburizing."

Contact Fatigue

Contact fatigue is the cracking of a surface subjected to alternating Hertzian stresses produced under controlled rolling and sliding loading conditions (Ref 5). In addition to cracking, contact fatigue can result in microstructural alterations, including changes in retained austenite, residual stress, and martensite morphology.

When sliding is imposed on rolling, the tangential forces and thermal gradient caused by friction alter the magnitude and distribution of stresses in and below the contact area. The alternating shear stress increases in magnitude and is moved nearer to the surface by sliding forces. Thus, initiation of contact-fatigue cracks in gear teeth, which are subjected to significant amounts of sliding adjacent to the pitchline, is found to be in the surface material. These cracks propagate at a shallow angle to the surface, and pits result when the cracks are connected to the surface by secondary cracks. If pitting is severe, the bending strength of the tooth may be decreased to the point at which fracturing can occur.

The existence of several distinctly different modes of contact-fatigue damage has been recognized, but the factors that control the nucleation and propagation of each type are only partially understood. The contact fatigue of hardened steel has been the subject of exten-

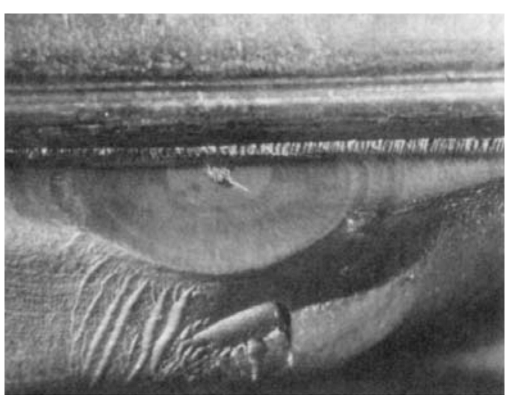


Fig. 5 Spur pinion failure. Tooth-bending fatigue is at midlength of the tooth at the root radius, but the origin is at an inclusion located in the case/core transition. Original magnification at 55×

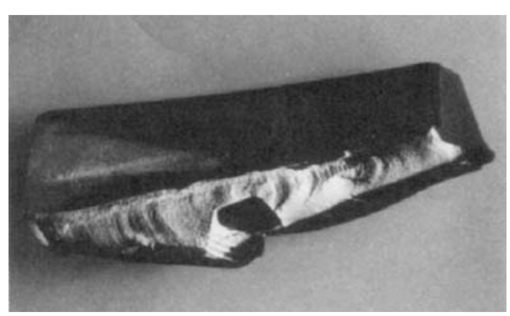


Fig. 6 Spiral bevel gear tooth failure. Tooth-bending fatigue with origin at the apex of the drilled bolt hole, which terminated just below the root radius. Original magnification at 0.5×

sive research using analytical and experimental methods, and publications have revealed a large variation in the contact fatigue strength of similar materials when tested under different conditions of contact geometry, speed, lubrication, temperature, and sliding in combination with rolling. These variations result from a poor understanding of the basic mechanism of contact fatigue. Although the cause of the fatigue and the locations of the origins are somewhat different, these are true fatigue fractures that are caused by shear-stress origins due to contact stresses between two metal surfaces.

Table 3 lists terminology used by the gear industry to describe the fracture appearance of contact-fatigue failed components. The following sections describe the types of damage from contact fatigue based in the categories in Table 3 as previously published in Ref 5.

The macro- and micropitting failures listed in Table 3 are divided into the following three categories based on the location of the origin:

- *Surface-origin pitting:* This is the type of pitting characteristic of combined rolling and sliding, such as occurs on gear teeth slightly away from the exact pitch line, where only rolling is present. The maximum shear stress is no longer buried within the metal, but is now brought to the surface because of the friction at the interface. Because the character of the surface is intimately involved, the lubricant becomes critical. It is for this reason that transmissions, final drives, and various other types of gear systems require lubricants with special additives. Everything possible must be done to reduce surface friction, expand the contact area, and reduce the load in order to reduce the contact stress and provide a hard, fatigue-resistant structure in the metal. Unfortunately, if those things are already near optimum, little can be done to improve the fatigue life. Only minimal improvements, if any, can be expected. Improvements can best be obtained, if at all, by changes in the lubricant.
- *Subsurface-origin pitting* is the result of fatigue below the surface in the region of highest shear stress. Subsurface origin pitting typically occurs only in cases where the elastohydrodynamic (EHD) lubricant film thickness is great enough to prevent significant asperity interaction. This pitting occurs in high-speed power gearing and in aerospace power drives. It is also typical in antifriction bearings (which operate with pure rolling as at the pitch diameter on spur gears). The origin is typically a nonmetallic inclusion or discontinuity in the structure, in the area of highest shear stress below the surface. Crack propagation is initially quite slow, until a crack penetrates the surface, allowing lubricant to enter. It can be difficult

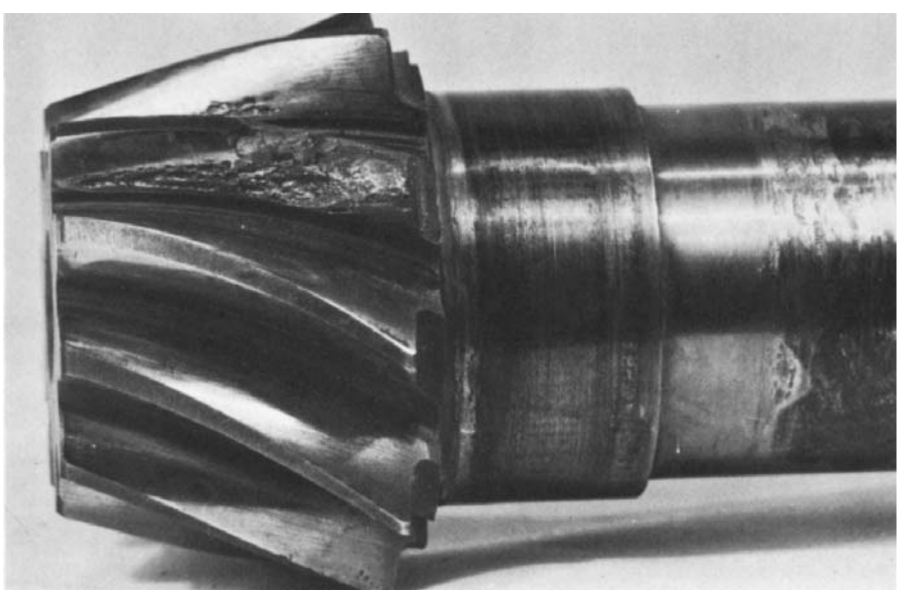


Fig. 7 Bending-fatigue fracture in the heel of one tooth of a spiral bevel pinion of AISI 8617 steel, carburized and hardened to 57 HRC in the case. Fracture resulted from severe pitting. Note that pitting had begun in an adjoining tooth (near top). Original magnification at 0.75×

to tell the difference between surface and subsurface origin pits, although subsurface origin pits often do not have the arrow shape typical of surface origin pits. As with surface origin pits, when the pit becomes large enough to cause a readily detectable increase in noise and vibration, it is considered a failure. Because most pits start small and progressively grow large, some critical high-speed drives incorporate a chip detector in the lubrication system to provide advance warning of imminent failure.

- *Subcase fatigue:* The previous types of contact-stress fatigue result in small, often tiny, pits in the contacting surfaces. Subcase-origin fatigue is different because very large cavities may be formed, apparently in a very short time. The fatigue originates near the case/core interface where the shear stresses. Because the basic cause is inadequate metal strength in the subcase region, the logical solution should be to strengthen the metal there. This can be done by increasing the case depth and/or the core hardness. This must be done carefully, however, because the changes could lead to through-hardening and brittle fracture, particularly on gear teeth.

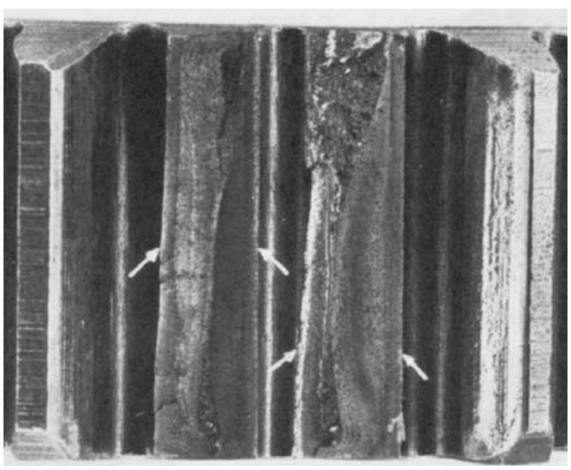


Fig. 8 Bending-fatigue fracture in two teeth of a reverse idler gear of AISI 8617 steel, carburized and hardened to 60 HRC in the case. Arrows point to the root fillets on both sides of each tooth, where fracture began due to excessive stress in these locations. Original magnification at ~2×

Macropitting

Macropitting is a general term that includes spalling and other forms of macroscale damage (Table 3) caused by Hertzian contact fatigue. Macropitting of gear teeth is generally due to contact fatigue, which occurs from localized plastic deformation, crack initiation, and finally macropitting from crack propagation in and near the contact surface.

Macropitting results from the subsurface growth of fatigue cracks, which may have a surface or subsurface origin. Figure 11 shows macropit formation by subsurface growth of surface-origin pitting on a gear tooth. The bottom of inclusion-origin macropits form first, and so forth. When circumstances cause surface-origin pitting, crack growth often occurs within and beyond the range of maximum shear stresses in the contact stress field (so-called *hydraulic-pressure propagation crack paths* that may be due to chemical reactions at the crack tip as well as to lubricant viscosity effects).

When fully developed, the craters exist at a depth comparable to that of the maximum alternating Hertzian shear stress. A macropit does not result from the gradual enlargement of a small cavity, but rather it results from the subsurface growth of a fatigue crack, which eventually separates from the main body of material. In the example of a gear damaged by crater formation due to surface-origin macropitting (Fig. 11), the craters are generally steep walled and essentially flat bottomed, as the walls and bot-

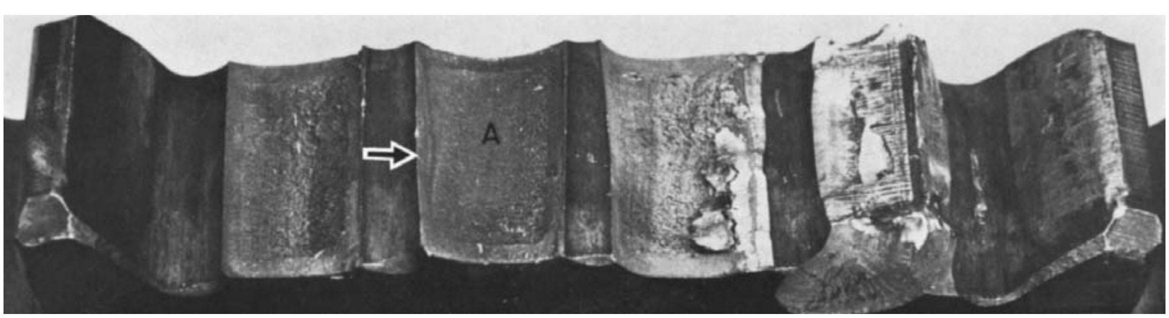


Fig. 9 Bending-fatigue fractures in several teeth of a spur gear of AISI 8620 steel, carburized and hardened to 60 HRC in the case. The tooth marked A apparently broke first, as the result of a fatigue crack that originated in the fillet to the left of the tooth (arrow). After this tooth broke off, fracturing of the teeth on each side of it was accelerated. Original magnification at 2×

tom are formed by fracture surfaces. The bottoms of macropits usually form first, and then the crack extends through the contact surface, creating the walls of the crater. However, there are circumstances in which macropits are initiated at or near the contact surface and the crack propagates through the Hertzian shear stress field, forming the walls and then the bottom of the crater (Ref 5).

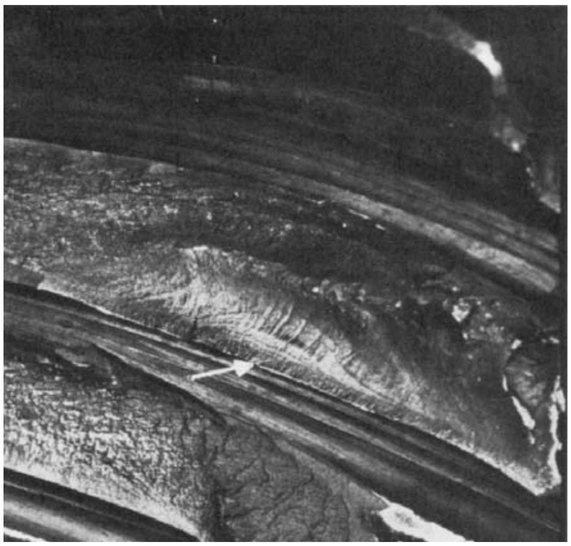


Fig. 10 Surface of a bending-fatigue fracture in a tooth (upper tooth in this view) of a large spiral bevel pinion of AISI 8620 steel carburized and hardened to 60 HRC at the surface. The arrow marks the fatigue-crack origin, in the root fillet. The absence of this tooth resulted in fracture of the tooth below by overload. Original magnification at 1.65×

Table 3 List of contact fatigue terminology used to describe the different fatigue mechanisms

Category	Fatigue mechanisms
Macroscale	
Macropitting	Pitting Initial pitting Destructive pitting Flaking Spalling Scabbing Shelling Fatigue wear
Subcase fatigue	Case crushing
Microscale	
Micropitting	Microspalling Frosting Glazing Gray staining Surface distress Peeling

Source: Ref 5

Macropitting fatigue life is inherently statistical because defect severity and location are randomly distributed among macroscopically identical contact components. Two major classes of macropitting damage are distinguished according to the location of the initiating defect: subsurface- and surface-origin macropitting.

Subsurface-origin macropitting results from defects in the bulk material subjected to the Hertzian cyclic stress field. Subsurface-origin macropits are characterized by a smooth area parallel to the contact surface with a steep wall exit (inclined by more than 45° with respect to the contact surface). Sometimes the subsurface defect that caused the damage remains visible after the macropit forms. The defects that cause subsurface-origin macropits are inclusions and/or material microstructure alterations. The defects that most likely produce this type of macropit are located above the depth of maximum alternating Hertzian shear stress. Variables governing the life of a component with respect to this type of macropit are Hertzian shear stress level, material matrix fatigue resistance, defect severity, and defect location.

Inclusion-origin fatigue is normally initiated at nonmetallic inclusions below the contact surface. The initiation and propagation of fatigue cracks are the result of cyclic stresses that are locally intensified by the shape, size, and distribution of nonmetallic inclusions in steels. Oxide inclusions from deoxidation, reoxidation, or refractory sources are the most frequently observed origins of inclusion-origin fatigue damage. Sulfide inclusions alone are rarely associated with contact fatigue cracks. The ductility of sulfide inclusions at hot working temperatures

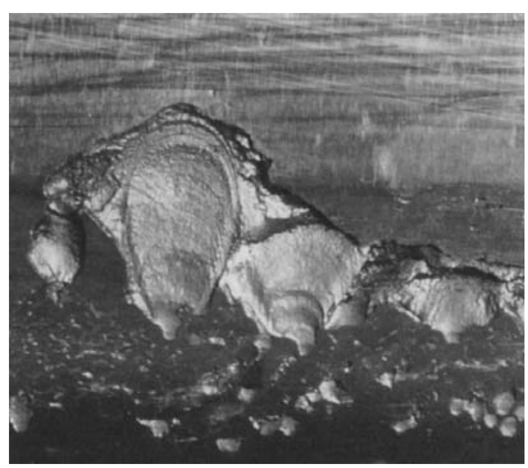


Fig. 11 Fractograph showing the advanced stages of macropitting fatigue for a helical pinion tooth

and a relatively low interfacial energy make them less effective stress concentrators than the harder, incoherent oxide inclusions. In addition, oxide inclusions are often present as stringers, or elongated aggregates of particles, which provide a much greater statistical probability for a point of stress concentration to be in an area of high contact stress.

When a nonmetallic inclusion as described above results in significant stress concentration, two situations are possible: (a) localized plastic flow occurs; and/or (b) a fatigue crack begins to propagate with no plastic flow as observed microscopically. If localized plastic deformation occurs, microstructural alterations begin to develop and the effective stress concentration of the inclusion is reduced. The probability of fatigue crack initiation/propagation is less than before the plastic flow occurred, so a macropit may not form. However, if a fatigue crack forms due to the inclusion, prior to any plastic deformation, the stress concentration is increased and crack propagation results in macropitting fatigue. This hypothesis includes the possibility of a fatigue crack originating from an inclusion that is associated with some sort of microstructural alteration, but also proposes that microstructural alterations are not a necessary step in the initiation/propagation of inclusion origin fatigue.

Early macroscopic propagation of inclusion-origin fatigue takes on a characteristic pattern. Many branching subsurface cracks are formed around the inclusion, with propagation often being most rapid perpendicular to the rolling direction, with a resulting elliptical shape for the incipient macropit. In some instances, the cracked material around the inclusion macropits early and further propagation is relatively slow, leaving a small macropit with very few subsurface cracks to cause further damage. However, when material has macropitted from the contact surface, the result may be debris bruising and denting of other contact areas with subsequent surface-origin macropitting.

Surface-origin macropitting is caused by defects in the immediate subsurface material subject to asperity scale cyclic stress fields and aggravated by surface tractive forces. These defects are either preexisting defects, such as nicks, dents, grinding furrows, surface discontinuities, and so forth, or microscale pits, which are described subsequently. The distinguishing features of a surface-origin macropit are in the entrance zone of the macropit. The entrance zone may exhibit a shallow-angle entry wall (inclined less than 30° to the contact surface), an arrow-head configuration (Fig. 12), the presence of a visible surface defect, or an association with a stress concentration due to design geometry. Surface-origin macropits have been classified by the nature of the defect: geometric stress concentration (GSC) and point surface origin (PSO).

Geometric stress concentration macropits are distributed on the contact surface at the ends of line contact. When the contact geometry, deflection under applied loads, and alignment cause the contact stress to be higher at the end of line contact, fatigue occurs within a narrow band in which the contact stresses are more severe than those associated with inclusions. The GSC macropits can propagate more rapidly in either direction—parallel or transverse to the rolling direction. This mode of contact fatigue is more frequently observed in life tests accelerated due to overload conditions because contact geometry is designed to produce uniform loading under expected service loads.

Crowning of gear teeth in line contact situations has been shown to be an effective method of prolonging life or preventing the GSC mode of macropitting. The fact that GSC macropits can occur simultaneously at both ends of a line contact indicates that this mode of contact fatigue is truly the result of end stress concentrations and not simply misalignment. However, misalignment can aggravate the situation by increasing end contact stresses. Early stages of

Fig. 12 Gear with arrow-shaped surface-origin pit. Source: Ref 4

GSC macropitting exhibit cracks extending to the surface, but it is not clear whether the first cracks always originate at the surface or slightly subsurface. However, the origins are rarely associated with a nonmetallic inclusion, but occasionally an inclusion at the end of line contact will serve as the nucleus for a GSC fatigue macropit.

Point surface origin macropits occur randomly distributed on the contact surface, similar to inclusion-type macropits. However, PSO macropits are different in two important aspects: first, there is no consistent association with nonmetallic inclusions; second, the origin is located at or near the contact surface, whereas with inclusion-origin macropits, the initiation site is located subsurface. These characteristics result in a distinct arrowhead appearance, in which the tip of the arrowhead is the origin and the advanced stages of the macropit develop in a fanned-out appearance with respect to the tip in the rolling direction.

Micropitting

Micropitting is damage of rolling/sliding contact metal surfaces by the formation of a glazed (burnished) surface, asperity scale microcracks, and asperity scale micropit craters. Micropitting damage is a gradual type of surface fatigue damage that is a complex function of surface topography and its interaction with the lubricant. (Even under high contact loads, a thin film of lubricant between the contacting surfaces from EHD lubrication is present.) Cumulative plastic deformation first results in a surface with a mirrorlike reflection known as a glazed surface. The surface and immediate subsurface material of a glazed surface are heavily deformed and with continued cyclic stressing, the ductility of the material decreases and microcracks form. The cracks tend to propagate parallel to the contact surface at depths comparable to that of asperity scale shear stresses. A microcracked surface is not usually distinguishable from a glazed surface by the unaided eye, but with microscopy, microcracks opening to the surface may be visible in glazed areas. As the microcracks multiply and propagate, the surface becomes undermined (in asperity dimensions), and multiple microscopic pits form. A micropitted surface appears frosted to the unaided eye, with black spots representing the micropits. An example is shown in Fig. 13.

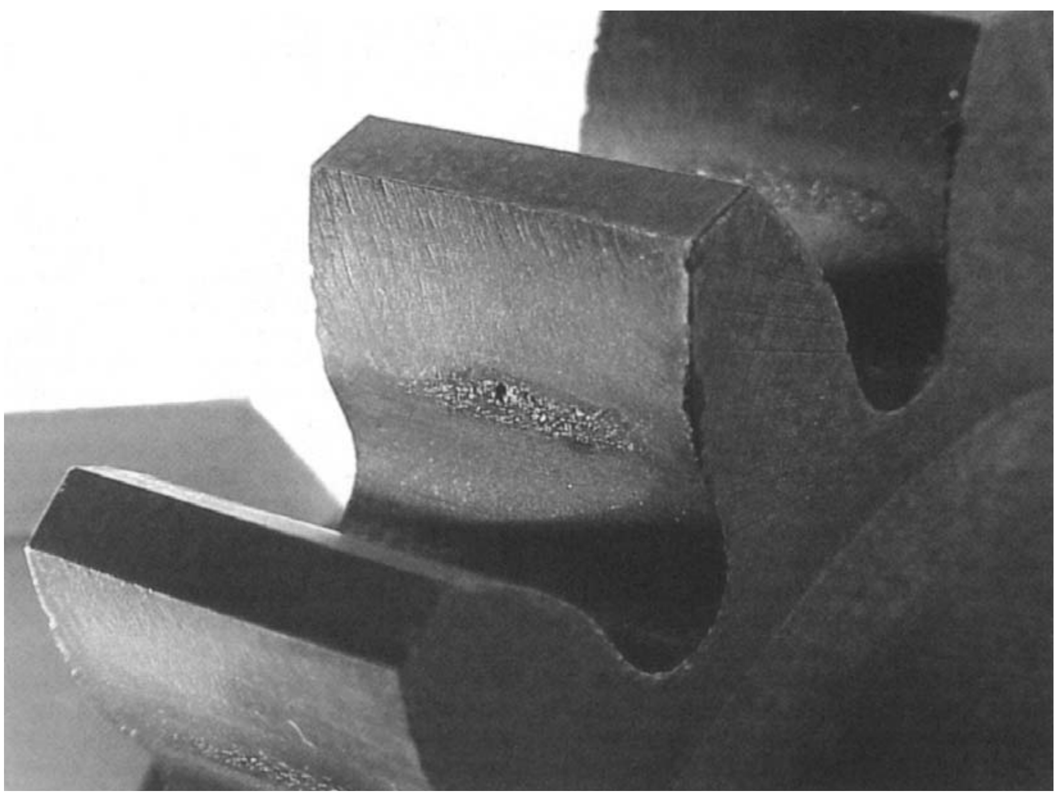


Fig. 13 Gear with micropitting (frosting) failure. Source: Ref 4

If extensive, the micropitting is considered a failure.

The occurrence of micropitting is controlled by the EHD film-thickness-to-roughness ratio and surface microgeometry. In the presence of a sufficiently high EHD film-thickness-to-surface-roughness height ratio, micropitting does not occur because the film prevents high-contact microstress in asperity interactions. However, film thickness may be sufficient to prevent generalized micropitting, but not local micropitting in the vicinity of surface defects such as nicks and dents that locally depressurize and thereby thin the EHD film. Tribology features that effect micropitting are both the height and sharpness (root-mean square height and slope) of the asperities. There is also some evidence that the normal wear in boundary-lubricated gear contacts removes incipient surface-origin pitting, which is one reason gears may be less prone to macropitting from surface-origin macropitting under some circumstances.

Micropitting is characterized by the limited depth of cracking and propagation over areas rather than propagation in depth. Micropitting fatigue cracks usually propagate parallel to the surface at depths from 2.5 to 13 mm (0.098 to 0.511 in.) and rarely as deep as 25 mm (0.984 in.). The high-stress gradient that exists in the vicinity of asperities explains why such cracks do not propagate to greater surface depths of the macro-Hertzian contact stress field. In essence, micropitting removes the asperity ridges and relieves the high-stress gradient. An additional visual characteristic is that the micropitting occurs more or less in the rolling direction, following finishing lines.

Macropitting fatigue is the most hazardous consequence of micropitting. The principal operating factor influencing the rate of macropitting from micropits is tractive interfacial stress. When friction is high, macropits form rapidly from the micropits on the surface. If early macropitting does not terminate the operation of a component that has suffered micropitting, then surface material delamination may occur. The original surface is thereby lost over wide areas, and the component becomes inoperative through loss of dimensional accuracy, noise, or secondary damage.

In many cases, micropitting is not destructive to the gear-tooth surface. It sometimes occurs only in patches and may stop after the tribological conditions have been improved by run-in. Run-in is defined as an initial transition process occurring in newly established wearing contacts, often accompanied by transients in coefficient of friction, wear rate, or both, that are uncharacteristic of the long-term behavior of the given tribological system. Gears appear less sensitive to the consequences of surface distress than rolling-element bearings. This may be due to the greater ductility of lower-hardness materials.

Subcase Fatigue

Subcase fatigue is fracture of case-hardened components by the formation of cracks below the contact surface within the Hertzian stress field. However, the depth at which the cracks form is much greater than those that cause macropitting fatigue, and it is a function of material strength in conjunction with the alternating Hertzian shear stresses. Subcase fatigue is also sometimes referred to as *case crushing,* but because it results from a fatigue crack that initiates below the effective case depth or in the lower-carbon portion of the case, the former nomenclature is used.

In case-hardened components—produced by carburizing, nitriding, or induction hardening—a gradient of decreasing hardness (shear strength) exists from the case to the core. If the material shear strength gradient is steeper than the gradient in Hertzian shear stress, then the applied-stress-to-strength ratio is least favorable at some depth below the normal maximum Hertzian shear-stress region. This depth is usually located at (or near) the case/core interface due to shear strength and stress gradients, in addition to unfavorable tensile residual stresses. The reversal in the sign of the residual stress from compression in the case to tension in the core occurs at or very near the case/core interface, and therefore the Hertzian shear stresses are increased due to the tensile residual stresses.

The fatigue originates near the case/core interface where the shear stresses, which are still fairly high, exceed the strength of the relatively low-strength core metal in that region. Fatigue cracks propagate parallel to the surface but underneath the case, then they join and form cracks that come up through the case to the surface, resulting in macropit formation. A typical example of this type of failure in gear teeth is shown in Figure 14.

Subcase fatigue is usually caused by too thin a case, such as on a nitrided gear tooth, by insufficient core hardness on carburized or induction-hardened parts, or in heavily loaded case-hardened rolling-element bearings. Factors that tend to promote the various types of contact fatigue are listed in Table 4. It should be noted that three of the five modes of fatigue damage are influenced by lubrication, including all surface-origin damage modes.

Location of Pitting Initiation on the Gear Tooth Profile

The initiation of pitting is mostly confined to three areas along the profile of a gear tooth. First, the pitchline is the only area of line contact that receives pure rolling pressure. The pits forming directly at the pitchline are usually very small and may not progress beyond the point of original initiation. Some analysts claim that this type of pitting repairs itself and is not detrimental. That may be true when the pitting is formed by the breakdown of a rough surface. Note, too, that lubrication will not deter the origin of pitting along the pitchline. Oil is noncompressible and will not cushion the pressures exerted in the area of pure rolling.

Second, the area immediately above or below the pitchline is very susceptible to pitting. Not only is the rolling pressure great at this point but sliding is now a real factor. The mechanics of surface-subsurface pitting can be best understood when looking at the resultant applied stresses illustrated in Fig. 15. The extra shear stress of the sliding component when added to that of the rolling component often results in near-surface fatigue at the point of maximum shear below the surface. In many instances, it is difficult to determine whether some pitting cracks actually initiate at or below the surface. Figure 16 shows initiation of pitting fatigue both at the pitchline of a helical-gear tooth and directly above the pitchline. Progression up the addendum in some areas makes it difficult to differentiate the two. A surface-pitted area near the pitchline is illustrated in Fig. 17. Although the surface was in a rolling-sliding area, the immediate surface shows no catastrophic movement. It is then most probable that cracking initiated by shearing fatigue below the surface. The freedom from surface movement also speaks well for the lubrication of the gear teeth.

Third, the lowest point of single-tooth contact is that point which receives the tip of the mating tooth as it makes first contact low on the active profile. Tip contact will produce high pressures, even if a small fraction of the actual load is transferred by the tip. Also, the maximum sliding speed, both in approach and recess, occurs at the tip contact.

There are two types of pits that prevail at this area. One is the type shown in Fig. 18, which appears to be a swift-shear and lift-out type.

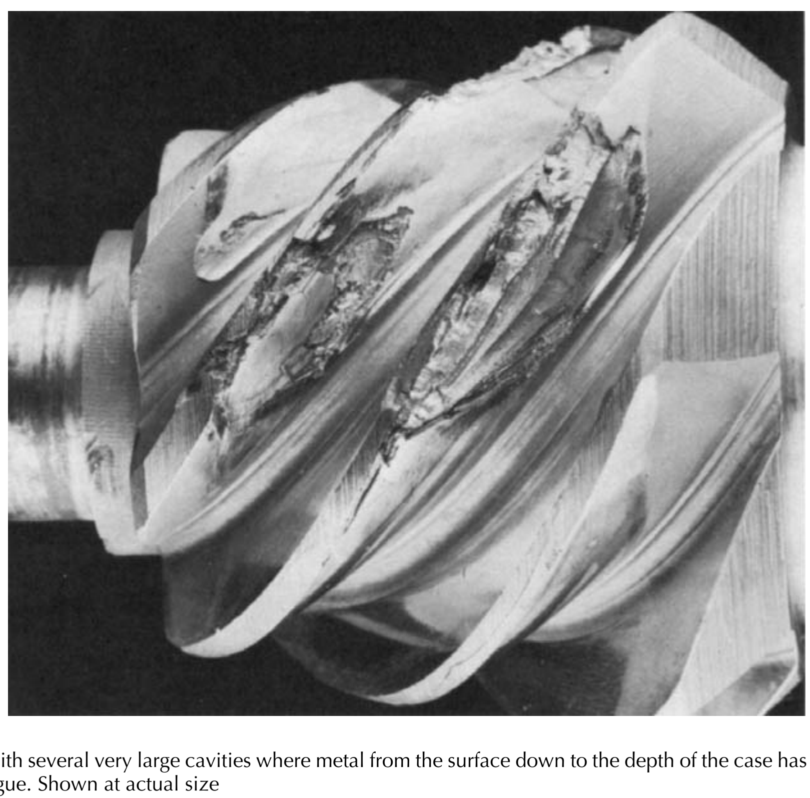


Fig. 14 Pinion with several very large cavities where metal from the surface down to the depth of the case has fallen out due to subcase fatigue. Shown at actual size

Perhaps the question whether this is fatigue or impact can be answered, but if it is shearing fatigue, it is certainly rapid and would call for a study of tip interference and lubrication. The second type of pit is the same as that illustrated in Fig. 17 and shows failure only by compressive loads. This is not uncommon in spur-gear teeth or helical-gear teeth, but, when found, is puzzling because one does not expect to find tip interference pits other than those previously discussed. However, it must be remembered that dynamic effects are to be taken into account. Gear vibration is a fact, but is seldom ever recognized as such. Studies indicate that gear vibration at high speeds, excited by static transmission error, may cause corner contact even though the tip relief is sufficient to prevent corner contact at low speed. Also, with sufficient lubrication, the pressures of tip interference will result in compressive-type pitting (Fig. 17) rather than the adhesive type (Fig. 18).

Other than the three specific and most common areas of surface-fatigue pitting just discussed, there are always the random areas that

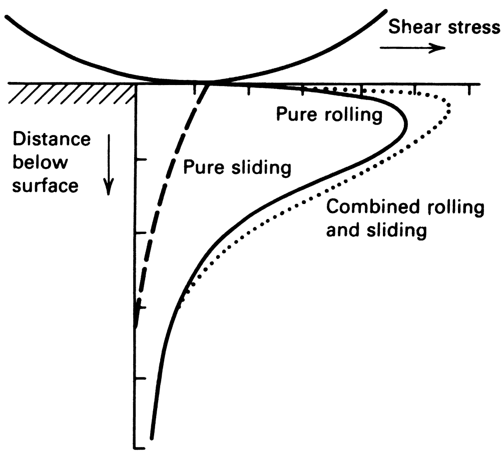


Fig. 15 Stress distribution in contacting surfaces due to rolling, sliding, and combined effect

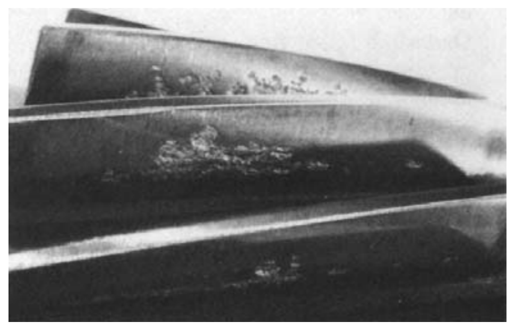


Fig. 16 Pitting initiated along and immediately above the pitchline in helical gear teeth. In some areas, the progression has been continuous. Shown at actual size

Table 4 List of contact fatigue modes and their controlling factors

Mode of failure	Factors that control occurrence
Subsurface	
Inclusion origin	Size and density of oxide or other hard inclusions
	Absence of other modes of failure
Subcase fatigue	Low core hardness
	Thin case depth relative to radius of curvature and load
Surface	
Point-surface origin	Low lubricant viscosity
	Thin EHD film compared with asperities in contact
	Tangential forces and/or gross sliding
Geometric stress concentration	End contact geometry
	Misalignment and deflections
	Possible lubricant film thickness effects
Micropitting	Low lubricant viscosity
	Thin EHD film compared with asperities in contact
	Loss of EHD pressure
	Slow operating speeds

Source: Ref 5

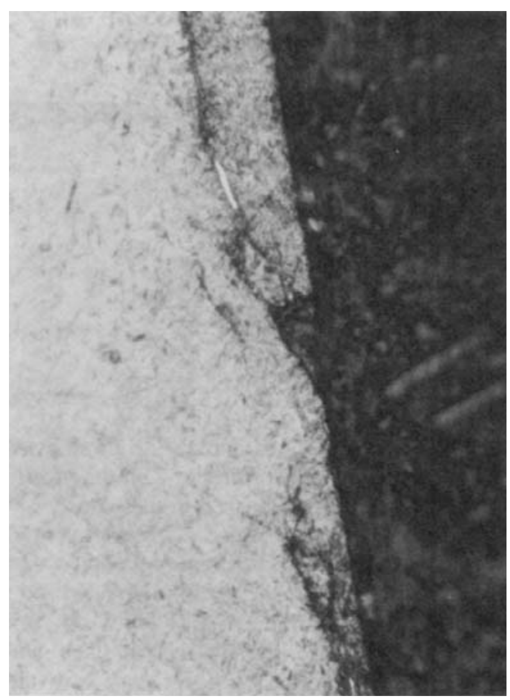


Fig. 17 Near-pitchline pitting fatigue in a spiral pinion tooth. Origin is subsurface at plane of maximum shear. Original magnification at 180×

crop up to demand recognition, such as pitting at only one end of the face, at opposite ends of opposite faces, along the top edge of the addendum, or at one area of one tooth only. The random incidences are without number and are most baffling; but each one is present because of a reason, and that reason must be found if at all possible.

Rolling-Contact Fatigue

Rolling-contact fatigue can only be described by using Fig. 15. Under any conditions of rolling, the maximum stress being applied at or very near the contact area is the shear stress parallel to the rolling surface at some point below the surface. For normally loaded gear teeth, this distance is from 0.18 to 0.30 mm (0.007 to 0.012 in.) below the surface just ahead of the rolling point of contact. If sliding is occurring in the same direction, the shear stress increases at the same point. If the shear plane is close to the surface, then light pitting will probably occur. If the shear plane is deep due to a heavy rolling load contact, then the tendency is for the crack propagation to turn inward (Fig. 19). The cracks will continue to propagate under repeated stress until heavy pitting or spalling takes place (Fig. 20).

There is always one characteristic (and often two) of rolling-contact fatigue that will distinguish it from other modes of surface-contact fatigue. Both characteristics can be observed only by an examination of the microstructure. In rolling contact, the surface will not show a catastrophic movement; it will remain as the original structure. For example, an unetched, polished sample taken near the origin of a subsurface fatigue crack (Fig. 21) very clearly shows undisturbed black oxides at the surface. The subsurface cracking could not have been caused by either abrasive or adhesive contact; it had to be by rolling. This is always the first evidence to look for when determining the type of applied stress.

The second characteristic is common only in a martensitic case that contains very little or no austenite and is found only at, along, or in line

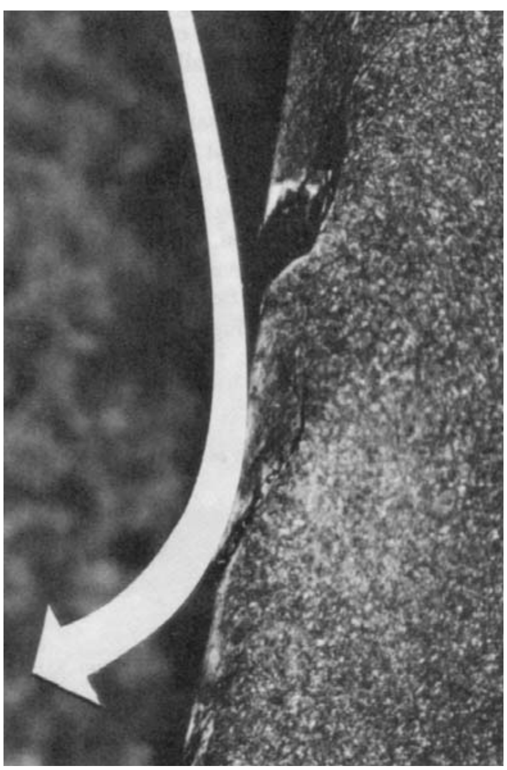


Fig. 18 Spiral bevel tooth failure. Pitting at the lowest point of single-tooth contact illustrating contact path of the tip of the mating tooth. Nital etch. Original magnification at 90×

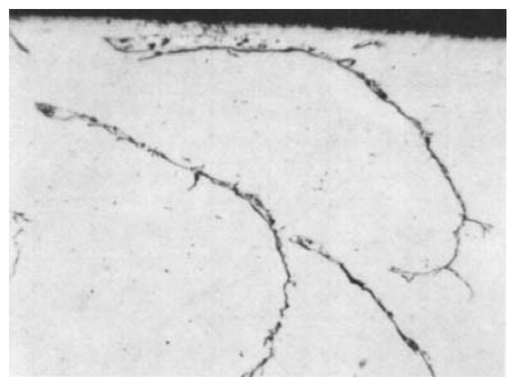


Fig. 19 Rolling-contact fatigue in a gear tooth section. Crack origin subsurface. Progression was parallel to surface and inward away from surface. Not etched. Original magnification at 60×

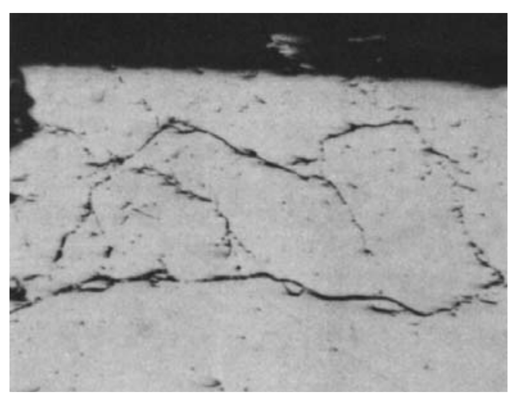


Fig. 20 Rolling-contact fatigue in a gear tooth section. Crack origin subsurface. Progression was parallel with surface, inward, and finally to the surface to form a large pit or spall. Not etched. Original magnification at 60×

with the shear plane. This is a microstructural feature that has been termed butterfly wings. If the sample in Fig. 21 is etched properly with 3% nital, the result is the microstructure shown in Fig. 22(a). Increasing the magnification to 310× shows more detail (Fig. 22b).

The original discussion presented by rolling-element bearing manufacturers had no explanation as to the origin of this microstructural alteration except that it was always in conjunction with and radiated from an inclusion. It is true that many times there is an inclusion present, but not always. One could argue that the fracture is a plane and not a line, such as could be seen on a polished sample; therefore, an inclusion could be present somewhere else along the plane being observed.

Two other extensive and very detailed studies of rolling-contact fatigue refer to the gray substructure as white bands of altered martensite. It is believed that these substructures are caused when, under an extreme shearing stress, movement is called for but is restrained and contained to such an extent that the energy absorbed institutes a change in the microstructure ahead of a progressing crack. They are never observed when significant amounts of austenite are present; austenite will quickly absorb the energy and will be converted to untempered martensite. They are also in an area that has not been deformed but has definitely been transformed. Each area has distinct boundaries, and the oncoming cracks appear to follow these boundaries. It has been noted that some academic studies refer to this same structure as being a transformed shear band product formed by adiabatic shear.

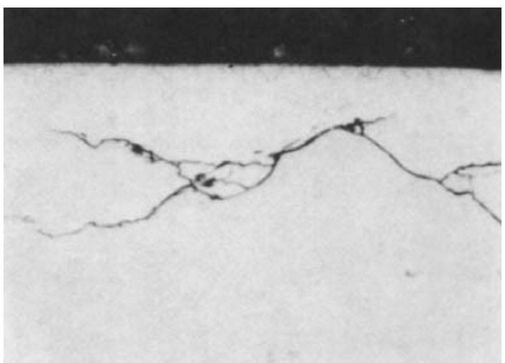


Fig. 21 Rolling-contact fatigue in a gear tooth section distinguished by subsurface shear parallel to surface. Note the undisturbed black oxides at the surface, indicating no surface-material movement. Not etched. Original magnification at 125×

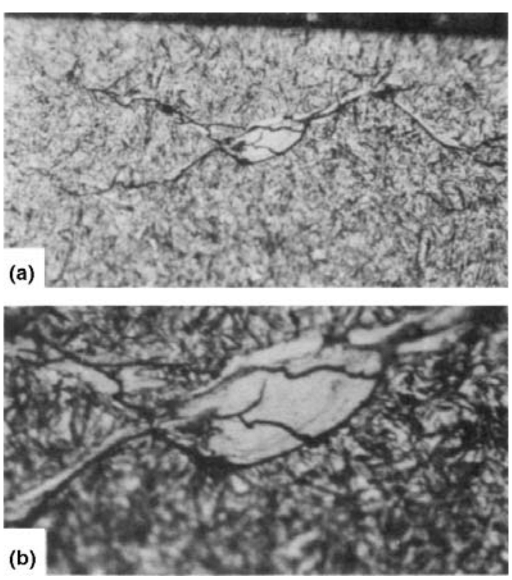


Fig. 22 Same sample as in Fig. 21, only etched in 3% nital, showing details of submicrostructure called butterfly wings. (a) Original magnification at 125×. (b) Original magnification at 310×

Spalling

Spalling, in general, is not considered an initial mode of failure but rather a continuation or propagation of macropitting and rolling-contact fatigue. It is very common to reference this failure mode as pitting and spalling. For example, Fig. 23 shows a spiral-gear tooth with pitting low on the profile that subsequently progressed until spalling occurred over the top face and back side profile. This apparent rapid and extensive progression can be referred to as the cyclone effect.

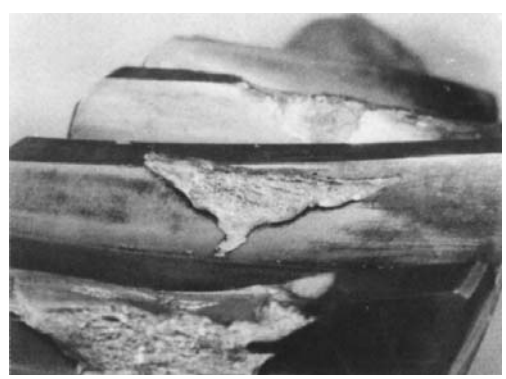


Fig. 23 Spalling of spiral bevel gear teeth. Original pitting low on the active profile gives initiation to a fast and extensive progression of spalling over the top face and down the back profile. This is often called the cyclone effect. Original magnification at 0.25×

Spalling, in the true sense of the word, is a mode of fatigue failure distinct in itself and unique in its origin. It originates below the surface, usually at or near the case/core transition zone. As illustrated in Fig. 24, the origin is at that point where the sum of all applied and misapplied stresses intersect the net strength of the part. The applied stresses are substantially in shear, and the intercept point is most likely under a carburized case. Fatigue fractures will generally progress under the case (Fig. 25 and 26) and will eventually spall away from the gear tooth.

Gear failures due to spalling in conjunction with other failure modes are also common. Figure 27 shows a pinion that failed because of spalling, abrasive wear, and plastic deformation.

Thermal Fatigue

Thermal fatigue in gearing, although not necessarily correct, is most often synonymous with frictional heat. This is perhaps the only type of alternating heating and cooling that is applied in the field. Then, too, it is not commonly found on the active profile of the gear teeth, but on a rotating face that is exerting a high amount of thrust. Figure 28(a) shows an example of high end thrust that resulted in frictional heat. The thermal expansion and contraction initiated a large number of radial cracks that progressed as a fatigue fracture (Fig. 28b) into the bore of the part and along the roots of the teeth.

Impact

Tooth-Bending Impact. When a tooth is removed from a gear within a very few cycles (usually one or two), the fracture is uniform in structure and does not show the striations common to the fatigue mode of failure. They are usually random, due to a sudden shock load, either in a forward or reverse direction, and do not necessarily originate at the root radius. In fact, if the fracture did originate at the root

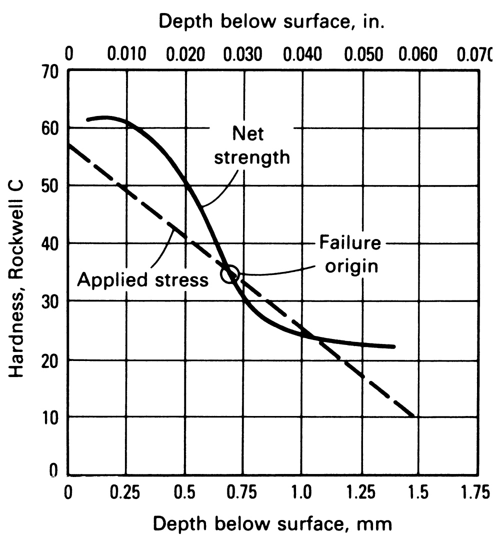


Fig. 24 Applied stress versus case depth (net strength)

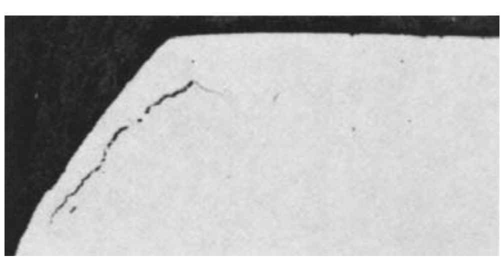


Fig. 25 Subsurface cracking that subsequently resulted in spalling at a gear-tooth edge. Unetched section of a carburized AMS 6260 steel gear tooth. Cracking initiated in the transition zone between the carburized case and the core. Original magnification at 500×

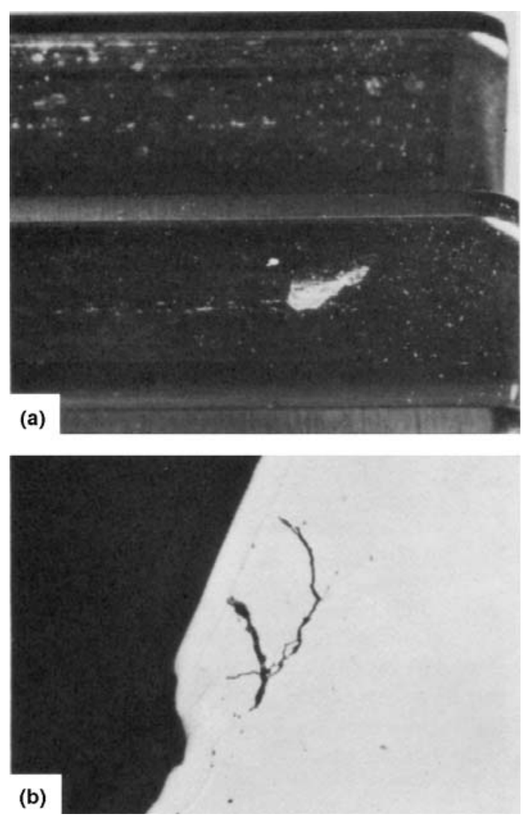


Fig. 26 Spalling on a tooth of a steel spur sun gear shaft. (a) Overall view of spalled tooth. (b) Micrograph of an unetched section taken through the spalled area showing progressive subsurface cracking. Original magnification at 100×

radius, it would follow a rather flat path across to the opposite root radius rather than travel downward toward the zero-stress point.

Tooth Shear. When the impact is very high and the time of contact is very short, and if the ductility of the material will allow it, the resultant tooth failure mode will be shear. The fractured area appears to be highly glazed, and the direction of the fracture will be from straight across the tooth to a convex shape. For example, a loaded gear and pinion set were operating at a high rate of speed when the pinion stopped instantaneously (Fig. 29). The momentum of the gear was strong enough to shear the contacting pinion teeth from the reverse direction, leaving the remaining teeth in excellent condition. The gear teeth were partially sheared and were all scrubbed over the top face from the reverse direction.

Most commonly, bending impact and shear are found in combination (Fig. 30). Tooth-bending impact fractures supersede the shearing fractures; thus, the direction of impact can be determined.

Tooth chipping is certainly a type of impact failure, but is not generally considered to be tooth-to-tooth impact. It is usually accomplished by an external force, such as a foreign object within the unit, a loose bolt backing out into the tooth area, or even a failed tooth from another gear. Figure 31 shows a chipping condition that existed during the operation of the part. Most chipping, however, is caused by mishandling before, during, or after the finished parts

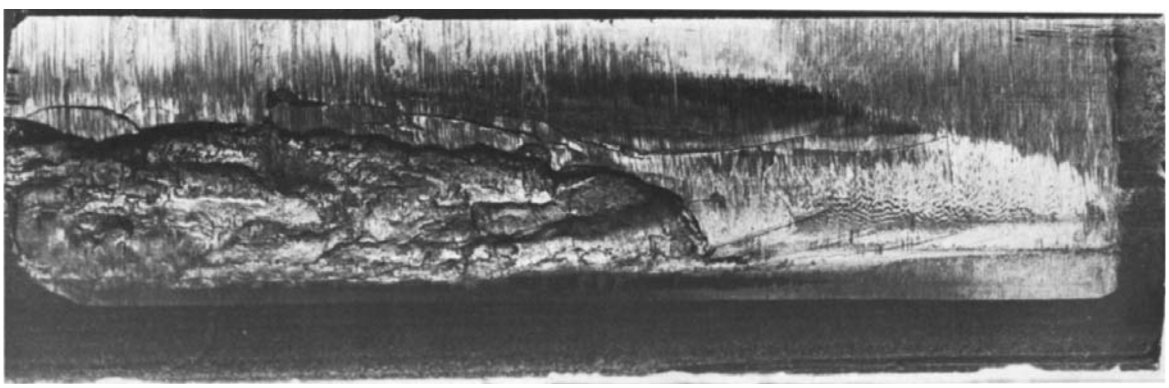


Fig. 27 Surface of a spalling-fatigue fracture in a single tooth of a heavily loaded final-drive pinion of AISI 8620 steel, carburized and hardened to 60 HRC in the case, showing vertical scratches, which indicate that appreciable abrasive wear took place also. The surface ripples at right suggest that a small amount of plastic flow occurred under the applied load. Original magnification at ~4×

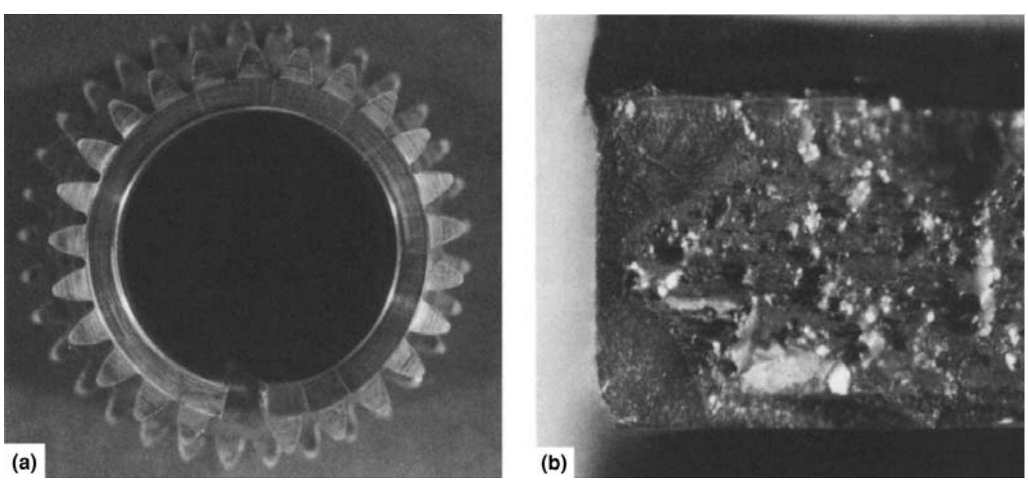


Fig. 28 Thermal fatigue cracking of a spur gear. (a) Radial cracking due to frictional heat against the thrust face. Original magnification at 0.4×. (b) Progression of thermal fatigue produced by the frictional heat. Original magnification at 1.5×

are shipped from the manufacturer and are not field failures. This is one of the modes that must be identified by a cause.

Wear

Surface deterioration of the active profile of the gear teeth is called wear. There are two distinct modes of wear—abrasive and adhesive. Additional information on these wear failure modes can be found in Chapter 2, "Gear Tribology and Lubrication."

Abrasive wear occurs as the surface is being freely cut away by hard abrasive particles. It can happen only as two surfaces are in sliding contact. The dissociated material must be continually washed away and not allowed to accumulate on the sliding surfaces, or adhesion may take place. The first evidence of abrasive wear is the appearance of light scratches on the surface, followed by scuffing. As the scuffing continues to deepen, the result is scoring. Figure 32 illustrates an excellent example of pure abrasive wear that shows the entire tooth profile cleanly cut away with no microstructural damage to the underlying material.

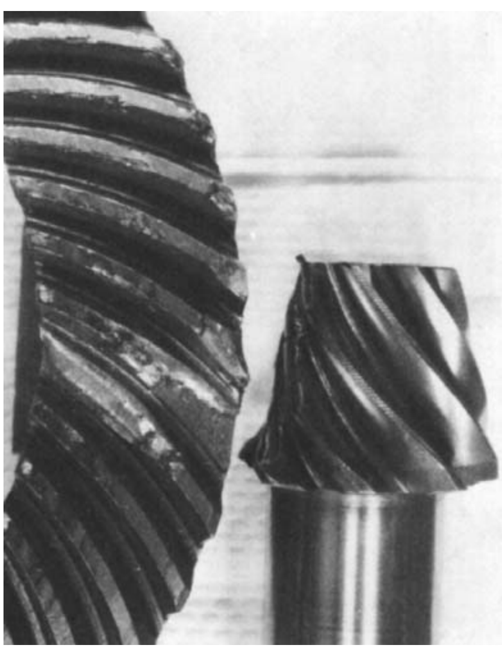


Fig. 29 Spiral bevel gear and pinion set that sheared in the reverse direction. The pinion came to a sudden and complete stop at the instant of primary failure of the unit, allowing the gear to shear the contacting teeth and to continue rotating over the failed area. Original magnification at 0.4×

Abrasive wear cannot be deterred by lubrication, because the lubricant is often the vehicle that contains and continually supplies the abrasive material as a contaminant. When this is the case, all moving parts within the assembly are affected, such as seals, spacers, bearings, pumps, and matching gears.

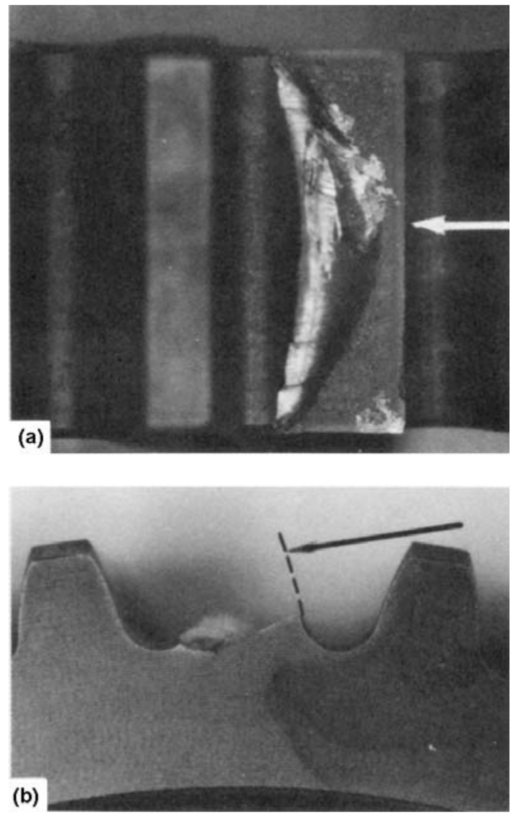


Fig. 30 Spur-gear tooth showing combination failure modes. (a) Tooth-bending impact. (b) Tooth shear. Arrows indicate direction of applied force.

Fig. 31 Tooth chipping resulting from impact of a foreign object within the assembled unit

When abrasive wear is isolated to only one part, that is, either gear or pinion, it is imperative to examine closely the surface of the matching part. For example, a very large amount of massive carbides embedded in the surface may easily cut into a softer matching surface.

Adhesive wear occurs on sliding surfaces when the pressure between the contacting asperities is sufficient to cause local plastic deformation and adhesion. Whenever plastic deformation occurs, the energy expended to produce the deformation is converted to heat.

The first indication of trouble is a glazed surface, followed by galling, then by seizure. A glazed surface may not have any dimensional changes, but the microstructure shows a catastrophic movement of surface material (Fig. 33). As frictional heat increases, the surface becomes softer, the adhesive tendency becomes greater, there is more plastic deformation, the local temperature becomes high enough to change completely the microstructure at the surface (Fig. 34), and galling occurs. Galling continues, the sliding surfaces begin welding together, and adhesion takes place. Large areas may be pulled away from the surface (Fig. 35). Some of these welded particles are so hard due to change of microstructure that they become sharp points that cut into the matching parts. As the adhesive wear continues with the parts still in service, there is the distinct possibility that the part will ultimately fail (Fig. 36).

Adhesive wear on gear teeth may not always point to that set as the culprit. There have been instances when a severe case of adhesion is of a secondary nature. In other words, particles of material from a primary failure of another component in the assembly may have impinged upon the gear-tooth surfaces and thus started the sequence of events leading to seizure and destruction.

Wear-Resistant Coatings (Ref 6). Although the gear industry has been slow to implement gear coatings, there have been a fairly wide variety of coating methods/materials that have been used to coat gears. Their uses range from cosmetic to doubling the wear life of a gear set. They include technologies that are just emerging, such as the coatings applied by physical vapor depostion (PVD) discussed below, as well as well-established coating systems that have been used for more than 50 years including black oxide conversion coatings, electroless nickel coatings, electroplated coatings (chrome and nickel), and phosphate coatings. The following represent some of the more recently developed wear-resistant options available for steel gears:

- *Boron carbide* (B_4C) is a very hard, amorphous ceramic material applied using the magnetron sputtering PVD process. This coating has been used for coating transmission gears for major auto makers. The coating thickness is about 2 to 3 μm.
- *Diamond-like carbon* (DLC) is a hard (1000–3000 HV) low-friction coating of an amorphous form of carbon with diamond-like bonds. It is applied by ion-beam PVD. This gear coating is the focus of intense research by automotive manufacturers and the aerospace community.

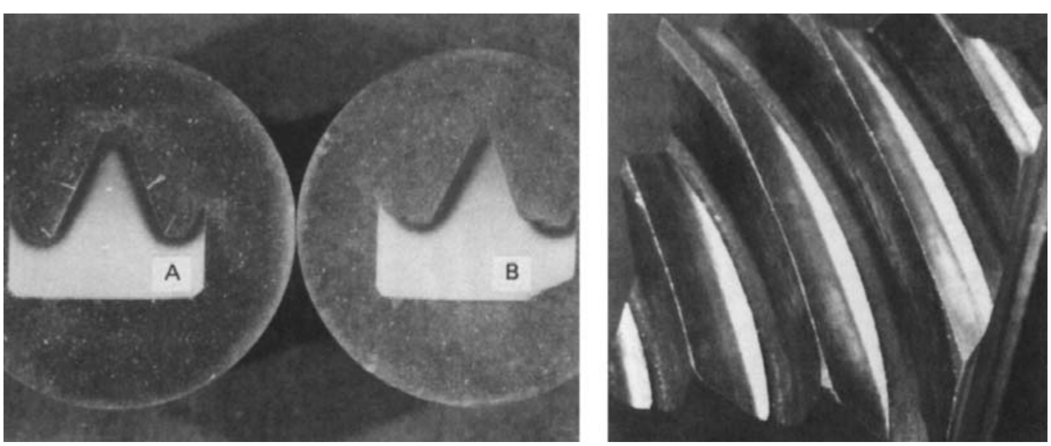


Fig. 32 Spiral bevel gear teeth showing contact wear. Insert A shows a tooth area exhibiting no wear. Insert B shows abrasive wear clearly cutting away 3.2 mm (⅛ in.) of the surface without damage to underlying material.

- *Molybdenum disulfide* (MoS_2) has become the workhorse of dry-film lubrication for gears. It combines a low coefficient of friction with high load-carrying capacity, and it works well in a vacuum. For lower precision gears, it can be used in powder form or applied by special techniques such as spraying or dipping, followed by curing in an oven. It can also be applied by PVD sputtering, which allows much tighter tolerances and thinner films.
- *Tungsten carbide-amorphous carbon* (WC/C) coatings are applied by the reactive sputtering PVD process. Application areas, both present and potential, for these coatings include motorcycle gears, concrete mixer gears, bevel gear actuators for aircraft landing gear flaps, highly-stressed, fast-running gears, and worm gear drives.

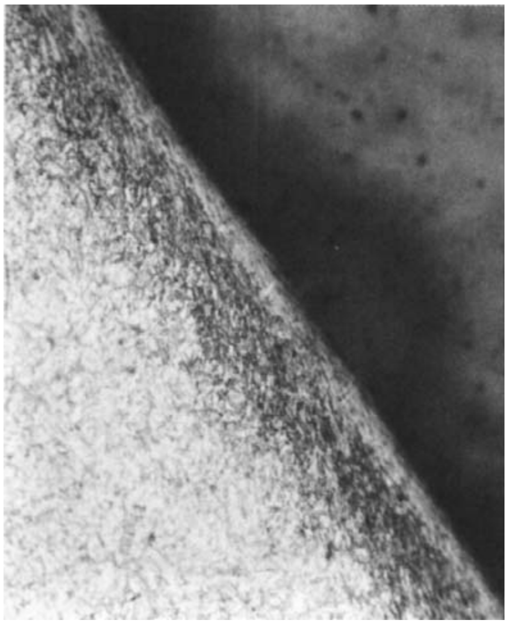


Fig. 33 Pinion tooth profile. Glazed surface showing the start of catastrophic movement of surface material. Frictional heat has already started to temper the surface. Original magnification at 75×

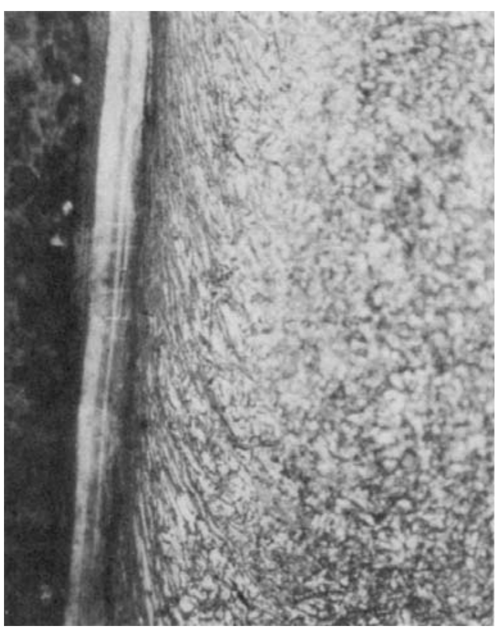


Fig. 34 Pinion tooth profile. The pinion was plastically deformed by frictional heat and sliding pressures. The surface layer has locally rehardened, and galling is evident. Original magnification at 80×

Scuffing

Scuffing, which is sometimes also referred to as scoring, is a lubrication failure, frequently brought on by an increase in the operating temperature. The lubricant film breaks down, allowing metal-to-metal contact at high spots (asperities) on the flank surfaces. Under sufficient conditions of load and temperature, this

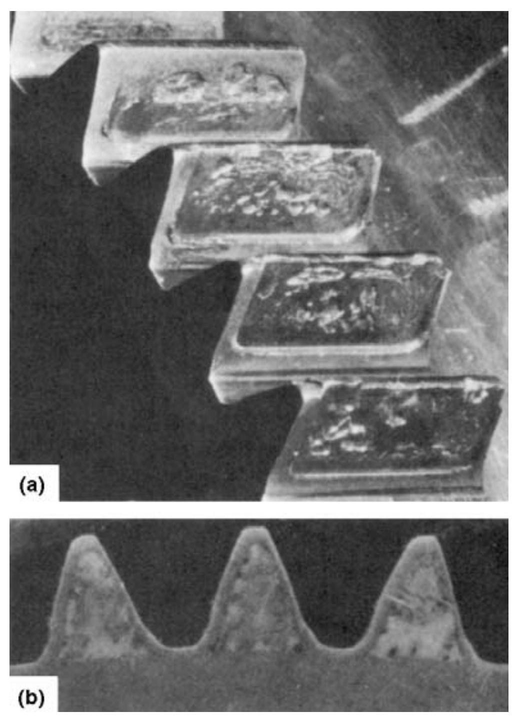


Fig. 35 Carburized AMS 6260 steel gear damaged by adhesive wear. (a) Overall view of damaged teeth. (b) Etched end face of the gear showing excessive stock removal from drive faces of teeth

metal-to-metal contact causes asperities to weld together and then tear apart as the motion continues. The surface is characterized by torn or furrowed markings in the direction of sliding. Figure 37 shows a typical scuffed gear. Similar damage can occur without the instantaneous welding of the contact surfaces. This damage is sometimes caused by foreign material being carried through the mesh but is more frequently caused by high asperities on one contacting surface plowing through the other. This damage is termed *cold scoring* (or *scuffing*), or *burnishing,* and is differentiated from that described previously by the absence of tearing. More detailed information on scuffing can be found in Chapter 2, "Gear Tribology and Lubrication."

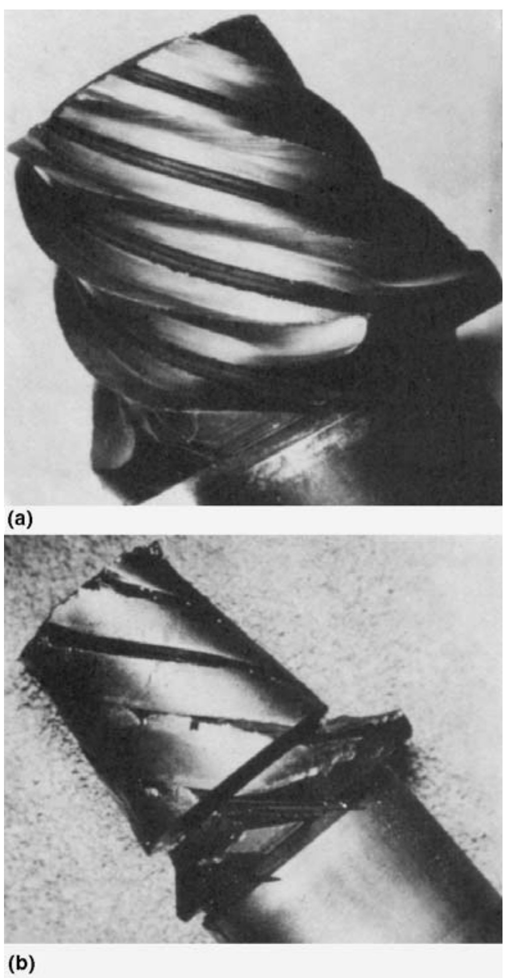


Fig. 36 Extreme wear of steel pinions. (a) Hypoid pinion with teeth worn to a knife-edge from both sides. (b) A similar pinion with teeth worn completely away by adhesive wear

Stress Rupture

When the internal residual stresses build to a magnitude beyond the strength of the material, the part will rupture. The rupture will occur at the point where this critical value is exceeded, either internally or externally.

Internal Rupture. The most likely point within a gear that would attract a buildup of residual stresses is the case/core interface near the top face or the corner of a tooth. Figure 38 shows an excellent example of case/core separation due to internal residual stresses exceeding the strength of material at the case/core transition zone. In severe cases, the entire top of the tooth might pop off (Fig. 39).

External rupture is much easier to understand because it almost always originates from a stress raiser. For example, a part not yet in service ruptured starting at the end face (Fig. 40) from a grinding crack; this was confirmed by examination of the fractured face (Fig. 41). The grinding cracks were not the entire cause of the failure, but they supplied the notch in the highly stressed area.

Causes of Gear Failures

Analysis of the actual part that failed will show the mode of failure, but the cause of failure is often found in the mating or matching part. One should always inspect both parts very closely for the correct answer to the problem. In fact, if there are several components to an assembly, each component must be suspect until eliminated.

The cause of a specific gear failure may be obvious enough to be recognized by anyone or subtle enough to defy recognition by the most experienced analyst. At times, one almost believes that the number of causes may equal the number of failures. This is close enough to being correct that each failure must be treated individually as a complete case in itself.

It is very difficult in many instances to separate cause from mode. In fact, a single cause can initiate several different modes, depending upon the forces being applied. Conversely, a specific mode of failure could have been initiated by one of several causes. It must be emphasized that one should not match a fracture pattern of a failed gear to a picture in this chapter and conclude that the cause and effect are the same. They may be, but not necessarily. The pictures of a reference book should be used only as a guide. Causes, as listed in Table 5, are under five major headings and several subheadings. Undoubtedly, there may be more. The headings are not listed in order of importance (they are all important), but in sequence of probable occurrence. An example of a gear failure due to improper heat treatment is shown in Fig. 42 and 43.

Conducting the Failure Analysis (Ref 7)

When gears fail, there may be incentive to repair or replace failed components quickly and return the gear system to service. However, because gear failures provide valuable data that may help prevent future failures, a systematic inspection procedure should be followed before repair or replacement begins.

The failure investigation should be planned carefully to preserve evidence. The specific approach can vary depending on when and where the inspection is made, the nature of the failure, and time constraints.

Ideally, the analyst should visit the site and inspect failed components as soon after failure as possible. If an early inspection is not possible, someone at the site must preserve the evidence based on instructions from the analyst.

The failure conditions can determine when and how to conduct an analysis. It is best to shut down a failing gearbox as soon as possible to limit damage. To preserve evidence, carefully plan the failure investigation and conduct in-situ inspections and plan to become involved in gearbox removal, transport, storage, and disassembly. If the gears are damaged but still functional, the company may decide to continue operation and monitor damage progression. In this case, be certain to become involved in establishing the gear system monitoring process. In most applications, inspection and monitoring include visual inspection and temperature, sound, and vibration measurements. Additionally, for critical applications, nondestructive inspection of the gears (e.g., magnetic particle inspection) should ensure the absence of cracks before operation is continued. Before the system is restarted, be certain to collect samples of lubricant for analysis, drain and flush lubricant reservoirs, and replace the lubricant. Examine the oil filter for wear debris and contaminants, and inspect magnetic plugs for wear debris.

The high cost of shutdown frequently limits time available for inspection. Such cases call for careful planning. Dividing tasks between two or more analysts may reduce time required and provide varied insight into the failure analysis task.

Prepare for Inspection

Before visiting the failure site, the analyst should interview a contact person and explain

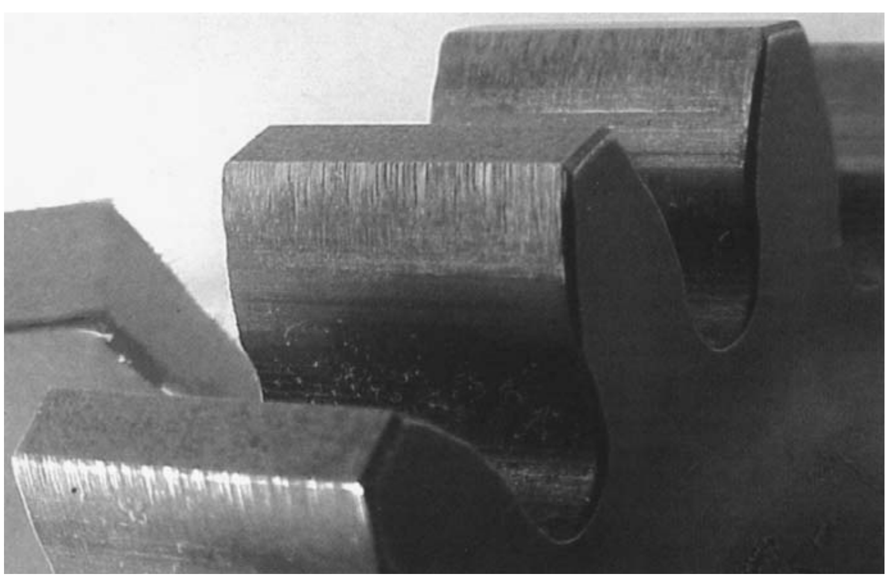


Fig. 37 Specimen gear with scuffing failure. Source: Ref 4

the failure analysis process and outline specific needs. Emphasize the need to inspect the gearbox, interview personnel, examine equipment, and assess working conditions.

A skilled technician should be requested to disassemble the equipment under the direction of the analyst. However, if safety permits, it is best if *no work is done on the gearbox until the analyst arrives.* This means no disassembly, cleaning, or draining oil. Otherwise, a well-meaning technician could inadvertently destroy evidence. Emphasize that failure investigation is different from a gearbox rebuild, and the disassembly process may reveal significant facts to a trained observer.

Verify that gearbox drawings, disassembly tools, and adequate facilities are available. Inform the contact person that privacy is required to conduct the investigation, and access to all available information is necessary.

Ask for as much background information as possible, including specifications of the manufacturer, service history, load data, and lubricant analyses. Send a questionnaire to the contact person to help expedite information gathering.

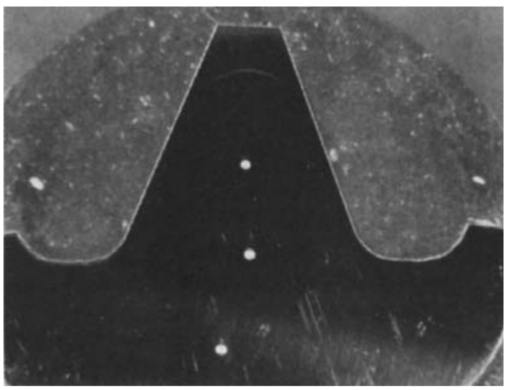


Fig. 38 An internal rupture in a gear tooth at the case/core transition zone. The rupture does not reach the surface. This condition can be discovered by ultrasonic testing.

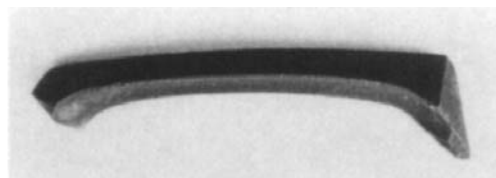


Fig. 39 Spiral bevel gear tooth failure. Internal rupture is lifting the entire top of a tooth.

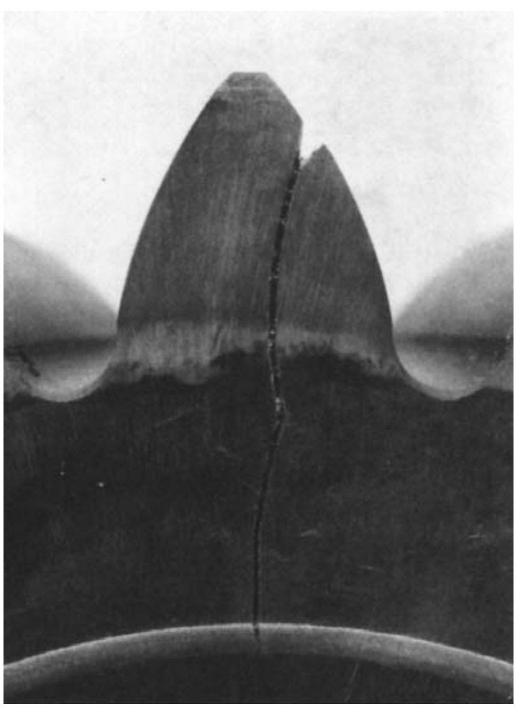


Fig. 40 Spur gear failure. External rupture (assembled, but not in service) with origin at the end face. See also Fig. 41.

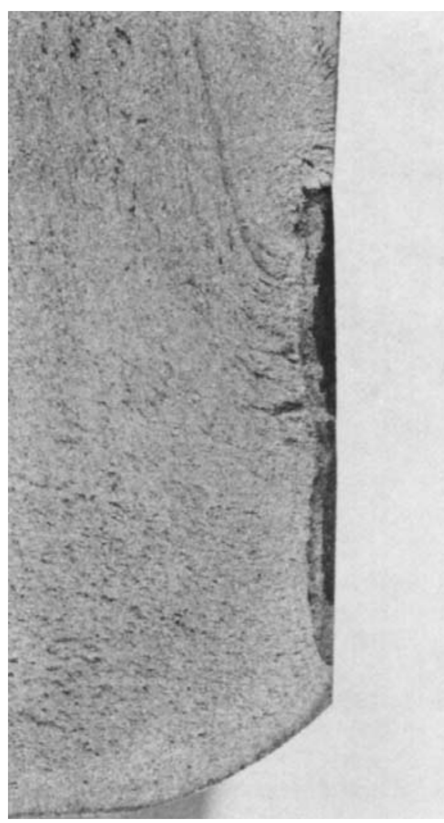


Fig. 41 Spur gear. External rupture originating from a grinding check. See also Fig. 40.

Inspect In Situ

Before starting the inspection, review background information and service history with the contact person. Try to interview those involved in design, installation, startup, operation and maintenance, and anyone present when failure of the gearbox occurred or was discovered. Encourage the interviewees to share everything they know about the gearbox and associated systems even if they feel it is not important.

External Examination. Before removing and disassembling the gearbox, take photographs and thoroughly inspect the exterior. Use an inspection form to ensure that important data (data that may be lost once disassembly begins) is recorded. For example, the condition of seals and keyways should be recorded before disassembly or it may be impossible to determine when these parts were damaged.

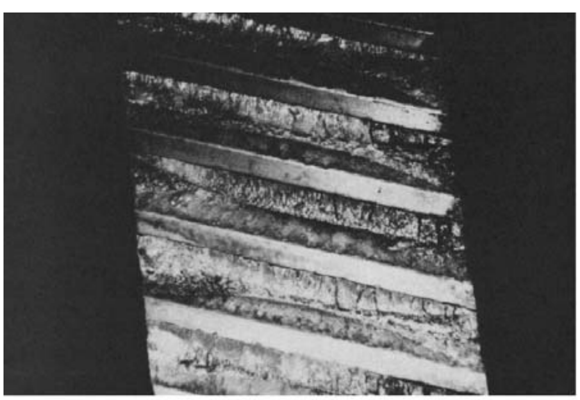
Before cleaning the exterior of the gear housing, inspect for signs of over-heating, corrosion, contamination, oil leaks, and damage, and photograph the areas of interest. Photographic documentation is frequently a key to any good failure analysis, including a gear failure analysis.

Gear Tooth Contact Patterns. To observe the condition of the gears, shafts, and bearings, clean the inspection port cover and the immediate area around it, and then remove the cover. Be careful not to contaminate the gearbox dur-

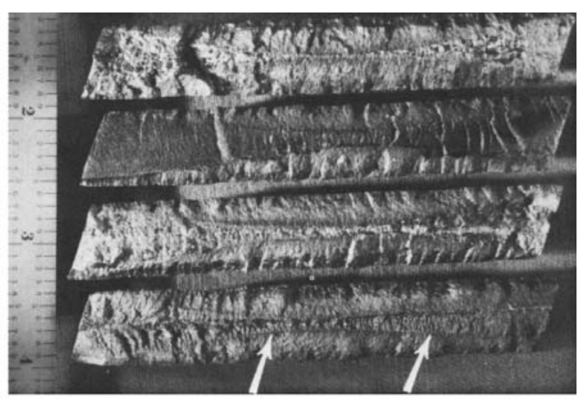
Table 5 Summary of a statistical report on causes of gear failures (%) over a 35-year period

Basic material	0.8
Steel	
Forgings	
Castings	
Engineering	6.9
Design	
Material selection	
Heat treat specifications	
Grinding tolerances	
Manufacturing	1.6
Tool undercutting, sharp notches	
Tooth characteristics	
Grinding checks, burns	
Heat treat changes	
Heat treatment	16.0
Case properties	
Core properties	
Case/core combinations	
Hardening	
Tempering	
Miscellaneous operations	
Service application	74.7
Set-matching	
Assembly, alignment, deflection, vibration	
Mechanical damage, mishandling	
Lubrication	
Foreign material	
Corrosion	
Continuous overloading	
Impact overloading	
Bearing failure	
Maintenance	
Operator error	
Field application	

Source: Ref 1



(a)



(b)

Fig. 42 Fatigue fracture of AISI 4140 bull gear due to improper heat treatment. The one-of-a-kind replacement gear had a service life of just 2 weeks. Heat treatment did not produce full hardness in gear teeth. Hardness at tooth face, 15 HRC; at tooth core, 82 HRB. (a) Outside diameter of gear showing where teeth broke off. Fatigue pattern on fracture surfaces is typical of reversed bending even though the gear was driven in only one direction. Original magnification at 0.6×. (b) Fracture surfaces of four teeth. Fatigue cracks propagated from both roots of each tooth. Arrows point to area of final overload on one tooth. Original magnification at 0.84×. See also Fig. 43.

ing cleaning or during the removal of the port cover.

The way gear teeth contact indicates how they are aligned. Record tooth contact patterns under loaded or unloaded conditions. No-load patterns are not as reliable as loaded patterns for detecting misalignment, because marking compound is relatively thick and no-load tests do not include misalignment caused by load, speed, or temperature. Therefore, follow no-load tests with loaded tests whenever possible. See ANSI/AGMA 2000 Appendix D for information regarding contact pattern tests.

No-Load Contact Patterns. For no-load tests, paint the teeth of one gear with soft marking compound and roll the teeth through mesh so compound transfers to the unpainted gear. Turn the pinion by hand while applying a light load to the gear shaft by hand or brake. Lift transferred patterns from the gear with clear tape and mount the tapes on white paper to form a permanent record.

Loaded Contact Patterns. For loaded tests, paint several teeth on one or both gears with machinist's layout lacquer. Thoroughly clean teeth with solvent and acetone, and *brush paint* with a thin coat of lacquer. Run the gears under load for sufficient time to wear off the lacquer and establish the contact patterns. Photograph patterns to obtain a permanent record.

Record loaded contact patterns under several loads, for example, 25, 50, 75, and 100% load. Inspect patterns after running approximately one hour at each load to monitor how patterns change with load. Ideally, the patterns should not change much with load. Optimum contact patterns cover nearly 100% of the active face of gear teeth under full load, except at extremes of teeth along tips, roots, and ends, where contact is lighter as evidenced by traces of lacquer.

Endplay and Backlash. Inspect endplay and radial movement of the input and output shafts and gear backlash.

Remove Gearbox

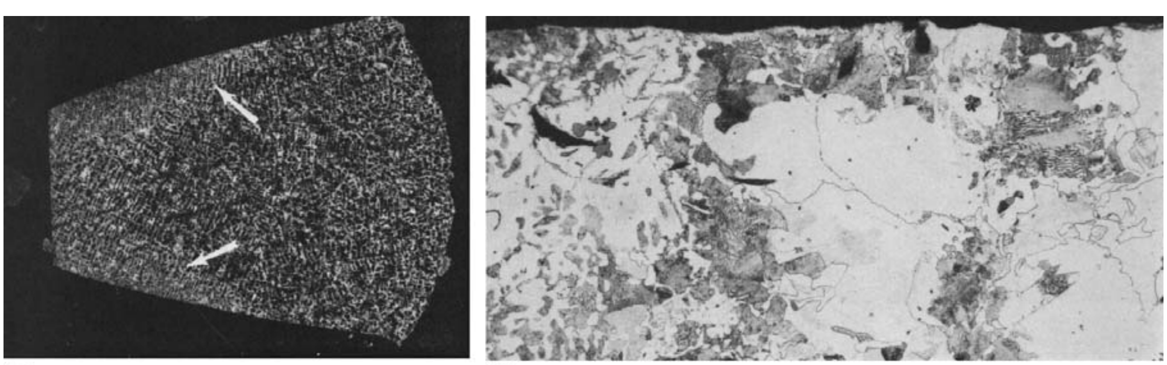
Mounting Alignment. Measure alignment of shaft couplings before removing the gearbox. Note the condition and loosening torque of all fasteners including coupling and mounting bolts. To check for possible twist of the gear housing, measure movement of the mounting feet as mounting bolts are loosened. Install four dial indicators, one at each corner of the gearbox. Each indicator will record the same vertical movement if there is no twist. If not, calculate the twist from relative movements.

Transport Gearbox

Fretting corrosion is a common problem that may occur during shipping. Ship the gearbox on an air-ride truck, and support the gearbox on vibration isolators to help avoid fretting corrosion. If possible, ship the gearbox with oil. To minimize contamination, remove the breather and seal the opening, seal labyrinth seals with silicone rubber, and cover the gearbox with a tarpaulin.

Store Gearbox

It is best to inspect the gearbox as soon as possible. However, if the gearbox must be stored, store it indoors in a dry, temperature-controlled environment.



(a) (b)

Fig. 43 Structure of fractured tooth from AISI 4140 bull gear in Fig. 42. (a) Tooth cross section. Arrows show depth to which temperature exceeded Ac_1 during heat treatment. (b) Near-surface hardened region of tooth. Large ferrite indicates that complete austenitization was not obtained during heat treating. 2% nital, original magnification at 200×

Disassemble Gearbox

Explain analysis objectives to the attending technician. Review the gearbox assembly drawings with the technician, checking for potential disassembly problems. Verify that the work will be done in a clean, well-lighted area, protected from the elements, and that all necessary tools are available. If working conditions are not suitable, find an alternate location for gearbox disassembly.

Because technicians usually are trained to work quickly, it is wise to remind him or her that disassembly must be done slowly and carefully.

After the external examination, thoroughly clean the exterior of the gearbox to avoid contaminating the gearbox when opening it. Disassemble the gearbox and inspect all components, both failed and undamaged.

Inspect Components

Inspect Before Cleaning. Mark relative positions of all components before removing them. Do not throw away or clean any parts until they are examined thoroughly. If there are broken components, do not touch fracture surfaces or fit broken pieces together. If fractures cannot be examined immediately, coat them with oil and store the parts so fracture surfaces are not damaged.

Examine functional surfaces of gear teeth and bearings and record their condition. Before cleaning the parts, look for signs of corrosion, contamination, and overheating.

Inspect After Cleaning. After the initial inspection, wash the components with solvents and re-examine them. This examination should be as thorough as possible because *it is often the most important phase of the investigation* and may yield valuable clues. A low-power magnifying glass and 30× pocket microscope are helpful tools for this examination.

It is important to inspect bearings because they often provide clues to the cause of gear failure. For example:

- Bearing wear can cause excessive radial clearance or endplay that misaligns gears.
- Bearing damage may indicate corrosion, contamination, electrical discharge, or lack of lubrication.
- Plastic deformation between rollers and raceways may indicate overloads.
- Gear failure often follows bearing failure.

Document Observations. Identify and mark each component (including gear teeth and bearing components) so it is clearly identified in written descriptions, sketches, and photographs. It is especially important to mark all bearings, including inboard and outboard sides, so their locations and positions in the gearbox are identified.

Describe components consistently. For example, always start with the same part of a bearing and progress through the parts in the same sequence. This helps to avoid over-looking any evidence.

Describe important observations in writing using sketches and photographs where needed. The following guidelines help maximize the chances for obtaining meaningful evidence:

- *Concentrate on collecting evidence, not on determining cause of failure.* Regardless of how obvious the cause may appear, do not form conclusions until all evidence is considered.
- *Document what is visible.* List all observations even if some seem insignificant or if the failure mode is not easily recognizable. Remember there is a reason for everything, and some details may become important later when all the evidence is considered.
- *Document what is not visible.* This step is helpful to eliminate certain failure modes and causes. For example, if there is no scuffing, it can be concluded that gear tooth contact temperature was less than the scuffing temperature of the lubricant.
- *Search the bottom of the gearbox.* Often, this is where the best-preserved evidence is found, such as when a tooth fractures and falls free without secondary damage.
- *Use time efficiently.* Be prepared for the inspection. Plan work carefully to obtain as much evidence as possible. Do not be distracted by anyone.
- *Control the investigation.* Watch every step of the disassembly. Do not let the technician proceed too quickly. Disassembly should stop while the analyst inspects and documents the condition of a component; then move on to the next component.
- *Insist on privacy.* Do not let anyone distract attention from the investigation. If asked about conclusions, answer that conclusions are not formed until the investigation is complete.

Gather Gear Geometry. The load capacity of the gears should be calculated. For this purpose, obtain the following geometry data from the gears and housing or drawings:

- Number of teeth
- Outside diameter
- Face width
- Gear housing center distance
- Whole depth of teeth
- Tooth thickness (both span and topland thickness)

Specimens for Laboratory Tests. During inspection, the analyst will begin to formulate hypotheses regarding the cause of failure. With these hypotheses in mind, select specimens for laboratory testing. Take broken parts for laboratory evaluation or, if this is not possible, preserve them for later analysis.

After the inspection is completed, be sure all parts are coated with oil and stored properly so that corrosion or damage will not occur.

Oil samples can be very helpful. However, an effective analysis depends on how well the sample represents the operating lubricant. To take samples from the gearbox drain valve, first discard stagnant oil from the valve. Then, take a sample at the start, middle, and end of the drain to avoid stratification. To sample from the storage drum or reservoir, draw samples from the top, middle, and near the bottom. These samples can uncover problems such as excessive water in the oil due to improper storage.

Ask whether there are new, unused components. These parts are helpful to compare with failed parts. Similarly, compare a sample of fresh lubricant to used lubricant.

Before leaving the site, be sure that all necessary items—completed inspection forms, written descriptions and sketches, photos, and test specimens—have been collected.

It is best to devote two days minimum for the failure inspection. This affords time after the first-day inspection to review the inspection and analyze collected data. Often, the initial inspection discloses a need for other data, which can be gathered on the second day.

Determine Failure Mode

Now it is time to examine all information and determine how the gears failed. As described earlier in this chapter, several failure modes may be present and an understanding of these modes—bending fatigue, contact fatigue, wear, scuffing, impact, and so on—will enable identification of the cause of failure.

Tests and Calculations. In many cases, failed parts and inspection data do not yield enough information to determine the cause of failure. When this happens, gear design calculations and laboratory tests are necessary to develop and confirm a hypothesis for the probable cause.

Gear Design Calculations. Gear geometry data aids in estimating tooth contact stress, bending stress, lubricant film thickness, and gear tooth contact temperature based on transmitted loads. Calculate values according to AGMA standards such as ANSI/AGMA 2001. Compare calculated values with AGMA allowable values to help determine risks of micropitting, macropitting, bending fatigue, and scuffing.

Laboratory Examination and Tests. Microscopic examination may confirm the failure mode or find the origin of a fatigue crack. Light microscopes and scanning electron microscopes (SEM) are useful for this purpose. A SEM with energy-dispersive x-ray is especially useful for identifying corrosion, contamination, or inclusions.

If the primary failure mode is likely to be influenced by gear geometry or metallurgical properties, check for any geometric or metallurgical defects that may have contributed to the failure. For example, if tooth contact patterns indicate misalignment or interference, inspect the gear for accuracy on gear inspection machines. Conversely, where contact patterns indicate good alignment and loads are within rated gear capacity, check teeth for metallurgical defects.

Conduct nondestructive tests before any destructive tests. These nondestructive tests, which aid in detecting material or manufacturing defects and provide rating information, include:

- Surface hardness and roughness
- Magnetic particle inspection
- Acid etch inspection
- Gear tooth accuracy inspection

Then, conduct destructive tests to evaluate material and heat treatment. These tests include:

- Microhardness survey
- Microstructural determination using acid etches
- Determination of grain size
- Determination of nonmetallic inclusions
- Scanning electron microscopy to study fracture surfaces

Form and Test Conclusions

When all calculations and tests are completed, the analyst should form one or more hypotheses for the probable cause of failure, then determine whether the evidence supports or disproves the hypotheses. Evaluate all evidence that was gathered, including:

- Documentary evidence and service history
- Statements from witnesses
- Written descriptions, sketches, and photos
- Gear geometry and contact patterns
- Gear design calculations
- Laboratory data for materials and lubricant

Results of this evaluation may make it necessary to modify or abandon initial hypotheses or pursue new lines of investigation.

Finally, after thoroughly testing the hypotheses against the evidence, a conclusion will be reached regarding the most probable cause of failure. In addition, secondary factors that contributed to the failure may be identified.

Report Results

The failure analysis report should describe all relevant facts found during analysis, inspections and tests, weighing of evidence, conclusions, and recommendations. Present data succinctly, preferably in tables or figures. Good photos are especially helpful for portraying failure characteristics.

If possible, include recommendations for repairing equipment or making changes in equipment design, manufacturing, or operation to prevent future failures.

Examples of Gear Failure Analysis

No failure examination has been successfully completed until an evaluation of the results is made and it is only when the failure mechanism is understood that effective corrective measures can be devised. The purpose now is to take the known facts of a specific failure, to place them in a systematic context to arrive at a logical conclusion, and to indicate a corrective measure. Obviously, one cannot detail the hundreds of examinations available, but a chosen few will be presented to illustrate a systematic approach to the examination, regardless of its apparent simplicity or complexity.

Example 1: Catastrophic Failure of a Hoist Worm Caused by Destructive Wear Resulting From Abusive Operation. A farm-silo hoist was used as the power source for a homemade barn elevator. The hoist mechanism consisted of a pulley attached by a shaft to a worm that, in turn, engaged and drove a worm gear mounted directly on the hoist drum shaft. The worm was made of leaded cold-drawn 1113 steel with a hardness of 80 to 90 HRB; the worm gear was made of class 35–40 gray iron that had been nitrided in an aerated salt bath. Driving power was applied to the pulley, which actuated the worm and worm gear and rotated the drum to which the elevator cable was attached. After less than 1 year of service, the hoist failed catastrophically from destructive wear of the worm (Fig. 44). The hoist was rated at 905 kg (2000 lb) at a 200-rpm input. It was determined that at the time of failure the load on the hoist was only 325 kg (720 lb).

Investigation. When examined, the gearbox was found to contain fragments of the worm teeth and shavings that resembled steel wool. It was determined that both fragments and shavings were steel of the same composition as the worm, ruling out the possibility that these materials may have derived from a source other than the gearbox. The teeth of the worm had worn to a thickness of approximately 1.6 mm ($\frac{1}{16}$ in.) from an original thickness of 3.2 mm ($\frac{1}{8}$ in.). More than half of the worm teeth had been sheared off.

Further investigation revealed that the drive pulley had been replaced with a pulley of different diameter and that the pulley was rotating at 975 rpm instead of at the rated maximum of 200 rpm. By calculation, with a load of 325 kg (720 lb), the gearbox was forced to develop 0.24 kW (0.32 hp)—almost twice the specified rating of 0.13 kW (0.18 hp) (2000 lb at 200 rpm).

Conclusions. The failure mode was adhesive wear of the steel worm, which resulted from operation of the gearbox at a power level that far exceeded the maximum specified by the gearbox manufacturer. The cause of failure was misuse in the field.

Example 2: Failure of Carburized Steel Impeller Drive Gears Due to Pitting and a Wear Pattern. Two intermediate impeller drive gears were submitted for metallurgical examination when they exhibited evidence of pitting and abnormal wear after production tests in test-stand engines. Both gears were made of AMS 6263 steel and were gas carburized, hardened, and tempered. Figure 45 shows the pitting and the wear pattern observed on the teeth of gear 1.

According to the heat-treating specification, the gears were required to satisfy the following requirements: (1) a carburized case depth of 0.4 to 0.6 mm (0.015 to 0.025 in.), (2) a case hardness of 77 to 80 HR30N, and (3) a core hardness of 36 to 44 HRC.

Investigation. Sections of both gears were removed with a cutoff wheel and examined for hardness, case depth, and microstructure of case and core. Results were as follows:

	Gear 1	Gear 2
Hardness		
Case, HR30N	70–73	77–78
Core, HRC	40	40
Case depth, mm (in.)	0.5–0.6 (0.02–0.024)	0.6–0.7 (0.024–0.028)
Microstructure		
Case	Lean carbon (<0.85% C)	Normal
Core	Normal	Normal

Both the pitting and the wear pattern were more severe on gear 1 than on gear 2.

Conclusions. Gear 1 failed by surface-contact fatigue (pitting) because the carbon content of the carburized case and consequently the case hardness were below specification and inadequate for the loads to which the gear was subjected. The pitting and wear pattern observed on gear 2 were relatively mild because case-carbon content and case hardness were within specified requirements. Nevertheless, it was evident that case depth, as specified, was not adequate for this application—even with case-carbon content and hardness being acceptable.

Recommendations. It was recommended that the depth of the carburized case on impeller drive gears be increased from 0.4 to 0.6 mm (0.015 to 0.025 in.) to 0.6 to 0.9 mm (0.025 to 0.035 in.), an increase that would still allow a minimum of 30% core material across the tooth section at the pitchline. This increase would ensure improved load-carrying potential and wear resistance. It was also recommended that hardness readings on gears be made using a 60 kg (132 lb) load (Rockwell A scale) and that the minimum case-hardness requirement be set at 81 HRA, an additional safeguard to ensure adequate load-carrying capacity.

Example 3: Failure of a Carburized Steel Generator-Drive Idler Gear by Pitting Due to Decarburization and Subsurface Oxidation. Following test-stand engine testing, an idler gear for the generator drive of an aircraft engine exhibited evidence of destructive pitting on the gear teeth in the area of the pitchline; as a consequence, the gear was submitted for metallurgical examination. The gear was made of AMS 6263 steel and was gas carburized to produce a case 0.4 to 0.6 mm (0.015 to 0.025 in.) deep.

Investigation. Sections of the gear were removed and examined for hardness, case depth, and microstructure of case and core. The results of the examination and the specified requirements were:

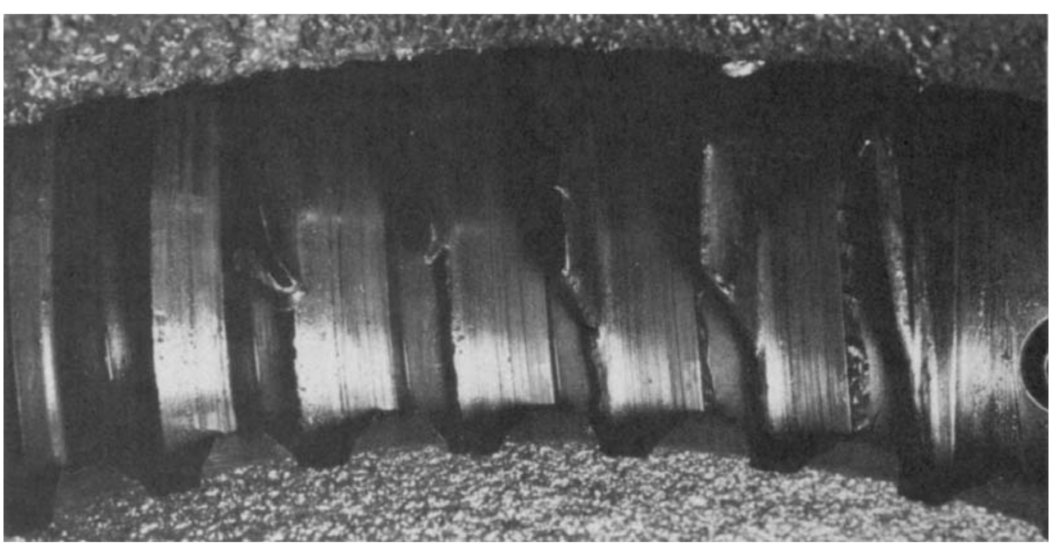


Fig. 44 Destructive wear of an 1113 steel worm used in a silo hoist

	Required	Actual
Case depth, mm (in.)	0.4–0.6 (0.015–0.025)	0.6–0.7 (0.024–0.028)
Hardness		
Case, HR30N	77–80	74–76 (HR15N88–89)
Core, HRC	36–44	40–41

When it was determined that the case hardness of the gear was below specification, additional hardness measurements were made using a 15 kg (33 lb) load. These measurements confirmed the low hardness and suggested that it had resulted from one or more surface defects.

A specimen from the pitted area of one gear tooth was prepared for metallographic examination. The results of this examination are shown in Fig. 46. When the specimen was etched in 2% nital for 15 to 20 s, a decarburized surface layer was observed in the vicinity of pitting (Fig. 46a). When the specimen was repolished and then etched in 2% nital for only 3 s, a heavy subsurface layer of oxide scale was observed in the vicinity of pitting (Fig. 46b).

Conclusions. Based on low case-hardness readings and the results of metallographic examination, it was concluded that pitting had resulted due to a combination of surface decarburization and subsurface oxidation. Because of previous failure analyses and thorough investigation of heat-treating facilities, the source of these defects was well known. The oxide layer had been developed during the carburizing cycle because the furnace retort had not been adequately purged of air before carburization. Decarburization had occurred during the austenitizing cycle in a hardening furnace containing an exothermic protective atmosphere because of defects in the copper plating applied to the gear for its protection during austenitizing. These defects had permitted leakage of exothermic atmosphere to portions of the carburized surface, and, because of the low carbon potential of the exothermic atmosphere, the leakage had resulted in decarburization.

Recommendations. It was recommended that steps be taken during heat treatment to ensure that furnaces are thoroughly purged before carburizing and that positive atmospheric pressure be maintained throughout the carburizing cycle. Further, it was recommended that more effective control be exercised over all aspects of copper plating of carburized gears, including final inspection, before releasing the gears for hardening.

Example 4: Fatigue Failure of a Carburized 4817 Steel Spiral Bevel Gear at Acute-Angle Intersections of Mounting Holes and Tooth-Root Fillets. A spiral bevel gear set was returned to the manufacturer because the gear

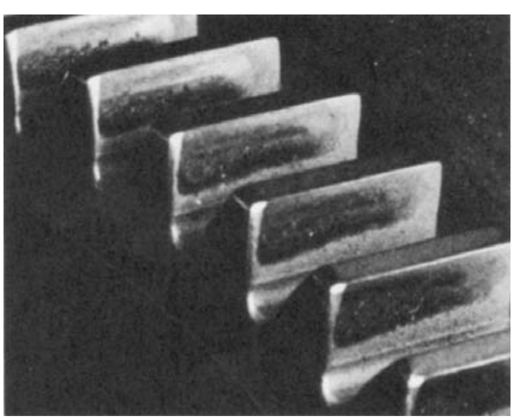


Fig. 45 Pitting and wear pattern on a carburized AMS 6263 steel impeller drive gear. Original magnification approximately 2.3×

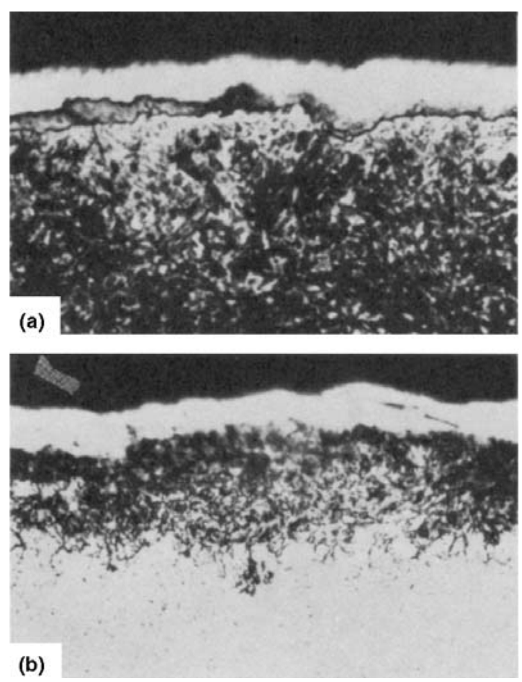


Fig. 46 Metallurgical causes of destructive pitting that occurred in a carburized AMS 6263 steel gear. (a) Specimen etched 15 to 20 s in 2% nital showing a surface layer of decarburized material. Original magnification at 500×. (b) Same specimen repolished and etched 3 s in 2% nital showing a heavy subsurface layer of oxide scale. Original magnification at 500×. The white band immediately above the decarburized and oxidized layers is electrodeposited nickel that was applied to prevent edge-rounding during polishing.

broke into three pieces after about 2 years of service. Damage to the pinion was minor. The gear broke along the root of a tooth intersected by three of the six 22 mm (0.875 in.) diam holes (Fig. 47) used to mount the gear to a hub. The intersection of the hole and root fillet created an acute-angle condition (Section A-A, Fig. 47).

The ring gear, machined from a forged blank made of 4817 steel, had been gas carburized at 925 °C (1700 °F), cooled to 815 °C (1500 °F), press quenched in oil at 60 °C (140 °F), then tempered at 175 °C (350 °F) for 1½ h to produce a case hardness of 61 to 62 HRC, as measured at a tooth land.

Investigation. Examination of the gear revealed fatigue progression for about 6.4 mm (¼ in.) at acute-angle intersections of three mounting holes with the root fillets of three teeth. Subsequent fracture progressed rapidly until the entire section failed. Cracking between the holes and the bore of the gear started before the tooth section failed. Magnetic-particle inspection revealed cracks at the intersections of the remaining three mounting holes and the adjacent tooth-root fillets.

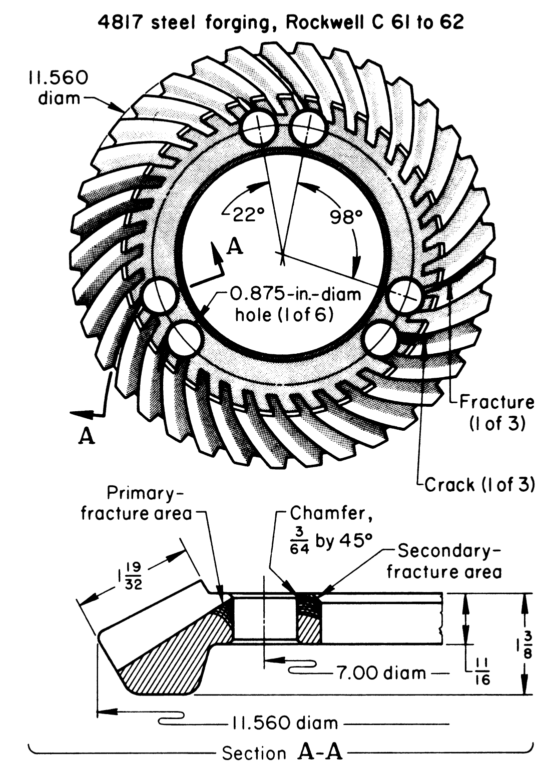


Fig. 47 Carburized 4817 steel spiral bevel gear failure. The gear broke from fatigue at acute-angle intersections of mounting holes and tooth-root fillets as a result of through hardening. Dimensions given in inches

Microstructure of the carburized and hardened case was fine acicular martensite with about 5% retained austenite. The core was composed principally of low-carbon martensite.

Metallographic examination revealed that the acute-angle intersections of the mounting holes and tooth-root fillets were carburized to a depth that comprised the entire cross section and consequently were through hardened.

Conclusions. The gear failed in fatigue, slowly to a depth of about 6.4 mm (¼ in.), then rapidly until final fracture. Design of the gear and placement of the mounting holes were contributing factors in the failure because they resulted in through hardening at the acute-angle intersections of the mounting holes and tooth-root fillets.

Example 5: A Spiral Bevel Pinion That Had Two Failure Modes Occurring Independently of Each Other. This spiral bevel set was the primary drive unit for the differential and axle shafts of an exceptionally large front-end loader in the experimental stages of development.

Visual Examination. The tooth contact area of the pinion appeared to have been heavy at the toe end and low on the profile of the concave (drive) side. The entire active profile showed surface rippling for more than half the length of the tooth from the toe end (Fig. 48). Pitting originated at the lower edge of the profile 50 mm (2 in.) from the toe end of the concave side (Fig. 48a) and subsequently progressed in both directions along the lower edge of the active profile and upward toward the top of the tooth, changing the mode from pitting to spalling (Fig. 48b). Both photographs show evidence of a double contact: the original setting, and a secondary pattern with the pinion moved out and away from the center of the gear. The convex (reverse) sides were slightly worn and scuffed high over the profile.

All of the gear teeth were intact and highly polished along the top of the profile toward the toe end of the convex (forward) side. Some scuffing was evident low on the profile of the concave sides.

Physical Examination. Magnetic-particle inspection showed no evidence of tooth-bending fatigue on either part. The several cracks indicated are associated with the spalling surfaces on the concave sides of the pinion teeth. The gear teeth showed no crack indications.

Tooth Characteristics. The original setting test charts taken on this set in December 1979 are on file and document a very good and normal tooth contact pattern. The secondary tooth pattern that prevails at this time could not be duplicated on the gear tester. It appears that the pinion had moved away from the center of the gear, thus activating a very heavy low-profile contact toward the toe end.

Surface-Hardness Tests. The surface hardness of both parts was 58 to 59 HRC, which is within the specification of 58 to 63 HRC.

Metallurgical Examination. The procedures to implement are determined as follows. The rippling (Fig. 48) was extensive and covered the entire area of the initial contact pattern. Rippling is a reflection of apparent movement or of a tendency to move. It can be a superficial adjustment of a surface being heavily rolled while sliding, or it can be an adjustment of the surface during absorption of energy by retained austenite. Rippling is not always associated with a failure, nor is it detrimental in itself. It does usually indicate, however, a rolling-sliding condition that is highly compressive.

The pitting originated in a localized area, and the spalling appeared to be a contiguous event, although this type of spalling could easily have been found as an independent mode without the association of surface pitting. It was, therefore, necessary to section the pinion tooth normal to the active profile near the spalled area shown in Fig. 48(b) and to prepare this section for a microhardness traverse of the case and for a microscopic examination.

Microscopic Examination. The core structure of the pinion showed an equal admixture of low-carbon martensite and lamellar pearlite. The case structure consisted of very fine acicular martensite retaining less than 5% austenite.

The pitted area low on the active profile showed evidence of a heavy metal-to-metal sliding action which tended to produce adhesion. The central profile area, being subjected to spalling, had a surface that was relatively unaltered (Fig. 49). In the same area, there was a subsurface display of butterfly wings reaching to a depth of 0.7 mm (0.027 in.) from the surface.

Case-Hardness Traverse. Figure 50 shows the case-hardness traverse of the section used for Fig. 49. The results are self-explanatory.

Conclusions. The material and process specifications were met satisfactorily. The primary mode of failure was rolling-contact fatigue of the concave (drive) active tooth profile. The spalled area was a consequence of this action. The pitting low on the profile appeared to have originated after the shift of the pinion tooth away from the gear center. The shift of the pinion is most often due to a bearing displacement or malfunction. The cause of this failure appeared to be continuous high overload, which may also have contributed to the bearing displacement.

Example 6: Fatigue Failure of a Carburized Steel Bevel Pinion Because of Misalignment. The bevel pinion shown in Fig. 51(a) was part of a drive unit in an edging mill. The pinion had been in service about 3 months when several teeth failed. Specifications required that the pinion be made from a 2317 steel forging

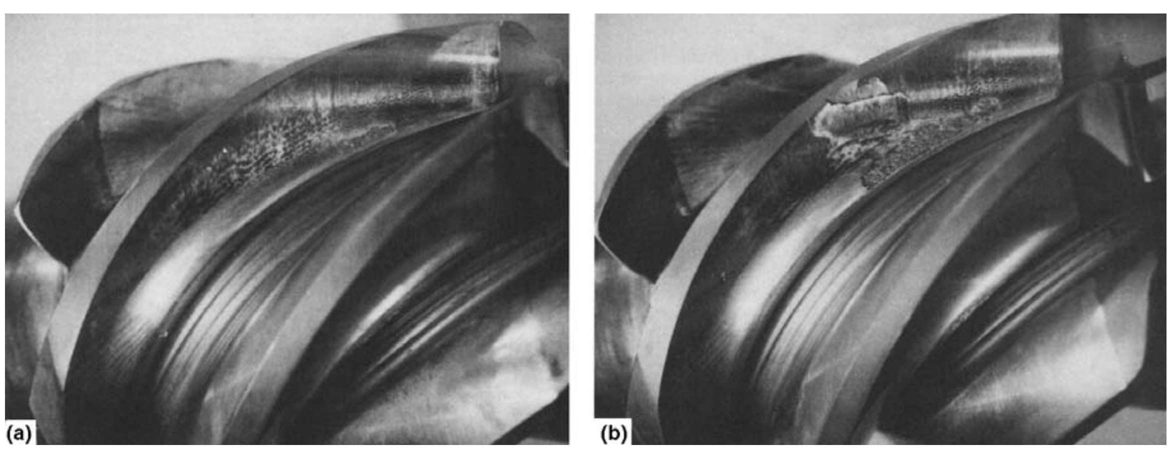


Fig. 48 Spiral bevel pinion of 4820H steel. (a) Rippled surface for three-fourths of the length of the tooth starting from the toe end. Pitting originated low on the active profile 50 mm (2 in.) from the toe end. (b) The pitting area extended in both directions and broadened. The central profile has an area of spalling that appears to be contiguous with the pitting. The microstructure indicates that the two modes occurred independently. Original magnification at 0.5×. See also Fig. 49 and 50.

and that the teeth be carburized and hardened to a case hardness of 56 HRC and a core hardness of 250 HB (24.5 HRC).

Investigation. Chemical analysis of the metal in the pinion showed that it was 2317 steel as specified. Case hardness was 52 to 54 HRC, and core hardness was 229 HB (20.5 HRC).

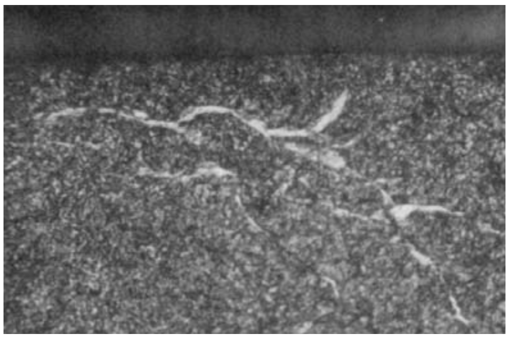


Fig. 49 Section normal to surface of tooth profile taken near the spalled area shown in Fig. 48(b). The surface shows no catastrophic movement; the butterfly wings are generally parallel to the surface, but extend 0.7 mm (0.027 in.) below the surface. Microstructure is very fine acicular martensite retaining less than 5% austenite. Original magnification at 200×

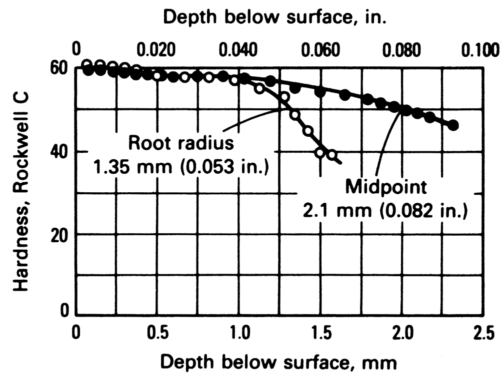


Fig. 50 Case-hardness traverse of section used for Fig. 49 taken from tooth shown in Fig. 48(b)

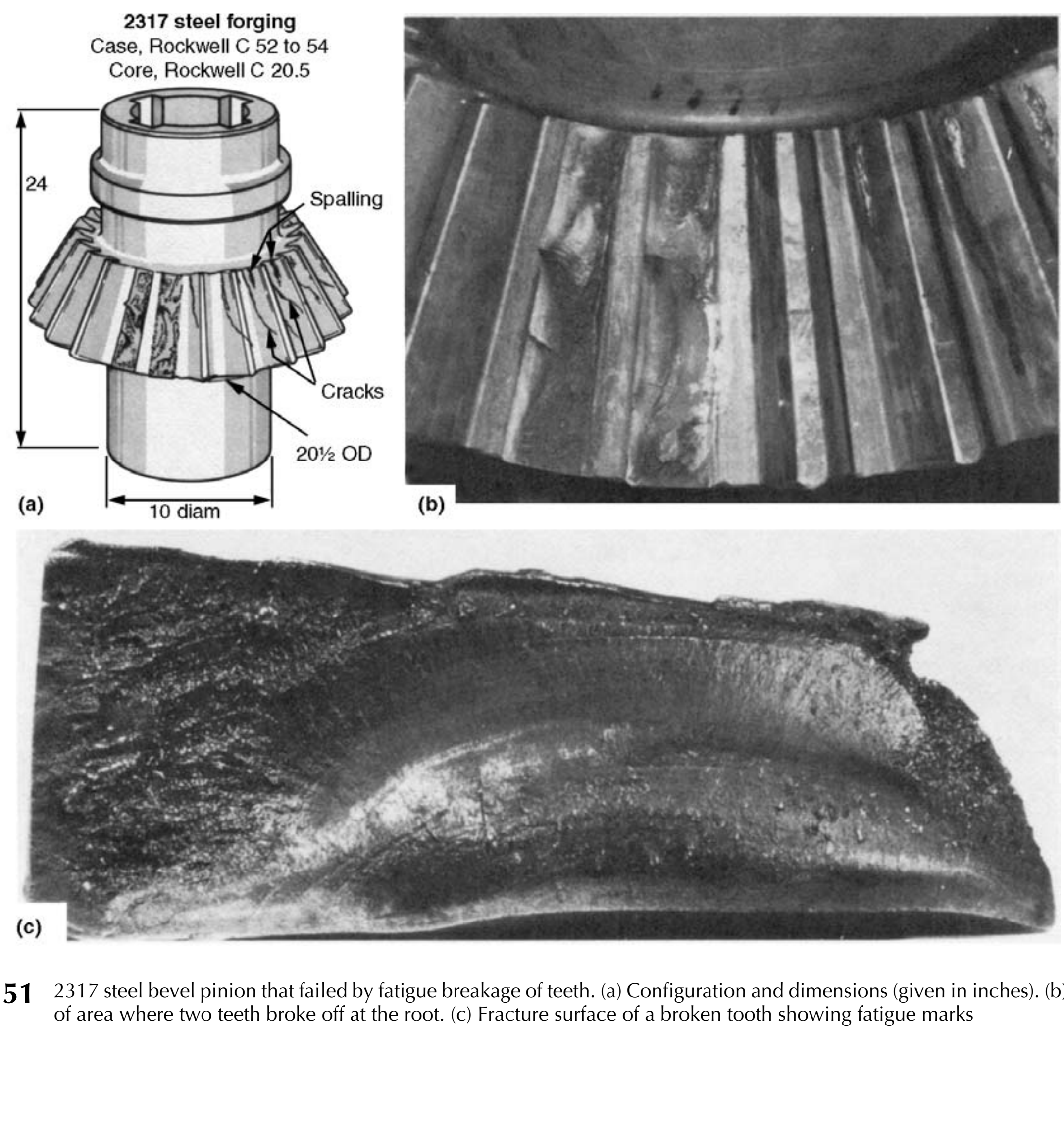


Fig. 51 2317 steel bevel pinion that failed by fatigue breakage of teeth. (a) Configuration and dimensions (given in inches). (b) View of area where two teeth broke off at the root. (c) Fracture surface of a broken tooth showing fatigue marks

Both hardness values were slightly lower than specified, but acceptable.

Visual inspection of the pinion showed that two teeth had broken at the root (Fig. 51b). The surfaces of these fractures exhibited fatigue marks extending across almost the entire tooth (Fig. 51c). Magnetic-particle inspection of the pinion showed that all of the teeth were cracked. Each crack had initiated along the tooth root, at the toe (small) end of the tooth, extending to the center of the crown. Spalling was also noted on the pressure (drive) side of each tooth at the toe end. A metallographic specimen taken transversely through a broken tooth showed it to be case carburized to a depth of 4.8 mm (3⁄16 in.), hardened, and tempered.

Conclusions. The pinion failed by tooth-bending fatigue. Some mechanical misalignment of the pinion with the mating gear caused a cyclic shock load to be applied to the toe ends of the teeth, as exhibited by the spalling. This continuous pounding caused the teeth to crack at the roots and to break off.

Example 7: Overload Failure of a Bronze Worm Gear. Overload failures, from the perspective of the materials failure analyst, refer to the ductile or brittle failure of a material when stresses exceed the load-bearing capacity of the material from either excessive applied stress or degradation of the load-bearing capacity of the material from damage, embrittlement, or other factors (Ref 8). Ductile overload failures are simply those that exhibit significant visible macroscopic plastic deformation. Brittle overload failures, in contrast to ductile overload failures, are characterized by little or no plastic deformation. For ductile and brittle fractures, distinguishing characteristics at different scales of observations (1× to >10,000×) are briefly summarized in Table 6. In this example, a mixed-mode failure (ductile plus brittle fracture) is described.

Background. A very large diameter worm gear that had been in service in a dam for more than 60 years exhibited cracks and was removed. It was reported that the cast bronze gear was only rarely stressed during service, associated with infrequent opening and closing of gates. Due to the age of the gear and the time frame of its manufacture, no original material specifications or strength requirements could be located. Likewise, no maintenance records of possible repairs to the gear were available.

Investigation. As part of the investigation, a number of representative sections excised from the gear were submitted for metallurgical analysis. The cracks were primarily located on the gear faces and were not associated with traditional regions of high service stresses, such as the tooth roots. Some portions of the gear face exhibited light discoloration, which was suggestive of weld repairs. The cracks were evident within the base metal, at the repair weld fusion lines, and within the weld metal. Figure 52(a) shows the opened crack features in an area where a weld and remaining casting imperfection were apparent. The macroscopic fracture features differed substantially between the base metal and the welds. Some of the cracking did not occur at casting imperfections or repair welds.

Scanning electron microscopy revealed the typical weld and base metal fracture features shown in Fig. 52(b) and (c), respectively. The weld exhibited fine features that consisted of both dimple rupture and grain-boundary dimple-rupture evidence. The base metal exhibited rather large-grain intergranular brittle cracking features with many types of inclusions.

Chemical analysis revealed a standard high-strength, manganese bronze composition, and the weld filler metal was considered compatible. Tension testing of specimens removed from gear segments remote to repairs revealed tensile strengths near 690 MPa (100 ksi), yield strengths near 517 MPa (75 ksi), and 5% elongation at fracture. Weld and base metal hardnesses were similar.

Table 6 Distinguishing characteristics of brittle versus ductile behavior depending on the scale of observation

Scale of observation	Brittle	Ductile
Structural engineer	Applied stress at failure is less than the yield stress	Applied stress at failure is greater than the yield stress
By eye (1×)	No necking, shiny facets, crystalline, granular	Necked, fibrous, woody
Macroscale (<50×)	"Low" RA or ductility	Medium to high RA
Microscale, scanning electron microscopy (100–10,000×)	Brittle microprocess, cleavage or intergranular	Ductile microprocess, microvoid coalescence
Transmission electron microscopy (>10,000×)	May have a large level of local plasticity	High amount of plasticity globally

RA, reduction of area. Source: Ref 8

The microstructure at the transition between the base metal and weld metal is shown in Fig. 52(d). The continuous grain-boundary phase in the weld metal responsible for the decohesive rupture features is evident.

Conclusions. It was concluded from the investigation that the bronze gear cracked via mixed-mode overload, rather than by a progressive mechanism such as fatigue or stress-corrosion cracking (SSC). The cracking was not associated with regions that would be highly stressed and did not appear to be consistently correlated to casting imperfections, repair welds, or the associated heat-affected zones. The cast gear was a high-strength, low-ductility bronze. Cracking across the gear face suggested bending forces from misalignment were likely responsible for the cracking, and further review of this potential root cause could be recommended.

ACKNOWLEDGMENTS

This chapter was adapted from:

- L.E. Alban, Failures of Gears, *Failure Analysis and Prevention,* Vol 11, *Metals Handbook,* 9th ed., American Society for Metals, 1986, p 586–601
- R. Errichello, Tutorial: How to Analyze Gear Failures, *Practical Failure Analysis,* Vol 2 (No. 6), December 2002, p 8–16
- R.S. Hyde, Contact Fatigue of Hardened Steel, *Fatigue and Fracture,* Vol 19, *ASM Handbook,* ASM International, 1996, p 691–703
- B.A. Miller, Overload Failures, *Failure Analysis and Prevention,* Vol 11, *ASM Handbook,* 2002, p 671–699
- D.R. McPherson and S.B. Rao, Mechanical Testing of Gears, *Mechanical Testing and*

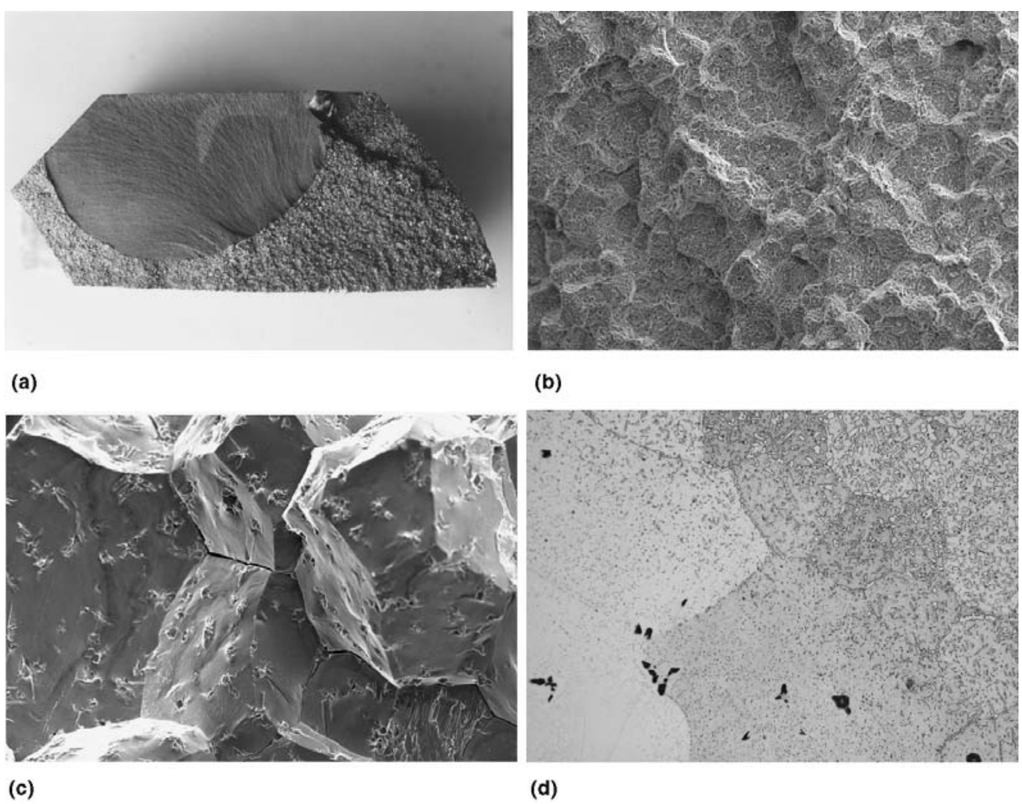


Fig. 52 Overload failure of a bronze worm gear (example 4). (a) An opened crack is shown with a repair weld, a remaining casting flaw, and cracking in the base metal. (b) Electron image of decohesive rupture in the fine-grain weld metal. Scanning electron micrograph. Original magnification at 119×. (c) Morphology in the large-grain base material at the same magnification as (b), showing intergranular brittle fracture features. Scanning electron micrograph. Original magnification at 119×. (d) Metallographic image showing the weak grain-boundary phase in the weld. Potassium dichromate etch. Original magnification at 297×

Evaluation, Vol 8, *ASM Handbook,* ASM International, 2000, p 861–872

REFERENCES

1. L.E. Alban, *Systematic Analysis of Gear Failures,* American Society for Metals, 1985
2. *Appearance of Gear Teeth—Terminology of Wear and Failure,* ANSI/AGMA 1010-E95, American Gear Manufacturers Association, 1995
3. R. Errichello, Friction, Lubrication, and Wear of Gears, *Friction, Lubrication, and Wear Technology,* Vol 18, *ASM Handbook,* ASM International, 1992, p 535–545
4. D.R. McPherson and S.B. Rao, Mechanical Testing of Gears, *Mechanical Testing and Evaluation,* Vol 8, *ASM Handbook,* ASM International, 2000, p 861–872
5. R.S. Hyde, Contact Fatigue of Hardened Steel, *Fatigue and Fracture,* Vol 19, *ASM Handbook,* ASM International, 1996, p 691–703
6. W.R. Scott, Myths and Miracles of Gear Coatings, *Gear Technology,* July/Aug 1999, p 35–44
7. R. Errichello, Tutorial: How to Analyze Gear Failures, *Practical Failure Analysis,* Vol 2 (No. 6), December 2002, p 8–16
8. B.A. Miller, Overload Failures, *Failure Analysis and Prevention,* Vol 11, *ASM Handbook,* 2002, p 671–699

SELECTED REFERENCES

- R. Errichello, "Analysis Techniques End Gear Damage," *Power Transmission Design,* March 1995, 37(3), p 23–26
- R. Errichello and J. Muller, "How to Analyze Gear Failures," *Power Transmission Design,* March 1994, 36(3), p 35–40

Gear Materials, Properties, and Manufacture
J.R. Davis, editor, p293-309
DOI:10.1361/gmpm2005p293

CHAPTER 14

Fatigue and Life Prediction

GEARS can fail in many different ways, and except for an increase in noise level and vibration, there is often no indication of difficulty until total failure occurs. In general, each type of failure leaves characteristic clues on gear teeth, and detailed examination often yields enough information to establish the cause of failure. The general types of failure modes (in decreasing order of frequency) include fatigue, impact fracture, wear, and stress rupture (Table 1). The leading causes of failure appear to be tooth-bending fatigue, tooth-bending impact, and abrasive tooth wear.

This chapter summarizes the various kinds of gear wear and failure and how gear life in service is estimated. Hopefully, gears are properly designed, made of good material, and accurately machined so that the life of the gear is adequate for the service intended. The manner in which gears wear in service and the various kinds of failure that may occur determine how gear life in service is estimated. In addition, the kinds of flaws in material that may lead to premature gear failure are discussed. Gear life is influenced primarily by geometric accuracy, gear-tooth contact conditions, and material condition or flaws. Metallurgical quality is just as important as gear-tooth geometric accuracy, and both must be under good control by those designing, making, or inspecting gears.

Table 1 Failure modes of gears

Failure mode	Type of failure
Fatigue	Tooth bending, surface contact (pitting or spalling), rolling contact, thermal fatigue
Impact	Tooth bending, tooth shear, tooth chipping, case crushing, torsional shear
Wear	Abrasive, adhesive
Stress rupture	Internal, external

In an analysis of more than 1500 studies, the three most common failure modes, which together account for more than half the failures studied, are tooth-bending fatigue, tooth-bending impact, and abrasive tooth wear. Source: *Metals Handbook,* 9th ed., Vol 11, *Failure Analysis and Prevention,* ASM International, 1986

Gear Tooth Contact

The way in which tooth surfaces of properly aligned gears make contact with each other is responsible for the heavy loads that gears are able to carry. In theory, gear teeth make contact along lines or at points; in service, however, because of elastic deformation of the surfaces of loaded gear teeth, contact occurs along narrow bands or in small areas. The radius of curvature of the tooth profile has an effect on the amount of deformation and on the width of the resulting contact bands. Depending on gear size and loading, the width of the contact bands varies from about 0.38 mm (0.015 in.) for small, lightly loaded gears to about 5 mm (0.2 in.) for large, heavily loaded gears.

Gear tooth surfaces are not continuously active. Each part of the tooth surface is in action for only short periods of time. This continual shifting of the load to new areas of cool metal and cool oil makes it possible to load gear surfaces to stresses approaching the critical limit of the gear metal without failure of the lubricating film.

The maximum load that can be carried by gear teeth also depends on the velocity of sliding between the surfaces, because the heat generated varies with rate of sliding as well as with pressure. If both pressure and sliding speeds are excessive, the frictional heat developed can cause destruction of tooth surfaces. This pressure-velocity factor, therefore, has a critical influence on the probability of galling and scoring of gear teeth. The permissible value of this critical factor is influenced by gear metal, gear design, character of lubricant, and method of lubricant application.

Lubrication is accomplished on gear teeth by the formation of two types of oil films. The reaction film, also known as the boundary lubricant, is produced by physical adsorption and/or chemical reaction to form a desired film that is soft and easily sheared but difficult to penetrate or remove from the surface. The elastohydrodynamic film forms dynamically on the gear tooth surface as a function of the surface speed. This secondary film is very thin, has a very high shear strength, and is only slightly affected by compressive loads as long as constant temperature is maintained.

Certain rules about a lubricant should be remembered in designing gearing and analyzing failures of gears:

- Load is transferred from a gear tooth to its mating tooth through a pressurized oil film. If not, metal-to-metal contact may be detrimental.
- Increasing oil viscosity results in a thicker oil film (keeping load, speed, and temperature constant).
- Heat generation cannot be controlled above a certain maximum viscosity (for a given oil).
- Breakdown of the oil film will occur when the gear tooth surface-equilibrium temperature has reached a specific value.

The scuffing load limit of mating tooth surfaces is speed dependent. With increasing speed, the load required to be supported by the reaction film decreases, while the load that can be supported by the increasing elastohydrodynamic film increases. The result is a decreasing scuffing load limit to a certain speed as the reaction film decreases; then, as the speed picks up to where the elastohydrodynamic film increases, the scuffing load limit increases. This allows an increase in overall load-carrying capacity (assuming no change in temperature that would change viscosity).

- At constant speed, surface-equilibrium temperature increases as load increases, which lowers the scuffing load limit of the reaction film (surface-equilibrium temperature is attained when the heat dissipated from the oil is equal to the heat extracted by the oil).

Damage to and failures of gears can and do occur as a direct or indirect result of lubrication problems.

Spur and Bevel Gears. Spur gear teeth are cut straight across the face of the gear blank, and the mating teeth theoretically meet at a line of contact (Fig. 1a) parallel to the shaft. Straight teeth of bevel gears also make contact along a line (Fig. 1b) that, if extended, would pass through the point of intersection of the two shaft axes. As teeth on either spur or bevel gears pass through mesh, the line of contact sweeps across the face of each tooth. On the driving tooth, it starts at the bottom and finishes at the tip. On the driven tooth, the line of contact starts at the tip and finishes at the bottom.

Helical, Spiral Bevel, and Hypoid Gears. Gear tooth contact on helical, spiral bevel, and hypoid gears is similar to that developed on a stepped spur gear (Fig. 2). Each section, or lamination, of the spur gear makes contact with its mating gear along a straight line; each line, because of the offset between sections, is slightly in advance of its adjacent predecessor. When innumerable laminations are combined into a smoothly twisted tooth, the short individual lines of contact blend into a smoothly slanted line (Fig. 1c) that extends from one side of the tooth face to the other and sweeps either upward or downward as the tooth passes through mesh. This slanted-line contact occurs between the teeth of helical gears on parallel shafts, spiral bevel gears, and hypoid gears.

The load pattern is a line contact extending at a bias across the tooth profile, moving from one end of the contact area to the other end. Under load, this line assumes an elliptical shape and thus distributes the stress over a larger area. Also, the purpose of spiral bevel gearing is to relieve stress concentrations by having more than one tooth enmeshed at all times.

The greater the angle of the helix or spiral, the greater the number of teeth that mesh simultaneously and share the load. With increased angularity, the length of the slanted contact line on each tooth is shortened, and shorter but more steeply slanted lines of contact sweep across the faces of several teeth simultaneously. The total length of these lines of contact is greater than the length of the single line of contact between straight spur-gear teeth of the same width. Consequently, the load on these gears is distributed not only over more than one tooth but also over a greater total length of line of contact. On the other hand, the increased angularity of the teeth increases the axial thrust load and thus increases the loading on each tooth. These two factors counterbalance each other; therefore, if the power transmitted is the same, the average unit loading remains about the same.

Helical gears on crossed shafts make tooth contact only at a point. As the teeth pass through mesh, this point of contact advances from below the pitchline of the driving tooth diagonally across the face of the tooth to its top, and from the top of the driven tooth diagonally across its face to a point below the pitchline. Even with several teeth in mesh simultaneously, this point contact does not provide sufficient area to carry an appreciable load. For this reason, helical gears at angles are usually used to transmit motion where very little power is involved.

Worm Gears. In a single-enveloping worm gear set, in which the worm is cylindrical in shape, several teeth may be in mesh at the same time, but only one tooth at a time is fully engaged. The point (or points) of contact in this type of gear set constitutes too small an area to carry an appreciable load without destruction of the metal surface. As a result, single-enveloping worm gear sets are used in applications similar to those for helical gears on crossed shafts: to transmit motion where little power is involved.

Considerable power must be transmitted by commercial worm gear sets; therefore, the gears of these sets are throated to provide a greatly increased area of contact surface. The gear tooth theoretically makes contact with the worm thread along a line curved diagonally across the gear tooth. The exact curve and slant depend on tooth design and on the number of threads on the worm relative to the number of teeth on the gear. Usually, two or more threads of the worm are in mesh at the same time, and there is a separate line of contact on each meshing tooth. As meshing proceeds, these lines of contact move inward on the gear teeth and outward on the worm threads. To secure smooth operation from a gear of this type, the teeth of the gear and sometimes the threads of the worm are usually altered from theoretically correct standard tooth forms. These alterations result in slightly wider bands of contact, thus increasing the load-carrying capacity of the unit. Load-carrying capacity also depends on the number of teeth in simultaneous contact. Exact tooth design varies from one manufacturer to another, and for this reason, the patterns of contact also vary.

In a double-enveloping worm gear set, the worm is constructed so that it resembles an hourglass in profile. Such a worm partly envelopes the gear, and its threads engage the teeth of the gear throughout the entire length of the worm. The teeth of both the worm and the gear have

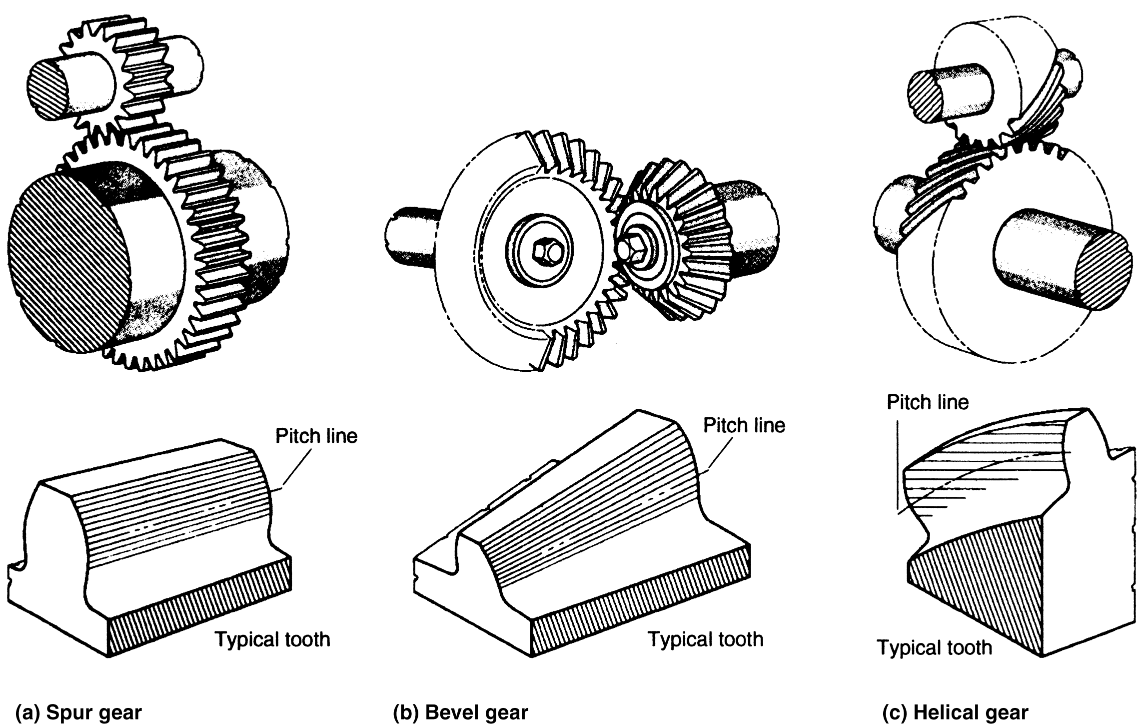


Fig. 1 Tooth contact lines on a spur gear (a), a bevel gear (b), and a low-angle helical gear (c). Lines on tooth faces of typical teeth are lines of contact.

straight-sided profiles like those of rack teeth, and in the central plane of the gear, they mesh fully along the entire length of the worm.

The exact pattern of contact in double-enveloping (or double-throated) worm gear sets is somewhat controversial and seems to vary with gear design and with method of gear manufacture. It is generally agreed, however, that contact is entirely by sliding with no rolling and that radial contact occurs simultaneously over the full depth of all the worm teeth.

Operating Loads. Gears and gear drives cover the range of power transmission from fractional-horsepower applications, such as hand tools and kitchen utensils, to applications involving thousands of horsepower, such as heavy machinery and marine drives. However, neither the horsepower rating nor the size of a gear is necessarily indicative of the severity of the loading it can withstand. For example, the severity of tooth loading in the gear train of a 186 W (¼ hp) hand drill may exceed that of the loading in a 15 MW (20,000 hp) marine drive. Factors other than horsepower rating and severity of loading can affect gear strength and durability, particularly duration of loading, operating speed, transient loading, and environmental factors such as temperature and lubrication.

Gear Tooth Surface Durability and Breakage

Metal gears are normally lubricated with an oil or a grease. Some small, nonmetallic gears have limited capability to run without lubrication, due to the self-lubricating properties of the material.

Scuffing. When metal gears are adequately lubricated with a clean lubricant, there is very little abrasive wear of the contacting tooth surfaces. However, the tooth surface may be quickly damaged by scuffing if (in combination with poor surface finish and low-viscosity lubricant) the load intensity, temperature, and rubbing speed are too high. Scuffing consists of small radial tears in the tooth surface. It is most apt to happen where the tip of one tooth is contacting the lower flank of a mating tooth (Fig. 3).

Pitting may occur after some millions or billions of tooth contacts. In general, pitting almost never occurs before 10,000 contacts. Pitting is a fatigue failure where small cracks form in the tooth surface and then grow to the point where small, round bits of metal break out of the tooth surface (Fig. 4).

Tooth breakage (such as in Fig. 5) is normally the situation where a crack starts in the root fillet, below the contacting surface, and then the crack grows such that a whole tooth breaks off in a cantilever-beam-type of failure. In wide-face gears, one end of the tooth may break off, leaving the tooth intact for the rest of the face width.

Traditionally, the gear designer first determines a pitch diameter and a face width for the pinion that are large enough for the pinion to last for the required service life with a probability of failure of no more than 1 in 100. This determination is based on a possible pitting fatigue failure. It is assumed in the beginning that the surface finish, the tooth accuracy, the

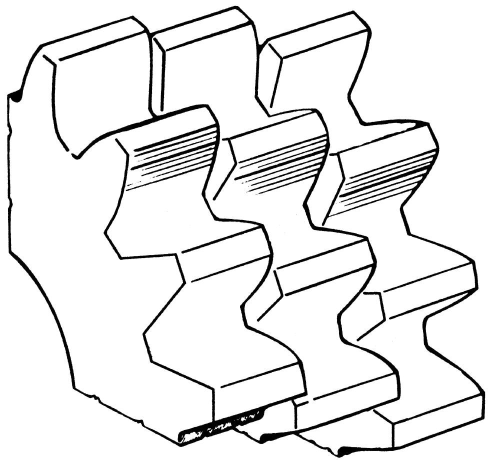


Fig. 2 Lines of contact on a stepped spur gear. The heavy line on a tooth face of each gear section represents the instantaneous line of contact for that section. This offset-contact pattern is typical for helical, spiral bevel, and hypoid gears. Lines on tooth faces are lines of contact.

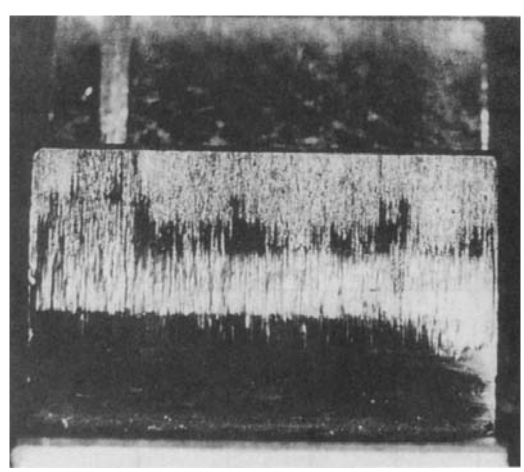


Fig. 3 An example of gear tooth scuffing. Note radial scratch lines.

lubrication, and the needed profile and helix modifications will all be carried out well enough to avoid any serious risk of scuffing.

Normally, the pinion is more apt to fail in pitting than the gear, so the sizing of the pinion tends to determine the needed size of the gear. (The gear pitch diameter equals the pinion pitch diameter multiplied by the ratio. If the gear has four times as many teeth as the mating pinion, then the pitch diameter of the gear is four times greater than the pinion pitch diameter.)

After the pitch diameter of the pinion has been determined, the size of the teeth are determined by calculations regarding a possible failure in tooth breakage. A broken tooth tends to be catastrophic to a gear unit, so the designer usually makes the teeth large enough so that they are definitely less apt to fail in breakage mode than in a pitting mode. This makes the design life of a gear unit primarily dependent on its surface fatigue capacity (pitting resistance) rather than on its cantilever beam capacity (capacity to resist tooth breakage).

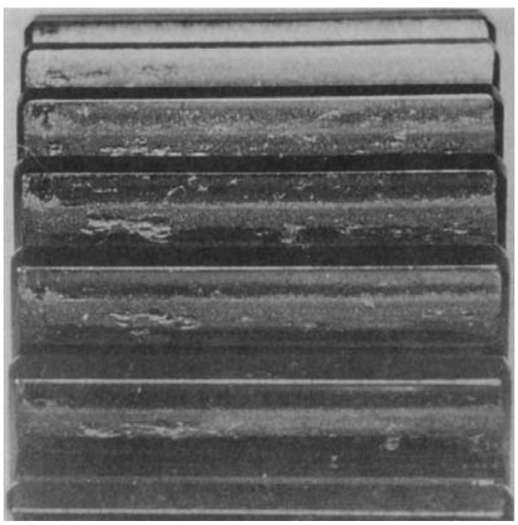


Fig. 4 Pitted gear teeth. Note micropitting at the pitch line, scattered macropits, and one area of gross pitting near the left end.

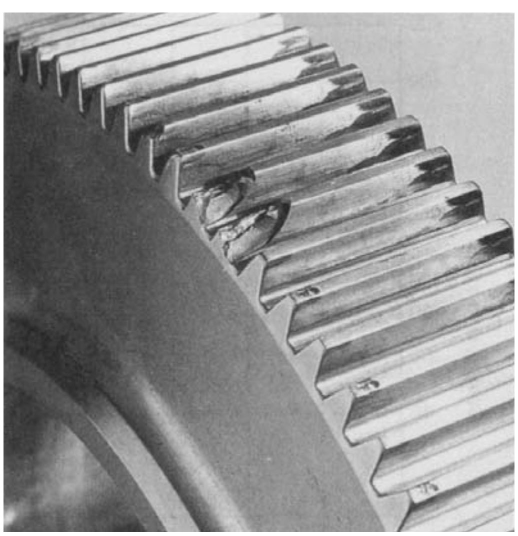


Fig. 5 Misaligned gear has tooth breakage at ends of teeth.

Life Determined by Contact Stress

The surface contact stress, S_c, between gear teeth (discussed further in Ref 1 with in-depth presentations of gear rating calculations and concepts) may be calculated by a simplified formula:

$$S_c \text{ (in psi)} = C_k (KC_d)^{0.5} \qquad \text{(Eq 1)}$$

where C_k is the geometry factor for durability (see Table 2), K is the index of tooth loading

Table 2 Some typical geometry factors (C_k) for durability

No. of pinion teeth	No. of gear teeth	Standard addendum, a_p = 1.000, a_G = 1.000, whole depth 2.35 in.	25% long addendum, a_p = 1.250, a_G = 0.750, whole depth 2.35 in.
Spur gears, 20° pressure angle			
25	35	5913	(a)
25	50	5985	5756
25	85	6057	5768
35	35	5756	(a)
35	50	5810	5722
35	85	5877	5726
35	275	5925	5732
Spur gears, 25° pressure angle			
20	35	5564	5371
20	50	5660	5419
20	85	5744	5479
25	35	5419	(a)
25	50	5503	5335
25	85	5575	5380
35	35	5262	(a)
35	50	5329	5269
35	85	5407	5298
35	275	5479	5323
Helical gears, 15° helix angle, 20° normal pressure angle			
25	35	4576	(a)
25	50	4540	4564
25	85	4479	4533
35	35	4516	(a)
35	50	4492	4499
35	85	4419	4469
Helical gears, 15° helix angle, 25° normal pressure angle			
25	35	4444	(a)
25	50	4419	(a)
25	85	4383	(a)
35	35	4395	(a)
35	50	4371	(a)
35	85	4335	(a)

(a) 25% long addendum design not used here. a_p, pinion addendum; a_G, gear addendum severity for pitting (see Eq 2), and C_d is the overall derating factor for durability (see Table 3).

The index of tooth loading for pitting is called the "*K*-factor." For spur or helical gears, the *K*-factor is:

$$K = \frac{W_t}{Fd}\left(\frac{m_G + 1}{m_G}\right) \quad \text{(Eq 2)}$$

where W_t is the tangential driving force (in pounds), F is the net face width in contact (in inches), d is the pinion pitch diameter (in inches), and m_G is the ratio of gear teeth to pinion teeth.

The tangential driving force is obtained from the horsepower being transmitted by the combination of pinion and gear, the speed of the pinion in revolutions per minute, and the pitch diameter of the pinion in inches. Thus:

$$W_t = \frac{P \times 126{,}050}{n_p \times d}\ \text{lb} \quad \text{(Eq 3)}$$

where P is the horsepower transmitted and n_p is the speed of the pinion (in rpm).

Example 1: Surface Durability of Gear. To show how the above equations work, consider a practical problem of an electric motor (running at a design speed of 1800 rpm) that drives a fan at 450 rpm and transmits power at a constant 30 hp. If the pinion driven by the motor has 22 teeth at a 15° helix angle and the normal diametral pitch of the teeth is 10, what are the calculated *K*-factor and contact stress when the mating gear has a face width 1.0 times the pinion pitch diameter? The normal pressure angle is 20°. The tooth design is 25% extra addendum of the pinion and 25% shorter addendum for the gear.

The first step is to obtain the pinion pitch diameter:

$$d = \frac{N_p}{P_n \cos \psi}\ \text{in.} \quad \text{(Eq 4)}$$

where N_p is the number of pinion teeth, ψ is the helix angle, and P_n is the normal diametral pitch. For the problem described above, then:

$$d = \frac{22}{10 \cos(15°)} = 2.2776\ \text{in.}$$

$$F = 1.0 \times 2.2776 = 2.28\ \text{in.}$$

$$W_t = \frac{30 \times 126{,}050}{1800 \times 2.2776} = 922.4\ \text{lb}$$

$$K = \frac{922.4}{2.2776 \times 2.28}\,\frac{(4.0 + 1)}{(4.0)} = 222.0$$

$$S_c = 4558\,(222.0 \times 2.5)^{0.5}$$

$$= 107{,}380\ \text{psi}$$

Table 3 Some typical overall derating factors (C_d)

	Accuracy level by AGMA			
Aspect ratio, *d/F*	**12 or 13**	**10 or 11**	**8 or 9**	**6 or 7**
For designs where pinion pitch diameter is about 2 in. (50 mm)				
0.25 or less	1.50	1.65	1.80	2.00
0.50 or less	1.65	1.75	2.00	2.25
0.75 or less	1.75	2.00	2.25	2.50
1.0 or less	2.00	2.25	2.50	2.75
For designs where pinion pitch diameter is about 5.0 in. (125 mm)				
0.25 or less	1.45	1.55	1.65	1.80
0.50 or less	1.55	1.65	1.80	2.00
0.75 or less	1.65	1.80	2.00	2.25
1.0 or less	1.80	2.00	2.25	2.50
For designs where pinion pitch diameter is about 12 in. (300 mm)				
0.25 or less	1.40	1.50	1.60	1.70
0.50 or less	1.50	1.60	1.70	1.80
0.75 or less	1.60	1.70	1.80	2.00
1.0 or less	1.70	1.80	2.00	2.25

Note: These derating factors should be considered illustrative only. For a prime mover that is rough running and a driven device that is rough running, the derating needs to be more than shown above. Likewise, for a very smooth-running prime mover and a low-inertia, smooth-running driven device, lower derating factors may be justified. Field experience in vehicles, aircraft, and ships tends to be the best guide for choosing the right derating factor.

Table 2 does not have 22/88 tooth numbers, so in calculating S_c, C_k was determined from Ref 2.

After the contact stress has been calculated, a rating curve sheet needs to be used to determine what life can be expected. Figure 6 shows typical contact stress values plotted against numbers of tooth contacts. This curve sheet is drawn for gearing having these qualifications:

- Material quality: Normal industry quality (Grade 1 per Ref 2)
- Probability of failure: Not over 1 in 100
- Lubrication: Full oil film, which is regime III
- Hardness: Both pinion and mating gear at least as hard as value shown on surface durability figure

Determination of Gear Life. To explain how gear life is determined, the discussion can be based on the same sample problem. The calculated contact stress was 107,380 psi. From Fig. 6 the following life values are obtained for steel gears (through hardened):

- 210 HB: 1.1×10^6 cycles
- 300 HB: 2.0×10^8 cycles
- 60 HRC: over 1.0×10^{10} cycles

The next step is to calculate the number of pinion cycles for some different lengths of time at full rated load, where:

$$\text{Cycles} = (\text{rpm}) \times (60\ \text{min/h}) \times (\text{No. of hours})$$

$$= 1800 \times 60 \times \text{No. of hours}$$

Taking some different numbers of hours, the following results are obtained:

Time at rated power, h	No. of pinion cycles
500	54×10^6
2,000	216×10^6
20,000	2.16×10^9

Determination of adequate gear life requires some reasoning. Inexpensive equipment that is used infrequently and is seldom used at full rated power might be satisfactory if it had only a 500-hour capability at full rating. The design under consideration would be quite inadequate for 500 hours if the teeth were at low hardness, such as 210 HB. However, if the pinion and gear were at 300 HB hardness, they would have enough capacity for 500 hours and almost enough for 2000 hours.

At full hardness of 60 HRC, the situation changes considerably. The calculated life capacity is well over 10 billion cycles (on the pinion). Only a little over 2 billion cycles are needed for a life of 20,000 hours. The gear has only one-quarter of the pinion cycles, which is just a little over 0.5 billion cycles. If the gear

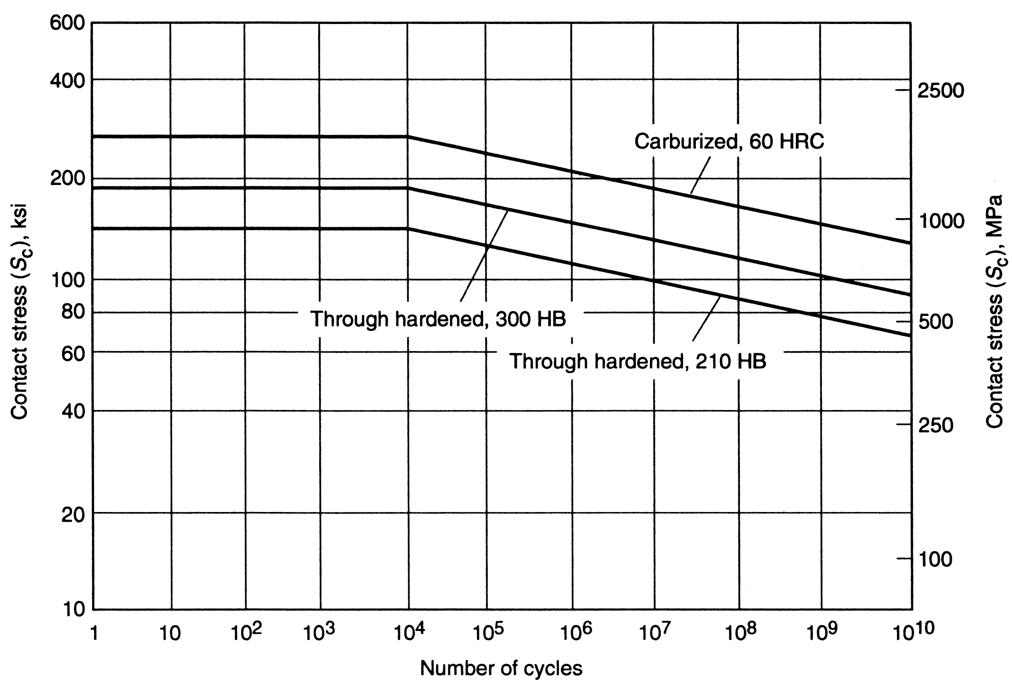


Fig. 6 Surface durability curve for gear life rating (contact stress vs. cycles) for normal industry quality material (Grade 1 per Ref 2)

hardness is the same as that of the pinion, the gear certainly has adequate capacity.

Derating Factors. In the above calculation, the derating factor (C_d) of 2.50 may seem high but can be explained. The values given in Table 3 are based on an assumption of *average* conditions and a failure from pitting conditions of situation 1, 2, or 3 (see Table 4 and later discussions). A derating factor lower than 2.0 may even be acceptable for a 2 in. (50 mm) pinion if the driving and driven units are very light (low mass) and run *very* smoothly. When a low derating factor such as 1.5 is used, there is usually fairly extensive pitting. If the concept of a "failure" is situation 4 or 5 (Table 4), field experience may show that the gears can survive even though the derating is too low for situations such as situations 1, 2, and 3 in Table 4.

If a derating factor of 1.50 had been used, the calculated stress at 10 billion cycles would be reduced by the square root of the change in the derating factor. This works out to $(1.50/2.50)^{0.50} = 0.775$. This makes the calculated contact stress drop to 83,220 psi. At this value, a through-hardened steel at 300 HB has a life of over 10 billion cycles. This is enough for a life of 20,000 hours.

Now consider the variables for the 2.50 derating factor. The first of three major factors is a derating factor of 1.4 for typical experience with industrial fan drives. These run somewhat rough, and gear people evaluate this as a 1.4 application factor.

With a face width equal to the pitch diameter, there will normally be some misfit across the wide face. In addition, the accuracy of tooth spacing will not be close to perfect. For an industrial application, the tooth spacing accuracy is probably AGMA quality level 9. This requires a derating factor of about 1.4 for small gearing such as a 2 in. (50 mm) pinion.

It is also necessary to consider dynamic overload due to the masses in the drive system and the tooth spacing accuracy. In this sample problem, the 2.125 pitch diameter pinion running at 1800 rpm has a pitch line velocity of 1073 ft/min (327 m/min). This kind of drive at this speed could be expected to have a dynamic overload factor of 1.00/0.80 = 1.25.

The overall derating factor is obtained by multiplying these three factors such that $1.4 \times 1.4 \times 1.25 = 2.45$ (this is rounded off to 2.50).

For the values shown in Fig. 6, the geometry factor (C_k) may look quite conservative when compared to values used in bearing design. Part of this is due to the relatively high component of sliding in gear tooth contacts. For spur and helical gearing, pure rolling occurs only at the pitch line. Above and below the pitch line there is appreciable sliding. Also, the geometric shape

Table 4 Situations and concepts of pitting failures in gears

Description	Recommended action
Situation 1	
Some micropitting and a few macropits on the pinion. No gear pitting.	Scrap the pinion, even though it is still quite runnable. It may not last until the next scheduled overhaul.
Situation 2	
Micropitting and appreciable macropitting (e.g., 3–10% of the tooth surface of the pinion). Almost no pitting on gear.	Scrap the pinion, because metal debris in the oil system is a serious hazard to the bearings. Premature bearing failure may occur at any time.
Situation 3	
Micropitting and considerable macropitting (e.g., 15–40% of the surface). One or more gross pits. Damage to both pinion and gear.	Scrap the gears. Damage to the tooth surface may lead to tooth breakage, starting in a badly pitted location.
Situation 4	
Macropitting over 50–100% of the pinion tooth surface. Gross pitting present. Removal of metal thins the teeth and disrupts load sharing between teeth. Gear unit has greatly increased noise and vibration.	Scrap the gears. Complete failure is imminent. Foundation bolts may break. Bearings are probably in the process of failure.
Situation 5	
Macropitting all over the teeth. Also considerable gross pitting. Teeth are thinned so much by wear that the tips are becoming sharp like a knife.	Scrap the gears. A thinned tooth may roll over and jam the drive—if it doesn't break off first.

of bearings is simpler than that of toothed gears. This helps in both quality of fit and quality control of material.

Another factor in comparing rolling element bearings to gears is probability of failure. Bearings are considered somewhat expendable and are generally designed with a probability of failure of 10 in 100 rather than the 1 in 100 that is normal for gearing. In some cases of expendable gearing, a factor of 10 failures in 100 is used for the rating curve. If Fig. 6 was redrawn for the 10 to 100 probability, all curves shown would allow about 17% more stress. The approximate life for a stress of 107,380 psi (740 MPa) works out as follows:

	Approximate life at probability of failure of:	
Hardness, HB	**1 out of 100**	**10 out of 100**
210	1.1 million cycles	20 million cycles
300	180 million cycles	3.5 billion cycles

Pitting Failure. The degree of risk to be assumed in a gear design is a most important consideration. The above results show more than a 15 times longer life by going from a very low risk of failure to a still somewhat low risk of failure. If a 10 out of 100 degree of risk was tolerable, then the gear set being considered would have adequate pitting life for 20,000 hours at 300 HB.

The concept and definition for contact stress of pitting failures is another very significant consideration. Pitting failure is not easy to define, and a gear designer needs to use judgment based on experience. In general, a *pitted* gear tooth will still run, whereas broken-off gear teeth will so damage a gear that it is either impossible or impractical to continue to operate the gear unit.

Table 4 shows five different levels of tooth damage due to pitting that govern whether or not a pinion and a mating gear have failed by pitting. In aircraft or space vehicle gearing, situation 1 is cause for failure. In long-life turbine gearing, situation 2 is quite apt to be considered a failure. According to the calculated life given earlier in this chapter, the gear failed when it reached situation 3.

In some factory and farm applications, gears that have reached situation 4 are still kept in service. The cost of replacement and a forgiving environment where noise and vibration can be tolerated may prompt the gear user to keep running the gears. Situation 5 is the ultimate. If the pinion won't turn under power or the pinion does turn and the gear does not turn, there is just no question about the gears having failed.

From what has been discussed so far, the method used in the gear trade to calculate gear life should be quite clear. It should be evident, though, that much experience and judgment are needed to properly decide when a pitted and worn gear should be taken out of service and scrapped.

Although there are good guidelines for evaluating the things that derate gears, it still takes much data and experience to set numerical values that are just right for a particular product application. For instance, long-life turbine gearing is very expensive, and downtime in rebuilding a broken main gear drive may be much more expensive than replacing a gear unit. This means that the designer of turbine gearing is very concerned about having an adequate overall derating factor and close control over gear material quality. Grade 2 material is often specified rather than grade 1 material. Grade 2 steel is rated to carry about 25% more contact stress than grade 1 when carburized but only about 10% more stress when through hardened.

In the vehicle field (trucks, tractors, and automobiles), gearing is not highly expensive and a replacement gear unit can often be installed within a few hours. The designer of the gears may use a relatively low overall derating. The designer learns by thousands of similar units in service how to set the derating factor and how to control geometric quality and material quality so that adequate gear life is obtained.

The standards of the American Gear Manufacturers Association (AGMA) give recommendations for how to rate many kinds of gears and how to specify geometric and material quality. AGMA standards define established gearing practices for the gear trade. In the industrial field, contracts for geared machinery generally specify that the gearing supplied must be in accordance with applicable AGMA standards.

The allowable contact stress for gears made of grade 1 material (per Ref 2) is drawn for the lubrication condition of regime III (full oil film). Many gears that are small and run at relatively slow pitch line speeds are in regime II (partial oil film) or regime I (wet with oil but with almost no elastohydrodynamic (EHD) oil film thickness).

As of December 1995, there is no national standard for how to rate gears for allowable contact stress when they are in regimes II or I. There is much field experience, though, with gears running with an inadequate EHD oil film thickness. Work is underway to add material to AGMA standards to cover the problem of an inadequate oil film thickness.

Figure 7 shows how the allowable contact stress may change for grade 1 carburized steel gears at 60 HRC. Note the dashed curves for regimes II and I.

Life Determined by Bending Stress

The bending stress of a pinion or a mating gear tooth (Ref 1, 2) may be estimated by a simplified formula:

$$S_t = K_t U_l K_d \tag{Eq 5}$$

where K_t is a geometry factor (see Table 5), U_l is the index of tooth loading severity for breakage (see Eq 6 below), and K_d is the overall derating factor for bending stress (see footnote in Table 3). The bending stress derating factor (K_d) is approximately the same as the overall derating factor (C_d) unless there is a wide face width. When the aspect ratio is around 0.25, the C_d for contact stress and the K_d for bending stress are about equal. At an aspect ratio of 1.0, K_d may be about 80% of C_d.

The index of tooth loading for breakage is often called "unit load." Its value for spur or helical teeth is:

$$U_l = \frac{W_t P_n}{F} \tag{Eq 6}$$

where W_t is the tangential driving force (same value as in Eq 3), F is the net face width in contact, and P_n is the normal diametral pitch (1/in.).

P_n is the inverse of the tooth size in a plane normal to a helical tooth. The diametral pitch (P_d) is the inverse of the tooth size in a plane normal to the axis of gear. The relation between the two is:

$$P_d = P_n \cos(\text{helix angle}, \Psi) \tag{Eq 7}$$

The tangential load for tooth strength is the same quantity as the tangential load used for contact stress calculations (Eq 3).

The derating factor for strength is sometimes taken as the same value as is used for surface durability. In some cases of gears with favorable contact ratios, there may be justification to use a somewhat lower derating factor for strength. The points of high beam stress are some distance away from the points of maximum contact stress. This tends to allow the beam stress to

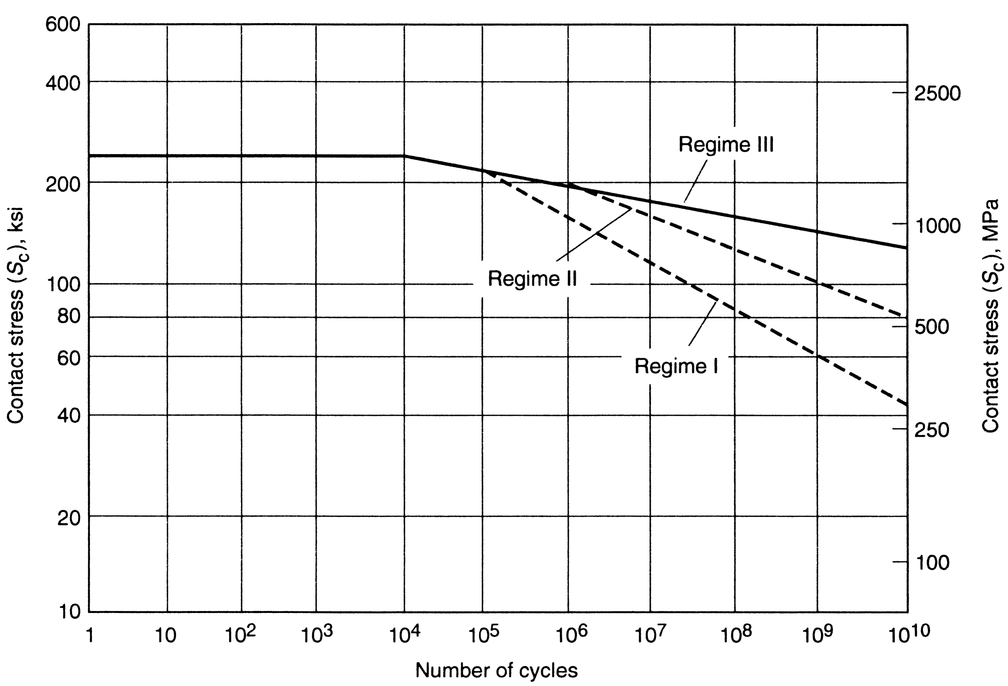


Fig. 7 Regimes of lubrication for carburized gears of normal industry quality material (Grade 1 material, per Ref 2)

spread out more than the contact stress. (In this brief chapter it is not practical to show how to reduce the derating factor for the tooth strength calculations.) For the sample problem below it is reasonable to use $K_d = 2.0$. (The pitting derating, C_d, was 2.5.)

Example 2: Rating Gear Life with Bending Stress. In Example 1 of this chapter, the numerical values are $W_t = 922.4$ lb, $P_n = 10$, and $F = 2.28$. For the data just given, the unit load index is:

$$U_l = \frac{922.4 \times 10}{2.28} = 4045.4 \text{ psi (27.9 MPa)}$$

The geometry factor is taken as 1.75 (see Table 5). The calculated bending stress becomes:

$$\begin{aligned} S_t &= 1.75 \times 4045.4 \times 2.0 \\ &= 14.159 \text{ psi (97.6 MPa)} \end{aligned}$$

The tooth bending stress can now be used to determine allowable cycles from a rating curve, such as Fig. 8. Figure 8 shows typical bending stresses plotted against tooth contacts. This curve sheet is drawn for gearing having these qualifications:

- Material quality: Grade 1
- Probability of failure: Not over 1 in 100
- Lubrication: Full oil film, which is regime III
- Hardness: Both pinion and mating gear at least as hard as value shown on bending strength figure (Fig. 8)

The expected life of both the pinion and the gear, from a tooth breakage standpoint, can now be determined for the calculated stress of 14,159 psi (97.6 MPa) for the pinion and a calculated stress of 14,321 psi (98.7 MPa) for the gear. (The gear has the same unit load as the pinion

Table 5 Some typical geometry factors (K_t) for strength

No. of pinion teeth	No. of gear teeth	Standard addendum		25% long addendum	
		a_p = 1.000,	a_G = 1.000,	a_p = 1.250,	a_G = 0.750,
		whole depth 2.35		whole depth 2.35	
Spur gears, 20° pressure angle					
25	35	2.72	2.55	2.37	3.00
25	50	2.68	2.42	2.33	2.74
25	85	2.62	2.22	2.30	2.54
35	35	2.50	2.50	2.21	2.90
35	50	2.42	2.30	2.18	2.70
35	85	2.36	2.15	2.15	2.54
35	275	2.20	2.00	2.10	2.45
Spur gears, 25° pressure angle					
20	35	2.42	2.16	2.08	2.42
20	50	2.38	2.01	2.05	2.22
20	85	2.35	1.90	2.03	2.09
25	35	2.26	2.10	2.00	2.40
25	50	2.23	1.97	1.95	2.20
25	85	2.20	1.86	1.91	2.05
35	35	2.06	2.06	1.88	2.34
35	50	2.02	1.92	1.84	2.15
35	85	1.99	1.80	1.80	2.00
35	275	1.90	1.70	1.75	1.82
Helical gears, 15° helix angle, 20° normal pressure angle					
25	35	1.93	1.83	1.74	2.05
25	50	1.90	1.76	1.72	1.88
25	85	1.87	1.67	1.71	1.75
35	35	1.82	1.82	1.68	1.97
35	50	1.78	1.73	1.65	1.82
35	85	1.75	1.64	1.62	1.73
Helical gears, 15° helix angle, 25° normal pressure angle					
25	35	1.60	1.52	...	...
25	50	1.57	1.47	...	...
25	85	1.54	1.40	...	...
35	35	1.50	1.50	...	...
35	50	1.42	1.42	...	...
35	85	1.35	1.34	...	...

and the same derating factor. However, the geometry factor for the gear from Ref 1 is 1.77, compared to 1.75 for the pinion.)

Based on Fig. 8, the pinion stress of 14,159 psi (97.6 MPa) is well below all the curves. If a pinion life of 20,000 hours was desired, the bending stress allowable could be about 23,000 psi (158.5 MPa) even for the through-hardened, 210 HB steel. This means that with 22 pinion teeth, the "life" of the pinion is determined by pitting and not by tooth breakage.

Suppose the pinion had the same diameter but had 40 teeth instead of 22 teeth. The bending stress would increase, because the normal diametral pitch would be about 18. The derating would not change for an increased number of teeth, and the geometry factor would tend to decrease a small amount. This means that a 40-tooth pinion meshing with a 160-tooth gear would have a calculated bending stress of about 22,300 psi (153.8 MPa).

If the normal diametral pitch was changed to 18.18182, the pitch diameters of the pinion and gear would not change with 40 pinion teeth and a 15° helix. The *K*-factor for pitting would not change, and the change in the geometry factor for durability would be a small decrease. This means that it is possible to change numbers of teeth to adjust bending stress and not have much change in the contact stress.

In this sample problem, "life" determined by contact stress is very short at 210 HB, but it is almost enough for 2000 hours at 300 HB. The pinion could have 40 teeth at 300 HB because the allowable bending stress is up to 32,000 psi (220 MPa) at 2000 hours (216×10^6 million cycles). With the bending stress at 24,274 psi (167.4 MPa) there is a modest amount of extra margin in bending strength, but not nearly as much as at 22 teeth.

From a practical standpoint, it is quite desirable to control the life of a gear set and have an appreciable extra margin of bending strength. Having lots of small teeth instead of fewer and larger teeth is not desirable from two other standpoints: in general, parts with fewer teeth cost less to make; and larger teeth can stand somewhat more wear.

In summary, this example shows that 22 teeth on a pinion is a reasonable design. There is no good reason to use 40 teeth, even though the bending stress would be allowable.

Gear Tooth Failure by Breakage after Pitting

The preceding section covered tooth breakage by the tensile stress in the root fillet. Another kind of common tooth breakage in

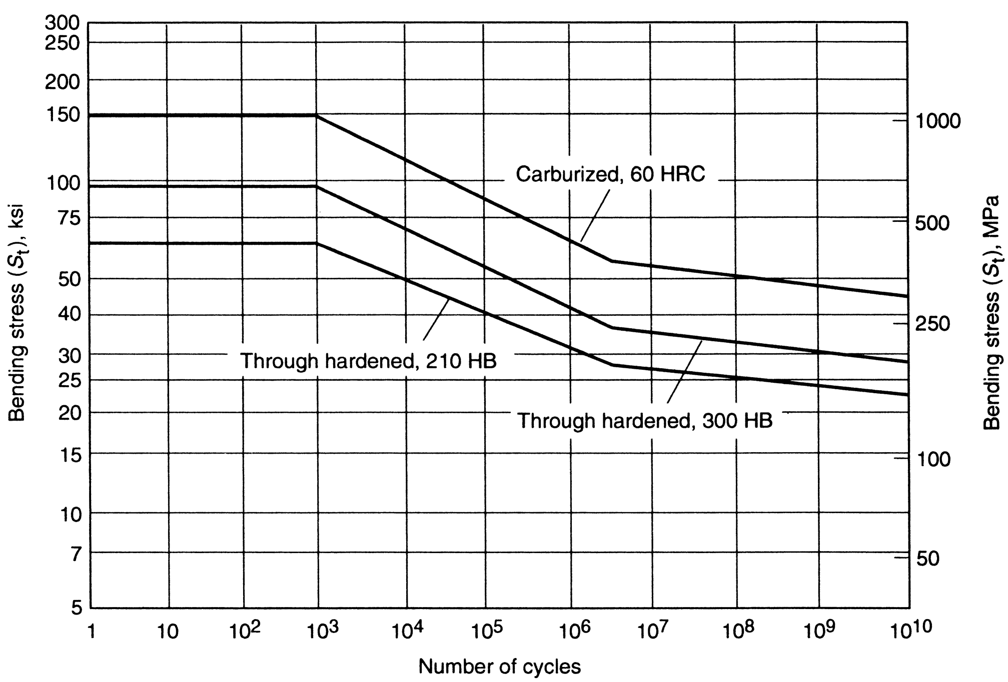


Fig. 8 Bending strength curve for gear life rating of normal industry quality material (Grade 1 per Ref 2)

gearing starts somewhat high on the tooth profile rather than in the root fillet. When a gear is damaged by pitting, a crack can start at a pit, spread lengthwise, and go all the way through the tooth. On wider-face-width gears, a triangular piece of tooth often breaks out. This piece may be as wide as one-third or one-half of the face width. The break may progress as deep as the root fillet at one end and run out at the outside diameter at the other end. The "eye" where the crack started may be at the pitch line, or it may be in the lower flank of the tooth.

If a pinion or gear continues in service after an appreciable part of the tooth is gone, the now-overloaded remainder of the tooth is likely to break off. Sometimes, even though several teeth have portions broken off, the gears continue in service because the increased noise and vibration are not enough to attract attention. Then one or more teeth are completely removed by successive chunks breaking off (see Fig. 9).

In narrow-face-width gears, such as those where the face width is about equal to the tooth height, a break starting at a pit tends to remove the tooth across the whole face width.

The normal calculation of bending stresses is not useful in designing gears to avoid tooth breakage starting at pits. The logic of this kind of trouble must be considered *an incident of tooth breakage caused by pitting.*

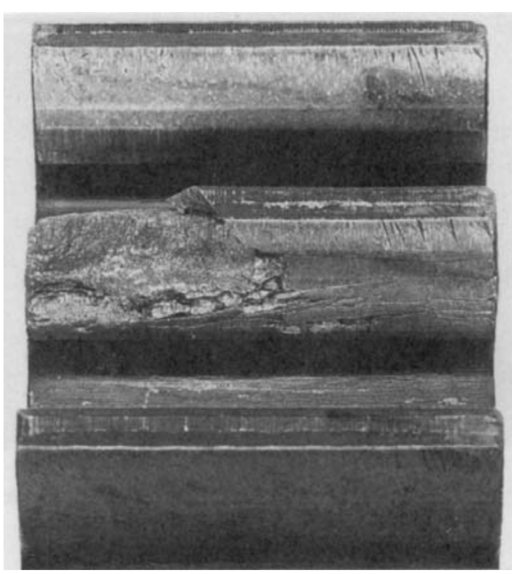


Fig. 9 End of tooth broken because of gross pitting. Note cracks in remaining part of tooth.

Gears that are case hardened by carburizing are the most susceptible to this trouble. Gears with a nitrided case are not quite as susceptible because the nitrided case depths tend to be thinner. A pit in a thin case is not quite as great a stress riser as a pit in a deep case. To prevent pitting altogether, though, deep carburized case depths may be needed.

Through-hardened gears in the 300 to 400 HB range are also quite critical. In general, gears are not used that are around 500 HB. At 450 HB and higher, a through-hardened gear is usually so brittle that it will fracture with even a small amount of surface distress.

The concepts of pitting failure given in Table 4 are quite useful in regard to the risk of tooth breakage stemming from pitting. In situation 1, the pitting is relatively minor, so there is very little danger of tooth breakage starting at a pit.

In situation 2, there is a moderate risk of some area on the tooth surface having enough macropits to start a tooth breakage crack when the pinion or gear has a carburized case. Through-hardened parts at 300 HB are not so notch sensitive and will probably not develop cracks from pits. Through-hardened parts at 210 HB are not notch sensitive. There is not much risk of cracks from pits at this level of hardness.

In situation 3, the much heavier pitting is quite risky for carburized gear teeth and moderately risky for through-hardened gears at 300 HB. However, through-hardened gears at 210 HB do not have much risk.

In some special cases where the pinion is quite hard (e.g., 360 HB) and the gear is much lower in hardness (e.g., 300 HB), there may be such serious *ledge wear* on the gear due to micro- and macropitting that a whole layer of metal is worn away from the gear tooth. This can result in the addendum of the gear being forced to carry all the load. This tends to double the stress in the gear root fillet, and gear teeth then break off due to the greatly increased stress.

Ledge wear that occurs in helical gears over a long period of time is not too dangerous if both the pinion and gear have about the same amount of wear. The big danger of tooth breakage comes when one pitted gear has little or no ledge wear and the mating gear has serious ledge wear. Figure 10 shows an example of a gear tooth with ledge wear and in danger of breaking. (The pinion that mated with this gear had some pitting but essentially no ledge wear.)

In situations 4 and 5, hard gears cannot survive without tooth breakage. Low-hardness gears that run at relatively slow pitch line speeds often survive for years without tooth breakage, even though the pitting and wear are severe.

In summary, tooth breakage due to pitting is a high risk with hard gears and may not be much of a risk if the gears are of low hardness and not running fast.

Flaws and Gear Life

Metallurgical flaws may seriously reduce the life of gears. The calculations for the life of gear teeth assume that the material in the gear is not flawed by a mistake in processing the part. If the material is grade 1 (normal industrial quality per Ref 2) the quality is not as good as for grades 2 (extra high quality for high speed, long life gearing per Ref 2) or 3 (super quality for critical aerospace gearing per Ref 2), but it is assumed that the quality is normal for the grade specified. For instance, the microstructure for grade 1 is not specified to a limit as close as that for grade 2. Also, the cleanliness of the steel in grade 1 does not have to be as good as the cleanliness for grades 2 or 3.

Some of the more common flaws that affect the life of gear teeth are:

- Case depth for carburized, nitrided, or induction-hardened teeth too thin or too thick
- Grinding burns on case-hardened gears
- Nonhomogenous structure, voids, hard spots, soft spots, etc.
- Alloy segregation
- Core hardness too low near the root diameter or at the middle of the face width
- Forging flow lines pronounced and in the wrong direction
- Quenching cracks
- Composition of steel not within specification limits

When gears fail prematurely, the normal first step of an investigator is to make some calculations of gear life to see if the gear unit size was large enough for the rated life. If these calculations show that a gear unit rated for something like 10 years of life was indeed large enough and had the right materials specified, then it becomes a matter of concern to investigate factors such as lubrication system, behavior of driving and driven equipment, operating environment, and material quality (perhaps some serious metallurgical flaw (or flaws) exist).

Quite often the case depth is not right. The gear drawing should specify the right case depth for the midpoint of the tooth flank. Besides the middle of the tooth, the case depth needs to be right at the lower flank of the tooth. The case depth at the tip of the tooth is also critical.

The case depth may be right at midheight but too thin at the lower flank and root fillet. This is often caused by inadequate carbon gas penetration between teeth. In large gears, heat soak back after quenching may reduce the hardness considerably. Too much case depth at the tooth tip may cause the tooth tips to be so brittle that they break off in service. (This has happened off and on in gears used in mining.)

If large, through-hardened gears are quenched too drastically, there may be hidden cracks in the gear body. On the other hand, an inadequate quench may cause the hardness of the gear teeth to be below specifications in the middle of the face width or near the root diameter of the teeth.

Case-hardened gears can be ground by several different methods. There is always a risk of

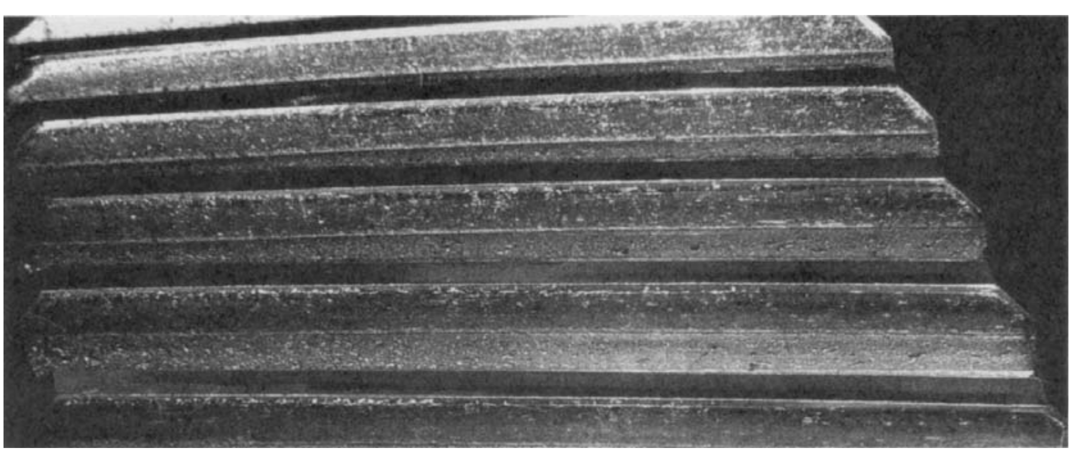


Fig. 10 Helical gear with serious ledge wear due to micro- and macropitting

grinding too fast or having inadequate cooling at the grinding location. Grinding burns and grinding cracks must be avoided to get the potential life out of the gears.

When scrap material is melted to make steel, there is a hazard that something in the melt will produce a local hard spot or a local soft spot. There may also be stringers of inclusions. There is a well-developed art of producing steel ingots that can be made into steel billets of uniform good material for a toothed gear. Good gears require good steel.

When gear parts are forged, the flow lines in the forging should not be in the wrong direction at critical stress points, such as the root fillet of a gear tooth or where a thin web meets a thin gear rim. Rolled bar stock that is not forged may have flow lines in the wrong direction for a critical gear part. Normal gear rating practice is based on the assumption that the forging or rolling of the stock to make the gear blank is done right for the configuration of the gear part.

Another flaw to be concerned about is the chemical composition of the steel in the gear. Sometimes mistakes are made and the steel for a gear is taken out of the wrong bin. Through-hardening steels have different compositions than case-hardening steels.

Sometimes an improper substitution is made. For example, it quite often happens in worldwide operations that a readily available local steel is substituted for the specified steel that was just right for the rating and method of manufacturing used in another country. It is then quite a shock to have gears failing in a year or two that should have lasted for 10 or more years.

Bores and Shafts

Besides proper gear design, geometric accuracy, and good gear material, associated components in a gear train can also be important. This section briefly reviews some components in the design and structure of each gear and/or gear train that must be considered in conjunction with the teeth.

A round bore, closely toleranced and ground, may rotate freely around a ground shaft diameter, rotate tightly against a ground diameter by having a press fit, or be the outer race of a needle bearing that gives freedom of rotation. Each application has unique problems. A ground bore is always subject to tempering, burning, and checking during the grinding operation. A freely rotating bore requires a good lubrication film, or seizing and galling may result. The bore used as a bearing outer race must be as hard as any standard bearing surface and is subject to all conditions of rolling contact, such as fatigue, pitting, spalling, and galling. The bore that is press fit onto a shaft is subject to a definite amount of initial tensile stress. Also, any tendency for the bore to slip under the applied rotational forces will set up a unique type of wear between the two surfaces that leads to a recognizable condition of fretting damage.

A spur of a helical round bore gear acting as an idler, or reversal, gear between the input member and the output member of a gear train has an extremely complicated pattern of stresses. The most common application is the planet pinion group in wheel-reduction assemblies or in planetary-type speed reducers. Typical stress patterns are as follows:

- The tensile and compressive stresses in the bore are due to bending stresses of the gear being loaded as a ring.
- The maximum tensile and compressive stresses in the bore increase as the load on the teeth increases.
- The maximum tensile and compressive stresses in the bore increase as the clearance between the bore and the shaft is increased.
- The maximum tensile and compressive stresses in the bore increase as the ratio of the size of the bore to the root diameter of the teeth is increased.
- During one revolution of the gear under load, the teeth go through one cycle of complete reversal of stresses, whereas each element of the bore experiences two cycles of reversals.

Three modifications of the round bore alter stress patterns considerably:

- *Oil holes* that extend into the bore are intended to lubricate the rotating surfaces. Each hole may be a stress raiser that could be the source of a fatigue crack.
- *Tapered bores* are usually expected to be shrink fitted onto a shaft. This sets up a very high concentration of stresses, not only along the ends of the bore but also at the juncture on the shaft.
- *A keyway* in the bore also creates a stress-concentration area. A keyway is also required to withstand a very high load that is continuous. In fact, the applied load to the side of the keyway is directly proportional to the ratio of tooth pitchline radius to the

radius of the keyway position. Fatigue failure at this point is common.

Splined Bores. The loads applied to the splined-bore area are also directly proportional to the ratio of pitchline radius of the teeth to the pitchline radius of the splines. However, the load is distributed equally onto each spline; therefore, the stress per spline is usually not excessive. It is possible for an out-of-round condition to exist that would concentrate the load on slightly more than two splines. Also, a tapered condition would place all loadings at one end of the splines, which would be detrimental both to the gear splines and to the shaft.

Heat treating of the splined area must be monitored closely for quench cracks. Grinding of the face against the end of the splines can also cause grinding checks to radiate from the corners of the root fillets.

Shafts. The shafts within a gear train, as well as the shank of a pinion that constitutes a shaft in function, are very important to load-carrying capacity and load distribution. They are continually exposed to torsional loads, both unidirectional and reversing. The less obvious stressed condition that is equally important is bending. Bending stresses can be identified as unidirectional, bidirectional, or rotational. When the type of stress is identified, one can explore the causes for such a stress.

There can be a number of stresses applied to a shaft that are imposed by parts riding on it. For example, helical gears will transmit a bending stress, as will straight and spiral bevel gears; round bores may be tight enough to cause scoring and galling; a splined bore may cause high stress concentration at the end face; runout in gears may cause repeated deflections in bending of the shaft; and loose bearings may cause excessive end play and more bending.

ACKNOWLEDGMENT

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REFERENCES

1. D.W. Dudley, *The Handbook of Practical Gear Design,* Technomics Publishing Co., Inc., Lancaster, PA, 1994
2. *Fundamental Rating Factors and Calculation Methods for Involute and Helical Gear Teeth,* ANSI/AGMA 2001-C95, American Gear Manufacturers Association

SELECTED REFERENCES

Fatigue of Gears

- B. Abersek, J. Flasker, and J. Balic, Theoretical Model for Fatigue Crack Propagation on Gears, *Conf. Proc. Localized Damage III: Computer-Aided Assessment and Control,* Computational Mechanics Publications, Billerica, MA, 1994, p 63–70
- A.Y. Attia, Fatigue Failure of Gears of Circular-Arc Tooth-Profile, *J. Tribol.,* Vol 112 (No. 3), 1990, p 453–459
- G.P. Cavallaro et al., Bending Fatigue and Contact Fatigue Characteristics of Carburized Gears, *Surf. Coat. Technol.,* Vol 71 (No. 2), 1995, p 182–192
- G. Dalpiaz and U. Meneghetti, Monitoring Fatigue Cracks in Gears, *Nondestr. Test. Eval. Int.,* Vol 24 (No. 6), 1991, p 303–306
- J. Flasker and B. Abersek, Determination of Plastification of Crack Tip of Short and Long Fatigue Crack on Gears, *Conf. Proc. Computational Plasticity: Fundamentals and Applications,* Pineridge Press Ltd., Mumbles, Swansea, U.K., 1992, p 1585–1595
- Y. Jia, Analysis and Calculation of Fatigue Loading of Machine Tool Gears, *Int. J. Fatigue,* Vol 13 (No. 6), 1991, p 483–487
- B. Jiang and E.A. Shao, Rapid Method for Measurement of Rolling Contact Fatigue Limit of Case-Hardened Gear Materials, *ASTM J. Test. Eval.,* Vol 18 (No. 5), 1990, p 328–331
- B. Jiang, X. Zheng, and M. Wang, Calculation for Rolling Contact Fatigue Life and Strength of Case-Hardened Gear Materials by Computer, *ASTM J. Test. Eval.,* Vol 21 (No. 1), 1993, p 9–13
- B. Jiang, X. Zheng, and L. Wu, Estimate by Fracture Mechanics Theory for Rolling Contact Fatigue Life and Strength of Case-Hardened Gear Materials with Computer, *Eng. Fract. Mech.,* Vol 39 (No. 5), 1991, p 867–874
- G. Mesnard, Super Puma MK II: Rotor and Gearbox Fatigue, *Conf. Proc. ICAF '91: Aeronautical Fatigue—Key to Safety and Structural Integrity,* Ryoin Co., Tokyo, 1991, p 195–215
- M. Nakamura et al., Effects of Alloying Elements and Shot Peening on Fatigue

Strength of Gears, *Kobelco Technol. Rev.,* No. 16, 1993, p 14–19

- Y. Obata et al., Evaluation of Fatigue Crack Growth Rate of Carburized Gear by Acoustic Emission Technique, *Acoustic Emission: Current Practice and Future Directions,* STP 1077, ASTM, 1989, p 261–270
- Y. Okada et al., Fatigue Strength Analysis of Carburized Transmission Gears, *JSAE Rev.,* Vol 11 (No. 3), 1990, p 82–84
- W. Pizzichil, Testing Gear Teeth for Low-Cycle Fatigue and Static Bending Strength, *Heat Treating,* Vol 24 (No. 2), 1992, p 14–20
- H.J. Rose, Vibration Signature and Fatigue Crack Growth Analysis of a Gear Tooth Bending Fatigue Failure, *Conf. Proc. Current Practices and Trends in Mechanical Failure Prevention,* Vibration Institute, Willowbrook, IL, p 235–245
- M. Roth and K. Sander, Fatigue Failure of a Carburized Steel Gear from a Helicopter Transmission, *Handbook of Case Histories in Failure Analysis,* Vol 1, ASM International, 1992, p 228–230
- G.M. Tanner and J.R. Harty, Contact Fatigue Failure of a Bull Gear, *Handbook of Case Histories in Failure Analysis,* Vol 2, ASM International, 1993, p 39–44
- D.P. Townsend, "Improvement in Surface Fatigue Life of Hardened Gears by High-Intensity Shot Peening," TM-105678, National Aeronautics and Space Administration, 1992
- D.P. Townsend and J. Shimski, "Evaluation of Advanced Lubricants for Aircraft Applications Using Gear Surface Fatigue Tests," TM-104336, National Aeronautics and Space Administration, 1991
- V.I. Tsypak, Fretting Corrosion and Fatigue of Gears, *Mater. Sci. (Russia),* Vol 29 (No. 6), 1993, p 680–681
- J.-G. Wang, Z. Feng, and J.-C. Zhou, Effect of the Morphology of Carbides in the Carburizing Case on the Wear Resistance and the Contact Fatigue of Gears, *J. Mater. Eng. (China),* Dec 1989, p 35–40
- T.P. Wilks et al., Conditions Prevailing in the Carburising Process and Their Effect on the Fatigue Properties of Carburised Gears, *J. Mater. Process. Technol.,* Vol 40 (No. 1–2), 1994, p 111–125
- J.J. Zakrajsek, D.P. Townsend, and H.J. Decker, An Analysis of Gear Fault Detection Methods as Applied to Pitting Fatigue Failure Data, *Conf. Proc. The Systems Engineering Approach to Mechanical Failure Prevention,* Vibration Institute, Willowbrook, IL, p 199–208

Gear Durability and Failure

- J.W. Blake and H.S. Cheng, A Surface Pitting Life Model for Spur Gears, Part II: Failure Probability Prediction, *J. Tribol.,* Vol 113 (No. 4), 1991, p 719–724
- R. Gnanamoorthy and K. Gopinath, Surface Durability Studies on As-Sintered Low Alloy Steel Gears, *Non-Ferrous Materials,* Vol 6, *Advances in Powder Metallurgy & Particulate Materials,* Metal Powder Industries Federation, 1992, p 237–250
- G.M. Goodrich, Failure of Four Cast Steel Gear Segments, *Handbook of Case Histories in Failure Analysis,* Vol 2, ASM International, 1993, p 45–52
- C. Madhavan and K. Gopinath, Surface Durability of Through-Hardened P/M Gears, *Powder Met. Sci. Technol.,* Vol 2 (No. 4), 1991, p 30–44
- T. Nakatsuji, A. Mori, and Y. Shimotsuma, Pitting Durability of Electrolytically Polished Medium Carbon Steel Gears, *Tribol. Trans.,* Vol 38 (No. 2), 1995, p 223–232
- H.V. Somasundar and R. Krishnamurthy, Surface Durability of HSLA Steel Gears, *Conf. Proc. Ninth National Conference on Industrial Tribology,* Central Machine Tool Institute, Bangalore, India, 1991, p M87–M95
- Y.L. Su, J.S. Lin, and S.K. Hsieh, The Tribological Failure Diagnosis of Spur Gear by an Expert System, *Wear,* Vol 166 (No. 2), 1993, p 187–196
- A. Verma and R.K. Sidhu, Failure Investigation of Gears, *Tool and Alloy Steels,* Vol 27 (No. 6), 1993, p 157–160
- K. Wozniak, Worm Gears of Increased Durability, *Archiwum Budowy Maszyn,* Vol 38 (No. 1), 1991, p 5–13

Gear Materials, Properties, and Manufacture
J.R. Davis, editor, p311-327
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CHAPTER 15

Mechanical Testing

MECHANICAL TESTS are performed to evaluate the durability of gears under load. One of the primary uses of gear testing is to generate performance data, which are used to develop design-allowable stresses. Design-allowable stresses are used as a guide for determining the size, geometry, material, manufacturing process, and process conditions for new designs during the design process. Gear testing is also used to compare the performance of a new attribute or characteristic introduced in the gear to a baseline, which defines the part performance with the old or existing attributes or characteristics. These new attributes or characteristics may be related to tooth geometry, gear material, manufacturing processes, and even process parameters used to manufacture the gear. In the aerospace industry, gear testing forms the minimum basis for accepting a change in these characteristics and is usually termed "qualification" of a proposed or recommended change.

Mechanical testing of gears is conducted under different scenarios. The first scenario is termed *rig testing,* which is extensively discussed and defined in this chapter. In rig testing, the gear is subjected to simulated loading that exercises a distinct mode of gear failure. While this simulation may not always be successful for a variety of reasons, data on the resistance of the gear to a distinct mode of failure allow the analysis of the specific characteristics that need to be altered to change its performance.

After rig testing is complete, it is customary to incorporate the gear into a transmission and subject the transmission to *bench testing.* In bench testing, the input and output of the transmission under test are simulated using various devices for power input and absorption. After bench testing, it is not unusual to subject the transmission to *full-scale testing,* where the transmission is subjected to controlled but actual operating conditions to determine the performance of the specific gear. Both bench and full-scale testing represent very specific application-oriented testing and are usually carried out by the organization manufacturing a specific product. It is obvious that rig testing, bench testing, and full-scale testing represent increasing orders of cost. Consequently, in most situations, extensive rig testing is followed by limited bench testing and even more limited full-scale testing. While the testing methods described in this chapter are particularly applicable to carburized and hardened steel gears, the described techniques are applicable to through-hardened steel and gears fabricated from other materials, with modifications.

Common Modes of Gear Failure

Gear tooth failures occur in two distinct regions—the tooth flank and the root fillet. Both of these types of failures are briefly described in this section. More detailed information can be found in Chapter 14, "Gear Failure Modes and Analysis," and in Ref 1–6.

Failure Modes on Tooth Flanks. In order to understand the common modes of failure on tooth flanks, it is important to study the kinematics of a gear mesh, illustrated in Fig. 1 for a pair of spur gears. For the direction of rotation shown, contact starts at the left side of the figure and progresses to the right. Initial contact occurs well below the pitch diameter of the driving tooth and well above the pitch diameter of the driven tooth. There is significant sliding along with rolling contact at this point. Sliding on the surface of the driving gear is in the same direction as rolling. As rotation continues, the point of contact moves toward the tip of the driving gear and toward the root of the driven gear. Sliding decreases as the point of contact moves toward the pitch diameter, reaching zero (pure rolling) at the pitch diameter. As the point of contact moves above the pitch diameter of the driving gear, the direction of sliding reverses. At the end of contact, significant sliding again occurs. Tests have shown that sliding in the same direction as rolling (as occurs below the pitch diameter of the driving gear) has a more severe effect on the surface durability of the material.

Failure modes on tooth flanks are caused by this combination of rolling and sliding. The more common failure modes are scoring (or scuffing), wear, and pitting.

Failure Modes in Root Fillets. Failure of the gear tooth also can occur in the root fillet. This failure is primarily due to bending fatigue but can be precipitated by sudden overloading (impact). Sudden overloads can be caused by a large foreign object (e.g., a broken tooth) being drawn through the mesh, the driving or driven shafts suddenly stopping, or sudden loss of alignment (failure of an adjacent bearing).

Stress Calculations for Test Parameters

Gear-rating standards, such as American National Standards Institute (ANSI)/American Gear Manufacturers Association (AGMA) 2001-C95 ("Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth") and International Organization for Standardization (ISO) 6336, base gear performance on contact and bending stresses. Nominal stresses are determined from first principles and are then modified to allow for the realities of manufactured gears running in actual gearboxes. This modification approximates a reasonable upper limit for the range of stress variation.

ANSI/AGMA 2001-C95 contains factors in the fundamental stress formulas for contact and bending to address overload, gear dynamics, size, load distribution, surface condition (contact only), and rim thickness (bending only). Design-allowable stress numbers are modified by further factors that address intended life, operating temperature, reliability, safety factor, and hardness ratio (contact only). It is normal practice in determining stresses to be reported with rig test results to take all of these factors as unity. The intent of rig testing is to compare the effect of various attributes or characteristics on performance. Thus, it is reasonable to determine the actual stress based on a value of unity for each factor, then determine suitable values for the factors based on test results.

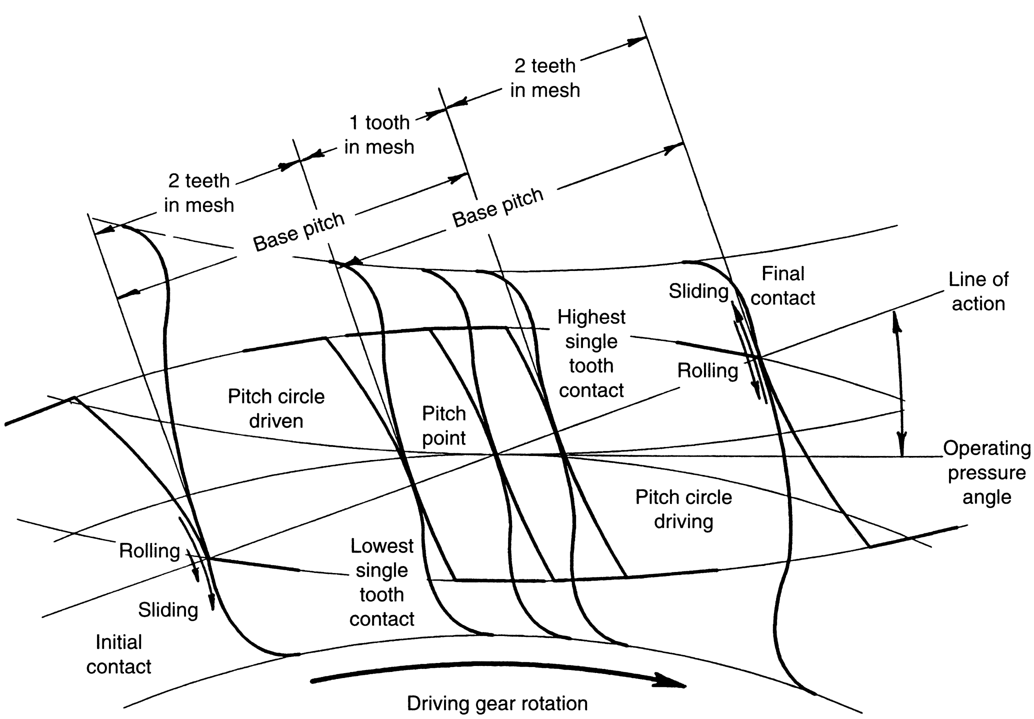

Fig. 1 Kinematics of a gear mesh

The factors that are in essence being ignored by this approach represent real phenomena that occur as gears operate. It is critical to address this fact in the design of test rigs and in their day-to-day operation. For example, it is important to conduct tests in a manner that precludes overloading; in a set of tests intended to evaluate the effect of overloading, only controlled overloads can be permitted. Most of the other factors can be handled in the same manner. However, gear dynamics and load distribution need to be addressed in a more direct fashion because they both can introduce uncontrolled variation in a set of test results.

Gear dynamics present a problem in determining the proper stress to be related to performance. Rating standards provide guidance on dynamic load variations in normal gearboxes. Gear test rigs, as described here, load one gearbox against another to keep power requirements to a minimum. Thus, there are two or more sets of dynamic variations, plus their interactions, to be considered. If the shafts connecting the gearboxes are quite rigid, the dynamic loading can be greater than the applied loading, even when precision gears are tested. Owing to the complexity of the situation, stresses for rig tests are computed making no allowance for dynamics. The fact that the dynamics of gear test rigs are not the same as real machines that use gears is a principal reason that bench testing and full-scale testing are required before a given gear design is adopted for critical applications.

The load distribution factor accounts for variations in alignment due to deflections of teeth, gearbox, and so on for variations in axial alignment of gear teeth, and for variations in load sharing between teeth. Much of the variation in load distribution can be minimized by straddle mounting test gears between bearings so their axes remain parallel under loading, modifying the profiles of test gears to optimize load sharing and so on. However, concentration of contact load at one end of a gear tooth (due to a small variation in axial alignment) produces a bending stress concentration. To avoid this condition, one or both test gears can be crowned to keep the load at the center of the face width. In this condition, the load-distribution factor is taken as unity for bending stress, and a suitable factor is estimated as shown in the following section for contact stress. However, crowning adds an additional variable that could affect test repeatability. It may be more expensive to accurately crown (and measure for verification) these test gears. Many times it may be more feasible to use high-accuracy standard test gears and very stiff (lateral) mounting to ensure uniform load distribution.

Contact Stress Computations for Gear Tooth Flank. Contact stresses on the surfaces of gear teeth are determined using Hertz's solution for the stress between contacting cylinders. The equation for contact stress taken from this solution, reduced to the simplest form applicable to spur gear teeth, is:

$$\text{Contact stress} = (P' \times B)^{1/2} \times EF \qquad \text{(Eq 1)}$$

where the contact force per unit width, P', is:

$$\frac{\text{Pinion torque}}{\text{Pinion base radius} \times \text{effective face width}} \qquad \text{(Eq 2)}$$

and

$$B = \frac{R_1 + R_2}{R_1 \times R_2} \qquad \text{(Eq 3)}$$

where B is the geometry factor, R_1 is the radius of the pinion contact surface at the point of interest, and R_2 is the radius of the gear contact surface at the point of interest.

$$EF = \left[\pi\left(\frac{1-\upsilon_1^2}{E_1} + \frac{1-\upsilon_2^2}{E_2}\right)\right]^{-1/2} \qquad \text{(Eq 4)}$$

where EF is the elasticity factor, E_1 is the pinion modulus of elasticity, υ_1 is the pinion Poisson's ratio, E_2 is the gear modulus of elasticity, and υ_2 is the gear Poisson's ratio. For pinion and gear (both steel), the EF is 2290 in U.S. customary units (P' in lb/in., B in in.$^{-1}$, and stress in psi), and EF is 190.2 in Système International d'Unités (SI) units (P' in N/mm, B in mm^{-1}, and stress in MPa).

The most critical combination of contact load and sliding in spur gears occurs on the flank of the driving gear at the lowest point of contact with one pair of teeth in mesh (the location labeled "lowest single tooth contact" in Fig. 1). The radii of the contact surfaces at this point can be determined from the corresponding roll angles for the gear and pinion. The radius of curvature is the roll angle (in radians) times the base radius. The radii used in determining the geometry factor are thus:

$$R_1 = (OR_P^2 - BR_P^2)^{1/2} - \frac{2 \times \pi \times BR_P}{N_P} \qquad \text{(Eq 5)}$$

$$R_2 = \sin(OPA) \times OCD - (OR_P^2 - BR_P^2)^{1/2} + \frac{2 \times \pi \times BR_G}{N_G} \quad \text{(Eq 6)}$$

where OR_P is the pinion outside the radius, BR_P is the pinion base radius, N_P is the number of pinion teeth, OCD is the operating center distance, BR_G is the gear base radius, N_G is the number of gear teeth, and OPA is the operating pressure angle, equal to $\cos^{-1}(BR_P + BG_G)/OCD$.

The contact stress returned by Eq 1 is a nominal stress and assumes uniform distribution of load. With crowned gears this uniform distribution will not be the case. The contact area becomes an ellipse, or in most cases, the center portion of an ellipse, with the highest contact stress at the center of the face width and progressively lower contact stress toward the ends. The proportions of the ellipse can be determined from a more complex form of Hertz's solution to the contact stress problem. A second geometry factor, A, related to the radius of curvature along the lead of the gears is needed:

$$A = \frac{R_1' + R_2'}{R_1' \times R_2'} \text{ or } A = \frac{1}{R'} \quad \text{(Eq 7)}$$

where R_1' is the radius of curvature along the pinion lead, R_2' is the radius of curvature along the gear lead, and R' is the radius of curvature along the lead (only one gear crowned).

The amount of crown is usually a very small proportion of the face width; thus, the radius of curvature is large, and the section that can be measured represents an infinitesimal angular segment of the circle. This geometry makes it difficult for coordinate measuring machines (CMMs) to determine the crown radius. A useable crown radius can be determined from lead traces and Eq 8:

$$\text{crown radius} = \frac{(\text{effective face width})^2}{8 \times (\text{average drop at ends of effective face})} \quad \text{(Eq 8)}$$

The following equations regarding the contact ellipse are empirical in nature and assume that the ratio B/A is appreciably greater than 20, which will be the case with most crowned gears. If this is not the case, a thorough presentation of Hertz's contact solution, such as Ref 7, should be consulted.

$$k = \left(\frac{B}{A}\right)^{-0.62} \quad \text{(Eq 9)}$$

where k is the ratio of ellipse minor to major axis.

$$\text{contact stress (crowned)} = \text{contact stress} \times \left[1 + \frac{B \times (\pi \times k \times EF \times \text{effective face width})^2}{48 \times (1 + k) \times P'}\right]^{1/2} \quad \text{(Eq 10)}$$

If the crown radii are small enough in proportion to the contact load and effective face width, the contact area will be an ellipse narrower than the effective face width. The width of the elliptical contact area and the corresponding contact stress for this case can be determined from Eq 11, 12, and 13.

$$\text{contact force} = \frac{\text{pinion torque}}{\text{pinion base radius}} \quad \text{(Eq 11)}$$

Width of contact ellipse, EW, is determined by:

$$2 \times \left[\frac{3 \times \text{contact force}}{B \times (\pi \times k \times EF)^2}\right]^{1/3} \quad \text{(Eq 12)}$$

$$\text{contact stress (crowned)} = \frac{6 \times \text{contact force}}{\pi \times k \times EW^2} \quad \text{(Eq 13)}$$

These formulas can compute contact stresses for rig test specimen gears far higher than would be predicted by the load distribution factors presented in gear-rating standards. The formulas are presented here to permit comparison of performance on a stress basis between gears with differing crown. Stresses determined in this manner should be reduced by a typical load distribution factor before being used to develop allowable design stresses.

In most cases, there will be variation in contact stress between specimen gears in the same lot (and from tooth to tooth on each gear) tested at the same load. The edge break at the ends of teeth is frequently applied in a manual operation; thus, the effective face width can vary. Crown can vary from tooth to tooth on each gear. Center distance can vary due to radial runout in shafts and bearings. The outside diameter of gears can vary (as required for manufacturing tolerance). It is important, therefore, to determine the mean contact stress and the probable range of variance.

Bending stress computations for root fillets are computed based on the assumption that the gear tooth is a cantilever beam with a stress concentration at its supported end. AGMA rating standards determine the form of the cantilever beam from the solution presented by W. Lewis (Ref 8) and use a corresponding stress concentration factor. ISO and Deutsche Industrie-Normen (German Industrial Standards) (DIN) use different proportions for the beam and determine the stress concentration factor in a different manner. Only the AGMA approach is discussed here.

Figure 2 shows a spur gear tooth with a point load applied at the highest point of single-tooth contact. This point of loading corresponds to the highest bending stress when there is effective load sharing between gear teeth. Specimen gears used in rig tests should have effective load sharing, so this is the appropriate point of loading for determining bending stress in rig tests. For gears tested in single-tooth bending fatigue, the actual point of loading established by the test fixture should be used in calculating bending stresses.

The Lewis parabola is drawn from the point at which the load line intersects the center of the gear tooth and is tangent to the root fillet. Methods used to lay out this parabola depend on how the root form is generated, the particulars of which are presented in detail elsewhere (Ref 9). The critical height and width are determined from the Lewis parabola as shown in Fig. 2. The angle between the load line and a normal-to-the-tooth center is termed the *load angle* (it differs from the pressure angle at the point of loading because of the thickness of the tooth). The bending stress is thus:

$$\frac{\text{load} \times \cos(\text{load angle})}{\text{face width}} \left[\frac{6 \times h}{s^2} - \frac{\tan(\text{load angle})}{s} \right] K_f \qquad \text{(Eq 14)}$$

where s is the critical width from the Lewis parabola, h is the critical height from the Lewis parabola, and K_f is the stress concentration factor:

$$H + \left(\frac{s}{r}\right)^L \left(\frac{s}{h}\right)^M \qquad \text{(Eq 15)}$$

where r is the minimum fillet radius, H is equal to 0.331 – 0.436 × (nominal pressure angle, in radians), L is equal to 0.324 – 0.492 × (nominal pressure angle, in radians), and M is equal to 0.261 + 0.545 × (nominal pressure angle, in radians).

Equation 14 is derived from first principles but also can be derived from AGMA standards by taking the forms of relevant formulas pertinent to spur gears and setting all design factors at unity. While a similar formula could be developed for helical gears, testing them usually represents a step between rig testing and bench testing, and such tests are negotiated between the client and the testing laboratory.

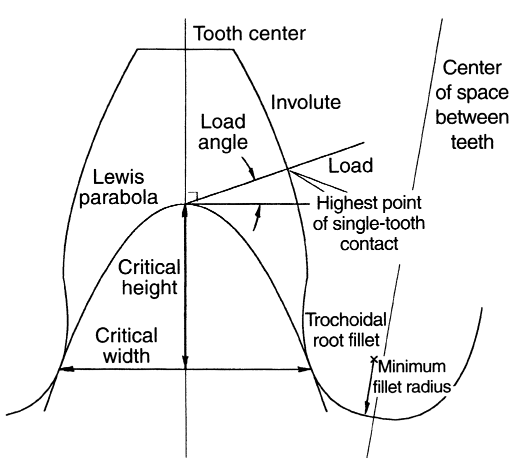

Fig. 2 Layout of Lewis parabola for tooth-bending stress calculation

Specimen Characterization

Specimen characterization is a critical part of any fatigue test program because it enables meaningful interpretation of the results. It is important to know that the specimens to be tested meet specifications, where key parameters fall in the specified range, and what the variations are. Characterizations fall into four areas: dimensional, surface finish texture, metallurgical, and residual stress.

Dimensional Characterization. Basic dimensional checks include size and alignment of mounting surfaces, size and alignment of test surfaces relative to mounting surfaces, and size and uniformity of edge breaks at the edges of test surfaces. These checks ensure that the specimens will fit into the test rig and either ensures that stresses will be consistent or provides data to determine variations.

The important dimensions to check on rolling contact fatigue (RCF) specimens are diameters of mounting trunnions and test surface, total indicator reading (TIR) test surface-to-mounting trunnions, and overall length. The important dimensions to check on the mating load rollers are inside and outside diameters, TIR inside to outside, crown radius and location of high point of crown, and thickness. In the case of nonstandard specimens where the entire width of the test surface is intended to make contact, the size and uniformity of the edge break needs to be checked.

The important dimensions to check on gear specimens fall into the areas of mounting surfaces, gear functional charts, and other specific features impacting stress. All of the gear specimens described in the following sections are mounted on their inside diameter. Thus, inside diameter, length between end faces, and perpendicularity of end faces to inside diameter need to be checked. For specimens with a splined bore, the pitch diameter, or inside diameter if so specified, counts as the inside diameter for alignment checks. The gear functional charts are developed with the gear mounted on the test mounting surfaces. Typical measurements include lead, profile, spacing, index, and runout. Other specific items to check include diameter over wires, root diameter, and size and uniformity of edge break at tooth ends and along the tip.

Surface Finish/Texture Characterization. It has been standard practice to specify surface finish and texture in terms of arithmetic average surface roughness (R_a). Because it is possible to manufacture surfaces with widely varying load capacity, all with the same R_a, more information than R_a is needed to interpret data. At a minimum, R_z (a measure of "average" peak roughness) should also be measured. A preferable method would be to examine surfaces with an optical interference system that maps the contour of an area rather than the traditional line contact. Test surfaces to be subject to contact fatigue should also be examined at 10× to ensure they are free of corrosion, dings, scratches, and so on.

Metallurgical Characterization. Both surface hardness and hardness gradients are to meet specification. For gear specimens, hardness should be checked at the tooth half height and at the root fillet. Details of the microstructure should be characterized at the same locations as hardness. Typical practice has been to prepare and document photomicrographs of the etched structure at 400× to 600×.

Residual Stress Measurement. Residual stresses vary among a lot of parts; therefore, enough measurements should be made to permit a statistical analysis. Residual stress gradients provide more useful information than surface measurements but at a much higher price. A complete study should include some residual stress gradient data. Typical practice is to measure surface residual stress at six points per variant and measure the residual stress gradient at one or two of these points. Contact fatigue specimens should be measured at the center of the contact surface. Gear specimens should be measured at the tooth half height (or lowest point of single-tooth contact) for surface durability tests and at the midpoint of the root fillet for bending strength tests.

Tests Simulating Gear Action

Tests that simulate gear action are the rolling contact fatigue test, the single-tooth fatigue test, the single-tooth single-overload test, and the single-tooth impact test.

Rolling Contact Fatigue (RCF) Test

The RCF test simulates the rolling/sliding action that occurs in a gear mesh, and depending on the test parameters employed, it can simulate the most severe condition in the mesh. Because the specimens are cylindrical, this test is the least expensive to run in both cost and time. Hence, it is frequently used as a screening test, and its results are used to plan further gear testing.

Test Equipment. Figure 3 shows a schematic of the rolling/sliding contact fatigue test. The specimen and load rollers are cylindrical. The outside diameter of the load roller is crowned to concentrate the load at the center of contact and eliminate the possibility of concentrated loading at the edge of contact due to misalignment. A normal load is applied by air pressure. Phasing gears, attached to the shafts on which the specimen and load rollers are mounted, control the extent of sliding at the specimen/load roller interface. The amount of sliding is determined by the gear ratio and can be fixed at any value (21 and 43% are commonly selected), simulating the rolling/sliding action at any desired location on gear teeth. This test is primarily intended to evaluate the resistance of candidate material systems to surface origin pitting, but other modes of surface failure can be studied. The surfaces of the specimen and load roller are sliding against each other, with lubrication, under high load. Therefore, some indication of the lubricated wear resistance of the material being tested can be obtained. In addition, the resistance of the material/surface finish/lubricant system to scuffing can be studied. Tests are conducted at very high overload to promote failure in real time; thus, it is possible to introduce plastic deformation (rippling) on the contact surface. With specially adapted equipment, these tests can be conducted at different temperatures up to 400 °F. Specimens and load rollers can be designed for a variety of testing requirements, depending on the exact material and objective of the test.

Test Procedure. The lubricant stream to the specimen/load roller interface is aimed at the discharge side of the mesh (to cool the surfaces after contact). The lubricant flow rate is adjusted (typically to about 2 L/min, or as specified for the project), with the lubricant at test temperature.

Alignment of the specimen and load roller is checked as each test is set up. One method often used is to coat the surface of the load roller with a thin layer of layout blue and ensure that contact occurs uniformly at the center of the load roller. Before the start of testing at load, a break-in run is typically conducted. This procedure is accomplished by running the machine for 10 min at each of a series of increasing loads up to the full test load. The intent of the break-in run is to gently polish away asperities on the contacting surfaces that might cause scuffing (i.e., localized scoring), burnishing, or wear to occur under full load.

Tests are stopped periodically, and the contacting surfaces of the specimen and load roller are inspected for signs of surface distress. The diameter at the center of the specimen is also measured to provide an indication of wear. The formation of large, progressing pits produces a strong vibration signal, which, in turn, shuts down the test. Lubricant temperature, lubricant pressure, and air pressure (to apply the contact load) are also monitored, and the test is stopped if irregularities are detected.

The results of contact fatigue tests exhibit more scatter than is typical with other fatigue tests. One gage of scatter is the Weibull slope, a measure of the increase in rate of failure with increasing life. For many fatigue tests, a Weibull slope of six or more is typical. A Weibull slope of six can be roughly translated into practical terms as follows. If 20 samples were randomly selected from a large group of theoretically identical parts and tested to failure under identical conditions, the longest test would, on the average, last about twice as long as the shortest. For contact fatigue testing with current ultraclean steels, a Weibull slope of 1.5 is typical. In this example, if the Weibull slope were 1.5, the longest of the 20 tests would last approximately

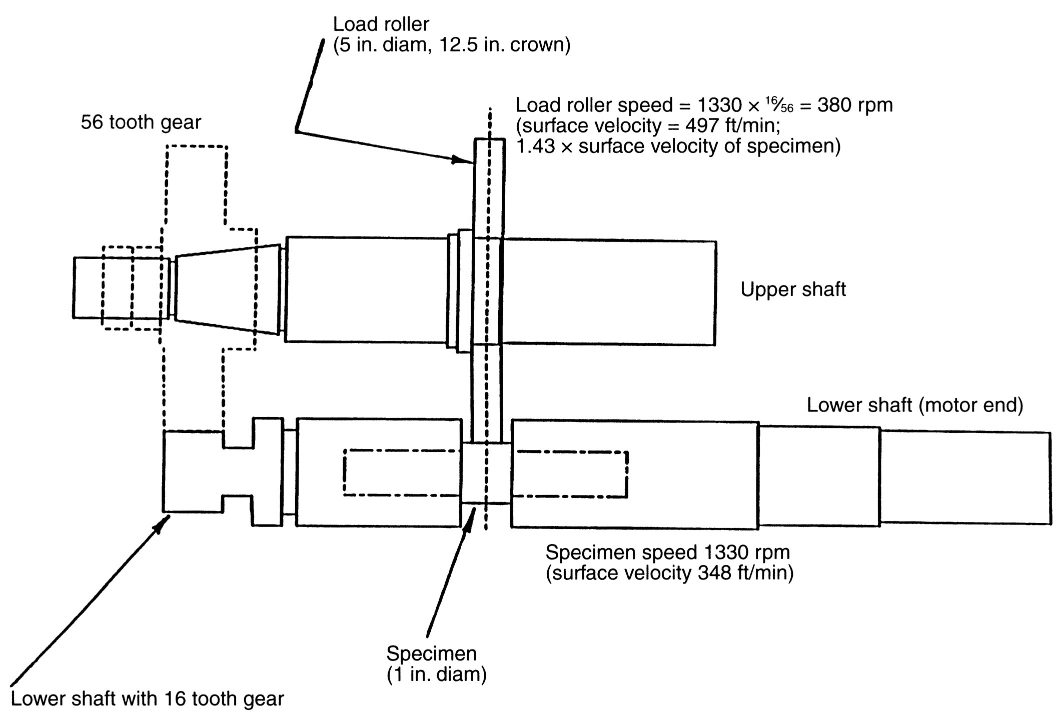

Fig. 3 Schematic of a rolling/sliding contact fatigue (RCF) test

20 times as long as the shortest. The impact on test programs is that many tests have to be conducted to develop enough data to make statistically meaningful comparisons. The minimum number of tests with each variant (at each load) to enable reasonable comparisons has typically been considered to be six. With six tests for each of two variants and Weibull slopes of 1.5, the G50 life of one variant would have to be twice that of the other to conclude that it is better. If the comparison were made on the basis of G10 life, the ratio would have to be 5 to 1.

Test Results. An example set of data, representative of the results of rolling/sliding contact fatigue testing, is shown in Table 1. In addition to specimen identification, load, and life to failure, other information is reported to aid in the interpretation of results. Several test parameters can be varied as needed; thus, the actual values are reported with the results. These parameters include lubricant, lubricant bulk temperature, nominal filter size, test speed, and phasing gear set/slide ratio. Also reported for each test are data regarding surface roughness, ratio of elastohydrodynamic lubricant film thickness to composite roughness, wear, and wear rate. A Weibull statistical analysis of this set of results is shown graphically in Fig. 4. Comparisons between variants are made on either a G10 life or a G50 life basis.

Single-Tooth Fatigue (STF) Test

The STF test is used to generate a statistically significant quantity of bending fatigue data at a comparatively low price. Teeth are tested one at a time with a fixed loading point, allowing the generation of bending fatigue data at comparatively high cycles without risk of losing tests to other modes of failure. Another cost-saving measure is that four or more tests can be conducted with each gear specimen.

Test Equipment. A gear is placed in a fixture so that one tooth at a time can be loaded while another tooth supports the reaction. The test is usually done in an electrohydraulic, servocontrolled universal test machine. The primary object of this test is to determine fatigue properties in bending. However, the same setup can be used to determine single overload properties (ultimate bending strength) as well. Frequently, enough teeth are tested to develop a stress-cycles diagram to define the bending fatigue characteristics of the material system.

Several arrangements for loading can be considered. One fixture arrangement is shown in Fig. 5. This illustration shows the Boeing flexural design, which appears to have found favor with the aerospace sector. This fixture is designed for a 32 tooth, 5.333 diametral pitch (DP). 8 mm (⅜ in.) face width spur gear with several teeth removed to provide access to test and reaction teeth. The gear is rigidly supported on a shaft. Load is applied through a carbide block contacting the test tooth at the highest point of single-tooth contact. The loading block is held in the specified orientation to the gear by a flexural loading arm. This flexural design ensures accurate loading of the gear tooth with minimal migration of the point of loading. Reaction is carried through a block contacting the reaction tooth at the lowest point of single-tooth contact. Load is cycled from the specified test load to a minimum load high enough to keep the slack in the system taken up (usually 10% of the test load). While most testing is conducted at 20 Hz, other frequencies also are possible. The fatigue test machine is instrumented to monitor instantaneous loads and tooth deflections. Changes in

Table 1 Example of rolling/sliding contact fatigue test data

Lubricant, automatic transmission fluid. Bulk temperature, 90 °C (194 °F). Filter, 10 mm (nominal). Test speed, 1330 rpm. Phasing gear set, 16 tooth/56 tooth. Slide/roll ratio, 43%. Contact stress, 400 ksi. Load, 3000 lb.

		Test duration		R_a(a)					
Test No.	Specimen No.	10^6 cycles	h	Specimen	Load roller	λ(b)	Wear(c), in.	Wear rate(d)	Comments
		1.173	14.7	13.5	3.8				
3	10-9/LR 1	1.603	20.2	8.5	. . .	0.58	0.0002	12	Surface origin pitting
8	10-11/LR 5	7.332	92.3	9.3	. . .	0.56	0.0001	1.4	Surface origin pitting
12	10-4/LR 13	7.565	95.3	. . .	5.0	0.57	0.0008	12	Surface origin pitting. Severe exfoliation and wear at center of wear track
18	10-2/LR 17	5.267	66.3	9.0	6.8	0.51	0.0002	3.8	Surface origin pitting
22	10-3/LR 22	6.322	79.7	. . .	5.0	0.56	0.0003	4.7	Surface origin pitting

(a) R_a, arithmetic average surface roughness measured in axial direction. μin. (b) λ, ratio of elastohydrodynamic film thickness to composite surface roughness. In cases where surface roughness was not measured for a particular specimen or load roller. λ is based on average surface roughness for group. (c) Diametral wear (i.e., change in diameter) on specimen. (d) Change in diameter, in. × 10^{-5}/cycles × 10^6

compliance can be used for monitoring crack initiation and propagation in the root fillet region. In addition, a crack wire can be incorporated to monitor catastrophic tooth failure. Typical fatigue load capacity of such types of equipment is in the range of 10 to 20 kips, although higher loads, up to 110 kips, can be used for single overload tests.

A second fixture arrangement (Fig. 6) appears to have found favor with many other industry segments. This fixture utilizes a 34 tooth, 6 DP, 25 mm (1 in.) face width spur gear and is derived directly from the Society of Automotive Engineers (SAE) Division 33 STF fixture. Fatigue test loads up to about 15 kips are feasible with this fixture, and single overload tests up to about 50 kips can be accommodated. These STF fixtures are compact enough to be immersed in heated fluid; thus, fatigue testing can be conducted at elevated temperatures (up to 400 °F).

Test Procedure. Special concern must be taken with regard to safety in conducting the STF test (in addition to general lab safety procedures). Setting up the test requires moving the hydraulic load ram while lining up the fixture, bypassing safety features, such as a light curtain to stop the machine, and so on, and considerable caution needs to be exercised during setup. Periodic calibration of the test fixture is important because there are many closely fitted parts that can wear and change the bending stress in specimen gears. A calibration gear is made from a standard specimen with strain gages fit at key points in the root fillets. This calibration gear is installed in the fixture and loaded to set loads periodically to ensure consistent loading. If specimens for a specific test program differ from the standard design established for the fixture, one of these specimens should also be fit-

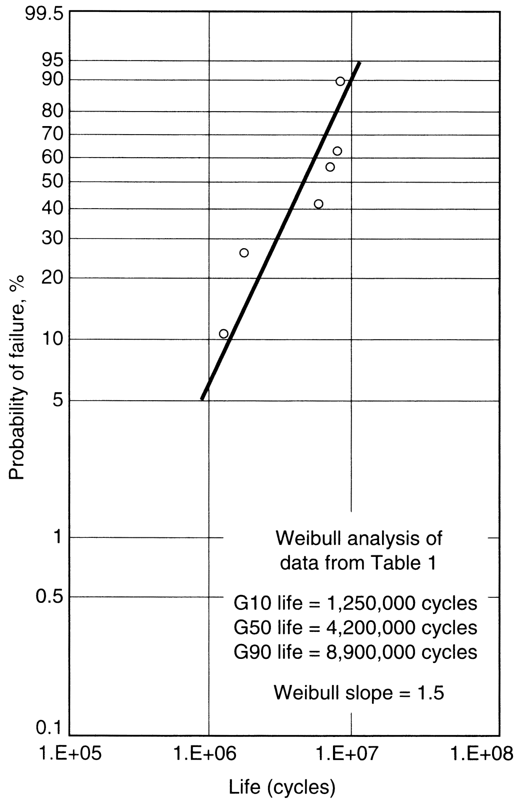

Fig. 4 Weibull analysis with RCF test data

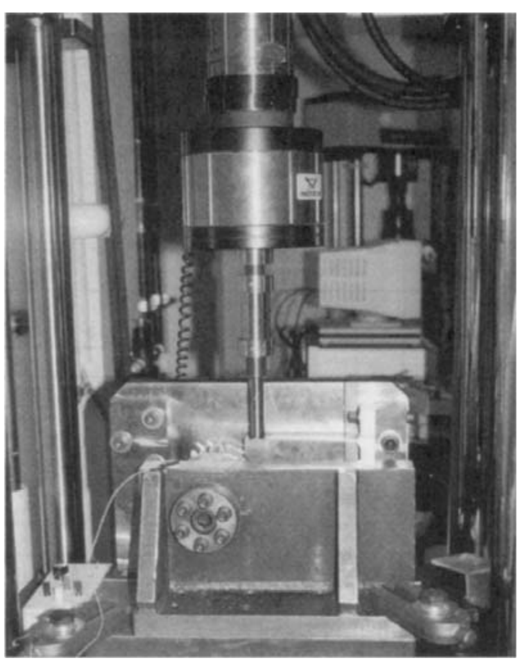

Fig. 5 Flexure arm single-tooth fatigue (STF) test fixture

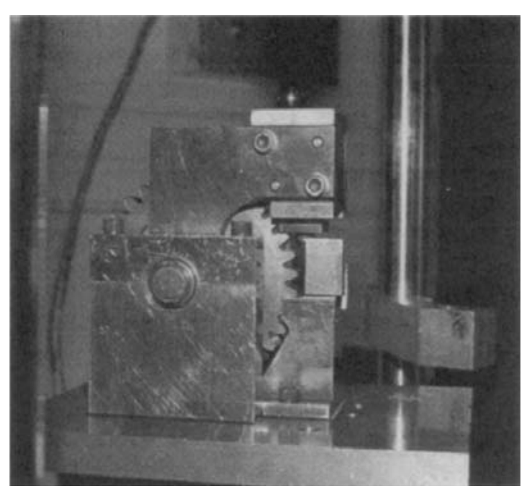

Fig. 6 SAE division 33 single-tooth fatigue (STF) test fixture

ted with strain gages, and calibrations should be conducted with both gears. Typical calibration intervals are from every six to every thirty tests, or any time the fixture is cleaned and repaired.

Other set-up items that can affect test results are mounting of the fixture on the universal test stand, tuning of the test-stand controller, and test frequency. If the SAE type fixture is used, it is important to ensure that the fixture floats freely on an oil film before starting each test. Other fixture designs can be bolted to the machine base, provided proper alignment between the load ram and loading point on the load arm is maintained. The feedback control should be tuned to suit test fixture and specimen compliance. Some systems do this automatically each time the machine is turned on; with others; this tuning may have to be done manually. At a minimum, tuning should be verified as often as fixture calibration. Some systems provide a feature to automatically adjust tuning as compliance changes. This feature should be turned off when running STF tests—a significant change in compliance is proof of failure. The maximum frequency for STF testing is limited by the capacity of the system to maintain a satisfactory load waveform. An inability to maintain peaks on the unload side of the wave at the specified value is a sign of testing at too high a frequency. All of the tests for a given project should be conducted at the same frequency, which is usually set at slightly less than the maximum for the highest anticipated load.

The other procedural item that can affect test results is the method used to detect failure and stop the test (and stop counting cycles). All systems can be set up to stop if the load goes out of the specified range. This deviation will happen when compliance changes, usually as a result of a crack in the root fillet. Many systems incorporate a linear variable differential transformer and can be set up to stop if the load ram moves beyond the specified range (also an indication of tooth failure). A limit switch can be set to trip when the load ram moves too far. Other failure detection devices, such as a crack wire bonded near the root fillet so that it will break when the tooth cracks, or an ultrasonic system to detect cracking, can be used to stop the test. At least two of these methods should be employed to ensure that the machine stops cycling load when the tooth breaks. If a computer is used to control loading during tests, one of the failure detection systems should be connected directly to the universal test-stand controller to guarantee that the machine stops if the computer crashes.

Test Results. Table 2 summarizes results from a typical set of STF tests. Testing was conducted in three phases. Initial searching tests were conducted to establish loads that would

Table 2 Results from a typical set of gear single-tooth fatigue tests showing overall testing sequence, including modified staircase ("up and down") tests

Legend: X, failure; O, runout.
Specimen serial numbers 2, 4, and 6. Specimens cut from bar stock, hobbed roots. *R*, loading; *R* = 0.1. Frequency, 25 Hz

Life, cycles $\times 10^6$	0.263	0.152	0.324	0.270	0.918	Run out	Run out	Run out	0.521	0.461	Run out	Run out	Run out	0.557	0.299	Run out	0.220	0.783	Run out	Run out	Run out	0.209	0.310	0.425	0.165	0.232	0.105	0.128		
Gear/tooth	2/1	4/1	6/1	2/2	4/2	6/2	2/3	4/3	6/3	2/4	4/4	6/4	2/5	4/5	6/5	2/6	4/6	6/6	2/7	2/8	4/7	2/9	2/10	6/7	4/8	4/9	6/8	6/9		
Load	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**	**21**	**22**	**23**	**24**	**25**	**26**	**27**	**28**	**29**	**Failure rate**
10,000 lb																														. . .
9,500 lb																														. . .
9,000 lb	X																								X	X	X	X		100%
8,500 lb		X												X								X	X	X						100%
8,000 lb			X						X				O		X		X			O										80%
7,500 lb				X				O		X		O				O		X												50%
7,000 lb					X		O				O								O											25%
6,500 lb						O														O	O									0%
6.000 lb																														. . .
Load	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**	**21**	**22**	**23**	**24**	**25**	**26**	**27**	**28**	**29**	**Failure rate**

Searching tests (tests 1–5) **Modified staircase test sequence** (tests 6–19) **Finite life and confirmation tests** (tests 20–28) result in failure in reasonable time. A "modified staircase sequence" of tests was conducted to develop data at a series of loads representing zero to 100% failure. Further tests were conducted to fill in the stress cycles relationship. Searching tests are started at a high load to ensure starting with a failure, then stepped down until the tooth survives the specified number of cycles (here, 5 million cycles has been selected as a run-out limit). The modified staircase sequence is conducted by testing three specimen gears in sequence. If the tested tooth breaks before the specified limit, the next test is conducted one load step lower. If it does not break by the specified limit, the next test is conducted one load step higher. After the modified staircase sequence is completed, additional tests are conducted to ensure that all the specimen gears are tested at the lowest load. More tests are conducted to develop enough data for Weibull analysis at two loads resulting in 100% failure.

The load-cycles diagram shown in Fig. 7 was developed from the data in Table 2. Results at 9000 and 8500 lb were analyzed via Weibull statistical analyses (similar to that shown in Fig. 4) to determine lives to 10, 50, and 90% failure. The failure rates at 5 million cycles for loads from 6500 to 8500 lb were analyzed using normal probability concepts to determine 10, 50, and 90% failure loads. The curves labeled G10, G50, and G90 were then fit visually using the results of these analyses as a guide. Results are reported in terms of load versus cycles, and load can be converted to stress using the method discussed previously. Comparisons can be made between groups of gears with the same geometry on a load-cycles basis or between gears with differing geometry on a stress-cycles basis. When converting to stress for comparison with running gear data, consideration must be given to the different stress ranges applied to STF gears and running gears. Also, consideration must be given to the statistical difference between four, eight, or more data points from a single STF specimen gear compared with one data point from a running gear specimen set.

Single-Tooth Single-Overload and Impact Testing

The single-tooth impact test is used to investigate impact strength of gear steels. Impact strength is an important property for steels to be used in high-speed gear drives or transmissions for off-highway equipment, which can be subjected to significant shock loading. It has been suggested that fatigue strength can be related to impact strength; however, results of tests to support this suggestion have been mixed.

Single overload tests are used to investigate the ultimate strength of gear materials. The results are useful in selecting loads for fatigue

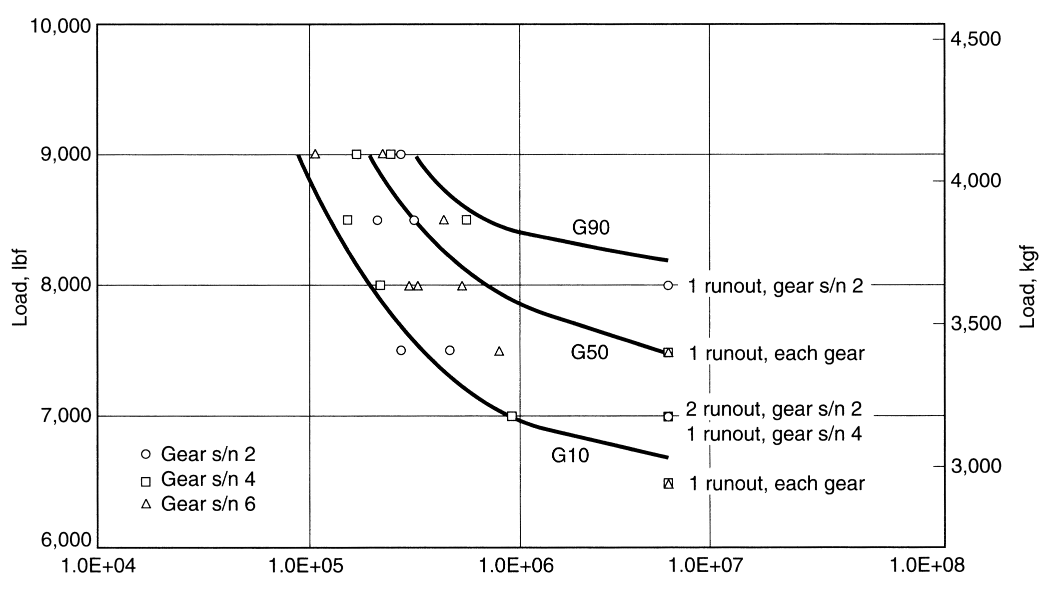

Fig. 7 Load cycles diagram constructed from STF data. See Table 2 for load-cycles relationship for data

tests and in constructing allowable stress-range diagrams (e.g., for use in comparing STF results with bending fatigue data developed from running gear tests).

Test Equipment. Figure 8 shows a closeup of the test area of a 4.5 m (15 ft) drop tower for single-tooth impact testing. A single-tooth fatigue test fixture, without loading block, load arm, and so on, is placed on a load cell and arranged so that the striking surface of the drop weight hits the load point on the test gear tooth. Stops are provided to prevent the forces required to stop the drop weight after the test tooth fractures passing through the load cell. A high-speed digital oscilloscope captures the signal from the load cell.

Single-tooth, single-overload tests are accomplished with the same equipment used in STF testing. The only change is that a high-speed digital oscilloscope captures the signal from the single event.

Test Procedure. The single-tooth fatigue test fixture, without loading block, loading arm, and so on, is placed on a load cell and installed under the drop weight. The striking surface on the drop weight is adjusted to the correct angle to strike the test tooth at the load point. This point is verified by coating the tooth with a thin layer of layout blue and allowing the drop weight assembly to contact the tooth statically. The digital oscilloscope is set up to record a single event using a change in signal as a trigger. Usually three to six tests per variant are conducted.

For single-overload tests, the STF fixture is installed in the universal test stand as for fatigue testing, and a preload is applied. For the SAE fixture, 1800 kg (4000 lb) is used, and for other fixtures the load should be high enough to take the slack out of the system. The function generator is set up to ramp up the load to break the tooth in approximately 30 s for slow bend tests and in approximately 0.050 s for fast bend tests. One slow bend and three fast bend tests usually are conducted per variant.

Test Results. A load deflection trace for a single-tooth impact test is shown in Fig. 9. Results from fast and slow bend tests are similar. This trace shows load increasing linearly with deflection up to 5900 kg (13,000 lb), then increasing at a slower rate until tooth breakage at 9000 kg (20,000 lb). This result is typical for many materials and is possibly related to the onset of plastic flow prior to fracture. Based on this, the stress corresponding to the change in slope of the load deflection (or load time) trace corresponds to the ultimate bending strength of the material at the critical point in the case. Maximum load, or energy required to break the tooth, represents the strength of the core rather than the strength of the case.

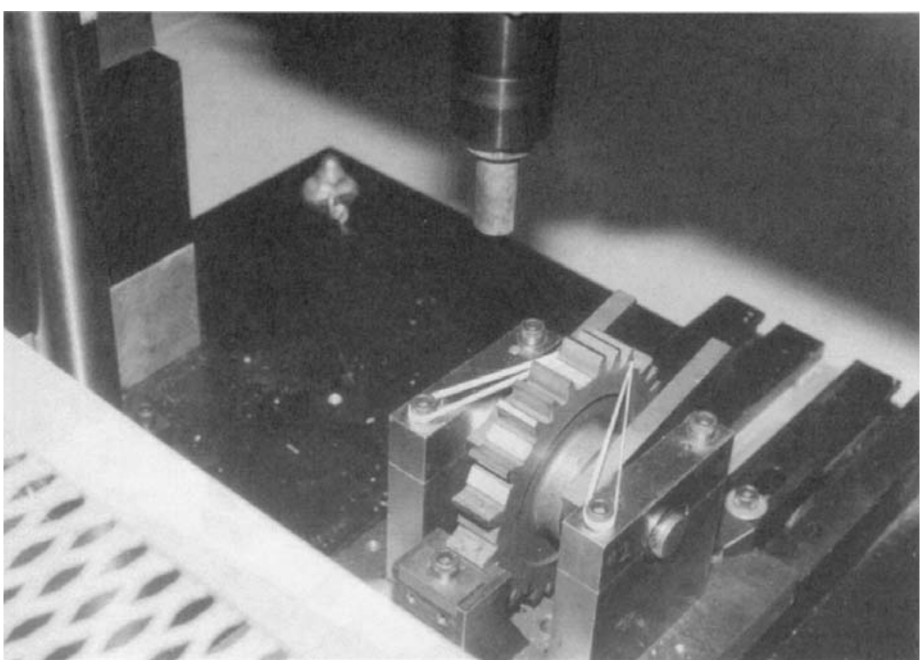

Fig. 8 Test area of an impact test tower

Gear Power-Circulating (PC), or Four-Square, Tests

A gear failure map is shown in Fig. 10. This map indicates that by varying load and speed, failure by any of several different modes can be induced. Thus, PC gear tests can be used to evaluate bending fatigue strength, surface durability with regard to pitting, or scoring resistance. A limiting factor is that tests targeted at one of the failure modes may have to be terminated because of failure by another mode. For example, bending fatigue tests conducted at a comparatively low load, targeting failure in several million cycles, may have to be terminated due to pitting failure. Consequently, PC gear testing is more expensive than the tests simulating gear action previously discussed. It is, however, less expensive than bench testing, and several standard specimen designs are available to allow tests targeted at particular failure modes.

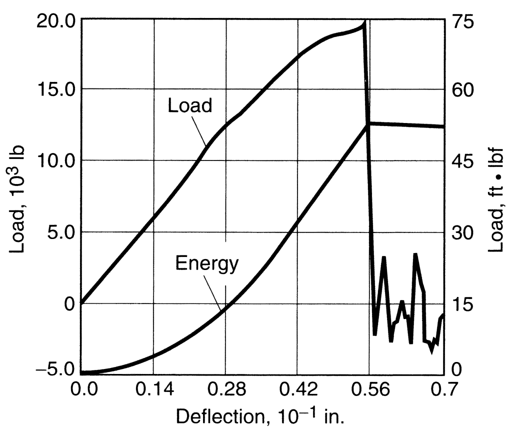

Fig. 9 Load deflection trace from single-tooth impact test

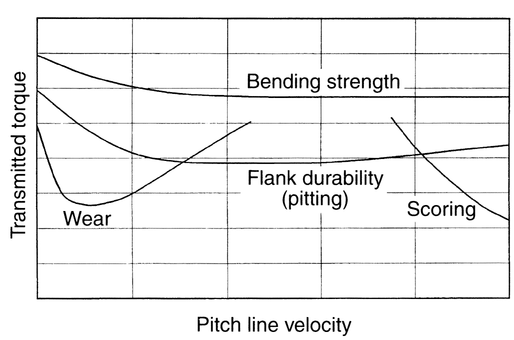

Fig. 10 Gear failure map

Test Equipment. A test and a torque-reversing gear set are mounted at opposite ends of the test bed. One shaft connects the test and torque-reversion pinions, and another connects the torque-reversing and test gears. One gearbox is loaded against the other by twisting the shafts to lock in torque. This rig is known as a power (re-) circulating, or four-square, test rig and is schematically illustrated in Fig. 11. The locked-in torque, which is fixed at various values to create the required tooth load, can be generated mechanically or hydraulically. The entire loaded arrangement is driven by an electric motor at the required speed, and the motor need only supply the friction losses. Significant transmitted power can be simulated in this system with comparatively little power input. Data from these tests can be directly translated for design use.

In a four-square test rig, the tooth accuracy of the torque-reversing gears, and their transmission error, has a significant impact on the dynamic loads on the test-gear pair. These dynamic loads due to torque reversing gear inaccuracy can be further amplified or attenuated by the longitudinal and torsional frequency response characteristics of the four-square-test rig. This intensification is of particular significance in high-speed testing where operating frequencies will be close to or at torsional and/or longitudinal natural frequencies of the four-square system. However, high-accuracy torque-reversing gears are expensive. Consequently, the PC test rigs can be classified into two categories. The first category of PC test rigs have low power, operate at low speeds, and generally have AGMA quality class Q9 or Q10 gears in the torque-reversing

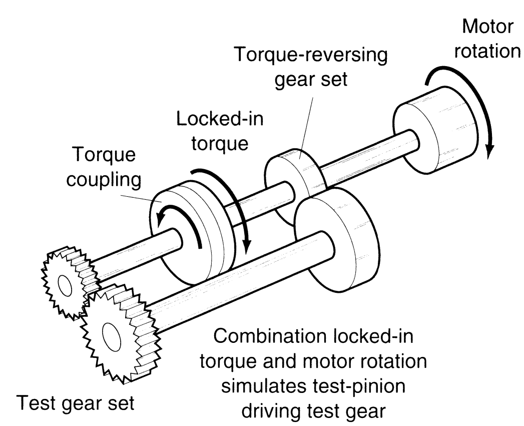

Fig. 11 Schematic of a power (re-) circulating (PC) test rig

gearbox that are quite adequate for many industry segments. The second category of PC test rigs operate at high power, high speed, and incorporate high-accuracy torque-reversing gears that are essential for gear testing in certain industry segments, such as the aerospace industry. Testing costs associated with each of the two categories of test rigs are significantly different.

The low-speed PC test rig illustrated in Fig. 12 nominally operate at 900 rpm but can be adapted to operate at 1500 rpm. The standard center distance is 100 mm (4 in.), which can be modified if necessary, although there are some costs associated with doing so. These test rigs can simulate the transmission of up to 100 hp when driven with a 3 hp motor. Load is applied via a mechanical loading arm and locked in with a special coupling that can be set at any angle. Thus, load cannot be varied as the rig is running. Depending on specimen design, contact stresses of up to 450 ksi at the lowest point of single-tooth contact and bending stresses up to 160 ksi can be generated. Splash lubrication is standard, and spray lubrication to either side of the mesh can be provided. These rigs can be adapted to conduct scoring tests.

Figure 13 shows a PC test rig with a 155 mm (6 in.) center distance that can operate up to 8000 rpm and simulate the transmission of up to 1400 hp. This test rig has two test gearboxes so that two pairs of gears can be tested simultaneously. In programs to evaluate materials for aerospace gearing, runout can be set as high as 400 million cycles for surface durability tests, making the project a lengthy exercise. In this case, obtaining two data points in one run can be a significant benefit. Figure 14 shows another PC test rig with a 90 mm (3.5 in.) center distance, capable of operating at 10,000 rpm and simulating transmission of up to 700 hp. Torque-reversing gears in these high-power rigs are in AGMA quality class Q13 and Q14. Load is applied via a servo-controlled hydraulic system and can be varied as required during tests. Torque is monitored continuously by a noncontact torque cell. This monitoring permits interpretation of test results in light of the actual load signature. Based on input lubricating oil temperature and load, surface durability (pitting), bending, and scoring tests can be conducted on these PC test rigs. Spray lubrication, to either side of the mesh, is standard; oil-mist lubrication can be provided.

Test Procedures for Surface Durability (Pitting) and Root Strength (Bending) Testing. One or both gears in the specimen set are coated with a thin layer of layout blue prior to installation in the test machine. After the gears are installed, they are run briefly at low load with room-temperature lubricant to determine the contact pattern. The low-speed test rigs allow some alignment adjustment to optimize contact pattern. The specimen mountings in the

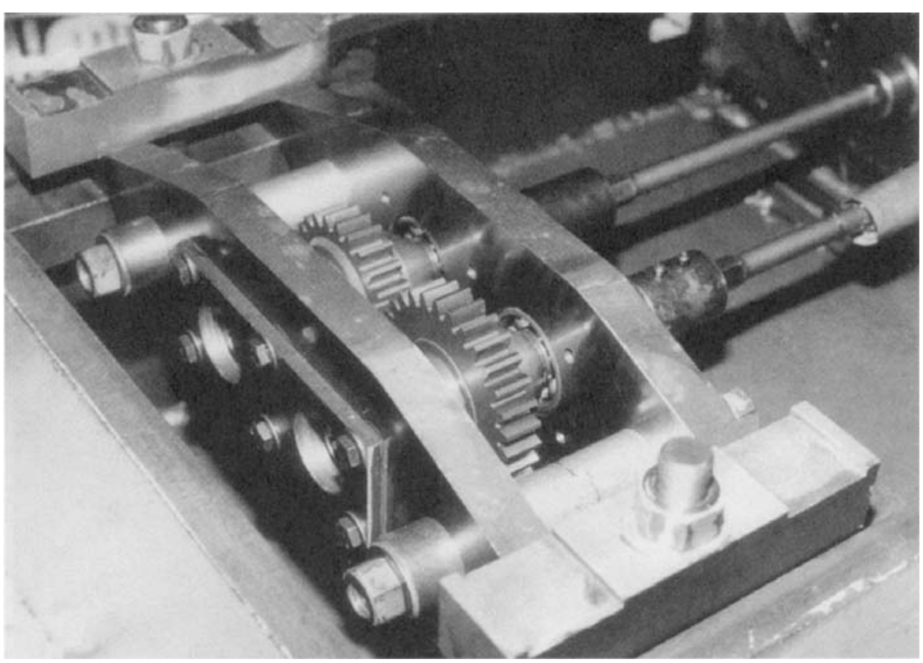

Fig. 12 Low-speed (4 in. center distance) PC test rig

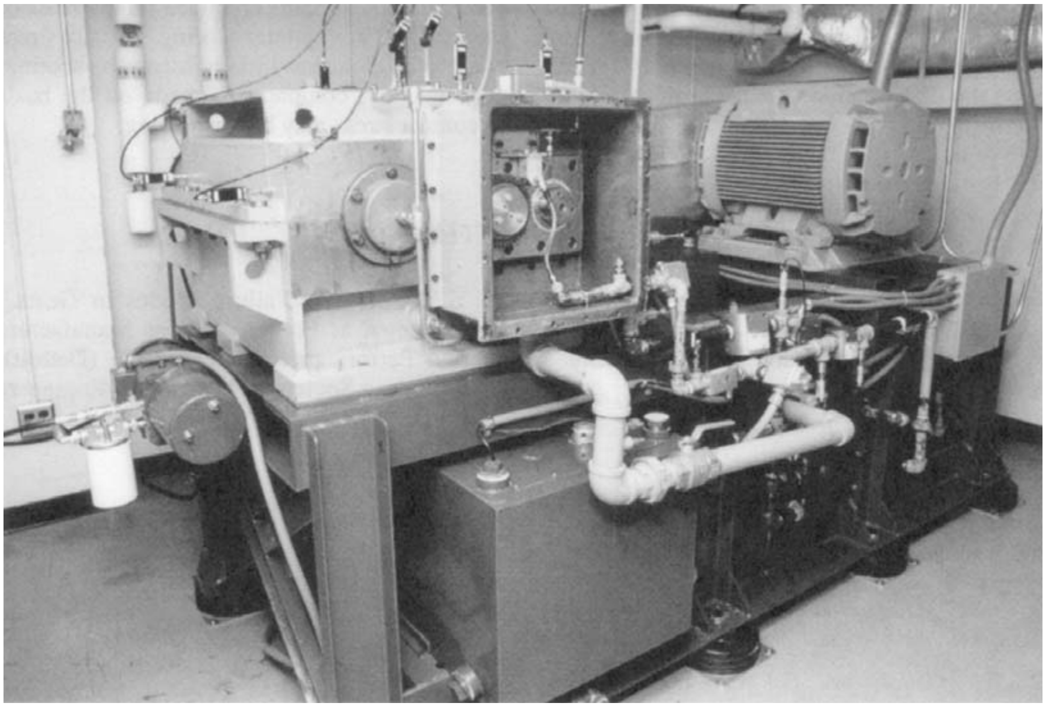

Fig. 13 High-speed (6 in. center distance) PC test rig

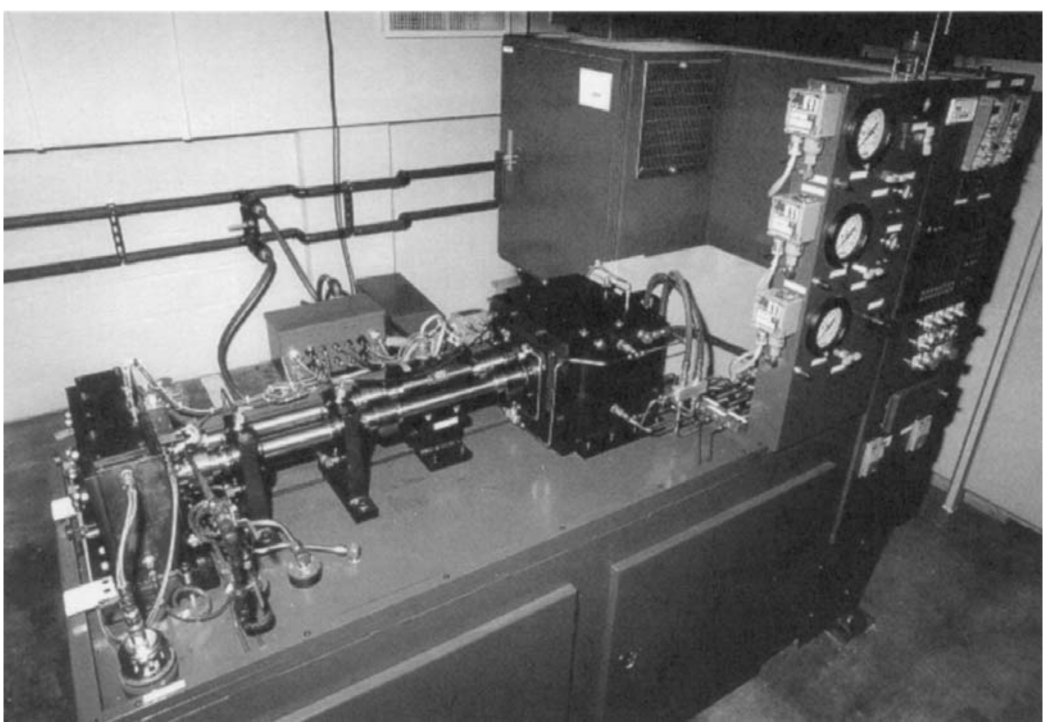

Fig. 14 High-speed (3.5 in. center distance) PC test rig

high-speed test rigs are rigid, so the only remedy for nonuniform contact is to change specimen gear sets.

A break-in run is generally conducted before the start of the test run. On the low-speed machines this run is accomplished by applying one-half of the test load and running for ½ h at test speed starting with room-temperature lubricant. The friction of the meshing gears warms the lubricant to near test temperature in this time. On the high-speed machines, load and speed are increased in a set pattern to accomplish break-in. As described previously, break-in is expected to increase the life of the gears. Thus, all specimens tested in a given project should be broken in using the same method. For root strength (bending fatigue) tests, break-in may not affect bending fatigue life but may delay the onset of pitting long enough to allow the gear to fail in bending.

In general, the main difference in surface durability (pitting) and root strength (bending) tests is the load. Most of the standard specimen gear sets are designed to allow surface durability testing at loads high enough to produce failure in less than 10 million cycles. Root strength tests, therefore, have to be conducted at loads high enough to produce failure in less than 1 million cycles to avoid surface durability failures. Thus, unless specially designed specimens are tested, the results of this test can only be compared to the lower-life portion of the stress-cycles relationship developed in the STF test.

Surface durability gear tests have similar problems with scatter as those previously described for RCF tests. However, each specimen gear has 18 or more teeth; therefore, the result of a single PC surface durability test represents the lowest performance of 18 or more items tested. This statistical difference causes the results of PC surface durability tests to exhibit less scatter than those of RCF tests, and Weibull slopes are generally three or higher. Six tests are still generally conducted with each variant at each load, the reduced scatter being used to allow more statistically reliable comparison of the results rather than to reduce the number of tests. A similar statistical difference exists between the results of STF tests and PC bending tests, and this difference must be considered when comparing STF and PC bending results.

Test Results. The end point of a surface-durability test may be somewhat ambiguous. Several points in the development of surface origin pitting may be considered failure. Test gear teeth are also subject to wear and localized scoring, both of which can affect the appearance of the contact surface in ways similar to some of the stages of pitting development. A composite failure criterion has to be adopted that specifies the level of damage in each of these categories that constitutes failure. A typical failure criterion for pitting is one pit 4 mm (0.16 in.) wide or several smaller pits with slightly greater total area. Typical failure criteria for micropitting is micropitting over half of the contact width of all the teeth and for wear, a loss of 12 μm (0.0005 in.) at any point on the profile.

A test program evaluating bending and surface durability performance of candidate materials would include 24 tests with each material. Six tests would be conducted at each of two loads intended to result in bending failure; the balance of the tests would be conducted at two loads intended to result in pitting failure. The results of tests at each of the four loads are analyzed with Weibull statistical analysis (as illustrated in Fig. 4). A load-cycles diagram is then constructed showing loads corresponding to 10, 50, and 90% failures at various lives for both bending and surface durability. Comparisons are made on the basis of life to 50% (or 10%) failures at a given load, or the load corresponding to 50% failures at a given life. It is useful to construct a stress-cycles diagram (determining contact stress as described here) to compare surface durability results obtained from specimens having differing crown.

Test Procedures for Scoring (or Scuffing) Testing. Scoring is typically most severe in the dedendum of the driving gear (at the start of contact) and at the tip of the driven gear. Therefore, to facilitate observation of scoring, the gears and loading are set up so that the specimen is the driven gear. Scoring is more probable at high speed, high load, and/or high-lubricant temperature. Because of this probability, the test machine is set up to run at high speed and high-lubricant temperature. Resistance to scoring is then measured by the load that will precipitate scoring. Load is increased in comparatively small steps until scoring starts. Because small variations in lead or crown can produce as much variation in unit loading at the most heavily loaded area as one or more load steps, specimens used for scoring should be those with the most consistent geometry that can be selected from the lot to minimize scatter. Gear alignment should be adjusted, if possible, to obtain the best contact pattern.

Break-in is a particularly critical part of this test. Therefore, all specimens should be broken in using an identical process, and control specimens should be included in the project test matrix to benchmark the results for comparison with prior tests. Typically, break-in is started at low load in room-temperature oil and run-in one to four load steps lasting 20,000 to 50,000 loading cycles each. The lubricant temperature is allowed to slowly rise due to the friction of the meshing gears during break-in. The condition of the lubricant also can affect the outcome of the test; thus, lubricant should be changed prior to the start of each test.

Test Results. After break-in, the surface roughness (R_a) is measured in the axial direction at the tips of several teeth on the specimen gear, and the surface condition is documented. The load is increased, and the lubricant temperature is raised to test temperature. The gear set is run 20,000 to 50,000 loading cycles for each load step. After each load step, surface roughness and surface condition are examined and recorded. The appearance of score marks, or an abrupt increase in surface roughness, indicates scoring failure. Typically, three tests are conducted per variant. Comparisons between variants are made on the basis of the load that precipitates scoring. Contact stress considered alone has little influence on scoring; consequently, comparisons made on the basis of contact stress may be misleading.

ACKNOWLEDGMENT

This chapter was adapted from D.R. McPherson and S.B. Rao, Mechanical Testing of Gears, *Mechanical Testing and Evaluation,* Vol 8, *ASM Handbook,* ASM International, 2000, p 861–872

REFERENCES

1. E.E. Shipley, "Failure Modes in Gears," presented at Forum on Gear Manufacture and Performance, 19 Oct 1973 (Detroit), American Society of Mechanical Engineers
2. D.W. Dudley, Gear Wear, *Wear Control Handbook,* M. Peterson and W. Winer, Ed., American Society of Mechanical Engineers, 1980
3. G. Niemann and H. Winter, Getriebeschäden und Abhilfe, Entwicklungstendenzen, *Maschinen-elemente,* Vol 2, 2nd ed., Springer-Verlag (Berlin), 1983
4. "Appearance of Gear Teeth—Terminology of Wear and Failure," ANSI/AGMA 1010-E95, American National Standards Institute/American Gear Manufacturers Association
5. R. Errichello, Friction, Lubrication, and Wear of Gears, *Friction, Lubrication, and Wear Technology,* Vol 18, *ASM Handbook,* ASM International, 1992, p 535–545
6. L. Alban, *Systematic Analysis of Gear Failures,* American Society for Metals, 1985
7. A.P. Boresi and O.M. Sidebottom, *Advanced Mechanics of Materials,* 4th ed., John Wiley & Sons, 1985
8. W. Lewis, Investigation of the Strength of Gear Teeth, *Proc. Eng. Club,* Philadelphia, PA, 1893
9. "Information Sheet: Geometry Factors for Determining the Pitting Resistance and Bending Strength of Spur, Helical, and Herringbone Gear Teeth," AGMA 908-B89 (R1995), American Gear Manufacturers Association, 1995

SELECTED REFERENCES

- C.W. Bowen, Review of Gear Testing Methods, *Gear Manufacture and Performance,* American Society for Metals, 1974, p 137–162
- D.H. Breen, "Physical and Analytical Modeling of Contact Fatigue Pits from Rolling/Sliding Tests," Technical Paper 87-FTM-8, American Gear Manufacturers Association, 1987
- L. Flamand, D. Berthe, and M. Godet, Simulation of Hertizian Contacts Found in Spur Gears With a High Performance Disk Machine, *Trans. ASME,* Vol 102, Jan, 1981, p 204
- D. Medlin, G. Krauss, D.K. Matlock, K. Burris, and M. Slane, "Comparison of Single Gear Tooth and Cantilever Beam Fatigue Testing of Carburized Steel," Technical Paper 950212, SAE International, 1995
- M.B. Slane, R. Buenneke, C. Dunham, M. Semenek, M. Shea, and J. Tripp, "Gear Single Tooth Bending Fatigue," Technical Paper 821042, SAE International, 1982
- M.N. Webster and C.J.J. Norbart, An Experimental Investigation of Micropitting Using a Roller Disk Machine, *STLE Trib. Trans.,* Vol 38, 1995, p 883

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