GEARS AND GEAR DRIVES

GEAR TYPES

Gears are compact, positive-engagement, power transmission elements that determine the speed, torque, and direction of rotation of driven machine elements. Gear types may be grouped into five main categories: Spur, Helical, Bevel, Hypoid, and Worm. Typically, shaft orientation, efficiency, and speed determine which of these types should be used for a particular application. Table 1 compares these factors and relates them to the specific gear selections. This section on gearing and gear drives describes the major gear types; evaluates how the various gear types are combined into gear drives; and considers the principle factors that affect gear drive selection.

Spur gears

Spur gears have straight teeth cut parallel to the rotational axis. The tooth form is based on the involute curve, Figure 1. Practice has shown that this design accommodates mostly rolling, rather than sliding, contact of the tooth surfaces.

The involute curve is generated during gear machining processes using gear cutters with straight sides. Near the root of the tooth, however, the tool traces a trochoidal path, Figure 2, providing a heavier, and stronger, root section. Because of this geometry, contact between the teeth occurs mostly as rolling rather than sliding. Since less heat is produced by this rolling action, mechanical efficiency of spur gears is high, often up to 99%. Some sliding does occur, however. And because contact is simultaneous across the entire width of the meshing teeth, a continuous series of shocks is produced by the gear. These rapid shocks result in some objectionable operating noise and vibration. Moreover, tooth wear results from shock loads at high speeds. Noise and wear can be minimized with proper lubrication, which reduces tooth surface contact and engagement shock loads.

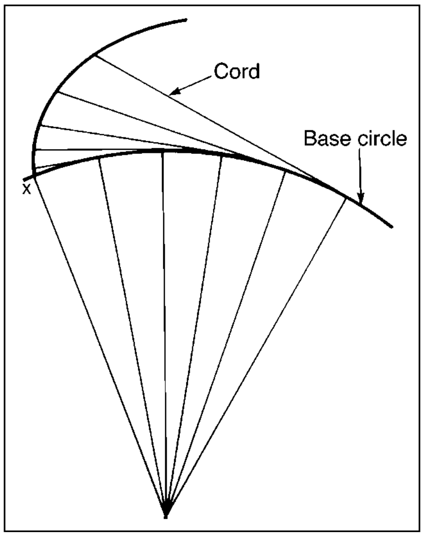

Figure I — Involute generated by unwrapping a cord from a circle.

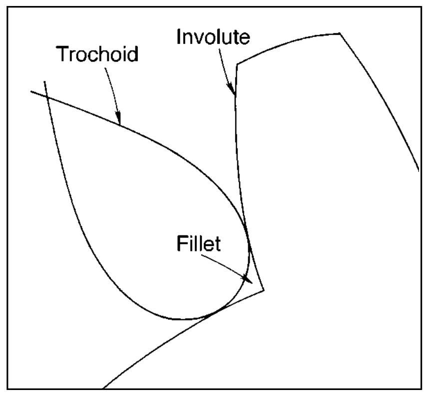

Figure 2 — Root fillet trochoid generated by straight tooth cutting tool.

Table 1 — Comparison of gear types

Gear type	Approx. range of efficiency,%	Range of reduction ratio	Max. pitch line velocity, fpm: High-precision	Max. pitch line velocity, fpm: Commercial
External spur gears	97-99	1:1-5:1	20,000	4,000
External helical gear	97-99	1:1-9:1	40,000	5,000
External herringbone or double helical gear	97-99	1:1-9:1	40,000	5,000
Straight bevel gear	97-99	1:1-10:1	10,000	1,000
Spiral bevel gear	97-99	1:1-10:1	25,000	5,000
Cylindrical worm	50-90	3:1-100:1	10,000	5,000
Double-enveloping worm	50-98	3:1-100:1	10,000	4,000
Hypoid	90-98	1:1-10:1	10,000	4,000

Spur gears are the least expensive to manufacture and the most commonly used, especially for drives with parallel shafts. The three main classes of spur gears are: external tooth, internal tooth, and rack-and-pinion.

External-tooth gears — The most common type of spur gear, Figure 3, has teeth cut on the outside perimeter of mating cylindrical wheels, with the larger wheel called the gear and the smaller wheel the pinion.

The simplest arrangement of spur gears is a single pair of gears called a single reduction stage, where output rotation is in a direction opposite that of the input. In other words, one is clockwise while the other is counter-clockwise.

Higher net reduction is produced with multiple stages in

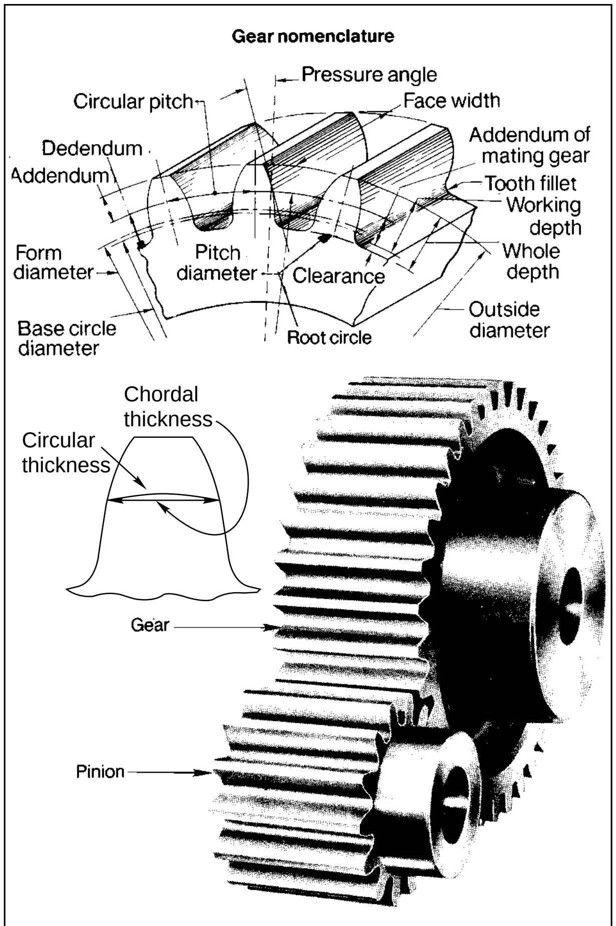

Figure 3 — Spur gears have straight teeth cut parallel to the rotational axis.

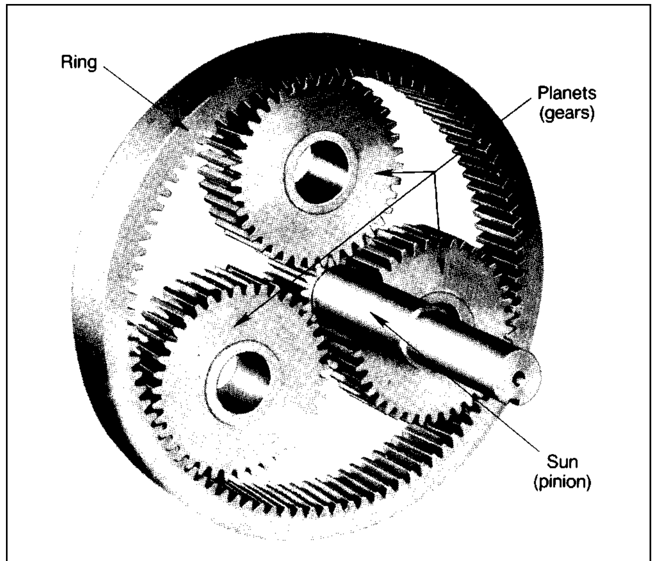

Figure 4 — Internal (ring) gears produce a complex form of output with a planetary configuration of sun, planets, and ring.

which the driven gear is rigidly connected to a third gear. This third gear then drives a mating fourth gear that serves as output for the second stage. In this manner, several output speeds on different shafts can be produced from a single input rotation.

Internal (ring) gears — Ring gears produce an output rotation that is in the same direction as the input, Figure 4. As the name implies, teeth are cut on the inside surface of a cylindrical ring, inside of which are mounted a single external-tooth spur gear or set of external-tooth spur gears, typically consisting of three or four larger spur gears (planets) usually surrounding a smaller central pinion (sun).

Normally, the ring gear is stationary, causing the planets to orbit the sun in the same rotational direction as that of the sun. For this reason, this class of gear is often referred to as a planetary system. The orbiting motion of the planets is transmitted to the output shaft by a planet carrier.

In an alternative planetary arrangement, the planets may be restrained from orbiting the sun and the ring left free to move. This causes the ring gear to rotate in a direction opposite that of the sun. By allowing both the planet carrier and the ring gear to rotate, a differential gear drive is produced, the output speed of one shaft being dependent on the other.

Rack-and-pinion gears — A straight bar with teeth cut straight across it, Figure 5, is called a rack. Basically, this rack is considered to be a spur gear unrolled and laid out fiat. Thus, the rack-and-pinion is a special case of spur gearing.

The rack-and-pinion is useful in converting rotary motion to linear and vice versa. Rotation of the pinion produces linear travel of the rack. Conversely, movement of the rack causes the pinion to rotate.

The rack-and-pinion is used extensively in machine tools, lift trucks, power shovels, and other heavy machinery where rotary motion of the pinion drives the straight-line action of a reciprocating part. Generally, the rack is operated without a sealed enclosure in these applications, but some type of cover may be provided to keep dirt and other contaminants from accumulating on the working surfaces.

Helical gears

Helical gearing differs from spur in that helical teeth are cut across the gear face at an angle rather than

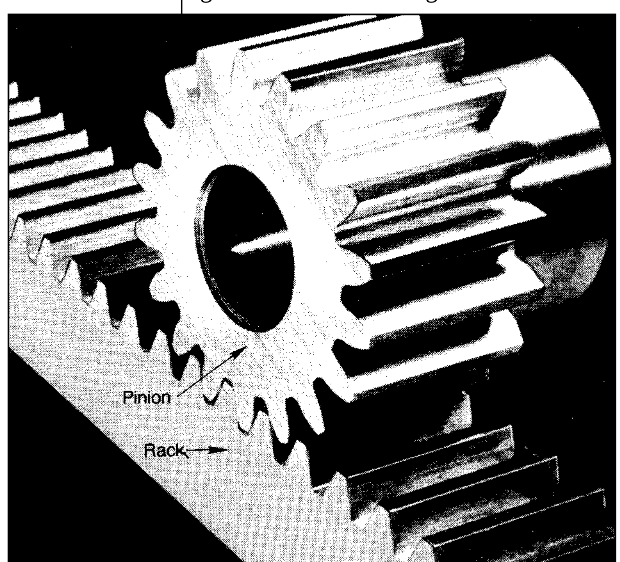

Figure 5 — Rack-and-pinion gearing produces linear travel from rotational input. Shown here is spur gearing. Helical gearing is also available, but is not as common because the helical teeth create thrust, which produces a force acting across the face of the rack. Worm rack is also available, the axis of the worm (pinion) being parallel to, rather than perpendicular to, the rack.

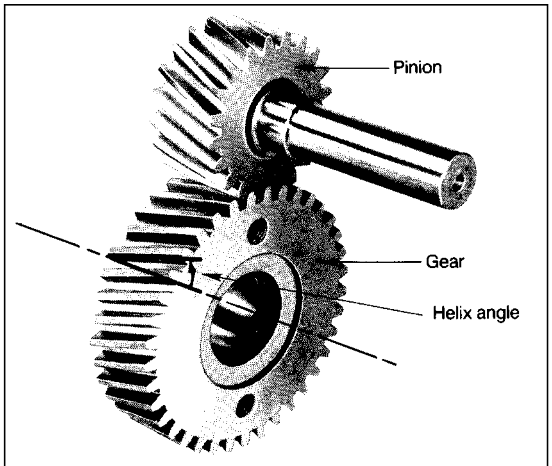

Figure 6 — Helical gears have teeth cut across the face at an angle for gradual loading.

straight, Figure 6. Thus, the contact line of the meshing teeth progresses across the face from the tip at one end to the root of the other, reducing the noise and vibration characteristic of spur gears. Also, several teeth are in contact at any one time, producing a more gradual loading of the teeth that reduces wear substantially.

The increased amount of sliding action between helical gear teeth, however, places greater demands on the lubricant to prevent metal-to-metal contact and resulting premature gear failure. Also, since the teeth mesh at an angle, a side thrust load is produced along each gear shaft. Thus, thrust bearings must be used to absorb this load so that the gears are held in proper alignment.

The three other principle classes of helical gears are: double-helical, herringbone, and cross-helical.

Double-helical gears — Thrust loading is eliminated by using two pairs of gears with tooth angles opposed to each other, Figure 7. In this way, the side thrust from one gear cancels the thrust from the other gear. These opposed gears are usually manufactured with a space between the opposing sets of teeth.

Herringbone gears — Teeth in these gears resemble the geometry of a herring spine, with ribs extending from opposite sides in rows of parallel, slanting lines, Figure 8. Herringbone gears have opposed teeth to eliminate side thrust loads the same as double helicals, but the opposed teeth are joined in the middle of the gear circumference. This arrangement makes herringbone gears more compact than double-helicals. However, the gear centers must be precisely aligned to avoid interference between the mating helixes.

Cross-helical gears — This type of gear is recommended only for a narrow range of applications where loads are relatively light. Because contact between teeth is a point instead of a line, the resulting high sliding loads between the teeth requires extensive lubrication. Thus, very little power can be transmitted with cross-helical gears.

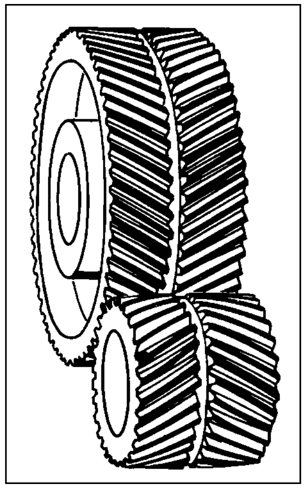

Figure 7 — Double helical gearing uses two pairs of opposed gears to eliminate thrust.

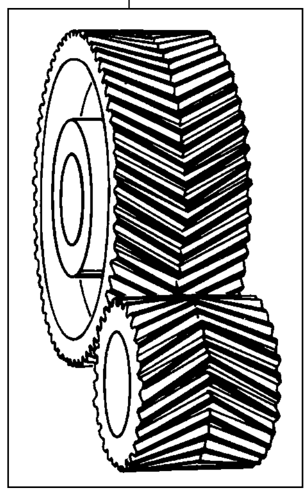

Figure 8 — Herringbone gears have opposed teeth joined in the middle.

Bevel gears

Unlike spur and helical gears with teeth cut from a cylindrical blank, bevel gears have teeth cut on an angular or conical surface. Bevel gears are used when input and output shaft centerlines intersect. Teeth are usually cut at an angle so that the shaft axes intersect at 90 deg, but any other angle may be used. A special class of bevels called miter gears have gears of the same size with their shafts at right angles.

Often there is no room to support bevel gears at both ends because the shafts intersect. Thus, one or both gears overhang their supporting shafts. This overhung load (OHL) may deflect the shaft, misaligning gears, which causes poor tooth contact and accelerates wear. Shaft deflection may be overcome with straddle mounting in which a bearing is placed on each side of the gear where space permits.

There are two basic classes of bevels: straight-tooth and spirals.

Straight-tooth bevels — These gears, also known as plain bevels, have teeth cut straight across the face of the gear, Figure 9. They are subject to much of the same operating conditions as spur gears in that straight-tooth bevels are efficient but somewhat noisy. They produce thrust loads in a direction that tends to separate the gears.

Spiral-bevels — Curved teeth provide an action somewhat like that of a helical gear, Figure 10. This produces smoother, quieter operation than straight-tooth bevels. Thrust loading depends on the direction of rotation and whether the spiral angle at which the teeth are cut is positive or negative.

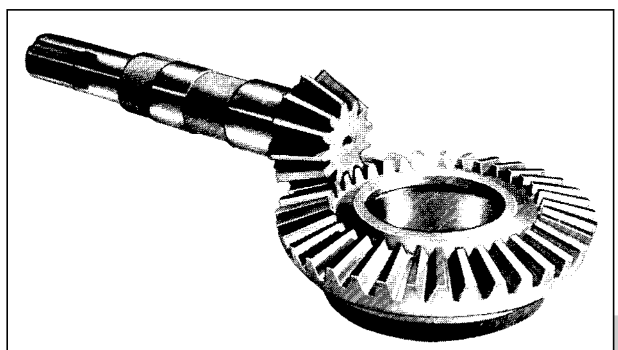

Figure 9 — Straight-tooth bevel gears are efficient but somewhat noisy.

Hypoid gears

Hypoid gears resemble spiral-bevels, but the shaft axes of the pinion and driven gear do not intersect, Figure 11. This configuration allows both shafts to be supported at both ends. In hypoid gears, the meshing point of the pinion with the driven gear is about midway between the central position

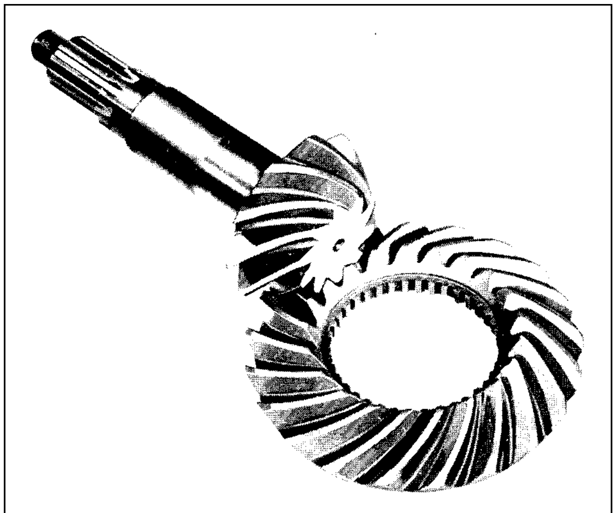

Figure 10 — Spiral bevel-gears have curved teeth for smoother operation.

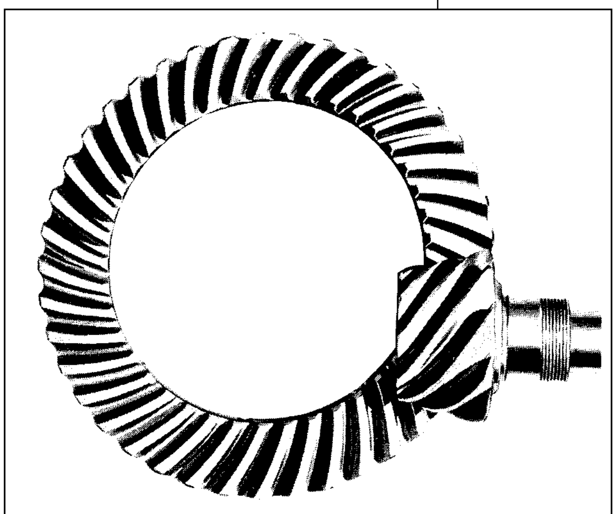

Figure 11 — Hypoid gears resemble spiral bevels, but the shaft axes do not intersect. Therefore, both shafts can be supported at both ends.

of a pinion in a spiral-bevel and the extreme top or bottom position of a worm. This geometry allows the driving and driven shafts to continue past each other so that end-support bearings can be mounted. These bearings provide greater rigidity than the support provided by the cantilever mounting used in some bevel gearing. Also adding to the high strength and rigidity of the hypoid gear is the fact that the hypoid pinion has a larger diameter and longer base than a bevel or spiral-bevel gear pinion of equal ratio.

Although hypoid gears are stronger and more rigid than most other types, they are one of the most difficult to lubricate because of high tooth-contact pressures. Moreover, the high levels of sliding between tooth surfaces reduces efficiency. In fact, the hypoid combines the sliding action of the worm gear with the rolling movement and high tooth pressure associated with the spiral bevel. In addition, both the driven and driving gears are made of steel, which further increases the demands on the lubricant. As a result, special extreme pressure lubricants with both oiliness and anti-weld properties are required to withstand the high contact pressures and rubbing speeds in hypoids.

Despite these demands for special lubrication, hypoid gears are used extensively in rear axles of automobiles with rear-wheel drives. Moreover, they are being used increasingly in industrial machinery.

Worm gearing

Worm gear sets, Figure 12, consist of a screw-like worm (comparable to a pinion) that meshes with a larger gear, usually called a wheel. The worm acts as a screw, several revolutions of which pull the wheel through a single revolution. In this way, a wide range of speed ratios up to 60:1 and higher can be obtained from a single reduction.

Most worms are cylindrical in shape with a uniform pitch diameter. However, a double-enveloping worm has a variable pitch diameter that is narrowest in the middle and greatest at the ends. This configuration allows the worm to engage more teeth on the wheel, thereby increasing load capacity.

In worm-gear sets, the worm is most often the driving member. However, a reversible worm-gear has the worm and wheel pitches so proportioned that movement of the wheel rotates the worm.

In most worm gears, the wheel has teeth similar to those of a helical gear, but the tops of the teeth curve inward to envelop the worm. As a result, the worm slides rather than rolls as it drives the wheel. Because of this high level of rubbing between the worm and wheel teeth, the efficiency of worm gearing is lower than other major gear types.

One major advantage of the worm gear is low wear, due mostly to the full-fluid lubricant film that tends to be formed between tooth surfaces by the worm sliding action. A continuous film that separates the tooth surfaces and prevents direct metal-to-metal contact is typically provided by a relatively heavy oil, which is often compounded with fatty or fixed oils such as acidless tallow oil. This adds film strength to the lubricant and further reduces friction by increasing the oiliness of the fluid.

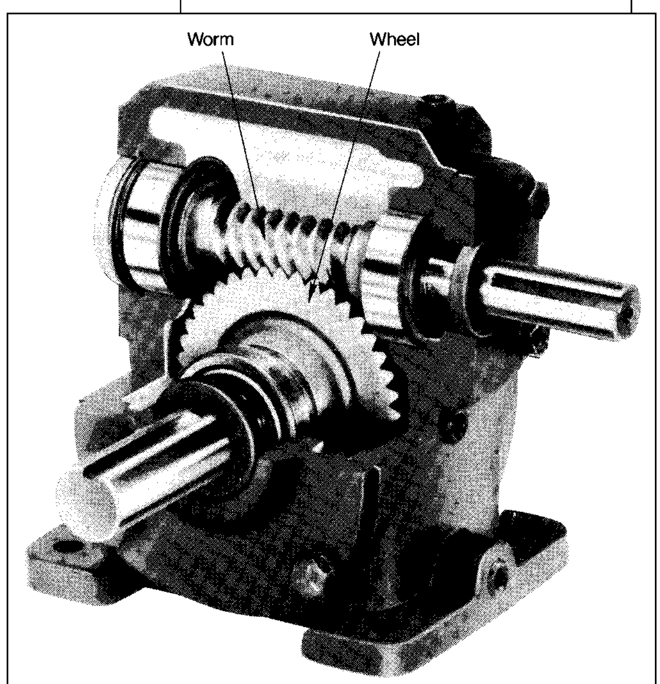

Figure 12 — Worm gearing has perpendicular, nonintersecting shafts in which the worm acts as a screw. Several revolutions of the worm pull the wheel through one revolution.

Worm-gear friction is further reduced through the use of metals with inherently low coefficients of friction. For example, the wheel is typically made of bronze and the worm of a highly finished, hardened steel. These low-friction materials can be used in worm gears because pressures are more uniformly distributed over the tooth surface than most other gear types.

Worm-gear shafts are perpendicular, non-intersecting, and may be positioned in a variety of orientations.

Noncircular gears

Though often overlooked, noncircular gears can provide several types of unusual motion or speed characteristics.

Cams and linkages can provide these special motion requirements as well, but noncircular gears often represent a simpler, more compact, or more accurate solution. Servo systems may also be able to do the job, but they are usually more expensive and require more expertise to solve motion problems.

Common requirements handled by noncircular gears include converting a constant input speed into a variable output speed, and providing several different constant-speed segments during an operating cycle. Other applications require combined translation and rotation, or stop-and-dwell motion.

Variable speed. Several types of noncircular gears, particularly elliptical gears, generate variable output speeds. Other, less commonly used types are triangular and square gears.

Elliptical. A set of like elliptical gears can run at a constant center distance, but deliver an output speed that changes as they rotate. Elliptical gears come in two basic types: unilobe, Figure 13, which rotates about one of two fixed points on its long axis, and bilobe, Figure 14, which rotates about its center. The speed-reduction ratio of these gears varies from 1/K to K during each cycle of rotation, where practical values of K range up to 3.

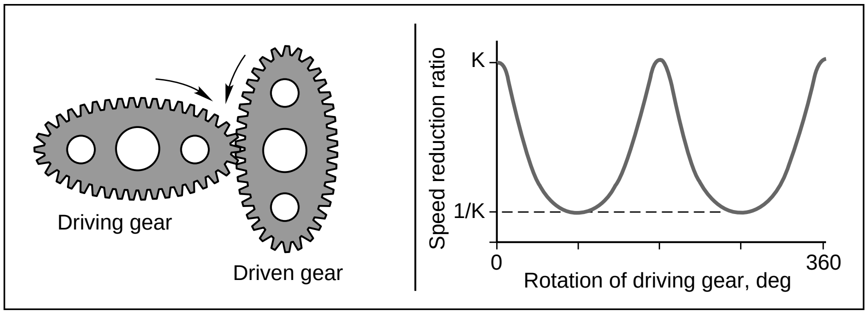

Figure 14 — Elliptical gears with two lobes (bilobes) provide twice as many periods of variable output speed.

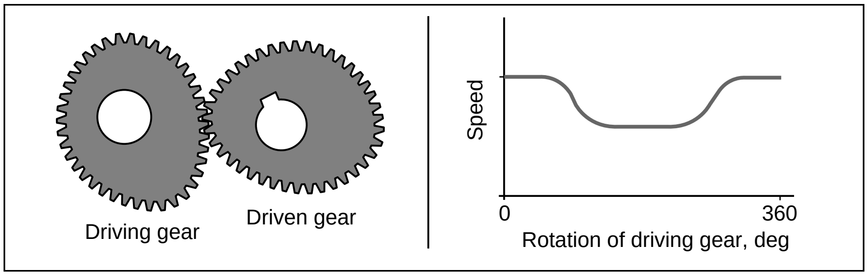

Figure 15 — Multispeed gears give one constant speed for part of a cycle and a different constant speed for a second part of the cycle.

As the gears in Figure 14 rotate, the radii of the driving and driven gears change, so that speed first decreases for $^{1}/_{4}$ revolution, then increases for $^{1}/_{4}$ revolution, etc. These periods of increasing or decreasing speed occur four times per revolution.

Elliptical gears are commonly used in packaging and conveyor applications.

Triangular. A pair of triangular gears has three lobes, or high points on the perimeter, rather than the two lobes in elliptical bilobe gears. So triangular gears deliver six periods of speed increase or decrease per revolution, rather than four.

Square. Gears that are square have four lobes, so they produce eight periods of speed increase or decrease per revolution.

Both triangular and square gears have a smaller range of speed ratios than elliptical gears.

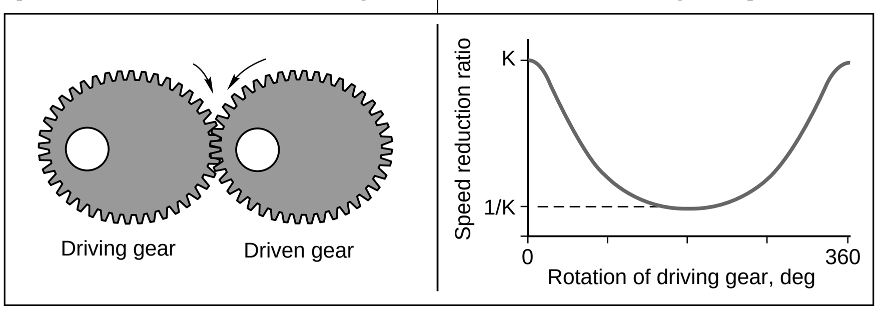

Figure 13 — Elliptical unilobe gears provide variable output speed.

Constant-speed segments. Where an application requires several constant-speed periods within a cycle, multispeed gears, Figure 15, may be the answer. These gears make the transition between speeds by using special function segments on the gear perimeter between the constant-speed sections.

Translation and rotation. For applications requiring both translational and rotational motion, certain gears serve as cam substitutes. Often used in labeling machines, the cam gear duplicates the shape of a part to be labeled and a cam-following rack carries the labeling device at a constant surface speed.

Stop-and-dwell motion. Some machines must provide either stop-and-dwell motion or reverse motion. This is achieved by combining noncircular gears with round gears and a differential (epicyclic gear train).

Stop-and-dwell motion is common in indexing mechanisms. Reverse motion is required where a transfer device must operate between two locations.

GEAR BASICS

Types of gears

Spur gears. Gears with teeth straight and parallel to the axis of rotation.

Helical gears. Gears with teeth that spiral around the body of the gear.

External gears. Gears with teeth on the outside of a cylinder.

Internal gears. Gears with teeth on the inside of a hollow cylinder. (The mating gear for an internal gear must be an external gear.)

Bevel gears. Gears with teeth on the outside of a conical-shaped body (nor-

Fundamental formulas

Spur gears

Pitch diameter $D = \frac{N}{P_d} = \frac{N - p_c}{\pi}$

Circular pitch $p_c = \frac{\pi}{P_d} = \frac{\pi D}{N}$

Diametral pitch $P_d = \frac{\pi}{p_c} = \frac{N}{D}$

Number of teeth $N = DP_d = \frac{\pi D}{p_c}$

Outside diameter $D_o = D + \frac{2}{P_d} = \frac{N+2}{P_d}$

Root diameter $D_R = D - 2b$

Base circle diameter $D_b = D\cos\phi$

Base pitch $p_b = p_c\cos\phi$

Tooth thickness at standard pitch diameter $T\,\text{std} = \frac{p_c}{2} = \frac{\pi D}{2N}$

Addendum $a = \frac{1}{P_d}$

Center distance $C = \frac{D_1 + D_2}{2} = \frac{N_1 + N_2}{2} = \frac{p_c(N_1 + N_2)}{2\pi}$

Contact ratio $m_p = \frac{\sqrt{R_o^2 - R_b^2} + \sqrt{r_o^2 - r_b^2} - C\sin\phi}{p_c\cos\phi}$

Backlash (linear) $B = 2\,(\overline{\Delta C})\tan\phi$

Backlash (linear) $B = \Delta T$

Backlash, linear along line-of-action $B_{LA} = B\cos\phi$

Backlash, angular $B_a = 6880\frac{B}{D}$ (arcmin)

Minimum number of teeth for no undercutting $N = \frac{2}{\sin^2\phi}$

Worm meshes

Pitch diameter of worm $d_w = \frac{n_w p_{cn}}{\pi\sin\lambda}$

Pitch diameter of worm gear $D_g = \frac{N_g p_{cn}}{\pi\cos\lambda}$

Lead angle $\lambda = \tan^{-1}\frac{n_w}{P_d d_w} = \sin^{-1}\frac{n_w p_{cn}}{\pi d_w}$

Lead of worm $L = n_w p_c = \frac{n_w p_{cn}}{\cos\lambda}$

Normal circular pitch $p_{cn} = p_c\cos\lambda$

Center distance $C = \frac{d_w + D_g}{2}$

Center distance $C = \frac{p_{cn}}{2\pi}\left(\frac{N_g}{\cos\lambda} + \frac{n_w}{\sin\lambda}\right)$

Velocity ratio $Z = \frac{N}{n_w}$

mally used on 90-deg axes).
Worm gears. Gearsets in which one member of the pair has teeth wrapped around a cylindrical body like screw threads. (Normally this gear, called the worm, has its axis at 90 deg to the worm-gear axis.)
Face gears. Gears with teeth on the end of the cylinder.

Bevel gearing

Velocity ratio	$Z = \frac{N_1}{N_2}$
Velocity ratio	$Z = \frac{D_1}{D_2}$
Velocity ratio	$Z = \frac{\sin \gamma_1}{\sin \gamma_2}$
Shaft angle	$\Sigma = \gamma_1 + \gamma_2$

Helical gearing

Normal circular pitch	$p_{cn} = p_c \cos \psi$
Normal diametral pitch	$P_{dn} = \frac{P_d}{\cos \psi}$
Axial pitch	$P_a = p_c \cot \psi = \frac{P_{cn}}{\sin \psi}$
Normal pressure angle	$\tan \phi_n = \tan \phi \cos \psi$
Pitch diameter	$D = \frac{N}{P_d} = \frac{N}{P_{dn} \cos \psi}$
Center distance (parallel shafts)	$C = \frac{N_1 + N_2}{2P_{dn} \cos \psi}$
Center distance (crossed shafts)	$C = \frac{1}{2P_{dn}} \left(\frac{N_1}{\cos \psi_1} + \frac{N_2}{\cos \psi_2} \right)$
Shaft angle (crossed shafts)	$\theta = \psi_1 + \psi_2$

Symbol nomenclature and definition

B	backlash, linear amount along pitch circle
B_{LA}	backlash, linear amount along line of action
B_a	backlash in minutes
C	center distance
D	pitch diameter, gear (D_g)
D_b	base circle diameter
D_o	outside diameter
D_R	root diameter
L	length, general; also lead of worm
N	number of teeth, usually gear (N_g)
P_d	diametral pitch
P_{dn}	normal diametral pitch
R	pitch radius, gear or general use
R_b	base circle radius, gear
R_o	outside radius, gear
T	tooth thickness, gear
Z	mesh velocity ratio
a	addendum
b	dedendum
d	pitch diameter, pinion
d_w	pitch diameter, worm
m_p	contact ratio
n_w	number of threads in worm
p_a	axial pitch
p_c	circular pitch
p_{cn}	normal circular pitch
r	pitch radius, pinion
r_b	base circle radius, pinion
r_o	outside radius, pinion
γ	pitch angle, bevel gear
θ	rotation angle, general
λ	lead angle, worm gearing
ϕ	pressure angle
ϕ_n	normal pressure angle
ψ	helix angle
Σ	shaft angle, bevel gearing

Hypoid gears. Similar in general form to bevel gears, but operate on non-intersecting axes.

Elements of gear teeth

Tooth surface. Forms the side of a gear tooth.

Tooth profile. One side of a tooth in a cross section between the outside circle and the root circle.

Flank. The working, or contacting, side of the gear tooth. The flank of a spur gear usually has an involute profile in a transverse section.

Top land. The top surface of a gear tooth.

Bottom land. The surface at the bottom of the space between adjacent teeth.

Crown. A modification that results in the flank of each gear tooth having a slight outward bulge in its center area. A crowned tooth becomes gradually thinner toward each end. A fully crowned tooth has a little extra material removed at the tip and root areas also. The purpose of crowning is to ensure that the center of the flank carries its full share of the load even if the gears are slightly misaligned or deflect under load.

Root circle. Tangent to the bottom of the tooth spaces in a cross section.

Pitch circle. Concentric to base circle and including pitch point. Pitch circles are tangent in mating gears.

Gear center. The center of the pitch circle.

Line of centers. Connects the centers of the pitch circles of two engaging gears; it is also the common perpendicular of the axes in crossed helical gears and worm gears.

Pitch point. The point of a gear-tooth profile which lies on the pitch circle of that gear. At the moment that the pitch point of one gear contacts its mating gear, the contact occurs at the pitch point of the mating gear, and this common pitch point lies on a line connecting the two gear centers.

Line of action. The path of action for involute gears. It is the straight line passing through the pitch point and tangent to the base circle.

Line of contact. The line or curve along which two tooth surfaces are tangent to each other.

Point of contact. Any point at which two tooth profiles touch each other.

Linear and circular measurements

Center distance. The distance between the parallel axes of spur gears or of parallel helical gears, or the crossed axes of crossed helical gears or of worms and worm gears. Also, it is the distance between the centers of the pitch circles.

Offset. The perpendicular distance between the axes of hypoid gears or offset face gears.

Pitch. The distance between similar, equally spaced tooth surfaces along a given line or curve.

Diametral pitch. 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. Diametral pitches usually range from 25 to 1.

Axial pitch. Linear pitch in an axial plane and in a pitch surface. In helical gears and worms, axial pitch has the same value at all diameters. In gearing of other types, axial pitch may be confined to the pitch surface and may be a circular measurement.

Base pitch. In an involute gear, the pitch on the base circle or along the line of action. Corresponding sides of involute gear teeth are parallel curves, and the base pitch is the constant and fundamental distance between them along a common normal in a plane of rotation.

Axial base pitch. The base pitch of helical involute tooth surfaces in an axial plane.

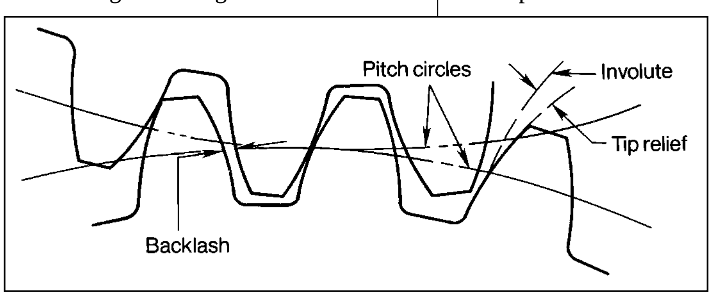

Figure 16 — Backlash and tip relief.

Lead. The axial advance of a thread or a helical spiral in 360 deg (one turn about the shaft axis).

Backlash. The amount by which a tooth space exceeds the thickness of the engaging tooth on the operating pitch circles.

Angular dimensions

Helix angle. The inclination of the tooth in a lengthwise direction. If the helix angle is 0 deg, the tooth is parallel to the axis of the gear and is really a spur-gear tooth.

Lead angle. The inclination of a thread at the pitch line from a line 90-deg to the shaft axis.

Shaft angle. The angle between the axes of two non-parallel gear shafts.

Pitch angle. In bevel gears, the angle between an element of a pitch cone and its axis.

Angular pitch. The angle subtended by the circular pitch, usually expressed in radians.

Ratios

Gear-tooth ratio. The ratio of the larger to the smaller number of teeth in a pair of gears.

Contact ratio. To assure smooth, continuous tooth action, as one pair of teeth passes out of action, a succeeding pair of teeth must have already started action. It is desired to have as much overlap as possible. A measure of this overlapping action is the contact ratio.

Hunting ratio. A ratio of numbers of gear and pinion teeth which ensures that each tooth in the pinion will contact every tooth in the gear before it contacts any gear tooth a second time. (13 to 48 is a hunting ratio; 12 to 48 is not a hunting ratio.)

General Terms

Runout. A measure of eccentricity relative to the axis of rotation. Runout is measured in a radial direction and the amount is the difference between the highest and lowest reading in 360 deg, or one turn. For gear teeth, runout is usually checked by either putting pins between the teeth or using a master gear. Cylindrical surfaces are checked for runout by a measuring probe that reads in a radial direction as the part is turned on its specified axis.

Undercut. When part of the involute profile of a gear tooth is cut away near its base, the tooth is said to be undercut. Undercutting becomes a problem when the number of pinion teeth is small.

Flash temperature. The temperature at which a gear tooth surface is calculated to be hot enough to destroy the oil film and allow instantaneous welding or scoring at the contact point.

Full depth teeth. Those in which the working depth equals 2.000 divided by normal diametral pitch.

Tip relief. A modification of a tooth profile, whereby a small amount of material is removed near the tip of the gear tooth to accommodate smooth engagement of the teeth.

SPEED REDUCERS

A speed reducer is a gearset assembled with appropriate shafting, bearings, and lubrication in a sealed housing, generally an oil-tight case. Such units are also sometimes referred to as gear boxes, speed increasers, and gear reducers.

Speed reducers are available in a broad range of power capacities and speed ratios depending on gear size and type, with most having a maximum speed limit of 3,600 rpm, and usually driven at a full-load speed of 1,750 rpm or less.

Lubrication of speed reducers is accomplished either by a splash or circulating system. Bearings may be lubricated automatically with the same oil as the gearset, or they may have separate systems. The specific lubricant required for a speed reducer depends on a number of factors including operating speed, ambient temperature, loads, and method of lubricant application. More information on lubrication is discussed in the PT Accessories Product Department.

Gear types — Most speed reducers use one or more of the common gear types previously discussed. However, the use of helical, worm, and bevel gearsets is particularly prevalent, especially in small and medium-sized speed reducers. Helical gears are often used in combination with spiral-bevel or worm gears.

Selection of a particular style of speed reducer for an application depends primarily on shaft arrangement, type of gearing, ratio range, and horsepower range. Three broad categories of speed reducers, grouped according to mounting arrangements, are base-mounted, shaft-mounted, and gearmotor, Figures 17 to 19.

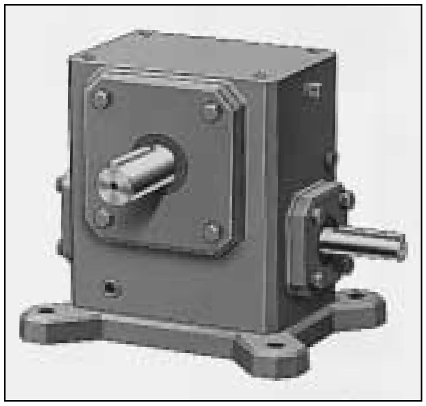

Figure 17 — Based-mounted speed reducer with shafts mounted at right angles.

Figure 18 — Shaft-mounted speed reducer with torque arm to prevent housing rotation.

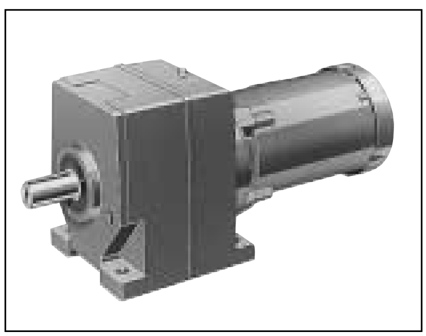

Figure 19 — Gearmotor.

Base-mounted speed reducers — Gear reducer units that have feet bolted to a stationary pad or other structural member, account for the largest number of speed reducer applications. In general, the prime mover is mounted on the same structure as the reducer or on the reducer itself, but can be mounted on a separate structure.

The input and output shafts of these base-mounted speed reducers can be arranged horizontally or vertically, and parallel to each other or at a right angle. Parallel shaft units typically use helical or spur gears, whereas right-angle units contain bevel, bevel and helical, or worm gearing.

A particular type of base-mounted reducer is the free-shaft type, which usually has one high-speed input shaft connected to a prime mover and a single or double output (low-speed) shaft connected to the load. In general this type is a good choice if the prime mover is remotely located or if only a simple change in direction in the power train is required. Free-shaft reducers are also recommended if a proper-sized gearmotor or motorized reducer is not available, or if several loads must be driven by a single prime mover.

Shaft-mounted reducers — Some gear reducers are furnished with a hollow output shaft that slips over the driven shaft, which in turn supports the reducer. Free rotation of the housing is prevented by some type of reaction member, often a torque arm or flange attached to a stationary portion of the machine. The gear unit may or may not support the prime mover. Shaft-mounted reducers generally use helical or spur gears for parallel-shaft arrangements and worm gearing for right-angle arrangements. Though many shaft-mounted reducers are now in use, they are generally available only in a limited range of sizes and are used most widely in material handling applications.

Gearmotors — A gearmotor combines an enclosed gearset with a motor. The frame of one component supports the other, and the motor shaft is typically common with or coupled directly to the gear input shaft. In one version, called an integral gearmotor, the speed reducer and motor share a common shaft. Another type uses an input flange, Figure 20, for connec-

Figure 20 — Input flange on a speed reducer accepts a C-face motor.

tion to a C-face motor.

Motorized reducers resemble gearmotors and perform much the same function. This reducer type commonly incorporates a scoop, Figure 21, which supports the motor. Generally, the distinction between motor-reducer units depends on whether the motor is an integral part (as in a gearmotor) or a modular part (as in a reducer-motor combination).

Figure 21 — Scoop-mount motor-reducer.

Gearmotors and motorized reducers often are specified when the reducer is the first component following the prime mover in a power train. Of the two, the motorized reducer is being used more extensively because the motor can be changed quickly and easily, standard motors are readily available for replacement, and a wide range of options can be provided from a few basic components.

Traction drives

Traction drives, Figure 22, transmit mechanical power from source to load by means of mating metal rollers, which may be considered gears with infinite numbers of teeth.

These rollers can be cones, cylinders, discs, rings, spheres, or toroids. The speed ratio is determined by the radius of rotation of the driver roller on the driven member, the distance between the rollers, or their orientation with respect to each other.

Traction drives are available in two types — dry and lubricated. Dry traction drives eliminate the need for lubricant and allow nearly 100% efficiency in power transmission. Slippage between driving and driven members is prevented by a spring-loaded system.

Other traction drives use synthetic fluids. Previously, conventional lubricants were used so that traction drives had to be designed with very high contact pressures between rolling members to carry the load without slip. Today's traction fluids reduce this required contact pressure so drives can be designed with longer lives and higher power capacities. Under the high pressure, viscosity of the fluid increases dramatically so the fluid behaves more like a plastic material than a liquid. This plastic-like film enables the drive to transmit power without appreciable metal-to-metal contact.

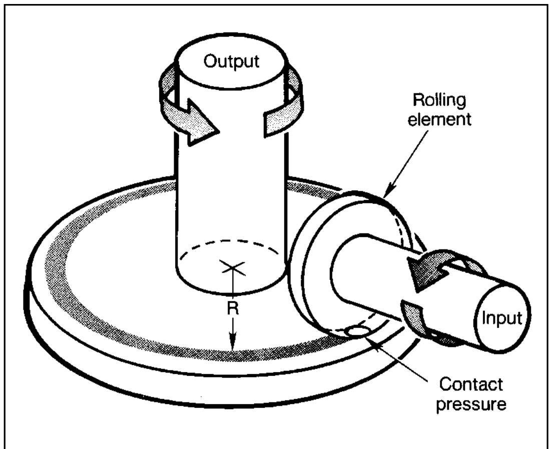

Figure 22 — Concept drawing shows how traction drives transfer power from the input to output shaft with a set of mating rollers. By varying radius, *R*, output speed can be adjusted while input speed remains constant.

Another contributor to traction drives' increased power capacity is improved high-grade bearing steel that resists fatigue, thereby extending the life of the drives.

Traction drives are compact and can be used instead of belts, gears, or chain drives where space is limited. Moreover, traction drives are quieter than most other mechanical drives because there is no engagement shock loading or backlash. Other inherent advantages include high speed reduction or multiplication, low vibration, excellent rotational accuracy, and high efficiency at all speeds.

Many traction drives are capable of adjusting the speed ratio infinitely throughout a range. These variable-ratio traction drives are discussed in the Adjustable-speed Drives Product Department of this handbook. More recently, constant-ratio traction drives have been developed to accommodate increased power, longer life, and higher speed ratios.

Constant-ratio and other roller drive designs use one or more rows of stepped planet rollers to provide a high speed ratio in a single planetary stage. The drive functions as either a speed reducer or increaser depending on whether power is introduced through the sun roller or the outer roller ring.

Cycloidal drives

Cycloidal drives, Figure 23, transmit power equal to that of gears, but in a smaller and more efficient package. In contrast to the circular motion of gears, cycloidal drives use noncircular or eccentric components to convert input rotation into a wobbly cycloidal motion. This cycloidal motion is then converted back into smooth, concentric output rotation. In the process, speed reduction occurs.

The term cycloidal is derived from hypocycloidal, which is defined as the curve traced by a point on the circumference of a circle that is rotating inside the circumference of a larger fixed circle. A common example of this motion is the path traced by a tooth of a planetary pinion rotating inside a ring gear.

Whereas worm gearing experiences a dramatic loss of efficiency in going from low to very high input/output speed ratios; and helical gearing loses efficiency at high ratios because two or more stages of reduction are required; cycloidal drives achieve reduction ratios as high as 200:1 in a single stage, while still maintaining moderately high efficiencies. Moreover, because cycloidal drive components interact in a rolling fashion, failure is generally not catastrophic. As in a bearing, fatigue in the rolling surfaces of a cycloidal drive causes noise levels to gradually increase, serving as a warning long before complete drive failure occurs.

Heat generation, attributable mainly to mechanical losses and the power being transmitted, is readily dissipated through the large surface area of other types of gears. But cycloidal drives, like worm gears, must dissipate heat through a smaller housing surface area. However, because efficiency in the cycloidal drive

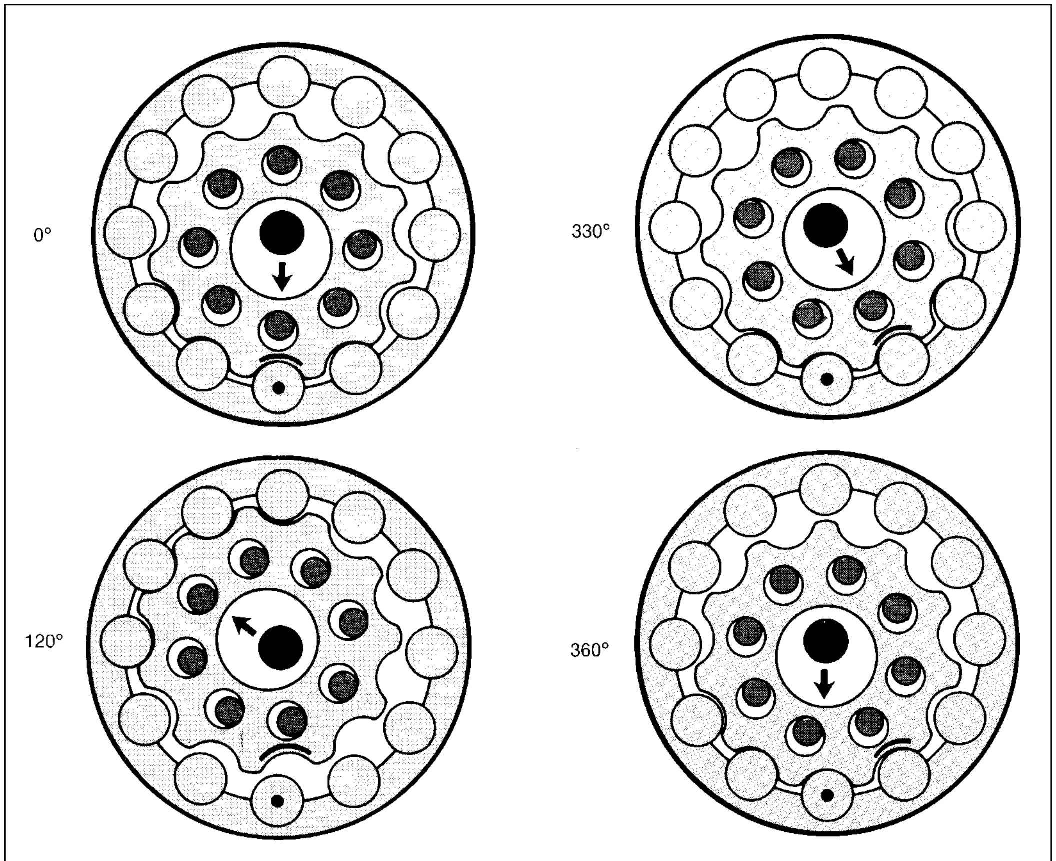

Figure 23 — Cycloidal drives, such as this planocentric unit, convert input rotation into a wobbly cycloidal motion. This cycloidal motion is then converted back into smooth circular output rotation, with speed reduction occurring in the process.

is higher than worm gearing of equal capacity and ratio, less heat is generated in cycloidal units. Consequently, the auxiliary cooling often required for worm units is usually not needed for cycloidal drives.

There are various types of cycloidal drives currently available. Harmonic and planocentric drives are general types of cycloidals in which speed reduction occurs in converting the input motion into cycloidal motion. Double-reduction cycloidals achieve greater ratios because speed is reduced twice: first in converting input rotational motion into cycloidal motion, and again in converting the cycloidal motion back into rotational output motion.

Although not as efficient as spur or helical gearing, cycloidal drives offer substantially higher efficiency than worm gearing. The concentric shaft orientation also proves valuable, as does the drive's compact size and high reduction capability.

SELECTING GEAR DRIVES

Gears can be selected, rated, installed, and maintained by most users through common standards and practices developed by the American Gear Manufacturers Association. However, the services of a gear-engineering specialist are generally required to make more detailed, in-depth analysis in cases where severe duty, extreme reliability, unusually long service, or other extraordinary conditions exist. In either case, the major selection factors include: shaft orientation, speed ratio, design style, nature of load, service factor, environment, mounting position, ratio, lubrication, and installation practices. All these factors must be carefully considered in selecting gears for optimal operation in a particular application.

Shaft orientation — Probably the first consideration in selecting a gear type is to determine the required orientation of the input to the output shaft. According to the type of gear selected, various shaft arrangements are available including: parallel shafts, shafts at right angles with intersecting axes, shafts at right angles with nonintersecting axes, skewed shafts, and concentric shafts.

Speed ratio — The ratio of input speed to that of the output is another significant factor dictating the type of gearing to be selected. By examining efficiency and gear type, the user can determine if a single high-ratio stage is sufficient or if multistage gearing is required.

Geared systems can be driven at constant or varying speeds, depending on the requirements of the application. When a geared system is selected for an adjustable-speed application, operating speed should be determined along with operating cycle and power requirements. From this information, a lubrication system can be selected, the heat dissipation evaluated, the effect of speed variation on the dynamic characteristics determined, and the need for special balancing determined.

Gears can be custom-designed to meet specific speed requirements, or standard ratios may be used from manufacturers' catalogs. Generally, a standard ratio results in a less expensive drive. Standard ratios established by AGMA are a series of values based on the 1.5 geometric progression, which is a modification of the ASA 10 series ranging from 1.225 to 1,810.0:1. Tolerances for these ratios are ±3% for single reduction, ±4% for double and triple reduction, and ±5% for quadruple reduction. There are no AGMA standards for worm-gear ratios. However, popular ratios for single reduction systems range from 5 to 70, and for double reduction from 75 to 5,000.

Design style — The application should be evaluated to determine if individual open gears will be sufficient or if an enclosed speed reducer is required. In general, individual gears require rigid shafting to keep the gears aligned. And shafts should not introduce external loads to the gearing. Typically, an enclosure around the gears with oil lubrication is the preferred design, but grease-lubricated open gears can be used in relatively clean environments.

Nature of load — Theoretically, gear teeth and bearings under stresses below the endurance limit and lubricated properly will last indefinitely, provided the operating loads are within the design specifications. Thus, determining the nature of the load is important in selecting gears for long life and reliable service. Gears may be subjected to occasional loads (e.g., 15 to 30 minutes per day), intermittent loads (applied several minutes per hour), or continuous service loads (10 to 24 hours per day). The life of the geared system is the period of operating time or cycles during which the system can transmit a required load. Life of the gear may end with a fracture or operational failure of a component or with the development of excessive noise, vibration, or heat.

Determining the nature of a gear load involves the consideration of maximum horsepower, drive inertia, overhung load, and speed limit of the gear.

Maximum horsepower or maximum torque of the prime mover is one of the most significant considerations on which selection of a geared unit is based. A gear is rated approximately as a constant-torque machine, with the horsepower rating varying almost directly with the input speed. Either constant-torque or constant-horsepower motors are used with gears, depending on the application. For example, constant-torque motors are required for applications such as conveyors, stokers, and reciprocating compressors. Constant-horsepower motors are required for lathes, boring mills, radial drill presses, etc.

Drive inertia also requires torque to overcome more than the drive load of the output, because a drive is seldom subjected to only a single-intensity load. For example, if an electric motor is the prime mover, peak starting torques as high as 400% of the motor rating can be applied to the drive during start-up. The geared unit is only one member of the system that may contain members with high inertia, unbalance, or torsional stiffness.

The shafts of geared units are typically subjected to torque loads, axial loads, and radial loads. These last two types are usually called externally applied thrust loads and externally applied overhung loads, and they can be caused by chains, pinions, V-belts, or flat belts.

The type of load introduced by the prime mover depends on the operational characteristics of the prime mover. Electric motors and turbines, for example, produce relatively smooth operation whereas an internal-combustion engine does not afford so smooth a load.

Overhung load (OHL), if produced by the drive, requires the use of outboard bearings or speed reducers that will accept this OHL. OHL is imposed on the shaft when a pinion, sprocket, sheave, pulley, or crank is mounted on the input or output shaft.

To calculate the magnitude of the load, multiply the transmitted force that is tangent to the pitch circle of the mounted member by the OHL factor. For a single or multiple-chain drive, this OHL factor is 1.00. The factor is 1.25 for a cut pinion run with gear teeth, 1.50 for a single or multiple V-belt drive, and 2.50 for a flat-belt drive.

OHL given by manufacturers are usually specified in catalogs at one shaft diameter from the housing face. Load-location factors are also given so that, regardless of where the load is acting, it can be converted to the reference position and compared with the cataloged value.

The magnitude of the OHL that can safely be applied depends upon several factors including bearing life or pressure, elastic shaft deflection, shaft strength, and bolt strength.

Speed limit is based on a maximum pitch line velocity of 5,000 fpm and pinion speeds of 3,600 rpm or less. For worm gearing, speed limits are based on a maximum sliding velocity of 6,000 fpm and worm speeds of 3,600 rpm and less. Occasionally, the speed limitation may be that for a bearing or contact oil seal, based on the manufacturer's limiting speed.

Service factor — Basic ratings take into account minimum design criteria for gears, lubricants, shafts, bearings, and other gearset components. Service factors are then used to adjust the basic rating to a service rating that is compatible with the required life, operating duty, and dynamic characteristics of the driving and driven machines. Typically, this service rating is determined by multiplying the required horsepower by the appropriate service factor based on equipment, duty cycle, and type of prime mover.

For most speed reducers (spur, helical, herringbone, and bevel), AGMA recognizes three load classifications for determining service factors: uniform, moderate shock, and heavy shock. Based on field experience, numerical values have been assigned to these classifications for intermittent service, for service between 3 and 10 hours per day, and for service beyond 10 hours per day. The factors are dependent upon the prime mover. Moreover, the load classification and resultant service factor varies with the application of different types of geared units.

Using these service factors, a maximum momentary or starting load of 200% of rated load can be allowed for most gears, with rated load defined as the unit rating with a service factor of 1.0.

Service factors for worm gears are determined somewhat differently, with normal starting or momentary peak loads up to 300% of rated load permissible.

Environment — The type of gear selected must also compensate for a less-than-ideal environment, which can adversely affect gear-system performance if proper precautionary steps are not taken. The most common types of hostile environments are dust, heat, wide variation in temperature, moisture, and chemicals. Generally, each has particular adverse effects on lubricant, gears, bearings, or seals. Dusty atmospheres may contaminate the lubricant, for example. Moreover, heat may accelerate lubricant breakdown, affect the performance of synthetic contact oils, or lower gear capacity by lowering material properties and distorting the gear. Wide temperature variation may cause improper lubrication for gears and bearings, thereby shortening the life of the unit. Moisture infiltration may accelerate lubricant breakdown, corrode components, and accelerate the wear of contact seals.

Contact seals should be used on input and output shafts when the unit operates in dusty environments or where water is splashed around the unit. In atmospheres laden with abrasive dust or in areas hosed down with water under pressure, two contact seals may be required on each shaft.

Mounting position — Mounting position is an important selection criteria, because most gear units are designed to operate in either a horizontal or vertical position only. In some applications, however, the geared unit may have to operate in a position inclined to either axis or inclined to both axes. In this case special considerations must be given in selecting the gear systems regarding oil level, air-vent position, and location of oil-drain holes.

Gear rating — All components of a commercial enclosed geared unit are usually rated by established practices, with the rating for the entire system determined by the lowest rating for any one part for a given set of operating conditions. Generally, there are two types of ratings for a geared unit: mechanical and thermal.

The mechanical rating is based on the strength of the gears, shafts, bolts, etc., or the load compatible with the required bearing life or pressure, or the resistance of the gears to pitting or scoring.

The thermal rating specifies the power that can be transmitted without exceeding a specified rise above ambient in the operating temperature of the unit. Typically, enclosed drives operate at temperature rises of 70 to 100 F above ambient temperature. Generally, a maximum oil-sump temperature of 200 F is permissible for gearmotors and shaft-mounted speed reducers, even though thermal ratings are not specified by manufacturers or defined by AGMA.

When the applied horsepower exceeds the thermal horsepower, auxiliary cooling methods are used including: use of an oil-exclusion pan to reduce churning of the oil and the resulting higher temperatures, air-cooling the housing, circulating water or some other cooling medium around the unit, using a separate oil sump for greater heat dissipation, or using a cooler mounted inside the gear housing.

Lubrication — The most important factor in a lubricating system is reliability, because failure to supply lubricant to the bearings and gears will result in their damage. As a result, the type of lubricating system should be chosen carefully, because this may be the most critical component of the entire system.

In addition, accepted practices must be observed by the user to maintain proper lubrication during the life of the geared unit. One of the most common causes for gear damage, for example, is the failure to fill the unit to the proper level with the first change of lubricant, so data on the lubrication plate and all other instructions must be followed carefully. Give special attention to warning tags.

The user should also know the temperature range in which the unit is designed to operate. If the unit is operating where temperatures vary widely, the oil viscosity should be changed to suit the conditions. For low-temperature operation, the oil should have a pour-point lower than that of the extreme minimum temperature encountered. And the oil may require pre-heating under extremely cold starting conditions.

Several lubrication system types are available. Enclosed gear units are commonly lubricated by splash systems in which one of the gears dips into an oil bath and transfers the lubricant to the contacting teeth as it rotates. For low-speed operation, scrapers close to the side of the gear may be used so that splash from the gears reaches oil-feed troughs to lubricate the bearings.

Pressure or forced-spray lubrication reduces oil churning. In this system, lubricant is pumped under pressure to the gear train. After passing through the gears, it is returned to the reservoir to be recirculated. This method uses the gear oil as a cooling medium. The lubricant may be delivered as a stream running over the gearing or as a spray from jet or spray nozzles. These nozzles may direct the lubricant to the entering or the trailing side of the gear mesh, depending on speed.

Many gear units (particularly vertical units) have grease-lubricated bearings. Grease is normally introduced at the factory and grease fittings are provided for lubrication in the field. Some units designed to operate in extremely adverse conditions have double seals mounted in cages. They allow grease to be purged from a chamber between the seals.

Installation practices

Successful performance of a gear unit is vitally dependent upon installation practices. The foundation must be rigid and level, the shafts aligned, the accessories properly installed, the required grade and quality of lubricant added, and any special instructions followed carefully.

Foundation — The gear unit must provide a level, secure base on which to operate. The base should be mounted horizontally, unless the unit has been designed for mounting in a tilted position. Moreover, the base should be precisely leveled, since only a few degrees of tilt might adversely affect lubrication efficiency. If a unit is to be mounted in a position different from that for which it was designed, changes might be necessary to provide proper lubrication.

Most gear units have a drain plug in the base for oil drainage. The unit should be mounted above floor level, or a sump hole used for this purpose. Drain plugs may be replaced with a valve or with pipe extensions for convenient drainage. However, guarding must be provided to prevent accidental breakage. Drain valves may be provided with a lock to prevent accidental or unauthorized opening.

When mounting a unit on structural steel beams, a base plate should be used between the two beams to which the unit is attached. The thickness of the base should be equal to or greater than the thickness of the unit feet, and it should extend under the entire unit. Mountings that are not sufficiently rigid can lead to excessive vibration and deflection.

Procedures — When installing the geared unit bring it into alignment by placing broad, flat shims under all mounting pads. Start at the low-speed end and level across the width and then along the length of the unit. Most units have leveling surfaces for reference. Use a feeler gage under all pads to make certain that all are carrying equal loads. This prevents distortion of the housing when foundation bolts are secured.

Units equipped with backstops should receive special attention during assembly. If the motor is connected to start in the wrong direction, the sudden shock can immediately break the backstop or cause premature failure shortly thereafter. Thus, disconnect the motor coupling and check the rotating direction of the backstop by rotating the gear unit by hand. Then check the rotation of the motor by an actual start.

The user should also make sure that all couplings, shims, and pinions are properly installed so that no destructive load is applied to the gear unit or to these parts because of misalignment. Hubs, pinions, sprockets, pulleys, and other components should be carefully installed (not driven on with a hammer) on shafts. Couplings should be aligned so that the angular and parallel alignment are within the limits specified by the coupling manufacturer. The distance between the shaft ends or coupling hubs should also be checked to ensure they agree with the specifications.

Maintenance — Proper maintenance is essential in keeping a gear unit running properly. After a week or more of service, check all external bolts and pipe plugs to make sure they are tight. Also, check chains, belts, sprockets, and pinions, to make sure that the load is being distributed evenly and that there is no unusual condition.

After a few hundred hours of operation, the user should drain all oil, flush the system with an oil of similar grade, and refill the lubricating system to the proper level. Of course, oil level should be periodically checked under all conditions.

ANALYZING GEAR FAILURES

By following a step-by-step procedure, engineers can diagnose gear failures and develop solutions. Here's how to conduct a failure analysis.

Procedures can vary depending on failure conditions. For example, if the gears are still able to function, you may continue their operation and monitor the rate at which damage progresses via periodic inspection and sound and vibration measurements. If reliability is crucial, examine the gears by magnetic particle inspection to ensure that they have no cracks.

Failure inspection

Before starting the inspection, collect background information, including part numbers, service history, and lubricant type. Interview those involved in the design, installation, operation, and maintenance of the gearbox. Encourage them to tell everything they know about the gearbox even if it seems unimportant.

Visual examination. Before disassembling the gearbox, inspect its exterior and record data that would otherwise be lost. For example, the condition of seals and keyways must be recorded before disassembly. Otherwise, it will be impossible to determine when these parts may have been damaged. Take gear tooth contact patterns before completely disassembling the gearbox (see next section).

After the external examination, disassemble the gearbox and inspect all components. Examine the gear teeth and bearings and record their condition. Look for signs of corrosion, contamination, and overheating.

After the initial inspection, wash the parts with solvent and re-examine them. This is often *the most important phase of the investigation* and may yield valuable clues. A low power magnifying glass and pocket microscope are helpful tools for this phase.

The bearings often provide clues as to the cause of gear failure:

- Bearing wear can cause excessive radial clearance or end play that misaligns the gears.
- Bearing damage may indicate corrosion, contamination, electrical discharge, or lack of lubrication.
- Deformation between rollers and raceways may indicate overloads.
- Gear failure often follows bearing failure.

Gear tooth contact patterns. (Complete this step before disassembling gearbox components). The way in which mating gear teeth contact indicates how well they are aligned, Figure 24. If the gears are still able to function, record tooth contact patterns under either loaded or unloaded conditions. For no-load tests, paint the teeth of one gear with marking compound. Then, roll the teeth through mesh so the contact pattern transfers to the unpainted gear.

For loaded tests, use machinist's layout lacquer to paint the teeth, then run the gears to wear off the lacquer and establish the contact patterns.

Document observations. Describe your observations in writing, using sketches and photographs where needed. Mark each component so it is clearly identified. Mark the bearings so their position in the gearbox can be determined later.

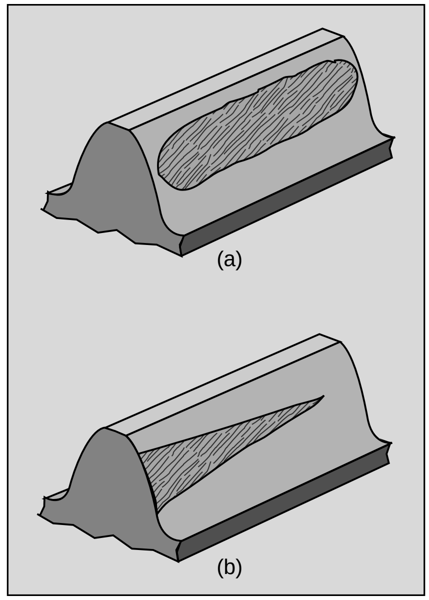

Figure 24 — Typical gear tooth contact patterns: (a), aligned, and (b), misaligned.

Gear geometry. Obtain the following data, which will be needed later to calculate the load capacity of the gearset:

- Number of teeth.
- Outside diameter.
- Face width.
- Gear housing center distance for each gearset.
- Whole depth of teeth.
- Tooth thickness (span and top land).

Test specimens. Take broken parts for laboratory testing. Oil samples can be very helpful. But, an effective oil analysis depends on having representative samples. For a gearbox drain or reservoir, take a sample from the top, middle, and bottom. Check the oil filter and magnetic plug for wear debris and contaminants.

Determine type of failure

Now examine all of the information to determine how the gear(s) failed. Several failure modes may be present. Identify the primary mode of failure, and any secondary modes that may have contributed to the failure. The four most common failure modes are bending fatigue, contact fatigue, wear, and scuffing.

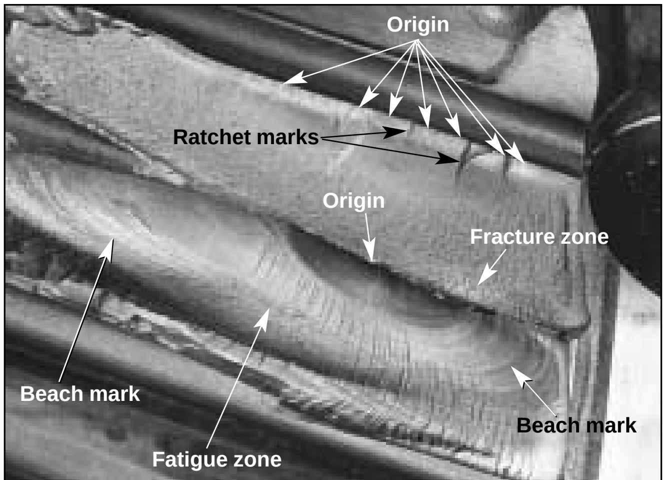

Figure 25 — Bending fatigue fracture surfaces of gear teeth. Upper tooth has multiple origins of failure.

Bending fatigue. This type of failure, caused by repeated loading, starts as a crack that grows until the part fractures. As a fatigue crack propagates, it leaves "beach marks" that correspond to positions where the crack stopped, Figure 25. The origin of the crack is usually surrounded by several of these beach marks.

Most fatigue failures occur in the tooth root fillet, Figure 26.

Contact fatigue. In another mode, called contact fatigue, repeated stresses cause cracks and detachment of metal fragments from the tooth contact surface, Figure 27. The most common types of contact fatigue are macropitting (visible to the naked eye) and micropitting.

With macropitting, fatigue cracks grow until they cause a piece of the surface to break out, forming a pit. Progressive macropitting consists of pits larger than 1 mm diam.

Micropitting has a frosted, matte, or gray stained appearance. Magnification shows the surface to be covered by pits less than 20 μm deep.

Wear. Tooth wear involves removal or displacement of material due to mechanical, chemical, or electrical action. The three major types are adhesion, abrasion, and polishing.

Adhesion is the transfer of material from one tooth to another due to welding and tearing, Figure 28. It is categorized as mild or moderate.

Mild adhesion usually occurs during gearset run-in and subsides after it wears imperfections from the surface. Moderate adhesion removes machining marks from the contact surface, and it can become excessive.

Abrasion is caused by contaminants in the lubricant such as scale, rust, machining chips, grinding dust, weld splatter, and wear debris. It appears as smooth, parallel scratches or gouges, Figure 29.

Severe abrasion removes all machining marks, and it may cause wear steps at the ends of the contact surface and in the dedendum. Tooth thickness may be reduced significantly.

Finally, polishing is fine abrasion that imparts a mirror-like finish to gear teeth, Figure 30. It is promoted by fine abrasives in the lubricant.

Severe polishing removes all machining marks. The surface may be wavy or it may have wear steps at the ends of the contact area and in the dedendum.

Scuffing. Severe adhesion (scuffing) transfers metal from one tooth to another, Figure 31. It usually occurs in bands along the direction of sliding. Surfaces have a rough or matte texture.

Mild scuffing is generally nonprogressive. Moderate scuffing occurs in patches, and it can be progressive.

Severe scuffing occurs on significant portions of a tooth (for example, entire addendum or dedendum), and is usually progressive. In some cases, material is deformed and displaced over the tooth tip or into the tooth root.

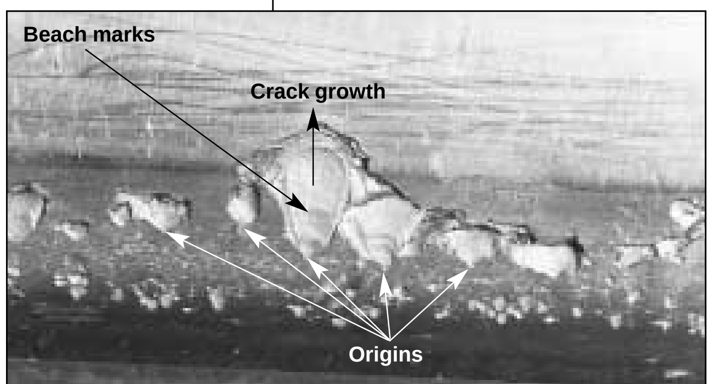

Figure 27 — Fatigue failure (pitting) in the contact surface of a gear tooth. Beach marks are visible in some of the larger pits.

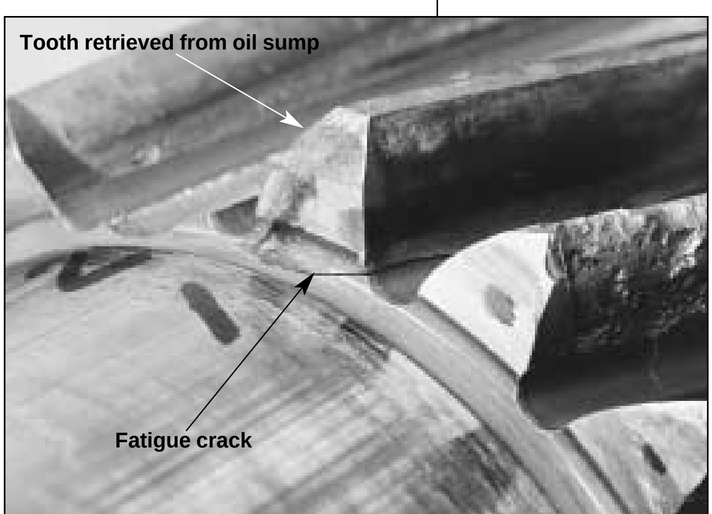

Figure 26 — Fatigue crack in a gear tooth root fillet.

Tests and calculations

In many cases, inspection information isn't enough to determine the cause of failure. When this happens, gear design calculations and laboratory tests are needed.

Design calculations. The gear geometry data collected earlier aids in estimating tooth contact stress, bending stress, lubricant film thickness, and tooth contact temperature. These values are calculated according to American Gear Manufacturers Association standards (ANSI/AGMA 2001-B88 for spur and helical gears). Comparing these calculated values with

Figure 28 — Adhesion type wear of gear teeth.

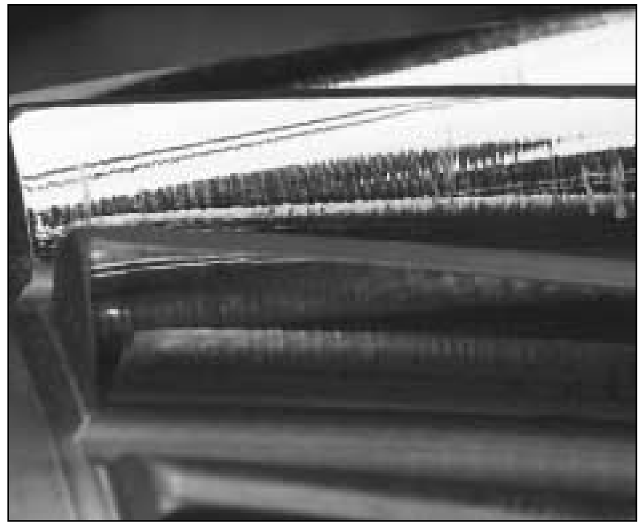

Figure 30 — Polishing type wear.

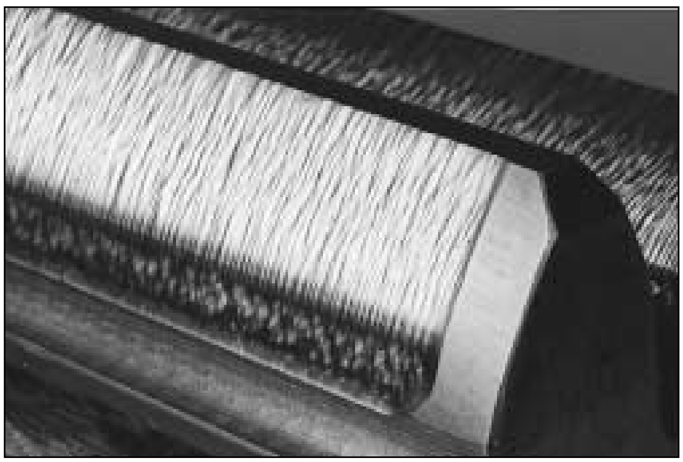

Figure 29 — Excessive abrasion type wear.

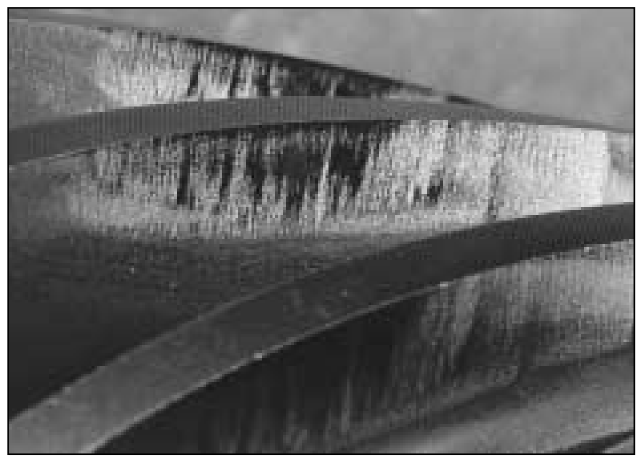

Figure 31 — Scuffing of gear tooth surfaces.

AGMA allowable values helps to determine the risk of macropitting, bending fatigue, and scuffing.

Laboratory tests. A microscopic examination is useful for confirming the failure mode or finding the origin of a fatigue crack.

If tooth contact patterns indicate misalignment, check the gear accuracy on gear inspection machines. Conversely, where contact patterns indicate good alignment, check for metallurgical defects.

Conduct nondestructive tests before any destructive tests. These nondestructive tests, which help detect material or manufacturing defects, 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 various acid etches.
- Determination of grain size and nonmetallic inclusions.
- Examination of fracture surfaces with scanning electron microscope (SEM).

Conclusions and report

When all calculations and tests are completed, form a hypothesis for the probable cause of failure, then determine if the evidence supports the hypothesis. Evaluate all of the evidence collected during the investigation.

Finally, after testing the hypothesis against the evidence, you reach a conclusion about the most probable cause of failure.

A failure analysis report should describe the inspections and tests, and give conclusions. It usually contains recommendations for repairing the equipment, or changing its design or operation to prevent future failures.

Extracted from an article by Geartech in the March 1994 issue of PTD. ■