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Gears

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SINCE THE DESIGN OF GEAR TEETH is covered extensively in the literature, this chapter will present information on selecting gearing and incorporating it into an overall drive system.

Gear Types

Gears can be classified according to the arrangement of their shaft axes in a simple, single-mesh gearset, Table 1.

Parallel Shafts: The following types of gears are used between parallel shafts.

Spur gears are the most common type, Fig. 1. Teeth are straight and parallel to the shaft axis.

Advantages are the absence of end thrust, economy of manufacture and maintenance, and simplicity. An external gear-and-pinion combination is most usual, but an external pinion with an internal gear permits a shorter center distance.

Helical gears are cylindrical gears, with teeth in the form of a helix, Fig. 2*a*.

Helical gears have a greater load-carrying capacity than equivalent-size spur gears, and can operate at higher pitchline velocity. Because of overlapping of the teeth, motion is transmitted more smoothly and quietly than with spur gears. Helical gears produce thrust along the axis of rotation. This end thrust increases with helix angle and requires use of thrust bearings.

Double-helical gears have two sets of opposed helical teeth. Thrust loads are counterbalanced, so that no resultant axial load is transmitted to the shaft and bearings. Space is provided between the two sets of teeth to allow for run-out of the tooth cutting tool. This space should be small because it reduces the active face width.

Herringbone gears are essentially the same as double-helical gears, Fig. 2*b*, and have no space separating the two opposed sets of helical teeth. Double-helical and herringbone gears are used for the transmission of heavy loads at high speeds in continuous service.

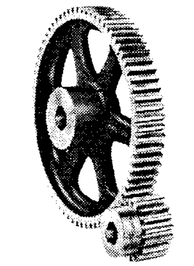
Planocentric Gears: A fixed internal-tooth gear meshing with an eccentrically mounted external-tooth gear only slightly smaller, Fig. 3, provides a high-reduction, high-torque drive for in-line shafts. Input is to the eccentric shaft, and output is taken from the pinion through a pin coupling which permits radial displacement. Under favorable conditions, efficiency of planocentric gears may be greater than 90%.

Harmonic drive[1] depends for its action for in-line shafts on controlled elastic deformation, Fig. 4. A fixed, rigid, internal-tooth spline meshes with the external teeth of a thin, flexible, inner ring of slightly smaller diameter. A third element, called a wave generator, rotates within the flexible spline and deflects it slightly from its natural circular shape so that it meshes with the fixed spline at two or more regions. The drive permits high speed ratios, and has high torque capacity and positional accuracy because of the many spline teeth in simultaneous contact.

Intersecting Shafts: The following types of gears are used between intersecting shafts.

[1]See list of tradenames at end of chapter.

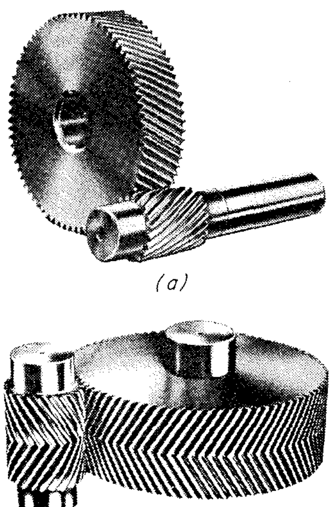
Straight-bevel gears can be used on shafts at any angle, although 90° is most usual. They are designed in pairs, hence are not always interchangeable. Because the teeth are tapered, bearing



(a)

(b)

Fig. 1—External spur-gear pair, *a*, and internal spur gear, *b*.



(a)

(b)

Fig. 2—Single-helical gear pair, *a*, and double-helical pair, herringbone type, *b*.

Table 1—Comparison of Single-Mesh Gears

Type of Gearing	Gear-Ratio Range	Efficiency at Rated Power (%)	Maximum Pitchline Velocity (ft/min)
Spur	1 to 10	98	2000
Helical & herringbone	1 to 15	98	5000
Helical and double helical, high speed	1 to 15	98	30,000
Crossed helical	1 to 10	98	4000
Straight bevel	1 to 6	98	1000
Spiral bevel	1 to 9	98	8000
Zerol	1 to 9	98	4000
Hypoid	1 to 9	98	4000
High-reduction hypoid	10 to 20	80	
Worm	3½ to 90	50 to 90	6000
Double-enveloping worm	3½ to 90	50 to 98	4000
Face	3 to 8	95 to 99	4000
Spiroid	10 to 100	50 to 97	6000
Helicon	3 to 100	50 to 98	6000

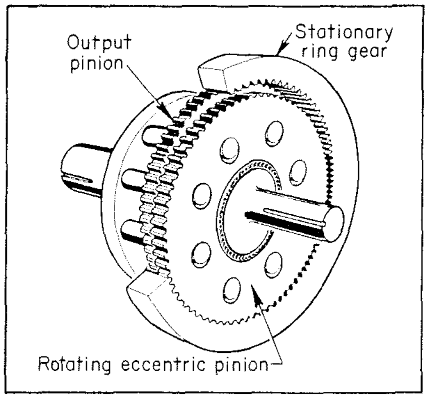


Fig. 3—Planocentric gear pair with output through second eccentric pinion.

surface and strength are reduced at the inner end. For this reason, the tooth contact bearing should be localized to avoid contact at either end. To maintain proper tooth contact, supporting shafts and bearings must not yield under load.

Straight-bevel gears, Fig. 5*a*, have straight teeth which if extended would pass through the point of intersection of their axes. They are considered satisfactory for pitchline velocities up to 1000 fpm.

Spiral-bevel gear teeth are curved and oblique, Fig. 5*b*. Because the teeth are at an angle to the axis of rotation, the loading is distributed over two or more teeth at any given instant. Such gears are suitable for pitchline velocities up to 8000 fpm. Ground teeth extend this limit to 18,000 fpm and beyond. Load capacity is superior to that of straight-bevel gears, and the gears run more quietly. Thrust loads are greater than with straight-bevel gears.

Basic Considerations

Gear Ratio: Defined as the ratio of the larger to the smaller number of teeth in a pair of gears, gear ratio is also equal to the ratio of the speed of the smaller gear to that of the larger. In reduction gears this is also the ratio of input to output speeds. The usual range of ratios for single-reduction gear drives is indicated in Table 1.

Gear Efficiency: The efficiency, or ratio of output power to input, is affected by friction losses at the gear mesh and at the bearings, as well as by windage and churning of lubricant. Total losses may change very little with transmitted load, hence the efficiency is likely to decrease at less than full power.

Typical values of efficiency per mesh or stage of reduction are shown in Table 1.

Speed Range: Controlling factor in establishing the operating speed limits of gearing is the pitchline velocity. Limits for various types of gears are shown in Table 1.

Operating speeds are limited by accuracy of the gear teeth and balance of rotating parts.

Power Range: With high-precision gear teeth and the proper materials, gears can be designed to transmit very high power. Approximate maximum power transmitted through a single mesh is 30,000 hp for high-speed double-helical gears. Single-helical and herringbone gears transmit up to 30,000 hp, spiral bevels up to 5000 hp, and worm-gear sets up to 750 hp. In special cases, power may be higher than these limits.

Zerol[2] *gears*, Fig. 6, similar to spiral-bevel gears, have curved teeth arranged in such a way that the effective spiral angle is zero. Thrust loads, therefore, are equivalent to those produced by straight-bevel gears. Tooth bearing is localized, so that stress concentration at the tips is eliminated.

Face gears, having teeth cut on the flat face of the blank, mesh at right angles with spur or helical pinions. When the shafts intersect, they are known as on-center face gears.

Other types available include a number of proprietary designs. Coniflex[2] gears are straight-bevel gears whose teeth are crowned in the section of the pitch surface. Formate[2] gears are bevel gears in which the gear member of the pair has nongenerated teeth, usually with straight tooth profiles, and in which the pinion member of the pair has generated teeth that are conjugate to the mating gear. Revacycle[2] gears are straight-bevel gears generated by a special process, with a special tooth form.

Nonparallel, Nonintersecting Shafts: Shafts which lie in parallel planes may be skewed at any angle between zero and 90°. The following types of gears are used between such shafts.

Crossed helical gears, also known as spiral gears, Fig. 7, are similar in appearance to helical gears used on parallel shafts. However, they can only transmit relatively small amounts of power because of sliding action and limited tooth contact area. They are principally used to allow for a wide variety of speed ratios without change of center distance or even gear size. They can be used at angles other than 90°.

Worm gears, because of their screw action, Fig. 8*a*, are quiet, vibration-free, and produce a smooth output. On a given center distance, much higher ratios can be obtained through a wormgear set than with other conventional types of gearing. If the lead angle of the worm is less than 5°, a wormgear set cannot usually be driven backward—that is, it is self-locking. Although there is sliding action, the contact area is greater than with crossed helicals; hence load capacity is high. Worm and wormgear shafts are almost invariably at right angles.

Cavex[3] *Worm Gearing:* In this type, a concave worm thread meshes with a convex gear tooth. This arrangement is designed to provide lower tooth contact surface stresses and better conditions of mesh lubrication.

Cone-drive[4] *gearing* is a proprietary, double-enveloping, wormgear drive, Fig. 8*b*. This gearing is used to connect shafts at right angles. In usual speed ratios, a Cone-Drive set resembles hourglass wormgearing. In some extreme ratios the drive may resemble right-angle, crossed, helical gearing.

Hypoid gears, Fig. 9, are similar to spiral-bevel gears, except that the shafts do not intersect. They operate more smoothly and quietly, and are stronger for a given ratio. Because the two supporting shafts do not intersect, bearings can be mounted on both sides of the

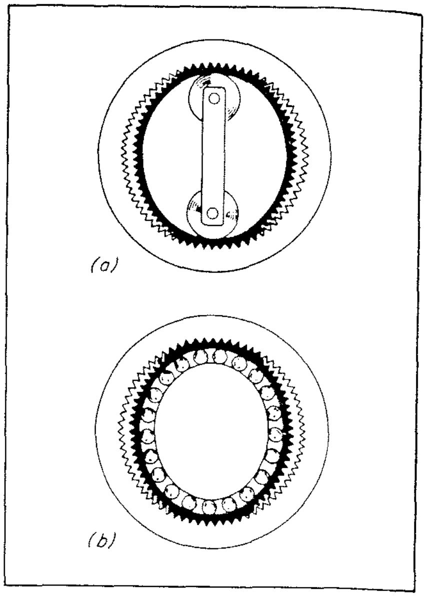


Fig. 4—Harmonic-drive principle, showing two means of generating the deflection wave—*a*, spaced rollers and, *b*, elliptoidal cam.

gear to provide extra rigidity. High-reduction hypoids (HRH)[2] permit ratios between 10:1 and 120:1, and even as high as 360:1 in fine pitches. Shaft angle is usually 90°, but other angles are possible.

Face gears, also used for intersecting shafts, can be offset to provide a right-angle nonintersecting-shaft drive.

Spiroid gears[5] are right-angle nonintersecting-axis gears having tapered pinions meshing with face-type gears, Fig. 10*a*. Having line contact across the full face width of the gear, with attendant load-carrying characteristics, Spiroid gears are suitable for ratios of 10:1 and larger. Ratios as high as 468:1 have been used.

Planoid gears[5], Fig. 10*b*, similar to hypoids, have load capacity comparable to Spiroid gears. They are suitable for ratios in the 2:1 to 10:1 range.

Helicon gears[5] are right-angle, nonintersecting gears having cylindrical pinions with threads of constant lead but unequal pressure angles meshing with teeth on the flat face of the gear, Fig. 10c. They have load-carrying characteristics similar to Spiroid gears. Helicon gears are suitable for ratios of 3:1 and higher. They have manufacturing advantages in low-ratio, high-quantity situations because the tooth form can be cast or molded in its correct theoretical form.

Beveloid gears[6] are involute gears with tapered tooth thickness, outside diameters, and root diameters, usually for intersecting axes but also employed for skew and parallel axes. They are suitable for precision instrument drives where their accuracy is an advantage and their limited load-carrying capacity is not a handicap. Accuracy of motion is not affected by mounting variations. A Beveloid gear will run with an involute spur or helical gear or rack. Unit cost is high except in large quantities.

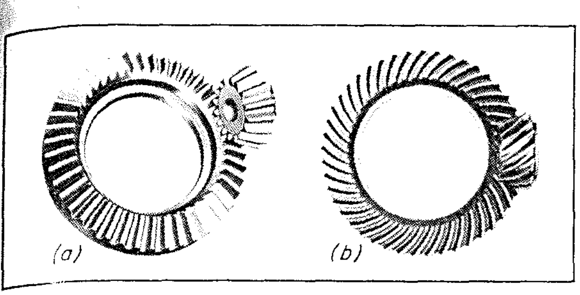


Fig. 5—Straight-bevel, *a*, and spiral-bevel pair, *b*.

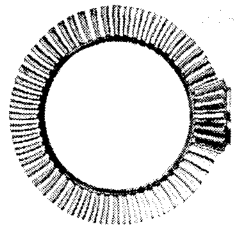


Fig. 6—Zerol gear pair.

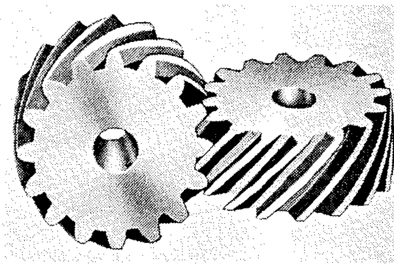


Fig. 7—Crossed helical gears, formerly called spiral gears.

Multiple Meshes

Combinations of two or more gear pairs on parallel and nonparallel, and on intersecting and nonintersecting shafts permit higher speed ratios and greater versatility than are possible with a single mesh, Table 2.

Series Trains: Overall ratio of any reduction gear train is the input shaft speed divided by the output speed. It is also the product of the individual ratios at each mesh, except in planetary arrangements. The ratio is most easily determined by dividing the product of the numbers of teeth of driven gears by the product of the numbers of teeth of driving gears. By manipulating numbers, any desired ratio can be obtained, either exactly or with an extremely close approximation.

In multiple-mesh series trains the forces transmitted through the gear teeth are higher at the low-speed end of the train. Therefore, the pitches and face widths of the gears are usually not the same throughout the train. In instrument gears, which transmit negligible power, this variation may not be necessary.

Speed increasers are gear trains that require great care in design, especially at high ratios. Because most gear sets and gear trains are intended for speed reduction, standards and published data in general apply to such drives. It is not safe to assume that these data can be applied without modification to a speed-increasing drive. Efficiency is sometimes lower in an increasing drive, which requires substantial input torques to overcome output load; in extreme instances, self-locking may occur. Clock and watch drives use special tooth profiles for minimum friction loss to avoid these difficulties.

Reverted trains are two sets of parallel-shaft gears so arranged that the output shaft is concentric with the input. The requirement of equal center distance for the two trains complicates determination of how many teeth should be in each gear to satisfy ratio requirements with standard pitches. Helical gears add flexibility to design through possible variation of helix angle.

Epicyclic Gearing: An epicyclic gear train is a reverted-gear arrangement in which one or more of the gears (planets) move around the circumference of coaxial gears, which may be fixed or rotating with respect to their own axes. The planet gears have a motion consisting of rotation about their own axes and rotation about the axes of the coaxial gears.

Epicyclic trains may incorporate spur or helical gears, external or internal, or bevel gears arranged in numerous ways. One common arrangement consists of: 1. Central sun gear with external teeth. 2. Ring gear with internal teeth. 3. Revolving planet pinions which engage the sun gear and the internal ring gear. 4. Planet carrier in which the planet pinions are supported.

Epicyclic trains can be classified as: *Single epicyclic trains*, which may be simple epicyclic trains, Fig. 11*a*, where coaxial gears *S* and *R* are connected by single intermediate gear *P* carried by planet carrier *A*; or compound epicyclic trains, Fig. 11*b*, where intermediate planet pinions *P* are compound gears.

Coupled epicyclic trains consist of two or more single epicyclic trains arranged so that two members in one train are common to the adjacent train.

Speed ratios are shown in Tables 3 and 4 for simple and compound epicyclic trains when one member is fixed and another is driving. With a suitable arrangement of clutches and brakes, an epicyclic train can be the basis of a change-speed transmission. With all members free and the gears locked to each other, an epicyclic train has a 1:1 speed ratio.

Differential Gearing: In a general sense, differential gearing is an arrangement in which the normal ratio of the unit can be changed by driving into the unit with a second drive. This arrangement, or one having two outputs and one input, is used to vary ratio. It is called the free type.

Simple differentials may use bevel gears, Fig. 12*a*, or spur gears, Fig. 12*b*. The bevel-gear type is used in automotive rear-end drives. Both types are used in analog-computer mechanisms where they perform addition or subtraction. The input-output speed relationship is $2\omega = \omega_1 - \omega_2$, where ω is the speed of the arm and ω_1 and ω_2 are the two shaft speeds.

Another type of differential (called the fixed type) has a large, fixed ratio. Such a drive is an evolution of the compound epicyclic train in Fig. 11*b*. If the ring gear is replaced with a sun gear which meshes with planet P_R, the equations in Table 4 apply, except that the plus signs change to minus. If the sun and planet gears are made almost but not exactly equal, the output speed is the small difference between two terms that are almost equal. Such a drive is good for ratios from 10:1 to 3000:1.

Using two ring gears instead of a ring and a sun in Fig. 11*b* results in a

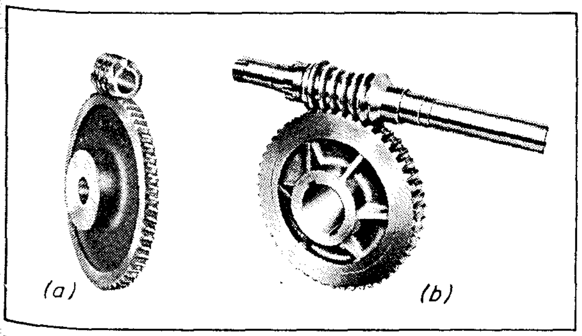


Fig. 8—Worm and wormgear pair, *a*; and double-enveloping Cone-Drive gearing, *b*.

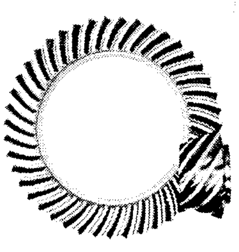


Fig. 9—Hypoid gears.

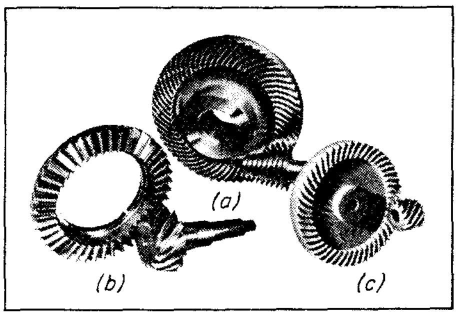


Fig. 10—Three types of gears used on crossed axes: *a*, Spiroid, *b*, Planoid, and *c*, Helicon.

Table 2—Speed Ratios Possible with Multiple Meshes

Type of Gear	Ratios
Helical, double-helical, herringbone gears	
Double reduction	10 to 75
Triple reduction	75 to 350
Spiral bevel (1st reduction), with helical gear	
Double reduction	9 to 50
Triple reduction	50 to 350
Wormgears	
Double reduction	100 to 8000
Helical with wormgears	
Double reduction	50 to 270
Planetary gearsets	
Simple	4 to 10
Compound	10 to 125

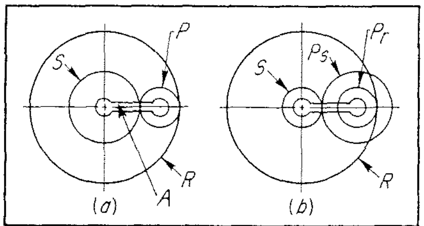


Fig. 11—Epicyclic trains: *a*, simple train and, *b*, compound train. In *b* planet gears are keyed to the same shaft.

fixed-differential drive suitable for ratios of 15:1 to 100:1.

Any epicyclic gear train can be designed for differential operation. For instance, instead of one of the elements, such as a sun or ring gear, being fixed, two of the elements may be driven independently. Output speed is then the net result of the two inputs.

Compound epicyclic trains can produce several input or output speeds by the addition of extra sun or ring gears meshing with the planet pinions. Thus, the compound epicyclic train in Fig. 11*b* may have another sun gear meshing with P_R and another ring gear meshing with P_S. Any one of the four (two suns and two rings) could be fixed, and the others free or available for input or output.

Closed-Loop Trains: Epicyclic gear trains with more than one planet pinion meshing with the same sun and ring gears have parallel paths through which power can flow. Other gear trains, too, sometimes use multiple-tooth contact to increase the capacity within a given space. Such gearing is sometimes called locked-train gearing.

Two considerations arise with multiple-tooth contact which are not present in open trains. One is the proper selection of tooth numbers and spacing to ensure assembly. The other is gear accuracy and adjustment to ensure equal distribution of the load to each mesh.

For single planetary arrangements, to make assembly possible, $(R + S)/Q$ must be a whole number, where R and S are the numbers of teeth in the ring and sun gears, and Q is the number of planets equally spaced around the sun.

For compound planetary trains the corresponding formula is $(RP_S - SP_R)/Q$, which must equal a whole number.

Multiple-mesh or locked-train gearing requires careful attention to tooth accuracy and support of the gears. Backlash and backlash tolerances should be as nearly equal as possible at each mesh to ensure equal load distribution.

Gear-Tooth Form

Gear-tooth profiles are commonly involutes. This shape gives a constant velocity ratio over the entire cycle of tooth contact, even if the center distance changes. Teeth are easily generated or formed with standard tools.

Involute tooth size and shape are affected by three variables—pitch, depth, and pressure (or profile) angle.

Pitch: Standard pitches are generally whole numbers when measured as diametral pitch P (number of teeth divided by pitch diameter of the gear in inches). Coarse-pitch gearing has teeth larger than 20 diametral pitch—usually 19.99 to 0.5. Fine-pitch gearing usually has teeth of diametral pitch 200 to 20.

Depth: This is standardized in terms of pitch. A standard full-depth tooth has a working depth equal to $2/P$. If the teeth have equal addenda (as in standard interchangeable gears) the addendum is therefore equal to $1/P$. Stub teeth have less working depth (usually 20% less) than full-depth teeth.

Full-depth teeth have a larger contact ratio than stub teeth. However, gears or pinions with small numbers of teeth are undercut. Undercut reduces the active profile and weakens the root of the tooth.

Pressure Angle: Standards are 20° and 25°. Earlier standards included a popular 14½° pressure angle which is still used. Pressure angle contributes to a force which tends to separate the gears. High pressure angle decreases the contact ratio but permits gears and pinions to have fewer teeth without undercutting.

Long and Short-Addendum Teeth: These permit pinions with even smaller numbers of teeth. The addendum of the teeth on the smaller gear (pinion) is increased and that of the teeth on the larger gear is decreased, leaving the whole depth, or sum of the two addenda, the same.

Standard Tooth Forms: Standards published by the American Gear Manufacturers Association (AGMA) establish tooth proportions and profiles.

Gear Precision and Quality

Because cost increases sharply with increasing accuracy, it is important to establish a degree of precision appropriate to the application. Silent gears and gears which transmit high power at high speed demand high precision and quality. So do instrument and computer gears, where speed or displacement ratios must be exactly maintained. Noncritical gears, such as might be used at slow speed in exposed locations, may be satisfactory if made to minimum quality standards.

Gear Classification: AGMA has established a gear classification system to assist in the selecton and specification of gears.

Gears are assigned an AGMA Class Number which combines a dimensional Quality Number (identifying tooth-element tolerances) and a Material and Treatment Number. A complete Class Number might read 8-H-14. The first number, 8, is the quality number; the *H* denotes the type of material and treatment; and the 14 is related to the hardness.

Quality Numbers range from 3 to 15 for coarse-pitch gearing (19.99 to 0.5 diametral pitch) and from 5 to 16 for fine-pitch gearing (200 to 20 diametral pitch). The higher the quality number, the more precise the gearing, and the closer the tolerances.

Tolerances for coarse-pitch gearing are described and listed in the AGMA *Gear Classification Manual:*

- Runout tolerance is the allowable variation in off-center relation between the axis of the pitch circle and the axis about which the gear rotates. It is twice the eccentricity, and includes the effect of side runout or wobble. It is applicable to all Numbers.
- Pitch tolerance is the allowable variation in spacing between corresponding sides of adjacent teeth, with measurements preferably made at or near the pitch circle in the plane of rotation. Applicable to Number 6 and higher.
- Profile tolerance is the allowable deviation of the profile of the actual tooth form from the theoretically correct profile, with measurement made in the

Table 3—Simple Epicyclic Train Ratios
(See Fig. 11*a*)

Condition	Revolution of — Sun	Arm	Ring
Sun fixed	0	1	$1 + \frac{S}{R}$
Arm fixed	1	0	$-\frac{S}{R}$
Ring fixed	$1 + \frac{R}{S}$	1	0

Table 4—Compound Epicyclic Train Ratios
(See Fig. 11*b*)

Condition	Revolution of — Sun	Arm	Ring
Sun fixed	0	1	$1 + \frac{SP_R}{P_S R}$
Arm fixed	1	0	$-\frac{SP_R}{P_S R}$
Ring fixed	$1 + \frac{P_S R}{SP_R}$	1	0

In the equations, R, S, P_R, and P_S denote either pitch diameter or number of teeth in the respective gears.

plane of rotation. Tip relief and any portion of the tooth surface below the active profile is not considered. Applicable to Number 7 and higher.

- Total composite tolerance is the allowable variation in center distance obtained when a gear is rotated through one complete revolution in tight mesh with a master gear. This includes the effects of all tooth and runout variations. Applicable to Numbers 8 to 12.
- Lead tolerance is the allowable deviation of the axis of the tooth from its true axis, with the measurement as a displacement normal to the axis. Applicable to Number 8 and higher.

For fine-pitch gearing, tooth-to-tooth composite tolerances and total composite tolerances are listed for Numbers 5 through 16.

Tooth-to-tooth composite tolerance is the allowable center distance variation obtained as the gear is rotated from tooth to tooth in tight mesh with a master gear for one complete revolution of the gear. It includes the effects of variations in circular pitch, tooth thickness, and profile.

In many applications requiring high-accuracy gearing it is necessary to match pinion and gear profiles, and pinion and gear leads, so that the mismatch in a set does not exceed the tolerance on profile or the tolerance on lead for the applicable Quality Number. Matched sets can be provided, but usually involve extra cost.

Certain control-gearing applications require gears having a high degree of accuracy in the circular spacing of teeth. For such applications, specification of the tooth spacing is required in addition to the accuracy class specifications.

Backlash of zero is required in some gearing applications. However, most applications only require free rotation of the engaging teeth in mating gears. In fact, a tight mesh may result in objectionable gear sound, more power losses, overheating, rupture of the lubricant film, overloaded bearings, and premature gear failure.

The following factors must be considered in establishing backlash tolerances: center-distance tolerance, parallelism of gear axes, side runout or wobble, tooth-thickness tolerance, pitchline runout tolerance, profile tolerance, pitch tolerance, lead tolerance, types of bearings and subsequent wear, deflection under load, gear-tooth wear, pitchline velocity, lubrication requirements, and thermal expansion of gears and housing.

For most gearing applications the AGMA-recommended backlash ranges provide proper running clearance between engaging teeth of mating gears. Recommended tolerances are a function of center distance and pitch.

Material and treatment are shown by the AGMA Material Number's two terms —a letter code designating material and treatment, and a number indicating hardness. The higher the number, the harder the material.

Preliminary Rating of Gear Sets

Standard calculation procedures are available for rating gears in noncritical applications. For high-speed, high-capacity gears operating under severe conditions, these procedures serve to approximate size and space requirements. Final dimensions and specifications should be checked by more rigorous methods.

Power that a gear set can transmit depends on the strength and wear resistance of the gear teeth. Ratings for strength and durability should be calculated separately and consideration given to each when a gear set is rated.

Strength and durability figures calculated from formulas determine the service rating for the gears after application of factors relating to duration and nature of service.

Durability and strength should be calculated, and the appropriate service factor should be applied to the smaller value, for the rating of the gear set. When it is necessary to design a gear set to satisfy given service conditions, power, speed, etc., it is necessary to assume dimensions and material, then check the assumed specifications.

Materials and Heat Treatment

Depending on the service, gears may be made of a number of materials including plastics, wood, bronze, iron, and steel. But those that transmit heavy loads are usually ferrous metal or bronze.

Considerations in choice of material are allowable stress, shock resistance, and wear resistance. Heat-treated steel meets most of these qualifications. Where wear is relatively severe, as with worm gearing, a high-quality, chill-cast, nickel bronze may be used for rim material. Smaller worm gears may be entirely of nickel bronze.

Heat Treatment: Gears may be case-hardened, through-hardened, nitrided, or precipitation-hardened for the proper combination of toughness and tooth hardness. Methods used to achieve these properties are furnace hardening, carburizing, induction hardening, and flame hardening.

The use of high-hardness heat-treated steels permits smaller gears for given loads. Also, hardening can increase service life up to ten times without increasing size or weight. But the gear must have at least the accuracies associated with softer gears and, for maximum service life, even greater precision.

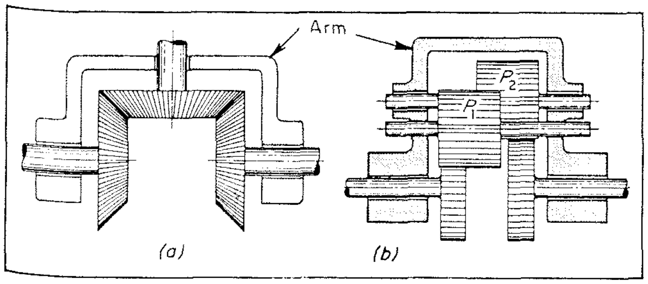
Heat-treat distortion must be minimized if the gear is to have increased service life. Several hardening techniques have proved useful. For moderate service-life increases, gears are hardened but kept within the range of machinability so that distortion produced by heat treatment can be machined away. Where maximum durability is required, surface hardening to 60 Rc is necessary. Carburizing, nitriding, and induction hardening are generally used. However, precision gearing (Quality 10 or better) can only be assured by finishing after hardening.

Full-contour induction hardening is an economical and effective method for surface-hardening gears. The extremely high but localized heat allows small sections to come to hardening temperatures while the balance of the gear dissipates heat. Thus, major distortions are eliminated.

While conventional methods such as flame hardening reduce wear by hardening the tooth flank, gear strength is not necessarily improved. In fact, stresses built up at the juncture of the hard and soft material may actually weaken the tooth. Induction hardening provides a hardened tooth contour with a heat-treated core to increase both surface durability and tooth strength. The uniformly hardened tooth surface extends from the flank, around the tooth, to the flank. No stress concentrations are developed to impair gear life.

Nitriding is a satisfactory method of hardening small and medium-size gears. Distortion is minimal because furnace temperatures are comparatively low. Hardening pattern is uniform but depth of hardness is limited. Best results are achieved when special materials, suited to nitriding, are specified.

Shot peening can add 10 to 15%



12—Simple differentials: *a*, bevel gears, and, *b*, spur gears.

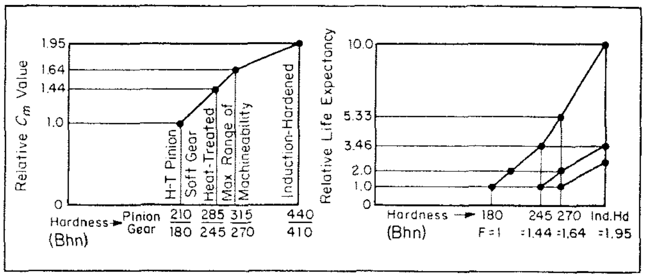


Fig. 13—Effect of gear hardness on durability and life.

additional strength to a gear. It is used for optimum strength.

Fig. 13 shows durability rating and life expectancy as functions of hardness.

Manufacturing Methods

Gears can be formed by various processes that can be classified under the general heads of milling, generating, and molding.

Milling: Almost any tooth form can be milled. However, only spur, helical, and straight-bevel gears are usually milled. Surface finish can be held to 125 μin.

Generating: In the generating process teeth are formed in a series of passes by a generating tool shaped somewhat like a mating gear. Either hobs or shapers can be used.

Hobbing can produce almost any external tooth form except bevel-gear teeth which are generated with face-mill cutters, side-mill cutters, and reciprocating tools. Hobbing closely controls tooth spacing, lead, and profile. Surface finishes as fine as 63 μin. can be obtained.

Precision hobbing methods have been developed to provide the capability of manufacturing gears of medium hardness to extreme accuracy levels (Quality 12 to 14) previously obtainable only by shaving and grinding methods.

Shapers are reciprocating pinion or rack-shaped tools. They can produce external and internal-spur, helical, herringbone, and face gears. Shaping is limited in the length of cut it can produce. Surface finishes as fine as 63 μin. are possible.

Molding: Large-volume production of gears can often be achieved by molding. Injection molding produces light gears of thermoplastic. Diecasting is a similar process using molten metal. Zinc, brass, aluminum, and magnesium gears are made by this process.

Sintering is used in small, heavy-duty gears for instruments and pumps. Iron and brass are the materials most used.

Investment casting and shell molding produce medium-duty iron and steel gears for rough applications.

Gear Finishing: To improve accuracy and finish, gears may be shaved. Shaving removes only a small amount of surface metal. A very hard mating gear with many small cutting edges is run with the gear to be shaved. Surface finish can be as fine as 32 μin.

Lapping corrects minute heat-treatment distortion errors in hardened gears. The gear is run in mesh with a gear-shaped lapping tool, or another mating gear. An abrasive lapping compound is used between them. Lapping improves tooth contact, but does not increase accuracy of the gear. Finish is on the order of 32 μin.

Grinding is the most accurate of tooth-finishing processes. Profiles can be controlled or altered to improve tooth contact. For example, barrelling or crowning the flanks of teeth promotes good center contact where the tooth is strong, and minimizes edge and corner contact where the tooth is unsupported.

Honing is used mainly for removing nicks and burrs, and improving surface finish. However, it has also been found valuable in certain instances to improve accuracy.

Enclosed Trains

Enclosures: Housings for industrial gear drives are generally high-quality gray iron, cast steel, or welded steel. For special requirements, malleable iron, ductile iron, aluminum,, and magnesium are used. Walls should be of sufficient thickness and adequately reinforced not only to withstand the most severe stresses encountered during operation, but also to provide the rigidity necessary to maintain gear alignment and mesh.

Bearings: Gear loads on shafting consist of three elements: tangential force transmitted between gears, radial force component due to pressure angle, and axial thrust. The tangential and radial forces combine to produce a result which must be supported by radial-type bearings. Axial loads need thrust bearings.

Antifriction (ball and roller) and sleeve-type bearings are used in gear drive systems. With gears which produce thrust loads, such as single helicals, bevel gears, and wormgears, bearings must take these thrusts as well as radial loads.

Shaft Oil Seals: Four basic types of shaft oil seals are used in gear housings: stuffing boxes, standard leather or synthetic-rubber lip-type seals, labyrinth seals using the centrifugal action of a rotating flinger as the means of retaining oil, and lapped-seat seals generally consisting of a lapped steel-to-graphite sealing surface.

Stuffing boxes and lapped-seat seals are satisfactory for retaining a head of oil, while lip-type and labyrinth-type seals keep splash from passing around the shaft. Labyrinth-type seals and lapped-seat seals are recommended for high shaft speeds.

Lubrication: Lubricating oils for enclosed gears should be high-grade, high-quality, refined, straight-mineral oils. For heavy-duty gears, mild extreme-pressure (EP) oils can be used. Lubricants for wormgears should have additions of 3 to 10% of acidless tallow or similar animal fats.

Lubricants should not be corrosive to gears or bearings, should have good defoaming properties, and be free from grit and abrasives. If high operating temperatures are encountered, resistance to oxidation is necessary. For low temperatures, a low pour point is necessary.

When possible, splash lubrication is used. When this is not effective to carry oil to gears and bearings, an oil pump is used for force-feed lubrication.

Bearings may also be greased, if they are shielded from the oil, through pressure fittings in the housing.

Grease should not be corrosive nor have grit, abrasive, or fillers; should not precipitate sediment; should not separate at temperatures to 300 F; should have moisture-resisting characteristics, and good resistance to oxidation.

Heat Dissipation: The rated power of the unit should not exceed the thermal rating—the maximum average power that can be transmitted continuously for three hours or more without undue rise in temperature and without auxiliary cooling.

The ability of the enclosure to dissipate heat will increase as the size of the unit is increased, all other conditions remaining the same. Fins are often used on the housings of worm-gear reducers to increase the surface area, and thus increase the heat-dissipating ability of the unit.

If the thermal power is greater than the rated power, external means must be used to keep the unit temperature down. This is usually accomplished with an oil cooler in the lubrication system.

Noise: At present it is not possible to manufacture gear units that are completely noise-free. However, it is possible to keep the noise level within acceptable limits.

Noise in gear drives is mainly caused by: 1. Inaccuracies in the gear action. 2. Vibrations set up in machine parts and foundations by irregular action of the gears or unbalance in rotating parts. 3. Resonant vibrations in gear units and foundations, both lateral and torsional, due to critical-speed conditions. 4. Bearing noises, particularly those produced by antifriction bearings. 5. Lubricant pumping and churning noises. 6. Windage.

Noise from the first three causes can be reduced by improving the precision of the teeth, by balancing, designing for rigidity, and avoiding critical speeds.

Tradenames

1. United Shoe Machinery Corp. (proprietary concept). 2. Gleason Works. 3. Philadelphia Gear Corp. 4. Cone-Drive Gears Div., Michigan Tool Co. 5. Spiroid Div., Illinois Tool Works. 6. Vinco Corp.

Acknowledgment

The authors acknowledge with appreciation the courtesy of the following companies in supplying illustrations for this article: Boston Gear Works—Fig. 1*a*, 8*a*; Ferguson Machine Co.—Fig. 3; Gleason Works—Fig. 5*b*, 6, 9; Harmonic Drive Div., United Shoe Machinery Corp.—Fig. 4; Illinois Tool Works—Fig. 10; Cone-Drive Gears Div., Michigan Tool Co.—Fig. 8*b*.