Fundamentals, Selection, Installation and Maintenance of Gearboxes (Gear Drives) - Part 1

By
K. P. Shah
Email: kpshah123[at]gmail.com (Please replace [at] with @)
Committed to improve the Quality of Life

The information contained in this booklet represents a significant collection of technical information on fundamentals, selection, installation and maintenance of gearboxes (gear drives). This information will help to achieve increased reliability at a decreased cost. Assemblage of this information will provide a single point of reference that might otherwise be time consuming to obtain. Most of information given in this booklet is taken from various sources for which information is given in each chapter and in the references list given at end of this booklet. For more information, please refer them. All information contained in this booklet has been assembled with great care. However, the information is given for guidance purposes only and the ultimate responsibility for its use and any subsequent liability rests with the end user. Please see the disclaimer uploaded on http://www.practicalmaintenance.net.

(Edition: January 2017)

Content

Notes:

Drives using gears should be called gear drives because a gear drive may not have a box (enclosure). However, as gear drives are commonly called gearboxes, I have used gearboxes instead of gear drives in chapters 1 & 7 of Part 2 and in the title.

For information on the following, please view Part 2.

- Storage, Installation and Commissioning of Gearboxes (Gear Drives)
- Tooth Contact Checking
- Lubrication of Gear Drives
- Lubricant Properties
- Lubrication System Maintenance
- Condition Monitoring of Gear Drives
- Inspection and Maintenance of Gearboxes (Gear Drives)
- Gear Wear and Failure Modes

Use of Gear Drives

Gear drives are widely used where changes of speed, torque or change in direction of rotation are required between a prime mover and the driven machinery.

In many power transmission applications, the preferred prime mover operates at a relatively high speed because of the better economy and efficiency of high speed motors, gas and steam turbines, etc. The driven equipment, however, often requires a much lower shaft speed and high torque. The gear drive not only reduces shaft speed to the value needed to operate the driven machine but also converts the relatively low torque output of the high speed prime mover to the high torque needed to drive the low speed driven device.

Rotary compressors operate more efficiently at high shaft speed and often require a speed increasing gear drive. In most cases, increasers are not simply speed reducers driven backward but involve design considerations different from those encountered in reducers.

In most industrial locations, gears are supplied within a gearbox. Gearboxes can be purchased that are single, double, or triple reduction, depending on the number of gears and shafts.

Information about gear nomenclature; types of gears; selection of enclosed gear drives; storage, installation and commissioning of gearboxes; lubrication of gear drives; tooth contact checking; condition monitoring of gear drives and inspection & maintenance of gearboxes is given in the following chapters (Part 1 and Part 2).

Gear Nomenclature and Tooth Proportions

Information about gear nomenclature and tooth proportions is given in this chapter. The information is illustrated through the spur gear tooth because it is the simplest and hence most comprehensible.

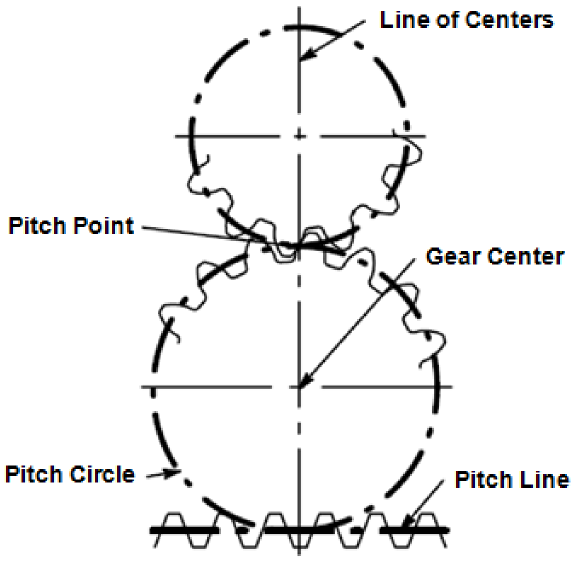
Gear Nomenclature

Following are the definitions of the commonly used gear terms. Most of the information given in this section is based on ANSI/AGMA 1012-G05.

Pitch Circles

Pitch circles are the imaginary circles that are in contact when two standard (also called theoretical or reference) gears are in correct mesh. The diameters of these circles are the pitch diameters (*D*) of the gears.

The **pitch line** corresponds, in the cross section of a rack, to the pitch circle in the cross section of a gear.



Gear Center, Line of Centers and Pitch Point

Above figure shows gear center, line of centers and pitch point.

Gear Center

A gear center is the center of the pitch circle.

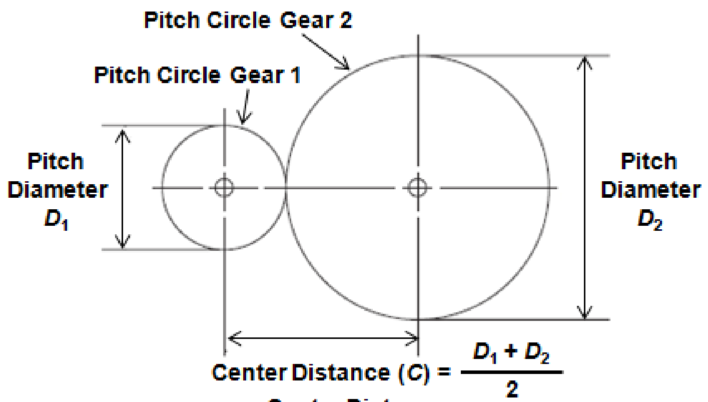
Line of Centers

The line of centers connects the centers of the pitch circles of two engaging gears. When one of the gears is a rack, the line of centers is perpendicular to its pitch line.

Pitch Point

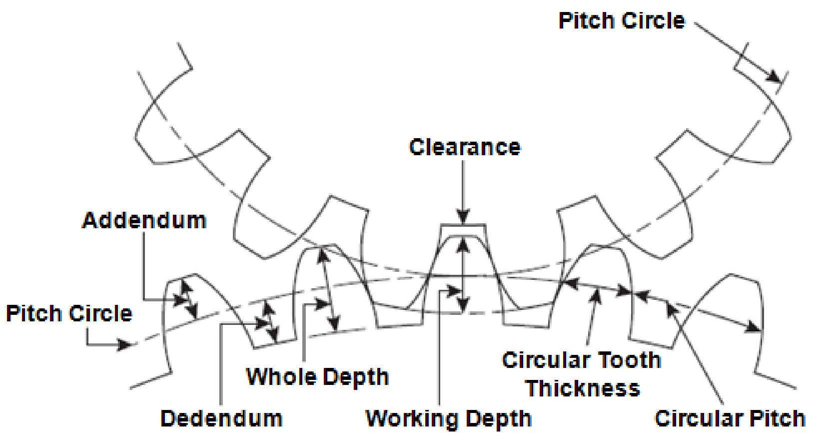
The pitch point is the point of tangency of two pitch circles and is on the line of centers.

Center Distance (*C*)



Center Distance

As shown in above figure, the center distance (*C*) of two gears in correct mesh is equal to one-half the sum of the two pitch diameters.



Gear Nomenclature

Above figure shows important nomenclature for a gear.

Addendum (*a*) is the distance a tooth projects beyond (outside for external, or inside for internal) the standard pitch circle or pitch line; also, the radial distance between the pitch circle and the addendum circle.

Dedendum (*b*) is the depth of a tooth space below, or inside of, the standard pitch circle or pitch line. It is normally greater than the addendum of the mating gear to provide clearance.

Clearance (*c*) is the amount by which the dedendum of a gear tooth exceeds the addendum of a mating gear tooth. It compensates for thermal expansion that occurs during operation and prevents the top of a gear tooth from interfering with the root of its mating gear tooth.

Whole depth (h_t) is the total height of a tooth or the total depth of a tooth space.

Working depth is the depth of tooth engagement of two mating gears. It is the sum of their addendums.

Tooth thickness is the length of arc between the two sides of a gear tooth on the pitch circle or the distance between the two sides of a gear tooth along the pitch line, unless otherwise specified.

Circular Pitch (*p*) is the distance along the pitch circle or pitch line between corresponding profiles of adjacent teeth.

Hence,

$p = (\pi \times D) / N$

Where,

p = circular pitch
N = number of teeth
D = pitch diameter

Diametral Pitch (P_d)

Diametrical pitch, usually a whole number, is the ratio of the number of teeth to the pitch diameter in inches. Expressed mathematically,

$P_d = N / D$

Where,

P_d = diametrical pitch
N = number of teeth
D = pitch diameter in inches

Thus, it specifies the number of teeth per inch of pitch diameter.

Stated another way, diametral pitch specifies the number of teeth in 3.1416 inch along the gear's pitch line (because, circumference of a circle = π × diameter = 3.1416 × diameter).

Therefore, the diametral pitch determines the size of the gear tooth. As a rule of thumb, teeth should be large and low in number for heavily loaded gears, small and numerous for smooth operation.

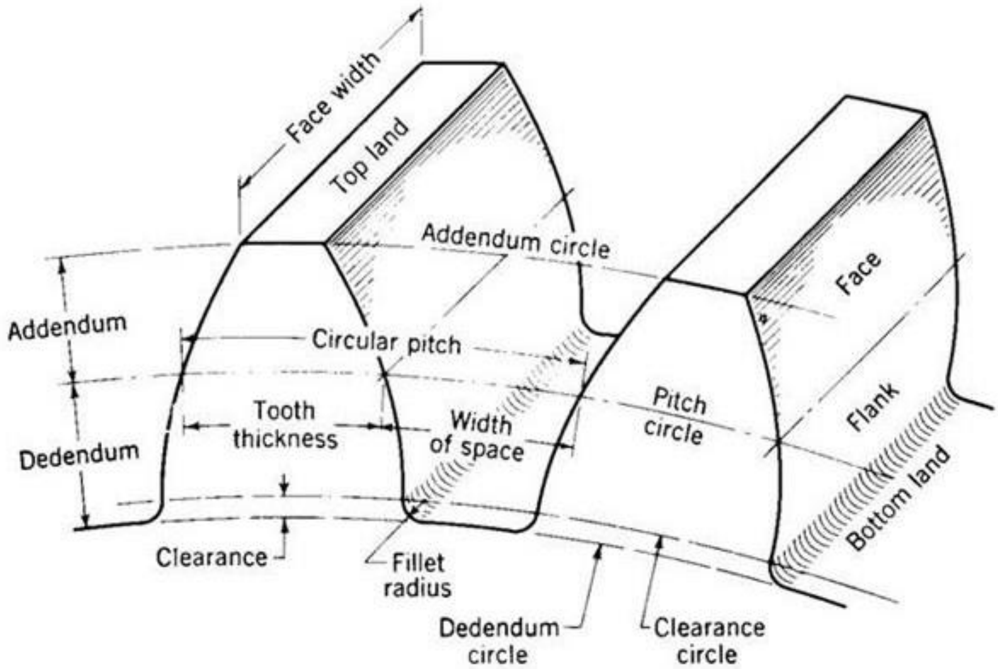
It is important to note that meshing gears must have the same diametrical pitch.

The diametral pitch system is used in the English system. Metric system uses the module in place of the diametral pitch.

Module (*m*)

The module is the metric index of tooth sizes and is always given in millimeters. It is the ratio of the pitch diameter (in mm) to the number of teeth. The relationship between module and diametrical pitch is as under.

$m = 25.4 / P_d$



Gear Nomenclature - Additional

Above figure shows some additional nomenclature for gears.

Addendum Circle coincides with the tops of the teeth and is concentric with the standard (reference) pitch circle and radially distant from it by the amount of the addendum.

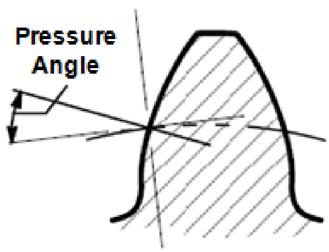
Fillet Radius is the radius of the fillet curve at the base of the gear tooth.

Land: The top land is the top surface of a gear tooth and the bottom land is the surface of the gear between the fillets of adjacent teeth.

Face/Top of tooth is the surface between the pitch circle and the top land.

Flank of tooth is the surface between the pitch circle and the bottom land, including the gear tooth fillet.

Pressure Angle (ϕ)



Pressure Angle

Pressure angle is in general the angle at a pitch point between the line of pressure (direction of force created by driving gear) which is normal to the tooth surface, and the plane tangent to the pitch surface.

The 14½° pressure angle was standard for many years; however, the use of the 20° pressure angle has grown. Today, 14½° gearing is generally limited to replacement work. In general, higher pressure angles provide higher strengths and a lower tendency for tooth tip interference, but are susceptible to noise and higher bearing loads.

Base Diameter (D_b)

Base diameter (D_b) is the diameter of the base circle. Base circle is the circle from which the involute tooth profile is generated or developed. Relationship between base diameter and pitch diameter is as under.

Base Diameter (D_b) = $D \cos\phi$

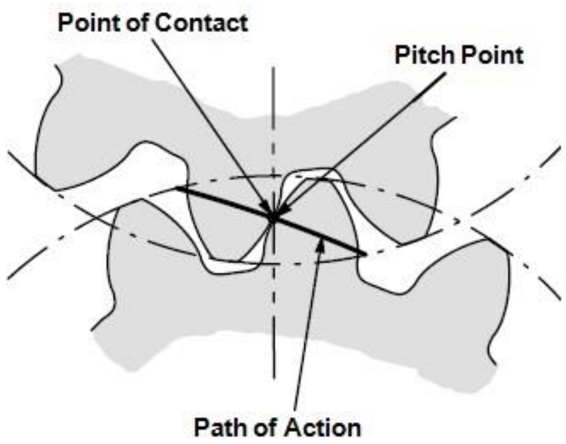
Where,

D = pitch diameter
ϕ = pressure angle

Point of Contact

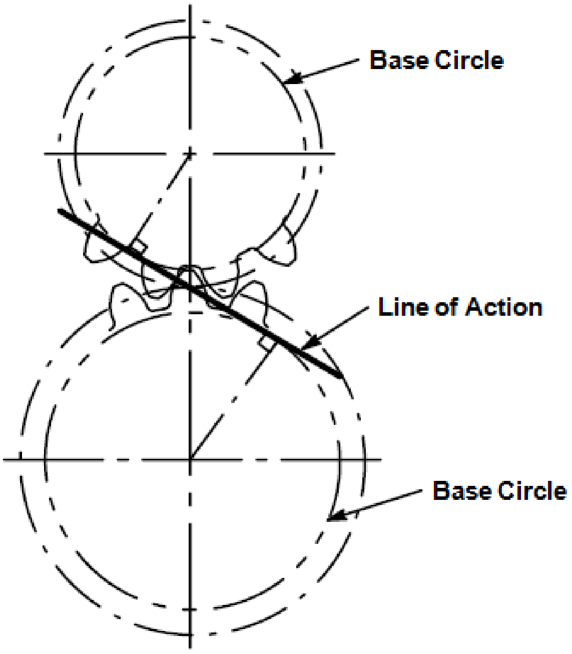
A point of contact is any point at which two tooth profiles touch each other.

Path of Action



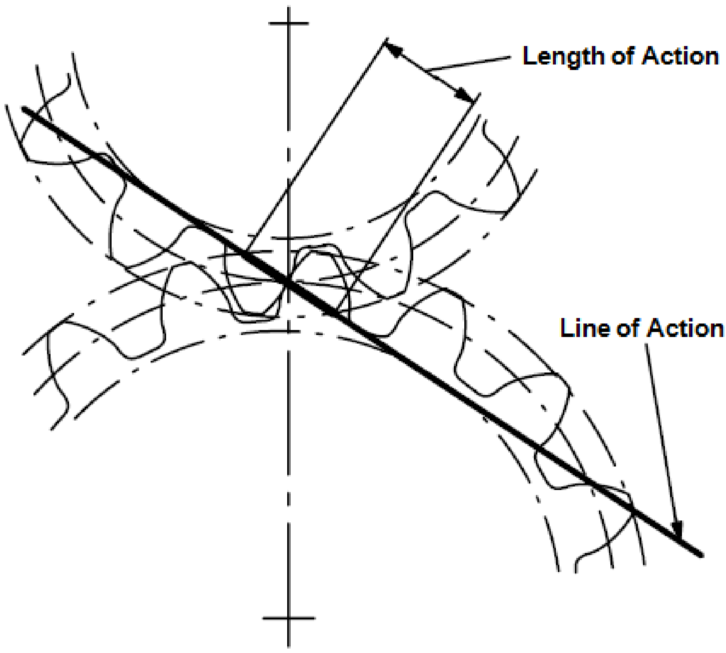
The path of action is the locus of successive contact points between a pair of gear teeth, during the phase of engagement.

Line of Action



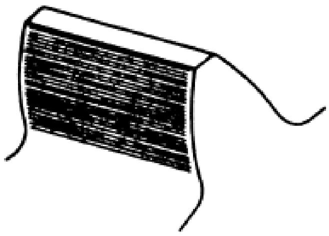
The line of action is the path of action for involute gears. It is the straight line passing through the pitch point and tangent to both base circles.

Length of Action



Length of action is the distance on the line of action through which the point of contact moves during the action of the tooth profile.

Active Profile



Active Profile

Active profile is a surface (shaded area in above figure) and is that portion of the surface of the gear tooth which at some phase of the meshing cycle contacts the active profile of the mating gear tooth.

Basic Considerations for Gear Tooth Design

Gear teeth are a series of cam surfaces that contact similar surfaces on a mating gear in an orderly fashion. To drive in a given direction and to transmit power or motion smoothly and with a minimum loss of energy, the contacting cam surface on mating gears must have the following properties:

- The height and the lengthwise shape of the active profiles of the teeth (cam surfaces) must be such that, before one pair of teeth goes out of contact during mesh, a second pair will have picked up its share of the load. This is called continuity of action.

- The shape of the contacting surfaces of the teeth (active profiles) must be such that the angular velocity of the driving member of the pair is smoothly imparted to the driven member in the proper ratio.

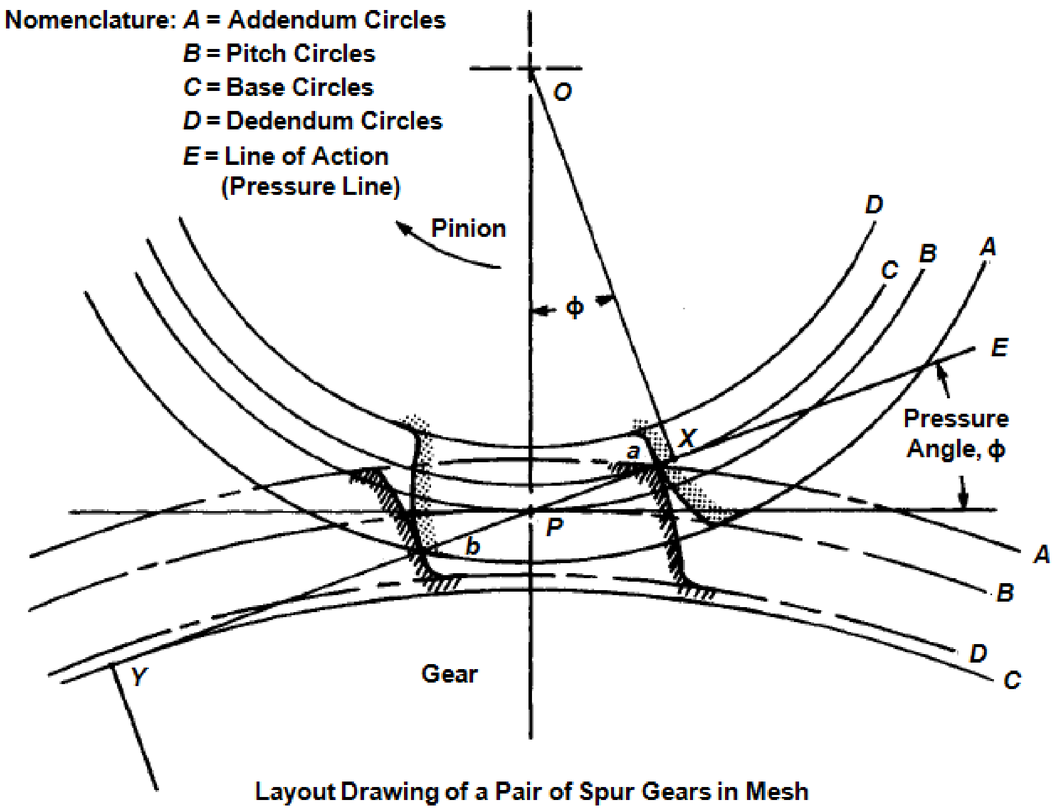
Many different shapes of surfaces can be used on the teeth to produce uniform transmission of motion. Curves that act on each other with a resulting smooth driving action and with a constant driving ratio are called conjugate curves. The fundamental requirements governing the shapes that any pair of these curves must have are summarized in Willis' "basic law of gearing" (1841), which states, "normals to the profiles of mating teeth must, at all points of contact, pass through a fixed point located on the line of centers".

In the case of spur and helical type gears, the curves used almost exclusively are those of the involute family. In this type of curve, the fixed point mentioned in the basic law is the pitch point. Since all contact takes place along the line of action, and since the line of action is normal to both the driving and driven involutes at all possible points of contact, and, lastly, since the line of action passes through the pitch point, it can be seen that the involute satisfies all requirements of the basic law of gearing. Many years ago, the cycloidal family of curves was in common use; however, clockwork gears are the only application of this type of tooth today. The involute curve has supplanted the cycloid because of its greater ease of design and because it is far less sensitive to manufacturing and mounting errors.

It may be noted that worm gearing, like bevel gearing is non-involute. The tooth form of worm gearing is usually based on the shape of the worm; that is, the teeth of the worm gear are made conjugate to the worm. In general, worms can be chased, as on a lathe, or cut by such processes as milling or hobbing, or can be ground. Each process, however, produces a different shape of worm thread and generally requires a different shape of worm gear tooth to run properly.

Pressure Angle, Undercutting and Contact Ratio in Involute Gearing

In involute teeth gears, pressure angle is often described as the angle between the line of action and the line tangent to the pitch circle (a line perpendicular to the line of centers).



Layout Drawing of a Pair of Spur Gears in Mesh

In above figure, line *OP* is the line of centers connecting the rotation axes of a pair of meshing gears. Line *E* is the line of action (the pressure line) and the angle ϕ is the pressure angle. The resultant force vector between a pair of operating gears acts along this line.

The line of action is tangent to both base circles *C*, at points *X* and *Y*.

Line *ab* is the length of action. Point *a* is the initial point of contact. This point is located at the intersection of the addendum circle of the gear with the line of action. Point *b* in the figure is the final point of contact. This point is located at the intersection of the addendum circle of the pinion with the line of action.

Line *aP* represents the approach phase of tooth contact and line *Pb* represents the recess phase. Tooth contact is a sliding contact throughout the line of action except for an instant at *P* (pitch point) when contact is pure rolling.

It can be seen that the maximum length of the length of action is limited by the length of the common tangent. Any tooth addendum that extends beyond the tangent points (*X* and *Y*) is not only useless, but interferes with the root fillet area of the mating tooth. This results in the undercut tooth. The undercut not only weakens the tooth, but also removes some of the useful involute adjacent to the base circle.

From the geometry of the limiting length of action (*X*-*Y*), it is evident that interference is first encountered by the addenda of the gear teeth digging into the mating pinion tooth flanks.

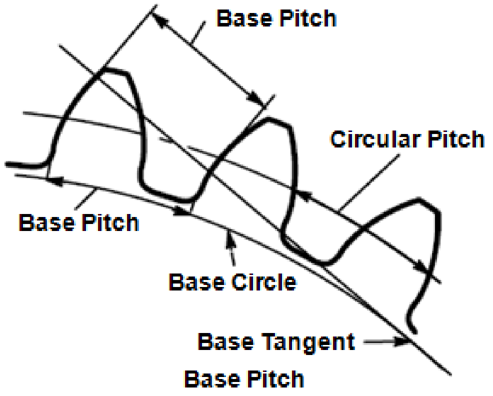
The operating diameters of the pitch circles depend on the center distance used in mounting the gears, but the base circle diameters are constant and depend only on how the tooth forms were generated because they form the base or the starting point of the involute profile.

In view of this, instead of using the standard/theoretical pitch circle as an index of tooth size, many times the base pitch p_b is used. The base pitch p_b is related to the circular pitch p by the following equation.

$p_b = p \cos\phi$

Where,

p_b = base pitch
p = circular pitch
ϕ = pressure angle



In the figure "layout drawing of a pair of spur gears in mesh", if the distance from *a* to *b* exactly equals the base pitch, then, when one pair of teeth is just beginning contact at *a*, the preceding pair will be leaving contact at *b*. Thus, for this special condition, there is never more or less than one pair of teeth in contact. If the distance *ab* is greater than the base pitch but less than twice as much, then when a pair of teeth come into contact at *a*, another pair of teeth will still be in contact somewhere along the length of action *ab*.

To assure smooth continuous tooth action, as one pair of teeth ceases contact, a succeeding pair of teeth must already have come into engagement. It is desirable to have as much overlap as possible. The measure of this overlapping is the contact ratio. This is a ratio of the length of action to the base pitch.

Hence, the contact ratio m_c can be defined by the following equation.

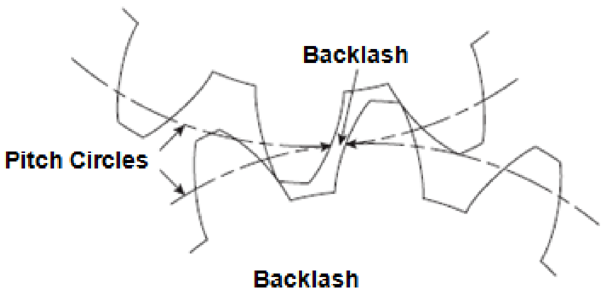
$m_c = L_{ab} / p_b$

Where L_{ab} = distance *ab*, the length of action.

Contact ratio in general is the number of angular pitches (angular pitch is the angle subtended by the circular pitch) through which a tooth surface rotates from the beginning to the end of contact. It is sometimes thought of as the average number of teeth in contact.

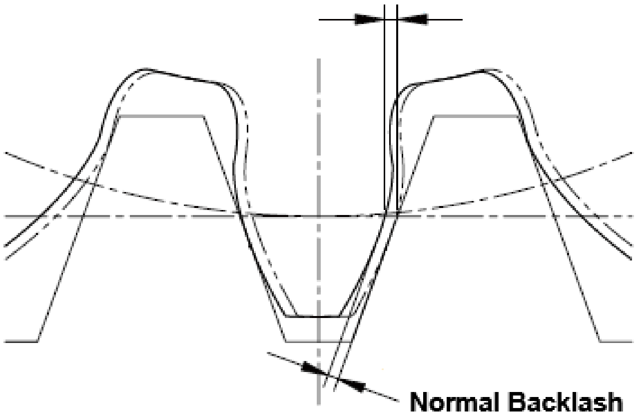
The American Gear Manufacturers Association (AGMA) recommends that the contact ratio for spur gears should not be less than 1.2.

Backlash



In terms of tooth dimensions, backlash is the amount by which the width of tooth space exceeds the thickness of the engaging tooth on the pitch circles as shown in above figure.

Normally there must be some backlash present in gear drives to provide running clearance. This is necessary as the binding of mating gears can result in heat generation, noise, abnormal wear and damage to the drive. It also provides space for lubrication of the gears. A small amount of backlash is also desirable because of the dimensional variations involved in manufacturing (tolerances) and to compensate for the effects of thermal expansion.



Transverse Backlash and Normal Backlash

Backlash is generally measured either in the direction of the pitch circles/lines (transverse / circumferential backlash) or on the line of action (normal backlash).

For spur and helical gears, the relationship between the backlashes is: Normal Backlash = Transverse Backlash x cosine of the pressure angle of gear teeth x cosine of the helix angle of helical gear.

On non-reversing drives, or drives with continuous load in one direction, the increase in backlash that results from tooth wear does not adversely affect operation.

Backlash is built into standard gears during manufacture by cutting the gear teeth thinner than normal by an amount equal to one-half the required figure. When two gears made in this manner are run together, at standard center distance, their allowances combine to provide the full amount of backlash required.

Following table lists the suggested transverse backlash for a pair of gears operating at the standard center distance.

Diametral Pitch	Backlash (inches)
3	0.013
4	0.010
5	0.008
6	0.007
7	0.006
8-9	0.005
10-13	0.004
14-32	0.003
33-64	0.0025

An increase or decrease in center distance will cause an increase or decrease in backlash.

Since, in practice, some deviation from the theoretical standard center distance is inevitable and will alter the backlash, such deviation should be as small as possible. For most applications, it would be acceptable to limit the deviation to an increase over the nominal center distance of one half the average backlash.

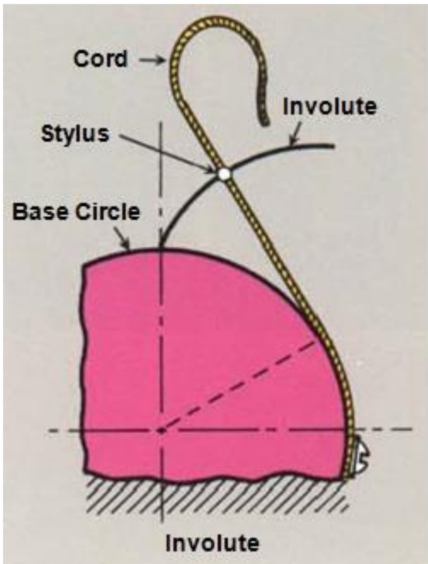
The approximate relationship between center distance and backlash change of 14½° and 20° pressure angle gears is shown below:

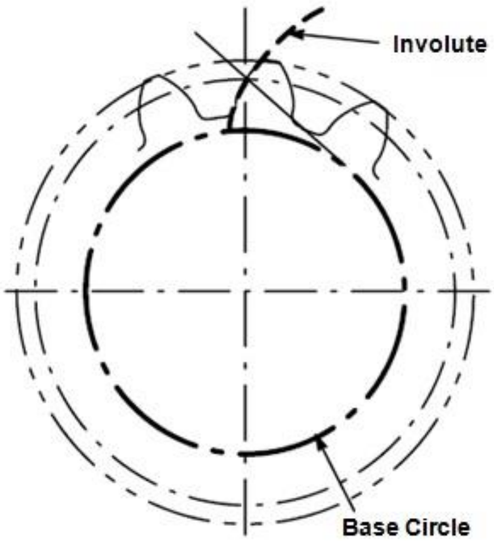
For 14½°: Change in Center Distance = 1.933 × Change in Backlash
For 20°: Change in Center Distance = 1.374 × Change in Backlash

From above, it is apparent that for a given change in center distance, 14½° gears will have a smaller change in backlash than 20° gears.

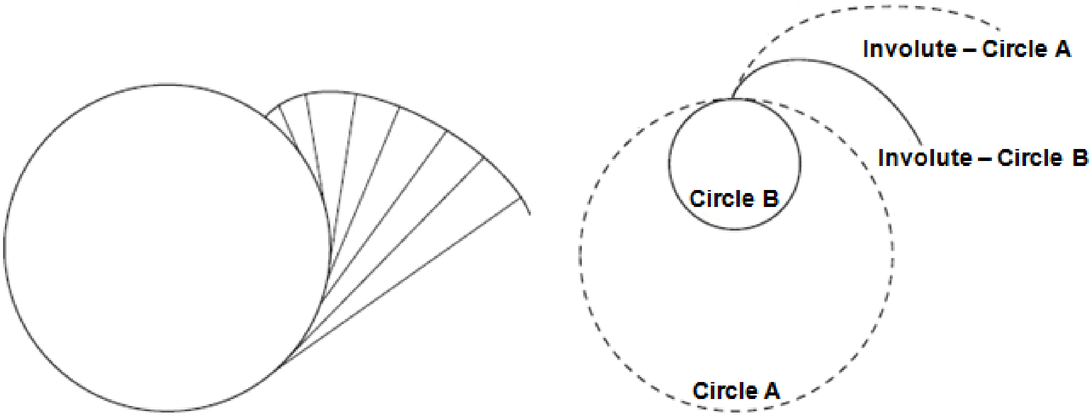
Involute Profile

Gear teeth could be manufactured with a wide variety of shapes and profiles. The involute profile is the most commonly used system for gearing.





An involute is a curve that is traced by a point (stylus) on a taut cord unwinding from a circle, which is called a base circle. The involute is a form of spiral, the curvature of which becomes straighter as it is drawn from a base circle and eventually would become a straight line if drawn far enough.



Involute Profile

As shown in above figure, an involute drawn from a larger base circle will be less curved (straighter) than one drawn from a smaller base circle.

Involute Profile on Gears and Rack

Hence as shown in above figure, the involute tooth profile of smaller gears is considerably curved, on larger gears is less curved (straighter), and is straight on a rack, which is essentially an infinitely large gear.

Properties of Involute Curve

The relative rate of motion between driving and driven gears having involute tooth curves is established by the diameters of their base circles.

If a gear tooth of involute curvature acts against the involute tooth of a mating gear while rotating at a uniform rate, the angular motion of the driven gear will also be uniform, even though the center to center distance is varied.

The point where the line of action intersects the line of centers establishes the radii of the pitch circles of these gears. Hence operating/true pitch circle diameters are affected by a change in the center distance.

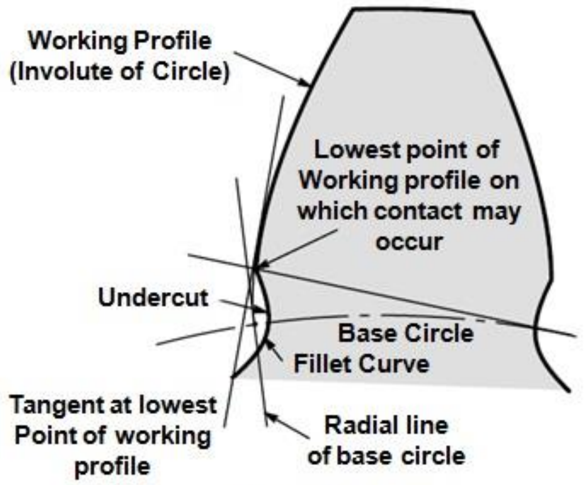
Since the base circle is related to the pressure angle and pitch diameter by the equation,

Base Diameter (D_b) = $D \cos\phi$

Where D (pitch diameter) and ϕ (pressure angle) are the standard values, exact values of operating pitch diameter (D') and operating pressure angle (ϕ') are given by,

Base Diameter (D_b) = $D' \cos\phi'$

Undercut



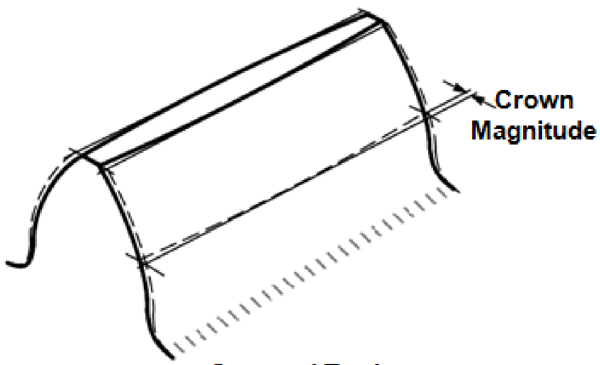
Undercut

Undercut is a condition in generated gear teeth when any part of the fillet curve lies inside of a line drawn tangent to the working profile at its point of juncture with the fillet.

When the number of teeth in a gear is small, the tip of the mating gear tooth may interfere with the lower portion of the tooth profile. To prevent this, the generating process removes material at this point. This results in loss of a portion of the involute adjacent to the tooth base, reducing tooth contact and tooth strength. Undercut may be deliberately introduced to facilitate finishing operations. With undercut the fillet curve intersects the working profile. Without undercut the fillet curve and the working profile have a common tangent.

On 20° pressure angle gears, undercutting occurs when number of teeth are less than 18. However, the gears with 17 teeth or under can be used if its strength or contact ratio does not pose any ill effect. Since the condition becomes more severe as tooth numbers decrease, it is recommended that the minimum number of teeth be 13 for 20° pressure angle. On 14½° pressure angle gears, undercutting occurs when number of teeth are less than 32.

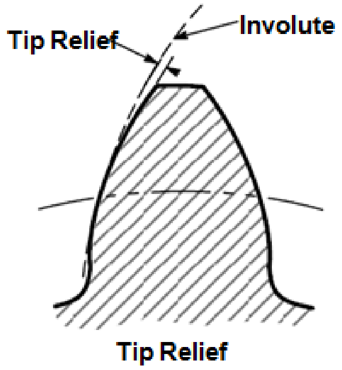
Crowned Teeth



Crowned Teeth

Crowned teeth have surfaces modified in the lengthwise direction to produce localized contact or to prevent contact at their ends. It ensures that the center of the flank carries its full share of the load even if the gears are slightly misaligned or distorted. Crowning can be applied to all types of teeth.

Tip Relief



It is an arbitrary modification of a tooth profile whereby a small amount of material is removed near the tip of the gear tooth. It is sometimes applied to lessen the chance of chipping, particularly in the case of hardened teeth.

Gear Ratio

The gear ratio or speed ratio is usually expressed as a speed reduction. In a pair of gears, it can be calculated by dividing the number of teeth in the large gear by the number of teeth in the small gear.

Gear and Pinion

Of two gears that run together, the one with the larger number of teeth is called the gear (wheel as per ISO 1122) and the one with the smaller number of teeth is called the pinion.

For more information on gear nomenclature and definitions of terms, please see ANSI/AGMA 1012-G05, Gear Nomenclature, Definitions of Terms with Symbols, published by the American Gear Manufacturers Association, 500 Montgomery Street, Suite 350, Alexandria, Virginia 22314, http://www.agma.org.

Tooth Proportions

There are several tooth form depth options for involute gearing in common usage, with the full depth design being the current standard because of its balance of strength and smoothness. Other involute options include modified (long and short) addendum teeth, and the obsolete stub tooth.

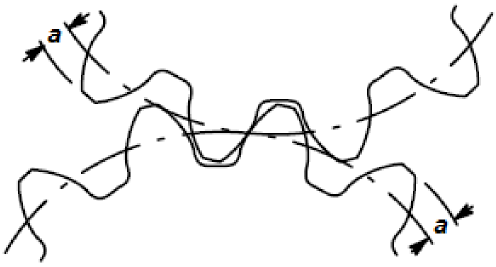
Full Depth Teeth

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

Stub Teeth

Stub teeth are those in which the working depth is less than 2.000 divided by the normal diametral pitch.

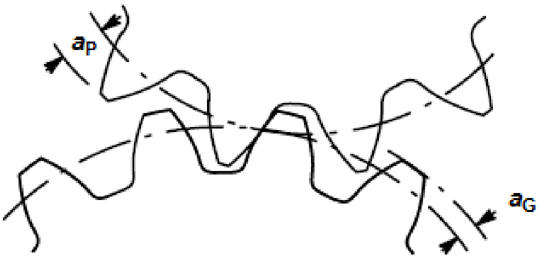
Equal Addendum Teeth



Equal Addendum Teeth

Equal addendum teeth are those in which two engaging gears have equal addendums.

Long and Short Addendum Teeth



Long and Short Addendum Teeth

Long and short addendum teeth are those in which the addendums of two engaging gears are unequal.

Tooth Form Modification - Stub Tooth

If tooth depth is shorter than full depth teeth, it is called a stub tooth. Stub teeth have more strength than full depth teeth but contact ratio is reduced.

The most widely used stub tooth, called American (AGMA) standard stub has an addendum = 0.8 / P_d and a dedendum = 1 / P_d (the most widely used stub tooth in metric system has an addendum = 0.8m and a dedendum = 1m).

The so-called "Fellows" stub tooth system, originated in 1906, is a combination of two diametral pitches. For example, 10/12 pitch; in which the first number (also called the numerator) is used to determine the number of teeth, pitch diameter and tooth thickness; and the second number (also called the denominator) is used to determine the addendum, dedendum, depth and clearance of the tooth.

Tooth Form Modification - Long and Short Addendum

Because the pinion is usually the smaller, driving member, pinion tooth strength is generally lower than that of the larger, driven gear, when standard tooth proportions are used. To provide increased strength, reduce undercutting and improve operating characteristics, the dedendum of a pinion tooth may be decreased and the addendum increased correspondingly. If the center distance remains the same, the addendum of the driven gear tooth must be decreased and the dedendum increased proportionally. Thus, tooth strengths are brought into balance and wear life of the gear set extended. This modification is called long and short addendum (as shown in above figure).

Addendum and Dedendum

Addendum and dedendum for full depth involute tooth having pressure angle of 20° in English system are as under.

Addendum (a) = 1 / P_d
Dedendum for coarse pitch gears (b) = 1.25 / P_d
Dedendum for fine pitch gears (b) = 1.20 / P_d + 0.002

A constant value, 0.002 in., is added to the dedendum of fine pitch gears, which allows space for the accumulation of foreign matter at the bottoms of spaces. This provision is particularly important in the case of very fine diametral pitches.

Above data are based on the information contained in AGMA standard 207.05 "20-Degree Involute Fine-Pitch System for Spur and Helical Gears" and AGMA standards 201.02 and 201.02A, "Tooth proportions for Coarse-Pitch Involute Spur Gears."

Diametral pitches range from 0.5 to 200. Coarse pitch gears are those with a diametral pitch of < 20. Fine pitch gears are those with a diametral pitch of ≥ 20.

Diametral pitches in general use are:

Coarse pitch: 2, 2¼, 2½, 3, 4, 6, 8, 10, 12, 16
Fine pitch: 20, 24, 32, 40, 48, 64, 80, 96, 120, 150, 200

Addendum and dedendum for full depth involute tooth having pressure angle of 20° in Metric system are as under.

Addendum (h_a) = m
Dedendum (h_f) = 1.25 m

Preferred modules in general use are: 1, 1.25, 1.5, 2, 2.5, 3, 4, 5, 6, 8, 10, 12, 16, 20, 25, 32, 40 and 50.

Finding Diametral Pitch

One can easily find the approximate diametral pitch of a gear without precision measuring tools or gauges. Measurements need not be exact because diametral pitch numbers are usually whole numbers. Therefore, if an approximate calculation results in a value close to a whole number, that whole number is the diametral pitch number of the gear. One of the methods to find the approximate diametral pitch is as under.

Count the number of teeth in the gear, add 2 to this number, and divide it by the outside diameter of the gear (diameter of the addendum circle). The approximate whole number is the diametral pitch of the gear.

Note:

The symbols used in this chapter are as per AGMA Symbols. The following table shows the equivalence of AGMA and ISO Symbols.

AGMA Symbols	ISO Symbols	Nomenclature / Description
a	h_a	Addendum
b	h_f	Dedendum
C	a	Center distance
D	d	Reference standard pitch diameter
h_t	h	Whole depth
N	z	Number of teeth
ϕ	α	Pressure angle

Types of Gears

Gears are machine elements that transmit motion by means of successively engaging teeth. There is a wide variety of types of gears in existence, each serving a range of functions. Information about various types of gears and terms related with them is given in this chapter.

Gear Types and Axial Arrangements

In order to understand types of gears, it is desirable to classify the important types in some way. One approach is by the relationship of the shaft axes on which the gears are mounted. As listed in the following table, gears may be classified in three categories:

- Parallel Axes Gears
- Intersecting Axes Gears
- Nonparallel and nonintersecting Axes Gears

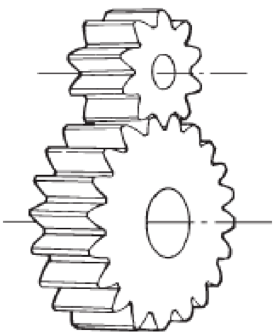
Types of Gears and Their Categories	
Gear Categories	**Types of Gears**
Parallel Axes Gears	Spur Gears
	Helical Gears
Intersecting Axes Gears	Straight Bevel Gears
	Spiral Bevel Gears
	Zerol Gears
Nonparallel and Nonintersecting Axes Gears	Hypoid Gears
	Worm Gears
	Crossed Helical Gears (Screw Gears)

Parallel Axes Gears

Spur and helical gears are the parallel axes gears.

Spur Gears

Spur gears are used to transmit motion between parallel shafts or between a shaft and a rack.

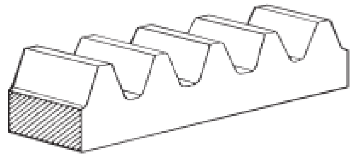


Spur Gears

Spur gear is a cylindrical shaped gear in which the teeth are parallel to the axis.

The advantages of spur gears are simplicity of design, economy of manufacture and absence of end thrust. They impose only radial load on the bearings. However they are relatively noisy when compared to other gears.

Spur Rack

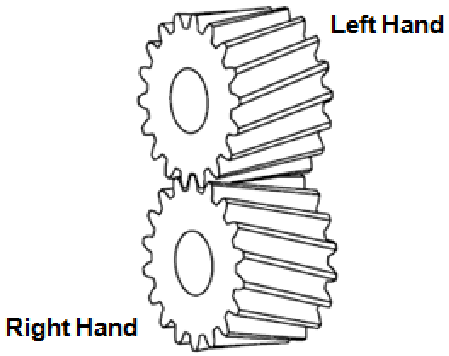


Spur Rack

Spur rack is a linear shaped gear which can mesh with a spur gear. The spur rack is a portion of a spur gear with an infinite radius.

Helical Gears

Helical gears are used to transmit motion between parallel or crossed shafts (screw gear) or between a shaft and a helical rack.

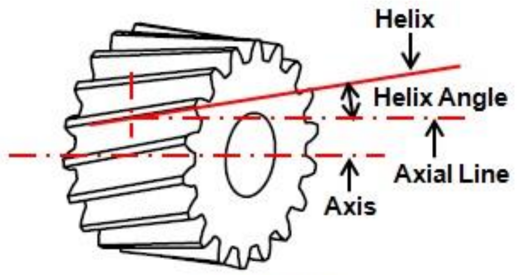


Helical Gears

Helical gear is a cylindrical shaped gear in which the teeth lie along a helix (at an angle to the axis of the shaft). Because of this angle, mating of the teeth occurs such that two or more teeth of each gear are always in contact. This condition permits smoother action than that of spur gears. However, single helical gears generate axial thrust, which causes slight loss of power and requires thrust bearings.

As shown in above figure, external helical gears on parallel axes have helices of opposite hands (a right hand pinion/gear meshes with a left hand gear). However if one of the members is an internal gear, the helices are of the same hand.

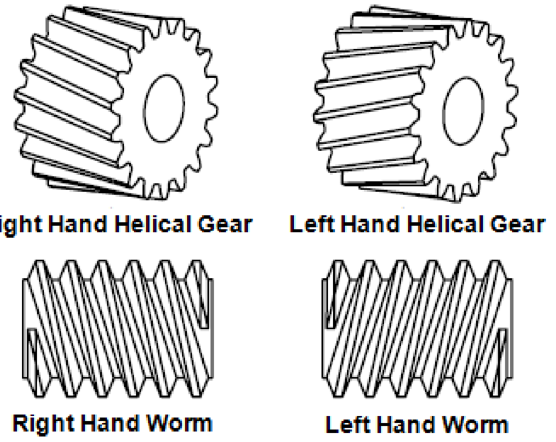
Helix Angle



Helix Angle

Helix angle is the angle between any helix and an axial line (line parallel to axis) on its cylinder.

Helical and Worm Hand

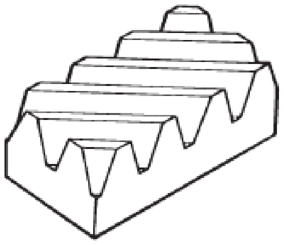


Helical and Worm Hand

A right hand helical gear or right hand worm is one in which the teeth twist clockwise as they recede from an observer looking along the axis.

A left hand helical gear or left hand worm is one in which the teeth twist counterclockwise as they recede from an observer looking along the axis.

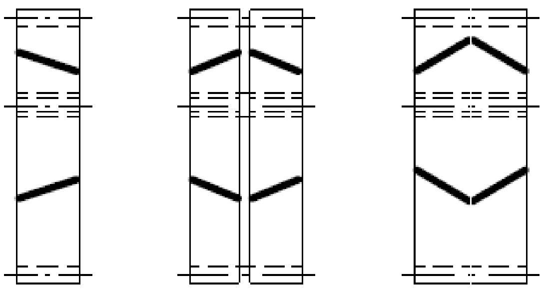
Helical Rack



Helical Rack

Helical rack is a linear shaped gear which meshes with a helical gear. It can be regarded as a portion of a helical gear with infinite radius.

Single and Double Helical Gears



Single and Double Helical Gears

Single helical gears have teeth of only one hand on each gear.

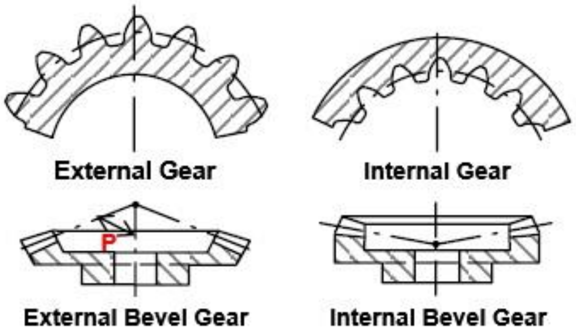
Double helical gears have teeth of both right hand and left hand on each gear. The teeth are separated by a gap between the helices. Where there is no gap, they are known as herringbone.

Herringbone gears have teeth of both right hand and left hand on each gear. However the teeth are continuous without a gap between the helices.

Because double helical gears have right hand and left hand helixes, they are usually not subject to end thrust.

The helix angle is usually about 20° for single helical gears and about 30-35° for double helical (herringbone) gears.

External and Internal Gears

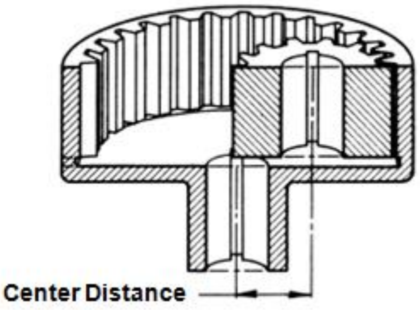


External and Internal Gears

An external gear is one with the teeth formed on the outer surface of a cylinder or cone.

An internal gear is one with the teeth formed on the inner surface of a cylinder or cone. For bevel gears, an internal gear is one with the pitch angle (P) exceeding 90°.

Internal gears are often used in planetary gear systems.

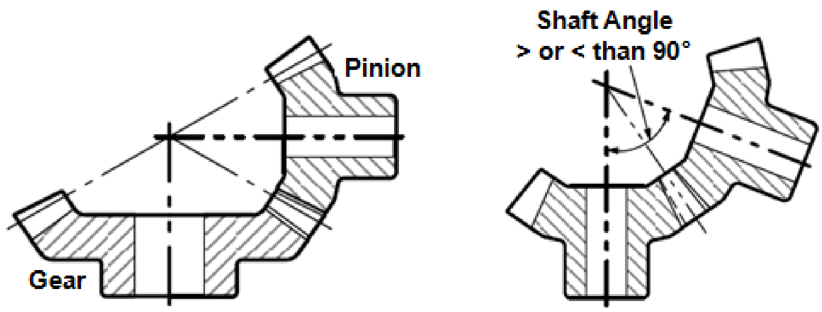


Spur Type Internal Gear

As shown in above figure, internal gears are sometimes used in compact designs because the center distance between the internal gear and its mating pinion is much smaller than that required for two external gears. An internal gear can be meshed only with an external pinion.

Intersecting Axes Gears

Bevel gears have conical pitch surfaces operating on intersecting or nonintersecting axes. Bevel gears that operate on nonintersecting axes are known as hypoid gears.

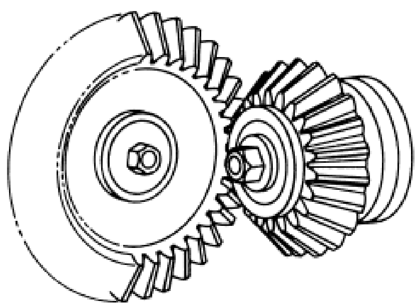


Bevel Gears and Angular Bevel Gears

In bevel gears, the shafts are at 90° to each other. Bevel gears in which the shaft axes are not at right angles are called angular bevel gears.

Miter gears are mating bevel gears with equal numbers of teeth and with axes at right angle.

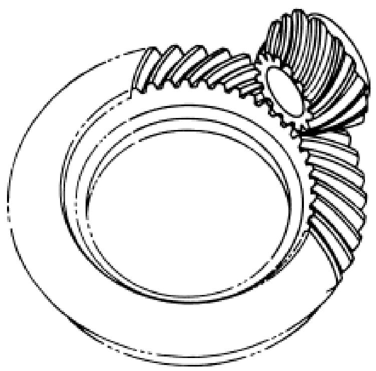
Straight Bevel Gears



Straight Bevel Gears

Straight bevel gears have straight teeth that, if extended inward, would intersect at the axis of the gear. Thus, the action between mating teeth resembles that of two cones rolling on each other. The teeth of straight bevel gears are tapered in both thickness and tooth height. The use of straight bevel gears is generally limited to drives that operate at low speeds and where noise is not important.

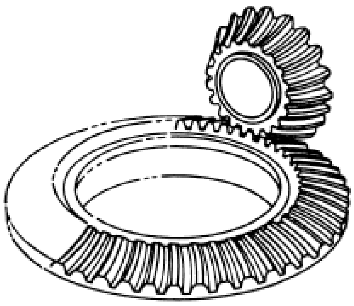
Spiral Bevel Gears



Spiral Bevel Gears

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

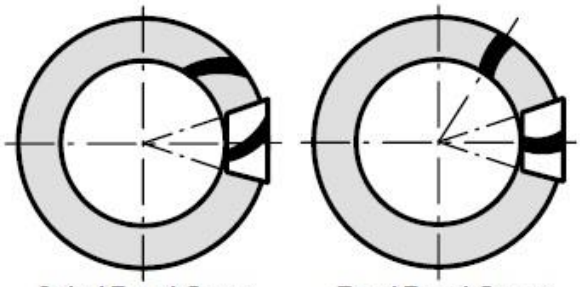
Zerol Bevel Gears



Zerol Bevel Gears

Zerol bevel gears are curved tooth bevel gears with zero spiral angle. They differ from spiral bevel gears in that the teeth are not oblique. They are used in the same way as spiral bevel gears, and they have somewhat greater tooth strength than straight bevel gears.

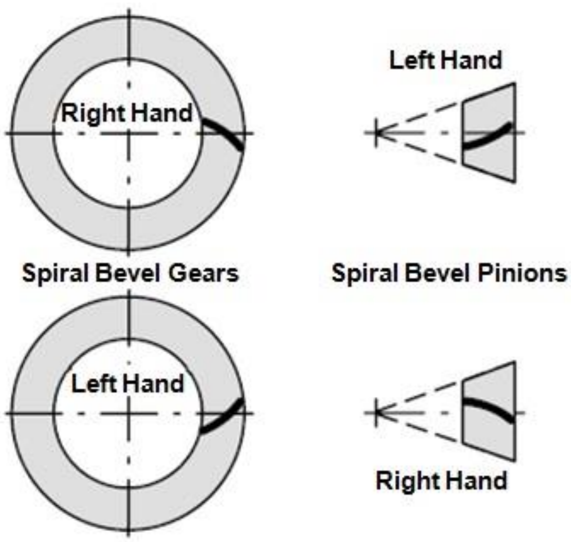
Following figure shows difference between teeth of spiral bevel gears and zerol bevel gears.



Spiral Bevel Gears Zerol Bevel Gears

Teeth of Spiral Bevel Gears and Zerol Bevel Gears

Spiral Bevel Hand



Spiral Bevel Hand

A right hand spiral bevel gear is one in which the outer half of a tooth is inclined in the clockwise direction from the axial plane through the midpoint of the tooth as viewed by an observer looking at the face of the gear.

A left hand spiral bevel gear is one in which the outer half of a tooth is inclined in the counterclockwise direction from the axial plane through the midpoint of the tooth as viewed by an observer looking at the face of the gear.

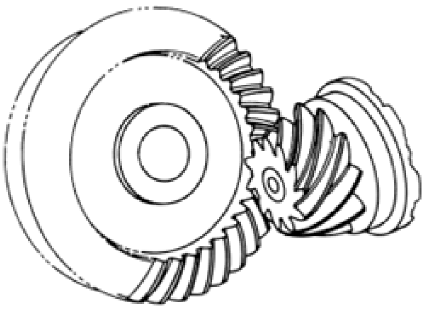
A spiral bevel gear and pinion are always of opposite hand, including the case when the gear is internal.

The designations right hand and left hand are applied similarly to spiral bevel gears, zerol bevel gears and hypoid gears.

Nonparallel and Nonintersecting Axes Gears

Hypoid gears, worm gears and crossed helical gears (screw gears) operate on nonparallel and nonintersecting shafts.

Hypoid Gears



Hypoid Gears

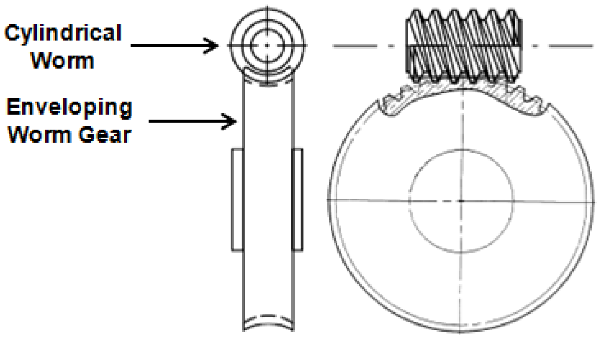
Hypoid gears are similar to spiral bevel gears in general appearance. The important difference is that the pinion axis of the hypoid pair of gears is offset somewhat from the gear axis.

Hypoid gear is a deviation from a bevel gear that originated as a special development for the automobile industry. This permitted the drive to the rear axle to be nonintersecting, and thus allowed the auto body to be lowered.

Worm Gear Sets

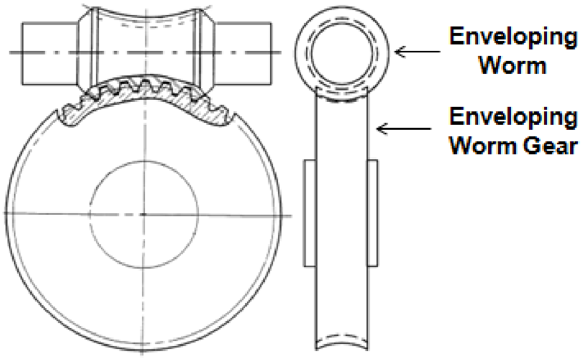
Worm Gear Set

Worm gear sets are usually right angle drives consisting of a worm and a worm gear (or worm wheel). The worm resembles a screw thread and the mating worm gear a helical gear.



Single Enveloping Worm Gear Set

A single enveloping worm gear set has a cylindrical worm but the gear is throated (the gear blank has a smaller diameter in the center than at the ends of the cylinder) so that it tends to envelope/wrap around the worm.

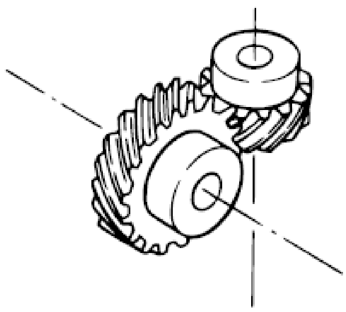


Double Enveloping Worm Gear Set

In the double enveloping worm gear set, both members are throated and both members envelope/wrap around each other.

The outstanding feature of a worm gear set is that it offers a very large gear ratio in a single mesh. However, transmission efficiency is very poor due to a great amount of sliding as the worm tooth engages with its mating worm gear tooth and forces rotation by pushing and sliding. With proper choices of materials and lubrication, wear can be contained and noise is reduced. A worm wheel is not feasible to drive a worm (self-locking feature) except for special occasions.

Crossed Helical Gears (Screw Gears)



Crossed Helical Gears

Crossed helical gears are conventional helical gears installation on crossed axes.

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

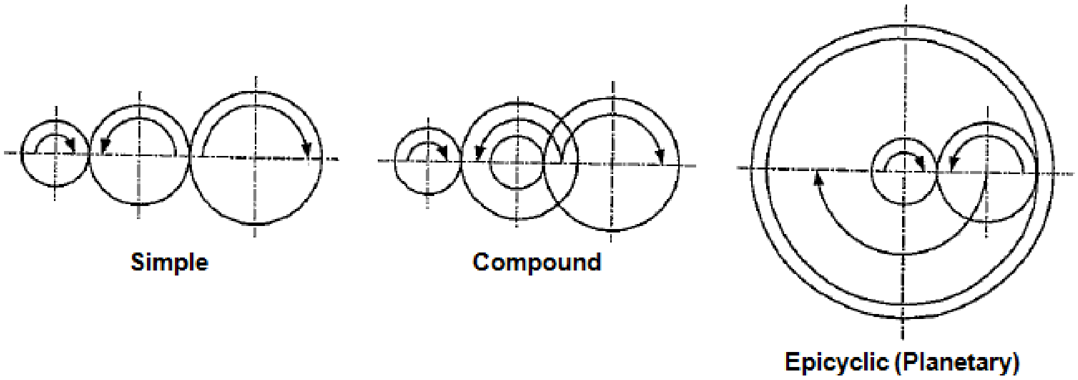
Gear Trains

The objective of gears is to provide a desired motion, either rotary (rotation) or linear. This is accomplished through either a simple gear pair or a more involved and complex system of several gear meshes called gear train. Also, related to this is the desired speed, direction of rotation and the shaft arrangement. Information about various types of gear trains is given in this chapter.

Types of Gear Trains

Two gears in mesh are called a gear pair. In a pair of gears, the gear transmitting input torque and power is called the driver whereas the gear transmitting output torque and power is known as the driven (or follower). Normally in mechanical power transmission, the driver is the pinion so that wheel rotates at a slower speed than the pinion, that is, there is a reduction in speed through the gear pair.

But very often the input and the output shaft are connected by more number of gears. Such combination of gears to connect the input to the output shaft is called a gear train.



Types of Gear Trains

Several types of gear train are used for mechanical power transmission. They are normally classified into three types as under.

- Simple gear train
- Compound gear train
- Epicyclic (commonly called planetary) gear train

In the simple and compound gear trains, the axes of the shafts over which the gears are mounted are fixed relative to each other. That means, the gear axes of all the gears are fixed in space. But in case of epicyclic gear trains, the axes of the shafts on which the gears are mounted may move relative to a fixed axis.

Speed Ratio (Velocity Ratio)

The speed ratio (or velocity ratio) of gear train is the ratio of the speed of the driver to the speed of the driven and ratio of speeds of any pair of gears in mesh is the inverse ratio of their number of teeth. The speed ratio of any pair of gears is given by:

$$\text{Speed Ratio} = \frac{N_1}{N_2} = \frac{T_2}{T_1}$$

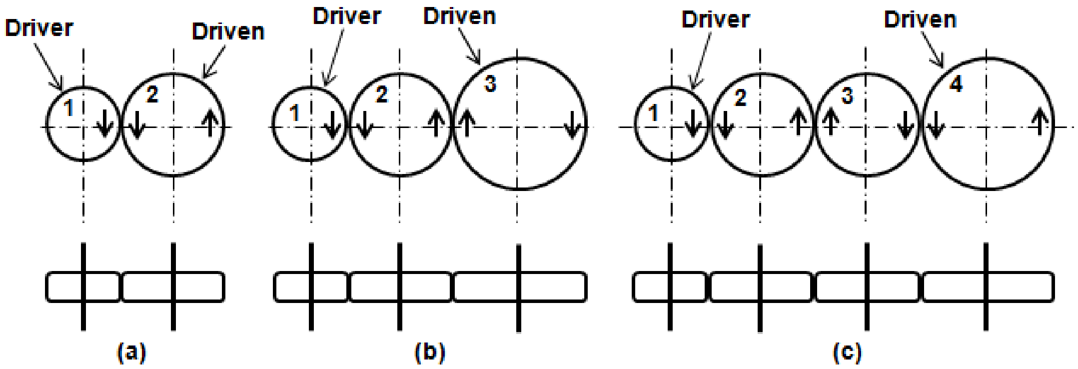
Where,

N_1 = Speed of gear 1 (or driver) in rpm.
N_2 = Speed of gear 2 (or driven or follower) in rpm.
T_1 = Number of teeth on gear 1
T_2 = Number of teeth on gear 2

Note:

As per AGMA, symbols for revolutions per unit of time and number of gear teeth are *n* and *N* respectively. However, as per common practice *N* and *T* are used for speed and number of teeth respectively in this article for easy of understanding.

Simple Gear Train



Simple Gear Train

When there is only one gear on each shaft, as shown in above figure, the assembly is known as simple gear train. The gears are represented by their pitch circles.

When the distance between two shafts is small, gear 1 and gear 2 are made to mesh with each other to transmit motion from one shaft to the other, as shown at (a) in above figure. Since gear 1 drives gear 2, gear 1 is called the driver and gear 2 is called the driven or follower. It may be noted that the motion of the driven gear is opposite to the motion of driving gear. If,

N_1 = Speed of gear 1 (or driver) in rpm.
N_2 = Speed of gear 2 (or driven or follower) in rpm.
T_1 = Number of teeth on gear 1
T_2 = Number of teeth on gear 2

The speed ratio is given by:

$$\text{Speed Ratio} = \frac{N_1}{N_2} = \frac{T_2}{T_1}$$

If the distance between two gears is large, the motion from one gear to another may be transmitted by either of the following two methods:

1. By providing larger sized gear.
2. By providing one or more intermediate gears.

A little consideration will show that the former method (i.e. providing large sized gears) is very inconvenient and uneconomical method whereas the latter method (i.e. providing one or more intermediate gear) is very convenient and economical.

Now consider a simple train of gears with one intermediate gear as shown at (b) in above figure.

Let,

N_1 = Speed of driver in rpm.
N_2 = Speed of intermediate gear in rpm.
N_3 = Speed of driven or follower in rpm.
T_1 = Number of teeth on driver
T_2 = Number of teeth on intermediate gear and
T_3 = Number of teeth on driven or follower

Since the driving gear 1 is in mesh with the intermediate gear 2, speed ratio for these two gears is:

$$\frac{N_1}{N_2} = \frac{T_2}{T_1} \quad \text{............ (i)}$$

Similarly, as the intermediate gear 2 is in mesh with the driven gear 3, speed ratio for these two gears is:

$$\frac{N_2}{N_3} = \frac{T_3}{T_2} \quad \text{............ (Ii)}$$

The speed ratio of the gear train shown at (b) in above figure is obtained by multiplying the equations (i) and (ii).

$$\frac{N_1}{N_2} \times \frac{N_2}{N_3} = \frac{T_2}{T_1} \times \frac{T_3}{T_2} \quad \text{or} \quad \frac{N_1}{N_3} = \frac{T_3}{T_1}$$

$$\text{i. e. Speed Ratio} = \frac{\text{Speed of driver}}{\text{Speed of driven}} = \frac{\text{No. of teeth on driven}}{\text{No. of teeth on driver}}$$

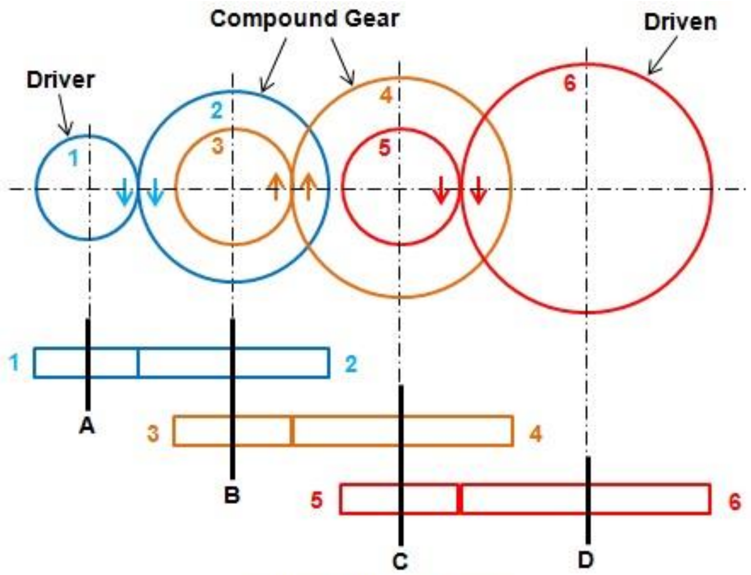
Similarly, it can be proved that the above equation holds good even if there are any number of intermediate gears. From above, we see that the speed ratio in a simple train of gears is independent of the size and number of intermediate gears. These intermediate gears are called idle gears (idlers), as they do not affect the speed ratio of the system. The idle gears are used for the following two purposes:

- To connect gears where a large centre distance is required.
- To obtain the desired direction of motion of the driven gear (i.e. clockwise or anticlockwise).

It may be noted that when the number of intermediate gears are odd (1, 3, 5, etc.) the motion of both the gears (i.e. driver and driven or follower) is similar as shown at (b) in above figure.

But if the number of intermediate gears are even (2, 4, 6, etc.), the motion of the driven or follower will be in the opposite direction of the driver as shown at (c) in above figure.

Compound Gear Train



Compound Gear Train

When there is more than one gear on a shaft, as shown in above figure, it is called a compound train of gear.

We have seen in previous section that the idle gears in a simple train of gears do not affect the speed ratio of the system. But these gears are useful in bridging over the space between the driver and the driven. But whenever the distance between the driver and the driven has to be bridged over by intermediate gears and at the same time a large speed ratio is required, then the advantage of intermediate gears is intensified by providing compound gears on intermediate shafts. In this case, each intermediate shaft has two gears rigidly fixed to it so that they have the same speed. As shown in above figure one of these two gears meshes with the driver and the other with the driven attached to the next shaft.

In a compound train of gears shown in above figure, gear 1 is the driving gear and is mounted on shaft A. Gears 2 and 3 are compound gears and are mounted on shaft B. The gears 4 and 5 are also compound gears and are mounted on shaft C. Gear 6 is the driven gear and is mounted on shaft D.

Let,

N_1 = Speed of driving gear 1, in rpm
T_1 = Number of teeth on driving gear 1,
N_2 ,N_3 ..., N_6 = Speed of respective gears in rpm., and
T_2 ,T_3..., T_6 = Number of teeth on respective gears.

Since gear 1 is in mesh with gear 2, its speed ratio is: $N_1 / N_2 = T_2 / T_1$ …….. (i)

Similarly, for gears 3 and 4, speed ratio is: $N_3 / N_4 = T_4 / T_3$ …………………. (ii)

And for gears 5 and 6, speed ratio is: $N_5 / N_6 = T_6 / T_5$ ……………………..... (iii)

The speed ratio of compound gear train is obtained by multiplying the equations (i), (ii) and (iii),

$$\frac{N_1}{N_2} \times \frac{N_3}{N_4} \times \frac{N_5}{N_6} = \frac{T_2}{T_1} \times \frac{T_4}{T_3} \times \frac{T_6}{T_5}$$

Since gears 2 and 3 are mounted on one shaft B, $N_2 = N_3$. Similarly as gears 4 and 5 are mounted on one shaft C, $N_4 = N_5$. Therefore,

$$\frac{N_1}{N_6} = \frac{T_2 \times T_4 \times T_6}{T_1 \times T_3 \times T_5}$$

i. e.

$$\text{Speed Ratio} = \frac{\text{Speed of the first driver}}{\text{Speed of the last driven}} = \frac{\text{Product of No. of teeth on the drivens}}{\text{Product of No. of teeth on the drivers}}$$

Thus in case of compound gear train, the overall speed ratio is the multiplication of the speed ratio of each gear pair.

While calculating the speed ration of a gear train by multiplying the equations (i), (ii) and (iii), a negative sign may be given to speed ratio in case of external gear pair and a positive sign may be given to internal gear pair to find direction of rotation of the driven gear.

For example, the direction of the driven gear for the compound gear train shown in above figure will be opposite to the driven gear as the product of the equations (i), (ii) and (iii) is negative.

$$\frac{N_1}{N_2} \times \frac{N_3}{N_4} \times \frac{N_5}{N_6} = \left[-\frac{T_2}{T_1}\right] \times \left[-\frac{T_4}{T_3}\right] \times \left[-\frac{T_6}{T_5}\right]$$

Since gears 2 and 3 are mounted on one shaft B, $N_2 = N_3$. Similarly as gears 4 and 5 are mounted on one shaft C, $N_4 = N_5$. Therefore,

$$\frac{N_1}{N_6} = -\left[\frac{T_2 \times T_4 \times T_6}{T_1 \times T_3 \times T_5}\right]$$

The advantage of a compound train over a simple gear train is that a much larger speed reduction from the first shaft to the last shaft can be obtained with small gears. If a simple gear train is used to get a large speed reduction, the last gear has to be very large. Usually for a speed reduction in excess of 5 to 1, a simple train is not used and a compound train or worm gearing is employed.

Limiting Speed Ratio

For a worm and wheel gear pair, the smallest speed ratio obtainable in practice is about 5. For all other gear pairs, the smallest ratio is 1. There is no theoretical upper limit to the speed ratio, however high speed ratios are undesirable because of the large number of teeth that need to be cut in the wheel. This makes accurate machining difficult (if not impossible) and requires large centre distances which are also not usually desirable.

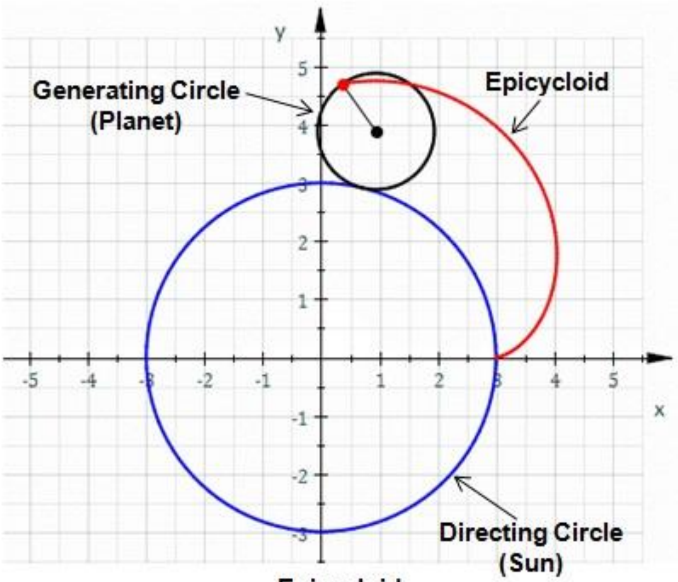
For example, consider a speed ratio of 125 obtained with single pair of spur gears. With a pinion of 17 teeth and PCD 17 mm, the wheel would require 2125 teeth and a PCD of 2125 mm! The total number of teeth to be cut is 2142!

The same velocity ratio can be obtained with a compound gear train with 3 pairs of gears in mesh, each with a velocity ratio of 5. Then the total number of teeth that need to be cut is 306 (7 times less) and the maximum wheel PCD is only 85 mm.

Rule-of-thumb limits for a gear pair are given in the following table.

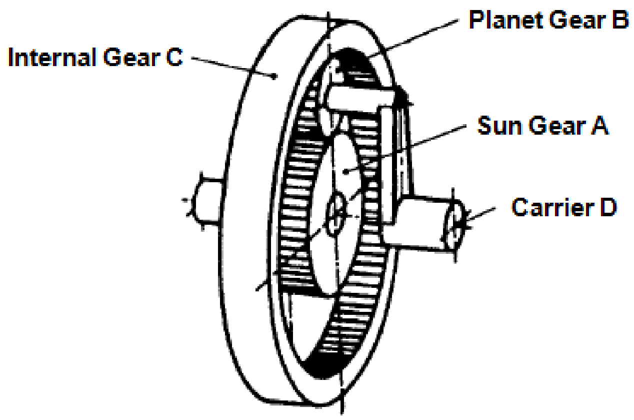
Rule-of-Thumb Speed Ratio Limits for a Gear Pair		
Type of Gear Pair	**Lower Limit of Speed Ratio**	**Upper Limit of Speed Ratio**
Worm and Wheel	5	60
All Others	1	5

Epicyclic (Planetary) Gear Train



Epicycloid

As shown in above figure, when a generating circle (for example, planet gear) rolls on the outside of another circle, called a directing circle (for example, sun gear), each point on the generating circle describes an epicycloid.



Epicyclic Gear Train

Above figure shows construction of an epicyclic (or planetary) gear train. The three basic components of the epicyclic gear train are:

- Sun Gear A: The central gear.
- Carrier D: It holds one or more peripheral Planet Gears B, all of the same size, meshed with the sun gear.
- Internal Gear C: An outer ring with inward facing teeth that meshes with the planet gear or gears.

In many epicyclic gearing trains, one of these three basic components is held stationary; one of the two remaining components becomes an input, providing power to the train, while the last component becomes an output, receiving power from the train.

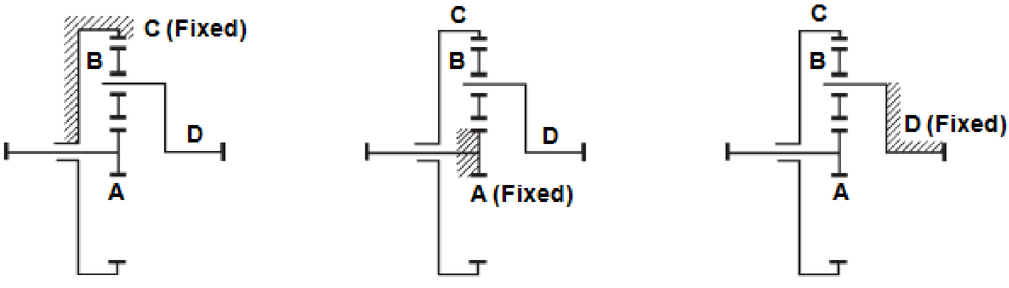
Epicyclic gear trains also referred to as planetary gear trains are those in which one or more gears (planet gears) orbit about the central axis (axis of sun gear) of the train. Thus, they differ from simple or compound gear train by having moving axes. The term "epicyclic" comes from the fact that points on gears with moving axes of rotation describe epicyclic paths.

Epicyclic Gearbox

Above figure shows typical photograph of an epicyclic gearbox.

Generally, more the number of planet gears, the greater is the torque capacity of the train. Epicyclic gear trains are useful for transmitting high/large speed ratios with gears of moderate size in a comparatively lesser space. For large speed ratios, it is recommended to use epicyclic gear trains as compared to worm gear sets because transmission efficiency of epicyclic gear trains is higher than transmission efficiency of worm gear sets.

In the case of simple and compound gear trains it is not difficult to visualize the motion of the gears and the determination of the speed ratio is relatively easy. However, in the case of epicyclic gear trains it is often difficult to visualize the motion of the gears.



(a) Planetary Type Gear Train **(b) Solar Type Gear Train** **(c) Star Type Gear Train**

Types of Epicyclic Gear Trains

Above figure shows three types of epicyclic gear trains depending upon which member is held stationary.

In an epicyclic gear train, the ratio of input rotation to output rotation called the speed ratio and the direction of rotation are dependent upon the number of teeth in each gear and upon which component is held stationary.

Planetary Type Gear Train

In this type, the internal gear is fixed. The input is the sun gear A and the output is carrier D. The direction of rotation of input and output axes are the same. The speed ratio is given by,

$$\text{Speed Ratio} = \frac{N_A}{N_D} = \frac{T_C}{T_A} + 1$$

Solar Type Gear Train

In this type, the sun gear A is fixed. The internal gear C is the input, and carrier D's axis is the output. The directions of rotation of input and output axes are the same. The speed ratio is given by,

$$\text{Speed Ratio} = \frac{N_C}{N_D} = \frac{T_A}{T_C} + 1$$

Star Type Gear Train

This is the type in which Carrier D is fixed. The planet gears B rotate only on fixed axes. In a strict definition, this train loses the features of an epicyclic train and it becomes simple gear train. The sun gear A is an input axis and the internal gear C is the output. The planet gears are merely idlers. Input and output axes have opposite rotations. The speed ratio is given by,

$$\text{Speed Ratio} = \frac{N_A}{N_C} = \frac{T_C}{T_A}$$

In above equations for speed ratios,

N_A, N_C and N_D are speeds of sun gear A, internal gear C and carrier D respectively and
T_A and T_C are the number of teeth of sun gear A and internal gear C respectively.

Note:

Some places speed ratio is defined as the ratio of driven gear's speed to driving gear's speed.

Selection of Enclosed Gear Drives

Enclosed gear drives (gearboxes) deliver power to industrial equipment such as bulk material handling conveyors, cooling towers, mixers, pumps, etc. The reliability that translates into higher uptime and long life begins with selecting and specifying the proper gear drives for these critical applications. Many variables such as service factor (mechanical rating), thermal capacity, drive ratio, overhang, thrust loads, etc. must be considered when selecting an enclosed gear drive. In view of this, introduction and detail information on selection of enclosed gear drives as per ANSI/AGMA 6010-F97, “Standard for Spur, Helical, Herringbone and Bevel Enclosed Drives”, ISO 9085 and IS 7403 is given in this chapter.

Introduction - Selection of Enclosed Gear Drives

Generally, the purpose of an enclosed gear drive is to reduce the input speed coming from the prime mover, usually an AC or DC motor, to a slower speed output through a gear reduction.

The term “enclosed gear drive” comes from the fact that the gears are contained in some type of an enclosure with all the necessary lubricant. The enclosure protects the machine operator from injury. Gearbox is the commonly called name of an enclosed gear drive.

When selecting an enclosed gear drive, a user normally knows at least three things - the input speed, the output speed or required gear ratio and the motor power. In addition, one may know the driven machine’s absorbed power, from experience or measurement on similar systems.

When referring to a catalogue of an enclosed gear drives, the user will find two or three different ratings - a mechanical rating, a thermal power rating and an allowable overhung load on the input and output shafts. The mechanical ratings should be modified to take care of the type of duty the gear drive will experience and the thermal power ratings should be modified for the ambient conditions the gear drive will experience. Then the highest required rating (highest of mechanical rating, thermal power rating and an allowable overhung load) should be used to select the gear drive.

Mechanical Rating

Mechanical rating is the maximum power or torque that an enclosed gear drive can transmit based on the strength and durability of its components.

The mechanical ratings in the catalogue are based on a service factor of 1.00 which typically applies to an enclosed gear drive driven by a uniform power source and driving a machine with a uniform load requirement for up to 10 hours of operation per day. In practice, this is seldom the case, so the required power for the gear drive must be increased to allow for the type of prime mover, the loading characteristics and the hours of use. This is accomplished by multiplying the prime mover power by the appropriate service factor, which will be found tabulated in the catalogue.

A larger size gear drive should be selected when peak running loads are substantially greater than normal operating loads.

As catalogue ratings generally allow 100% overload at starting, braking or momentarily during operation, the selected gear drive must have a catalogue rating at least equal to half the maximum overload.

Thermal Rating

Thermal rating is the maximum power or torque that an enclosed gear drive can transmit continuously, based on its ability to dissipate heat generated by friction.

Again, the thermal ratings in the catalogue usually have to be modified for actual ambient conditions. The tabulated ratings usually apply to a gear drive at sea level in a large enclosed space at an ambient temperature of 20 or 24° C. Factors for different altitudes, locations and temperatures are given in the catalogue and the thermal ratings in the catalogue is multiplied by these factors to find out the required thermal rating for the gear drive pertaining to the actual conditions.

Overhung Load and Axial Thrust Capacities

The input or the output shaft of a speed reducer can be subject to an overhung load; that is, to a force applied at right angles to the shaft, beyond its outermost bearing. Such a force is a shaft bending load resulting from a gear, pulley, sprocket or other external drive member. Besides the tendency to bend the shaft, the overhung load (that is, the radial force on the shaft) is reacted to by the shaft in its bearings. Therefore, the overhung load creates loads that the bearings must be able to support without damage.

Axial thrust or thrust load is the force imposed on a shaft parallel to the shaft axis. It is often encountered on shafts driving mixers, fans, blowers and similar machines. When a thrust load acts on a speed reducer, you must be sure that the thrust load rating of the reducer is high enough that its shafts and bearings can absorb the load.

Allowable overhung loads (radial loads) and axial loads are tabulated in the catalogue.

Design Considerations

For most applications, it is recommended to select gear drive for running torque rather than starting torque. The AC motor will normally produce a 200 percent starting torque. A gear drive is built to take at least 200% momentary overload to overcome normal starting inertia.

Selecting/sizing a gear drive based on the motor rating may be simpler and results in a gear drive that works, but it will result in the purchase of a larger gear drive than is needed. Sizing to the load will ensure a gear drive that fits the application.

Where total gear ratios/reductions are small, 50 to 1 or less, power (hp/kW) figures are commonly used. Higher ratios require that torque figures be used to select drive components because with large ratios, a small motor can produce extremely high torque at the final low speed.

A 20% safety factor in selection can double the life of a gear drive - better economy in the long run. This rule of thumb will help compensate for unexpected shock and vibration, and add substantially to wear life.

Gear drives used in variable speed applications require special consideration to insure adequate lubrication under all operating conditions. Excess speed can cause overheating, and low speeds can cause inadequate lubrication. These applications should be referred to the manufacturer.

Selection of Enclosed Gear Drives as per ANSI/AGMA 6010-F97

ANSI/AGMA 6010-F97 includes design, rating, lubrication, testing and selection information for spur, helical, herringbone and bevel gears when using enclosed speed reducers or increasers. Units covered include those with a pitch line velocity below 7000 feet per minute or rotational speeds no greater than 4500 rpm.

Application Limitations of ANSI/AGMA 6010

In this standard the unit rating is defined as the mechanical capacity of the gear unit components determined with a unity service factor.

Units rated as per this standard can accommodate the following peak load conditions:

- Each peak shall not exceed 200 percent of the unit rating (service factor = 1.0).
- A limited number of stress cycles, typically less than 10^4.

For applications exceeding above conditions an appropriate service factor should be selected.

Some applications may require selecting a gear drive with increased mechanical rating in order to accommodate adverse effects of environmental conditions, thermal capacity of the unit and external loading (overhung load, thrust load, etc.).

If units are to be operated below minus 20°F, care must be given to select materials which have adequate impact properties at the operating temperature.

The system of connected rotating parts must be free from critical speeds, torsional or other types of vibration, within the specified operating speed range.

Unit Rating

Historically, many terms (like service rating, nameplate rating, equivalent rating, catalog rating, mechanical rating, brake rating, unity rating, transmitted horsepower, calculated horsepower, allowable horsepower, application horsepower, etc.) have been used to denote conditions of operations - both calculated and actual. These terms have resulted in confusion as to the actual capability of the enclosed drive. In view of this, for purposes of this standard, where component capacities are being determined, the calculations are specifically related to the unit rating as defined below.

The unit rating is the overall mechanical power rating of all static and rotating elements within the enclosed drive. The minimum rated component (weakest link, whether determined by gear teeth, shafts, bolting, housing, etc.) of the enclosed drive determines the unit rating.

The unit rating implies that all items within the gear drive have been designed to meet or exceed the unit rating.

Unit ratings may also include allowable overhung load values which are usually designated to act at a distance of one shaft diameter from the face of the housing or enclosure component.

The required unit rating of an enclosed drive is a function of the application and assessment of variable factors that affect the overall rating. These factors include environmental conditions, severity of service and life.

Hence, the application of the enclosed drive requires that its capacity as defined by its unit rating; i.e., its minimum rated component power, P_{mc}, be related to the actual service conditions.

$$P_A \leq \frac{P_{mc}}{K_{sf}}$$

where,

P_A = the application power of enclosed drive, hp
P_{mc} = the minimum component power rating (the unit rating), hp
K_{sf} = the service factor

Service Factors

Before an enclosed gear drive can be selected for any application, an equivalent unit power rating (service factor = 1.0) must be determined. This is done by multiplying the specified power by the service factor. Since the service factor represents the normal relationship between the gear unit rating and the required application power, it is suggested that the service factor be applied to the nameplate rating of the prime mover or driven machine rating, as applicable.

It is necessary that the gear drive selected have a rated unit capacity equal to or in excess of this "equivalent unit power rating".

Service factors may be selected from Table A.2, annex A of the standard or may be determined by an analytical method.

The Table A.2 of the standard for the service factors has been developed from the experience of manufacturers and users of gear drives for use in common applications.

For ready reference and as an illustration, service factors for some common applications are shown in the following table (as per Table A.2 of the standard).

Service Factors for Enclosed Gear Drives
Driven by Motors (Hydraulic or Electric) or Turbines (Steam or Gas)
[As per Table A.2 of ANSI/AGMA 6010-F97}

Application	**Load Duration**		
	Up to 3 hours per day	3 to 10 hour per day	Over 10 hours per day
Agitators (mixers)			
• Pure liquids	1.00	1.00	1.25
• Liquids and solids	1.00	1.25	1.50
• Liquids - variable density	1.00	1.25	1.50
Blowers			
• Centrifugal	1.00	1.00	1.25
• Lobe	1.00	1.25	1.50
• Vane	1.00	1.25	1.50
Clarifiers	1.00	1.00	1.25
Compressors			
• Centrifugal	1.00	1.00	1.25
• Lobe	1.00	1.25	1.50
• Reciprocating, multi-cylinder	1.50	1.50	1.75
• Reciprocating, single-cylinder	1.75	1.75	2.00
Crusher - Stone or ore	1.75	1.75	2.00

Fans			
• Centrifugal	1.00	1.00	1.25
• Cooling towers	2.00	2.00	2.00
• Forced draft	1.25	1.25	1.25
• Induced draft	1.50	1.50	1.50
• Industrial & mine	1.50	1.50	1.50
Feeders			
• Apron	1.00	1.25	1.50
• Belt	1.00	1.25	1.50
• Disc	1.00	1.00	1.25
• Reciprocating	1.50	1.75	2.00
• Screw	1.00	1.25	1.50
Hammer mills	1.75	1.75	2.0
Pumps			
• Centrifugal	1.00	1.00	1.25
• Proportioning	1.25	1.25	1.50
• Reciprocating - Single acting, 3 or more cylinders	1.25	1.25	1.50
• Reciprocating - Double acting, 2 or more cylinders	1.25	1.25	1.50
• Rotary - Gear type	1.00	1.00	1.25
• Rotary - Lobe type	1.00	1.00	1.25
• Rotary - Vane type	1.00	1.00	1.25
Screens - Traveling water intake	1.00	1.00	1.25

For information on other applications (cranes, lumber industry, metal mills, etc.), please refer Table A.2 of the standard.

Following points should be considered when selecting service factors.

Applications such as high torque motors, extreme repetitive shock, or where high energy loads must be absorbed, as when stalling, require special consideration. In view of this, Table A2 should be used with caution. Much higher values have occurred in some applications and values as high as ten have been used. On some applications up to six times nominal torque can occur, such as: Turbine/Generator drives, Heavy Plate and Billet rolling mills.

Electric motors that have electric power interrupted and then re-applied before induced magnetic fields have dissipated can produce very high torques.

When a gear drive is equipped with a “working” brake that is used to decelerate the motion of the system, select the drive based on the brake rating or the transmitted power, whichever is greater. If the brake is used for holding only (non-working), and is applied after the motion of the system has come to rest, the brake rating should be less than 200 percent of the base unit rating. If the brake rating is greater than 200 percent of the unit rating, or the brake is located on the output shaft of the gear drive, special analysis is required.

Applications requiring a high degree of reliability/dependability or unusually long life should be given careful consideration before assigning a service factor.

Service factors shown in Table A.2 are for gear drives driven by motors (electric or hydraulic) and turbines (steam or gas).

Some different types of prime movers are electric or hydraulic motors, steam or gas turbines and single or multiple cylinder internal combustion engines. Each of these prime movers is designed to produce some nominal power, but each will produce this power with some variation over time. The variation of power output with time may be lower or higher depending on the prime mover. As any variation over nominal power is an overload, it must be considered.

In view of above, when the driver is a single cylinder or multi-cylinder engine, the service factors from Table A.2 must be converted to the values from the following table (Table A.1 of the standard) for the appropriate type of prime mover.

Conversion Table for Single or Multi-Cylinder Engines to find Equivalent Single or Multi-Cylinder Service Factor (Table A.1 of ANSI/AGMA 6010-F97)		
Steam and Gas Turbines, Hydraulic or Electric Motor	Single Cylinder Engines	Multi-Cylinder Engines
1.00	1.50	1.25
1.25	1.75	1.50
1.50	2.00	1.75
1.75	2.25	2.00
2.00	2.50	2.25
2.25	2.75	2.50
2.50	3.00	2.75
2.75	3.25	3.00
3.00	3.50	3.25

Example:

If the application is a centrifugal pump for load duration of over 10 hours per day, the service factor from Table A.2 is 1.25 for a motor or turbine. Table A.1 converts this value to 1.50 for a multi-cylinder engine and 1.75 for a single cylinder engine.

Note:

For service factors of high speed helical gear units, please use Table A.1 of ANSI/AGMA 6011. The standard includes design, lubrication, bearings, testing and rating for single and double helical external tooth, parallel shaft speed reducers or increasers. Units covered include those operating with at least one stage having a pitch line velocity equal to or greater than 35 meters per second or rotational speeds greater than 4500 rpm and other stages having pitch line velocities equal to or greater than 8 meters per second.

To give an idea of difference between ANSI/AGMA 6010 and ANSI/AGMA 6011, service factors for blowers as per Table A.1 of ANSI/AGMA 6011-I03 are shown in the following table.

Service Factors for Blowers as per Table A.1 of ANSI/AGMA 6011-I03				
Application	Service Factor, with Prime Mover			
	Synchronous Motors	Induction Motors	Gas or Steam Turbine	Internal Combustion Engine (Multi-Cylinder)
Blowers - Centrifugal	1.7	1.4	1.6	1.7
Blowers - Lobe	2.0	1.7	1.7	2.0

Thermal Power Rating

Selection of a gear drive must consider the thermal rating also as maintaining an acceptable temperature in the oil sump of a gear drive is critical to its life. This consideration is necessary to maintain proper lubrication. If a gear drive's capacity to dissipate thermal energy is insufficient, it will overheat and severe damage may occur.

Unacceptably high oil sump temperatures lead to increase in the oxidation rate of the oil and decrease in its viscosity. The decreased viscosity translates into reduced oil film thickness on the gear teeth and bearing contacting surfaces which may result in reducing the life of these elements. Hence to achieve the required life and performance of a gear drive, the operating oil sump temperatures must be evaluated and limited.

Thermal rating is defined as the maximum power that can be continuously transmitted through a gear drive without exceeding a specified oil sump temperature. The thermal rating must equal or exceed the transmitted power. Service factors are not used when determining thermal requirements because thermal rating depends upon the specifics of the drive, operating conditions, the maximum allowable sump temperature and the type of cooling employed.

Thermal ratings of gear drives rated by this standard (ANSI/AGMA 6010-F97) are limited to a maximum allowable oil sump temperature of 200°F. However, based on the gear manufacturer's experience or application requirements, selection can be made for oil sump temperatures above or below 200°F.

The basic thermal rating (P_T) is established by test or calculation under the following conditions.

- Oil sump temperature at 200°F (approximately 93°C)
- Ambient air temperature of 75°F (approximately 24°C)
- Ambient air velocity of $\leq$ 275 fpm in a large indoor space
- Air density at sea level
- Continuous operation

When the actual operating conditions for a specific application are different from the standard conditions (conditions given above), the thermal rating (P_T) may be modified (P_{Thm}) for the application as follows:

$$P_{Thm} = P_T\, B_{ref}\, B_V\, B_A\, B_T\, B_D$$

B_{ref} and B_A may be applied to natural or shaft fan cooling. B_V may be applied only to natural cooling.

Ambient Temperature Modifiers, B_{ref}	
Ambient Temperature, °F	**B_{ref}**
50	1.15
65	1.07
75	1.00
85	0.93
100	0.83
110	0.75
120	0.67

When the ambient air temperature is below 75°F, B_{ref} allows an increase in the thermal rating. Conversely, with an ambient air temperature above 75°F, the thermal rating is reduced.

Ambient Air Velocity Modifier, B_V	
Ambient Air Velocity, fpm	**B_V**
≤ 100	0.75
> 100 ≤275	1.00
> 275 < 725	1.40
≥ 725	1.90

When the surrounding air has a steady velocity in excess of 275 fpm due to natural or operational wind fields, the increased convection heat transfer allows the thermal rating to be increased by applying B_V. Conversely, with an ambient air velocity of ≤ 100 fpm, the thermal rating is reduced.

Altitude Modifier, B_A	
Altitude, ft	**B_A**
0 - Sea level	1.00
2500	0.95
5000	0.90
7500	0.85
10000	0.81
12500	0.76
15000	0.72
17500	0.68

At high altitudes, the decrease in air density results in the derating factor B_A.

Maximum Allowable Oil Sump Temperature Modifier, B_T	
Maximum Oil Sump Temperature, °F	**B_T**
185	0.81
200	1.00
220	1.13

The standard maximum allowable oil sump temperature is 200°F. A lower sump temperature requires a reduction in the thermal rating using B_T. A maximum allowable sump temperature in excess of 200°F will increase the thermal rating and can provide acceptable gear drive performance in some applications.

Operation Time Modifier, B_D	
Operation Time per Each Hour	**B_D**
100% (Continuous)	1.00
80%	1.05
60%	1.15
40%	1.35
20%	1.80

When a gear drive sees less than continuous operation with periods of zero speed, the resulting “cool-off” time allows the thermal rating to be increased by B_D.

The ability of a gear drive to operate within its thermal power rating may be reduced when adverse conditions exist. Some examples of adverse environmental conditions are:

- An enclosed space.
- A buildup of material that may cover the gear drive and reduce heat dissipation.
- A high ambient temperature, such as boiler, machinery or turbine rooms, or in conjunction with hot processing equipment.
- The presence of solar energy or radiant heat.

Auxiliary cooling should be used when the thermal rating is insufficient for operating conditions. The oil may be cooled by a number of means, some of which are:

- Fan cooling. The fan shall maintain the fan cooled thermal power rating.

- Heat exchanger. The heat exchanger used shah be capable of absorbing generated heat that cannot be dissipated by the gear drive by convection and radiation.

For more information on mechanical rating/capacity and thermal power rating, please see ANSI/AGMA 6013 Standard for Industrial Enclosed Gear Drives. This standard includes design, rating, lubrication, testing and selection information for enclosed gear drives, including foot mounted, shaft mounted, screw conveyor drives and gearmotors. These drives include spur, helical, herringbone, double helical, or bevel gearing in single or multistage arrangements, and wormgearing in multistage drives, as either parallel, concentric or right angle configurations. This standard combines and replaces the information previously found in ANSI/AGMA 6009-A00 and ANSI/AGMA 6010-F97.

Class of Service

As per ANSI/AGMA 6013, gear drives that are supplied in combination with electric motors (motorized reducer) may be designated with a service class number such as I, II or III, rather than a numerical service factor. Service classes I, II or III are equivalent to service factor values of 1.0, 1.41 or 2.0.

Following table shows service classes and service factors for various operating conditions.

AGMA Class of Service	Service Factor	Operating Conditions
I	1.00	Moderate Shock - not more than 15 minutes in 2 hours. Uniform Load - not more than 10 hours per day.
II	1.25	Moderate Shock - not more than 10 hours per day. Uniform Load - more than 10 hours per day.
	1.50	Heavy Shock - not more than 15 minutes in 2 hours. Moderate Shock - more than 10 hours per day.
III	1.75	Heavy Shock - not more than 10 hours per day.
	2.00	Heavy Shock - more than 10 hours per day.

Mechanical Rating as per ISO 9085:2002

ISO 9085 “Calculation of load capacity of spur and helical gears - Application for industrial gears” is equivalent to ANSI/AGMA 6113-A06, Metric Edition of ANSI/AGMA 6013-A06.

As per ISO 9085, mechanical rating of an enclosed gear drive can be selected as under.

Enclosed gear drives are designed to nominal load ratings (catalogue ratings) for sale from stock because the operating conditions are not exactly known at the time of design. Hence to take into account loads additional to nominal loads which are imposed on the gear drives from external sources, they should be evaluated and used to select an appropriately sized drive from the catalogue.

If it is not possible to determine the additional load by comprehensive system analysis, it is recommended to use the application factors (K_A), empirical guidance values to modify (reduce) the catalogue ratings as under.

$$\textbf{Absorbed Power} \leq \frac{\textbf{Catalogue Power Rating}}{\textbf{Application Factor}}$$

Recommended values of application factors (K_A) are given in the following table.

Working Characteristics of the Driving Machine	**Working Characteristics of the Driven Machine**			
	Uniform	Light Shocks	Moderate Shocks	Heavy Shocks
	Application Factors, K_A			
Uniform	1.00	1.25	1.50	1.75
Light Shocks	1.10	1.35	1.60	1.85
Moderate Shocks	1.25	1.50	1.75	2.00
Heavy Shocks	1.50	1.75	2.00	2.25 or Higher

Examples for driving machines with various working characteristics are given in the following table.

Working Characteristics	**Driving Machine**
Uniform	Electric motor (e.g. d.c. motor), steam or gas turbine with uniform operation and small rarely occurring starting torques
Light Shocks	Steam turbine, gas turbine, hydraulic or electric motor (large, frequently occurring starting torques)
Moderate Shocks	Multiple cylinder internal combustion engines
Heavy Shocks	Single cylinder internal combustion engines

Examples for driven machines with various working characteristics are given in the following table.

Working Characteristics	**Driven Machine**
Uniform	Steady load current generator; uniformly loaded conveyor belt or platform conveyor; worm conveyor; light lifts; packing machinery; feed drives for machine tools; ventilators; lightweight centrifuges; centrifugal pumps; agitators and mixers for light liquids or uniform density materials; shears; presses, stamping machines; vertical gear, running gear.
Light Shocks	Non-uniformly (i.e. with piece or batched components) loaded conveyor belts or platform conveyors; machine tool main drives; heavy lifts; crane slewing gear; industrial and mine ventilator; heavy centrifuges; centrifugal pumps; agitators and mixers for viscous liquids or substances of non-uniform density, multi-cylinder piston pumps, distribution pumps; extruders (general); calendars; rotating kilns; rolling mill stands (continuous zinc and aluminium strip mills, wire and bar mills).
Moderate Shocks	Rubber extruders; continuously operating mixers for rubber and plastics; ball mills (light); woodworking machine (gang saws, lathes); billet rolling mills; lifting gear; single cylinder piston pumps.
Heavy Shocks	Excavators (bucket wheel drives), bucket chain drives; sieve drives; power shovels, ball mills (heavy); rubber kneaders; crushers (stone, ore); foundry machines; heavy distribution pumps; rotary drills; brick presses; debarking mills; peeling machines; cold strip; briquette presses; breaker mills.

The application factors should be used with caution since much higher values have occurred in some applications. Values as high as 10 have been used.

The Recommended values of application factors only apply to transmissions which operate outside the resonance speed range under relatively steady loading. If operating conditions involve unusually heavy loading, motors with high starting torques, intermittent service or heavy repeated shock loading, the safety of the static and limited-life load capacity of the gears shall be verified (see ISO 6336-1, ISO 6336-2 and ISO 6336-3).

Example:

If a stone crusher is running with a 10 kW electric motor, the application factor will be 1.75 (because the working characteristics of the driving machine, electric motor is uniform and working characteristics of the driven machine, stone crusher is heavy shocks). As the absorbed power is 10 kW, catalogue power rating (mechanical rating) of the gear drive should be more than or equal to 10 × 1.75 kW = 17.5 kW.

Gear Drive Selection as per IS 7403: 1974

In India, many gear drive manufacturers are designing gear drives based on IS 7403 “Code of Practice for Selection of Standard Worm and Helical Gear Boxes” (or they follow methodology similar to IS 7403). In view of this, information about the standard is given in this section.

Service Factors

A gear box is rated to a specific application by the use of service factors. Service factor (*S*) is the ratio of maximum power rating of a particular size of gear box to the power required for driving the machine or equipment to which the gear box is coupled.

$$\text{Service Factor} = \frac{\text{Maximum Rated Power}}{\text{Actual Power Required}}$$

Each application has its own conditions and operating requirements, and following table (Table 1 of the standard) covers the list of service factors.

Prime Mover	Duration of Service Hours/Day	Service Factors (Table 1 of IS 7403)			
		Nature of Load on Gear Unit from Driven Machine			
		Uniform	Moderate Shock	Heavy Shock, Worm	Heavy Shock, Helical and Spur
Electric motor or steam turbine	2	0.75	0.90	1.25	1.40
	4	0.80	1.00	1.30	1.50
	8	0.90	1.10	1.45	1.65
	12	1.00	1.25	1.55	1.75
	24	1.25	1.50	1.75	2.00
Multi-cylinder internal combustion engine	2	0.90	1.10	1.25	1.65
	4	1.00	1.25	1.40	1.75
	8	1.10	1.35	1.60	1.90
	12	1.25	1.50	1.75	2.00
	24	1.50	1.75	2.00	2.25
Single-cylinder internal combustion engine	2	1.10	1.35	1.75	1.90
	4	1.25	1.50	1.85	2.00
	8	1.35	1.65	1.95	2.15
	12	1.50	1.75	2.05	2.25
	24	1.75	2.00	2.25	2.50

The service factors given in above table are based on the assumption that the system is free from severe critical or torsional vibrations, and that the maximum momentary or starting loads do not exceed 200 percent of the normal load (100 percent overload).

Application Classification

The load characteristics of various machines and equipment in which reduction gear boxes are used, have been classified according to their operating conditions in Table 2 of the standard. This table is intended as a guide to assist in deciding the appropriate service factor for a particular duty, and should be used in conjunction with service factors given in Table 1 of the standard. As an illustration, load characteristics values for some common applications from Table 2 of the standard are given in the following table.

Load Characteristics Values for Some Common Application
(As per Table 2 of IS 7403)

U = Uniform Load
M = Moderate Shock Load
H = Heavy Shock Load
HI = High Inertia Load. Manufacturer to be consulted.

Driven Machine	Type of Load
Agitators:	
• Pure liquids	U
• Liquids and solids	M
• Liquids - variable density	M
Blowers:	
• Centrifugal	U
• Lobe	M
• Vane	U
Clarifiers:	U
Compressors:	
• Centrifugal	U
• Lobe	M
• Reciprocating, multi-cylinder	M
• Reciprocating, single-cylinder	H
Conveyors:	
• Apron, uniformly loaded or fed	U
• Apron, heavy duty not uniformly fed	M
• Belt, uniformly loaded or fed	U
• Belt, heavy duty not uniformly fed	M
Crusher:	
• Ore	H
• Stone	H
Elevators:	
• Bucket, uniform load	U
• Bucket, heavy load	M
Fans:	
• Centrifugal	U
• Induced draft	M
• Large - mine, etc	U
• Large - industrial	M
• Light - smaller diameter	U
Feeders:	
• Apron	M
• Belt	M
• Screw	M

Hammer mills	H
Hoists:	
• Heavy duty	H
• Medium duty	M
Mills - rotary type:	
• Ball	M
• Cement kilns	M
Printing presses	HI
Pumps	
• Centrifugal	U
• Proportioning	M
• Reciprocating - Single acting, 3 or more cylinders	M
• Reciprocating - Double acting, 2 or more cylinders	M
• Reciprocating - Single acting, 1 or 2 cylinders	HI
• Rotary - Gear type	U
• Rotary - Lobe type	U
• Rotary - Vane type	U
Screens - Traveling water intake	U

For information on other applications (cranes, lumber industry, metal mills, etc.), please refer Table 2 of the standard.

Basic Steps for Selection

Determine Service Factor *S* - The appropriate service factor for the application is determined by referring to tables given above (Tables 1 and 2 of the standard).

Calculate Equivalent Power - Equivalent power is equal to the actual power required for driving the machine at desired speed multiplied by service factor.

$$\text{Equivalent power} = \text{Actual power} \times S$$

Determine Speed Ratio - This is obtained by dividing the speed of high speed shaft, by the speed of low speed shaft.

Select Size from Rating Table - The rated power for the size of unit selected should be equal to, or greater than the calculated equivalent power for the given speed ratio of the gear box selected.

Check for Momentary or Peak Load - The gear box selected can take momentary loads of twice the rated power. However, in case the peak power requirements of the driven machine exceeds by more than 100 percent the rated power, the size selected should have rated power equal to or greater than half of the peak power. Momentary load means a load which acts for a period of not more than 15 seconds.

Note:

As per one leading Indian manufacturer, all the components of gear drives manufactured by them are so designed that they can withstand overloads as under.

- 100 per cent overload for 15 seconds
- 50 per cent overload for one minute
- 40 per cent overload for 30 minutes and
- 25 per cent overload for two hours.

Above values may be used for different overload conditions. Many manufacturers also limit 100% overload at starting, braking or momentarily during operation - up to 10 times per day, not more than 5 times per hour, etc. Follow the manufacturer's recommendation.

Check Brake Torque - When a motor or engine is equipped with brake, whose torque rating exceeds that of the motor or engine, the torque rating of gear box selected should exceed brake torque, all other things being equal.

Check Overhung Load - If a sprocket, pulley or spur pinion is mounted on the output shaft, overhung load is imposed on the shaft. The equivalent overhung load in kg is calculated as under:

$$P = \frac{A \times 975000 \times K}{N \times R}$$

Where,

P = equivalent overhung load in kg
A = actual power carried by the shaft in kW
K = load factor
N = shaft speed in rpm
R = pitch radius of sprocket, pulley or spur pinion in mm.

The load factors are given in the following table.

Overhung Member	Load Factor, *K*
Sprocket	1.00
Spur pinion	1.25
V-belt sheave	1.50
Flat belt pulley	3.00

The calculated overhung loads should be less than the maximum permissible overhung loads on gear boxes.

Note:

Overhung loads can be reduced by increasing the diameter of the sprocket, gear etc. If the maximum permissible overhung load is exceeded, the sprocket, gear, etc. should be mounted on a separate shaft, flexibly coupled and supported in its own bearings. Alternatively, a larger gear is often a less expensive solution.

Number of Starts Factor

If a gear drive is subjected to torque reversals or frequent start-ups or overloads, a further check in its selection is required. For this, many times a starts factor, dependant on the frequency of starting, is given in the catalogue.

In such cases, after selecting a gear drive's mechanical power rating based on its service condition (using service factor), the input power capacity of the gear drive should be checked for its adequacy to take care of torque reversals or frequent start-ups or overloads as under.

$$\text{Input power capacity of gear drive (kW)} \geq \frac{Tm \times Fs \times n}{2 \times 9550}$$

Where,

Tm = motor starting torque (Nm) or rating of torque limiting device, fluid coupling, etc
n = input speed (rev/min)
Fs = number of starts factor

Following table gives values for number of starts factors (Fs).

Start / Stops per hour*	Up to 1	5	10	40	60	≥ 200
Unidirectional	1.0	1.03	1.06	1.10	1.15	1.20
Reversing	1.40	1.45	1.50	1.55	1.60	1.70

* Intermediate values are obtained by linear interpolation

The larger of the two powers thus calculated is then used to select a gear drive.

Note:

The values of starts factors (Fs) given in above table are as per recommendation of a reputed gear drive manufacturer.

In case when such table is not provided in the catalogue, please consult the manufacturer.

Efficiency of Enclosed Gear Drives

The approximate full load efficiencies in percent (%) of parallel / right angle shaft enclosed gear drives designed as per good engineering practice would be as per the following table.

No. of Reductions	Parallel Shafts	Right Angle Shaft
Single	99.0	-
Double	98.0	97.5
Triple	97.5	97.0
Quadruple	97.0	96.5

Acknowledgement

Most of the information given in this chapter is extracted from ANSI/AGMA 6010-F97, Standard for Spur, Helical, Herringbone and Bevel Enclosed Drives, with the permission of the publisher, the American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, Virginia 22314.

Gear Materials and Heat Treatment

A variety of cast irons, powder metallurgy materials, nonferrous alloys, and plastics are used in gears, but steels, because of their high strength-to-weight ratio and relatively low cost, are the most widely used gear materials for heavy duty, power transmission applications. In view of this, information about steels used for making gears, heat treatment operations and bronzes is given in this chapter.

Steels for Gears

There are a number of steels used for gears, ranging from plain carbon steels through the highly alloyed steels and from low to high carbon contents. The choice depends on number of factors, including size, service and design.

The common steels [AISI (American Iron and Steel Institute) Steel Grades] listed in the following table are used for making gears. The recommended heat treatments for them is also given in the table.

Typical Gear Materials - Wrought Steel (Table 4-1, ANSI/AGMA 2004-B89)		
Steel / Alloy Grades	**Common Heat Treat Practice***	**General Remarks/Application**
1045	T-H, I-H, F-H	Low Hardenability
4130	T-H	Marginal Hardenability
4140	T-H, T-H&N, I-H, F-H	Fair Hardenability
4145	T-H, T-H&N, I-H, F-H	Medium Hardenability
8640	T-H, T-H&N, I-H, F-H	Medium Hardenability
4340	T-H, T-H&N, I-H, F-H	Good Hardenability in Heavy Sections
Nitralloy 135 Mod.	T-H&N	Special Heat Treatment
Nitralloy G	T-H&N	Special Heat Treatment
4150	I-H, F-H, T-H, TH&N	Quench Crack Sensitive Good Hardenability
4142	I-H, F-H, T-H&N	Used when 4140 exhibits Marginal Hardenability
4350	T-H, I-H, F-H	Quench Crack Sensitive, Excellent Hardenability in Heavy Sections
1020	C-H	Very Low Hardenability
4118	C-H	Fair Core Hardenability
4620	C-H	Good Case Hardenability
8620	C-H	Fair Core Hardenability
4320	C-H	Good Core Hardenability
8822	C-H	Good Core Hardenability in Heavy Sections
3310	C-H	Excellent Hardenability (in Heavy Sections)
4820	C-H	Excellent Hardenability (in Heavy Sections)
9310	C-H	Excellent Hardenability (in Heavy Sections)

* C-H = Carburize Harden, F-H = Flame Harden, I-H = Induction Harden, T-H = Through Harden and T-H&N = Through Harden then nitride.

Above table is extracted from AGMA 2004-B89, Gear Materials and Heat Treatment Manual, with the permission of the publisher, the American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, Virginia 22314.

In gear terminology, some steels are considered to be alloy steels and some are considered to be plain carbon steels. The steels used for gears tend to vary from those with a small amount of alloys to steels rich in alloys. From a practical standpoint, the steels that are considered to be plain carbon usually have some alloy content, such as manganese and silicon. The steels that are normally thought of as alloy steels usually have chromium, nickel, and molybdenum as well as manganese and silicon.

The alloy content helps out in several very important ways:

- The cooling rate in quenching can be considerably slower. This makes it possible to get a good metallurgical structure in the large gears. (A large gear with low alloy content does tend to be weaker because a good metallurgical structure is not obtained.)

- Gears with high alloy content can be carburized with too rich or too lean a carburizing atmosphere and still come out fairly good. Nickel, in particular, makes heat treating operations less sensitive to precise control.

- Certain alloy combinations are helpful in developing fracture toughness. (With these combinations, a small crack grows very slowly, or perhaps ceases to grow. This is important in gears that suffer some surface damage but could run a long time if the surface damage did not lead to tooth breakage.)

- In general, the impact properties are considerably improved by alloy content. Nickel and molybdenum are particularly valuable for impact strength. (Many gears suffer occasional heavy shock loads and therefore need good impact strength.)

In short, the composition of the steel used in gears is very important. A poor choice of alloy content has often led to early failure in gears built with good precision and sized large enough to meet appropriate design standards.

Heat Treatment

Heat treatment refers to operations involving the heating and cooling of a metal in the solid state for the purpose of obtaining certain desired conditions or properties. Information about various heat treatment processes is given in this section.

Heat treating is one of the most important steps in the manufacture of precision gearing. Its contribution is vitally important for cost control, durability, and reliability. Heat treating represents about 30% of a typical gear manufacturing cost.

Common heat treatments for ferrous materials include:

Preheat Treatments - Anneal, Normalize or normalize and temper, Quench and temper, Stress relief

Heat Treatments - Through harden (anneal, normalize, or normalize and temper, and quench and temper), Surface harden profile heated (flame and induction harden), Surface harden profile chemistry modified (carburize, carbonitride and nitride)

Post Heat Treatment - Stress relieve

Gear steels are heat treated for two general purposes. First, they must be put in condition for proper machinability. Second, the necessary hardness, strength, and wear resistance for the intended use must be developed.

Normalizing and Annealing

Steel in the as rolled or as forged condition may be coarser grained and has non-uniform hardness as a result of uncontrolled cooling after the forging or rolling operations. Therefore, heat treatment followed by controlled cooling is used to develop the type of metallurgical structure most suited to the subsequent machining operations. Gear forgings of carburizing steels are usually normalized or normalized and tempered to develop a uniform microstructure and reduce their tendency to distort during later hardening operations. Normalizing consists of heating the steel to well above the upper critical temperature (Ac_3), that is the temperature at which austenite begins to transform to ferrite during cooling, and subsequently cooling in still or circulated air to room temperature.

Alloy steels are normally tempered at 1000-1250°F (538-677°C) after normalizing for uniform hardness, dimensional stability and improved machinability.

Another treatment similar to normalizing is annealing. The part to be annealed is heated above the upper critical temperature, just as in normalizing, but it is cooled at a slow rate either by controlling the furnace cooling rate or by allowing the furnace and load to cool off together with the doors closed to around 315°C. Slow cooling in the furnace usually produces a pearlitic or lamellar structure that provides good surface finishes after machining.

It may be noted that a normalized part is very machinable but harder than an annealed part.

Quenching (Hardening) and Tempering

Steel parts are hardened by quenching and tempering to develop the combination of strength, toughness, hardness, and wear resistance that may be needed to make the part function properly. Proper hardening consists of cooling the steel from above the upper critical temperature (the temperature at which the steel is completely austenized) quickly enough to form a fully hardened structure and to prevent the formation of undesired structures, which could occur at intermediate temperatures if the cooling rate is too slow. The critical cooling rate is the rate of cooling that is just fast enough to produce a fully hardened structure in a particular steel composition. After quenching, steels are usually tempered or stress-relieved. The tempering operation may be used to reduce the hardness of a part and increase the toughness. By properly adjusting the tempering temperature, a wide range of hardness may be obtained. Even when no reduction in hardness is desired, a low temperature (250-350°F) tempering operation is desirable to reduce stresses in the steel and produce a kind of martensite that is tougher than the kind produced immediately upon quenching.

The carbon content of a steel establishes the maximum hardness that can be reached in the fully hardened condition, while the alloying elements determine the critical cooling rate necessary for full hardening and therefore the section thickness that will harden with the quench that is available. For example, plain carbon steels have such high critical cooling rates that they must be water or brine quenched to be fully hardened even when the section is relatively small. Alloy steels transform more slowly and can be hardened with an oil quench. In the case of some high alloy tool and die steels, even cooling in still air will develop high hardness!

Stress Relieving

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

Through (Direct) Hardening

Through hardening is a term used to collectively describe methods of heat treatment of steel other than surface hardening techniques. These include: annealing, normalizing (or normalizing and tempering) and quenching and tempering. Depth of hardening is dependent upon hardenability, section size and heat treat considerations.

It may be noted that through hardening does not imply that the part has equivalent (same) hardness throughout the entire cross section. Since the outside of a gear is cooled faster than the inside, there will be a hardness gradient developed. The final hardness is dependent on the amount of carbon in the steel and the depth of hardness depends on the hardenability of the steel as well as the quench severity.

There are generally three methods of heat treating through hardened gears. In ascending order of hardness for a particular type of steel they are: annealing, normalizing (or normalizing and tempering), and quenching and tempering.

The quench and temper process on ferrous alloys involves heating to form austenite at 1475-1600°F (802-871°C), followed by rapid quenching. The rapid cooling causes the gear to become harder and stronger by formation of martensite. The gear is then tempered to a specific temperature, generally below 1275°F(691°C), to achieve the desired mechanical properties. Tempering reduces the material hardness and mechanical strength but improves the material ductility and toughness (impact resistance). Selection of the tempering temperature must be based upon the specified hardness range, material composition, and the as quenched hardness. The tempered hardness varies inversely with tempering temperature. Parts are normally air cooled from tempering temperatures.

The hardness and mechanical properties achieved from the quench and temper process are higher than those achieved from the normalize or anneal process.

Through hardening can be performed either before or after the gear teeth are cut. When gear teeth will be cut after the part has been hardened, surface hardness and machinability become important factors especially in light of the fact that machining will remove some or most of the higher hardness material at the surface.

Case Hardening

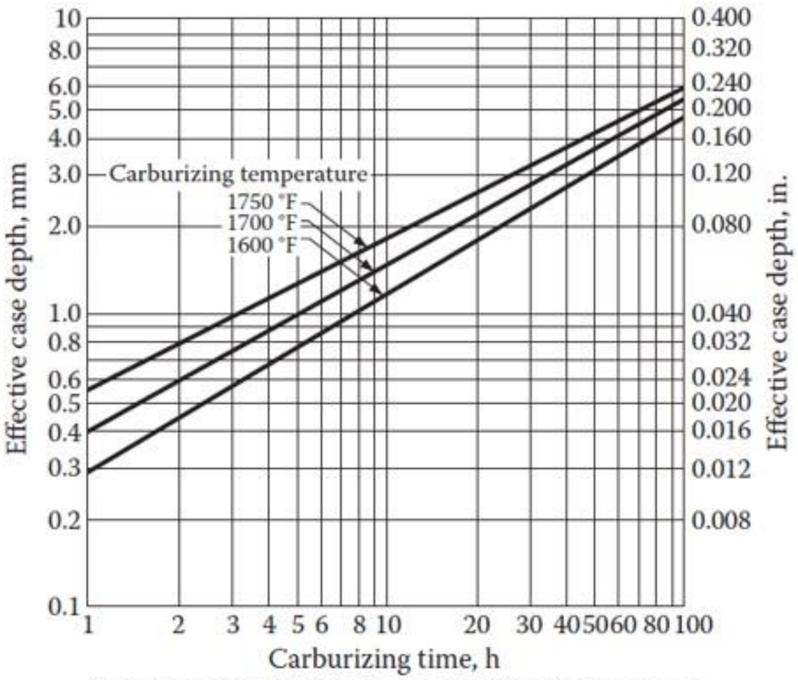
Case hardening produces a hard, wear resistant case or surface layer on top of a ductile, shock resistant interior, or core. The idea behind case hardening is to keep the core of the gear tooth at a level around 30 to 40 HRC to avoid tooth breakage while hardening the outer surface to increase pitting resistance. The higher the surface hardness value, the greater the pitting resistance.

Several methods are used either to case harden only the gear teeth or to harden the surfaces of gear teeth and leave the inside part of the tooth at an intermediate hardness. Carburizing, nitriding, induction hardening and flame hardening can be used to produce gear teeth that are much harder than the gear blank that supports the teeth.

Carburizing

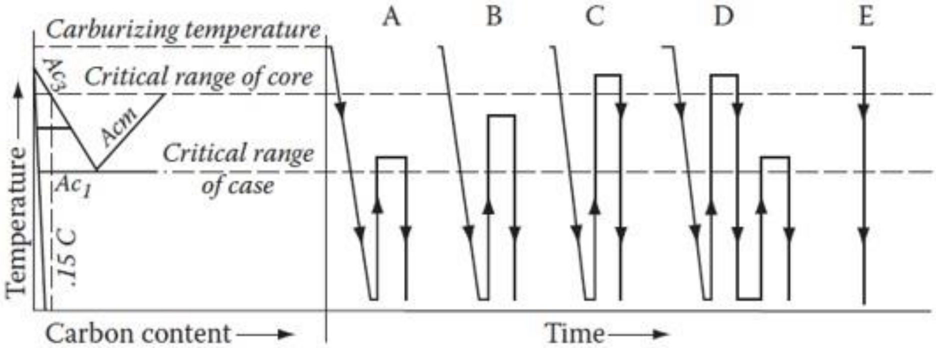
Carburizing is the oldest and probably the most widely used process for hardening gear teeth. It consists of heating a 0.10% to 0.25% carbon steel at a temperature above its critical range in a gaseous, solid, or liquid medium capable of imparting carbon to the steel. The surface layer becomes enriched in carbon and therefore is capable of developing a high degree of hardness after quenching.

The concentration of carbon in the surface layer is determined by many factors. If uncontrolled, it may reach 1.20%. For best strength and toughness, though, the concentration should be kept under 1.0%, preferably around 0.80% to 0.90%. Control of the richness of the carbon case is obtained by controlling the richness of the carburizing atmosphere.



Nominal Time and Temperature Requirements for Different Case Depths by Carburizing

The development of the case depends on the diffusion of carbon into the steel. Therefore, time and temperature are the main factors that control the case depth. Steel composition has no great influence on the rate of carbon penetration. Above figure shows the approximate effect of temperature and time on the depth of case obtained in a gaseous carburizing operation.



Different Treatments Subsequent to Carburization

Carburized parts may be heat treated in several ways to achieve a variety of case and core properties. Several of the most important treatments are illustrated in above figure (From International Nickel Co., NY, USA.). For heavy duty gearing, treatment C generally provides both a good case and a strong core.

For best load carrying capacity, the case should be up to 700 HV (60 HRC), and the core should be in the range of 340 to 415 HV (35 to 42 HRC).

Besides surface hardness, the carburized pinion or gear needs an adequate case depth. There are subsurface stresses that are strong enough to cause cracks in the region of case to core interface if the case is too thin.

Carbonitriding

Typically, carbonitriding is carried out at lower temperatures, 1550-1650°F (843-899°C), and for shorter times than gas carburizing. Shallower case depths are generally specified for carbonitriding than is usual for production carburizing.

Normally 2.5 to 5 percent anhydrous ammonia is added to the carburizing atmosphere when carbonitriding. Because nitrogen inhibits the diffusion of carbon, what generally results is a shallower case than is typical for carburized parts. A carbonitrided case is usually between 0.075 to 0.75 mm (0.003 to 0.030 in.) deep.

Use of carbonitriding is more restricted than carburizing. It is limited to shallower cases for finer pitch gearing since the process must be conducted at lower temperatures than carburizing. One of the advantages of carbonitriding is better case hardenability in lower alloy or plain carbon steels. The carbonitrided case has better wear and temper resistance than a straight carburized case. However, if higher core hardness and deeper case depths are required for bending resistance, carbonitriding may not be applicable.

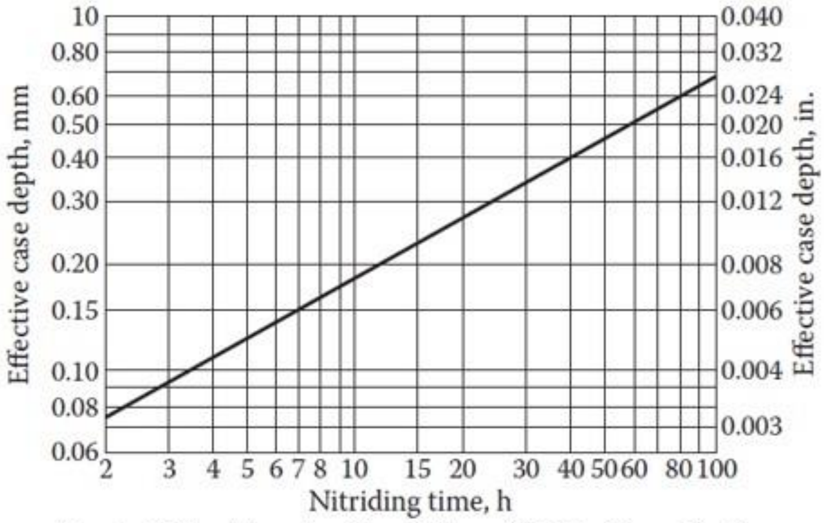
Nitriding

Nitriding is a case hardening process in which the hardening agents are nitrides formed in the surface layers of steel through the absorption of nitrogen from a nitrogenous medium, usually dissociated ammonia gas.

Almost any steel composition will absorb nitrogen, but useful cases can be obtained only on steels that contain appreciable amounts of aluminum, chromium or molybdenum. Other elements, such as nickel and vanadium, may be needed for their special effects on the properties of the nitrided steel.

A nitride case does not form as fast as a carbon case. Following figure shows the nominal relation between nitriding time and case depth obtained. Nitriding, like carburizing, is a difficult process, but because the rate of penetration is slower, the time cycles are quite long.

For best results, parts to be case hardened by nitriding should be rough machined and then quenched and tempered. The tempering temperature is usually between 1000°F and 1150°F. After heat treatment, the part is finish machined, stress relieved at about 1100°F, and then nitrided.



Nominal Time Required for Different Nitride Case Depths

Conventional gas nitride hardening of gears, which had a quench and temper pretreatment and are usually finish machined, involves heating and holding at a temperature between 950-1060°F (510-571°C) in a controlled cracked ammonia atmosphere (10 to 30 percent dissociation). The modern practice is to start the process with 30% ammonia dissociation for the first several hours, then to allow dissociation to increase to 85%. Nitride hardening can also be achieved with the ion nitriding process. During nitriding, nitrogen atoms are absorbed into the surface to form hard iron and alloy nitrides. The practical limit on case depth is about 0.040 inch (1.0 mm) maximum. Typically, case depths are between 0.20 to 0.65 mm (0.008 to 0.025 in.) and take from 10 to 80 hours to produce.

Since the nitriding temperature is lower than the original tempering temperature of the steel, hardening occurs with minimum distortion. The method is therefore suitable for complex parts such as gears that can be machined while in the medium hardness region and then hardened without enough distortion to require grinding. Parts that are too flimsy to stand a quench and draw without serious distortion can often be successfully brought up to full hardness by nitriding.

For best results, parts to be case hardened by nitriding should be rough machined and then quenched and tempered. The tempering temperature is usually between 1000°F and 1150°F. After heat treatment, the part is finish machined, stress relieved at about 1100°F, and then nitrided. If the blank has residual stresses before nitriding, the many hours at nitriding temperature will relieve these stresses and cause the part to warp.

Nitrided gears are used when gear geometry and tolerances do not allow use of other case hardening methods because of distortion, and when through hardened gears do not provide sufficient wear and pitting resistance. Nitrided gears are used on applications where thin, high hardness cases can withstand applied loads. Nitrided gears should not be specified if shock loading is present, due to inherent brittleness of the case.

However, the nitrided gear, because of its high hardness, resists scoring and abrasion better than other types of gears.

Flame and Induction Hardening

Flame or induction hardening of a gear involves heating of the gear teeth to 1450-1600°F (788-871°C) followed by quench and tempering. An oxyfuel burner is used for flame hardening. An encircling coil or tooth by tooth inductor is used for induction hardening. These processes develop a hard wear resistant case on the gear teeth. When only the surface is heated to the required depth, only the surface is hardened during quenching. Material selection and heat treat condition prior to flame or induction hardening significantly affects the hardness and uniformity of properties which can be obtained.

Flame and induction hardening have been used successfully on most gear types; e.g., spur, helical, bevel, herringbone, etc. These processes are used when gear teeth require high surface hardness, but size or configuration does not lend itself to carburizing and quenching the entire part. These processes may also be used when the maximum contact and bending strength achieved by carburizing is not required.

Bronzes

The metals copper, zinc, tin, aluminum, and manganese are used in various combinations as gear materials. The most important, perhaps, is the alloy called bronze.

Like the terms “steel” and “cast iron,” “bronze” is really the name of a family of materials that may vary over a wide range of composition and properties. A bronze family is an alloy of copper and tin, as compared with “brass,” which is an alloy of copper and zinc.

Most of them are used in worm gearing where the reduced coefficient of friction between dissimilar materials and increased malleability are desired.

The bronzes are very important in worm gear work because of their ability to withstand high sliding velocity and to wear in to fit hardened steel worms. They are also very useful in applications in which corrosion is a problem. The ease with which bronze can be worked makes it a good choice where small gear teeth are produced by stamping or by drawing rods through dies.

The basic bronze is an alloy of 90% copper and 10% tin that exhibits the desired two-phase structure. The best strength and bearing properties are developed if the hard and soft constituents are finely and intimately dispersed. This is usually accomplished by carefully chilling the casting.

The tin bronzes deoxidized with phosphor are usually the best for gear applications. The term phosphor bronze literally means a bronze deoxidized with phosphor. Tin bronze commonly designated as SAE C90700 (obsolete SAE 65) was the predominant bronze alloy for gear manufacturing throughout the 20th century, and continues to prevail today.

For more information on gear materials and heat treatment, please see the following publications.

ANSI/AGMA 2004-B89, Gear Materials and Heat Treatment Manual, publisher by the American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, Virginia 22314.

Dudley’s Handbook of Practical Gear Design and Manufacture, published by CRC Press, Taylor & Francis Group, 6000 Broken Sound Parkway NW, Suite 300, Boca Raton, FL 33487-2742 USA.

Gear Materials, Properties, and Manufacture, published by ASM International®, Materials Park, OH 44073-0002, USA.

References

Gearbox and Gear Drive Maintenance Guide, EPRI (Electric Power Research Institute, Inc. USA), Palo Alto, CA: 2004. 1009831.

Gear Materials, Properties, and Manufacture, ASM International®, Materials Park, OH 44073-0002 USA | website: www.asminternational.org

Handbook of Practical Gear Design and Manufacture, Second Edition, published by CRC Press, Taylor & Francis Group, 6000 Broken Sound Parkway NW, Suite 300, Boca Raton, FL 33487-2742, USA.