ENGINEERING INFORMATION

SPUR GEARS
GEAR NOMENCLATURE

ADDENDUM (a) is the height by which a tooth projects beyond the pitch circle or pitch line.

BASE DIAMETER (D_b) is the diameter of the base cylinder from which the involute portion of a tooth profile is generated.

BACKLASH (B) is the amount by which the width of a tooth space exceeds the thickness of the engaging tooth on the pitch circles. As actually indicated by measuring devices, backlash may be determined variously in the transverse, normal, or axial-planes, and either in the direction of the pitch circles or on the line of action. Such measurements should be corrected to corresponding values on transverse pitch circles for general comparisons.

BORE LENGTH is the total length through a gear, sprocket, or coupling bore.

CIRCULAR PITCH (p) is the distance along the pitch circle or pitch line between corresponding profiles of adjacent teeth.

CIRCULAR THICKNESS (t) is the length of arc between the two sides of a gear tooth on the pitch circle, unless otherwise specified.

CLEARANCE-OPERATING (c) is the amount by which the dedendum in a given gear exceeds the addendum of its mating gear.

CONTACT RATIO (m_c) in general, the number of angular pitches through which a tooth surface rotates from the beginning to the end of contact.

DEDENDUM (b) is the depth of a tooth space below the pitch line. It is normally greater than the addendum of the mating gear to provide clearance.

DIAMETRAL PITCH (P) is the ratio of the number of teeth to the pitch diameter.

FACE WIDTH (F) is the length of the teeth in an axial plane.

FILLET RADIUS (r_f) is the radius of the fillet curve at the base of the gear tooth.

FULL DEPTH TEETH are those in which the working depth equals 2.000 divided by the normal diametral pitch.

GEAR is a machine part with gear teeth. When two gears run together, the one with the larger number of teeth is called the gear.

HUB DIAMETER is outside diameter of a gear, sprocket or coupling hub.

HUB PROJECTION is the distance the hub extends beyond the gear face.

INVOLUTE TEETH of spur gears, helical gears and worms are those in which the active portion of the profile in the transverse plane is the involute of a circle.

LONG- AND SHORT-ADDENDUM TEETH are those of engaging gears (on a standard designed center distance) one of which has a long addendum and the other has a short addendum.

KEYWAY is the machined groove running the length of the bore. A similar groove is machined in the shaft and a key fits into this opening.

NORMAL DIAMETRAL PITCH (P_n) is the value of the diametral pitch as calculated in the normal plane of a helical gear or worm.

NORMAL PLANE is the plane normal to the tooth surface at a pitch point and perpendicular to the pitch plane. For a helical gear this plane can be normal to one tooth at a point laying in the plane surface. At such point, the normal plane contains the line normal to the tooth surface and this is normal to the pitch circle.

NORMAL PRESSURE ANGLE ($ø_n$) in a normal plane of helical tooth.

OUTSIDE DIAMETER (D_o) is the diameter of the addendum (outside) circle.

SPUR GEARS
GEAR NOMENCLATURE (Continued)

PITCH CIRCLE is the circle derived from a number of teeth and a specified diametral or circular pitch. Circle on which spacing or tooth profiles is established and from which the tooth proportions are constructed.

PITCH CYLINDER is the cylinder of diameter equal to the pitch circle.

PINION is a machine part with gear teeth. When two gears run together, the one with the smaller number of teeth is called the pinion.

PITCH DIAMETER (D) is the diameter of the pitch circle. In parallel shaft gears, the pitch diameters can be determined directly from the center distance and the number of teeth.

PRESSURE ANGLE (ø) is the angle at a pitch point between the line of pressure which is normal to the tooth surface, and the plane tangent to the pitch surface. In involute teeth, pressure angle is often described also as the angle between the line of action and the line tangent to the pitch circle. Standard pressure angles are established in connection with standard gear-tooth proportions.

ROOT DIAMETER (D_r) is the diameter at the base of the tooth space.

PRESSURE ANGLE—OPERATING ($ø_r$) is determined by the center distance at which the gears operate. It is the pressure angle at the operating pitch diameter.

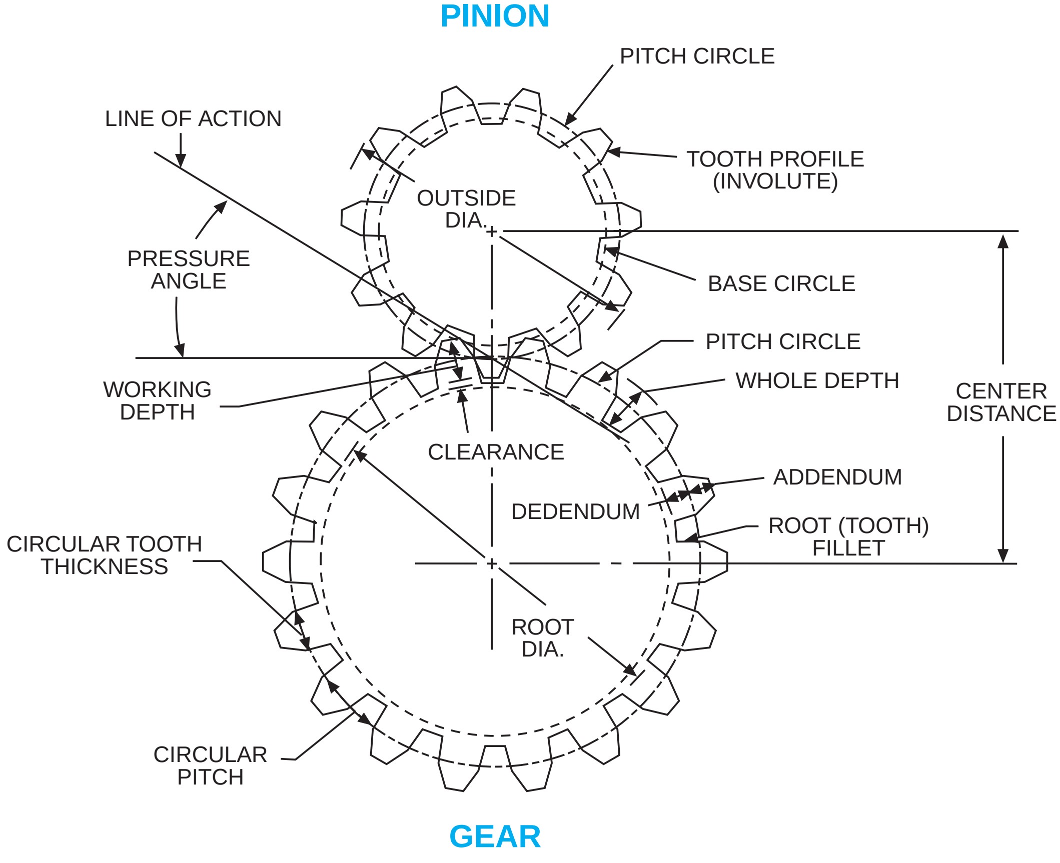
TIP RELIEF is an arbitrary modification of a tooth profile whereby a small amount of material is removed near the tip of the gear tooth.

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

WHOLE DEPTH (h_t) is the total depth of a tooth space, equal to addendum plus dedendum, equal to the working depth plus variance.

WORKING DEPTH (h_k) is the depth of engagement of two gears; that is, the sum of their addendums.

TOOTH PARTS

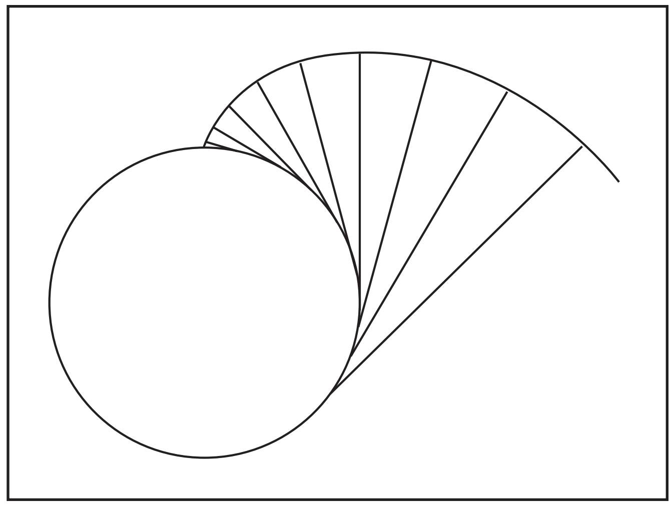


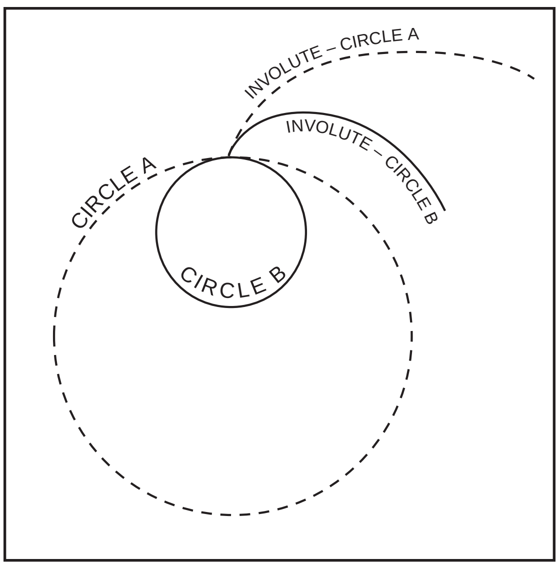
SPUR GEARS
INVOLUTE FORM

Gear teeth could be manufactured with a wide variety of shapes and profiles. The involute profile is the most commonly used system for gearing today, and all Boston spur and helical gears are of involute form.

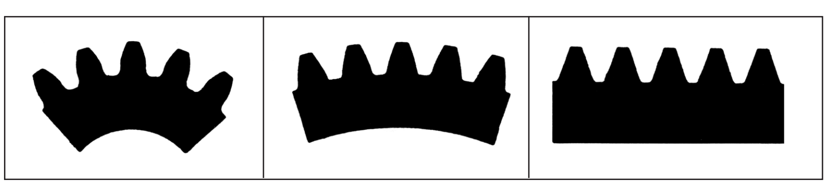
An involute is a curve that is traced by a point on a taut cord unwinding from a circle, which is called a BASE CIRCLE. The involute is a form of spiral, the curvature of which becomes straighter as it is drawn from a base circle and eventually would become a straight line if drawn far enough.

An involute drawn from a larger base circle will be less curved (straighter) than one drawn from a smaller base circle. Similarly, 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.





Involute gear tooth forms and standard tooth proportions are specified in terms of a basic rack which has straight-sided teeth, for involute systems.



20 TEETH 48 TEETH RACK

SPUR GEARS

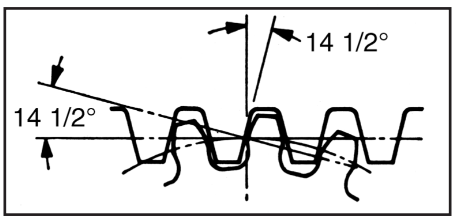
DIAMETRAL PITCH SYSTEM

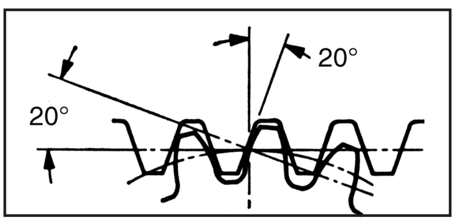
All stock gears are made in accordance with the diametral pitch system. The diametral pitch of a gear is the number of teeth in the gear for each inch of pitch diameter. Therefore, the diametral pitch determines the size of the gear tooth.

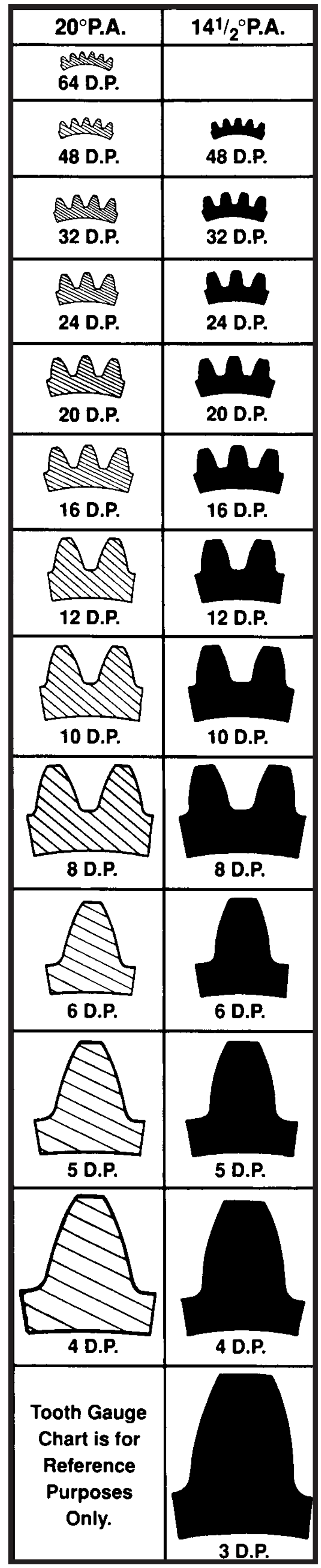
PRESSURE ANGLE

Pressure angle is the angle at a pitch point between the line of pressure which is normal to the tooth surface, and the plane tangent to the pitch surface. The pressure angle, as defined in this catalog, refers to the angle when the gears are mounted on their standard center distances.

Boston Gear manufactures both 14-1/2° and 20° PA, involute, full depth system gears. While 20°PA is generally recognized as having higher load carrying capacity, 14-1/2°PA gears have extensive use. The lower pressure angle results in less change in backlash due to center distance variation and concentricity errors. It also provides a higher contact ratio and consequent smoother, quieter operation provided that undercut of teeth is not present.







TOOTH DIMENSIONS

For convenience, Tooth Proportions of various standard diametral pitches of Spur Gears are given below.

Diametral Pitch	Circular Pitch (Inches)	Thickness of Tooth on Pitch Line (Inches)	Depth to be Cut in Gear (Inches) (Hobbed Gears)	Addendum (Inches)
3	1.0472	.5236	.7190	.3333
4	.7854	.3927	.5393	.2500
5	.6283	.3142	.4314	.2000
6	.5236	.2618	.3565	.1667
8	.3927	.1963	.2696	.1250
10	.3142	.1571	.2157	.1000
12	.2618	.1309	.1798	.0833
16	.1963	.0982	.1348	.0625
20	.1571	.0785	.1120	.0500
24	.1309	.0654	.0937	.0417
32	.0982	.0491	.0708	.0312
48	.0654	.0327	.0478	.0208
64	.0491	.0245	.0364	.0156

SPUR GEARS

BACKLASH

Stock spur gears are cut to operate at standard center distances. The standard center distance being defined by:

$$\text{Standard Center Distance} = \frac{\text{Pinion PD} + \text{Gear PD}}{2}$$

When mounted at this center distance, stock spur gears will have the following average backlash:

Diametral Pitch	Backlash (Inches)	Diametral Pitch	Backlash (Inches)
3	.013	8-9	.005
4	.010	10-13	.004
5	.008	14-32	.003
6	.007	33-64	.0025
7	.006		

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. Varying the center distance may afford a practical means of varying the backlash to a limited extent.

The approximate relationship between center distance and backlash change of 14-1/2° and 20° pressure angle gears is shown below:

For 14-1/2°–Change in Center Distance = 1.933 x Change in Backlash
For 20° –Change in Center Distance = 1.374 x Change in Backlash

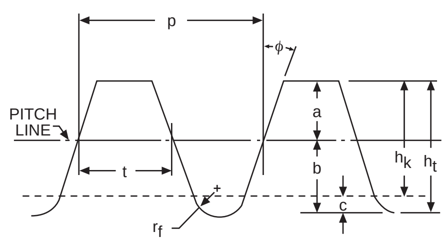
From this, it is apparent that a given change in center distance, 14-1/2° gears will have a smaller change in backlash than 20° gears. This fact should be considered in cases where backlash is critical.

UNDERCUT

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.

On 14-1/2°PA gears undercutting occurs where a number of teeth is less than 32 and for 20°PA less than 18. Since this condition becomes more severe as tooth numbers decrease, it is recommended that the minimum number of teeth be 16 for 14-1/2°PA and 13 for 20°PA.

In a similar manner INTERNAL Spur Gear teeth may interfere when the pinion gear is too near the size of its mating internal gear. The following may be used as a guide to assure proper operation of the gear set. For 14-1/2°PA, the difference in tooth numbers between the gear and pinion should not be less than 15. For 20°PA the difference in tooth numbers should not be less than 12.



a = ADDENDUM
b = DEDENDUM
c = CLEARANCE
h_k = WORKING DEPTH
h_t = WHOLE DEPTH
p = CIRCULAR PITCH
r_f = FILLET RADIUS
t = CIRCULAR TOOTH THICKNESS
φ = PRESSURE ANGLE

SPUR GEAR FORMULAS

FOR FULL DEPTH INVOLUTE TEETH

To Obtain	Having	Formula
Diametral Pitch (P)	Circular Pitch (p)	$P = \frac{3.1416}{p}$
	Number of Teeth (N) & Pitch Diameter (D)	$P = \frac{N}{D}$
	Number of Teeth (N) & Outside Diameter (D_o)	$P = \frac{N + 2}{D_o}$ (Approx.)
Circular Pitch (p)	Diametral Pitch (P)	$p = \frac{3.1416}{P}$
Pitch Diameter (D)	Number of Teeth (N) & Diametral Pitch (P)	$D = \frac{N}{P}$
	Outside Diameter (D_o) & Diametral Pitch (P)	$D = D_o - \frac{2}{P}$
Base Diameter (D_b)	Pitch Diameter (D) and Pressure Angle (ø)	Db = Dcosø
Number of Teeth (N)	Diametral Pitch (P) & Pitch Diameter (D)	N = P x D
Tooth Thickness (t) @Pitch Diameter (D)	Diametral Pitch (P)	$t = \frac{1.5708}{P}$
Addendum (a)	Diametral Pitch (P)	$a = \frac{1}{P}$
Outside Diameter (D_o)	Pitch Diameter (D) & Addendum (a)	$D_o = D + 2a$
Whole Depth (h_t) (20P & Finer)	Diametral Pitch (P)	$h_t = \frac{2.2}{P} + .002$
Whole Depth (h_t) (Courser than 20P)	Diametral Pitch (P)	$h_t = \frac{2.157}{P}$
Working Depth (h_k)	Addendum (a)	$h_k = 2(a)$
Clearance (c)	Whole Depth (h_t) Addendum (a)	$c = h_t - 2a$
Dedendum (b)	Whole Depth (h_t) & Addendum (a)	$b = h_t - a$
Contact Ratio (M_c)	Outside Radii, Base Radii, Center Distance and Pressure Angle+C.P.	$M_c = \frac{\sqrt{R_o^2 - R_b^2} + \sqrt{r_o^2 - r_b^2} - C\sin ø^*}{p \cos ø}$
Root Diameter (D_r)	Pitch Diameter (D) and Dedendum (b)	$D_r = D - 2b$
Center Distance (C)	Pitch Diameter (D) or No. of Teeth and Pitch	$C = \frac{D_1 + D_2}{2}$ or $\frac{N_1 + N_2}{2P}$

$^*R_o$ = Outside Radius, Gear
r_o = Outside Radius, Pinion
R_b = Base Circle Radius, Gear
r_b = Base Circle Radius, Pinion

SPUR GEARS

LEWIS FORMULA (Barth Revision)

Gear failure can occur due to tooth breakage (tooth stress) or surface failure (surface durability) as a result of fatigue and wear. Strength is determined in terms of tooth-beam stresses for static and dynamic conditions, following well established formula and procedures. Satisfactory results may be obtained by the use of Barth's Revision to the Lewis Formula, which considers beam strength but not wear. The formula is satisfactory for commercial gears at Pitch Circle velocities of up to 1500 FPM. It is this formula that is the basis for all Boston Spur Gear ratings.

METALLIC SPUR GEARS

$$W = \frac{SFY}{P}\left(\frac{600}{600 + V}\right)$$

W = Tooth Load, Lbs. (along the Pitch Line)
S = Safe Material Stress (static) Lbs. per Sq. In. (Table II)
F = Face Width, In.
Y = Tooth Form Factor (Table I)
P = Diametral Pitch
D = Pitch Diameter
V = Pitch Line Velocity, Ft. per Min. = .262 x D x RPM

For NON-METALLIC GEARS, the modified Lewis Formula shown below may be used with (S) values of 6000 PSI for Phenolic Laminated material.

$$W = \frac{SFY}{P}\left(\frac{150}{200 + V} + .25\right)$$

TABLE II–VALUES OF SAFE STATIC STRESS (s)

Material		(s) Lb. per Sq. In.
Plastic		5000
Bronze		10000
Cast Iron		12000
Steel	.20 Carbon (Untreated)	20000
Steel	.20 Carbon (Case-hardened)	25000
Steel	.40 Carbon (Untreated)	25000
Steel	.40 Carbon (Heat-treated)	30000
Steel	.40 C. Alloy (Heat-treated)	40000

Max. allowable torque (T) that should be imposed on a gear will be the safe tooth load (W) multiplied by $\frac{D}{2}$ or $T = \frac{W \times D}{2}$

The safe horsepower capacity of the gear (at a given RPM) can be calculated from $HP = \frac{T \times RPM}{63{,}025}$ or directly from (W) and (V);

$$HP = \frac{WV}{33{,}000}$$

For a known HP, $T = \frac{63025 \times HP}{RPM}$

TABLE I TOOTH FORM FACTOR (Y)

Number of Teeth	14-1/2° Full Depth Involute	20° Full Depth Involute
10	0.176	0.201
11	0.192	0.226
12	0.210	0.245
13	0.223	0.264
14	0.236	0.276
15	0.245	0.289
16	0.255	0.295
17	0.264	0.302
18	0.270	0.308
19	0.277	0.314
20	0.283	0.320
22	0.292	0.330
24	0.302	0.337
26	0.308	0.344
28	0.314	0.352
30	0.318	0.358
32	0.322	0.364
34	0.325	0.370
36	0.329	0.377
38	0.332	0.383
40	0.336	0.389
45	0.340	0.399
50	0.346	0.408
55	0.352	0.415
60	0.355	0.421
65	0.358	0.425
70	0.360	0.429
75	0.361	0.433
80	0.363	0.436
90	0.366	0.442
100	0.368	0.446
150	0.375	0.458
200	0.378	0.463
300	0.382	0.471
Rack	0.390	0.484

HELICAL GEARS

GEAR NOMENCLATURE

The information contained in the Spur Gear section is also pertinent to Helical Gears with the addition of the following:

HELIX ANGLE (ψ) is the angle between any helix and an element of its cylinder. In helical gears, it is at the pitch diameter unless otherwise specified.

LEAD (L) is the axial advance of a helix for one complete turn, as in the threads of cylindrical worms and teeth of helical gears.

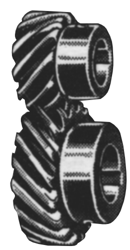
NORMAL DIAMETRAL PITCH (P_n) is the Diametral Pitch as calculated in the normal plane.

HAND – Helical Gears of the same hand operate at right angles, see Fig. 1
Helical Gears of opposite hands run on parallel shafts. Fig. 2

TWO RIGHT-HAND HELICAL GEARS

TWO LEFT-HAND HELICAL GEARS

Figure 1



LEFT-HAND AND RIGHT-HAND HELICAL GEARS

Figure 2

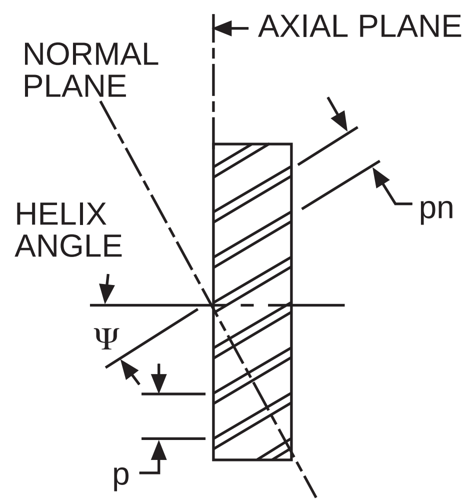
LEFT HAND HELICAL GEAR

The teeth of a LEFT HAND Helical Gear lean to the left when the gear is placed flat on a horizontal surface.

RIGHT HAND HELICAL GEAR

The teeth of a RIGHT HAND Helical Gear lean to the right when the gear is placed flat on a horizontal surface.

HELIX ANGLE—



p = AXIAL CIRCULAR PITCH
pn = NORMAL CIRCULAR PITCH

All Boston Helicals are cut to the Diametral Pitch system, resulting in a Normal Pitch which is lower in number than the Diametral Pitch.

INVOLUTE—The Helical tooth form is involute in the plane of rotation and can be developed in a manner similar to that of the Spur Gear. However, unlike the Spur Gear, which may be viewed as two-dimensional, the Helical Gear must be viewed as three-dimensional to show change in axial features.

Helical gears offer additional benefits relative to Spur Gears, those being:

- Improved tooth strength due to the elongated helical wrap-around.
- Increased contact ratio due to the axial tooth overlap.
- Helical Gears thus tend to have greater load carrying capacity than Spur Gears of similar size.
- Due to the above, smoother operating characteristics are apparent.

HELICAL GEARS

HELICAL GEAR FORMULAS

To Obtain	Having	Formula
Transverse Diametral Pitch (P)	Number of Teeth (N) & Pitch Diameter (D)	$P = \frac{N}{D}$
	Normal Diametral Pitch (P_n) Helix Angle (ψ)	$P = P_N \cos\psi$
Pitch Diameter (D)	Number of Teeth (N) & Transverse Diametral Pitch (P)	$D = \frac{N}{P}$
Normal Diametral Pitch (P_N)	Transverse Diametral Pitch (P) & Helix Angle (ψ)	$P_N = \frac{P}{\cos\psi}$
Normal Circular Tooth Thickness (τ)	Normal Diametral Pitch (P_N)	$\tau = \frac{1.5708}{P_N}$
Transverse Circular Pitch (p_t)	Diametral Pitch (P) (Transverse)	$p_t = \frac{\pi}{P}$
Normal Circular Pitch (p_n)	Transverse Circular Pitch (p)	$p_n = p_t \cos\psi$
Lead (L)	Pitch Diameter and Pitch Helix Angle	$L = \frac{\pi D}{\tan\psi}$

TRANSVERSE VS. NORMAL DIAMETRAL PITCH FOR BOSTON 45° HELICAL GEARS

P Transverse Diametral Pitch	P_N Normal Diametral Pitch
24	33.94
20	28.28
16	22.63
12	16.97
10	14.14
8	11.31
6	8.48

HELICAL GEAR LEWIS FORMULA

The beam strength of Helical Gears operating on *parallel shafts* can be calculated with the Lewis Formula revised to compensate for the difference between Spur and Helical Gears, with modified Tooth Form Factors Y.

$$W = \frac{SFY}{P_N}\left(\frac{600}{600 + V}\right)$$

W = Tooth Load, Lbs. (along the Pitch Line)
S = Safe Material Stress (static) Lbs. per Sq. In. (Table III)
F = Face Width, Inches
Y = Tooth Form Factor (Table IV)
P_N = Normal Diametral Pitch
(Refer to Conversion Chart)
D = Pitch Diameter
V = Pitch Line Velocity, Ft. Per Min. = .262 x D x RPM

TABLE III–VALUES OF SAFE STATIC STRESS (S)

Material		(s) Lb. per Sq. In.
Bronze		10000
Cast Iron		12000
Steel	.20 Carbon (Untreated)	20000
Steel	.20 Carbon (Case-hardened)	25000
Steel	.40 Carbon (Untreated)	25000
Steel	.40 Carbon (Heat-treated)	30000
Steel	.40 C. Alloy (Heat-treated)	40000

TABLE IV—VALUES OF TOOTH FORM FACTOR (Y)

FOR 14-1/2°PA—45° HELIX ANGLE GEAR			
No. of Teeth	Factor Y	No. of Teeth	Factor Y
8	.295	25	.361
9	.305	30	.364
10	.314	32	.365
12	.327	36	.367
15	.339	40	.370
16	.342	48	.372
18	.345	50	.373
20	.352	60	.374
24	.358	72	.377

HORSEPOWER AND TORQUE

Max. allowable torque (T) that should be imposed on a gear will be the safe tooth load (W) multiplied by $\frac{D}{2}$ or $T = \frac{W \times D}{2}$

The safe horsepower capacity of the gear (at a given RPM) can be calculated from $HP = \frac{T \times RPM}{63{,}025}$ or directly from (W) and (V);

$$HP = \frac{WV}{33{,}000}$$

For a known HP, $T = \frac{63025 \times HP}{RPM}$

HELICAL GEARS

When Helical gears are operated on other than Parallel shafts, the tooth load is concentrated at a point, with the result that very small loads produce very high pressures. The sliding velocity is usually quite high and, combined with the concentrated pressure, may cause galling or excessive wear, especially if the teeth are not well lubricated. For these reasons, the tooth load which may be applied to such drives is very limited and of uncertain value, and is perhaps best determined by trial under actual operating conditions. If one of the gears is made of bronze, the contact area and thereby the load carrying capacity, may be increased, by allowing the gears to "run-in" in their operating position, under loads which gradually increase to the maximum expected.

THRUST LOADS

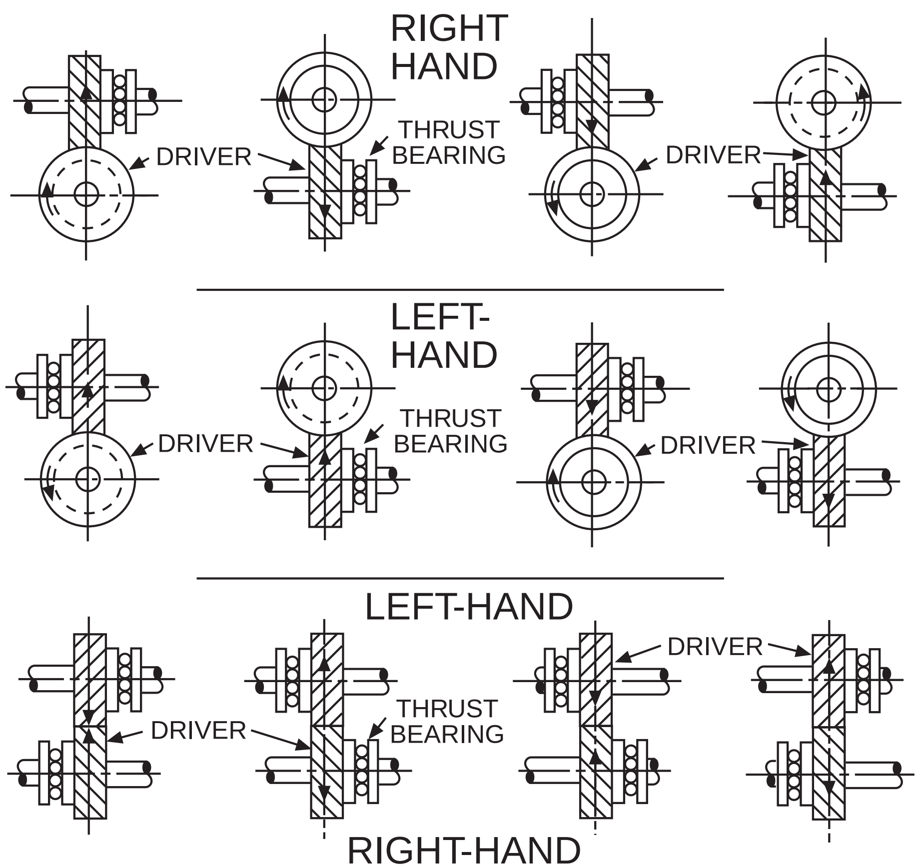
As a result of the design of the Helical Gear tooth, an axial or thrust load is developed. Bearings must be adequate to absorb this load. The thrust load direction is indicated below. The magnitude of the thrust load is based on calculated Horsepower.

$$\text{Axial Thrust Load} = \frac{126{,}050 \times \text{HP}}{\text{RPM} \times \text{Pitch Diameter}}$$

Boston Helicals are all 45° Helix Angle, producing a tangential force equal in magnitude to the axial thrust load. A separating force is also imposed on the gear set based on calculated Horsepower.

Separating Load = Axial Thrust Load x .386

Above formulae based on Boston 45° Helix Angle and 14-1/2° Normal Pressure Angle.



See page 118 for hardened and ground Thrust Washers.

ENGINEERING INFORMATION

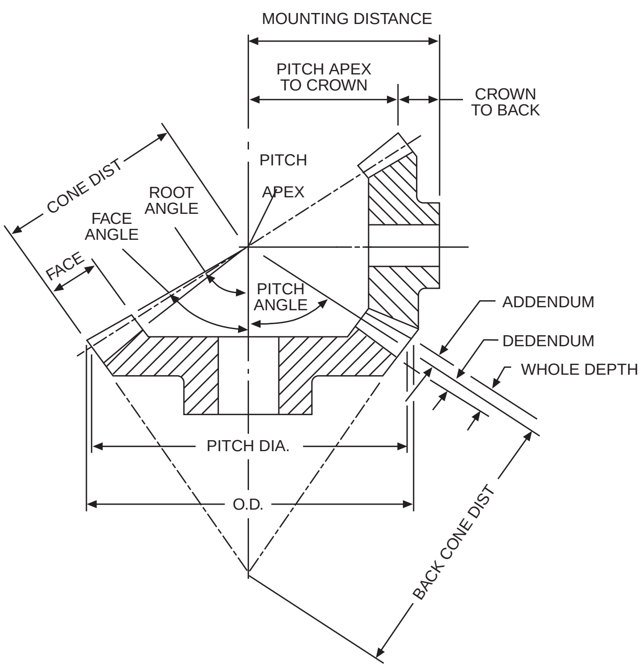
MITER AND BEVEL GEARS

Gear geometry for both straight and spiral tooth Miter and Bevel gears is of a complex nature and this text will not attempt to cover the topic in depth.

The basic tooth form is a modification to the involute form and is the common form used in production today. All Boston standard stock Miter and Bevel gears are manufactured with a 20° Pressure Angle. Bevel gears are made in accordance with A.G.M.A. specifications for long and short Addendum system for gears and pinions (pinion is cut long Addendum) which serves to reduce the amount of pinion tooth undercut and to nearly equalize the strength and durability of the gear set.

NOMENCLATURE

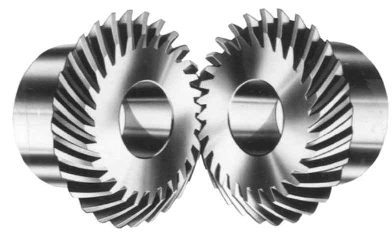
Nomenclature may best be understood by means of graphic representation depicted below:



Stock gears are cut to operate on an exact Mounting Distance with the following average backlash:

Diametral Pitch	Backlash (Inches)
4	.008
5	.007
6	.006
8	.005
10	.004
12-20	.003
24-48	.002

Similar in nature to Helical gearing, Spiral Miters and Bevels must be run with a mating pinion or gear of opposite hand.



The teeth of a Left Hand gear lean to the left when the gear is placed on a horizontal surface.

The teeth of a Right Hand gear lean to the right when the gear is placed flat on a horizontal surface.

All Boston Spiral Miter and Bevel gears are made with 35° spiral angles with all pinions cut left hand.

Straight Tooth Miter and Bevel Gear Formulas

To Obtain	Having	Formula: Pinion	Formula: Gear
Pitch Diameter (D,d)	No. of Teeth and Diametral Pitch (P)	$d = \frac{n}{P}$	$D = \frac{n}{P}$
Whole Depth (h_T)	Diametral Pitch (P)	$h_T = \frac{2.188}{P} + .002$	$h_T = \frac{2.188}{P} + .002$
Addendum (a)	Diametral Pitch (P)	$a = \frac{1}{P}$	$a = \frac{1}{P}$
Dedendum (b)	Whole Depth (h_T) & Addendum (a)	$b = h_T - a$	$b = h_T - a$
Clearance	Whole Depth (n_T) & Addendum (a)	$c = h_T - 2a$	$c = h_T - 2a$
Circular Tooth Thickness (τ)	Diametral Pitch (P)	$\tau = \frac{1.5708}{P}$	$\tau = \frac{1.5708}{P}$
Pitch Angle	Number of Teeth In Pinion (N_p) and Gear (N_G)	$L_p = \tan -1\left(\frac{N_p}{N_G}\right)$	$L_G = 90 - L_p$
Outside Diameter (D_o, d_o)	Pinion & Gear Pitch Diameter ($D_p + D_G$) Addendum (a) & Pitch Angle ($L_p + L_G$)	$d_o = D_p + 2a(\cos L_p)$	$D_o = D_G + 2a(\cos L_G)$

MITER AND BEVEL GEARS

Straight tooth bevel (and miter) gears are cut with generated tooth form having a localized lengthwise tooth bearing known as the “Coniflex”® tooth form. The superiority of these gears over straight bevels with full length tooth bearing, lies in the control of tooth contact. The localization of contact permits minor adjustment of the gears in assembly and allows for some displacement due to deflection under operating loads, without concentration of the load on the end of the tooth. This results in increased life and quieter operation.

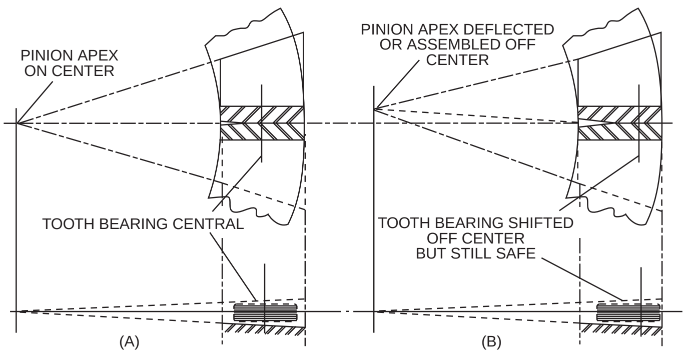


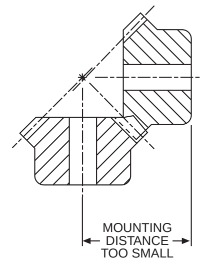
ILLUSTRATION OF LOCALIZED TOOTH BEARING
IN STRAIGHT BEVEL CONIFLEX® GEARS

Boston Gear Bevel and Miter Gears will provide smooth, quiet operation and long life when properly mounted and lubricated. There are several important considerations in mounting these gears.

1. All standard stock bevel and miter gears must be mounted at right angles (90°) for proper tooth bearing.
2. Mounting Distance (MD) is the distance from the end of the hub of one gear to the center line of its mating gear. When mounted at the MD specified, the gears will have a proper backlash and the ends of the gear teeth will be flush with each other (see drawings).
3. All bevel and miter gears develop radial and axial thrust loads when transmitting power. See page 148. These loads must be accommodated by the use of bearings.

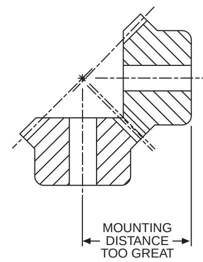
Incorrect

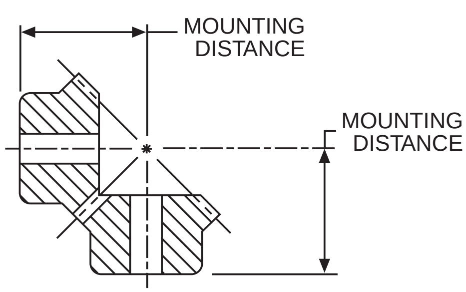
If Mounting Distance of one or both gears is made less than dimension specified, the teeth may bind. Excessive wear or breakage can result. Drawing below shows gears mounted incorrectly with the Mounting Distance too short for one gear.

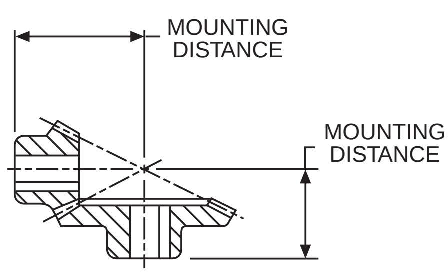


Incorrect

If Mounting Distance of either gear is made longer than dimension specified, as shown in drawing below, the gears will not be in full mesh on a common pitch line and may have excessive backlash. Excessive backlash or play, if great enough, can cause a sudden impulse or shock load in starting or reversing which might cause serious tooth damage.







®Registered in the U.S. Patent Office.

MITER AND BEVEL BEARS
TOOTH STRENGTH (Straight Tooth)

The beam strength of Miter and Bevel gears (straight tooth) may be calculated using the Lewis Formula revised to compensate for the differences between Spur and Bevel gears. Several factors are often combined to make allowance for the tooth taper and the normal overhung mounting of Bevel gears.

$$W = \frac{SFY}{P}\left(\frac{600}{600+V}\right).75$$

W = Tooth Load, Lbs. (along the Pitch Line)
S = Safe Material Stress (static) Lbs. per Sq. In. (Table 1)
F = Face Width, In.
Y = Tooth Form Factor (Table I)
P = Diametral Pitch
D = Pitch Diameter
V = Pitch Line Velocity, Ft. per Min. = .262 x D x RPM

TABLE I VALUES OF SAFE STATIC STRESS (s)

Material		(s) Lb. per Sq. In.
Plastic		5000
Bronze		10000
Cast Iron		12000
Steel	.20 Carbon (Untreated)	20000
	.20 Carbon (Case-hardened)	25000
	.40 Carbon (Untreated)	25000
	.40 Carbon (Heat-treated)	30000
	.40 C. Alloy (Heat-treated)	40000

TABLE II TOOTH FORM FACTOR (Y)

20°P.A.—LONG ADDENDUM PINIONS SHORT ADDENDUM GEARS

No. Teeth	Ratio 1		1.5		2		3		4		6	
Pinion	Pin.	Gear	Pin.	Gear	Pin.	Gear	Pin.	Gear	Pin.	Gear	Pin.	Gear
12	—		—	—	.345	.283	.355	.302	.358	.305	.361	.324
14	—		.349	.292	.367	.301	.377	.317	.380	.323	.405	.352
16	.333		.367	.311	.386	.320	.396	.333	.402	.339	.443	.377
18	.342		.383	.328	.402	.336	.415	.346	.427	.364	.474	.399
20	.352		.402	.339	.418	.349	.427	.355	.456	.386	.500	.421
24	.371		.424	.364	.443	.368	.471	.377	.506	.405	—	—
28	.386		.446	.383	.462	.386	.509	.396	.543	.421	—	—
32	.399		.462	.396	.487	.402	.540	.412	—	—	—	—
36	.408		.477	.408	.518	.415	.569	.424	—	—	—	—
40	.418		—	—	.543	.424	.594	.434	—	—	—	—

HORSEPOWER AND TORQUE

Max. allowable torque (T) that should be imposed on a gear will be the safe tooth load (W) multiplied by $\frac{D}{2}$ or $T = \frac{W \times D}{2}$

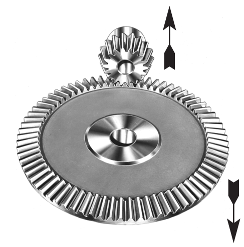
The safe horsepower capacity of the gear (at a given RPM) can be calculated from $HP = \frac{T \times RPM}{63{,}025}$ or directly from (W) and (V);

$$HP = \frac{WV}{33{,}000}$$

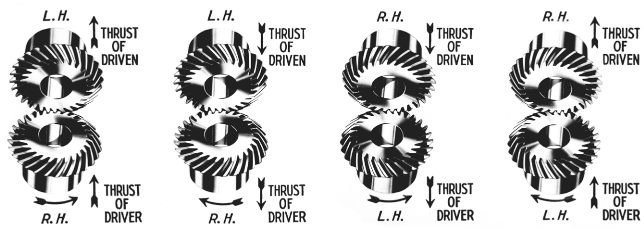
For a known HP, $T = \frac{63025 \times HP}{RPM}$

THRUST

The axial thrust loads developed by straight tooth miter and bevel gears always tend to separate the gears.



For Spiral Bevel and Miter Gears, the direction of axial thrust loads developed by the driven gears will depend upon the hand and direction of rotation. Stock Spiral Bevel pinions cut Left Hand only, Gears Right Hand only.



The magnitude of the thrust may be calculated from the formulae below, based on calculated HP, and an appropriate Thrust Bearing selected.

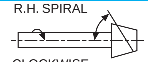
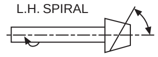
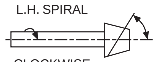
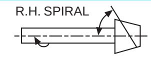
Straight Bevels and Miters

$$\text{Gear Thrust} = \frac{126{,}050 \times HP}{\text{RPM} \times \text{Pitch Diameter}} \times \tan\alpha\cos\beta$$

$$\text{Pinion Thrust} = \frac{126{,}050 \times HP}{\text{RPM} \times \text{Pitch Diameter}} \times \tan\alpha\sin\beta$$

Spiral Bevels and Miters

Thrust values for Pinions and Gears are given for four possible combinations.

R.H. SPIRAL  CLOCKWISE	$T_P = \frac{126{,}050 \times HP}{RPM \times D}\left(\frac{\tan\alpha\sin\beta}{\cos\gamma} - \tan\gamma\cos\beta\right)$
L.H. SPIRAL  C. CLOCKWISE	$T_G = \frac{126{,}050 \times HP}{RPM \times D}\left(\frac{\tan\alpha\cos\beta}{\cos\gamma} + \tan\gamma\sin\beta\right)$
L.H. SPIRAL  CLOCKWISE	$T_P = \frac{126{,}050 \times HP}{RPM \times D}\left(\frac{\tan\alpha\sin\beta}{\cos\gamma} + \tan\gamma\cos\beta\right)$
R.H. SPIRAL  C. CLOCKWISE	$T_G = \frac{126{,}050 \times HP}{RPM \times D}\left(\frac{\tan\alpha\cos\beta}{\cos\gamma} + \tan\gamma\sin\beta\right)$

α = Tooth Pressure Angle

β = 1/2 Pitch Angle

$$\text{Pitch Angle} = \tan^{-1}\left(\frac{N_P}{N_G}\right)$$

γ = Spiral Angle = 35°

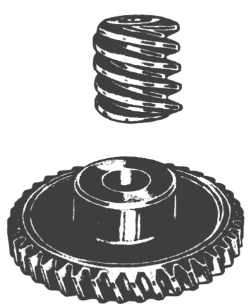
WORMS AND WORM GEARS

Boston standard stock Worms and Worm Gears are used for the transmission of motion and/or power between non-intersecting shafts at right angles (90°). Worm Gear drives are considered the smoothest and quietest form of gearing when properly applied and maintained. They should be considered for the following requirements:

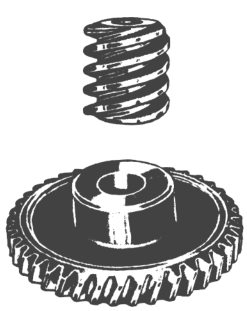
- HIGH RATIO SPEED REDUCTION
- LIMITED SPACE
- RIGHT ANGLE (NON-INTERSECTING) SHAFTS
- GOOD RESISTANCE TO BACK DRIVING

General nomenclature having been applied to Spur and Helical gear types, may also be applied to Worm Gearing with the addition of Worm Lead and Lead Angle, Number of Threads (starts) and Worm Gear Throat diameter.

HOW TO TELL A LEFT-HAND OR RIGHT-HAND WORM OR WORM GEAR



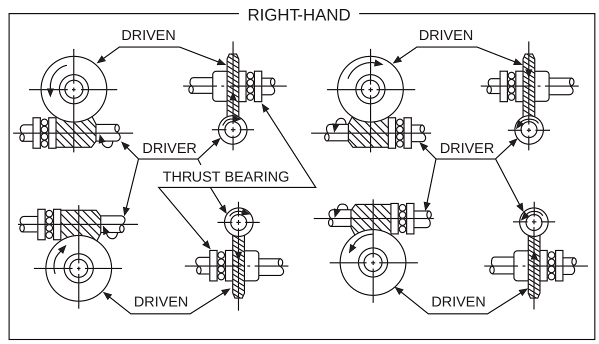
Threads of LEFT-HAND lean to the Left when standing on either end:

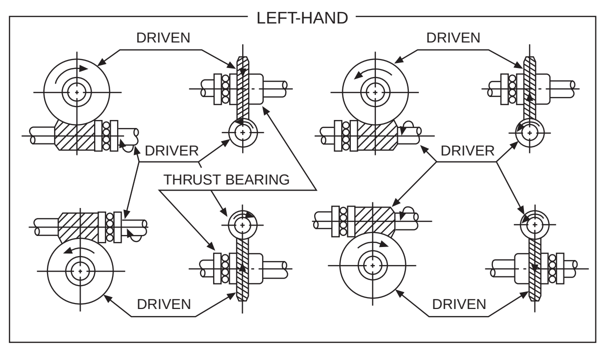


Threads of RIGHT-HAND lean to the Right when standing on either end:

THRUST LOADS

As is true with Helical and Bevel gearing, Worm gearing, when operating, produces Thrust loading. The Chart below indicates the direction of thrust of Worms and Worm Gears when they are rotated as shown. To absorb this thrust loading, bearings should be located as indicated.





EFFICIENCY

The efficiency of a worm gear drive depends on the lead angle of the worm. The angle decreases with increasing ratio and worm pitch diameter. For maximum efficiency the ratio should be kept low.

Due to the sliding action which occurs at the mesh of the Worm and Gear, the efficiency is dependent on the Lead Angle and the Coefficient of the contacting surface. A common formula for estimating efficiency of a given Worm Gear reduction is:

$$\text{EFFICIENCY} = E = \frac{\tan\gamma\,(1 - f\tan\gamma)}{f + \tan\gamma}$$

where γ = Worm Lead Angle
f = Coefficient of Friction

For a Bronze Worm Gear and hardened Steel Worm, a Coefficient of Friction in the range of .03/.05 may be assumed for estimated value only.

WORMS AND WORM GEARS

WORM AND WORM GEAR FORMULAS

To Obtain	Having	Formula
Circular Pitch (p)	Diametral Pitch (P)	$p = \frac{3.1416}{P}$
Diametral Pitch (P)	Circular Pitch (p)	$P = \frac{3.1416}{p}$
Lead (of Worm) (L)	Number of Threads in Worm & Circular Pitch (p)	L = p(No. of Threads)
Addendum (a)	Diametral Pitch (P)	$a = \frac{1}{P}$
Pitch Diameter (D) of Worm (D_W)	Outside Diameter (d_o) & Addendum (a)	$D_W = d_o - 2a$
Pitch Diameter of Worm Gear (D_G)	Circular Pitch (p) & Number of Teeth (N)	$D_G = \frac{N_{Gp}}{3.1416}$
Center Distance Between Worm & Worm Gear (CD)	Pitch Diameter of Worm (d_w) & Worm Gear (D_G)	$CD = \frac{d_w + D_G}{2}$
Whole Depth of Teeth (h_T)	Circular Pitch (p)	$h_T = .6866\ p$
	Diametral Pitch (P)	$h_T = \frac{2.157}{P}$
Bottom Diameter of Worm (D_r)	Whole Depth (h_T) & Outside Diameter (d_w)	$d_r = d_o - 2h_T$
Throat Diameter of Worm Gear (D_T)	Pitch Diameter of Worm Gear (D) & Addendum (a)	$D_T = D_G + 2a$
Lead Angle of Worm (γ)	Pitch Diameter of Worm(D) & The Lead (L)	$\gamma = \tan^{-1}\left(\frac{L}{3.1416d}\right)$
Ratio	No. of Teeth on Gear (N_G) and Number of Threads on Worm	$\text{Ratio} = \frac{N_G}{\text{No. of Threads}}$
Gear O.D. (D_O)	Throat Dia. (D_T) and Addendum (a)	$D_O = D_T + .6a$

SELF-LOCKING ABILITY

There is often some confusion as to the self-locking ability of a worm and gear set. Boston worm gear sets, under no condition should be considered to hold a load when at rest. The statement is made to cover the broad spectrum of variables effecting self-locking characteristics of a particular gear set in a specific application. Theoretically, a worm gear will not back drive if the friction angle is greater than the worm lead angle. However, the actual surface finish and lubrication may reduce this significantly. More important, vibration may cause motion at the point of mesh with further reduction in the friction angle.

Generally speaking, if the worm lead angle is less than 5°, there is reasonable expectation of self-locking. Again, no guarantee should be made and customer should be advised. If safety is involved, a positive brake should be used.

WORM GEAR BACK-DRIVING

This is the converse of self-locking and refers to the ability of the worm gear to drive the worm. The same variables exist, making it difficult to predict. However, our experience indicates that for a hardened worm and bronze gear properly manufactured, mounted and lubricated, back-driving capability may be expected, if the lead angle is greater than 11°. Again, no guarantee is made and the customer should be so advised.

RATING

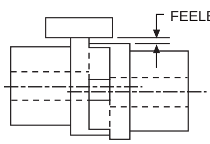
The high rate of sliding friction that takes place at the mesh of the Worm and Gear results in a more complex method of rating these Gears as opposed to the other Gear types. Material factors, friction factors and velocity factors must all be considered and applied to reflect a realistic durability rating.

COUPLINGS

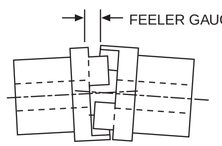
ALIGNMENT

Alignment of Boston couplings should be performed by the following steps to meet lateral and angular misalignment specifications below.

1. Align shafts and supports to give minimum lateral and angular misalignment.
2. Assemble coupling halves to shaft.
3. Slide couplings together and check lateral misalignment using straight edge and feeler gauge over coupling outside diameter (On BF Series couplings, spider must be removed.) This should be within specifications below.
4. Lock couplings on shaft and check distance using feeler gauges between drive lug on one half and space between on other coupling half. Rotate coupling and check gap at a minimum of 3 other coupling positions. The difference between any two readings should be within specifications below.



LATERAL MISALIGNMENT



ANGULAR MISALIGNMENT

MISALIGNMENT TOLERANCES

Coupling Series	Lateral	Angular
FC—Bronze Insert FC—Urethane Insert FC—Rubber Insert	.001 .002 .002	See Chart below
BF	.002	1-1/2°
BG (Shear Type)	1/32	2°
FA	.002	2°
FCP (Plastic)	.003	3°

FC SERIES ANGULAR MISALIGNMENT

Chart reflects maximum angular misalignment of 1-1/2° for rubber, 1° for urethane and 1/2° for bronze.

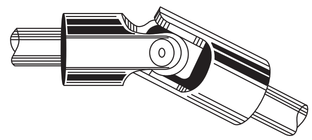
MAXIMUM READING DIFFERENTIAL

Size	Rubber	Insert Urethane	Bronze
FC12	.033	.022	.011
FC15	.039	.026	.013
FC20	.053	.035	.018
FC25	.066	.044	.022
FC30	.078	.052	.026
FC38	.097	.065	.032
FC45	.117	.078	.039

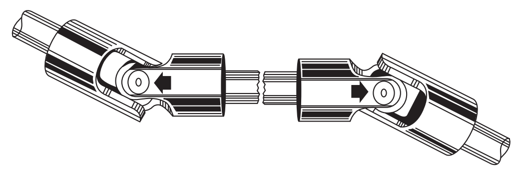
UNIVERSAL JOINTS

MOUNTING

A single universal joint (rotating at uniform speed) operating at an angle will introduce periodic variations of angular velocity to the driven shaft. These cyclic speed fluctuations (two per revolution) cause vibration, higher shaft stresses and bearing loads which will be more severe with larger angles of operation.



The detrimental effects of these rotational deviations can be reduced, and uniform speed restored by using two joints (and an intermediate shaft) to connect shafts at an angle or misaligned in a parallel direction.



For connecting shafts in the same plane the joints should be arranged to operate at equal angles and with the bearing pins of the yokes on the intermediate shaft in line with each other.

LUBRICATION

PIN and BLOCK TYPE

These universal joints are not lubricated when shipped.

Many applications are considered severe when in harsh environments and when a combination of speed, dirt contamination and inaccessible locations make it impractical to maintain proper lubrication.

It is in these instances when the Boot Kits become a desirable alternative. For satisfactory performance, all booted joints should be used with a LITH-EP-000 grease for an ambient temperature range of 40° to 225°F.

VOLUME OF LUBRICATION FOR BOOTED JOINTS

Size	Volume (Ozs.)	Size	Volume (Ozs.)	Size	Volume (Ozs.)
37	.4	100	2.0	250	25.0
50	.5	125	3.5	300	30.0
62	.75	150	4.5	400	50.1
75	1.0	175	7.0		
87	1.5	200	15.0		

Note: Joints should be initially lubricated with a 90 weight oil before being packed with grease.

FORGED AND CAST TYPE

Universal Joints are not lubricated when shipped.
Lubricate these joints with a Lith EP-2 grease or equivalent. The center cross of these joints holds a generous supply of lubricant which is fed to the bearings by centrifugal action. Light-duty, low-angle operation may require only occasional lubrication. For high-angle, high-speed operation or in extreme dirt or moist conditions, daily regreasing may be required.

GENERAL

MOUNTING

SPUR & HELICAL

For proper functioning gears, gears must be accurately aligned and supported by a shaft and bearing system which maintains alignment under load. Deflection should not exceed .001 inch at the tooth mesh for general applications. The tolerance on Center Distance normally should be positive to avoid possibility of gear teeth binding. Tolerance value is dependent on acceptable system backlash. As a guide for average application, this tolerance might vary from .002 for Boston Gear's fine pitch gears to .005 for the coarsest pitch.

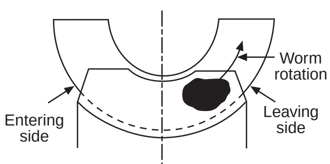
WORMS AND WORM GEAR

It is important that the mounting assures the central plane of the Worm gear passes essentially through the axis of the Worm. This can be accomplished by adjusting the Worm Gear axially. Boston Worm Gears are cut to close tolerancing of the Center Line of the Gear tooth to the flush side of the Gear. When properly mounted Worm Gears will become more efficient after initial break-in period.

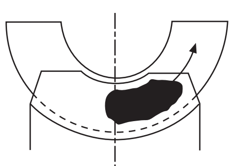
HOW WORM GEARS "ADJUST" THEMSELVES

The gear in a worm gear reducer is made of a soft bronze material. Therefore, it can cold-work and wear-in to accommodate slight errors in misalignment.

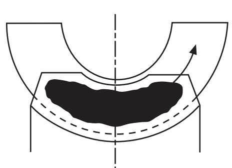
Evolution of Contact in a Worm Gear



Initially, contact is concentrated on the leaving side of the worm.



After several hours or running under load, gear has cold-worked to spread area of contact.



After many hours of operation, contact has spread to cover the entire working area of the tooth.

ALTERATIONS

Boston Gear Service Centers are equipped to alter catalog sprockets (rebore, keyway, setscrew, etc.). For customers, choosing to make their own alterations, the guidelines listed below should be beneficial. Alterations to hardened gears should not be made without consultation with factory.

In setting up for reboring the most important consideration is to preserve the accuracy of concentricity and lateral runout provided in the original product. There are several methods for accomplishing this. One procedure is: mount the part on an arbor, machine hub diameter to provide a true running surface, remove from arbor and chuck on the hub diameter, check face and bore runout prior to reboring. As a basic rule of thumb, the maximum bore should not exceed 60% of the Hub Diameter and depending on Key size should be checked for minimum wall thickness. A minimum of one setscrew diameter over a keyway is considered adequate.

Boston Gear offers a service for hardening stock sprockets. This added treatment can provide increased horsepower capacity with resultant longer life and/or reduction in size and weight.

Customers wishing to do the hardening operation should refer to "Materials" below for information.

LUBRICATION

The use of a straight mineral oil is recommended for most worm gear applications. This type of oil is applicable to gears of all materials, including non-metallic materials.

Mild E.P. (Extreme Pressure) lubricants may be used with Iron and Steel Gears. E.P. lubricants normally should be selected of the same viscosity as straight mineral oil, E.P. lubricants are not recommended for use with brass or bronze gears.

SAE80 or 90 gear oil should be satisfactory for splash lubricated gears. Where extremely high or low speed conditions are encountered, consult a lubricant manufacturer. Oil temperature of 150°F should not be exceeded for continuous duty applications. Temperatures up to 200°F can be safely tolerated for short periods of time.

Many specialty lubricants have been recently developed to meet the application demands of today's markets, including synthetics and both high and low temperature oils and greases. In those instances where Bath or Drip Feed is not practical, a moly-Disulphide grease may be used successfully, for low speed applications.

ENGINEERING INFORMATION

GENERAL

MATERIALS

Boston Gear stock steel gears are made from a .20 carbon steel with no subsequent treatment. For those applications requiring increased wearability. Case-hardening produces a wear resistant, durable surface and a higher strength core. Carburizing and hardening is the most common process used. Several proprietary nitriding processes are available for producing an essentially distortion-free part with a relatively shallow but wear-resistant case. Boston stock worms are made of either a .20 or .45 carbon steel. Selection of material is based on size and whether furnished as hardened or untreated.

Stock cast iron gears are manufactured from ASTM-CLASS 30 cast iron to Boston Gear specifications. This provides a fine-grained material with good wear-resistant properties.

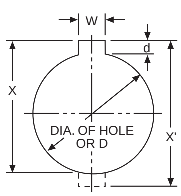
Bronze worm and helical gears are produced from several alloys selected for bearing and strength properties. Phosphor bronze is used for helicals and some worm gears (12P and coarser). Finer pitch worm gears are made from several different grades of bronze, dependent on size.

Non-metallic spur Gears listed in this Catalog are made from cotton reinforced phenolic normally referred to as Grade “C.”

Plastic Gears listed are molded from either Delrin®, Acetal or Minlon®.

STANDARD KEYWAYS AND SETSCREWS

Diameter of Hole	Standard W	Standard d	Recommended Setscrew
5/16 to 7/16”	3/32”	3/64”	10-32
1/2 to 9/16	1/8	1/16	1/4-20
5/8 to 7/8	3/16	3/32	5/16-18
15/16 to 1-1/4	1/4	1/8	3/8-16
1-5/16 to 1-3/8	5/16	5/32	7/16-14
1-7/16 to 1-3/4	3/8	3/16	1/2-13
1-13/16 to 2-1/4	1/2	1/4	9/16-12
2-5/16 to 2-3/4	5/8	5/16	5/8-11
2-13/16 to 3-1/4	3/4	3/8	3/4-10
3-5/16 to 3-3/4	7/8	7/16	7/8-9
3-13/16 to 4-1/2	1	1/2	1-8
4-9/16 to 5-1/2	1-1/4	7/16	1-1/8-7
5-9/16 to 6-1/2	1-1/2	1/2	1-1/4-6



FORMULA:

$X = \sqrt{(D/2)^2 - (W/2)^2} + d + D/2$

$X' = 2X - D$

EXAMPLE:

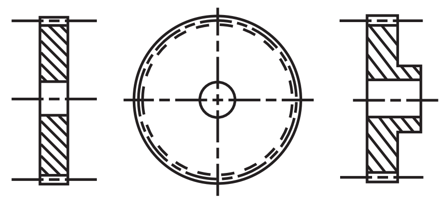
Hole 1”; Keyway 1/4” wide by 1/8” deep.

$X = \sqrt{(1/2)^2 - (1/8)^2} + 1/8 + 1/2 =$ **1.109”**

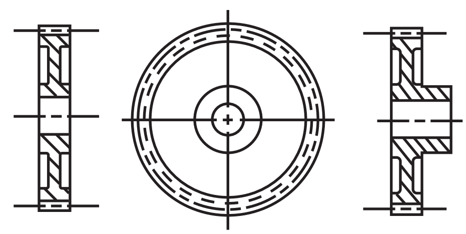
X’ = 2.218 – 1.000 = **1.218”**

STYLES

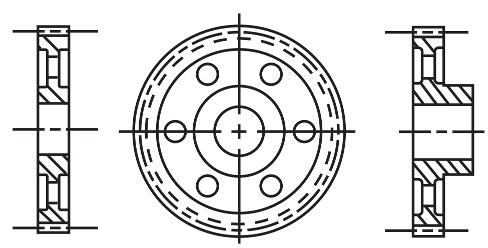
Boston Spur, Helical, and Worm Gears are carried in Plain, Web, or Spoke styles, as illustrated.



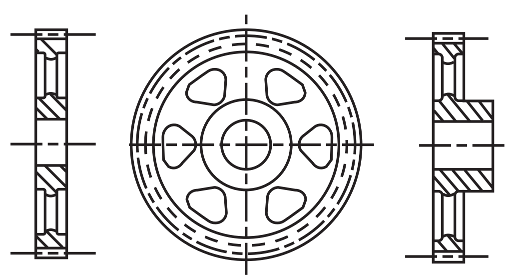
PLAIN – A



WEB – B



WEB WITH
LIGHTNING HOLES – C



SPOKE – D

ENGINEERING INFORMATION

HOW TO FIGURE HORSEPOWER AND TORQUE

TO OBTAIN	HAVING	FORMULA
Velocity (V) Feet Per Minute	Pitch Diameter (D) of Gear or Sprocket – Inches & Rev. Per Min. (RPM)	V = .2618 x D x RPM
Rev. Per Min. (RPM)	Velocity (V) Ft. Per Min. & Pitch Diameter (D) of Gear or Sprocket—Inches	$RPM = \frac{V}{.2618 \times D}$
Pitch Diameter (D) of Gear or Sprocket — Inches	Velocity (V) Ft. Per Min. & Rev. Per Min. (RPM)	$D = \frac{V}{.2618 \times RPM}$
Torque (T) In. Lbs.	Force (W) Lbs. & Radius (R) Inches	T = W x R
Horsepower (HP)	Force (W) Lbs. & Velocity (V) Ft. Per Min.	$HP = \frac{W \times V}{33000}$
Horsepower (HP)	Torque (T) In. Lbs. & Rev. Per Min. (RPM)	$HP = \frac{T \times RPM}{63025}$
Torque (T) In. Lbs.	Horsepower (HP) & Rev. Per Min. (RPM)	$T = \frac{63025 \times HP}{RPM}$
Force (W) Lbs.	Horsepower (HP) & Velocity (V) Ft. Per Min.	$W = \frac{33000 \times HP}{V}$
Rev. Per Min. (RPM)	Horsepower (HP) & Torque (T) In. Lbs.	$RPM = \frac{63025 \times HP}{T}$

POWER is the rate of doing work.

WORK is the exerting of a FORCE through a DISTANCE. ONE FOOT POUND is a unit of WORK. It is the WORK done in exerting a FORCE OF ONE POUND through a DISTANCE of ONE FOOT.

THE AMOUNT OF WORK done (Foot Pounds) is the FORCE (Pounds) exerted multiplied by the DISTANCE (Feet) through which the FORCE acts.

THE AMOUNT OF POWER used (Foot Pounds per Minute) is the WORK (Foot Pounds) done divided by the TIME (Minutes) required.

$$\text{POWER (Foot Pounds per Minute)} = \frac{\text{WORK (Ft. Lbs.)}}{\text{TIME (Minutes)}}$$

POWER is usually expressed in terms of HORSEPOWER.

HORSEPOWER is POWER (Foot Pounds per Minute) divided by 33000.

$$\text{HORSEPOWER (HP)} = \frac{\text{POWER (Ft. Lbs. per Minute)}}{33000}$$

$$= \frac{\text{WORK (Ft. Pounds)}}{33000 \times \text{TIME (Min.)}}$$

$$= \frac{\text{FORCE (Lbs.)} \times \text{DISTANCE (Feet)}}{33000 \times \text{TIME (Min.)}}$$

$$= \frac{\text{FORCE (Lbs.)} \times \text{DISTANCE (Feet)}}{33000 \times \text{TIME (Min.)}}$$

Cut on Dotted Lines
and Keep for Quick Reference

APPLICATION FORMULAS

1 hp = 36 lb-in. @ 1750 rpm
1 hp = 3 lb-ft. @ 1750 rpm

$hp = \frac{\text{Torque (lb.-in.)} \times \text{rpm}}{63{,}025}$

$hp = \frac{\text{Force (lb)} \times \text{Velocity (ft/min.)}}{33{,}000}$

Velocity (ft/min.) = 0.262 x Dia. (in.) x rpm
Torque (lb.-in) = Force (lb) x Radius (in.)

$\text{Torque (lb.-in.)} = \frac{hp \times 63{,}025}{rpm}$

$\text{Mechanical Efficiency} = \frac{\text{Output hp}}{\text{Input hp}} \times 100\%$

$\text{Output hp} = \frac{\text{OT (lb-in.)} \times \text{Output rpm}}{63{,}025}$

OT = Input Torque x Ratio x Efficiency
OT = Output Torque

$\text{Output rpm} = \frac{\text{Input rpm}}{\text{Ratio}}$

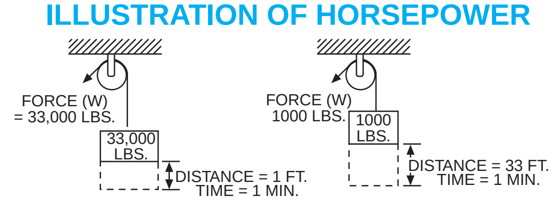
$OHL = \frac{2\ TK}{D}$

OHL = Overhung Load (lb)
T = Shaft Torque (lb-in.)
D = PD of Sprocket, Pinion or Pulley (in.)
K = Overhung Load Factor

Overhung Load Factors:
Sprocket or Timing Belt 1.00
Pinion & Gear Drive 1.25
Pulley & V-Belt Drive 1.50
Pulley & Flat Belt Drive 2.50
Variable Pitch Pulley 3.50

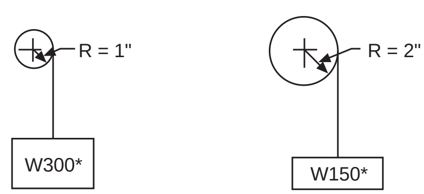
kW = hp x 0.7457
in. = mm/25.4
Temp. °C = (°F - 32) x 0.556
Temp. °F = (°C x 1.8) + 32
Torque (lb-in.) = 86.6 x kg•m
Torque (lb-in.) = 8.85 x N•m
Torque (lb-in.) = 88.5 x daN•m

ILLUSTRATION OF HORSEPOWER



$$HP = \frac{33{,}000 \times 1}{33{,}000 \times 1} = 1\ HP \qquad HP = \frac{1000 \times 33}{33{,}000 \times 1} = 1\ HP$$

TORQUE (T) is the product of a FORCE (W) in pounds, times a RADIUS (R) in inches from the center of shaft (Lever Arm) and is expressed in Inch Pounds.



T=WR=300 x 1=300 In. Lbs. T=WR=150 x 2=300 In. Lbs.

If the shaft is revolved, the FORCE (W) is moved through a distance, and WORK is done.

$$\text{WORK (Ft. Pounds)} = W \times \frac{2\pi R}{12} \times \text{No. of Rev. of Shaft.}$$

When this WORK is done in a specified TIME, POWER is used.

$$\text{POWER (Ft. Pounds per Min.)} = W \times \frac{2\pi R}{12} \times RPM$$

Since (1) HORSEPOWER = 33,000 Foot Pounds per Minute

$$\text{HORSEPOWER (HP)} = W \times \frac{2\pi R}{12} \times \frac{RPM}{33{,}000} = \frac{WxRxRPM}{63{,}025}$$

but TORQUE (Inch Pounds) = FORCE (W) X RADIUS (R)

$$\text{Therefore HORSEPOWER (HP)} = \frac{\text{TORQUE (T)} \times RPM}{63{,}025}$$