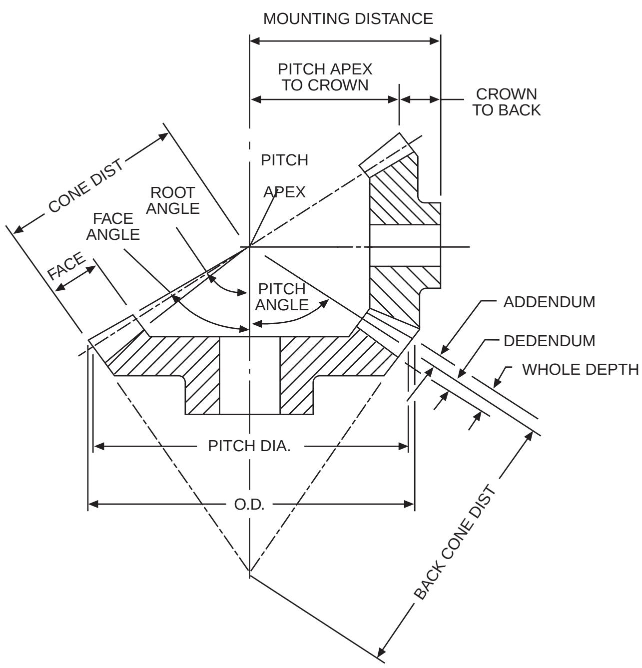
MITER AND BEVEL GEARS

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

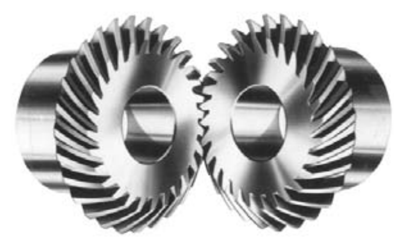
Nomenclature may best be understood by means of graphic representation depicted below:



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	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.

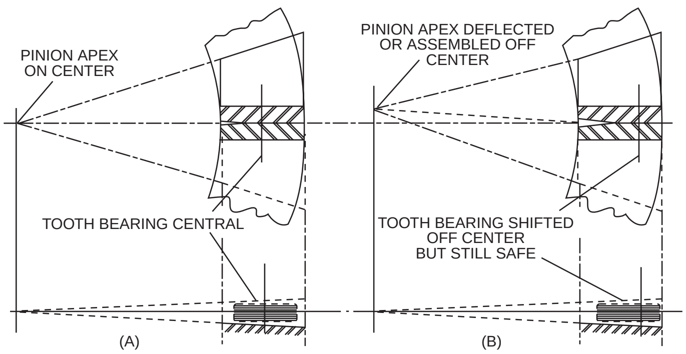


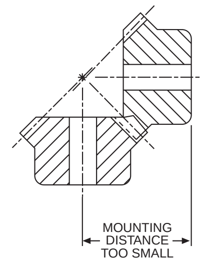
ILLUSTRATION OF LOCALIZED TOOTH BEARING IN STRAIGHT BEVEL CONIFLEX® GEARS

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.

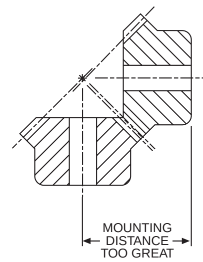
Incorrect

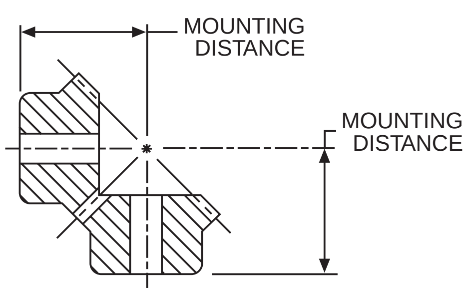
If Mounting Distance of one or both gears is made less than dimension specified, the teeth may bind. Excessive wear or breakage can result. Drawing below shows gears mounted incorrectly with the Mounting Distance too short for one gear.

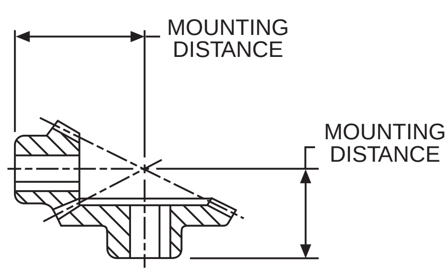


Incorrect

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.







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ENGINEERING INFORMATION

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 $T = \frac{W \times D}{2}$

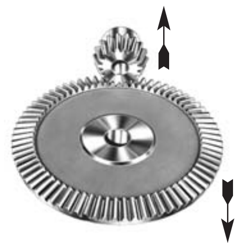
The safe horsepower capacity of the gear (at a given RPM) can be calculated from $HP = \frac{T \times RPM}{63{,}025}$ or directly from (W) and (V);

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

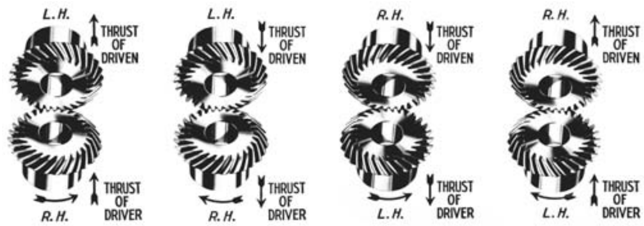
For a known HP, $T = \frac{63025 \times HP}{RPM}$

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.

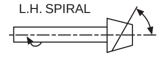
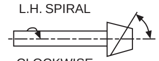
Straight Bevels and Miters

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

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

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°

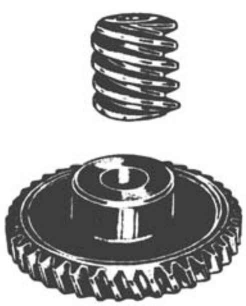
WORMS AND WORM GEARS

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:

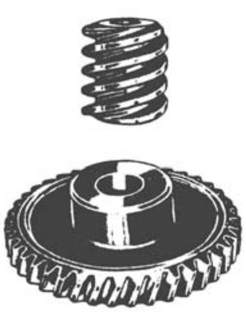
- HIGH RATIO SPEED REDUCTION
- LIMITED SPACE
- RIGHT ANGLE (NON-INTERSECTING) SHAFTS
- GOOD RESISTANCE TO BACK DRIVING

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



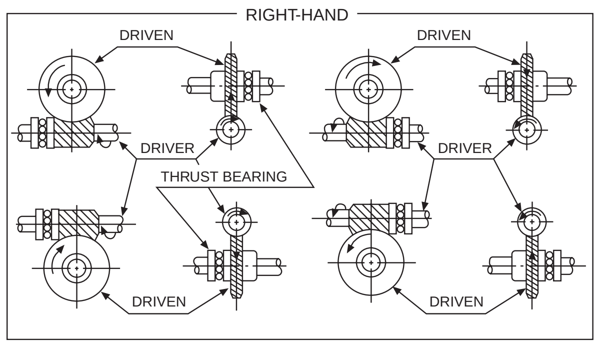
Threads of LEFT-HAND lean to the Left when standing on either end:

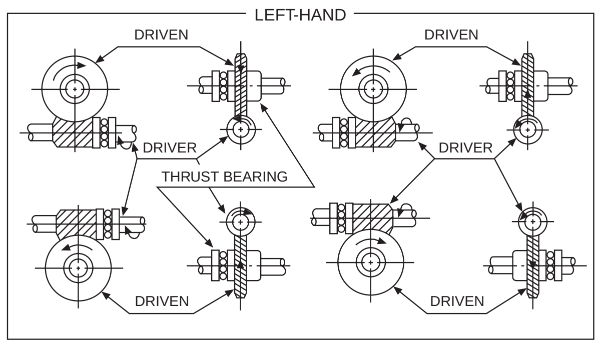


Threads of RIGHT-HAND lean to the Right when standing on either end:

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.

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

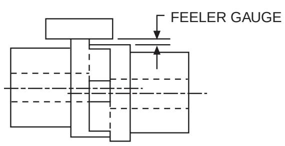
The high rate of sliding friction that takes place at the mesh of the Worm and Gear results in a more complex method of rating these Gears as opposed to the other Gear types. Material factors, friction factors and velocity factors must all be considered and applied to reflect a realistic durability rating.

COUPLINGS

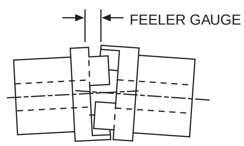
ALIGNMENT

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.

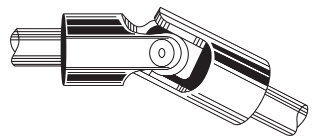
MAXIMUM READING DIFFERENTIAL

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

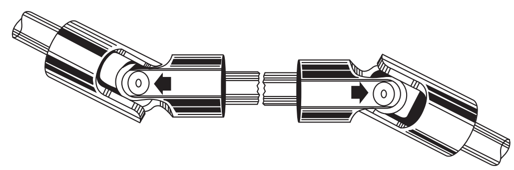
UNIVERSAL JOINTS

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.

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.

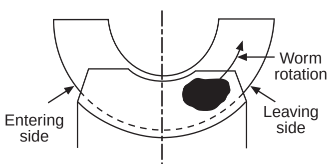
WORMS AND WORM GEAR

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.

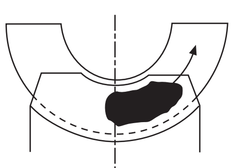
HOW WORM GEARS "ADJUST" THEMSELVES

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.

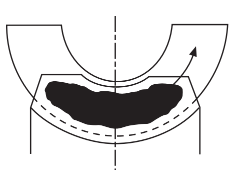
Evolution of Contact in a Worm Gear



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.

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.

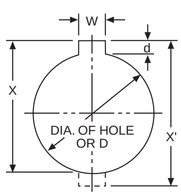
Bronze worm and helical gears are produced from several alloys selected for bearing and strength properties. Phosphor bronze is used for helicals and some worm gears (12P and coarser). Finer pitch worm gears are made from several different grades of bronze, dependent on size.

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:

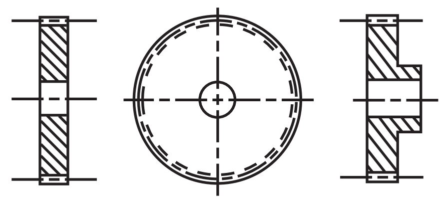
Hole 1”; Keyway 1/4” wide by 1/8” deep.

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

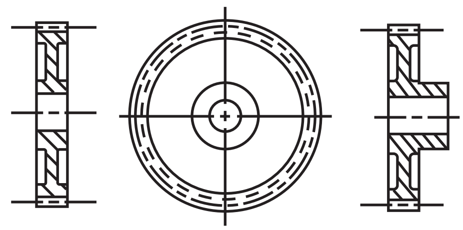
$$X' = 2.218 - 1.000 = \mathbf{1.218''}$$

STYLES

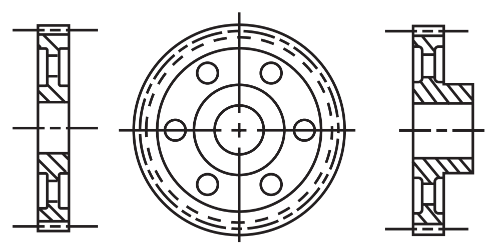
Boston Spur, Helical, and Worm Gears are carried in Plain, Web, or Spoke styles, as illustrated.



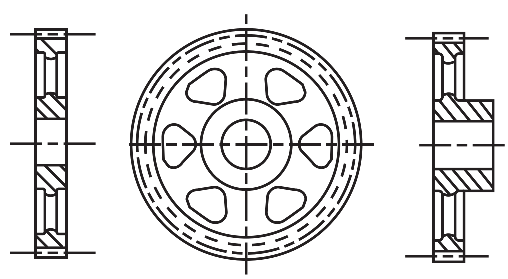
PLAIN – A



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)	$V = .2618 \times D \times RPM$
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	$T = W \times R$
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

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

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

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

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

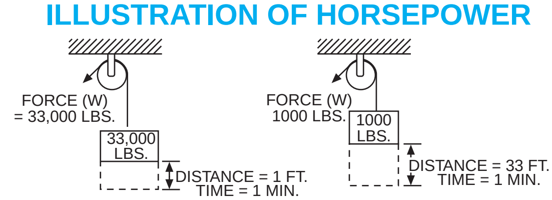
$$OHL = \frac{2\ TK}{D}$$

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

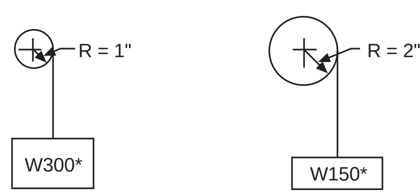
kW = hp x 0.7457
in. = mm/25.4
Temp. °C = (°F - 32) x 0.556
Temp. °F = (°C x 1.8) + 32
Torque (lb-in.) = 86.6 x kg•m
Torque (lb-in.) = 8.85 x N•m
Torque (lb-in.) = 88.5 x daN•m

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)

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