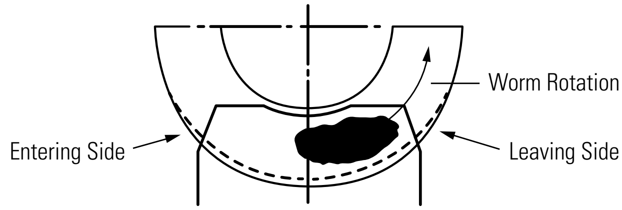
Worm Bearing Fits And Adjustments

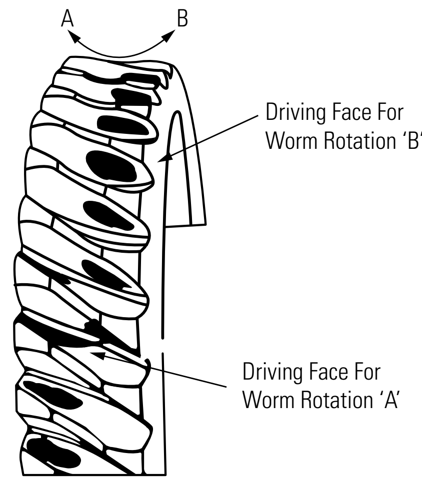
Dimensions, "MC", "MD", "MG" and "MH" (2" through 7" sizes) on pages 46 and 47 illustrate case bore dimensions for designer use in specifying correct component fit. Dimension "MH" represents the bearing manufacturer's recommended bore tolerance and dimension "MG" the bearing clearance bore. The same bores are used on both sides of the assembly. Bearing endplay should be adjusted to the following by use of shims at the bearing covers:

2" through 4" size .003 to .007"

5" through 7" size .006 to .010"

Housing bore dimensions on pages 40 through 52 (8" through 48" sizes) again represent recommended case machining tolerances. Dimension "MH" (8" through 12" sizes) is the thrust bearing manufacturer's recommended fit and the proper radial bearing housing case bore. Sizes 14" through 20" use a tapered roller radial bearing housing mounted in case bore dimension "MH". Sizes 21.837" through 48" use tapered roller bearings mounted directly in case housing bores "MH" and "MN". Worm thrust bearings with spacers are not adjustable and the bearing retainer should be pulled up tight to assure clamping of the cups and cup spacer. Proper end-play has been designed into the bearing spacer. No adjustment is required on the spherical or tapered roller bearings on the 8" through 48" sizes.





Location Of Contact

Provision for adjustment of the gearing at assembly cannot be overlooked. The worm, having threads that are continuous in form, is not critical in regard to endwise location. The gear, however, must be precisely located in an axial position. In most assemblies the accumulation of tolerances on the dimensions of housings, shafts, bearings and gear makes it impractical to control the location of the gear by accuracy of machining alone. Correct positioning is normally achieved by shimming the gear at assembly.

All gears are produced to allow for deflection and to provide an entry gap for lubricant on the "entering side" of the gear teeth. Therefore, contact on the driving face of the worm gear tooth is required on the "leaving side" as shown in the top figure. Contact should be checked after the worm and gear have been installed by coating the threads of the worm with Prussian blue and turning gears in mesh by hand.

The contact surface can easily be shifted by changing shims at the opposite end of the gear shaft to move the gear to the right or left of the worm. In doing this, once bearing adjustment is made, shims should not be added or removed simply moved from under one cover to the other. When assembling a worm gear which has to run in both directions of rotation, it is necessary to consider both driving faces of the teeth and to aim at contact as shown in the bottom figure. Note that both faces have a leaving side contact in relation to the corresponding direction of rotation of the worm. This is an inherent feature of Delroyd worm gearing as the worm deflects under load, contact moves toward the center of the gear tooth but still maintains some gap for lubricant on the entering side.

Lubrication

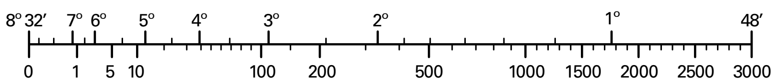
Worm gear performance depends on the ability of the lubricant to reduce friction on, and carry heat from, the working surfaces. Recommended lubricants are those meeting requirements out-lined in Table #8, American Gear Manufacturers Association Specification #250.04, for cylindrical worm gearing. In the usual lubrication system, oil contained within the housing is directed by splash to the bearings and to the zone of tooth and thread contact. Natural splash can be augmented by flingers, scrapers and cups attached to the gear. Channels or ribs can be placed inside the housing to help direct oil to bearings. For splash lubrication the recommended lubricant levels are (1) worm-below-gear: level at center of worm: (2) worm-above-gear: level a 1/6 of gear diameter: and (3) worm-beside-gear: level at center of worm and gear. To avoid excessive oil churning, pressure lubrication is advisable where speeds are high. This method is also advisable for worm-mover-gear arrangements operation at speeds too slow to assure satisfactory lubrication. With this system an oil cooler in the pipeline can be used to good advantage.

BEARING SELECTION

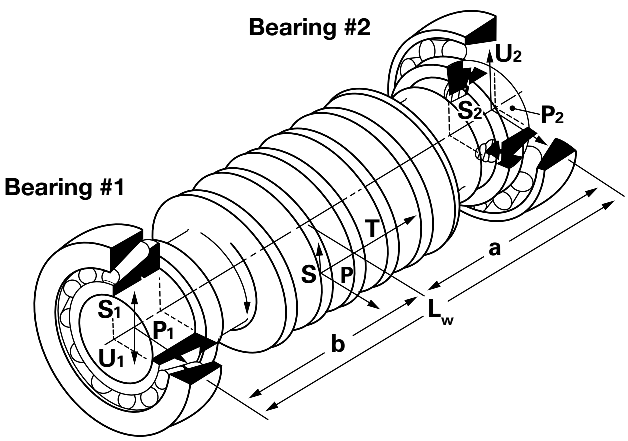
Careful consideration should be given to selection of bearings in order to locate adequately the gearing, and support the loads involved. A procedure for calculating bearing loads is presented on this page. The use of standards is recommended wherever design considerations permit.

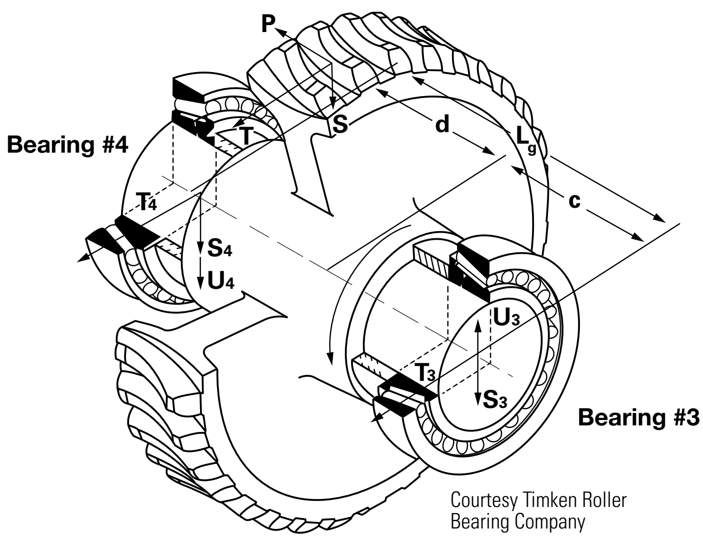
HOW TO CALCULATE BEARING LOADS

Friction Angle ø



V_S – Rubbing Speed (Feet per Minute)





Courtesy Timken Roller Bearing Company

GEARS

Integral gear and hub assemblies from standard Delroyd reducers are available as shown on page 53. This tabulation also shows flanged rim designs, size 3 1/2" center distance and larger, suitable for bolting to any adaptable center. Flanged rims fit the center in the counterbore only and a shrink fit is recommended where the gear is to transmit rated loads. This is necessary due to the differential rate of expansion between bronze and the center material, and insures positive location of the rim at maximum operating temperature. The flanged rim locating dimension is shown on page 53 as dimension "K". Also listed is the hub outside diameter "U" necessary to provide a recommended ASA Class FN-2 shrink fit. Assembly can easily be facilitated by heating the rim to about 200°F. This will cause sufficient expansion to permit center insertion for bolt hole alignment. Lightly loaded applications, where operating temperatures stay reasonably under the normal 180° F range, can use a free fit. The interference fit should also be avoided on applications involving less than standard backlash.

Principal forces and bearing loads in a worm and gear set.		
P	=	Q/WPR
S	=	P tan NPA/sin (LA+ ø)
T	=	P / tan (LA+ ø)
V_s	=	.262 (WPD) (RPM) / cos LA

where

WPD	is	pitch diameter of worm, inches
WPR	is	pitch radius of worm, inches
GPR	is	pitch radius of gear, inches
LA	is	lead angle of worm, degrees
P	is	tangential force of worm, pounds
Q	is	torque input to worm, inch pounds
S	is	separating force, pounds
T	is	axial thrust of worm, pounds
NPA	is	normal pressure angle, degrees*
ø	is	friction angle for worm driving, degrees
RPM	is	worm speed, RPM
V_S	is	rubbing speed, feet per minute

*20° for C.D. 2.000", 3.500", and 14.000" through 48.000".
$22\frac{1}{2}$° for C.D. 3.000", and 4.000" through 12.000".

Bearing Loads

Resulting from	Bearing #1	Bearing #2	Bearing #3	Bearing #4
P	$P(a)/L_w = P1$	$P(b)/L_w = P2$	$P(GPR)/L_g = U3$	$P(GPR) / L_g = U4$
S	$S(a)/L_w = S1$	$S(b)/L_w = S2$	$S(d)/L_g = S3$	$S(c)/L_g = S4$
T	$T(WPR)/L_w = U1$	$T(WPR)/L_w = U2$	$T(d)/L_g = T3$	$T(c)/L_g = T4$
Radial Load	$\sqrt{P_1^2+(S1-U1)^2}$ =R1	$\sqrt{P_2^2+(S2+U2)^2}$ =R2	$\sqrt{T_3^2+(U3-S3)^2}$ =R3	$\sqrt{T_4^2+(S4+U4)^2}$ =R4
Thrust Load		T		P

Approximate WR^2 Values

Assuming uniform acceleration (or deceleration), the motor torque required to accelerate - or the brake torque required to decelerate - in a given time can be determined by the following:

$$T = \frac{WR^2_{WS} \times \Delta N}{3690t}$$

where

T	=	Torque in inch pounds
WR^2_{ws}	=	lb-in^2
ΔN	=	RPM change
t	=	Time in seconds

Listed in the table are approximate WR2 values for standard single extended worm gear assemblies and standard low speed shaft assemblies for sets shown in this catalog.

$$WR2_{WS} = \frac{WR^2_{LSS} \times WR^2_{DL}}{(ratio)^2} + WR^2_{WA} + WR^2_{M}$$

where

WR^2_{WS}	is	WR^2 of the system with respect to input shaft
WR^2_{LSS}	is	WR^2 of the reducer low speed shaft assembly
WR^2_{DL}	is	WR^2 of the driven load
WR^2_{WA}	is	WR^2 of the worm assembly
WR^2_{M}	is	WR^2 of the motor

Center Distance	WR^2 Worm Assembly lb-in^2	WR^2 Low Speed Shaft Assembly* lb-in^2
2.500"	.36	10.0
3.000"	1.40	20.0
3.500"	1.58	53.0
4.000"	3.30	114
5.000"	8.52	265
6.000"	15.4	633
7.000"	16.3	1,220
8.000"	38.7	2,250
9.000"	55.3	4,020
10.000"	76.4	6,700
12.000"	151	17,300
14.000"	369	23,800
17.000"	513	66,300
20.000"	1,920	174,000
21.837"	2,150	215,000
24.000"	3,560	487,000
27.000"	6,510	566,000
30.000"	7,910	945,000
36.000"	10,600	2,140,000
42.000"	26,800	4,420,000
48.000"	31,700	8,030,000

*Below 12" center distance, this value represents a complete gear and not just a flanged rim. For 12" and greater center distances, the WR^2_{LSS} consists of a flanged rim plus a gear center approximating the size of the gear center used in standard Delroyd units.

Service Factors

Tables in ths catalog provide mechanical ratings in terms of input horsepower and inch-pounds output torque. Mechanical ratings reflect gearing wear capacity. Values in the rating tables apply for continuous service, free from recurrent shock loading, and of total duration up to ten hours per day. Normal starting or momentary peak loads up to 300% of this rating are permissible for a maximum period of two seconds duration. The total number of 300% peak loads is limited to 25,000 over the life of the reducer. Use of service factors is necessary dependent on actual nature and duration of service. The terms specified in the service factor table **"intermittent"** and **"occasional"** refer to total operating time per day while the term **"frequent starts and stops"** refers to more than ten starts per hour.

Service Factors

		Driven Machine AGMA Load Classification		
		Uniform	Moderate Shock	Heavy Shock
Prime Mover	Duration of Service	(Peak Load of 100% of Driver Hp.)	(Peak Load of 125% of Driver Hp.)	(Peak Load of 150% of Driver Hp.)
Electric Motor	occasional 1/2 hr/day	0.80	0.90	1.00
	intermittent 2 hr/day	0.90	1.00	1.25
	10 hr/day	1.00	1.25	1.50
	24 hr/day	1.25	1.50	1.75
Multi-cylinder internal combustion engine	occasional 1/2 hr/day	0.90	1.00	1.25
	intermittent 2 hr/day	1.00	1.25	1.50
	10 hr/day	1.25	1.50	1.75
	24 hr/day	1.50	1.75	2.00
Single cylinder internal combustion engine	occasional 1/2 hr/day	1.00	1.25	1.50
	intermittent 2 hr/day	1.25	1.50	1.75
	10 hr/day	1.50	1.75	2.00
	24 hr/day	1.75	2.00	2.25
FOR FREQUENT STARTS AND STOPS (more than 10 per hour)				
Electric motor	occasional 1/2 hr/day	0.90	1.00	1.25
	intermitten 2 hr/day	1.00	1.25	1.50
	10 hr/day	1.25	1.50	1.75
	24 hr/day	1.50	1.75	2.00