Gear Reference Guide

W.M. Berg manufactures several styles of gears. Each gear has and serves its own particular application. Listed below are brief descriptions and application notes for the variety of available styles. Further information can be obtained from numerous gear and mechanical design handbooks, or by contacting our engineering department

Gear Types

Spur Gears are the most recognized style of gear. Spur Gears are used exclusively to transmit rotary motion between parallel shafts, while maintaining uniform speed and torque. The involute tooth form, being the simplest to generate, permits high manufacturing tolerances to be attained.

Internal Spur Gears, unlike spur gears, have the teeth generated on the I.D. of the blank. They are generally stronger and more efficient than the mating pinion gear. The pitch diameter of the internal gear must be at least 1.5 times the P.D. of the mating pinion.If this condition is not met, interferences between the tips of the teeth will occur. Internal gears provide the designer with the ability to achieve higher contact and drive ratios than standard spur gears at shorter center distances. They also enable a velocity change without a directional change. This would require an idler gear with standard spurs.

Helical Gears are similar to spur gears with the exception that the teeth are cut at an angle to the axis of the shaft (helix angle). The helix cut creates a wider contact area enabling higher strengths and torques to be achieved.Though helical gearsets operate quieter and smoother than spur gears, they are slightly less efficient. Helical gears can run on parallel shafts or may be offset as much as the helix angle will permit.Axial thrust loads are developed during operation and must be considered when selecting bearings and mounting arrangements.

Racks are best described as spur gears of infinite pitch radius. They will translate rotary motion to linear motion (rack driven by pinion) and vice versa. Racks will mate pinions of the same pitch.

Gear
Reference Data

Bevel Gear are used exclusively to transmit rotary motion between intersecting shafts. Through commonly seen in right angle drives, bevel gears can be cut to drive any angle. A cross section of the gear tooth reveals a profile similar to a spur gear. However, as the tooth is generated. the cross section section decreases the closer it gets to the center of the gear. Bevel gear sets will produce axial thrust loads which must be compensated for when selection bearings and designing mounting fixtures. Bevel gears of 1:1 ratio are referred to as miter gears.

Worm and Worm Wheel: Best choice of gearing when high drive reduction is required. Worm Wheels resemble helical gears with the addition of a throat cut into the O.D. of wheel. The throat permits the worm wheel to fully envelope the threads of the worm. Threads, not teeth are cut on the worm, and by adjusting the number of threads, different ratios can be achieved without altering mounting arrangements. A unique feature of Worm and Wheel assemblies is their ability to prohibit back driving. Certain pitches and leads of the worm will not permit the worm wheel to drive the worm. This is useful when an application requires the output to lock-up should the application operate in the opposite direction. The worm is self locking when the helix angle is less than 5˚. The worm is back drivable when the helix angle is greater than 10˚. Worm and Worm Wheel assemblies must be mounted on perpendicular, non-intersecting shafts.

Many **Gearing Assemblies** can be developed from W.M. Berg's extensive inventory, however, we have designed several styles of gear boxes that are "Ready to Install". Refer to catalog for styles and selection, then contact our Sales department for availability and pricing.

Gear/Gear Assembly Efficiencies:

	EFFICIENCY RANGE %	RATIO RANGE	PITCH LINE VELOCITY (FEET/MIN)	PITCH LINE VELOCITY (METERS/MIN)
SPUR	97 TO 99	1:1 TO 10:1	10,000	3,000
BEVEL	96 TO 98	5:1 TO 400:1	10,000	3,000
WORM	50 TO 90	5:1 TO 400:1	25,000	7,600
HELICAL	96 TO 98	1:1 TO 8:1	10,000	3,000

Gear Reference Guide

GEAR TOOTH STRENGTH

Many factors must be considered when designing a gear train. The information listed on this page should be used as a general guideline for your application. If more critical strength calculation is required W.M.Berg suggests that you consult our engineering department or any one of the many gear handbooks that are readily available.

When a gear train is transmitting motion, it is save to assume that all of the load is being carried by one tooth.This is because as the load approaches the end of the tooth, where the bending force would be the greatest, a second tooth comes into mesh to share the load. Simple results can be obtained from the Lewis bending strength equation.

$$W_t = \frac{SFY}{D.P.}$$

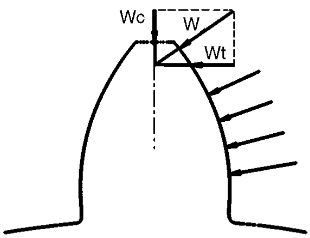
W_t = Maximum transmitted load (lbs or N)

S = Maximum bending tooth stress (taken as 1/3 of the tensile strength) See Table C on Page 5

F = Face width of gear (in.or mm)

D.P. = Diametral Pitch = 1/module (for equation only)

Y = Lewis Factor (See Table)



NOTE: The maximum bending tooth stress (S) is valid for well lubricated, low shock applications. For high shock, poorly lubricated applications, the safe stress could be as low as .025S. If your design calls for an unfriendly environment for gears, you might want to lower S to assure a reasonable amount of gear life.

LEWIS FACTOR - Y

NO. OF TEETH	14 1/2° INVOLUTE	20° 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

Table C

Mechanical Properties (Machine Design in Mechanical Design - Robert L.Mott)

MATERIAL	Hardness (Rockwell)	Tension Modulus (E)	Tensile Strength	Yield Strength	Shear Strength	Endurance Limit
303 Stainless Steel	B75-90	28×10^6 PSI 194 GPA	90×10^3 PSI 623 MPA	35×10^3 PSI 242 MPA	75×10^3 PSI 517 MPA	35×10^3 PSI 242 MPA
17-4 PH Stainless Steel	C28-35	28.5×10^6 PSI 197 GPA	150×10^3 PSI 1040 MPA	125×10^3 PSI 865 MPA	83×10^3 PSI 574 MPA	90×10^3 PSI 623 MPA
416 Stainless Steel	C26-36	29×10^6 PSI 201 GPA	75×10^3 PSI 519 MPA	40×10^3 PSI 277 MPA	75×10^3 PSI 517 MPA	40×10^3 PSI 277 MPA
416 Stainless Steel - Hardened	C36-42	29×10^6 PSI 201 GPA	135×10^3 PSI 930 MPA	105×10^3 PSI 725 MPA	75×10^3 PSI 517 MPA	40×10^3 PSI 277 MPA
12 L14 Steel	B75-90	30×10^6 PSI 208 GPA	78×10^3 PSI 540 MPA	60×10^3 PSI 415 MPA	50×10^3 PSI 345 MPA	--
2024T4 Aluminum	--	10.6×10^6 PSI 73.4 GPA	68×10^3 PSI 470 MPA	47×10^3 PSI 325 MPA	40×10^3 PSI 276 MPA	--
464 Brass Alloy	--	18×10^6 PSI 125 GPA	57×10^3 PSI 395 MPA	25×10^3 PSI 173 MPA	40×10^3 PSI 276 MPA	--
360 Brass Alloy	--	14×10^6 PSI 97 GPA	49×10^3 PSI 339 MPA	18×10^3 PSI 125 MPA	30×10^3 PSI 205 MPA	--

Special Bore

+ .0005
Tol. - .0000

Designator	**Designator**	**Designator**
.0781 = B	.1873 = H	.3748 = M
.0900 = V	.1875 = HH	.3750 = MM
.0937 = D	.2405 = J	.4998 = R
.1200 = E	.2498 = K	.5000 = RR
.1248 = F	.2500 = KK	.6248 = T
.1250 = FF	.3123 = L	.6250 = TT
.1562 = G	.3125 = LL	.6871 = W

Example:
Stock Number P48S28-120 (1/4" Bore to be rebored to .3748)
Specify as follows:P48S28-120-M

In the table listed below are the basic standard Metric Motor Shaft and Metric Bearings Bore diameters most commonly used.

For modification of W.M. Berg, Inc. U.S. Standard Bore components to Metric Bores select nearest standard to desired metric bore and modify same. Modification charge will apply. Note: True Metric available, see Berg Metric Catalog.

W.M. Berg, Inc. U.S. Standard Bore Diameters		Standard Metric System Bores		Recommended Rebore Dimensions & Tolerances
Fractional	**Decimal**	**mm**	**Metric Tolerances**	
1/8	.1248 + .0005	4	H7	.1573 + .0005
3/16	.1873 + .0005	5	H7	.1966 + .0005
3/16	.1873 + .0005	6	H7	.2360 + .0005
1/4	.2498 + .0005	7	H7	.2757 + .0006
1/4	.2498 + .0005	8	H7	.3148 + .0006
5/16	.3123 + .0005	9	H7	.3541 + .0006
3/8	.3748 + .0005	10	H7	.3935 + .0006
3/8	.3748 + .0005	12	H7	.4725 + .0007
1/2	.4998 + .0005	14	H7	.5510+ .0007
1/2	.4998 + .0005	15	H7	.5907+ .0007

Other basic Berg code designators
W = Worm Wheel
H = Helical Gears
M = Miter & Bevel Gears
R = Racks Spur
S = Shafting

AP = Anti-backlash Pin Hub
AC = Anti-backlash Clamp Hub
PH = Pin Hub
CH = Clamp Hub
CG = Clamps - Gears

Note: Most gears as specified by Berg numbering system are stock.Any others, not listed in our catalogs, are considered specials and are gears cut to order using basic stock blanks.

All prices and quantity discounts are available on request.

No exchanges or returns are expected on special non-stock parts as all such parts are made to your particular specification and have no resale value.

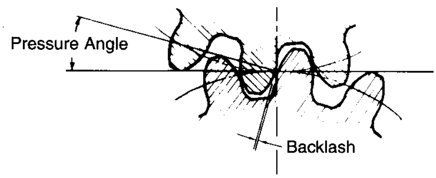
Gear Reference Guide

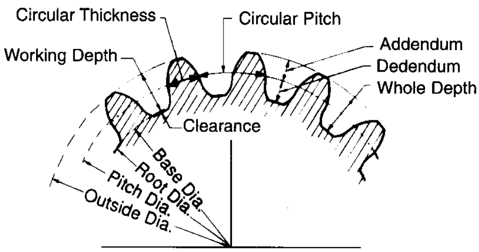
Table of Gear Tooth Proportions

DIAMETRAL PITCH	CIRCULAR PITCH	CIRCULAR THICKNESS	ADDENDUM	DEDENDUM	WORKING DEPTH	WHOLE DEPTH
12	.26180	.13090	.0833	.1020	.1667	.1853
16	.19635	.09818	.0625	.0770	.1250	.1395
20	.15708	.07854	.0500	.0620	.1000	.1120
24	.13090	.06545	.0417	.0520	.0833	.0937
1/10	.10000	.05000	.0318	.0402	.0637	.0720
32	.09817	.04909	.0313	.0395	.0625	.0708
48	.06545	.03272	.0208	.0270	.0417	.0487
1/20	.05000	.02500	.0159	.0211	.0318	.0370
64	.04909	.02454	.0156	.0208	.0313	.0364
72	.04363	.02182	.0139	.0187	.0278	.0362
80	.03927	.01963	.0125	.0170	.0250	.0295
96	.03272	.01636	.0104	.0145	.0208	.0249
100	.03142	.01571	.0100	.0140	.0200	.0240
120	.02618	.01309	.0083	.0120	.0167	.0203
200	.01571	.00785	.0050	.0080	.0100	.0130

Table A

Gear Terms and Abbreviations





Base Diameter - (B.D.) The diameter of the circle from which the involute is generated

Backlash - Is the amount by which the width of a tooth space exceeds the thickness of the engaging tooth of a mating gear, when both gears are at nominal center distances.

Center Distance - (C.D.) Distance between the centers of mating gears.

Circular Pitch - (C.P.) The distance, along the Pitch Circle, between corresponding points of adjacent teeth.

Circular Thickness - Thickness of tooth on pitch circle.

Diametral Pitch - (D.P.) Number of teeth in a gear having one inch pitch diameter. Ex.: A gear having 48 teeth and a 1”pitch diameter is a 48 diametral pitch.

Number of Teeth - (N)

Outside Diameter - (O.D.) Diameter measuring on tops of teeth.

Pressure Angle - (P.A.) The angle between a line tangent to the pitch circle and a line perpendicular to the tooth profile at the point of contact.

Pitch Diameter - (P.D.) Diameter of the pitch circle.

Pitch Circle - An imaginary circle, whose diameter is equal to the number of teeth divided by diameter pitch.

Testing Diameter - (T.D.) A diameter, established by inspection with a master gear of known size. It is equal to twice the difference between the tight mesh center distance, and the sum of the master gear testing diameter, divided by 2.

$$\text{T.D.} = 2 \times \left(\frac{\text{Tight Mesh}}{\text{Center Distance}} - \frac{\text{T.D. Master Gear}}{2} \right)$$

Gear Reference Guide

Rules and Formula for Standard Full Depth Spur Gears

Table B

TO GET	HAVING	RULE	FORMULA
The diametral pitch	THE CIRCULAR PITCH	DIVIDE 3.1416 BY THE CIRCULAR PITCH	$DP=\frac{3.1416}{CP}$
The diametral pitch	THE PITCH DIAMETER AND THE NUMBER OF TEETH	DIVIDE NUMBER OF TEETH BY PITCH DIAMETER	$DP=\frac{N}{PD}$
The diametral pitch	THE OUTSIDE DIAMETER AND THE NUMBER OF TEETH	DIVIDE NUMBER OF TEETH PLUS 2 BY THE OUTSIDE DIAMETER	$DP=\frac{N+2}{OD}$
Pitch Diameter	THE NUMBER OF TEETH AND THE DIAMETRAL PITCH	DIVIDE THE NUMBER OF TEETH BY THE DIAMETRAL PITCH	$PD=\frac{N}{DP}$
Pitch Diameter	THE OUTSIDE DIAMETER AND THE DIAMETRAL PITCH	SUBTRACT FROM THE OUTSIDE DIAMETER THE QUOTIENT OF 2 DIVIDED BY THE DIAMETRAL PITCH	$PD=OD-\frac{2}{DP}$
Outside Diameter	THE NUMBER OF TEETH AND THE DIAMETRAL PITCH	DIVIDE NUMBER OF TEETH PLUS 2 BY THE DIAMETRAL PITCH	$OD=\frac{N+2}{DP}$
Outside Diameter	THE PITCH DIAMETER AND THE NUMBER OF TEETH	DIVIDE THE NUMBER OF TEETH PLUS 2 BY THE QUOTIENT OF NUMBER OF TEETH DIVIDED BY THE PITCH DIAMETER	$OD=\frac{N+2}{\frac{N}{PD}}$
Number of teeth	THE PITCH DIAMETER AND THE DIAMETRAL PITCH	MULTIPLY PITCH DIAMETER BY THE DIAMETRAL PITCH	$N=PD \times DP$
Thickness of tooth	THE DIAMETRAL PITCH	DIVIDE 1.5708 BY THE DIAMETRAL PITCH	$T=\frac{1.5708}{DP}$
Addendum	THE DIAMETRAL PITCH	DIVIDE 1 BY THE DIAMETRAL PITCH	$ADD.=\frac{1}{DP}$
Addendum	THE PITCH DIAMETER AND THE NUMBER OF TEETH	DIVIDE THE PITCH DIAMETER BY THE NUMBER OF TEETH	$ADD.=\frac{PD}{N}$
Working depth	THE DIAMETRAL PITCH	DIVIDE 2 BY THE DIAMETRAL PITCH	$WDE=\frac{2}{DP}$
Whole depth	THE DIAMETRAL PITCH	DIVIDE 2.2 BYTHE DIAMETRAL PITCH AND ADD .002	$WD=\frac{2.2}{DP}+.002$
Clearance	THE DIAMETRAL PITCH	DIVIDE .2 BYTHE DIAMETRAL PITCH AND ADD .002	$C=\frac{.2}{DP}+.002$
Standard center distance	NUMBERS OF TEETH IN MATING GEARS & DIAMETRAL PITCH	ADD THE NUMBERS OF TEETH IN THE MATING GEARS & DIVIDE BY 2 TIMES THE DIAMETRAL PITCH	$\frac{N1+N2}{2 \times DP}$
Backlash	PRESSURE ANGLE, STANDARD CENTER DISTANCE, & MEASURED CENTER DISTANCE	MULTIPLY 2 TIMES THE TANGENT OF THE PRESSURE ANGLE TIMES THE DIFFERENCE BETWEEN THE STANDARD & THE MEASURED CENTER DISTANCES.	B = 2 (TAN P.A)X DIFF.

Explanation of Class "C" Backlash

Many designers tend to use tolerances much tighter than necessary to "play safe" in obtaining final backlash tolerance.

It is obvious that this is a very costly procedure, and not very realistic.There is not much use in paying for quality 12 or 14 gears if quality 10 will do the job just as well.Of course, there will be times where output backlash tolerances will require the use of more precise gears, but, their use will be justified.

You will notice that none of the gears in this catalog will show a pitch diameter tolerance. The pitch diameter is theoretical, and therefore, should not be toleranced. Tolerancing of pitch diameter will usually cause binding between mating gears at one or more points which results in excessive tooth wear on gears, improper lubrication, distortion of shafts, and overload of bearings. Naturally these conditions have detrimental effects, and the life and accuracy of the unit is impaired.

Instead, the common practice is to use a letter, A, B, C or D, as shown on the following tables. This letter, along with the quality class numbers and diametral pitch, are used to make the "center distance inspection" as outlined further. With this method there is no possibility of interference between mating gears at standard center distance. Of course, the center distance tolerance will increase or decrease backlash values depending on whether they are plus or minus. Our gears are all cut to class "C" backlash in quality 10 and 20˚ pressure angles.

Standard AGMA Center Distance Inspection Procedure

The following shows the excursion of an indicating device when checking the total composite error and size of a gear on a variable center distance fixture, with a master of known accuracy.

It is therefore recommended that the tolerances for tooth thickness shown in Table D be used in conjunction with pin measurements in order to obtain closer correlation with a size determination made by means of a master gear.

Column 1 gives the quality class. Column 2 gives the diametral pitch range for backlash values shown in column 8.Column 3 gives the backlash values for two gears of equal tooth thickness derived from column 8.Column 4 shows the total composite error for each of the classes shown in column 1.

Column 5 shows the minimum reduction on standard tooth thickness. This value is obtained by adding one half of the minimum value of column 2 (which gives the tooth thickness reduction per gear) to one half of the total composite error converted from a radial displacement to an equivalent tooth thickness. This is accomplished by multiplying the radial displacement to an equivalent tooth thickness. This is accomplished by multiplying the radial displacement by 2 Tan x of operating pressure angle.

Column 6 gives the maximum deduction in standard tooth thickness obtained in a similar manner.

Column 7 gives the maximum indicator reading which is obtained by converting the values in column 5 to radial displacements and subtracting from them one half of the total composite error shown in column 4.

Column 8 gives the minimum indicator reading.This is obtained by converting the values in column 6 to radial displacements and adding one half of the total composite error.

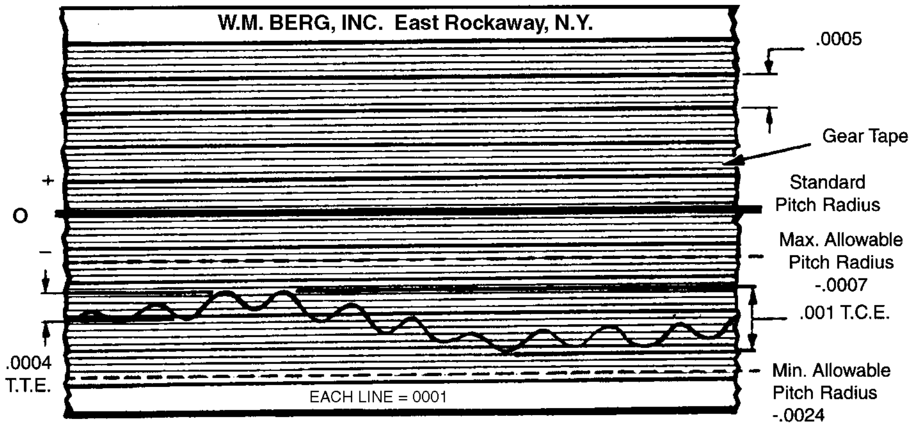
The values in columns 5 and 6 are taken through the middle of the total composite error and are therefore in closer agreement with a determination of tooth thickness made by means of pins which ignore tooth to tooth error and runout.

The foregoing procedure, along with the accompanying tables, is the method used to check all **W.M. Berg,Inc.** precision gears. We have found this method, which takes into consideration most problems commonly encountered in gear train design (such as; bind, backlash, center distance tolerance, runout, composite error, etc.) to be the one which gives the most realistic results. Of course, it does not take into account, fits of gears, shafts and bearings, these problems are dealt with later on in this section under "recommended practices".All setups on gear checking fixtures are set up with "certified" master gears, and class "xx" accuracy carbide measuring wires are used.

Class C Backlash
Table D

AGMA QUALITY CLASS	DIAMETRAL PITCH	BACKLASH IN MATING GEARS	TOTAL COMPOSITE ERROR	MINUMUM REDUCTION IN STD. TOOTH THICKNESS (Δt Min.)	MAXIMUM REDUCTION IN STD. TOOTH THICKNESS (Δt Max.)	INDICATOR LIMITS (GAGE ZEROED AT STANDARD PITCH RADIUS ALL VALUES MINUS)	
						MINUMUM	MAXIMUM
QUALITY 10 QUALITY 12 QUALITY 14	16 TO 48	.001-.002	PER AGMA STANDARD 2000-A88	.0009 .0007 .0006	.0014 .0012 .0011	.0007	.0024 .0019 .0017
QUALITY 10 QUALITY 12 QUALITY 14	1/20 TO 120	.0008-.0015		.0007 .0005 .0004	.0011 .0009 .0008	.0005	.0020 .0015 .0013

Typical Inspection Tape for A 48 D.P., 20° P.A.Quality 10C Gear



Recommended Practices

When determining materials, it is usually preferable to have the pinions of stainless steel and the mating gear made of aluminum.

The fact is the pinion usually rotates through more cycles, therefore, the tooth wear is greater on the pinion.Meshing stainless steel pinion with an aluminum gear tends to minimize this wear.

The face widths of pinions are usually wider than face widths of mating gears to insure full face contact without critical adjustment.

Where minimum backlash is a factor, it is important that fits between the bore of gears, inside diameter of bearings, and the outside diameter of shafting be held as close as possible. The accuracy of precision gears is lost unless these fits are held closely. This means that a form of selective fitting must be used, because it is too costly, if not impossible to hold tolerances that will allow perfect fit assembly. The extent of this selective fitting is determined by the accuracy requirements of the finished assembly

(continued from previous page)

Factors which control backlash are:

1. Precision class of gears.
2. Center distance tolerance.
3. Type of fit between gears, shafts and bearings.
4. Precision accuracy of bearings.
5. Straightness and adequate support of shafts.

It is important when drilling pears to shafts that the shafts be properly supported. Failure to do so can result in bending shafts with the resultant runout conditions.

Note: Gears with 14 1/2˚ pressure angle will not mesh properly with gears of 20˚ pressure angle.

It is imperative that the gears are not damaged in handling.Our gears are packaged to avoid handling damage. If they can be left in this package until used the danger should be minimized. If they are removed they should be put on tote boards and covered to prevent gears from contacting each other, and to keep them clean.

Gear Train Design

The following section deals with backlash which we believe is one of the basic problems in designing fine pitch gear trains. Backlash in mating gears or in a gear train can be broken down into three basic factors.

The quality class of the gears, i.e. quality 10, 12 or 14 (former precision standards 1, 2 or 3):the letter which designates backlash, i.e. “C” and the center distance and tolerance on which these gears operate. This section will cover AGMA type precision fine pitch gears. AGMA is an acronym for the “American Gear Manufactures Association” who publish standards which are the recognized and accepted gear standards throughout the country.

Berg precision stock gears are manufactured to AGMA classes, quality 10C, unless otherwise specified.

The following section is divided into two parts, the first dealing with backlash calculations of the non-critical gear trains, the second with backlash calculations of critical gear trains.

The accepted definition of the term backlash is “ the amount by which the width of a tooth space exceeds the tooth thickness of the engaging tooth on the pitch circles.” At first glance this meaning might lead us to think that backlash is a function of the gear cutting operation only. Actually, the teeth of a gear contribute but very little to its overall backlash value.

A complete understanding of all the elements that induce backlash is mandatory in order to properly and economically design a gear train.The following factors must be individually considered for their own parameter:

1. Standard Center Distance
2. Center Distance Tolerance
3. Size and Tolerance of Mating Gears
4. Total Composite Error of Gears
5. Fits Between Bores, Shafts and Bearings
6. Bearing Accuracy
7. Radial Play of Bearing
8. Shaft Straightness and Alignment
9. Fits Between Electrical and/or Mechanical Component Pilot Diameters and Housing Bores
10. Eccentricity and Radial Play of Electrical and/or Mechanical Component Shafts

Each of the foregoing, except standard center distance, tend to induce a “change in the center distance” which will push together or pull apart mating gears. Consequently, this push pull action produces two backlash values, minimum at the point of tightest mesh and maximum at the point of loosest mesh.

Standard Center Distance.

Standard center distance can be considered the starting point in the calculation of overall backlash values.

Standard **(theoretical)** center distance is calculated by taking one-half (1/2) the sum of the **(theoretical)** pitch diameters of mating gears.

Example:

$$C.D. = 5\,(P.D._1 + P.D._2) \text{ or } C.D. = \frac{P.D.1 + P.D.2}{2}$$

Two (2) 96 gears having theoretical P.D.s of one inch (1") and two inches (2") respectively have a standard center distance of:

$$.5 \times (1" + 2") = 1.5000 \text{ or } \frac{1" + 2"}{2} = 1.5000 \text{ C.D}$$

Center Distance Tolerance

The problem of center distance and its tolerance is usually and extremely important area for consideration. It is important to remember that the minimum and maximum backlash values between mating gears, as outlined in the previous tables are based on "standard" center distance mountings. Obviously then, if the center distance is in excess of the standard value, backlash will be increased. By the same token, if the C.D. is less than standard backlash will be decreased.However, caution must be exercised to **avoid interference** between mating gears as a result of this decrease.

Table E

ΔC (DIFFERENCE BETWEEN STANDARD AND ACTUAL CENTER DISTANCE)	B (BACKLASH IN INCHES)
.0001	.00007
.0002	.00014
.0003	.00022
.0004	.00029
.0005	.00036
.0006	.00044
.0007	.00051
.0008	.00058
.0009	.00066
.0010	.00073
.0011	.00080
.0012	.00087
.0013	.00095
.0014	.00102
.0015	.00109
.0016	.00116
.0017	.00124
.0018	.00131
.0019	.00138
.0020	.00146

The relationship of "change in center distance" which can be positive or negative depending upon C.D. tolerance, to backlash is expressed by the formula:

$$B = 2\ \text{Tan}\,\phi \times \Delta C \quad (1)$$

In which:

B = backlash in inches

ø = pressure angle

ΔC = difference between standard (theoretical) and actual center distance

Examples:

(tan 20° = .36397)

1) Standard centers + .001 tolerance
B = 2 tan 20° x ΔC
B = 72794 x .001 = + .00073 backlash

2) Standard centers - .0005 tolerance
B = 2 tan 20° x ΔC
B = .72794 x .0005 = -.00036 backlash

Backlash calculations can be divided into two categories:

1.Systems where backlash is not critical.
2.Systems where backlash is critical and exact totals must be known.

Section 1 - Backlash Calculations for Non-Critical Gear Trains

Let us set up two test problems falling into the non-critical category. These two problems will deal with items 2, 3 and 4 as previously specified and as they collectively contribute the greatest amount of change in center distance.

1. Two mating gears, 96 D.P. -20˚ P.A.having standard centers with A + .001 -.000 tolerance.

Step A - Referring to BERG quality 10 gears yield a maximum of .0015 backlash at standard centers.

Step B - With an actual center distance .001 greater than standard, an additional .0073 backlash is introduced.(See Table E)

Step C - The total probable maximum backlash would be .0015 + .00073 = .00223

2.Again two mating gears, 96 D.P. -20˚ P.A. must have no more than .0015 backlash measured at the loosest point of mesh.(See Table D)

Step A - Referring to BERG quality 10C gears yield a maximum of .0015 backlash at standard centers.

Step B - Actual center distance .0005 greater than standard yields an additional .00036 backlash.

Step C - The total probable backlash would be .0015 + .00036 = .008186 or .00036 above limit.

One solution at this point would be to use BERG quality 12C gears with the same center distance and tolerance. Again we find that the maximum backlash for quality 12C gears is .00109 and the center distance tolerance results in an additional .00036 for a total of .00145 or .00005 below limit.

Another means of overcoming this problem would be to incorporate anti-backlash or spring loaded gears. When this type of gear is used, center distance tolerance can be increased thereby reducing the cost of machining the housing.The increase in center distance when using anti-backlash gears should not exceed oz./D.P.

Before deciding on either of these two possibilities, let us see what can be done using BERG quality 10C gears.

Referring to the minimum backlash value we have .0008.This indicates that there is a minimum of .0011 clearance (see Table E).Now, because the center distance was dimensioned +.0000/-.0005 minimum backlash would be reduced to <u>.00036</u> ,
.00044

and the maximum could still be .00109 or .0045 below the limit, using BERG gears.

Section 2 - Backlash Critical Gear Trains

If the above section does not pinpoint the amount for backlash that your system will possess accurately enough for you then there are many steps that can be employed during the manufacturing process that can limit the amount of backlash in a gear train.These include cutting gears to class “D” or “E”backlash, and/or AGMA Quality 12 or 14.

If backlash is critical and BERG anti-backlash gears will not suffice, consult W.M. Berg, Inc. Engineering Department for how these changes will influence backlash.

Gear Reference Guide

Procedure for Inspection of Fine Pitch Gears.

There has been much discussion and confusion regarding the center distance method of testing fine pitch gears. W.M. Berg, Inc. center distance method of testing uses the limits and tolerances as outlined in AGMA standards.

Figures 1 and 2 are schematic diagrams showing the basic requirements for making a center distance check.The diagrams are not to be interpreted as a recommended or suggested means of construction.

In order to make this check the following items are required.

1. Rolling Fixture.
   - A. This fixture must incorporate provisions for accurately mounting both the master and the gear to be tested.
   - B. A means of accurately adjusting the weights to the correct testing pressure.
2. Master Gear
   - A. With a maximum total composite error of .0001 and a standard pitch diameter of ± .0001 tolerance over wires.
3. Hardened or ground pins or carbide lapped pins (preferred) for mounting both the master and gear to be inspected.

Lets assume that we wish to inspect a gear having the following characteristics:

64	**Diametral Pitch**
20	**Degree Pressure Angle**
.2500	**+.0000/-.0002 Bore**
80	**Teeth**
1.2500	**Theoretical Pitch Diameter**
	AGMA Class 12 C

We first select the proper master gear and correct size carbide gage pin to fit this master. since the masters we use have a .5000" $^{+.0001}_{-.0000}$ bore, we use a lapped pin of .4995 diameter to assure free movement without wobble.

We follow the same procedure in selecting a pin on which to mount the gear to be inspected. In this case the lapped pin diameter would be .24975.

The studs or pins are then mounted on the rolling fixture and caution is used to assure parallelism between the studs after mounting.

We next set our rolling fixture to the proper testing pressure which in the case of 64 C.P. is 12 ounces, per gear standards. The testing pressure is an important factor in the rolling check to assure uniform pressure and correct mesh during the entire test.

The rolling fixture must now be set to the proper center distance by the uses of "jo-blocks".The master gear we are using has a pitch diameter of 1.5000 and the gear to be inspected has a theoretical pitch diameter of 1.2500.These two pitch diameters are added together and divided by two. This gives us a figure of 1.375. We then add together and divide by two the pin diameters on which the master gear and the test gear (.49995 + .24975 respectively) are mounted. This gives us a figure of .3749. The figure is subtracted from the 1.375 figure to arrive at our "jo-block"setting of 1.001".

While holding the "jo-blocks" between the two studs or pins on the rolling fixture, the dial indicator is set to zero. This setting is the nominal or (set up) center distance and is made under the "testing pressure". All gears being tested must read to the minus side of the indicator otherwise interference between mating gears, with a standard distance, may result at assembly.

Because the gear to be tested is 64 diametral pitch AGMA quality 12C we refer to our data section to obtain the dial indicators limits.

We find in column 7 that the maximum limit is -.0005 and in column 8 the minimum is -.0015.Please note that all figures given are minus values. Theses limits can never be exceeded or the gears will not be in tolerance.

Figure 2 shows these limits as seen on a dial indicator as well as how they would be recorded on a paper chart . For the sake of clarity, however, we have not drawn the .0001" graduation lines across the chart.

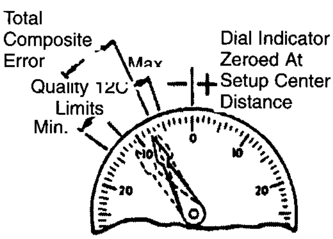
It must be remembered that although our dial limits have a range of from -.0005 to -.0015 the total composite error of an AGMA 12C gear is .0005 as shown in column 4.This means, therefore, that our dial fluctuation cannot exceed .0005 within the -.0005 to -.0015 dial limit range. Should the pointer movement not exceed the .0005 maximum total composite error, the gear would be an AGMA quality 10C. This is true even though the gear is within the dial limits on a quality 12C.

Figure 1, shows the limits for AGMA quality 10C gear. The same procedure as used to test for 12C is followed. It must be remembered, however that the dial indicator limits and the total composite error are greater. (Refer to previous page for limits.)

Measuring and Checking Forces

When making measurement-over-wire measurements or when checking the gears on a variable-center-distance device, the amount of force applied to the measuring wires or applied to maintain intimate contact between the gear to be checked and the master gear shall be as listed in the table below.

Diametral Pitch	Checking Force Oz.
16	30 To 34
20	26 To 30
24	26 To 30
1/10	22 To 26
32	22 To 26
48	18 To 22
1/20	10 To 14
64	10 To 14
72	10 To 14
80	6 To 10
96	7 To 9
100	3 To 5
120	3 To 5
200	2 To 4



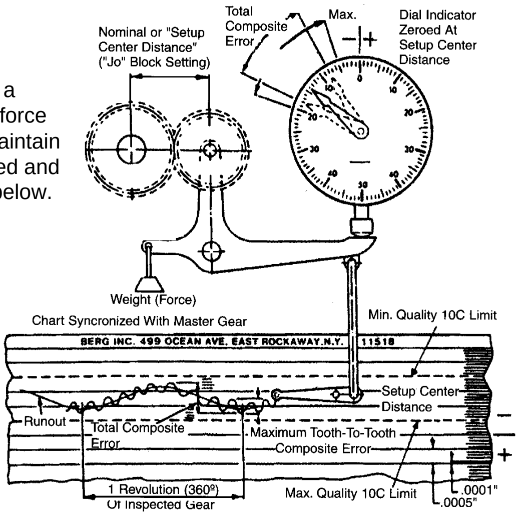


Figure 1 - AGMA Quality 10C Chart

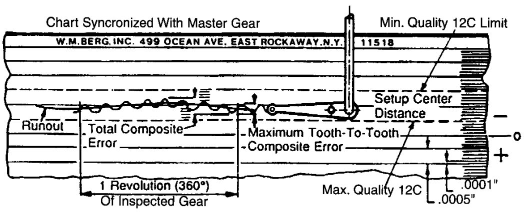


Figure 2 - AGMA Quality 12C Chart

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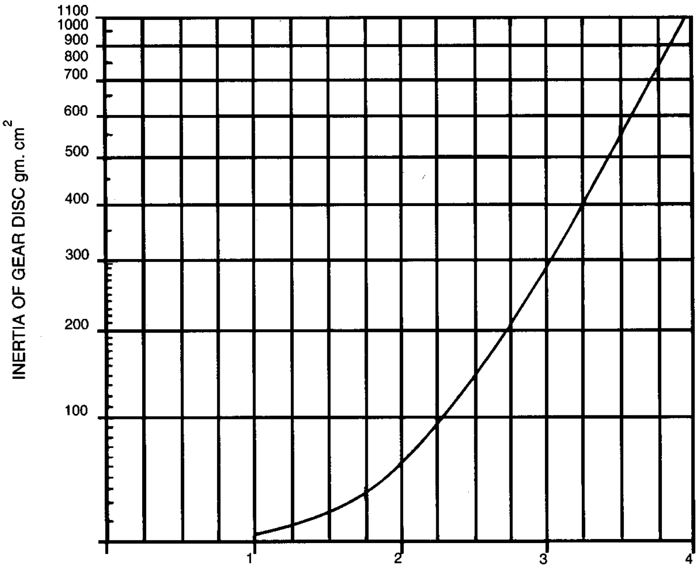
Gear Reference Guide

Inertia of Gears

To obtain the inertia of 1/8 face width aluminum gears, read directly from graph at right.



Diameter of Gear Disc in inches

To obtain the inertia of gears with other face and/or made of stainless steel read from the graph at the given diameter then multiply by the appropriate factor.

FACE WIDTH	1/16	3/32	.104	3/16	1/4	3/8	1/2
FACTOR	.5	.75	.83	1.5	2.0	3.0	4.0

MATERIAL	ALUMINUM	ST. STEEL
FACTOR	1	2.82

Example: 2" Diameter - 1/4" Face Stainless Steel Gear
From Graph 2" Dia.is 56 gm cm^2
56 x 2.0 x 2.82 = 318 gm cm^2

Inertia of Gear Clamps

CLAMP STK. NO.	INERTIA gm cm^2	CLAMP STK. NO.	INERTIA gm cm^2	CLAMP STK. NO.	INERTIA gm cm^2
CG1-4	1.91	CG1-18	61.55	CG2-5	4.95
CG1-5	2.30	CG1-19	61.55	CG2-6	4.95
CG1-8	8.86	CG1-20	0.28	CG3-1	2.01
CG1-9	11.84	CG1-21	1.50	CG3-2	2.01
CG1-11	8.86	CG2-1	2.00	CG3-3	4.01
CG1-12	11.84	CG2-2	2.00	CG3-4	4.01
CG1-14	32.35	CG2-3	2.91	CG3-5	4.97
CG1-15	61.55	CG2-4	2.91	CG3-6	4.97
CG1-17	32.35	-----	-----	CG4-1	0.18