Technical section index — Page

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Standard Gear Range

Reliance precision instrumentation gears are manufactured using high accuracy gearcutting equipment. Standard gears are produced in stainless steel, hardened stainless steel, aluminium alloy and brass (wormwheels only). Alternative materials such as PEEK™ plolymer or Delrin are available on request.

GEAR TOLERANCES

Gears are generally offered as Quality 10 (see the individual product pages). Higher qualities are available as shown in the table below. Most gears in the catalogue can be produced in these qualities to order.
Reliance standard tolerances are largely based on AGMA 390-03 backlash.

Reliance Standard Gear Qualities — Table values in 0.001mm (0.0001")

Quality Class	Modular Range	Total Composite Error	Tooth to Tooth Composite Error	Indicator Limits (Gauge zeroed at std. pitch rad.) Max	Min	Gear Quality Code
AQ10	0.8 to 0.5mod	26 (10)	13 (5)	-18 (-7)	-61 (-24)	-
AQ11		18 (7)	10 (4)		-53 (-21)	C
AQ12		13 (5)	8 (3)		-48 (-19)	B
AQ14		7 (2.7)	3.6 (1.4)		-41 (-16)	A
AQ10	0.4 to 0.2mod	26 (10)	13 (5)	-13 (-5)	-51 (-20)	-
AQ11		18 (7)	10 (4)		-43 (-17)	C
AQ12		13 (5)	8 (3)		-38 (-15)	B
AQ14		7 (2.7)	3.6 (1.4)		-33 (-13)	A

The above table values refer to measurements obtained by means of the dual flank tester.

To specify a gear other than the standard quality, add the quality code to the gear part number.
Example of a quality 12 gear - **P05S1B10F6A-100 B**
Quality code

Comparison of National Gear Quality Standards

Reliance Quality Class	American AGMA 390.03 (1980)	British BS. 4582 (pt.1 : 1970)	German DIN. 867 & 3963	International ISO	Japan JIS	Admiralty BR.6001
AQ10	Q 10	Class B	Q 7	7	3	Class 2
AQ11	Q 11	Class A	Q 6	6	2	Class 1
AQ12	Q 12	Class A	Q 5	5	1	†
AQ14	Q 14	†	Q 3	3	0	†

† Reliance quality higher than any equivalent in this specification.
Table applies to gears up to 50mm diameter.

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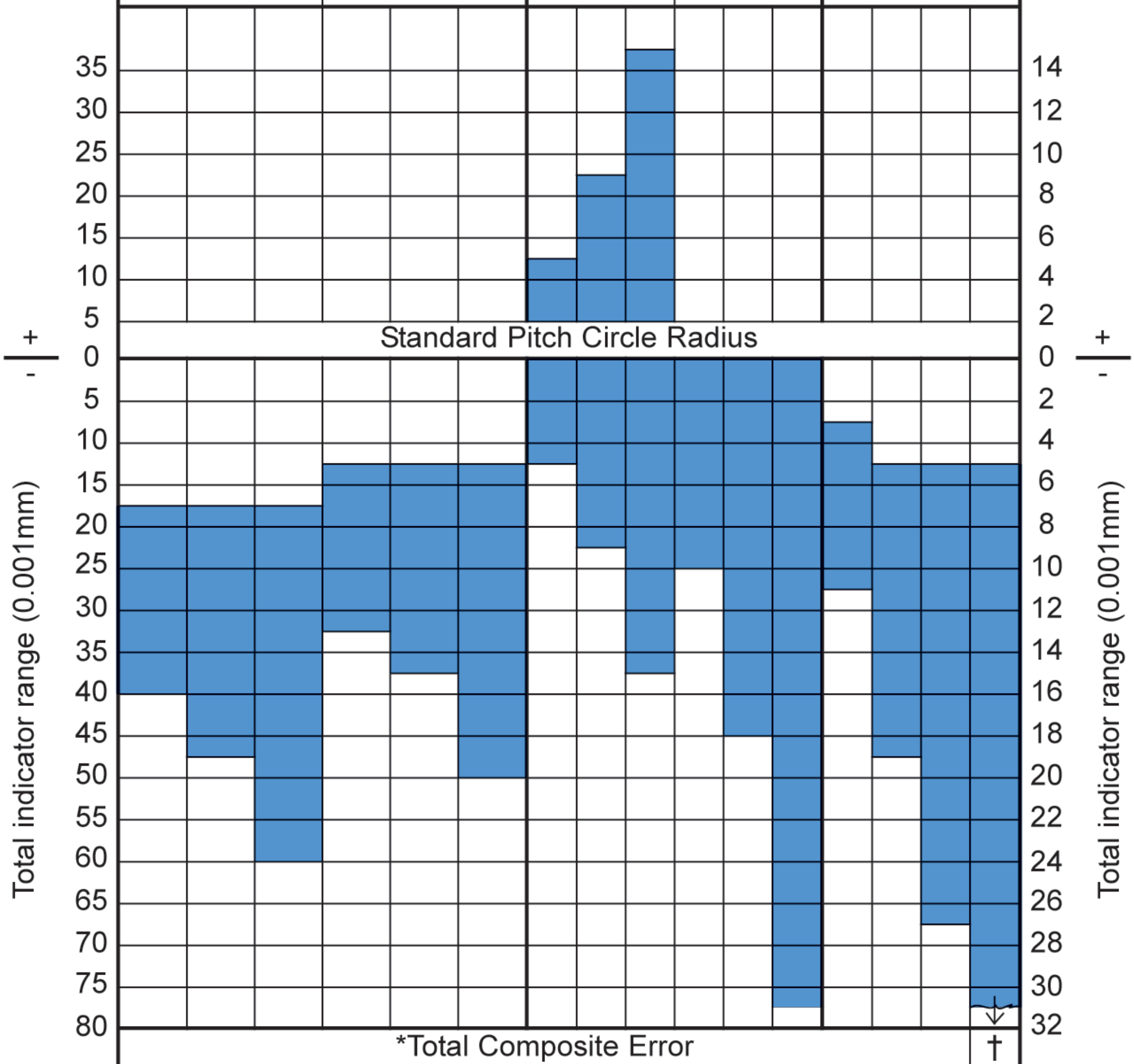
RELIANCE GEAR STANDARDS FOR FINE PITCH GEARS

The table below is a comparison between Reliance (AGMA) and equivalent UK specifications.

AGMA 390.03 1980 Backlash Designation "C"						Admiralty Standard BR 6001(1)						BS 4582 Pt1 (BS 978 Pt1)			
Quality Number						Class						Class			
14	12	10	14	12	10	1	2	3	1	2	3	A	B	C	D
0.8 to 0.5 mod			0.4 to 0.2 mod			Method 'A'			Method 'B'						



	14	12	10	14	12	10	1	2	3	1	2	3	A	B	C	D	
	*Total Composite Error															†	
(0.001mm)	7	13	26	7	13	26	18	31	48	18	31	48	15	25	40	100	
	2.7	5	10	2.7	5	10	7	12	19	7	12	19	6	10	16	40	(0.0001")
	*Tooth to Tooth Composite Error																
(0.001mm)	3.6	8	10	3.6	8	10	8	13	23	8	13	23	5	13	23	58	
	1.4	3	4	1.4	3	4	3	5	9	3	5	9	2	5	9	23	(0.0001")

*AGMA values quoted are for over 20T up to 50mm (2") diameter Admiralty & B.S. Tooth to tooth errors are for over 30T.
† Minimum indicator level 0.006" or 0.15mm.

For numbers of teeth outside the range consult the relevant specification.

STANDARD MODULES AND CIRCULAR PITCHES - METRIC

Reliance precision instrumentation spur gears are available as standard in the following modules and circular pitches, being those most commonly used in the design and manufacture of gear control mechanisms and instruments:

Module 0.2, 0.25, 0.3, 0.4, 0.5, 0.6.
Circular pitch 1, 2.

Pressure Angle and Rack Form

Except where stated otherwise, gears in this catalogue are cut to 20º pressure angle involute form teeth. Reliance standard gears will mesh satisfactorily with gears of the same module cut to the following standards:-

(i) BS 4582 (1970) Part 1, Figure 1.

(ii) DIN 867 and 58412.

(iii) AGMA 207.06 (Assuming the pitch is cut to an equivalent module)

The gears will not mesh satisfactorily with gears cut to DIN 58400 unless the outside diameter of the latter is reduced to PCD + (2 x module).

DIN 58400 tooth proportions are:

Addendum	1.1 x Module
Dedendum	1.5 x Module for pitch 0.1 to 0.6 Module

Metric Tooth Proportions (Dimensions in mm)

Module	Circular Pitch	Addendum	Dedendum	Working Depth	Whole Depth	Equivalent Inch Diametral Pitch
1.5	4.712	1.5	1.875	3.0	3.375	16.933
1.25	3.927	1.25	1.563	2.5	2.8125	20.300
1.0	3.142	1.0	1.400	2.0	2.4	25.400
0.8	2.513	0.8	1.120	1.6	1.92	31.750
0.6	1.885	0.6	0.840	1.2	1.44	42.333
0.5	1.571	0.5	0.700	1.0	1.2	50.800
0.4	1.257	0.4	0.560	0.8	0.96	63.500
0.3	0.942	0.3	0.420	0.6	0.72	84.667
0.25	0.785	0.25	0.350	0.5	0.6	101.600
0.2	0.628	0.2	0.280	0.4	0.48	127.000
0.318	1.0	0.318	0.446	0.637	0.764	79.796
0.637	2.0	0.637	0.891	1.273	1.528	39.898
0.796	2.5	0.796	1.114	1.592	1.910	31.919
0.955	3.0	0.955	1.337	1.910	2.292	26.599

The above list is by no means exhaustive. Please enquire if you require a special module as Reliance holds a large stock of non-standard cutters.

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MATERIALS AND SPECIFICATIONS

Reliance precision instrumentation gears are manufactured from the materials listed below. We reserve the right to change the actual material to an equivalent specification without notice depending on availability.

Reliance Standard Gear Materials

Material	Specification		Used on	Material Code
Stainless steel	303S31 (303S21) or 303S42 (303S42) or 302S31 (302S25) or 303 to MIL QQ-S-764	BS 970	Pin hub gears Clamp hub gears Hubless gears Worms Gear clamp & hubs	S1
Stainless steel (hardened)	17-4PH hardened to 36-40 HRc		Hardened pin hub gears	S9
Aluminium alloy	L168 or HE 15-TF or 2024-T4 to MIL QQ-A-225/6	BS 1474	Pin hub gears Clamp hub gears Hubless gears Gear clamp & hubs	A1
Phosphor bronze	PB 102	BS 2874	Worm wheels	B1
Brass (Naval)	Alloy 464 to MIL QQ-B-637		Worm wheels	B3

Finishes

Stainless steel, bronze and brass gears remain in their natural condition. Passivation to DEF STAN 03-2, process M can be carried out if required. Aluminium components are anodised to specification DEF STAN, 03-24 (chromic acid process) or DEF STAN 03-25 (sulphuric acid process). Gear teeth are not normally anodised.

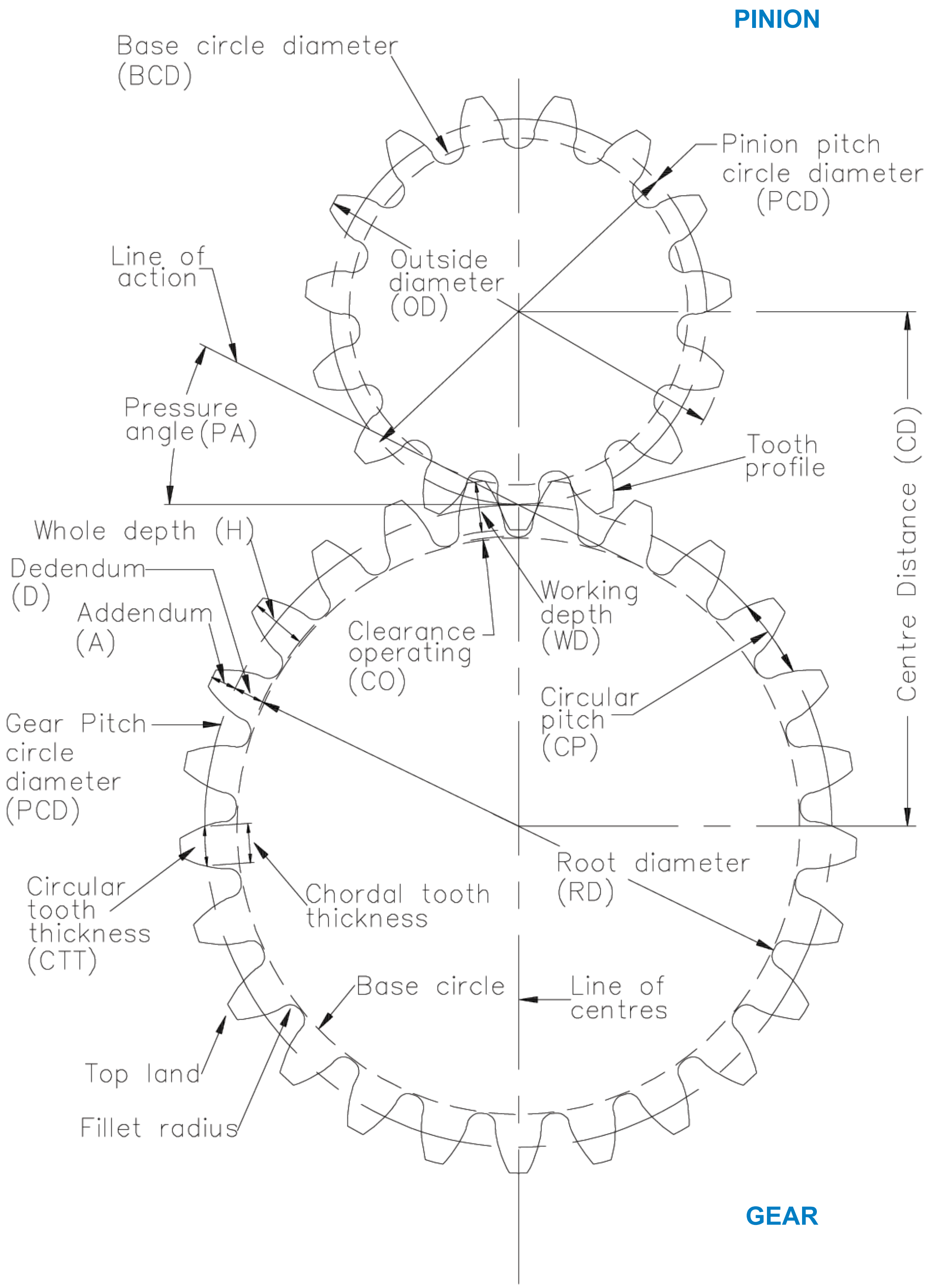
Anti-backlash Gears

Materials and finishes of standard anti-backlash gear components.

Where possible, circlips, anti-backlash springs, shims and set screws will be stainless steel. However some smaller pinions may have beryllium copper or zinc plated carbon steel circlips as standard.

SPUR GEAR GEOMETRY

A basic description of gear tooth terms is shown below. General formulae to enable correct understanding of spur gear geometry is shown overleaf.



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TERMINOLOGY FOR METRIC SPUR GEARS

TERM	DEFINITION	FORMULAE
Addendum (A)	The radial distance between the pitch circle and the outside diameter.	A=M
Addendum modification (K)	A method of modifying low tooth number gears to avoid undercutting and alter gear size to allow use of non standard centres.	See page 63
Backlash (B)	The circumferential clearance between mating gear teeth.	See page 70
Base circle diameter (BCD)	The diameter of the base cylinder from which the involute is generated.	BCD = N·Mcos PA
Base pitch (BP)	The pitch along the base circle or line of action.	BP = π Mcos PA
Basic rack	The straight sided rack shape from which teeth are generated.	See BS 4582.
Centre distance (CD)	Distance between the axes of rotation of mating spur gears.	$CD = \frac{PCD_{pinion}}{2} + \frac{PCD_{gear}}{2}$
Circular pitch (CP)	The distance along the pitch circle between corresponding points on adjacent teeth.	CP = π M
Circular tooth thickness (CTT)	The distance between opposite faces on the same tooth, measured at the pitch circle diameter.	$CTT = \frac{\pi M}{2}$
Clearance operating (CO)	The amount by which the dedendum in a given gear exceeds the addendum of the mating gear.	CO = D - A
Dedendum (D)	The radial distance between the pitch circle and the root diameter.	D = 1.4M
Diametral pitch (DP)	The size of the tooth expressed in teeth per inch of pitch diameter.	
Face width	The width of the tooth in an axial plane.	
Fillet radius	The radius of the fillet curve at the base of the gear tooth.	
Length of action	The distance on an involute line of action through which the point of contact moves during the action of the tooth profiles.	

Indicator limits	The size band of manufacture for the gear measured radially from the PCD.	
Module (M)	The size of the tooth expressed in mm of pitch diameter.	
Number of teeth (N)	Number of teeth on the gear.	
Outside diameter (OD)	The diameter over the tops of the teeth.	OD = PCD + (2M)
Pitch circle diameter (PCD)	An imaginary circle whose diameter is formed by meshing gears so that the circles actually touch each other, as if gears were driven purely by the friction of the circles.	PCD = (N+2K)·M Note: for unmodified gears K=0
Pressure angle (PA)	The angle between a line tangential to the pitch circles and a line perpendicular to the tooth profiles at the point of contact. (Equal to the side angle of the basic rack for standard gears).	Standard = 20º
Root diameter (RD)	The diameter of the base of the teeth.	RD = OD - (2H)
Total composite error (TCE)	The total error in the gear measured by the dual flank gear test. TTCE and pitch line runout are included.	
Tooth to tooth composite error (TTCE)	The change in error of each tooth on the gear measured by the dual flank tester.	
Undercut	The loss of profile in the vicinity of the involute start at the base circle due to tool cutter action generating gear with low tooth numbers. (N_{min} = minimum teeth for no undercut)	$N_{min} = \frac{2}{\sin^2 PA}$
Whole depth (H)	The total depth of a tooth space.	H = A + D
Working depth (WD)	The depth of engagement between mating gear teeth.	WD = 2A

Note: for imperial gears to BS 978 Part 1, Equivalent Module = $\frac{25.4}{DP}$

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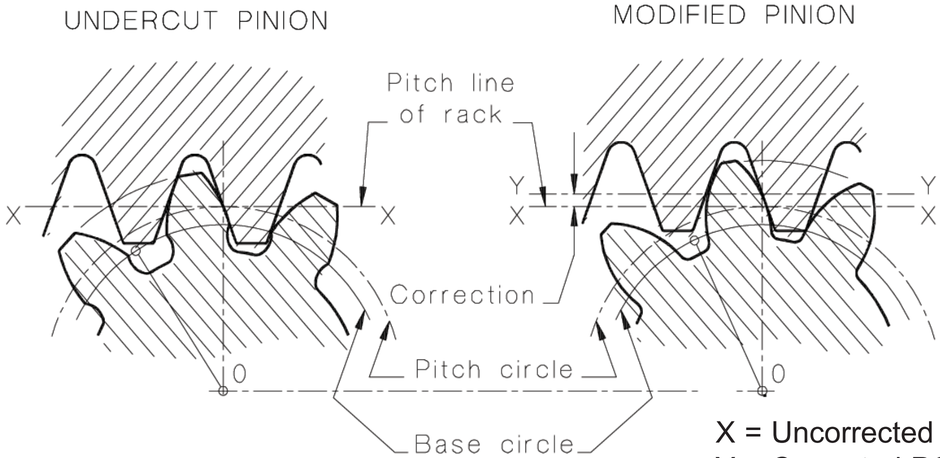
GEARS WITH SMALL NUMBER OF TEETH

Unless otherwise requested, all gears in this catalogue having 16 teeth or under will be enlarged by applying addendum modification in accordance with BS4582 Part 1 (metric) as shown in the table below. These gears are indicated (†) against the appropriate tooth numbers on the product pages.

A small amount of backlash will be introduced between corrected pinions and mating gears when the modification sum is other than zero and the nominal centre distance is adjusted only by an amount equal to the modification sum.

For minimum backlash it will be necessary to either reduce the centre distance further, or to apply a secondary correction to the pinion or wheel. See the above B.S specification for details



X = Uncorrected PCD
Y = Corrected PCD

Data for Addendum Modified Gears of Unit Module and Unit DP			
No. of Teeth	**Addendum Modification**	**Enlarged PCD**	**Enlarged OD (PCD+2)**
10	0.4151	10.8302	12.8302
11	0.3566	11.7132	13.7132
12	0.2982	12.5964	14.5964
13	0.2397	13.4794	15.4794
14	0.1812	14.3624	16.3624
15	0.1227	15.2454	17.2454
16	0.0642	16.1284	18.1284

Example. (Module)
Find P.C.D. and O.D. of enlarged gear having 13 teeth, 0.6 module

P.C.D. =13.4794 (from table) x 0.6 module =8.088mm
(Standard P.C.D. would be 13 x 0.6 =7.800mm)

Similarly,

O.D. =15.4794 (from table) x 0.6 module = 9.288mm
(Standard O.D. would be 7.8 + (2 x 0.6) = 9.00mm)

Note:
For Imperial (diametral pitch) gears, divide the PCD or OD value in the table by the diametral pitch. The answer will be in inches.

ENGINEERING DATA

For instrumentation Reliance normally recommend stainless steel pinions mating with aluminium alloy gears. Generally the pinion is subjected to most wear since it experiences a higher number of stress cycles than the wheel. This combination of materials tends to balance the wear between the pinion and the gear.

1. Gear materials

Stainless steel
The 300 series stainless steels are used for gears when maximum corrosion resistance is required. They are 'true' stainless steels containing 18% chromium and 8% nickel. Gears made from 303 stainless steel are essentially nonmagnetic and cannot be hardened by heat treatment. They are recommended for low torque applications as their mechanical properties and resistance are low.

Aluminium alloy
Gears made from aluminium alloy are widely used in measuring applications. Its light weight offers reduced inertia. The inertia of an aluminium alloy gear is approximately 35% that of a steel gear. In particular, aluminium alloy L168 offers excellent corrosion resistance when anodised, moderately good mechanical properties and good stability.

Hardened stainless steel
17-4PH is a precipitation hardening stainless steel that offers a remarkable combination of high strength and hardness. Its high chromium content, (15-17.5%), makes it an excellent material for arduous environments.

Phosphor bronze
As a gear material phosphor bronze has a fine grain and good resistance to tooth sliding wear hence its use as a worm wheel material.

2. Installation

Gears in this catalogue are designed to be a slide fit on the shafts. The gears are available with four fixing methods: standard clamp, Reli-a-Grip™ clamp, pins and set screws.

Traditional clamp hub style gears have a gear hub with a relatively thin wall partially split. The clamp is a close fit on the hub and is compressed when the clamp screw is tightened. Clamping gears on to the shaft offers extremely easy assembly with the best assembled accuracy. However, as the fastening depends upon friction it can only be used in low torque applications.

Reli-a-Grip™ clamps are designed as an integral clamp alternative to the standard separate gear clamp, reducing component count and improving ease of assembly yet maintaining a high level of assembled accuracy. When the clamp screw is tightened the clamp deflects in such a manner providing an even pressure over the contact area. As with the standard clamp, this fastening depends on friction and should only be used in lower torque applications.

Pin type gears are supplied as standard with a set screw and a sub-drilled hole. The set screw should be used to position the gear on the shaft during the drilling and pinning operation and can be removed once the gear is secure.

The sub-drilled hole provides a lead in for the drilling operation. It is recommended that drilling and pinning is completed outside the gear box and the gear is thoroughly cleaned afterwards.

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In less critical applications the set screw may be used to retain the gear on the shaft. To avoid damaging the shaft and to make removal of the gear easier the set screw should seat on a small flat, or dimple on the shaft.

3. Lubrication

All gears should be lubricated, but there are variations in degree.

Highly loaded precision gears should be in enclosed assembles with complete lubrication to obtain the best possible hydrodynamic film. The system can be splash, spray or force fed, depending on the application. Moderately loaded precision gears, such as fractional horsepower systems, should be enclosed with oil or grease lubrication which can be spread by splash or dip lubrication.

Lightly loaded gears in instrumentation systems only need to have a marginal boundary lubrication as provided by periodically wiped on oils or grease. In many instances a light coat of Rocol MT-LM or similar molybdenum disulphide grease will suffice for the life of the system. Anti-backlash gears should not be directly lubricated except via a very light application on the mating pinion.

Negligibly loaded fine instrument gears only need a brushed on film of light oil as a simple means of reducing friction.

4. Speed

The maximum pitch line velocity for stainless steel meshing with aluminium alloy with boundary lubrication is approximately 5,300 mm/sec, (for a pair of meshing actuation gears correctly lubricated, this rises to approximately 8,000 mm/sec). This represents 5,000 rpm on measurement gears of 20mm diameter, (and 7,500 rpm on actuation gears of 20mm diameter).

For speeds in excess of this and other material combinations please consult Reliance technical sales.

5. Gear Loading

The gears in this catalogue can be used for both feedback and actuation systems. The loads and material selection will depend on the application. In general a feedback system is designed to maintain accuracy and an actuation system is designed to transmit power.

Actuation Gears

The following analysis is intended to give a guide to the load capacity of a pair of spur gears. To simplify the calculations, a number of assumptions have been made. It must be noted that in many applications this will give a conservative estimate of the gear capacity, therefore, in critical applications an exact analysis must be completed. Please consult the relevant gear standards or Reliance technical sales.

The analysis is based on AGMA 2001-B88 and assumes the following:

1. The gears are simply supported in rolling element bearings.
2. Pinion revolutions $>10^7$.
3. Gears are grease lubricated.
4. Reliability of 1 failure in 100 is acceptable.
5. Gear material is 17-4PH hardened.

The basic load capacity (F_b) of a pair of spur gears is defined as the maximum tangential force at which they can operate indefinitely.

F_b has two values: one calculated from tooth root strength, (F_{bs}) and one for tooth flank pitting (F_{bw}). The useful or transmitted load capacity, F_t, is usually less than F_b due to transient or dynamic loads generated within the mechanism.

For tooth root strength	$F_{ts} = F_{bs}/K_a$	K_a & C_a = Application factors
For tooth flank pitting (wear)	$F_{tw} = F_{bw}/C_a$	

Both calculation should be made and the lower value used.

The application factors K_a and C_a make allowance for any externally applied loads in excess of the nominal tangential force Fb and they are most accurately determined by direct measurement. In determining application factors, consideration should be given to the fact that many prime movers develop momentary peak torques appreciably greater than those determined by the nominal ratings of either the prime mover or the driven equipment. There are many possible sources of overload which should be considered, including system vibrations, acceleration torques, overspeeds, variations in system operation and changes in process load. Impact loads due to reversing across backlash can be significant in servo systems.

As a general guide application factors for a motor gear system range from 1.0 for uniform loads up to 1.75 where heavy shock loads are anticipated.

For strength $F_{bs} = 177.7 \times J \times F \times M \times K$ [N]

For wear $F_{bw} = 14.64 \times N \times I \times F \times M \times K_v$ [N]

F_b = Basic load capacity (F_{bs} & F_{bw})
N = Number of teeth
J = Geometry factor, strength
I = Geometry factor, wear
F = Face width of smallest gear
M = Module
K_v = Dynamic factor

(i) Number or teeth - this is the number of teeth in the gear being analysed.

(ii) Geometry factors, I and J
These factors take account of the effect of tooth proportions on stress. The bending strength geometry factor, (J) takes account of the shape of the tooth. The wear resistance geometry factor, (I) takes account of the radii of curvature of the contacting tooth profiles. Please see the graphs on page 67 and 68.

(iii) Face width, F
This is the face width of the smallest gear in mm.

(iv) Module, M
This is the gear module expressed as shown on the respective gear pages.

(v) Dynamic factor, Kv
This accounts for internally generated gear tooth loads which are induced by the non-conjugate meshing action of the gear teeth.

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$$K_v = \left(\frac{84}{84 + \sqrt{200V_t}} \right)^{0.4}$$

For quality 10 gears only
V_t = Pitch line velocity (m/s)

Example calculation to find the theoretical load capacity of a 5:1 pass of 17-4PH spur gears as follows:

Pinion - P06S9B6F4A-25
Gear - P06S9B8F6A-125

Pinion speed is 500rpm.

(i) Number of teeth from part number = 25

(ii) Geometry factors from graph

J = 0.37
I = 0.118

(iii) Smallest gear face width from part number

F = 4

(iv) Gear module from part number

M = 0.6

(v) Dynamic factor from equation

$$K_v = \left(\frac{84}{84 + \sqrt{200V_t}} \right)^{0.4}$$

$$\text{where : } V_t = \frac{\text{rpm} \times \pi \times N \times M}{60000} \left[\text{m/s} \right]$$

F_{bs} = 177.7 x 0.37 x 4 x 0.6 x 0.96 = 151.5N

F_{bw} = 14.64 x 25 x 0.118 x 4 x 0.6 x 0.96 = 99.5N

For alternative materials the above values need to be modified as shown below.

Gear Material Modification Factors

Material	Specification	Strength	Wear
Hardened Stainless steel	17-4PH	1.00	1.00
Stainless steel	303S31	0.43	0.15
Stainless steel	316S31	0.47	0.20
Aluminium alloy	L168	0.37	0.10
Brass	CZ121	0.35	0.13

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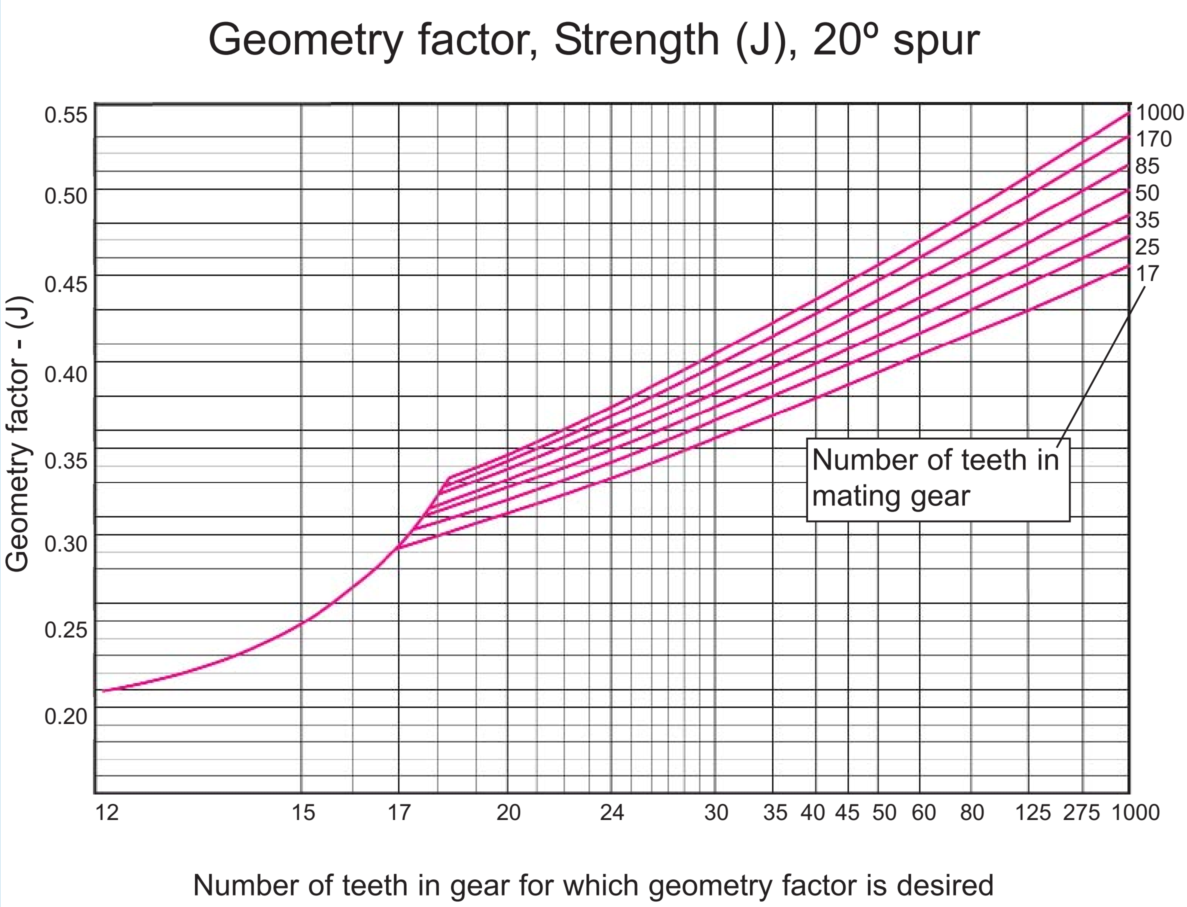
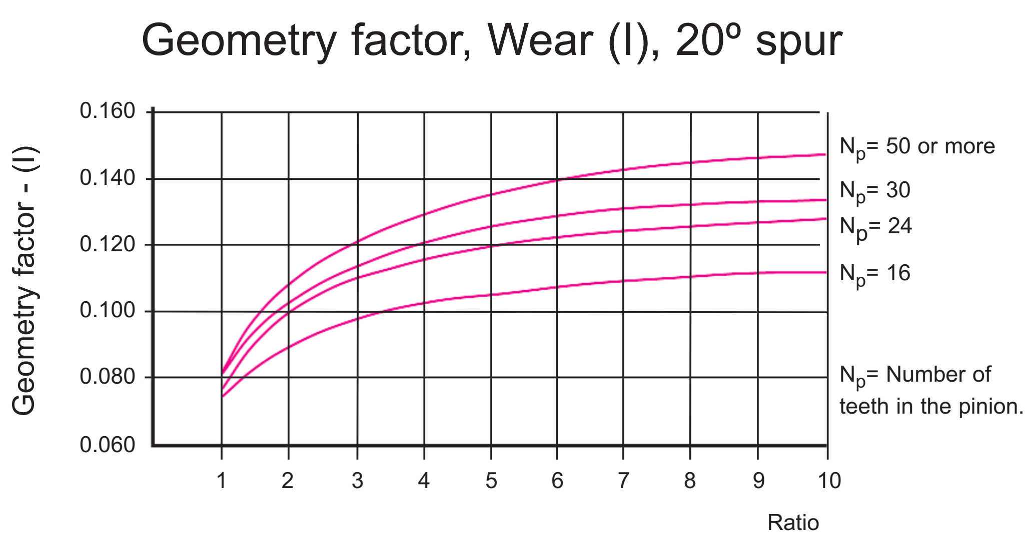
Example:
If the gears in the example on the previous page were made from 303S31

Fbs = 151.5 x 0.43 = 65.1N
Fbw = 99.5 x 0.15 = 14.9N

The application factors should be applied after the reduction for material.





Instrumentation and Feedback Systems

Gears and components designed for the precise transmission of angular position generally work at the low torque levels normally associated with servo components such as synchros, resolvers, optical encoders etc. Tooth loads of 1.2N per mm face width should result in an adequate accurate life. Higher loads will tend to increase deflections of gear teeth, shafts, bearings etc, resulting in significant values of lost motion and a decrease in life.

eg. To find the maximum advisable torque on a gear 40mm diameter x 3mm face width.

Torque = force x radius = 1.2 x 3 x 0.04/2 = 0.072Nm (10oz.in.)

Anti-backlash Gear Spring Tension

In order for anti-backlash gears to function as anti-backlash devices, it is necessary to ensure that the spring tension will provide sufficient torque to overcome the friction and acceleration torque in the system, ie the spring torque must be capable of driving/accelerating the gear train and any associated components.

The spring tension capability of anti-backlash gears listed in this catalogue will adequately cope with the low torques normally encountered.

As a general guide, torque settings on anti-backlash gears of 108 to 180 gmcm (1.5 to 2.5 oz.in.) will suffice in most applications. Ideally the spring torque should be set to the minimum at which the anti-backlash gear performs satisfactory, thus avoiding unnecessary high preload on the gear teeth and premature wear.

6. Lost Motion and Backlash Control

The following section deals with lost motion, which we know to be one of the basic problems in designing fine pitch gear trains. The accepted definition of lost motion is the amount by which the output shaft may be turned without turning the input shaft.

It may be thought that lost motion is a function of the gear cutting operation alone, but, in fact the teeth of the gears may contribute very little to the overall lost motion value. A complete understanding of all the elements which induce lost motion is essential in order to achieve a well designed gear train. The following factors must be individually considered for their own contribution to overall lost motion in the gear train.

(a) Nominal centre distance.

(b) Centre distance tolerance.

(c) Size and tolerance of mating gears.

(d) Total composite error of gears.

(e) Fits between bores, shafts and bearings.

(f) Bearing accuracy (class).

(g) Radial play of bearings.

(h) Shaft straightness and alignment.

(i) Fits between electrical and/or mechanical component spigot diameters, and housing bores.

(j) Eccentricity and radial play of electrical and/or mechanical component shafts.

(k) Torsional elasticity.

(l) Differential expansion.

Each of the above, except nominal centre distance, tend to induce a change in centre distance which will push together or pull apart the mating gears. This push-pull action produces two backlash values, minimum at the point of the tightest mesh, and maximum at the point of loosest mesh.

a) Calculation of Nominal Centre Distance

Nominal centre distance can be considered as the starting point in the calculation of overall backlash values. Nominal centre distance is calculated by taking half the sum of the (theoretical) pitch diameters of the mating gears.

i.e. $$CD = \frac{PCD_1 + PCD_2}{2}$$

(b) Centre Distance Tolerance

Centre distance tolerance is an extremely important area for consideration. Any increase in centre distance in excess of the nominal value will increase the backlash. A decrease in nominal centre distance will decrease the backlash. In this case caution must be exercised to avoid interference between mating gears as a result of this decrease.

The relationship between centre distance change to backlash for 20º PA spur gear is given by:

$B = 2\,\mathrm{Tan}\,ø \cdot \Delta C$ where

B = Circumferential backlash
$ø$ = Pressure angle (Tan 20º = 0.36397)
ΔC = Distance between theoretical nominal and actual centre distance

Note: Maximum Angular Backlash (minutes of arc) = $\dfrac{\text{Maximum Circumferential Backlash} \times 57.3 \times 60}{\text{Pitch Circle Radius}}$

7. Gear Error

The error function of a gear is approximately sinusoidal and for practical considerations can be assumed to be so. The first derivative of the time displacement curve yields the velocity function, therefore, the output velocity variable will also be an approximate sinusoid but the maximum velocity error will be displaced 90º from the maximum position error.

In summation, pitch circle runout will cause a sinusoidal error which is revealed as an output transmission error when meshed with a mating gear. The magnitude is given by the following example:

In the example on page 71, if the small pinion were not a perfect gear its error would be seen superimposed on the large gear error cycling at pinion frequency.

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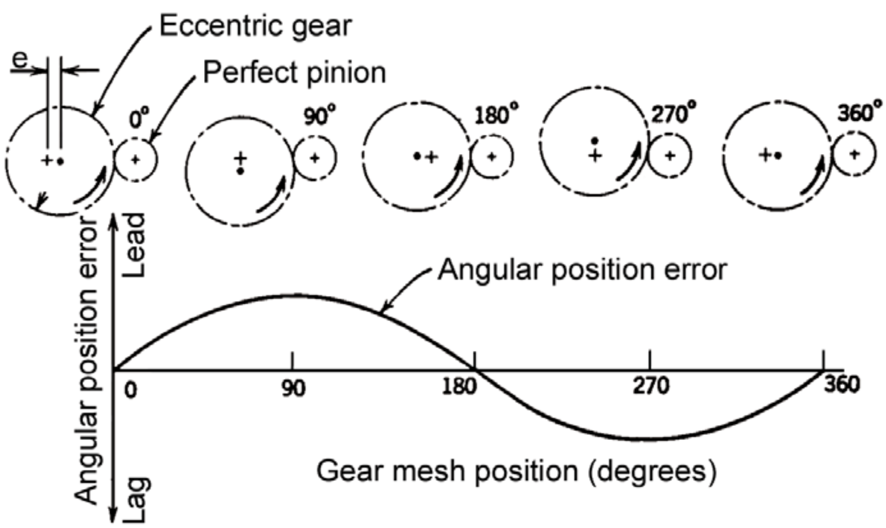


Angular position error $E_A = \frac{e}{R}\sin\theta$ Linear position error $E_L = e\sin\theta$

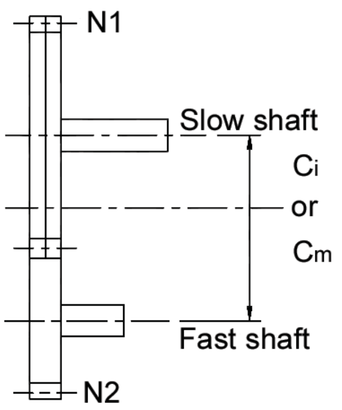
8. Transmission Accuracy of Gear Trains

The following section is based on work carried out by Reliance Gear Company to provide some guidance in the design of accurate data transmission gearing.

The transmission error referred to by equations 1 and 2 below represent the maximum statistical point to point error during a forward and reverse cycle of a single pass of quality 14 anti-backlash gearing assembled in a data transmission gearbox.

For quality 10 or 12 gearing add 50% or 30% respectively to the error calculated for quality 14 gearing.

For average transmission error substitute the numbers 3.25 and 83 in the equations for the numbers 4.4 and 112 respectively.



$$\Sigma_1 = \left(1 + \frac{N_2}{N_1}\right)\cdot\left(\frac{112}{C_m} \text{ or } \frac{4.4}{C_i}\right) \text{------} (1)$$

$$\Sigma_2 = \left(1 + \frac{N_1}{N_2}\right)\cdot\left(\frac{112}{C_m} \text{ or } \frac{4.4}{C_i}\right) \text{------} (2)$$

C_m and C_i = Centre distance in mm and inches respectively.

N_2 and N_1 = Number of teeth in pinion and wheel respectively.

Σ_1 and Σ_2 = Maximum statistical transmission error in minutes of arc measured at the slow and fast shafts respectively.