1.3.4 Conversion

For those wishing to ease themselves into working with metric gears by looking at them in terms of familiar inch gearing relationships and mathematics, Table 1-5 is offered as a means to make a quick comparison.

Table 1-5 Spur Gear Design Formulas

To Obtain	From Known	Use This Formula*
Pitch Diameter	Module	$D = mN$
Circular Pitch	Module	$P_c = m\pi = \frac{D}{N}\pi$
Module	Diametral Pitch	$m = \frac{25.4}{P_d}$
Number of Teeth	Module and Pitch Diameter	$N = \frac{D}{m}$
Addendum	Module	$a = m$
Dedendum	Module	$b = 1.25m$
Outside Diameter	Module and Pitch Diameter or Number of Teeth	$D_o = D + 2m = m(N + 2)$
Root Diameter	Pitch Diameter and Module	$D_R = D - 2.5m$
Base Circle Diameter	Pitch Diameter and Pressure Angle	$D_b = D \cos \phi$
Base Pitch	Module and Pressure Angle	$P_b = m \pi \cos \phi$
Tooth Thickness at Standard Pitch Diameter	Module	$T_{std} = \frac{\pi}{2} m$
Center Distance	Module and Number of Teeth	$C = \frac{m(N_1 + N_2)}{2}$
Contact Ratio	Outside Radii, Base Circle Radii Center Distance, Pressure Angle	$m_p = \frac{({}_1R_o - {}_1R_b)^{1/2} + ({}_2R_o - {}_2R_b)^{1/2} - C\sin\phi}{m \pi \cos \phi}$
Backlash (linear)	Change in Center Distance	$B = 2(\Delta C)\tan \phi$
Backlash (linear)	Change in Tooth Thickness	$B = \Delta T$
Backlash (linear) Along Line-of-action	Linear Backlash Along Pitch Circle	$B_{LA} = B \cos \phi$
Backlash, Angular	Linear Backlash	${}_aB = 6880 \frac{B}{D}$ (arc minutes)
Mm. No. of Teeth for No Undercutting	Pressure Angle	$N_c = \frac{2}{\sin^2\phi}$

* All linear dimensions in millimeters
Symbols per Table 1-4

SECTION 2 INTRODUCTION TO GEAR TECHNOLOGY

This section presents a technical coverage of gear fundamentals. It is intended as a broad coverage written in a manner that is easy to follow and to understand by anyone interested in knowing how gear systems function. Since gearing involves specialty components, it is expected that not all designers and engineers possess or have been exposed to every aspect of this subject. However, for proper use of gear components and design of gear systems it is essential to have a minimum understanding of gear basics and a reference source for details.

For those to whom this is their first encounter with gear component it is suggested this technical treatise be read in the order presented so as to obtain a logical development of the subject. Subsequently, and for those already familiar with gears, this material can be used selectively in random access as a design reference.

2.1 Basic Geometry Of Spur Gears

The fundamentals of gearing are illustrated through the spur gear tooth, both because it is the simplest, and hence most comprehensible, and because it is the form most widely used, particularly for instruments and control systems.

The basic geometry and nomenclature of a spur gear mesh is shown in Figure 2-1. The essential features of a gear mesh are:

1. Center distance.
2. The pitch circle diameters (or pitch diameters).
3. Size of teeth (or module).
4. Number of teeth.
5. Pressure angle of the contacting involutes.

Details of these items along with their interdependence and definitions are covered in subsequent paragraphs.

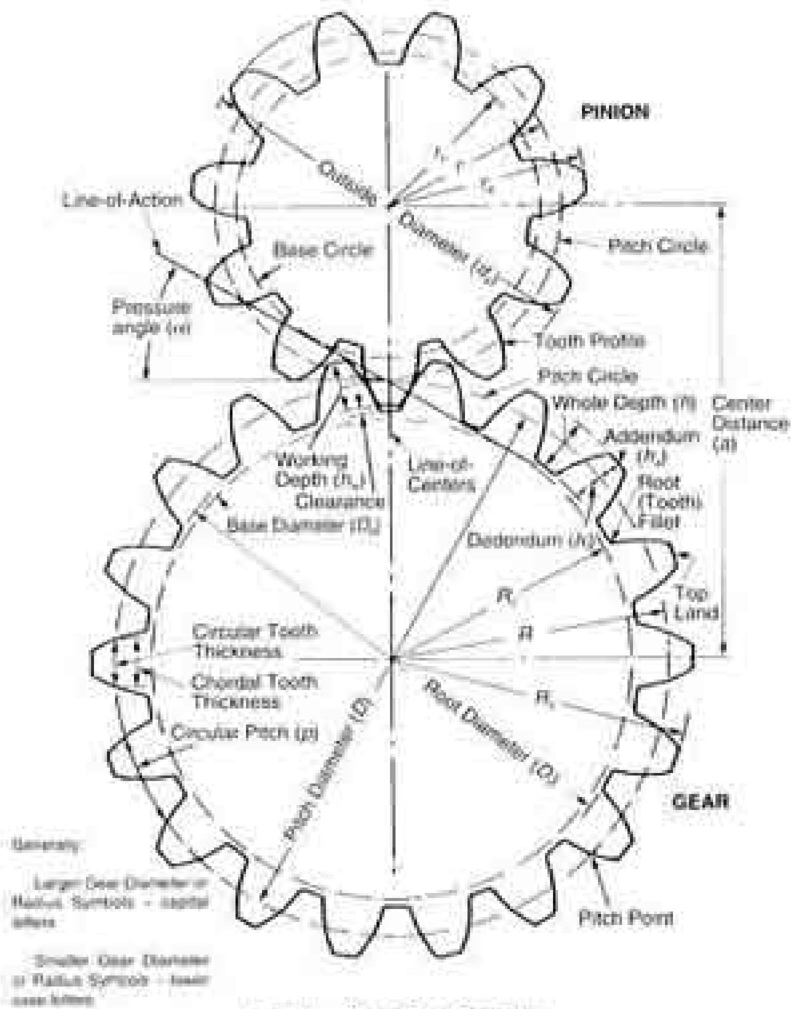


Fig. 2-1 Basic Gear Geometry

2.2 The Law Of Gearing

A primary requirement of gears is the constancy of angular velocities or proportionality of position transmission. Precision instruments require positioning fidelity. High-speed and/or high-power gear trains also require transmission at constant angular velocities in order to avoid severe dynamic problems.

Constant velocity (i.e., constant ratio) motion transmission is defined as "conjugate action" of the gear tooth profiles. A geometric relationship can be derived (2, 12)* for the form of the tooth profiles to provide conjugate action, which is summarized as the Law of Gearing as follows:

"A common normal to the tooth profiles at their point of in all positions of the contacting teeth, pass through a fixed point on the line-of-centers called the pitch point."

Any two curves or profiles engaging each other and satisfying the law of gearing are conjugate curves.

2.3 The Involute Curve

There is almost an infinite number of curves that can be developed to satisfy the law of gearing, and many different curve forms have been tried in the past. Modern gearing (except for clock gears) is based on involute teeth. This is due to three major advantages of the involute curve:

1. Conjugate action is independent of changes in center distance.

2. The form of the basic rack tooth is straight-sided, and therefore is relatively simple and can be accurately made; as a generating tool it imparts high accuracy to the cut gear tooth.

3. One cutter can generate all gear teeth numbers of the same pitch.

The involute curve is most easily understood as the trace of a point at the end of a taut string that unwinds from a cylinder. It is imagined that a point on a sting, which is pulled taut in a fixed direction, projects its trace onto a plane that rotates with the base circle. See Figure 2-2. The base cylinder, or base circle as referred to m gear literature, fully defines the form of the involute and in a gear it is an inherent parameter, though invisible.

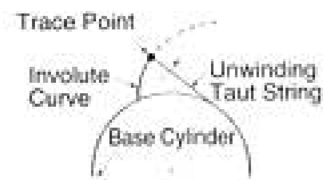
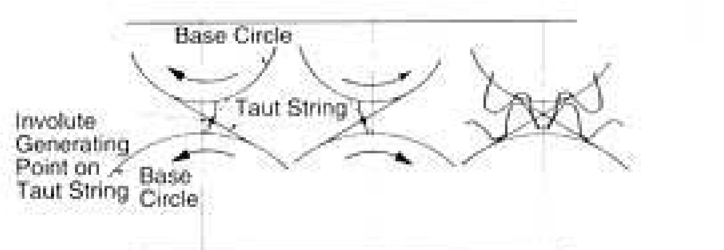


Fig. 2-2 Generation of an Involute by a Taut String

The development and action of mating teeth can be visualized by imagining the taut string as being unwound from one base circle and wound on to the other, as shown in Figure 2-3a. Thus, a single point on the string simultaneously traces an involute on each base circle's rotating plane. This pair of involutes is conjugate, since at all points of contact the common normal is the common tangent which passes through a fixed point on the line-of-centers. If a second winding/unwinding taut string is wound around the base circles in the opposite direction, Figure2-3b, oppositely curved involutes are generated which can accommodate. motion reversal. When the involute pairs are properly spaced, the result is the involute gear tooth, Figure 2-3c.



(a) Left-Hand Involutes (b) Right-Hand Involutes (c) Complete Teeth Generated by Two Crossed Generating Taut Strings contact must,

Fig. 2-3 Generation and Action of Gear Teeth

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* Numbers in parenthesis refer to references at end of text.

2.4 Pitch Circles

Referring to Figure 2-4, the tangent to the two base circles is the line of contact, or line-of-action in gear vernacular. Where this line crosses the line-of-centers establishes the pitch point, P. This in turn sets the size of the pitch circles or as commonly called, pitch diameters The ratio of the pitch diameters gives the velocity ratio:

Velocity ratio of gear 2 to gear 1 is:

$$i = \frac{d_1}{d_2} \qquad (2\text{-}1)$$

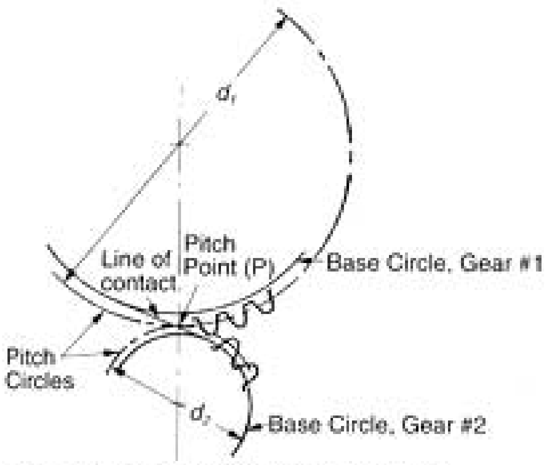

Fig. 2-4 Definition of Pitch Circle and Pitch Point

25 Pitch And Module

Essential to prescribing gear geometry is the size, or spacing of the teeth along the pitch circle. This is termed pitch, and there are two basic

Circular pitch - A naturally conceived linear measure along the pitch circle of the tooth spacing. Referring to Figure 2-5, it is the linear distance (measured along the pitch circle arc) between corresponding points of adjacent teeth. It is equal to the pitch-circle circumference divided the number of teeth:

p=circular pitch

$$= \frac{\text{Pitch Circle Circumference}}{\text{number of teeth}} = \frac{\pi d}{z} \qquad (2\text{-}2)$$

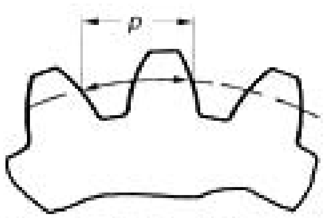

Fig. 2-5 Definition of Circular Pitch

Module - Metric gearing uses the quantity module m in place of the American inch unit, diametral pitch. The module is the length of pitch diameter per tooth. Thus:

$$m = \frac{d}{z} \qquad (2\text{-}3)$$

Relation of pitches: From the geometry that defines the two pitches, that shown that module and circular pitch are related by the expression:

module is as follows:

$$m = \frac{25.4}{P_d} \qquad (2\text{-}5)$$

2.6 Module Sizes And Standards

Module m represents the size of involute gear tooth. The unit of module is mm. Module is converted to circular pitch p, by the factor ,π.

$$p = \pi m \qquad (2\text{-}6)$$

Table 2-1 is extracted from JIS B 1701-1973 which defines the tooth profile and dimensions of involute gears. It divides the standard module into three series. Figure 2-6 shows the comparative size of various rack teeth.

Table 2-1 Standard Values of Module unit: mm

Series1	Series2	Series3	Series1	Series2	Series3
0.1					
				3.5	
	0.15				
0.2			4		3.75
	0.25			4.5	
0.3			5		
	0.35			5.5	
0.4			6		
	0.45				
0.5					6.5
				7	
			8		
	0.55				
0.6					
				9	
		0.65	10		
	0.7				
	0.75			11	
			12		
0.8					
	0.9			14	
			16		
1					
				18	
1.25			20		
1.5					
	1.75			22	
			25		
2					
	2.25			28	
			32		
2.5					
	2.75			36	
			40		
3					
				45	
		3.25	50		

Note: The preferred choices are in the series order beginning with 1.

Circular pitch, *p*, is also used to represent tooth size when a special desired spacing is wanted, such as to get an integral feed in a mechanism. In this case, a circular pitch is chosen that is an integer or a special fractional value. This is often the choice in designing position control systems. Another particular usage is the drive of printing plates to provide a given feed.

Most involute gear teeth have the standard whole depth and a standard pressure angle $\alpha = 20°$. Figure 2-7 shows the tooth profile of a whole depth standard rack tooth arid mating gear. It has an addendum of $h_a = 1m$ and dedendum $hf \geq 1.25m$. If tooth depth is shorter than whole depth it is called a stub tooth and it deeper than whole depth it is a "high" depth tooth.

The most widely used stub tooth has an addendum $h_a = 0.8m$ and dedendum $h_f = 1$ m. Stub teeth have more strength than a whole depth gear, but contact ratio is reduced. On the

$$\frac{P}{m} = \pi \qquad (2\text{-}4)$$

This relationship is simple to remember and permits an easy transformation from one to the other.

Diametral pitch P_d is widely used in England and America to resent the tooth size. The relation between diametral pitch and other hand, a high depth tooth can increase contact ratio, but weakens the tooth.

In the standard involute gear, pitch p times the number of teeth becomes the length of pitch circle:

$$d\pi = \pi m z$$

Pitch diameter d is then:

$$d = mz \qquad (2\text{-}7)$$

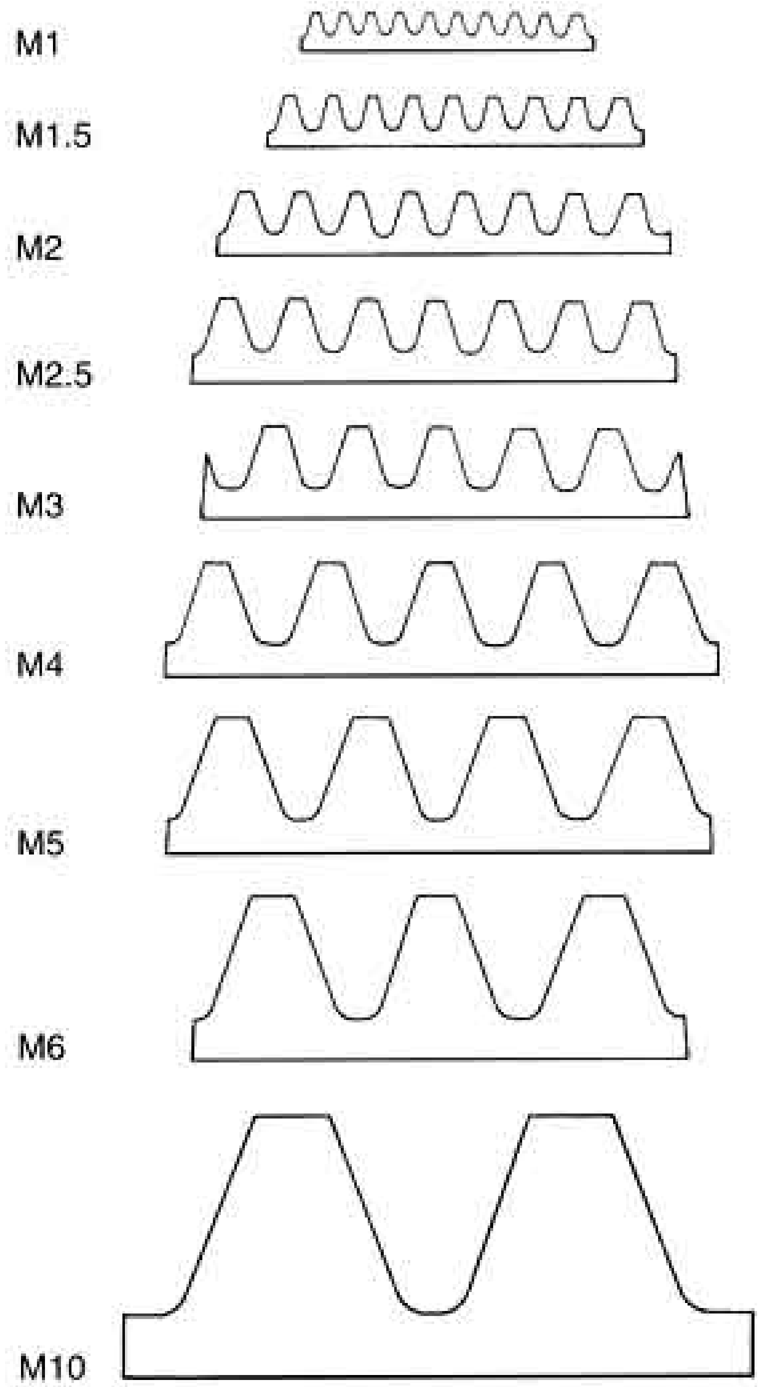

Fig. 2-6 Comparative Size of Various Rack Teeth

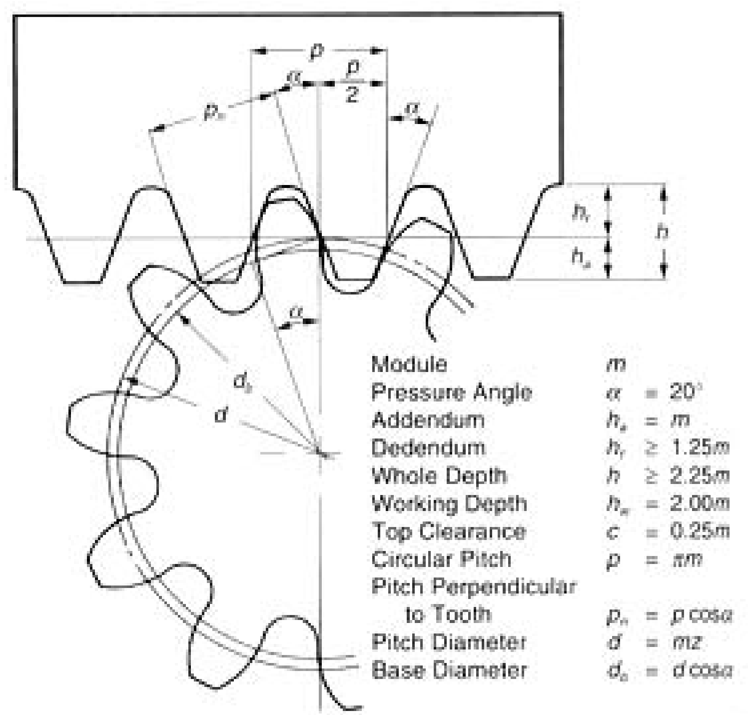

Fig. 2-7 The Tooth Profile and Dimension of Standard Rack

Metric Module and Inch Gear Preferences: Because there is no direct equivalence between the pitches in metric and inch systems, it is not possible to make direct substitutions. Further, there are preferred modules in the metric system. As an aid in using metric gears, Table 2-2 presents nearest equivalents for both systems, with the preferred sizes in bold type.

Diametral Pitch, P	Module, m	Circular Pitch		Circular Tooth Thickness		Addendum	
		in	mm	in	mm	in	mm
203.2000	0.125	0.0155	0.393	0.0077	0.196	0.0049	0.125
200	0.12700	0.0157	0.399	0.0079	0.199	0.0050	0.127
180	0.14111	0.0175	0.443	0.0087	0.222	0.0056	0.141
169.333	0.15	0.0186	0.471	0.0093	0.236	0.0059	0.150
150	0.16933	0.0209	0.532	0.0105	0.266	0.0067	0.169
127.000	**0.2**	0.0247	0.628	0.0124	0.314	0.0079	0.200
125	0.20320	0.0251	0.638	0.0126	0.319	0.0080	0.203
120	0.21167	0.0262	0.665	0.0131	0.332	0.0083	0.212
101.600	**0.25**	0.0309	0.785	0.0155	0.393	0.0098	0.250
96	0.26458	0.0327	0.831	0.0164	0.416	0.0104	0.265
92.3636	0.275	0.0340	0.864	0.0170	0.432	0.0108	0.275
84.6667	**0.3**	0.0371	0.942	0.0186	0.471	0.0118	0.300
80	0.31750	0.0393	0.997	0.0196	0.499	0.0125	0.318
78.1538	0.325	0.0402	1.021	0.0201	0.511	0.0128	0.325
72.5714	0.35	0.0433	1.100	0.0216	0.550	0.0138	0.350
72	0.35278	0.0436	1.108	0.0218	0.554	0.0139	0.353
67.733	0.375	0.0464	1.178	0.0232	0.589	0.0148	0.375
64	0.39688	0.0491	1.247	0.0245	0.623	0.0156	0.397
63.500	**0.4**	0.0495	1.257	0.0247	0.628	0.0157	0.400
50.800	**0.5**	0.0618	1.571	0.0309	0.785	0.0197	0.500
50	0.50800	0.0628	1.596	0.0314	0.798	0.0200	0.508
48	0.52917	0.0655	1.662	0.0327	0.831	0.0208	0.529
44	0.57727	0.0714	1.814	0.0357	0.907	0.0227	0.577
42.333	**0.6**	0.0742	1.885	0.0371	0.942	0.0236	0.600
40	0.63500	0.0785	1.995	0.0393	0.997	0.0250	0.635
36.2857	**0.7**	0.0866	2.199	0.0433	1.100	0.0276	0.700
36	0.70556	0.0873	2.217	0.0436	1.108	0.0278	0.706
33.8667	0.75	0.0928	2.356	0.0464	1.178	0.0295	0.750
32	0.79375	0.0982	2.494	0.0491	1.247	0.0313	0.794
31.7500	**0.8**	0.0989	2.513	0.0495	1.257	0.0315	0.800
30	0.84667	0.1047	2.660	0.0524	1.330	0.0333	0.847
28.2222	0.9	0.1113	2.827	0.0557	1.414	0.0354	0.900
28	0.90714	0.1122	2.850	0.0561	1.425	0.0357	0.907
25.4000	**1**	0.1237	3.142	0.0618	1.571	0.0394	1.000
24	1.0583	0.1309	3.325	0.0654	1.662	0.0417	1.058
22	1.1545	0.1428	3.627	0.0714	1.813	0.0455	1.155
20.3200	**1.25**	0.1546	3.927	0.0773	1.963	0.0492	1.250
20	1.2700	0.1571	3.990	0.0785	1.995	0.0500	1.270
18	1.4111	0.1745	4.433	0.0873	2.217	0.0556	1.411
16.9333	**1.5**	0.1855	4.712	0.0928	2.356	0.0591	1.500
16	1.5875	0.1963	4.987	0.0982	2.494	0.0625	1.588
15	1.6933	0.2094	5.320	0.1047	2.660	0.0667	1.693
14.5143	1.75	0.2164	5.498	0.1082	2.749	0.0689	1.750
14	1.8143	0.2244	5.700	0.1122	2.850	0.0714	1.814
13	1.9538	0.2417	6.138	0.1208	3.069	0.0769	1.954
12.7000	**2**	0.2474	6.283	0.1237	3.142	0.0787	2.000
12	2.1167	0.2618	6.650	0.1309	3.325	0.0833	2.117
11.2889	2.25	0.2783	7.069	0.1391	3.534	0.0886	2.250
11	2.3091	0.2856	7.254	0.1428	3.627	0.0909	2.309
10.1600	**2.50**	0.3092	7.854	0.1546	3.927	0.0984	2.500
10	2.5400	0.3142	7.980	0.1571	3.990	0.1000	2.540

Diametral Pitch, P	Module, m	Circular Pitch		Circular Tooth Thickness		Addendum	
		in	mm	in	mm	in	mm
9.2364	2.75	0.3401	8.639	0.1701	4.320	0.1083	2.750
9	2.8222	0.3491	8.866	0.1745	4.433	0.1111	2.822
8.4667	**3**	0.3711	9.425	0.1855	4.712	0.1181	3.000
8	3.1750	0.3927	9.975	0.1963	4.987	0.1250	3.175
7.8154	3.25	0.4020	10.210	0.2010	5.105	0.1280	3.250
7.2571	3.5	0.4329	10.996	0.2164	5.498	0.1378	3.500
7	3.6286	0.4488	11.400	0.2244	5.700	0.1429	3.629
6.7733	3.75	0.4638	11.781	0.2319	5.890	0.1476	3.750
6.3500	**4**	0.4947	12.566	0.2474	6.283	0.1575	4.000
6	4.2333	0.5236	13.299	0.2618	6.650	0.1667	4.233
5.6444	4.5	0.5566	14.137	0.2783	7.069	0.1772	4.500
5.3474	4.75	0.5875	14.923	0.2938	7.461	0.1870	4.750
5.0800	**5**	0.6184	15.708	0.3092	7.854	0.1969	5.000
5	5.0800	0.6283	15.959	0.3142	7.980	0.2000	5.080
4.6182	5.5000	0.6803	17.279	0.3401	8.639	0.2165	5.500
4.2333	**6**	0.7421	18.850	0.3711	9.425	0.2362	6.000
4	6.3500	0.7854	19.949	0.3927	9.975	0.2500	6.350
3.9077	6.5000	0.8040	20.420	0.4020	10.210	0.2559	6.500
3.6286	7	0.8658	21.991	0.4329	10.996	0.2756	7.000
3.5000	7.2571	0.8976	22.799	0.4488	11.399	0.2857	7.257
3.1750	**8**	0.9895	25.133	0.4947	12.566	0.3150	8.000
3.1416	8.0851	1.0000	25.400	0.5000	12.700	0.3183	8.085
3	8.4667	1.0472	26.599	0.5236	13.299	0.3333	8.467
2.8222	9	1.1132	28.274	0.5566	14.137	0.3543	9.000
2.5400	**10**	1.2368	31.416	0.6184	15.708	0.3937	10.000
2.5000	10.160	1.2566	31.919	0.6283	15.959	0.4000	10.160
2.3091	11	1.3605	34.558	0.6803	17.279	0.4331	11.000
2.1167	**12**	1.4842	37.699	0.7421	18.850	0.4724	12.000
2	12.700	1.5708	39.898	0.7854	19.949	0.5000	12.700
1.8143	14	1.7316	43.982	0.8658	21.991	0.5512	14.000
1.5875	**16**	1.9790	50.265	0.9895	25.133	0.6299	16.000
1.5000	16.933	2.0944	53.198	1.0472	26.599	0.6667	16.933
1.4111	18	2.2263	56.549	1.1132	28.274	0.7087	18.000
1.2700	**20**	2.4737	62.832	1.2368	31.416	0.7874	20.000
1.1545	22	2.7211	69.115	1.3605	34.558	0.8661	22.000
1.0583	24	2.9684	75.398	1.4842	37.699	0.9449	24.000
1.0160	**25**	3.0921	78.540	1.5461	39.270	0.9843	25.000
1	25.400	3.1416	79.796	1.5708	39.898	1.0000	25.400
0.9407	27	3.3395	84.823	1.6697	42.412	1.0630	27.000
0.9071	28	3.4632	87.965	1.7316	43.982	1.1024	28.000
0.8467	30	3.7105	94.248	1.8553	47.124	1.1811	30.000
0.7938	**32**	3.9579	100.531	1.9790	50.265	1.2598	32.000
0.7697	33	4.0816	103.673	2.0408	51.836	1.2992	33.000
0.7500	33.867	4.1888	106.395	2.0944	53.198	1.3333	33.867
0.7056	36	4.4527	113.097	2.2263	56.549	1.4173	36.000
0.6513	39	4.8237	122.522	2.4119	61.261	1.5354	39.000
0.6350	**40**	4.9474	125.664	2.4737	62.832	1.5748	40.000
0.6048	42	5.1948	131.947	2.5974	65.973	1.6535	42.000
0.5644	45	5.5658	141.372	2.7829	70.686	1.7717	45.000
0.5080	**50**	6.1842	157.080	3.0921	78.540	1.9685	50.000
0.5000	50.800	6.2832	159.593	3.1416	79.796	2.0000	50.800

NOTE: Bold face diametral pitches and modules designate preferred values.

2.7 Gear Types And Axial Arrangements

in accordance with the orientation of axes, there are three catagories of gears:

1. Parallel Axes Gears
2. Intersecting Axes Gears
3. Nonparallel and Nonintersecting Axes Gears

Spur and helical gears are the parallel axes gears. Bevel gears are the intersecting axes gears. Screw or crossed helical, worm and hypoid gears handle the third category. Table 2-3 lists the gear types per axes orientation.

Also, included in Table 2-3 is the theoretical efficiency range of the various gear types. These figures do not include bearing and lubricant losses. Also, they assume ideal mounting in regard to axis orientation and center distance. Inclusion of these realistic considerations will downgrade the efficiency numbers.

Table 2-3 Types of Gears and Their Categories

Categories of Gears	Types of Gears	Efficiency (%)
Parallel Axes Gears	Spur Gear Spur Rack Internal Gear Helical Gear Helical Rack Double Helical Gear	98 ... 99.5
Intersecting Axes Gears	Straight Bevel Gear Spiral Bevel Gear Zerol Gear	98 ... 99
Nonparallel and Nonintersecting Axes Gears Hypoid	Worm Gear Screw Gear Gear	30 ... 90 70 ... 95 96 ... 98

2.7.1 Parallel Axes Gears

1. Spur Gear

This is a cylindrical shaped gear in which the teeth are parallel to the axis. It has the largest applications and, also, it is the easiest to manufacture.

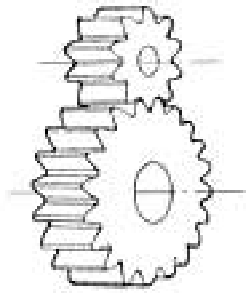

Fig. 2-8 Spur Gear

2. Spur Rack

This is a linear shaped gear which can mesh with a spur gear with any number of teeth. The spur rack is a portion of a spur gear with an infinite radius.

Fig. 2-9 Spur Rack

3. Internal Gear

This is a cylindrical shaped gear but with the teeth inside the circular ring. It can mesh with a spur gear. Internal gears are often used in planetary gear systems.

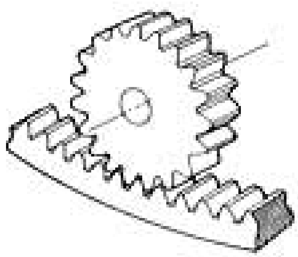

Fig. 2-10 Internal Gear and Spur Gear

4. Helical Gear

This is a cylindrical shaped gear with helicoid teeth. Helical gears can bear more load than spur gears, and work more quietly. They are widely used in industry. A negative is the axial thrust force the helix form causes.

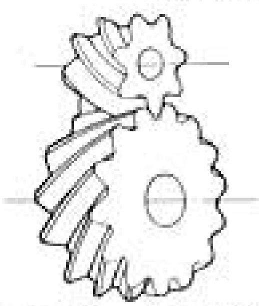

Fig. 2-11 Helical Gear

2.7.2 Intersecting Axes Gears

1. Straight Bevel Gear

This is a gear in which the teeth have tapered conical elements that have the same direction as the pitch cone base line (generatrix). The straight bevel gear is both the simplest to produce and the most widely applied in the bevel gear family.

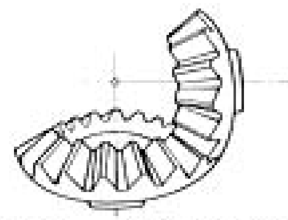

Fig. 2-14 Straight Bevel Gear

2. Spiral Bevel Gear

This is a bevel gear with a helical angle of spiral teeth. It is much more complex to manufacture, but offers a higher strength and lower noise.

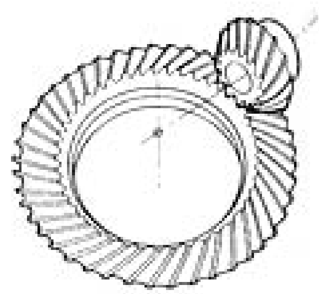

Fig. 2-15 Spiral Bevel Gear

3. Zerol Gear

Zerol gear is a special case of spiral bevel gear. It is a spiral bevel with zero degree of spiral angle tooth advance. It has the characteristics of both the straight and spiral bevel gears. The forces acting upon the tooth are the same as for a straight bevel gear.

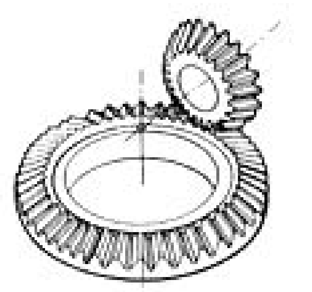

Fig. 2-16 Zerol Gear

5. Helical Rack

This is a linear shaped gear which meshes with a helical gear. Again, it can be regarded as a portion of a helical gear with infinite radius.

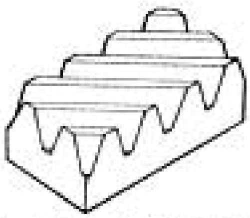

Fig. 2-12 Helical Rack

6. Double Helical Gear

This is a gear with both left-hand and right.hand helical teeth. The double helical form balances the inherent thrust forces.

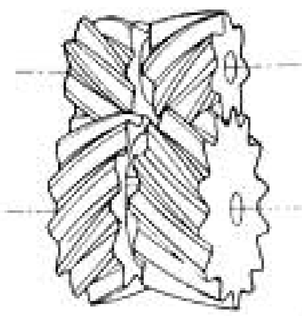

Fig. 2-13 Double Helical Gear

2.7.3 Nonparallel and Nonintersecting Axes Gears

1. Worm and Worm Gear

Worm set is the name for a meshed worm and worm gear. The worm resembles a screw thread; and the mating worm gear a helical gear, except that it is made to envelope the worm as seen along the worm's axis. The outstanding feature is that the worm offers a very large gear ratio in a single mesh. However, transmission efficiency is very poor due to a great amount of sliding as the worm tooth engages with its mating worm gear tooth and forces rotation by pushing and sliding. With proper choices of materials and lubrication, wear is contained and noise is low.

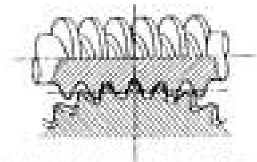

Fig. 2-17 Worm Gear

2. Screw Gear (Crossed Helical Gear)

Two helical gears of opposite helix angle will mesh if their axes are crossed. As separate gear components, they are merely conventional helical gears. Installation on crossed axes converts them - to screw gears. They offer a simple means of gearing skew axes at any angle. Because they have point contact, their load carrying capacity is very limited.

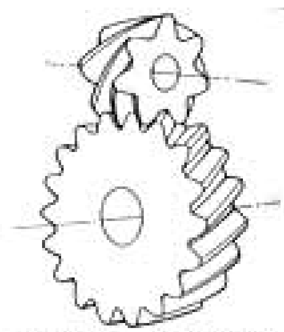

Fig. 2-18 Screw Gear

2.7.4 Other Special Gears

1. Face Gear

This is a pseudobevel gear that is limited to 900 intersecting axes. The face gear is a circular disc with a ring of teeth cut in its side face; hence the name face gear. Tooth elements are tapered towards its center. The mate is an ordinary spur gear. It offers no advantages over the standard bevel gear, except that it can be fabricated on an ordinary shaper gear generating machine.

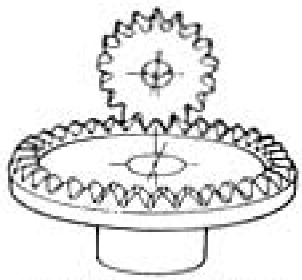

Fig. 2-19 Face Gear

2. Double Enveloping Worm Gear

This worm set uses a special worm shape in that it partially envelops the worm gear as viewed in the direction of the worm gear axis. Its big advantage over the standard worm is much higher load capacity. However, the worm gear is very complicated to design and produce, and sources for manufacture are few.

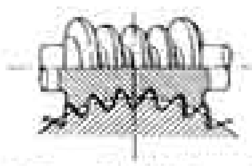

Fig. 2-20 Double Enveloping Worm Gear

3. Hypoid Gear

This is a deviation from a bevel gear that originated as a special development for the automobile industry. This permitted the drive to the rear axle to be nonintersecting, and thus allowed the auto body to be lowered, It looks very much like the spiral bevel gear. However, it is complicated to design and is the most difficult to produce on a bevel gear generator.

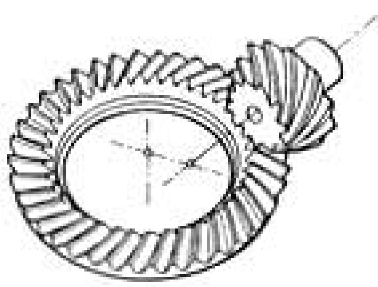

Fig. 2-21 Hypoid Gear

SECTION 3 DETAILS OF INVOLUTE GEARING

3.1 Pressure Angle

The pressure angle is defined as the angle between the line-of-action (common tangent to the base circles in Figures 2-3 and 2-4) and a perpendicular to the line-of-centers. See Figure 3-1. From the geometry of these figures, it is obvious that the pressure angle varies (slightly) as the center distance of a gear pair is altered. The base circle is related to the pressure angle and pitch diameter by the equation:

$$d_b = d \cos \alpha \qquad (3\text{-}1)$$

where d and α are the standard values, or alternately:

$$d_b = d' \cos \alpha' \qquad (3\text{-}2)$$

where d' and α' are the exact operating values.

The basic formula shows that the larger the pressure angle the smaller the base circle. Thus, for standard gears, 14.5° pressure angle gears have base circles much nearer to the roots of teeth than 20° gears. It is for this reason that 14.5° gears encounter greater undercutting problems than 20° gears. This is further elaborated on in SECTION 4.3.

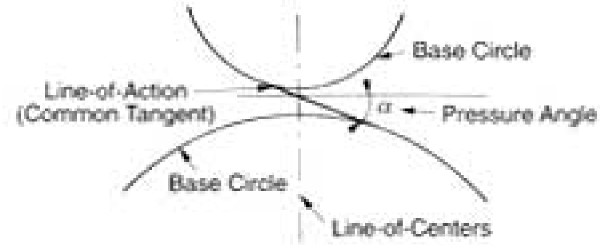

Fig. 3-1 Definition of Pressure Angle

3.2 Proper Meshing And Contact Ratio

Figure 3-2 shows a pair of standard gears meshing together. The contact point of the two involutes, as Figure 3-2 shows, slides along The common tangent of the two base circles as rotation occurs. The common tangent is called the line-of-contact, or line-of-action.

A pair of gears can only mesh correctly if the pitches and the pressut angles are the same. Pitch comparison can be module m, circular p, base P_b

That the pressure angles must be identical becomes obvious trot the following equation for base pitch:

$$P_b = \pi \text{ m COS } \alpha \qquad (3\text{-}3)$$

Thus, if the pressure angles are different, the base pitches cannot b identical.

The length of the line-of-action is shown as ab in Figure 3-2.

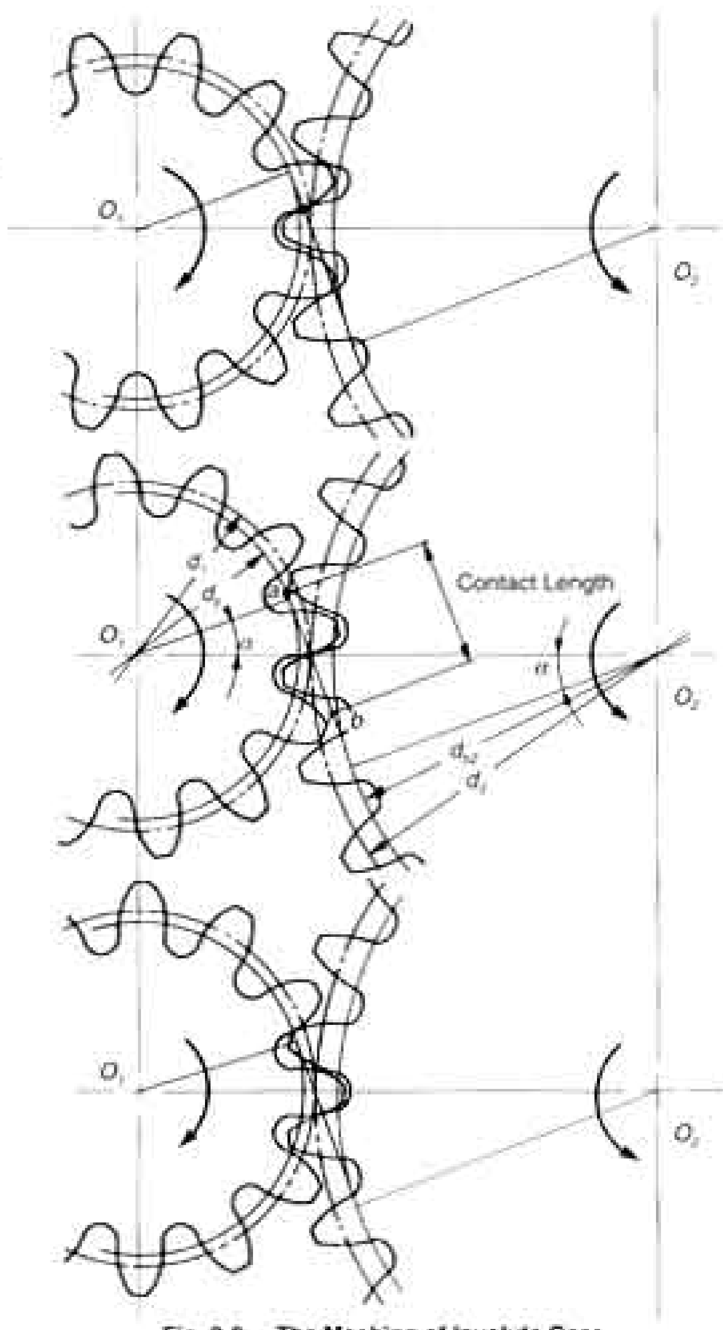

Fig. 3-2 The Meshing of Involute Gear