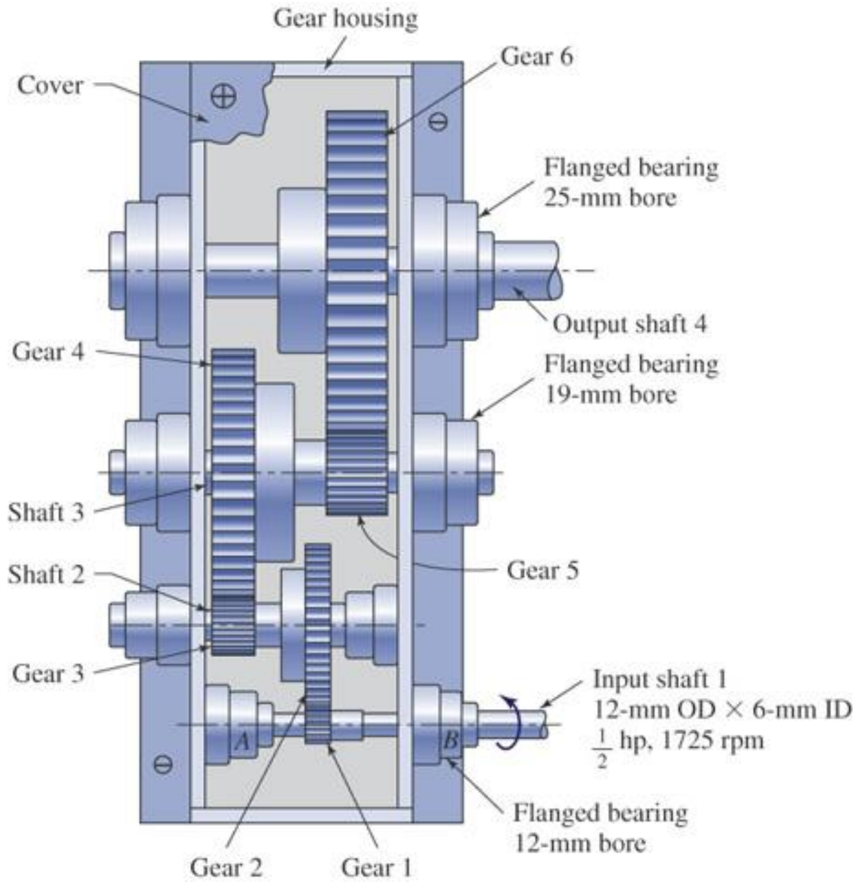
Gears

What we need to Know about them.

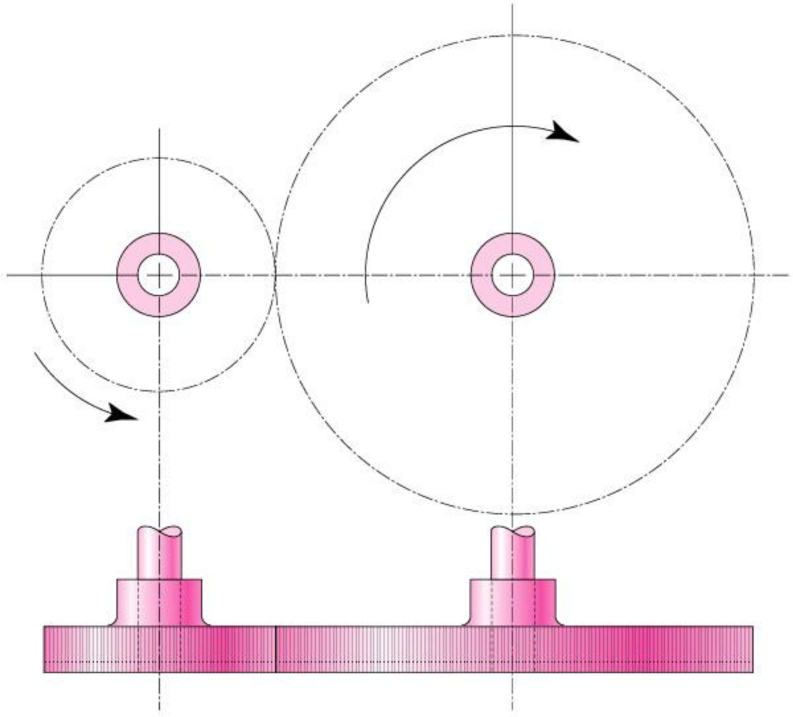
1. Type of gears
2. Terminologies or nomenclatures
3. Forces transmitted
4. Design of a gear box



Type of Gears

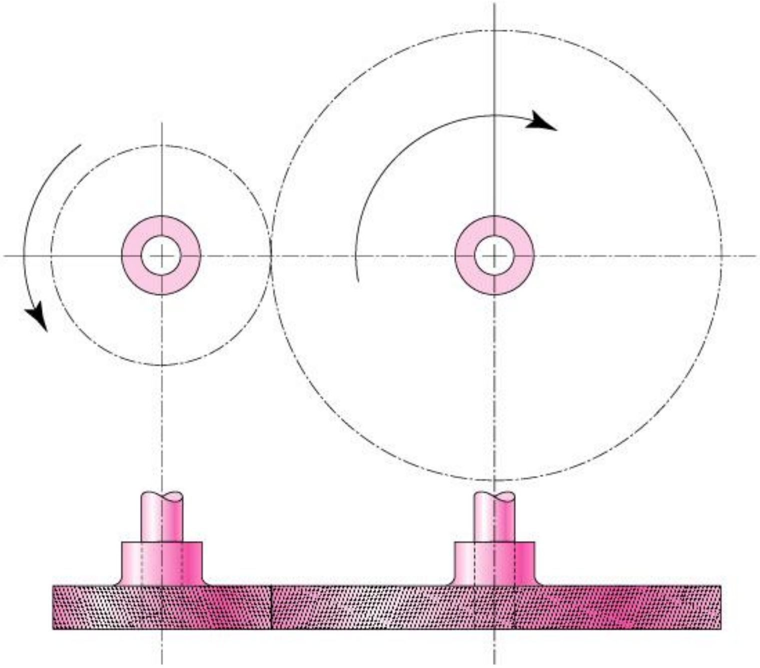
- Spurs
- Helical
- Bevel
- And Worm Gears

Spur Gears



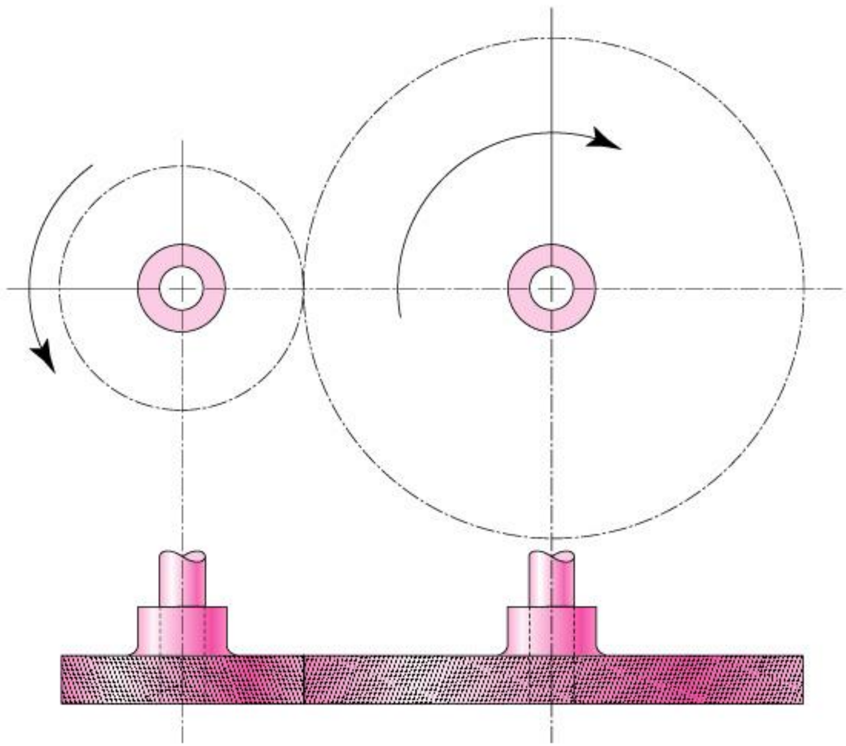
Are used in transmitting torque between parallel shafts

Helical Gears



Are used in transmitting torques between parallel or non parallel shafts, they are not as noisy as spur gears

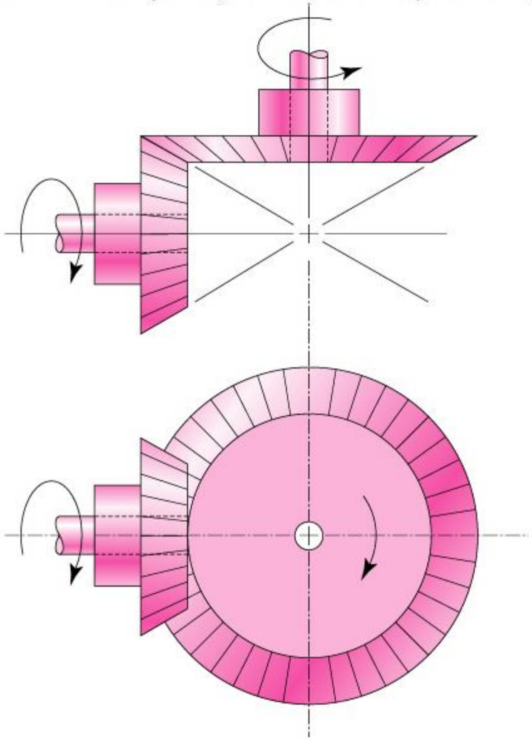
Fig. 13.2



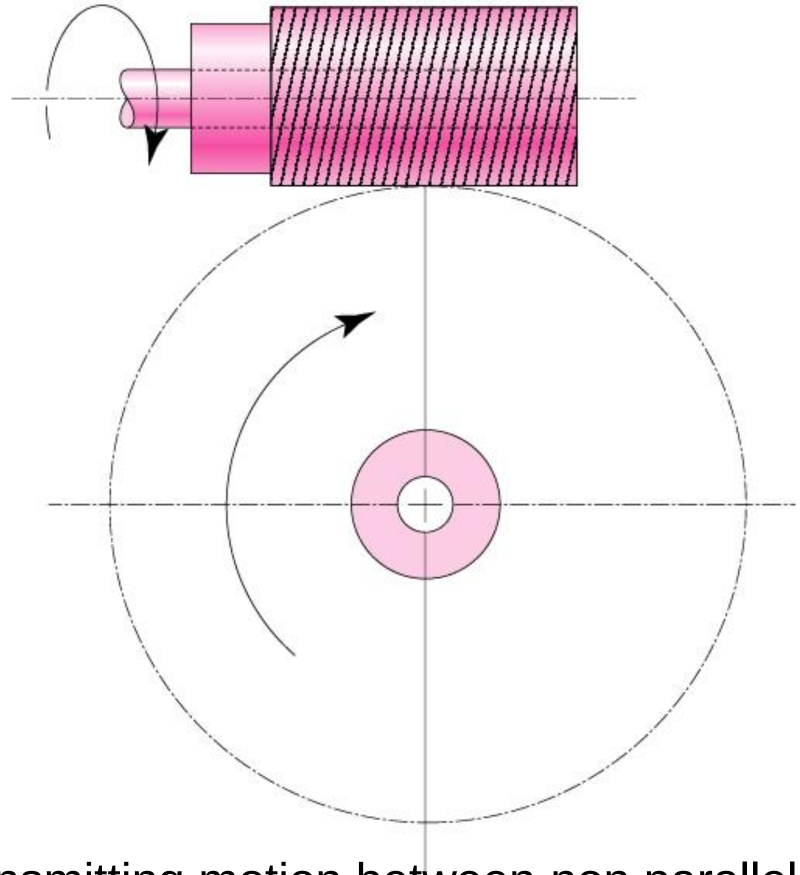
Bevel Gears

- Are used to transmit rotary motion between intersecting shafts

Teeth are formed on conical surfaces, the teeth could be straight or spiral.

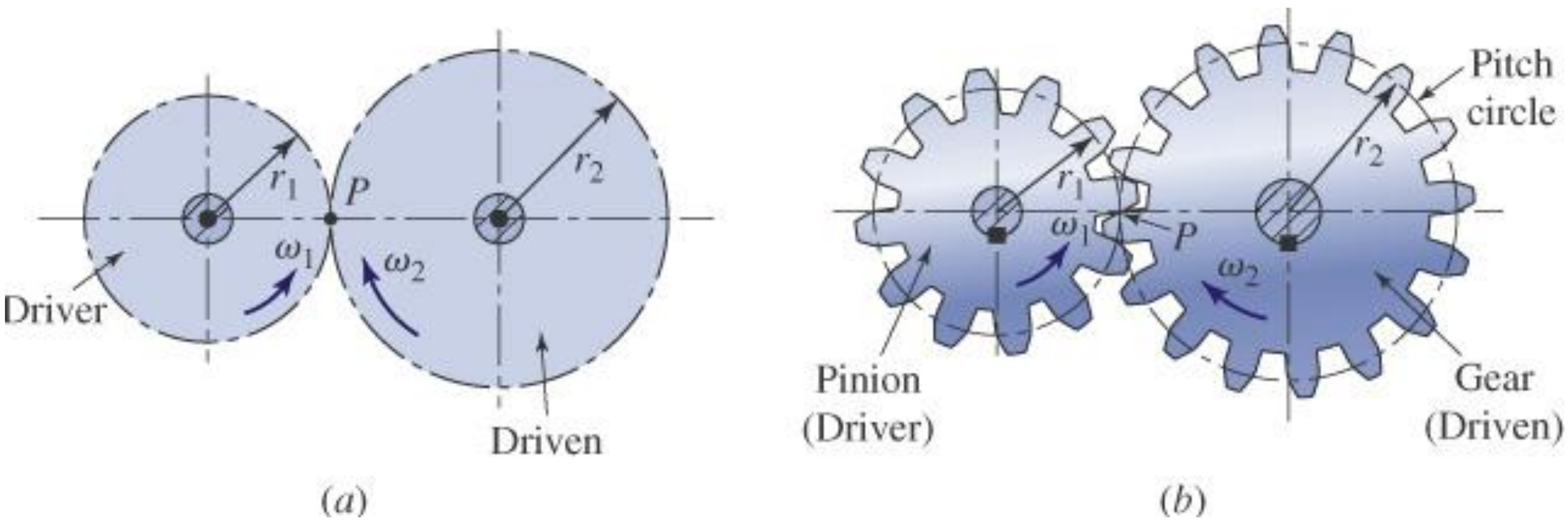


Worm Gears



Are used for transmitting motion between non parallel and non transmitting shafts, Depending on the number of teeth engaged called single or double. Worm gear mostly used when speed ratio is quiet high, 3 or more

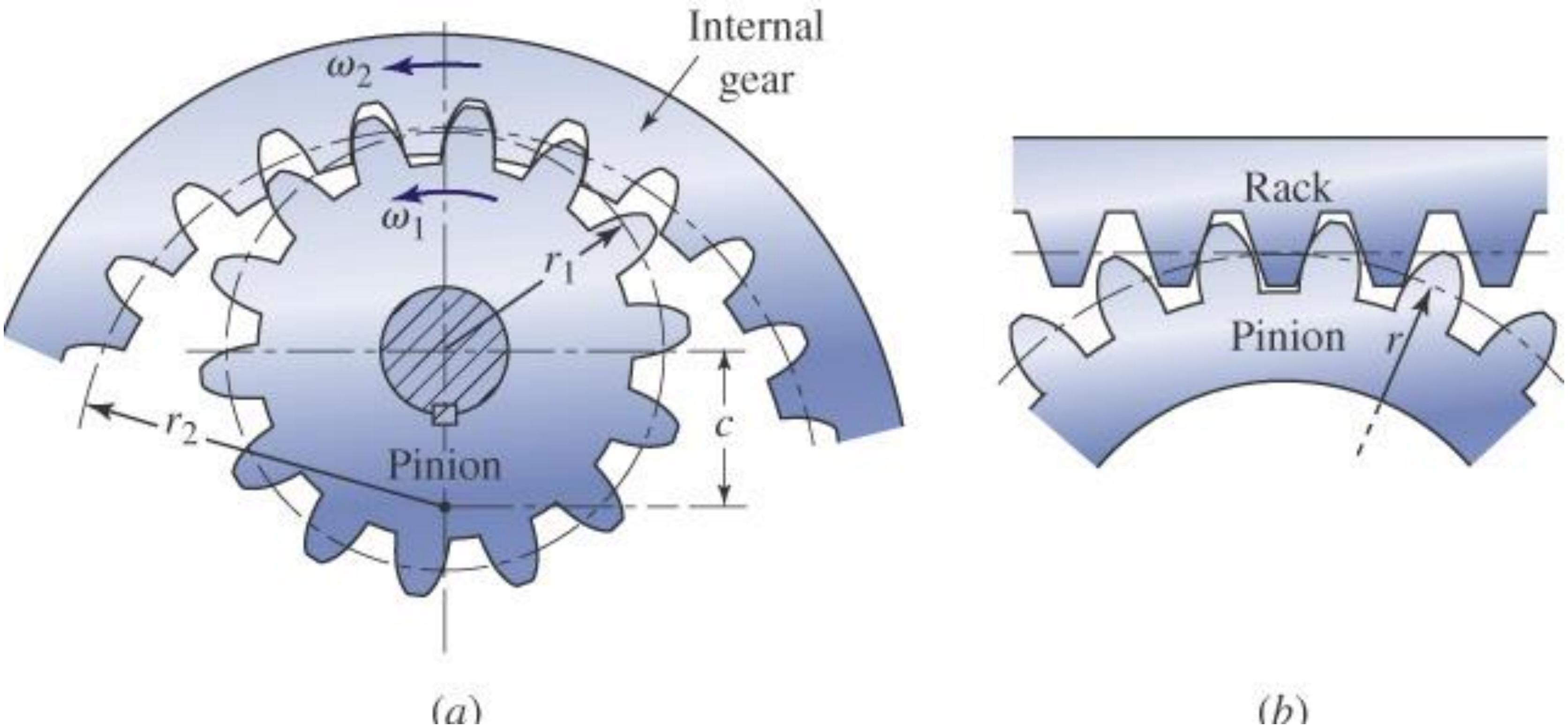
Nomenclature

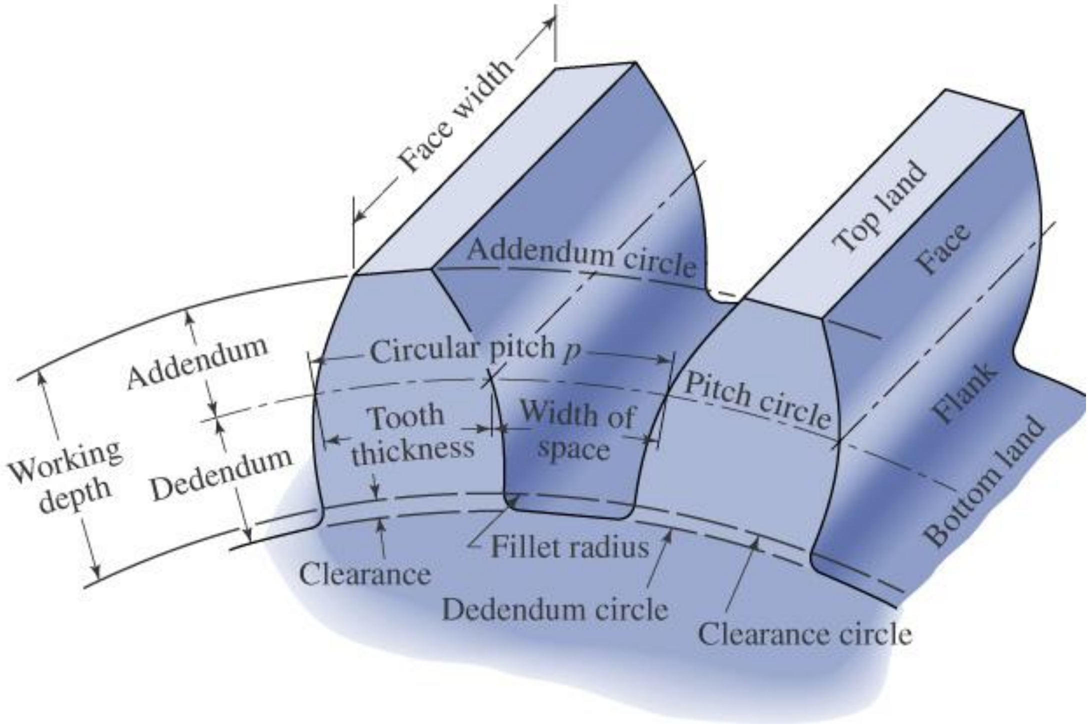


Smaller Gear is Pinion and Larger one is the gear

In most application the pinion is the driver, This reduces speed but it increases torque.

Internal Spur Gear System





pitch circle, theoretical circle upon which all calculation is based

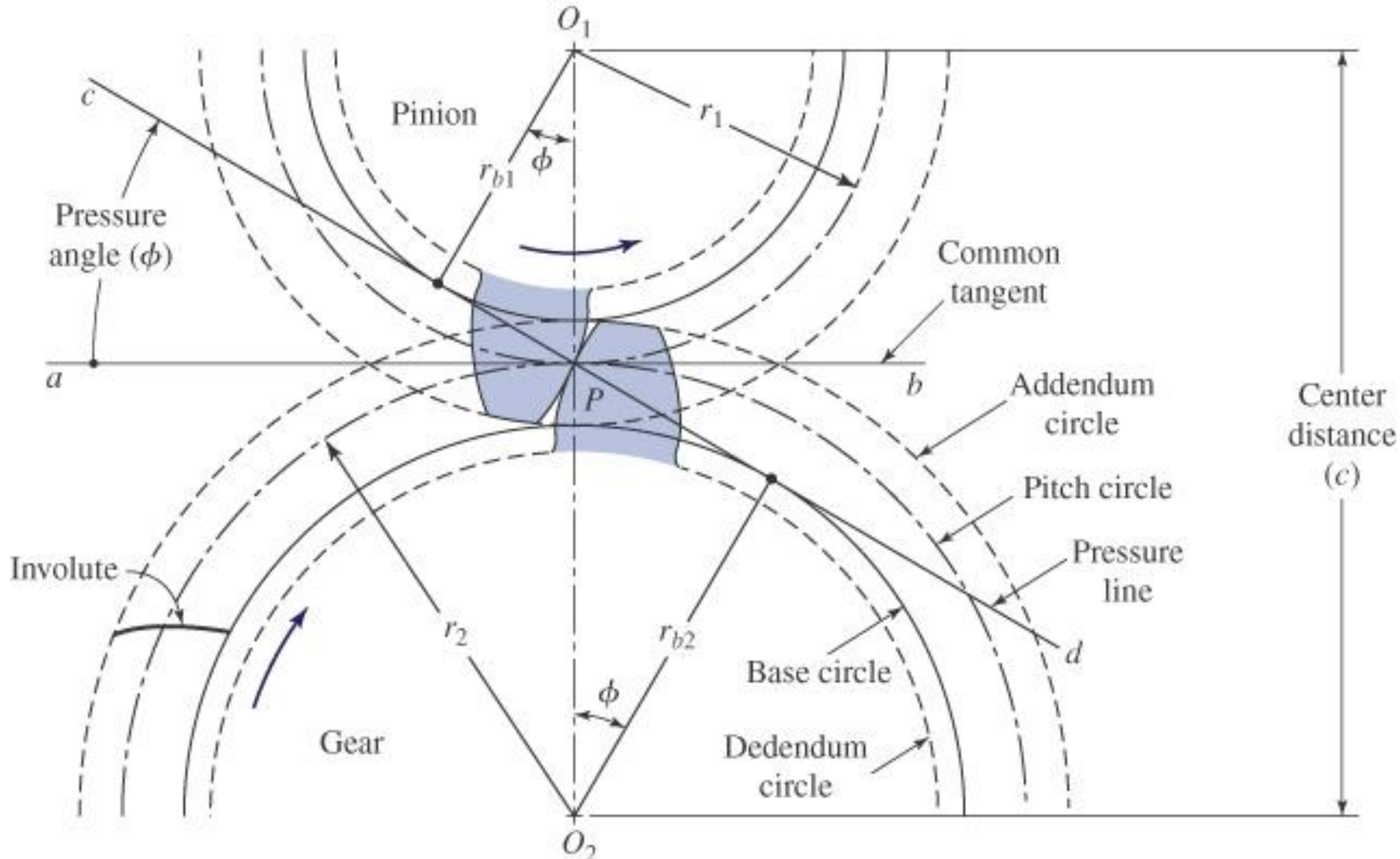
p, Circular pitch, p the distance from one teeth to the next, along the pitch circle. $p = \pi d/N$

m, module=d/N pitch circle/number of teeth

$p = \pi m$

P, Diametral Pitch $P = N/d$

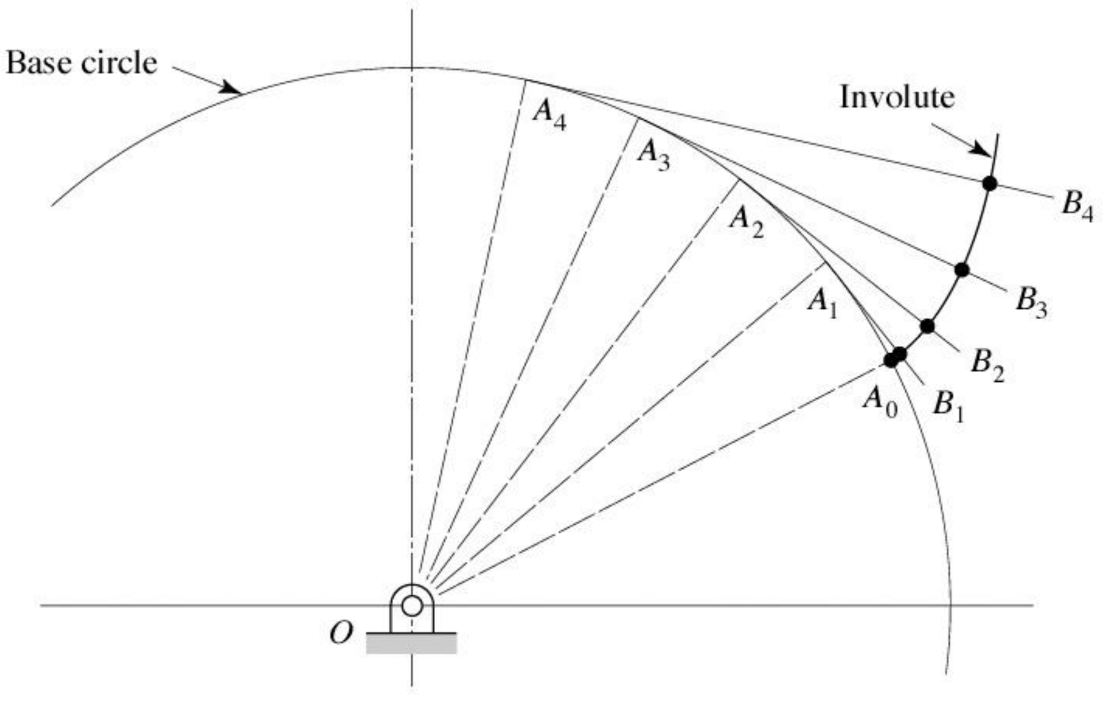
$pP = \pi$



Angle Φ has the values of 20 or 25 degrees. Angle 14.5 have been also used.

Gear profile is constructed from the base circle. Then additional clearance are given.

How Gear Profile is constructed

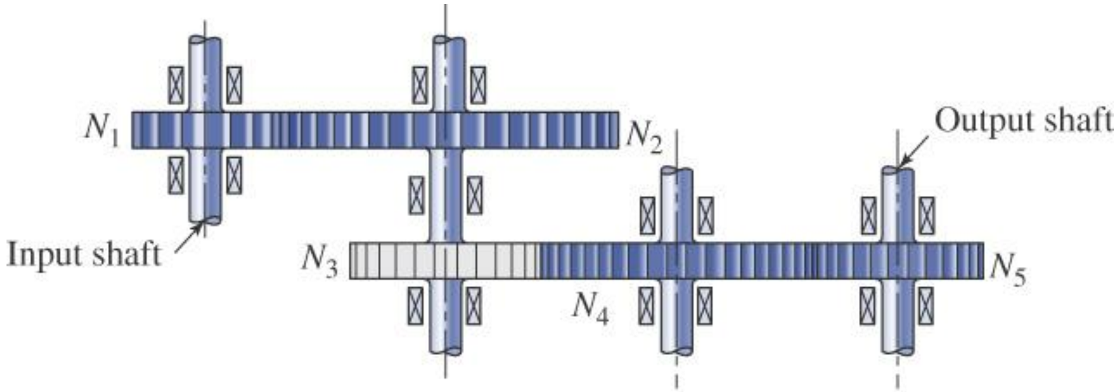


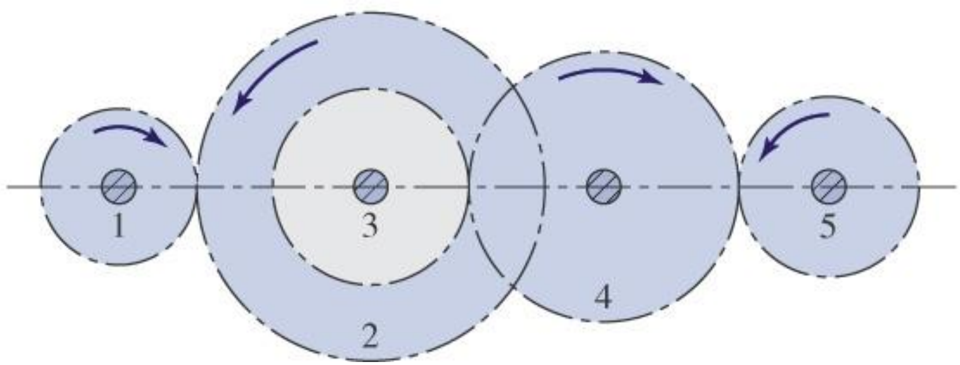
$A_1B_1=A_1A_0$, $A_2B_2=2\ A_1A_0$, etc

Standard Gear Teeth

Item	20^{o} full depth	20^{o} Stub	25^{o} full depth
Addendum a	1/P	0.8/P	1/P
Dedendum	1.25/P	1/P	1.25/P
Clearance f	0.25/P	0.2/P	0.25/P
Working depth	2/P	1.6/P	2/P
Whole depth	2.25/P	1.8/P	2.25/P
Tooth thickness	1.571/P	1.571/P	1.571/P
Face width	$9/P<b<13/P$	$9/P<b<13/P$	$9/P<b<13/P$

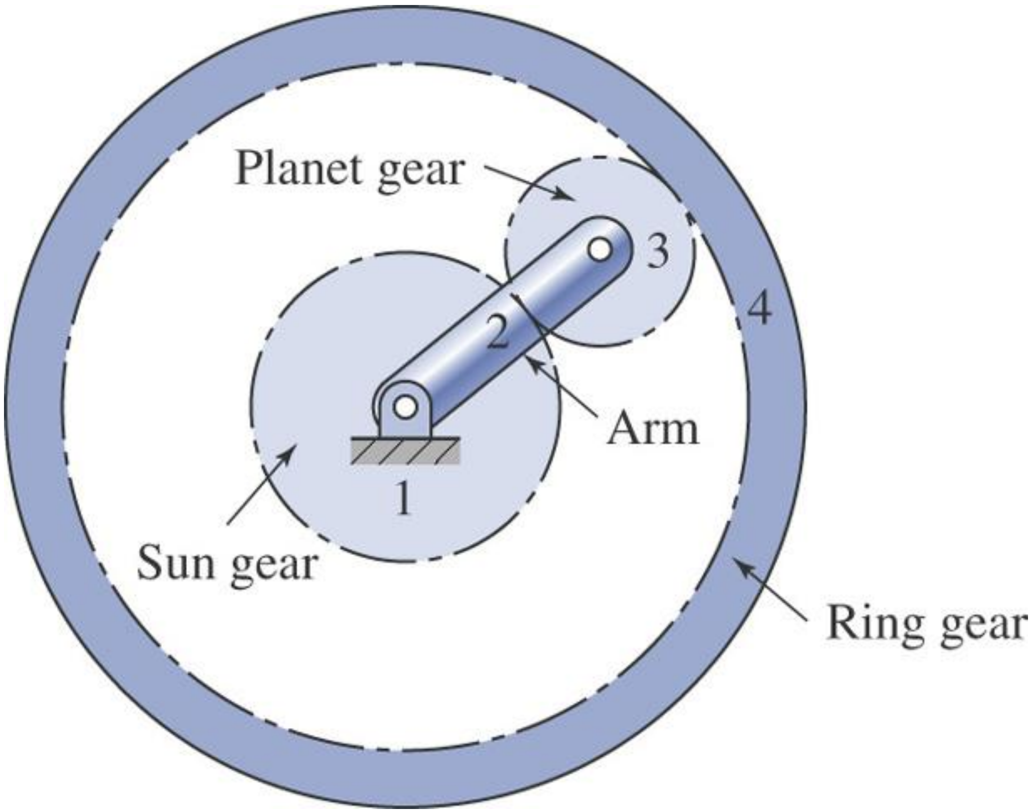
Gear Trains





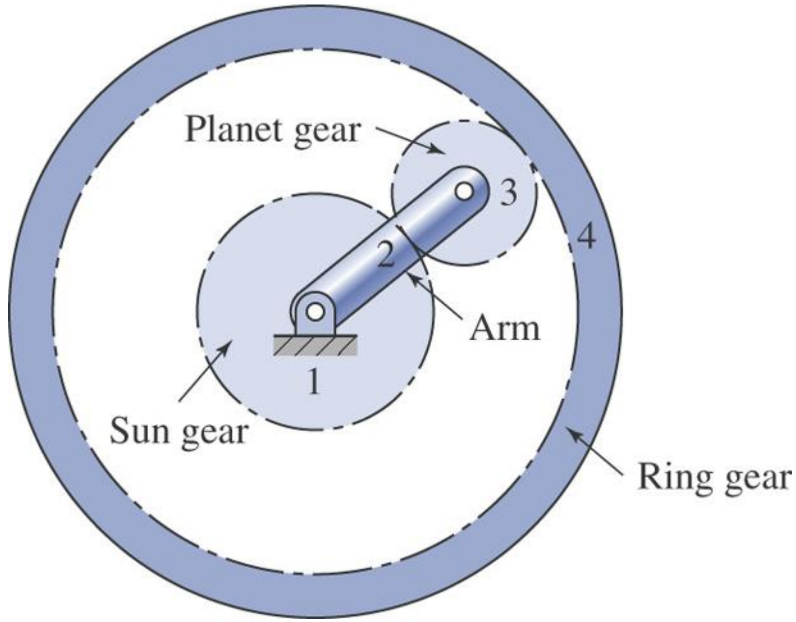
$$\frac{n_5}{n_1} = \left(-\frac{N_1}{N_2}\right)\left(-\frac{N_3}{N_4}\right)\left(-\frac{N_4}{N_5}\right)$$

Planetary Gear train
You can get high torque ratio in a smaller space



There are two inputs to the planetary gears, RPM of sun and Ring,
The out put is the speed of the arm.

Example of planetary Gear train



Gear 1, sun , RPM 1200, Number of teeth 20,

Planet Gear , Number of teeth 30

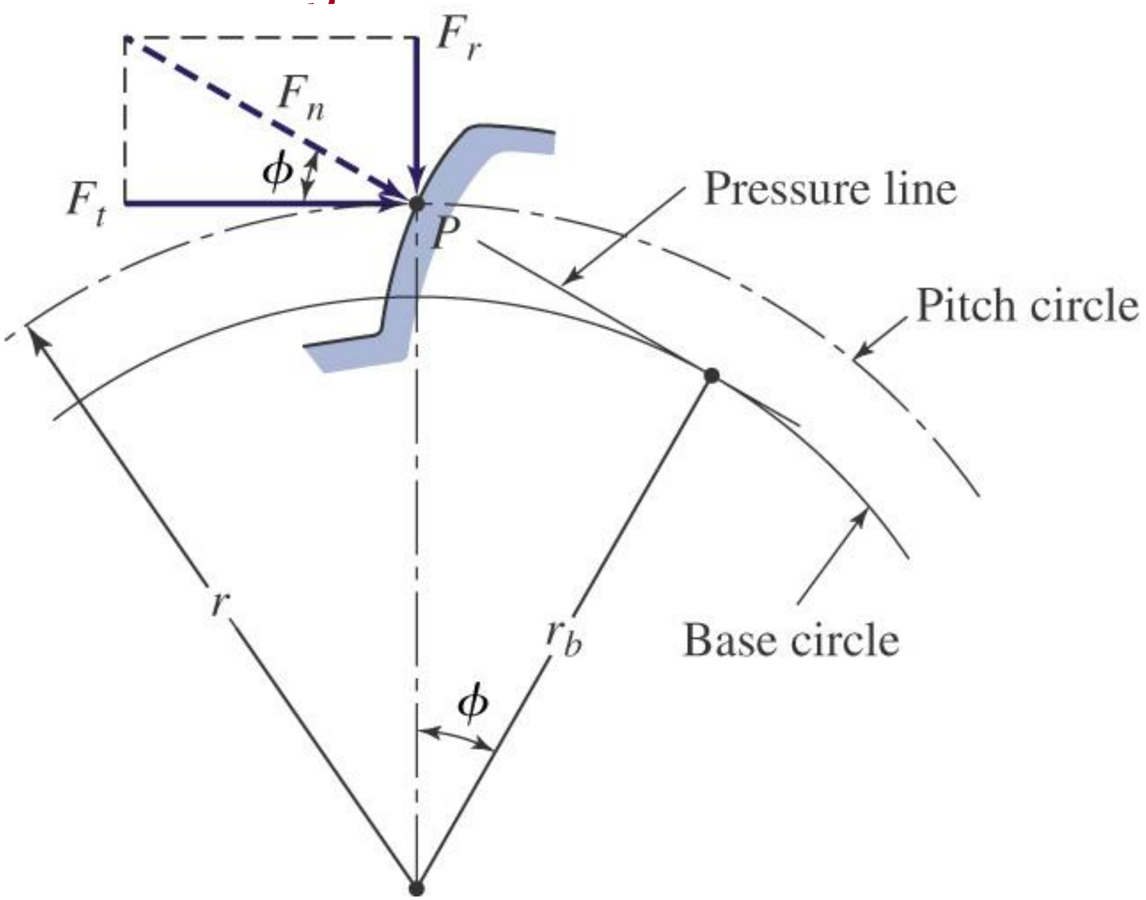
Ring Gear, Rotates RPM 120, and teeth of 80,

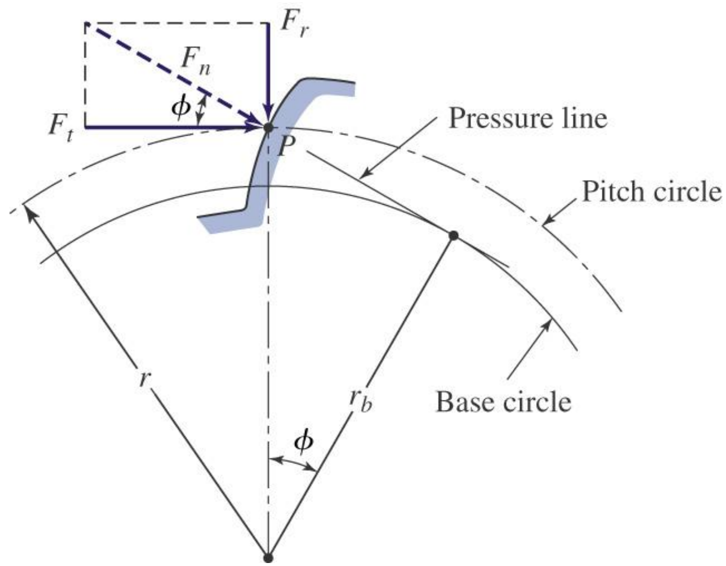
¼ horse power, find the speed of the arm and torque on the ring.

Alternatively you may have Certain Out put Torque requirements

Transmitted Load

- With a pair of gears or gear sets, Power is transmitted by the force developed between contacting Teeth





$$F_t = F_n \cos\phi$$

$$F_r = F_n \sin\phi$$

$$V = d/2\omega = d * \frac{2\pi RPM}{60}$$ d in, RPM rev./min, V in/sec

$$V = \frac{\pi dn}{12}$$ d in, n rpm, V fpm

$$hp = \frac{Tn}{63000}$$ Toque lb-in

$$F_t = \frac{33000hp}{V}$$ V fpm

$$KW = \frac{F_t V}{1000} = \frac{Tn}{9549}$$ T= N.m, V m/s, F Newton

These forces have to be corrected for dynamic effects , we discuss later, considering AGMA factors

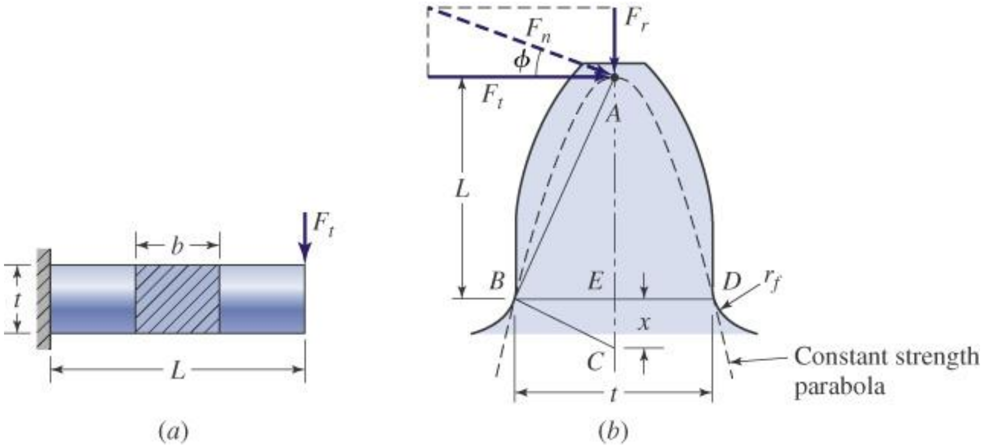
Some Useful Relations

- F=33000hp/V   V fpm   English system

- Metric System
- KW=(FV)/1000=Tn/9549
- F newton, V m/s, n rpm, T, N.m
- hp= FV/745.7=Tn/7121

Bending Strength of the a Gear Tooth

$$\sigma = \frac{Mc}{I} = \frac{(F_t L)t/2}{bt^3/12} = \frac{6F_t}{bt^2}$$



Earlier Stress Analysis of the Gear Tooth was based on

A full load is applied to the tip of a single tooth

The radial load is negligible

The load is uniform across the width

Neglect frictional forces

The stress concentration is negligible

This equation does not consider stress concentration, dynamic effects, etc.

Design for the Bending Strength of a Gear Tooth: The AGMA Method

$$\sigma = F_t K_0 K_v \frac{P}{b} \frac{K_s K_m}{J}$$ U.S. Customary

$$\sigma = F_t K_0 K_v \frac{1.0}{bm} \frac{K_s K_m}{J}$$ SI units

$\sigma =$ Bending stress at the root of the tooth

$F_t =$ Transmitted tangential load

$K_0 =$ Overload factor

$K_v =$ Velocity factor

$P =$ Diameteral pitch, P

$b =$ Face width

$m =$ Metric modue

$K_s =$ Size factor

$K_m =$ Mounting factor

$J =$ Geometry factor

Your stress should not exceed allowable stress

$$\sigma_{all} = \frac{S_t K_L}{K_T K_R}$$

$\sigma_{all} =$ Allowable bending stress

$S_t =$ Bending Strength

$K_L =$ Life factor

$K_T =$ Temperature factor

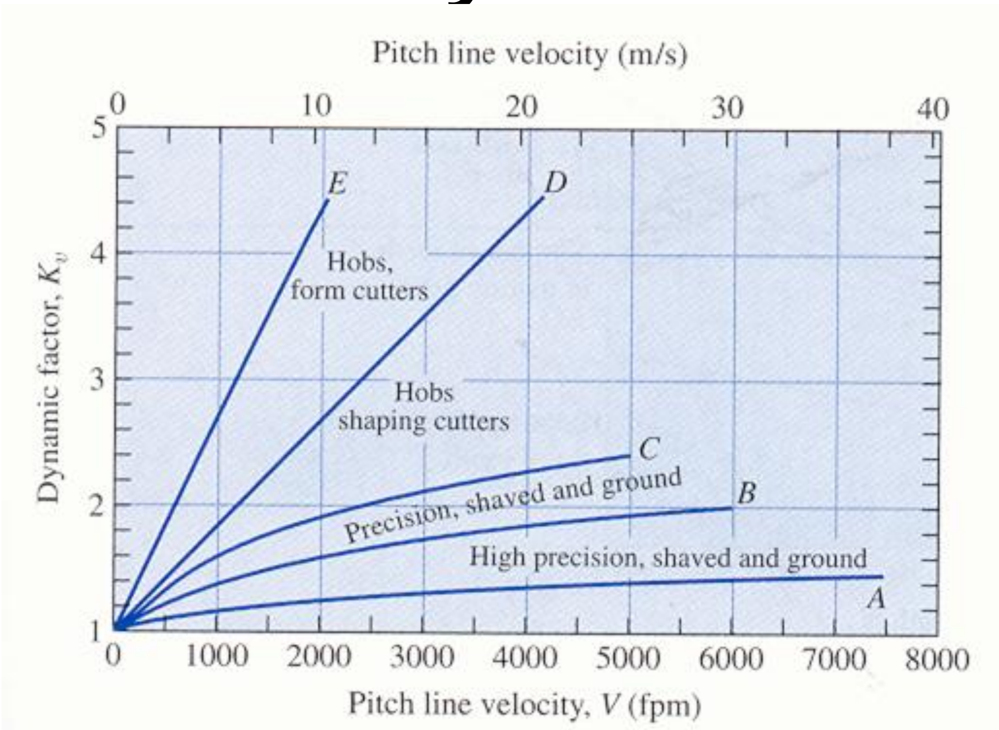
$K_R =$ Reliability factor

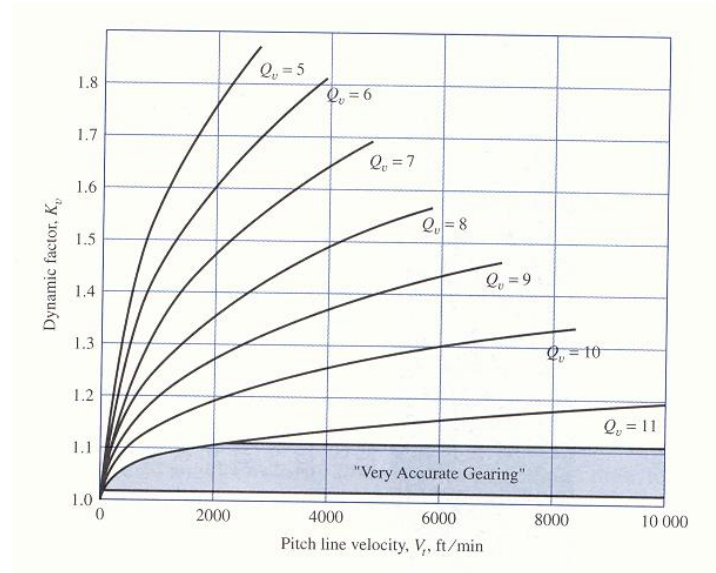
Overload Factor - K_o

Table 11.4 Overload correction factor K_o

Source of power	Load on driven machine		
	Uniform	Moderate shock	Heavy shock
Uniform	1.00	1.25	1.75
Light shock	1.25	1.50	2.00
Medium shock	1.50	1.75	2.25

Dynamic Factor - K_v





-Even with steady loads tooth impact can cause shock loading

-Impact strength depends on quality of the gear and the speed of gear teeth (pitch line velocity)

-Gears are classified with respect to manufacturing tolerances:

- -Q_v 3 – 7, commercial quality
- -Q_v 8 – 12, precision

-Graphs are available which chart K_v for different quality factors

Load Distribution Factor - K_m

Table 11.5 Mounting correction factor K_m

Condition of support	Face width (in.) 0 to 2	6	9	16 up
Accurate mounting, low bearing clearances, maximum deflection, precision gears	1.3	1.4	1.5	1.8
Less rigid mountings, less accurate gears, contact across the full face	1.6	1.7	1.8	2.2
Accuracy and mounting such that less than full-face contact exists		Over 2.2		

- -Failure greatly depends on how load is distributed across face
    - -Accurate mounting helps ensure even distribution
- -For larger face widths even distribution is difficult to attain
- -Note formula depends on face width which has to be estimated for initial iteration
    - -Form goal: $b < D_p$; $6 < b*P < 16$

Reliability Factor - K_R

Table 11.8 Reliability factor K_R

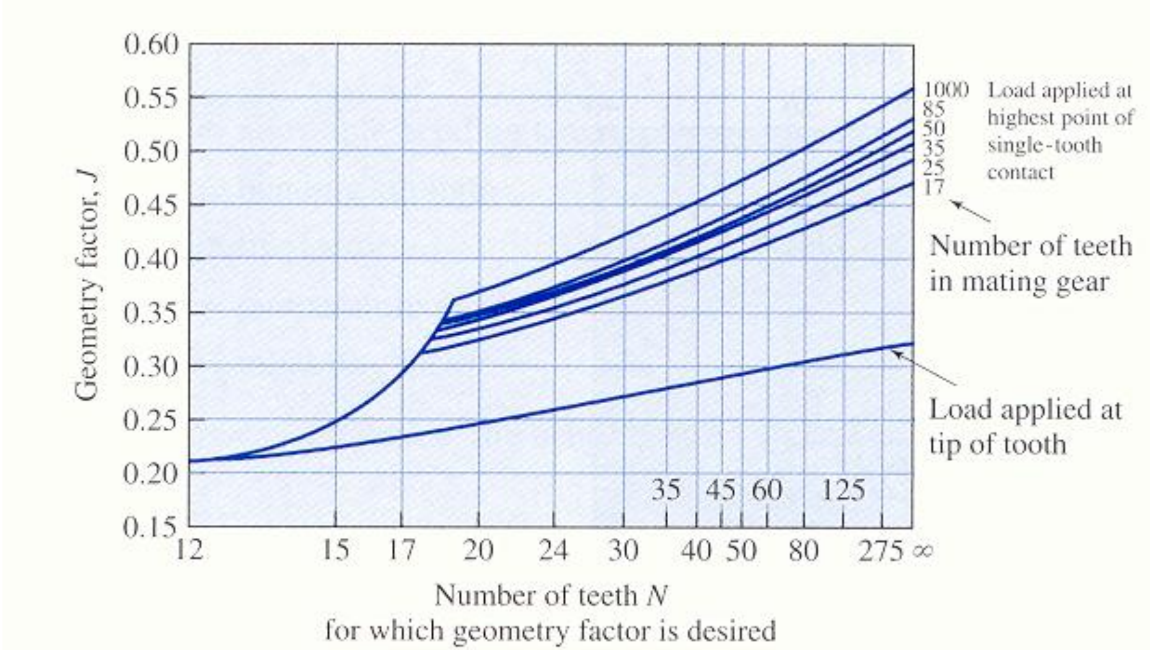
Reliability (%)	90	99	99.9	99.99
Factor K_R	0.85	1.00	1.25	1.50

SOURCE: The AGMA.

-Adjusts for reliability other than 99%

- $K_R = 0.658 – 0.0759 \ln (1-R)$ $0.5 < R < 0.99$
- $K_R = 0.50 – 0.109 \ln (1-R)$ $0.99 < R < 0.9999$

AGMA Geometry Factor - J



-Updated Lewis Form Factor includes effect of stress concentration at fillet

- -Different charts for different pressure angles

-Available for Precision Gears where we can assume load sharing (upper curves)

- -HPSTC – highest point of single tooth contact
- -Account for meshing gear and load sharing (contact ratio > 1)

-Single tooth contact conservative assumption (bottom curve)

- -J = 0.311 ln N + 0.15 (20 degree)
- -J = 0.367 ln N + 0.2016 (25 degree)

Bending Strength No. – $S_{t,}$ Fatigue bending strength

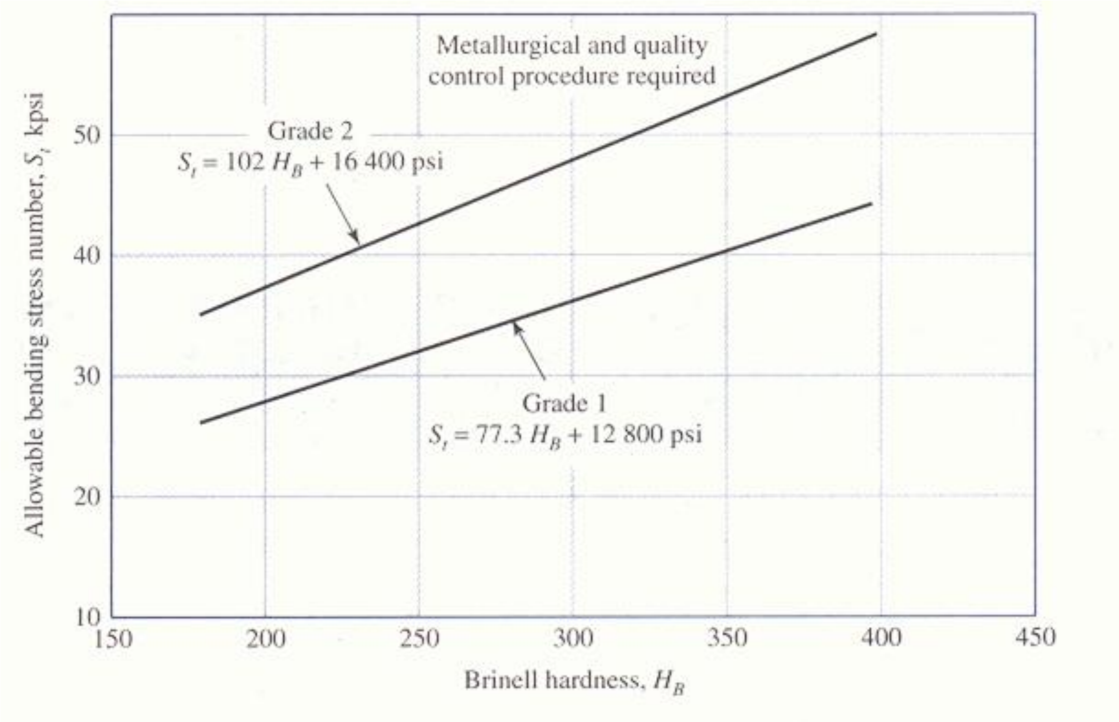
Table 11.6 Bending strength S_t of spur, helical, and bevel gear teeth

Material	Heat treatment	Minimum hardness or tensile strength	S_t ksi	S_t (MPa)
Steel	Normalized	140 Bhn	19–25	(131–172)
	Q&T	180 Bhn	25–33	(172–223)
	Q&T	300 Bhn	36–47	(248–324)
	Q&T	400 Bhn	42–56	(290–386)
	Case carburized	55 R_C	55–65	(380–448)
		60 R_C	55–70	(379–483)
	Nitrided AISI-4140	48 R_C case	34–45	(234–310)
		300 Bhn core		
Cast iron				
AGMA Grade 30		175 Bhn	8.5	(58.6)
AGMA Grade 40		200 Bhn	13	(89.6)
Nodular iron ASTM Grade:				
60-40-18			15	(103)
80-55-06	Annealed		20	(138)
100-70-18	Normalized		26	(179)
120-90-02	Q&T		30	(207)
Bronze, AGMA 2C	Sand cast	40 ksi (276 MPa)	5.7	(39.3)

SOURCE: AGMA 218.01.
Q&T = Quenched and tempered.

-Tabulated Data similar to fatigue strength
-Range given because value depends on Grade
-Based on life of 10^7 cycles and 99% reliability

S_t – Analytical Estimate



-Through hardened steel gears

  -Different charts for different manufacturing methods

-Grade 1 – good quality

  $St = 77.3\ H_B + 12{,}800$

-Grade 2 – premium quality

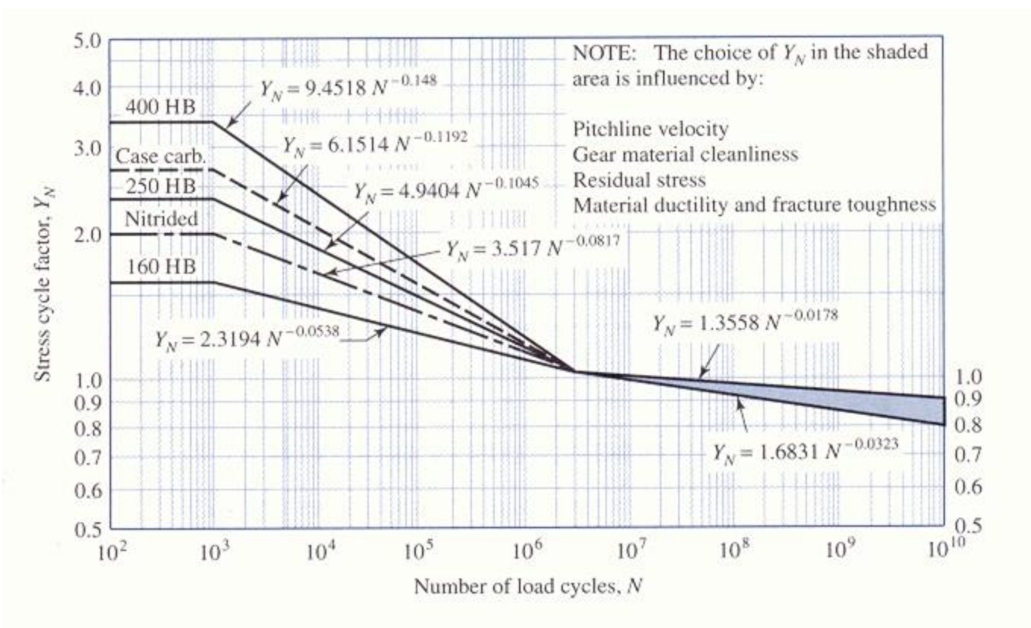
  $St = 102\ H_B + 16{,}400$

Bending Strength Life Factor- K_L

Table 11.7 Life factor K_L for spur and helical steel gears

Number of cycles	160 Bhn	250 Bhn	450 Bhn	Case carburized (55–63 R_C)
10^3	1.6	2.4	3.4	2.7–4.6
10^4	1.4	1.9	2.4	2.0–3.1
10^5	1.2	1.4	1.7	1.5–2.1
10^6	1.1	1.1	1.2	1.1–1.4
10^7	1.0	1.0	1.0	1.0

SOURCE: AGMA 218.01.

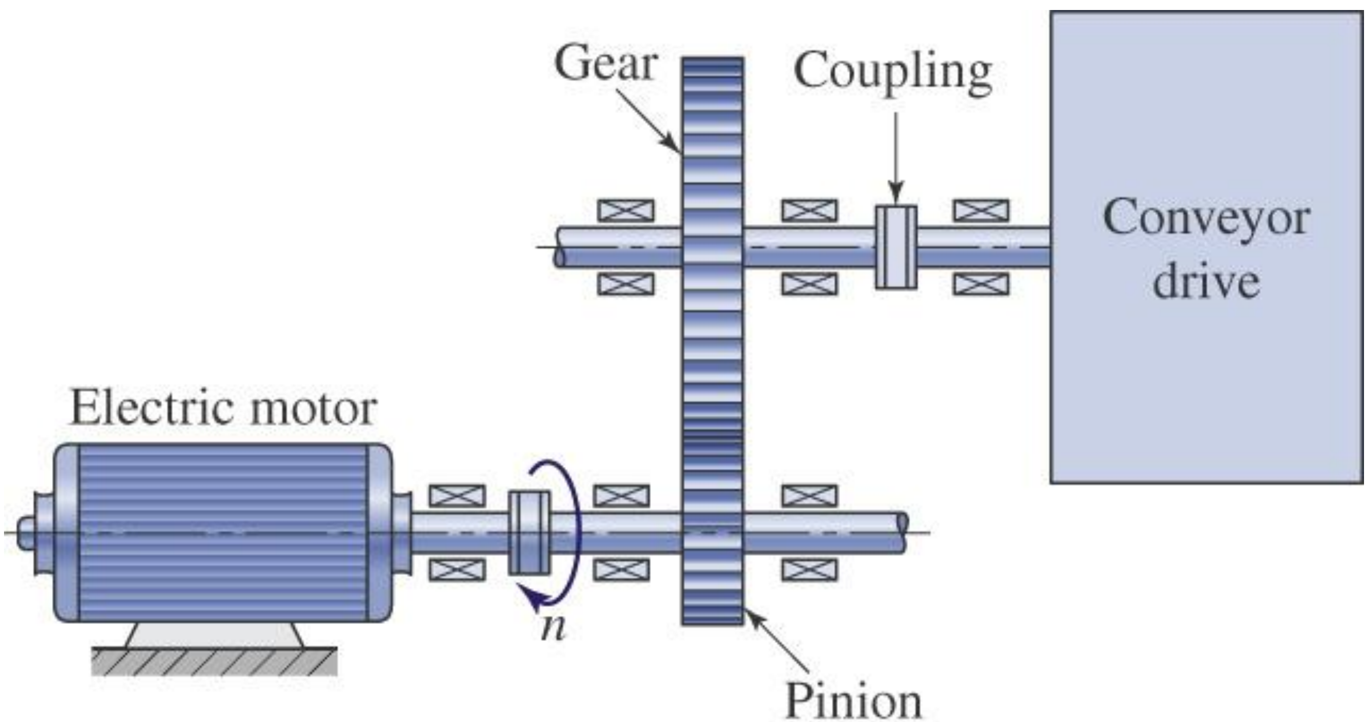


-Adjusts for life goals other than 10^7 cycles
-Fatigue effects vary with material properties and surface finishes
-$K_L = 1.6831\ N^{-0.0323}$ N>3E6

Note: @ 2000 rpm reach 3 million cycles in 1 day of service

Example:

A conveyor drive involving heavy-shock torsional loading is operated by an electric motor, the speed ratio is 1:2 and the pinion has Diameteral pitch P=10 in^{-1}, and number of teeth N=18 and face width of b=1.5 in. The gear has Brinnel hardness of 300 Bhn. Find the maximum horsepower that can be transmitted, using AGMA formula.



Gear Box Design