Kinematics And Dynamics Of Machines

(K & DM)

Semester : 4TH

Branch : Mechanical Engineering

Module - II

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CHAPTER : GEARS

Different Types Of Gears :

Spur gear

Basic Purpose of Use of Gears

Gears are widely used in various mechanisms and devices to transmit power
And motion positively (without slip) between parallel, intersecting (axis) or
Non-intersecting non parallel shafts,
• Without change in the direction of rotation
• With change in the direction of rotation
• Without change of speed (of rotation)
• With change in speed at any desired ratio

Often some gearing system (rack – and – pinion) is also used to transform Rotary motion into linear motion and vice-versa

- **A SPUR GEAR** is cylindrical in shape, with teeth on the outer circumference that are straight and parallel to the axis (hole) There are a number of variations of the basic spur gear, including pinion wire, stem pinions, rack and internal gears

Fig

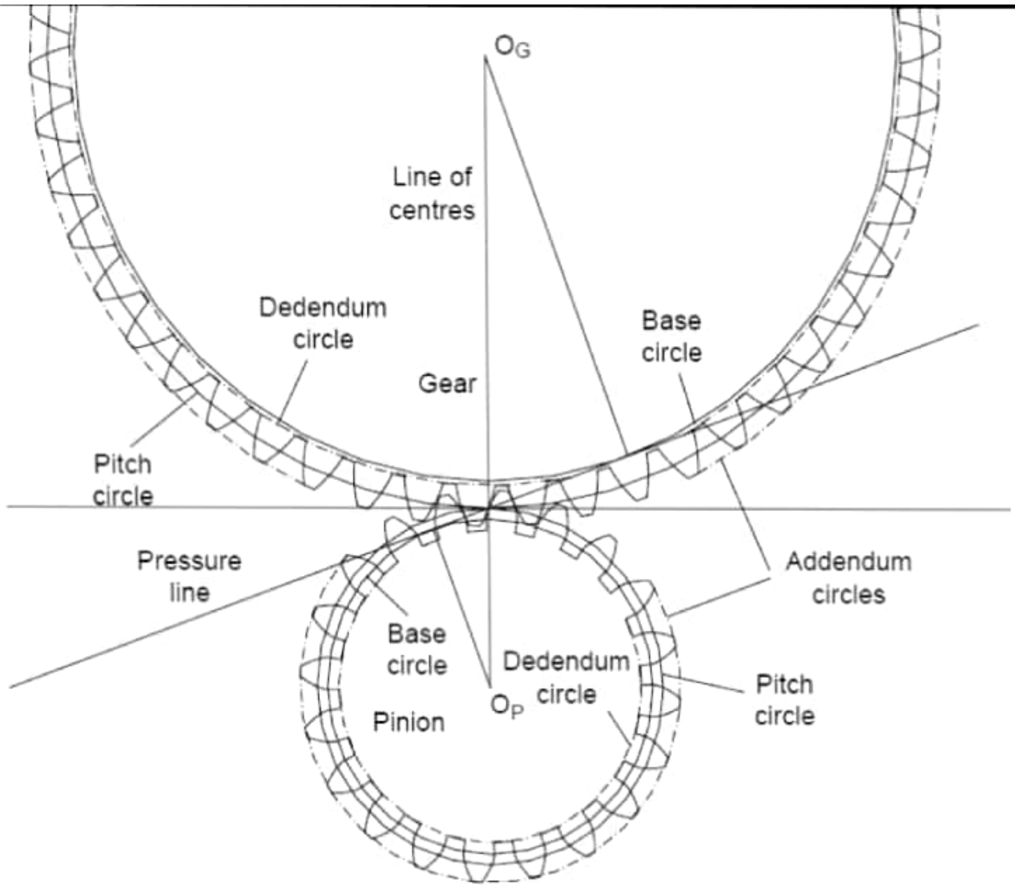


Figure- Layout of a pair of meshing spur gears

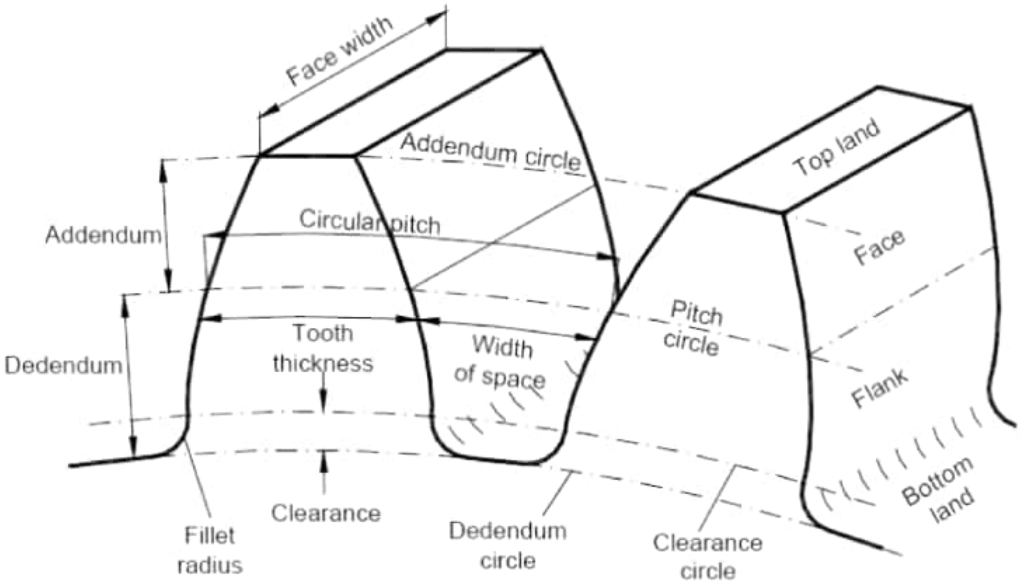


Figure- Spur gear schematic showing principle terminology

For a pair of meshing gears, the smaller gear is called the 'pinion', the larger is called the 'gear wheel' or simply the 'gear'.

Pitch circle

This is a theoretical circle on which calculations are based. Its diameter is called the pitch diameter.

$$d = mT$$

Where d is the pitch diameter (mm); m is the module (mm); and T is the number of teeth.
Care must be taken to distinguish the module from the unit symbol for a meter.

Circular pitch

This is the distance from a point on one tooth to the corresponding point on the adjacent tooth measured along the pitch circle.

$$p = \pi m = \frac{\pi d}{T}$$

Where p is the circular pitch (mm); m the module; d the pitch diameter (mm); and T the Number of teeth.

Module.

This is the ratio of the pitch diameter to the number of teeth. The unit of the module should be millimeters (mm). The module is defined by the ratio of pitch diameter and number of teeth. Typically the height of a tooth is about 2.25 times the module. Various modules are illustrated in figure.

$$m = \frac{d}{T}$$

- **Addendum**, (*a*). This is the radial distance from the pitch circle to the outside of the tooth.
- **Dedendum**, (*b*). This is the radial distance from the pitch circle to the bottom land.

Clearance (C) is the amount by which the dedendum in a given gear exceeds the addendum of its mating gear.

Backlash

BACKLASH is the distance (spacing) between two "mating" gears measured at the back of the driver on the pitch circle. Backlash, which is purposely built in, is very important because it helps prevent noise, abnormal wear and excessive heat while providing space for lubrication of the gears.

- The backlash for spur gears depends upon (i) module and (ii) pitch line velocity.
- Factor affected by changing center distance is backlash.

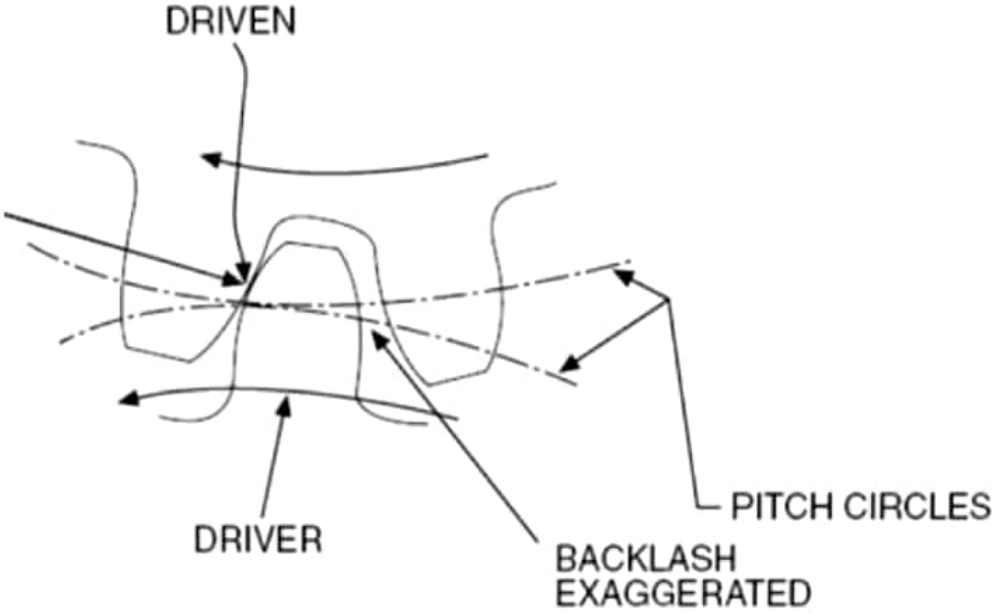


Fig.

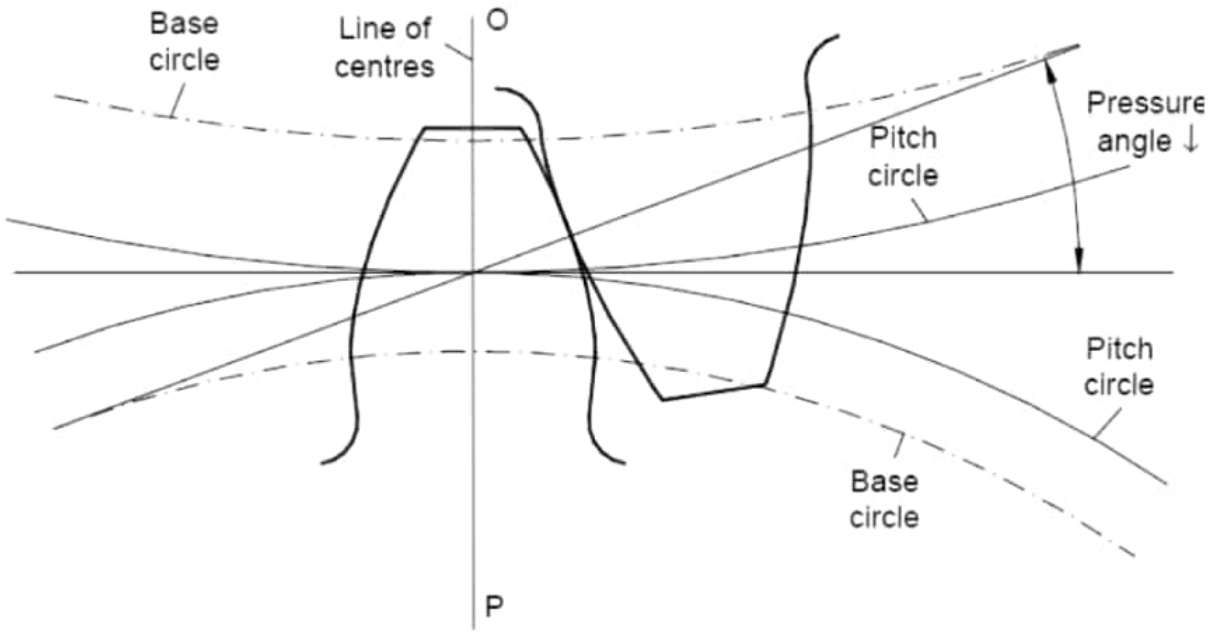


Figure- Schematic showing the pressure line and pressure angle

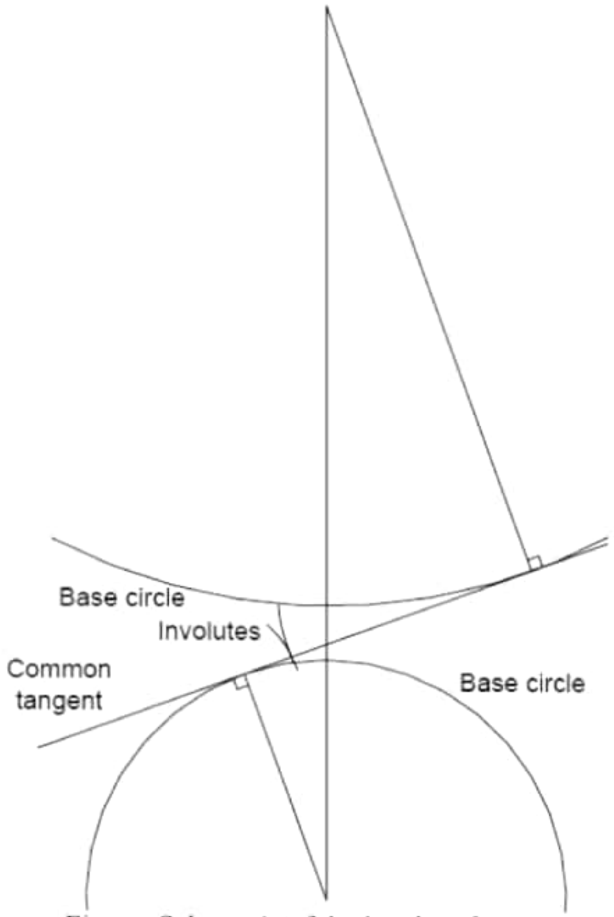


Figure- Schematic of the involute form

Pitch Circle and pitch point

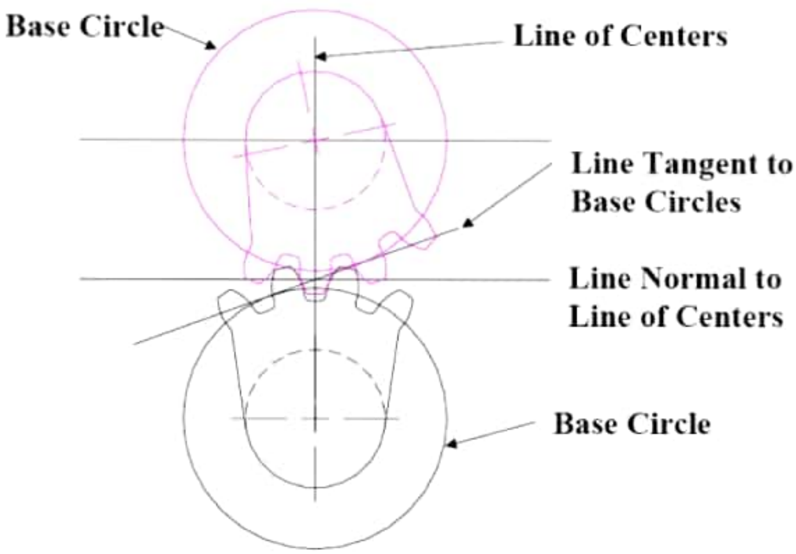


Fig.

Line of Action – Line tangent to both base circles

Pitch Point – Intersection of the line of centers with the line of action

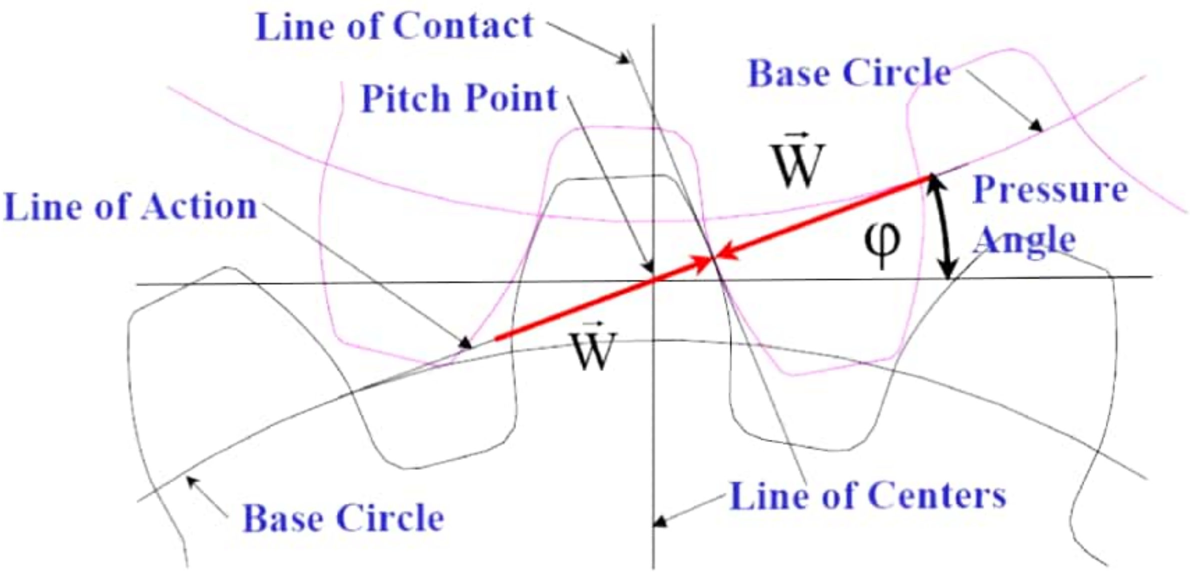


Fig.

Pitch Circle – Circle with origin at the gear center and passing through the pitch point.

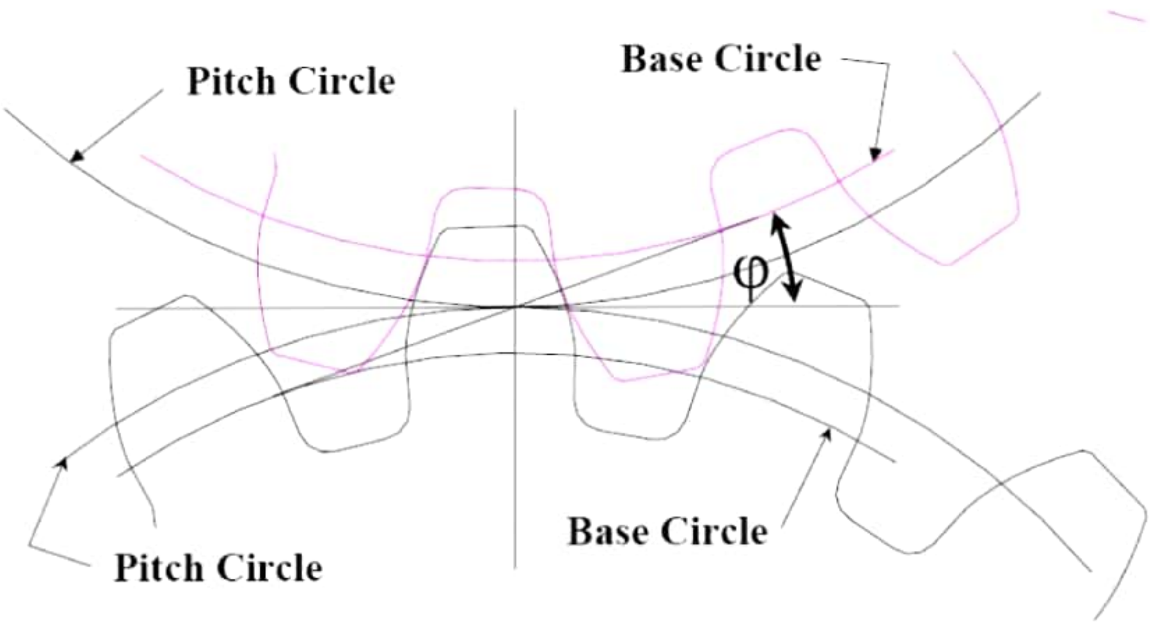


Fig.

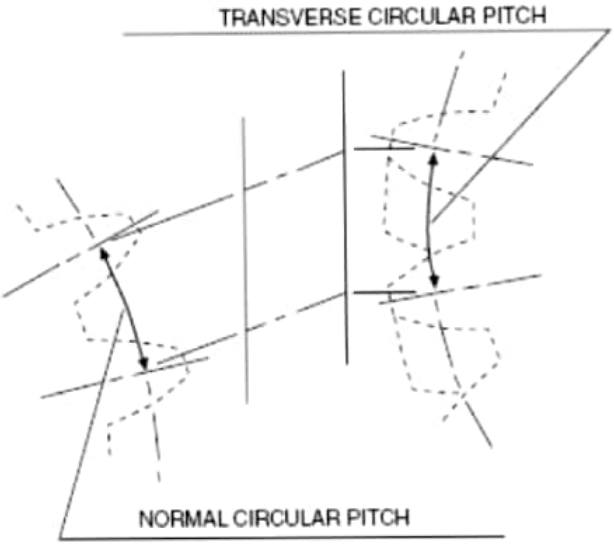


Fig.

Pressure angle – Angle between the line normal to the line of centers and the line of action.

- The pressure angle of a spur gear normally varies **from 14° to 20°**
- The value of pressure angle generally used for involute gears are **20°**
- Relationship Between Pitch and Base Circles

$$r_b = r\cos\phi$$

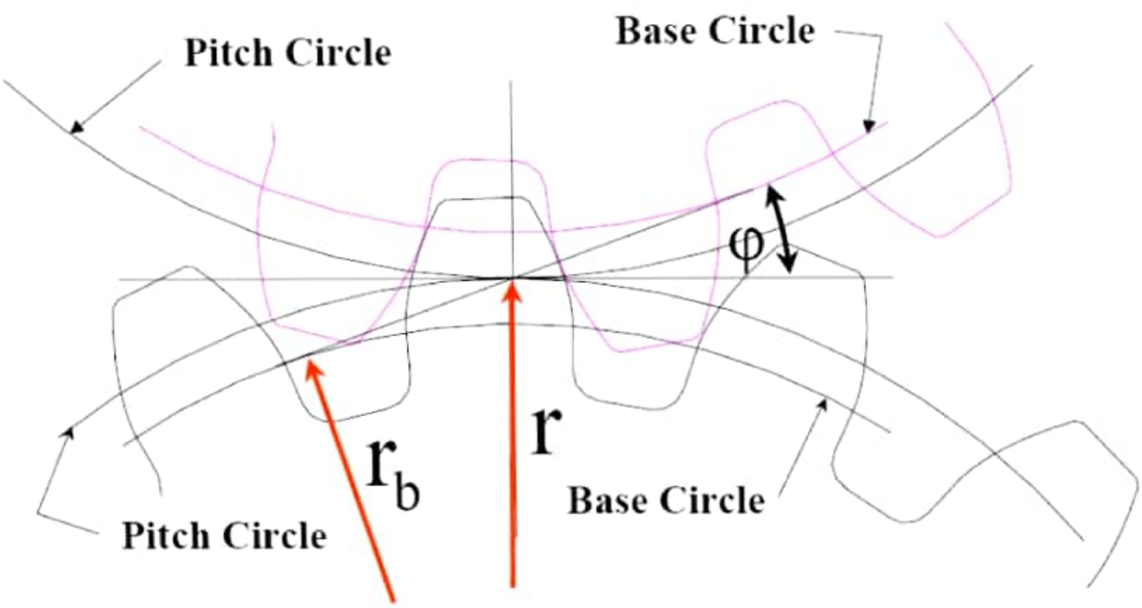


Fig.

The following four systems of gear teeth are commonly used in practice.

1.	$14\frac{1}{2}^{\circ}$	**Composite system.**

2.	$14\frac{1}{2}^0$	**Full depth involute system.**
3.	20^0	**Full depth involute system**
4	20^0	**Stub involutes system.**

The $14\frac{1}{2}^0$ ***composite system*** is used for general purpose gears. It is stronger but has no interchangeability. The tooth profile of this system has cycloidal curves at the top and bottom and involute curve at the middle portion. The teeth are produced by formed milling cutters or hobs. The tooth profile of the $14\frac{1}{2}^0$ ***full depth involute system*** was developed for use with gear hobs for spur and helical gears.

The tooth profile of the **20°** ***full depth involute system*** may be cut by hobs. The increase of the pressure angle from $14\frac{1}{2}^0$ to 20° results in a stronger tooth, because the tooth acting as a beam is wider at the base. The **20°** ***stub involute system*** has a **strong tooth** to take heavy loads.

Classification of Gears

Gears can be divided into several broad classifications.

1. Parallel axis gears:
(a) Spur gears
(b) Helical gears
(c) Internal gears.

2. Non-parallel, coplanar gears (intersecting axes):
(a) Bevel gears
(b) Face gears,
(c) Conical involute gearing.

3. Non-parallel, non- coplanar gears (nonintersecting axes):
(a) Crossed axis helical
(b) Cylindrical worm gearing
(c) Single enveloping worm gearing,
(d) Double enveloping worm gearing,
(e) Hypoid gears,
(f) Spiroid and helicon gearing,
(g) Face gears (off centre).

4. Special gear types:
(a) Square and rectangular gears,
(b) Elliptical gears.

RACK

RACKS are yet another type of spur gear. Unlike the basic spur gear, racks have their teeth cut into the surface of a straight bar instead of on the surface of a cylindrical blank.

Fig. Rack

Helical gear

The helical gears may be of ***single helical type*** or ***double helical type.*** In case of single helical gears there is some axial thrust between the teeth, which is a disadvantage. In order to eliminate this axial thrust, double helical gears (*i.e.* **herringbone gears**) are used. It is equivalent to two single helical gears, in which equal and opposite thrusts are provided on each gear and the resulting axial **thrust is zero.**

Herringbone gears

Figure-Herringbone gear

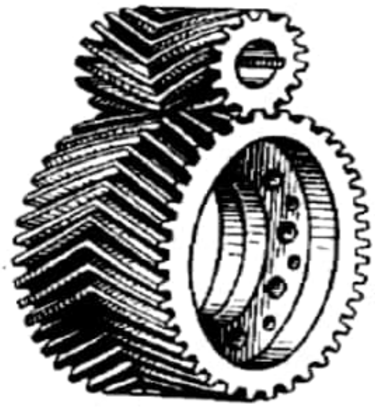


Figure- Herringbone gear

Figure- Crossed axis helical gears

- In spur gears, the contact between meshing teeth occurs along the entire face width of the tooth, resulting in a sudden application of the load which, in turn, results in impact conditions and generates noise.

- In helical gears, the contact between meshing teeth begins with a point on the leading edge of the tooth and gradually extends along the diagonal line across the tooth. There is a gradual pick-up of load by the tooth, resulting in smooth engagement and silence operation.

Bevel Gears

Fig.

Fig.

Worm Gear

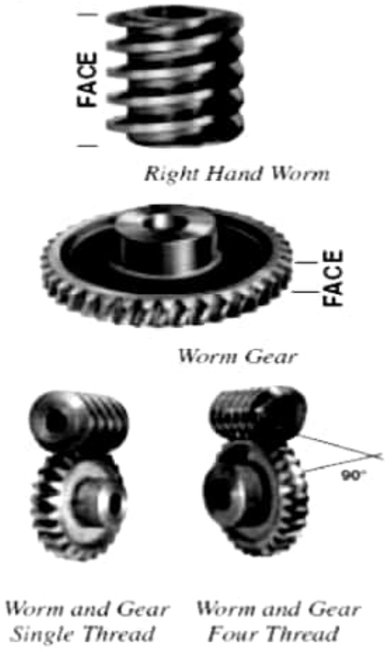


Fig.

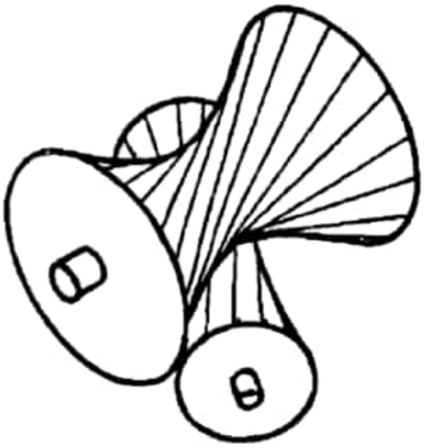
Fig.

Hypoid Gears

Hypoid gears resemble bevel gears and spiral bevel gears and are used on crossed-axis shafts. The distance between a hypoid pinion axis and the axis of a hypoid gear is called the *offset*. Hypoid pinions may have as few as five teeth in a high ratio set. Ratios can be obtained with hypoid gears that are not available with bevel gears. High ratios are easy to obtain with the hypoid gear system.

Hypoid gears are matched to run together, just as zero or spiral bevel gear sets are matched. The geometry of hypoid teeth is defined by the various dimensions used to set up the machines to cut the teeth.

- Hypoid gears are similar in appearance to spiral-bevel gears. They differ from spiral-bevel gears in that the axis of the pinion is offset from the axis of the gear.



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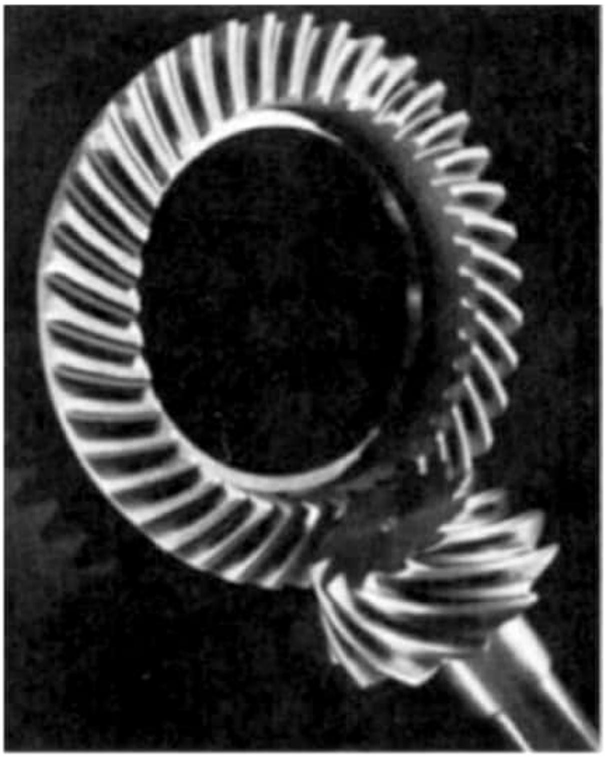


Figure- Hypoid gear

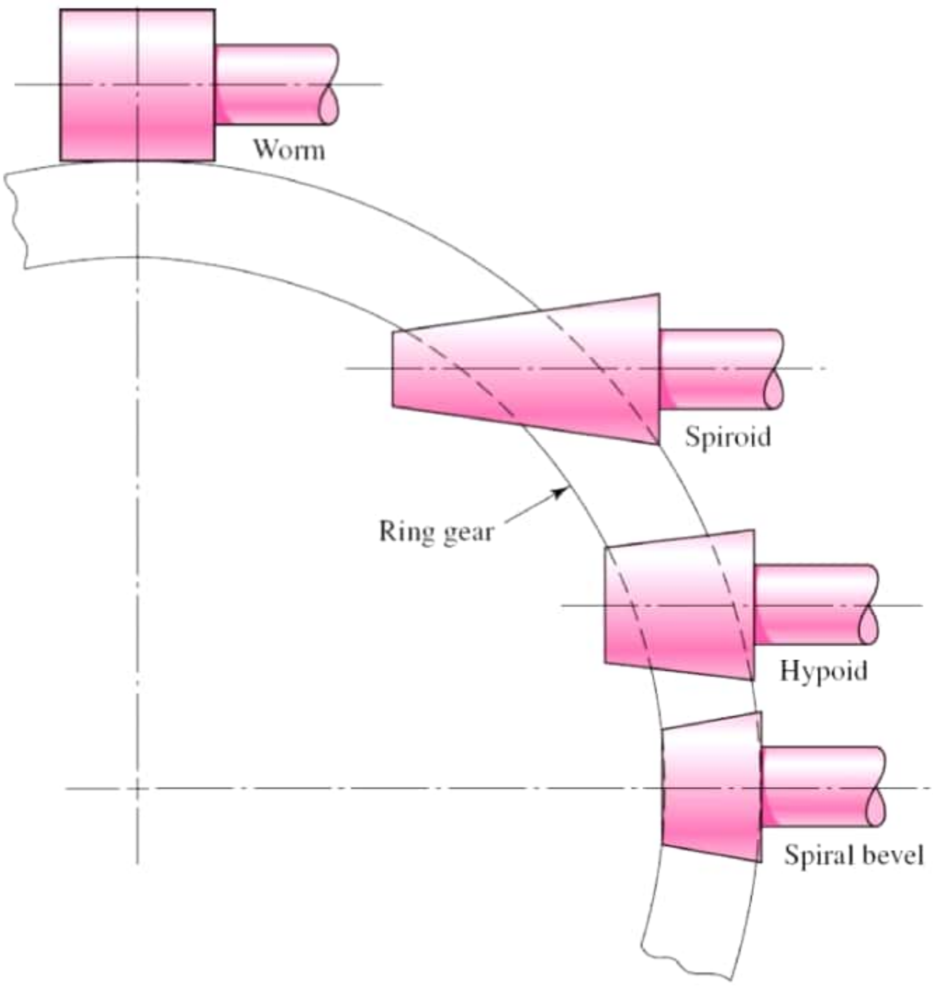


Figure- Comparison of intersecting and offset-shaft bevel-type gearings

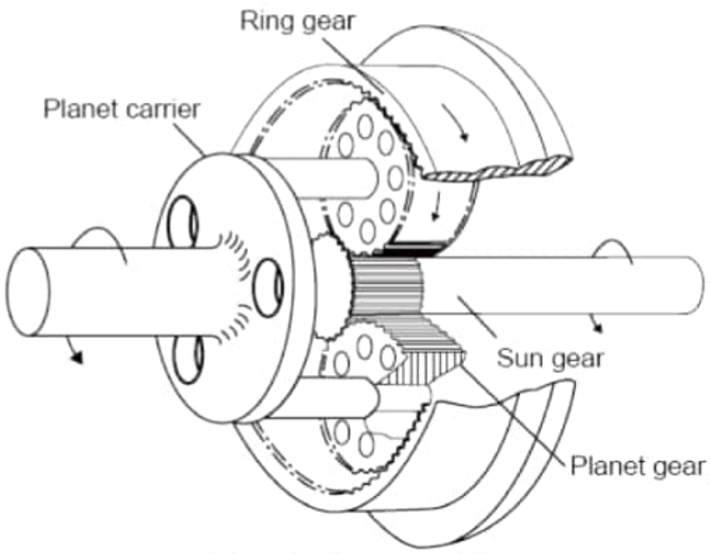


Figure-Epicyclic gears

Mitres gear

Miter gears are identical to bevel gears except that in a miter gear set, both gears always have the same number of teeth. Their ratio, therefore, is **always 1 to 1.** As a result, miter gears are not used when an application calls for a change of speed.

- **When equal bevel gears (having equal teeth) connect two shafts whose axes are mutually perpendicular, then the bevel gears are known as *mitres*.**

Straight Tooth *Spiral Tooth*

Figure- Miter gears

Minimum Number of Teeth on the Pinion in Order to Avoid Interference

The number of teeth on the pinion (T_p) in order to avoid interference may be obtained from the following relation:

$$\boxed{T_p = \frac{2A_w}{G\left[\sqrt{1+\frac{1}{G}\left(\frac{1}{G}+2\right)\sin^2\phi}-1\right]}}$$

Where A_W = Fraction by which the standard addendum for the wheel should be Multiplied, (generally $A_W = 1$)

G = Gear ratio or velocity ratio = $T_G / T_P = D_G / D_P$.

ϕ = Pressure angle or angle of obliquity.

- Minimum number of teeth for involute rack and pinion arrangement for pressure angle of **20°** is

$$T_{min} = \frac{2A_R}{\sin^2\theta} = \frac{2\times 1}{\sin^2 20^o} = 17.1 \quad as > 17 \qquad \text{So, } T_{min} = 18$$

- The minimum number of teeth on the pinion to operate without interference in standard full height involute teeth gear mechanism with 20° pressure angle is **18.**
- In **full depth** $14\frac{1}{2}^o$ degree involute system, the smallest number of teeth in a pinion which meshes with rack without interference is **32.**

Forms of teeth

Cycloidal teeth

A ***cycloid*** is the curve traced by a point on the circumference of a circle which rolls without slipping on a fixed straight line. When a circle rolls without slipping on the outside of a fixed circle, the curve traced by a point on the circumference of a circle is known as ***epicycloid.*** On the other hand, if a circle rolls without slipping on the inside of a fixed circle, then the curve traced by a point on the circumference of a circle is called ***hypocycloid.***

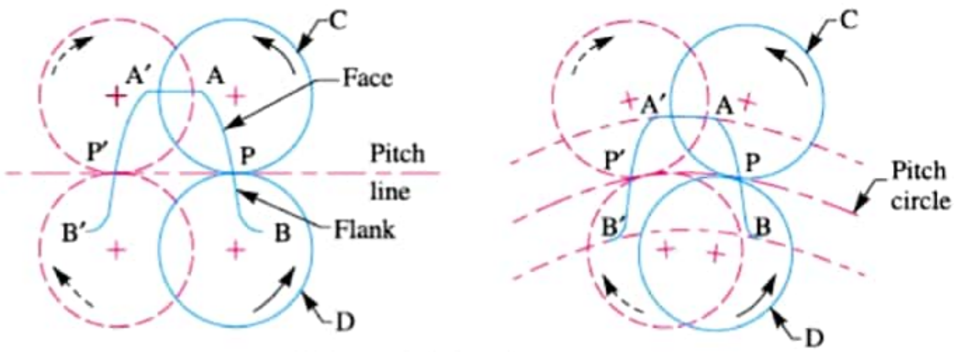


Fig. cycloidal teeth of a gear

Advantages of cycloidal gears

Following are the advantages of cycloidal gears:

1. Since the cycloidal teeth have wider flanks, therefore the cycloidal gears are stronger than the involute gears for the same pitch. Due to this reason, the cycloidal teeth are preferred especially for cast teeth.

2. In cycloidal gears, the contact takes place between a convex flank and concave surface, whereas in involute gears, the convex surfaces are in contact. This condition results in less wear in cycloidal gears as compared to involute gears. However the difference in wear is negligible.

3. In cycloidal gears, **the interference does not occur** at all. Though there are advantages of cycloidal gears but they are outweighed by the greater simplicity and flexibility of the involute gears.

Involute teeth

An involute of a circle is a plane curve generated by a point on a tangent, which rolls on the circle without slipping or by a point on a taut string which is unwrapped from a reel as shown in figure below. In connection with toothed wheels, the circle is known as base circle.

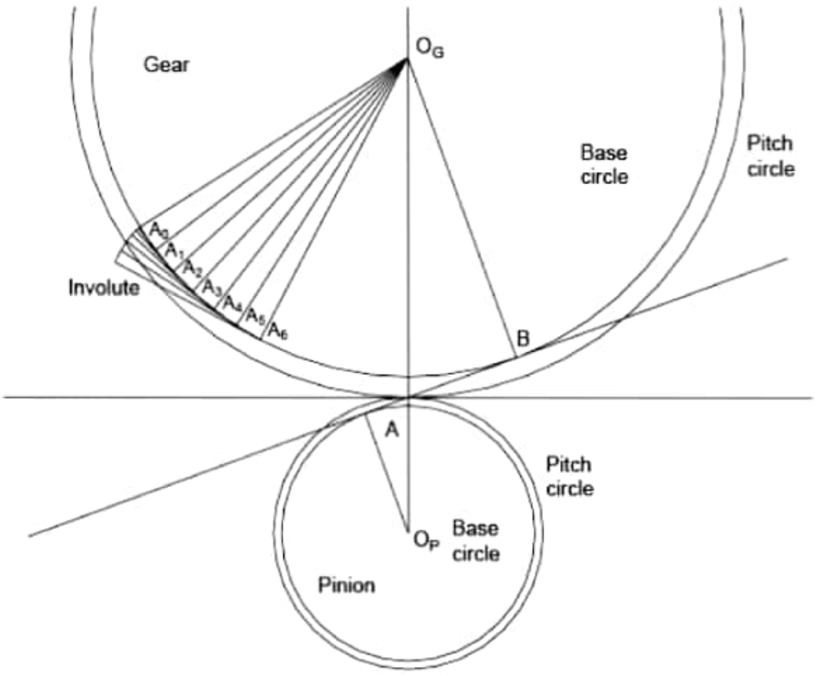


Fig.

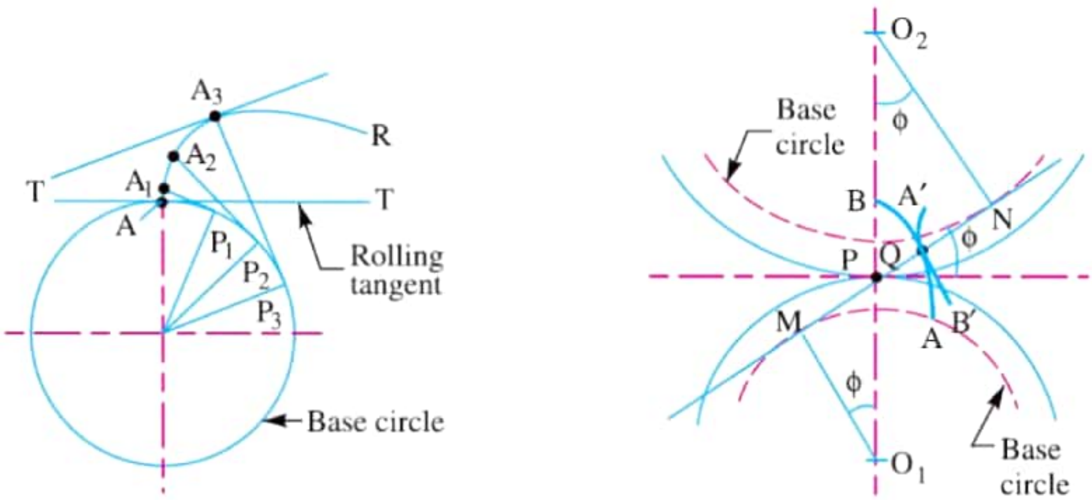


Figure-involute teeth

- The tooth profile most commonly used in gear drives for power transmission is an involute. It is due to easy manufacturing.

Advantages of involute gears

Following are the advantages of involute gears:

1. The most important advantage of the involute gears is that the centre distance for a pair of involute gears can be varied within limits without changing the velocity ratio. This is not true for cycloidal gears which require exact centre distance to be maintained.

2. In involute gears, the pressure angle, from the start of the engagement of teeth to the end of the engagement, remains **constant.** It is necessary for smooth running and less wear of gears. But in cycloidal gears, the pressure angle is maximum at the beginning of engagement, reduces to zero at pitch point, starts increasing and again becomes maximum at the end of engagement. This results in less smooth running of gears.

3. The face and flank of involute teeth are generated by a single curve whereas in cycloidal gears, double curves (*i.e.* epicycloids and hypocycloid) are required for the face and flank respectively.
Thus the involute teeth are easy to manufacture than cycloidal teeth. In involute system, the basic rack has straight teeth and the same can be cut with simple tools.

Note: The only disadvantage of the involute teeth is that **the interference occurs** with pinions having smaller number of teeth. This may be avoided by altering the heights of addendum and dedendum of the mating teeth or the angle of obliquity of the teeth.

Contact ratio

Note: The ratio of the length of arc of contact to the circular pitch is known as ***contact ratio*** *i.e.* number of pairs of teeth in contact.

$$\textbf{Contact ratio} = \frac{\text{length of arc of contact}}{\text{circular pitch}}$$

$$= \frac{\sqrt{R_{A^2} - R^2\cos^2\phi} + \sqrt{r_{A^2} - r^2\cos^2\phi} - (R+r)\sin\phi}{P_c(\cos\phi)}$$

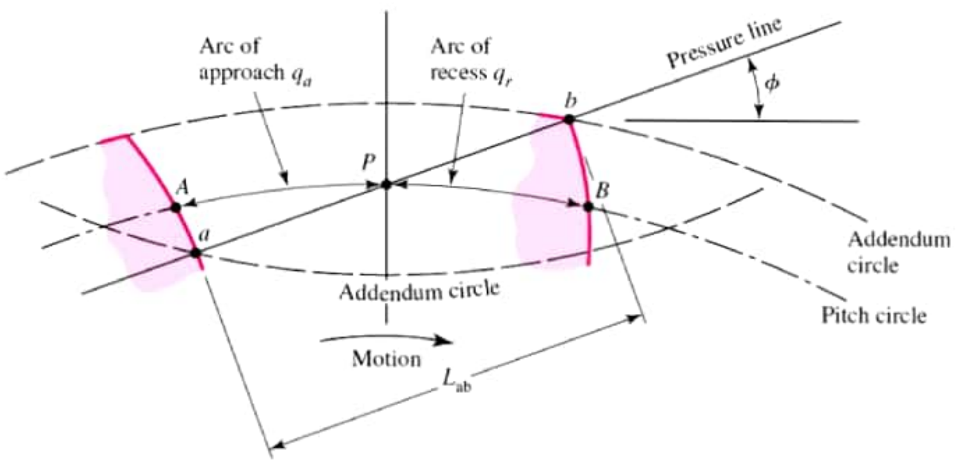


Fig.

The zone of action of meshing gear teeth is shown in figure above. We recall that tooth Contact begins and ends at the intersections of the two addendum circles with the pressure line. In figure above initial contact occurs at *a* and final contact at *b*. Tooth profiles drawn through these points intersect the pitch circle at *A* and *B*, respectively. As shown, the distance *AP* is called **the *arc of approach*** (q_a), and the distance *P B*, **the *arc of recess*** (q_r). The sum of these is the *arc of action* (q_t).

- The ratio of the length of arc of contact to the circular pitch is known as ***contact ratio*** *i.e.* number of pairs of teeth in contact. The contact ratio for gears is greater than one. **Contact ratio should be at least 1.25.** For maximum smoothness and quietness, the contact ratio should be between 1.50 and 2.00. High-speed applications should be designed with a face-contact ratio of 2.00 or higher for best results.

Interference

- **The contact of portions of tooth profiles that are not conjugate is called *interference*.**

- Contact begins when the tip of the driven tooth contacts the flank of the driving tooth. In this case the flank of the driving tooth first makes contact with the driven tooth at point A, and this occurs *before* the involute portion of the driving tooth comes within range. In other words, contact is occurring below the base circle of gear 2 on the *noninvolute* portion of the flank. The actual effect is that the involute tip or face of the driven gear tends to dig out the noninvolute flank of the driver.

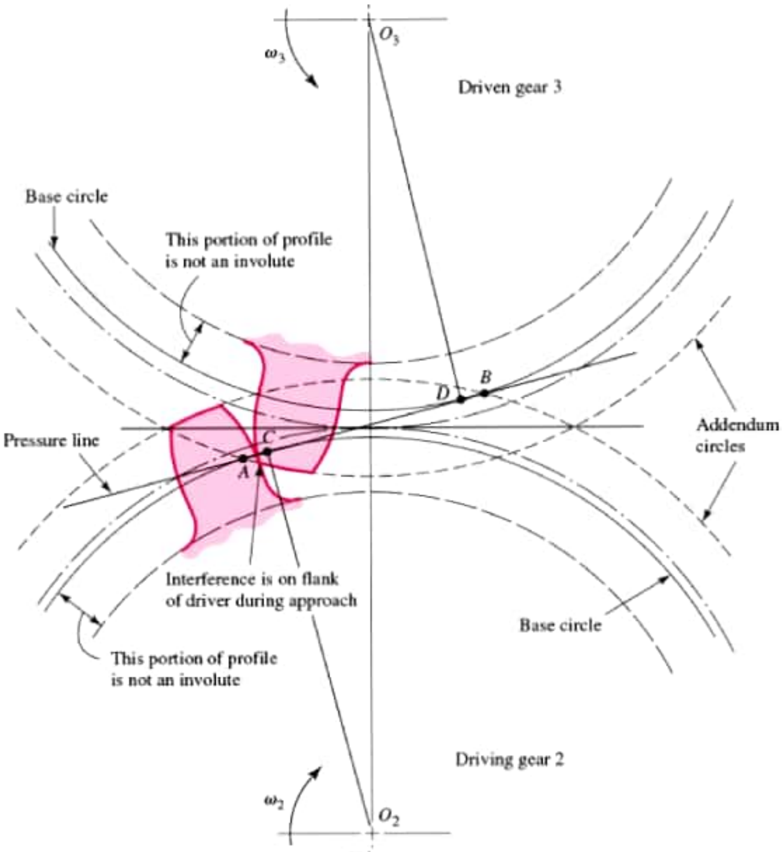


Fig.

- Interference can be eliminated by using more teeth on the pinion. However, if the pinion is to transmit a given amount of power, more teeth can be used only by increasing the pitch diameter.

- Interference can also be reduced by using a larger pressure angle. This results in a smaller base circle, so that more of the tooth profile becomes involute.

- The demand for smaller pinions with fewer teeth thus favors the use of a 25^0 pressure angle even though the frictional forces and bearing loads are increased and the contact ratio decreased.

- **There are several ways to avoid interfering:**
  i. Increase number of gear teeth
  ii. Modified involutes
  iii. Modified addendum
  iv. Increased centre distance.

Face Width

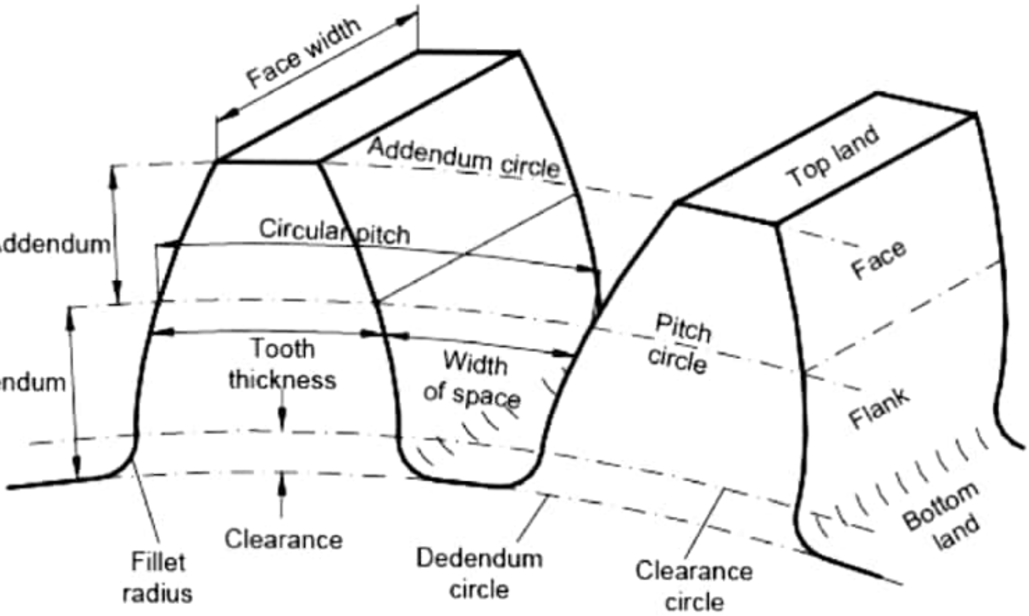
Face width. It is the width of the gear tooth measured parallel to its axis.

Face width.

We know that **face width,**

$$b = 10\ m$$

Where, **m is module.**



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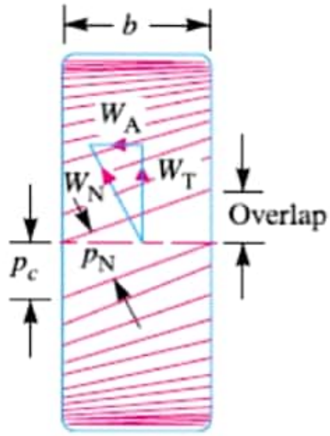


Fig. Face width of helical gear.

Beam Strength of Gear Tooth

The beam strength of gear teeth is determined from an equation **(known as Lewis equation**) and the load carrying ability of the toothed gears as determined by this equation gives satisfactory results. In the investigation, Lewis assumed that as the load is being transmitted from one gear to another, it is all given and taken by one tooth, because it is not always safe to assume that the load is distributed among several teeth, considering each tooth as a cantilever beam.

Notes: (*i*) The **Lewis equation** am applied only to the weaker of the two wheels (*i.e.* pinion or gear).

(*ii*) When both the pinion and the gear are made of the same material, then pinion is the weaker.

(*iii*) When the pinion and the gear are made of different materials, then the product of $(\sigma_w \times y)$ or $(\sigma_o \times y)$ is the deciding factor. The Lewis equation is used to that wheel for which $(\sigma_w \times y)$ or $(\sigma_o \times y)$ is less.

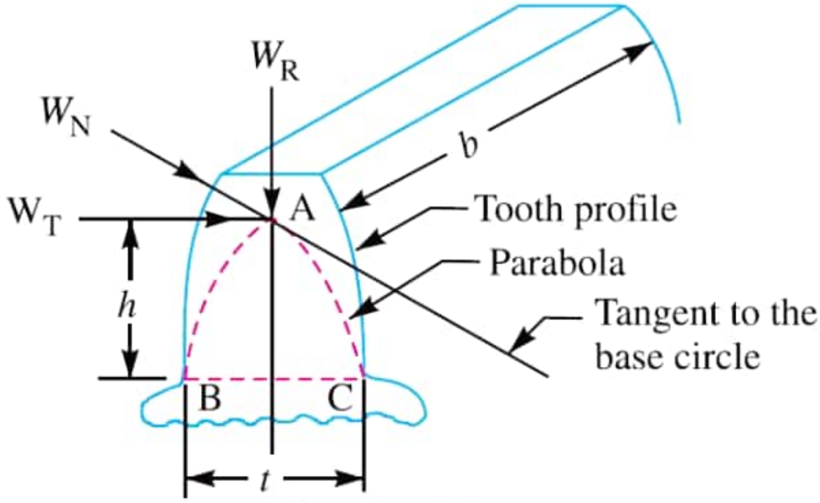


Figure- Tooth of a gear

The maximum value of the bending stress (or the permissible working stress):

$$\sigma_w = \frac{(W_T \times h)t/2}{b.t^3/12} = \frac{(W_T \times h)\times 6}{b.t^2}$$

Where
M = Maximum bending moment at the critical section $BC = W_T \times h$,
W_T = Tangential load acting at the tooth,
h = Length of the tooth,
y = Half the thickness of the tooth (t) at critical section $BC = t/2$,
I = Moment of inertia about the centre line of the tooth = $b.t^3/12$,
b = Width of gear face.
Lewis form factor or tooth form factor

$$W_T = \sigma_w \cdot b \cdot p_c \cdot y = \sigma_w \cdot b \cdot \pi m \cdot y$$

The quantity y is known as ***Lewis form factor*** or ***tooth form factor*** and W_T (which is the tangential load acting at the tooth) is called the ***beam strength of the tooth.***

Lewis form factor or ***tooth form factor***

$$y = 0.124 - \frac{0.684}{T}, \text{for } 14\tfrac{1}{2}^0 \text{ composite and full depth involute system.}$$

$$= 0.154 - \frac{0.912}{T}, \text{for } 20^0 \text{ full depth involute system.}$$

$$= 0.175 - \frac{0.841}{T}, \text{for } 20^0 \text{ stub system.}$$

Example: A spur gear transmits 10 kW at a pitch line velocity of 10 m/s; driving gear has a diameter of 1.0 m. find the tangential force between the driver and the follower, and the transmitted torque respectively.

Solution: Power transmitted = Force × Velocity

$$\Rightarrow \quad 10 \times 10^3 = \text{Force} \times 10$$

$$\Rightarrow \quad \text{Force} = \frac{10 \times 10^3}{10} = 1000\ \text{N/m}$$

$$\text{Torque Transmitted} = \text{Force} \times \frac{\text{diameter}}{2}$$

$$= 1000 \times \frac{1}{2} = 1000 \times 0.5$$

$$= 500\,\text{N} - \text{m} = 0.5\ \text{kN} - \text{m}$$

Wear Strength of Gear Tooth

Wear strength $(\sigma_w) = bQdpK$,

$$\text{Where, } Q = \frac{2T_g}{T_g + T_p} \qquad \text{for external gear}$$

$$= \frac{2T_g}{T_g - T_p} \qquad \text{for internal gear}$$

$$\textbf{load - stress factor}(k) = \frac{\sigma_c^2 \sin\phi\cos\phi}{1.4}\left(\frac{1}{E_1} + \frac{1}{E_2}\right)$$

$$= 0.16\left(\frac{BHN}{100}\right)^2$$

Gear Lubrication

All the major oil companies and lubrication specialty companies provide lubricants for gearing and other applications to meet a very broad range of operating conditions. General gear lubrication consists of high-quality machine oil when there are no temperature extremes or other adverse ambient conditions. Many of the automotive greases and oils are suitable for a broad range of gearing applications.

For adverse temperatures, environmental extremes, and high-pressure applications, consult the lubrication specialty companies or the major oil companies to meet your particular requirements or specifications.

The following points refer especially to spiral and hypoid bevel gears:

(a) Both spiral and hypoid bevel gears have combined rolling and sliding motion between the teeth, the rolling action being beneficial in maintaining a film of oil between the tooth mating surfaces.

(b) Due to the increased sliding velocity between the hypoid gear pair, a more complicated lubrication system may be necessary.

CHAPTER : GEAR TRAINS

Different Types Of Gear Trains :

1. Simple Gear Train
2. Compound Gear Train
3. Reverted Gear Train
4. Epicyclic Gear Train

Simple Gear train

A gear train is one or more pairs of gears operating together to transmit power. When two gears are in mesh, their pitch circles roll on each other without slippage.

If r_1 is pitch radius of gear 1, r_2 is pitch radius of gear 2; ω_1 is angular velocity of gear 1, and ω_2 is angular velocity of gear 2 then the pitch line velocity is given by

$$V = |r_1\omega_1| = |r_2\omega_2|$$

The velocity ratio is

$$\left|\frac{\omega_1}{\omega_2}\right| = \frac{r_2}{r_1}$$

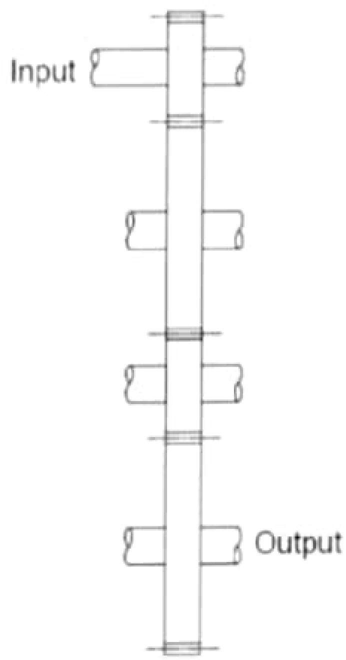


Figure- Simple gear train

Compound gear train

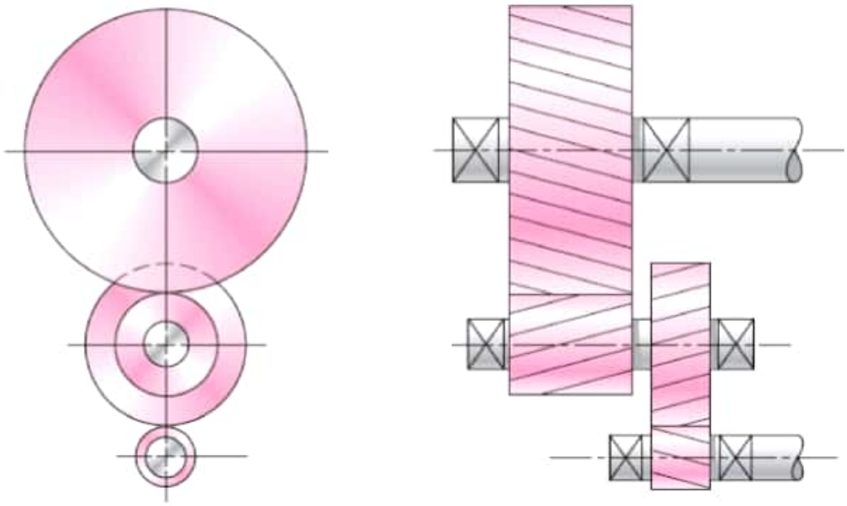


Figure -Compound gear train

The velocity ratio in the case of the compound train of wheels is equal to

$$= \frac{\text{Product of teeth on the followers}}{\text{Product of teeth on the drivers}}$$

The velocity ratio of the following gear train is

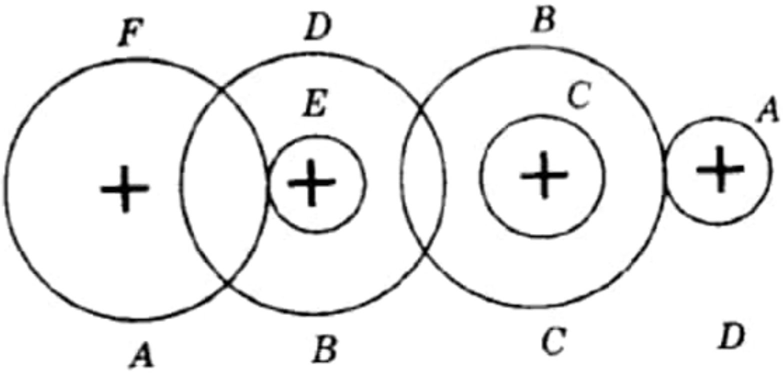


Figure- **velocity ratio**

$$\boxed{\frac{N_F}{N_A} = \frac{T_A \times T_C \times T_E}{T_B \times T_D \times T_F}}$$

Reverted gear train

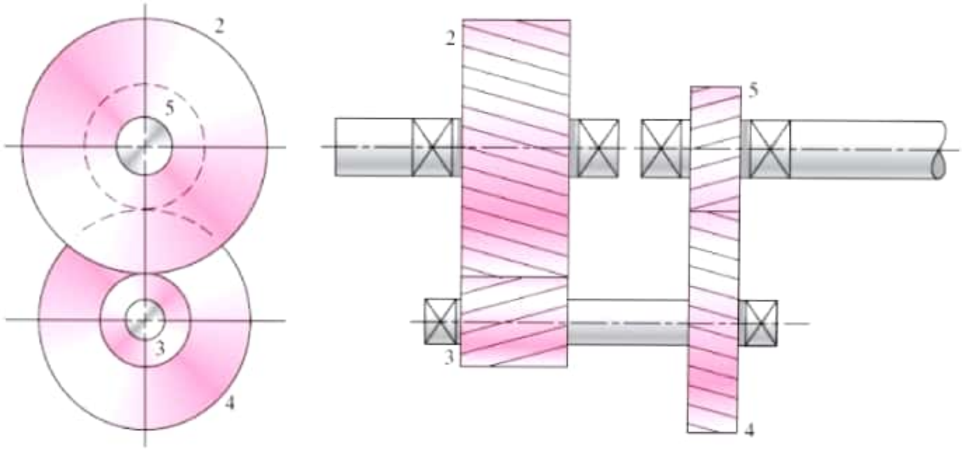


Figure- reverted gear train or a compound reverted gear train

It is sometimes desirable for the input shaft and the output shaft of a two-stage compound gear train to be in-line, as shown in Fig above. This configuration is called a *compound reverted gear train*. This requires the distances between the shafts to be the same for both stages of the train.

The distance constraint is

$$\frac{d_2}{2}+\frac{d_3}{2}=\frac{d_4}{2}+\frac{d_5}{2}$$

The diametric pitch relates the diameters and the numbers of teeth, $P = T/d$. Replacing All the diameters give

$$T_2/(2P)+T_3/(2P)=T_4/(2P)+T_5/(2P)$$

Assuming a constant diametral pitch in both stages, we have the geometry condition Stated in terms of numbers of teeth:

$$T_2 + T_3 = T_4 + T_5$$

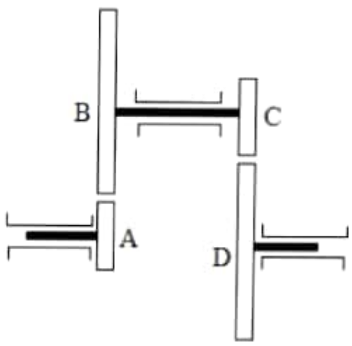
This condition must be exactly satisfied, in addition to the previous ratio equations, to Provide for the in-line condition on the input and output shafts.

In the compound gear train shown in the figure, gears A and C have equal numbers of teeth and gears B and D have equal numbers of teeth.

From the figure $r_A + r_B = r_C + r_D$ or $T_A + T_B = T_C + T_D$ and as $N_B + N_C$ it must be $T_B = T_D$ & $T_A = T_C$

Or $\frac{N_B}{N_A} = \frac{N_D}{N_C}$ or $N_C = \sqrt{N_A N_D}$

$[\text{where } N_B = N_c]$



Epicyclic gear train

Consider the following Epicyclic gear trai

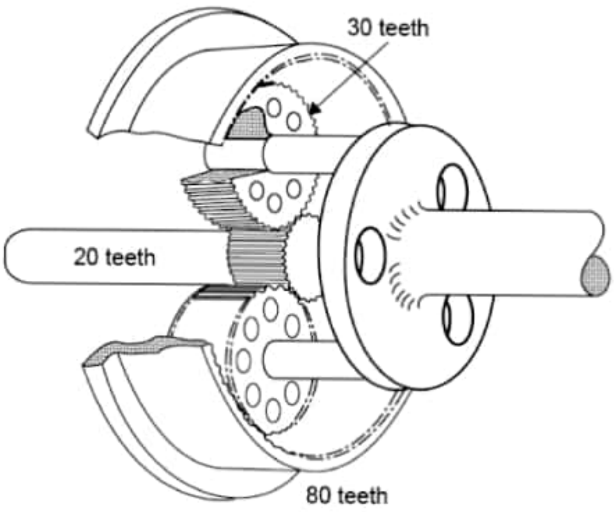


Figure- Epicyclic gear train

For the epicyclic gearbox illustrated in figure, determine the speed and direction of the final drive and also the speed and direction of the planetary gears. The teeth numbers of the sun, planets and ring gear are 20, 30 and 80, respectively. The speed and direction of the sun gear is 1000 rpm clockwise and the ring gear is held stationary.

Solution

$$n_{arm} = \frac{n_{sun}}{(80/20)+1} = \frac{-1000}{5} = -200\,rpm$$

The speed of the final drive is 200 rpm clockwise. The reduction ratio for the gearbox is Given by $n_{sun}/n_{arm} = 1000/200 = 5$. To determine the speed of the planets use

The planets and sun are in mesh, so

$$\frac{n_{planet} / n_{arm}}{n_{sum} / n_{arm}} = -\frac{N_S}{N_P}$$

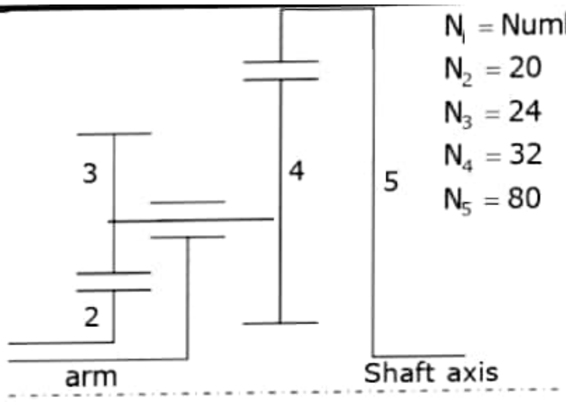
$$\frac{n_{planet} - n_{arm}}{n_{sum} - n_{arm}} = -\frac{N_S}{N_P}$$

$$\frac{n_{planet} - (-200)}{-1000-(-200)} = -\frac{20}{30}$$

$$n_{planet} = -\frac{20}{30} \times (-800) - 200 = 333\,\text{rpm}$$

The speed of rotation of the planetary gears is 333 rpm counter-clockwise.

Now make a table for **the epicyclic gear arrangement shown in the figure below.**



N_i = Number of teeth for gear i
$N_2 = 20$
$N_3 = 24$
$N_4 = 32$
$N_5 = 80$

	Arm	2	3	4	5
1.	0	+x	$\frac{-N_2}{N_3}x$	$\frac{-N_2}{N_3}x$	$\frac{-N_4}{N_5}\times\frac{N_2}{N_3}x$
2.	y	y	y	y	y
	y	x + y	$y - \frac{N_2}{N_3}x$		$y - \frac{N_4}{N_5}\times\frac{N_2}{N_3}x$

Formula List for Gears:

(a) Spur Gear

Name	Speed ratio	
1. Spur & Helical	6:1 to 10:1	for high speed helical For high speed spur.
2. Bevel	1:1 to 3:1	
3. Worm	10:1 to 100:1 provided $\angle 100$ KW	

SPUR GEAR

(i) Circular pitch (p) = $\frac{\pi d}{T}$

(ii) Diametral pitch (P) = $\frac{T}{d}$

(iii) $pP = \pi$

(iv) Module (m) = $\frac{d}{T} = \frac{1}{P}$ or $d = mT$

(v) Speed ratio (G) = $\frac{\omega_p}{\omega_g} = \frac{T_g}{T_p}$

(vi) centre-to-centre distance = $\frac{1}{2}(d_g + d_p)$

$= \frac{1}{2}m(T_g + T_p)$

(vii) Addendum $(h_a) = 1m$

$\mathbf{(h_f) = 1.25m}$

Clearance (C) = 0.25 m

(viii) $P_t = \frac{2T}{d}$

$$P_r = P_t \tan\alpha$$

$$P_N = \frac{P_t}{\cos\alpha}$$

(ix) Minimum number of teeth to avoid interference

$$T_{min} = \frac{2A_w}{\sin^2\phi} \quad \Big\| \text{ For } 20° \text{ full depth } T = 18 \text{ to } 20$$

(x)

$$T_{min,\,pinion} = \frac{2 \times A_w}{G\left[\sqrt{1 + \frac{1}{G}\left(\frac{1}{G} + 2\right)\sin^2\phi} \; -1\right]}$$

$$G = \textbf{Gear ratio} = \frac{z_g}{z_p} = \frac{\omega_p}{\omega_g}$$

$$A_p = \text{fraction of addendum to module} = \frac{h_a}{m} \text{ for pinion}$$

$$A_w = \text{fraction of addendum to module} = \frac{h_f}{m} \text{ for gear (generally 1)}$$

(xi) Face width 8m <b <12m; usually b = 10m

(xii) Beam strength $\sigma_b = mb\sigma_b Y$ → **Lewis equation**

Where $\sigma_b = \frac{\sigma_{ult}}{3}$

Lewis form factor, $Y = \left(0.154 - \frac{0.912}{z}\right)$ **for 20° full depth gear.**

(xiii) Wear strength (σ_ω) = **bQdpK**

$$\text{Where } Q = \frac{2T_g}{T_g + T_p} \quad \textbf{for external gear}$$

$$= \frac{2T_g}{T_g - T_p} \quad \textbf{for internal gear}$$

$$\textbf{load - stress factor}(k) = \frac{\sigma_c^2 \sin\phi\cos\phi}{1.4}\left(\frac{1}{E_1} + \frac{1}{E_2}\right)$$

$$= 0.16\left(\frac{BHN}{100}\right)^2$$

(xiv)

$$P_{eff} = \frac{C_s}{C_v} P_t \quad \text{where } C_v = \frac{3}{3+V}, \qquad \text{for ordinary cut gear } v < 10\,m/s$$

$$= \frac{6}{6+v}, \qquad \text{for hobbed generated } v > 20\,m/s$$

$$= \frac{5.6}{5.6+\sqrt{v}}, \qquad \text{for precision gear } v > 20\,m/s$$

(xv)

Spott's equation, $(P_{eff}) = C_s P_t + P_d$

$$\text{where } P_d = \frac{e n_p T_p b r_1 r_2}{2530\sqrt{r_1^2 + r_2^2}} \text{ for steel pinion and steel gear}$$

$$e = 16.00 + 1.25\phi \text{ for grade 8}$$

$$\phi = m + 0.29\sqrt{d} \text{ for all.}$$

(b) Helical Gear

(1) $\mathbf{p_n = p\cos\alpha}$ where p = transverse circular pitch.
p_n = normal circular pitch.

(2) $\mathbf{P_n = \dfrac{P}{\cos\alpha}}$ where P_n = normal diametral pitch.
P = transverse diametral pitch.
α = helix angle.

(3) $pP = \pi$

(4) $m_n = m\cos\alpha \left[P = \dfrac{1}{m} \right]$ m = transverse module.
m_n = normal module.

(5) $p_a = \dfrac{p}{\tan\alpha} = \dfrac{\pi m}{\tan\alpha}$; p_a = axial pitch.

(6) $\cos\alpha = \dfrac{\tan\phi_n}{\tan\phi}$ ϕ_n = normal pressure angle (usually 20°).
ϕ = transverse pressure angle.

(7) $d = \dfrac{TP}{\pi} = zm = \dfrac{Tm_n}{\cos\alpha}$; d = pitch circle diameter.

(8) $a = \dfrac{m_n}{2\cos\alpha}\{T_1 + T_2\}$ → centre – to - centre distance.

(9) $T' = \dfrac{T}{\cos^3\alpha}$; $d' = \dfrac{d}{\cos^2\alpha}$

(10) An imaginary spur gear is considered with a pitch circle diameter of d' and module m_n is called 'formative' or 'virtual' spur gear.

(11) Helix angle α varies from **15 to 25°.**

(12) Preference value of normal modus (m_n) = 1, 1.25, 1.5, 2, 3, 4, 5, 6, 8

(13) Addendum $(h_a) = m_n$; dedendum $(h_f) = 1.25\ m_n$, clearence $= 0.25 m_n$

(14)

$$P_t = \frac{2M_t}{d} = P\cos\phi_n\cos\alpha;\ M_t = \frac{60 \times 10^6 \times (KW)}{2\pi N}\ \text{N - mm}$$

$$P_r = P_t\left(\frac{\tan\phi_n}{\cos\alpha}\right) = P\sin\alpha\phi_n$$

$$P_a = P_t\tan\alpha = P\cos\phi_n\sin\alpha$$

(15) Beam strength $S_b = m_n b\sigma_b Y'$

(16) Wear strength $S_w = \dfrac{bQd_pK}{\cos^2\alpha}$

(17)

$$P_{eff} = \frac{C_s P_t}{C_v} \; ; \; C_v = \frac{5.6}{5.6+\sqrt{v}} \; ; \; P_d = \frac{e n_p T_p b r_1 r_2}{2530\sqrt{r_1^2 + r_2^2}}$$

$$P_{eff} = C_s P_t + P_d \cos\alpha_n \cos\psi$$

(c) Worm Gear

(i) Specified and designated by $T_1 / T_2 / q / m$

Where: q is diametric quotient $= \frac{d_1}{m}$

(ii) The threads of the worm have an involute helicoids profile.

(iii) Axial pitch (p_x) = distance between two consecutive teeth-measured along the axis of the worm.

(iv) The lead (l) = when the worm is rotated one revolution, a distance that a point on the helical profile will move.

(v) $l = p_x \times T_1$; $d_z = mT_2$

(vi) Axial pitch of the worm = circular pitch of the worm wheel

$$P_x = \frac{\pi d_2}{T_2} = \pi m \qquad \text{[ICS - 04]}$$

$$l = P_x T_1 = \pi m T_1$$

(vii) Lead angle $(\delta) = \tan^{-1}\left(\frac{T_1}{q}\right) = \tan^{-1}\left(\frac{l}{\pi d_1}\right)$

(viii) centre-to-centre distance (a) $= \frac{1}{2}(d_1 + d_2) = \frac{1}{2}m(T_1 + T_2)$

(ix) Preferred values of q: 8, 10, 12.5, 16, 20, 25

(x) Number of starts T_1 usually taken 1, 2, or 4

(xi)

addendum $(h_{a_1}) = m$	$h_{a_2} = m(2\cos\delta - 1)$
dedendum $(h_{f_1}) = (2.2\cos\delta - 1)m$	$h_{f_2} = m(1 + 0.2\cos\delta)$
clearance $(c) = 0.2m\cos\delta$	$c = 0.2\, m \cos\delta$

(xii) $F = 2m\sqrt{(q+1)}$ effective face width of the root of the worm wheel.

(xiii)

$\delta = \sin^{-1}\left(\frac{F}{d_{a_1} + 2C}\right)$; $l_r = (d_{a_1} + 2C)\sin^{-1}\left(\frac{F}{d_{a_1} + 2C}\right)$ = length of the root of the worm wheel teeth

(xiv) $\left(P_1\right)_t = \frac{2m_t}{d_1}$; $\left(P_1\right)_a = \left(P_1\right)_t \frac{\cos\alpha\cos\delta - \mu\sin\delta}{\cos\alpha\sin\delta + \mu\cos\delta}$; $\left(P_1\right)_r = \left(P_1\right)_t \frac{\sin\alpha}{\cos\alpha\sin\delta + \mu\cos\delta}$

(xv) Efficiency $(\eta) = \frac{\text{Power output}}{\text{Power input}} = \frac{\cos\alpha - \mu\tan\delta}{\cos\alpha + \mu\tan\delta}$

(xvi) Rubbing velocity $\left(V_s\right) = \frac{\pi d_1 n_1}{60000\cos\delta}$ m / s (remaining 4 cheak)

(xvii) Thermal consideration $H_g = 1000(1-\eta)\times(KW)$

$$H_d = K\left(t - t_0\right)A$$

$$KW = \frac{K\left(t - t_0\right)A}{1000(1-\eta)}$$

Objective Questions (IES, IAS, GATE)

Previous 20-Years GATE Questions

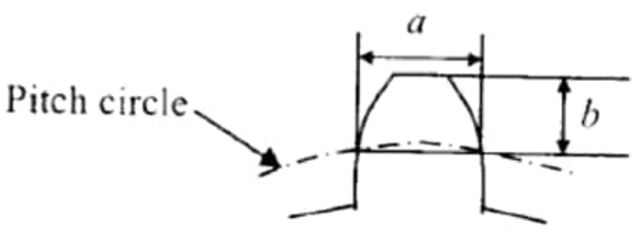
Spur gear

GATE-1. Match the type of gears with their most appropriate description. [GATE-2008]

Type of gear	**Description**
P Helical	**1. Axes non parallel and intersecting**
Q Spiral	**2. Axes parallel and teeth are inclined to the axis**
R Hypoid	**3. Axes parallel and teeth are parallel to the axis**
S Rack and pinion	**4. Axes are perpendicular and intersecting, and teeth are inclined to the axis**
	5. Axes are perpendicular and used for large speed reduction
	6. Axes parallel and one of the gears has infinite radius

(a) P-2, Q- 4, R- 1, S- 6
(b) P-1, Q- 4, R- 5, S- 6
(c) P-2, Q- 6, R- 4, S- 2
(d) P-6, Q- 3, R- 1, S- 5

GATE-2. One tooth of a gear having 4 module and 32 teeth is shown in the figure. Assume that the gear tooth and the corresponding tooth space make equal intercepts on the pitch circumference. The dimensions *'a'* and *'b'*, respectively, are closest to [GATE-2008]



(a) 6.08 mm, 4 mm
(b) 6.48 mm, 4.2 mm
(c) 6.28 mm, 4.3 mm
(d) 6.28 mm, 4.1

Classification of Gears

GATE-3. Match the following [GATE-2004]

Type of gears	**Arrangement of shafts**
P. Bevel gears	**1. Non-parallel off-set shafts**
Q. Worm gears	**2. Non-parallel intersecting shafts**
R. Herringbone gears	**3. Non-parallel non-intersecting shafts**
S. Hypoid gears	**4. Parallel shafts**

(a) P-4 Q-2 R-1 S-3
(b) P-2 Q-3 R-4 S-1
(c) P-3 Q-2 R-1 S-4
(d) P-1 Q-3 R-4 S-2

Pitch point

GATE-4. In spur gears, the circle on which the involute is generated is called the
(a) Pitch circle
(b) clearance circle **[GATE-1996]**

(c) Base circle (d) addendum circle

Minimum Number of Teeth

GATE-5. The minimum number of teeth on the pinion to operate without interference in standard full height involute teeth gear mechanism with 20° pressure angle is [GATE-2002]

(a) 14 (b) 12 (c) 18 (d) 32

Interference

GATE-6 Tooth interference in an external in volute spur gear pair can be reduced by

(a) Decreasing center distance between gear pair **[GATE-2010]**
(b) Decreasing module
(c) Decreasing pressure angle
(d) Increasing number of gear teeth

GATE-7. Interference in a pair of gears is avoided, if the addendum circles of both the gears intersect common tangent to the base circles within the points of tangency. [GATE-1995]

(a) True (b) False
(c) Insufficient data (d) None of the above

GATE-8. Twenty degree full depth involute profiled 19-tooth pinion and 37-tooth gear are in mesh. If the module is 5 mm, the centre distance between the gear pair will be [GATE-2006]

(a) 140 mm (b) 150 mm
(c) 280 mm (d) 300 mm

Beam Strength of Gear Tooth

GATE-9. A spur gear has a module of 3 mm, number of teeth 16, a face width of 36 mm and a pressure angle of 20°. It is transmitting a power of 3 kW at 20 rev/s. Taking a velocity factor of 1.5, and a form factor of 0.3, the stress in the gear tooth is about [GATE-2008]

(a) 32 MPa (b) 46 MPa
(c) 58 MPa (d) 70MPa

Statement for Linked Answer GATE-10 and GATE-11:
A 20° full depth involute spur pinion of 4 mm module and 21 teeth is to transmit 15 kW at 960 rpm. Its face width is 25 mm.

GATE-10. The tangential force transmitted (in N) is [GATE -2009]

(a) 3552 (b) 261 1 (c) 1776 (d) 1305

GATE-11. Given that the tooth geometry factor is 0.32 and the combined effect of dynamic load and allied factors intensifying the stress is 1.5; the minimum allowable stress (in MPa) for the gear material is [GATE -2009]

(a) 242.0 (b) 166.5 (c) 121.0 (d) 74.0

Simple Gear train

Note: - Common Data for GATE-12 & GATE-13.
A gear set has a opinion with 20 teeth and a gear with 40 teeth. The pinion runs at 0 rev/s and transmits a power of 20 kW. The teeth are on the 20° full –depth system and have module of 5 mm. The length of the line of action is 19 mm.

GATE-12. **The center distance for the above gear set in mm is** **[GATE-2007]**
(a) 140 (b) 150 (c) 160 (d) 170.

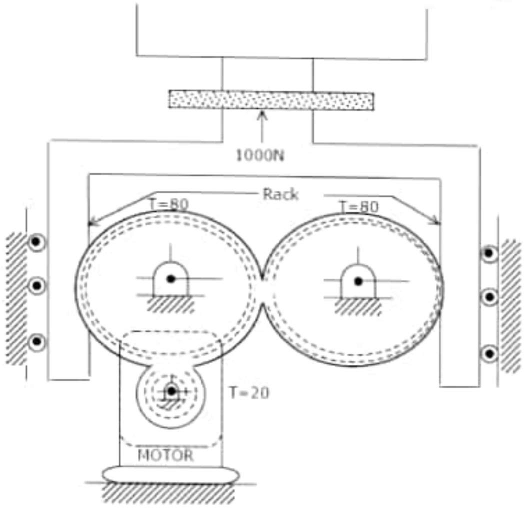
GATE-13 **The contact ratio of the contacting tooth** **[GATE-2007]**
(a) 1.21 (b) 1.25 (c) 1.29 (d) 1.33

GATE-14. **The resultant force on the contacting gear tooth in N is:** **[GATE-2007]**
(a) 77.23 (b) 212.20 (c) 225.80 (d) 289.43

Compound gear train

Data for GATE-15 & GATE-16 are given below. Solve the problems and choose correct answers.

A compacting machine shown in the figure below is used to create a desired thrust force by using a rack and pinion arrangement. The input gear is mounted on tile motor shaft. The gears have involute teeth of 2 mm module.



GATE-15. **If the drive efficiency is 80%, then torque required on the input shaft to create 1000 N output thrust is** **[GATE-2004]**
(a) 20 Nm (b) 25 Nm (c) 32 Nm (d) 50 Nm

GATE-16. **If the pressure angle of the rack is 20°, then force acting along the line of action between the rack and the gear teeth is** **[GATE-2004]**
(a) 250 N (b) 342 N (c) 532 N (d) 600 N

Reverted gear train

Data for GATE-17 & GATE-18 are given below. Solve the problems and choose correct answers.

The overall gear ratio in a 2 stage speed reduction gear box (with all spur gears) is 12. The input and output shafts of the gear box are collinear. The countershaft which is parallel to the input and output shafts has a gear (Z_2 teeth) and pinion (Z_3 = 15 teeth) to mesh with pinion (Z_1 = 16 teeth) on the input shaft and gear (Z_4 teeth) on the output shaft respectively. It was decided to use a gear ratio of 4 with 3 module in the first stage and 4 module in the second stage.

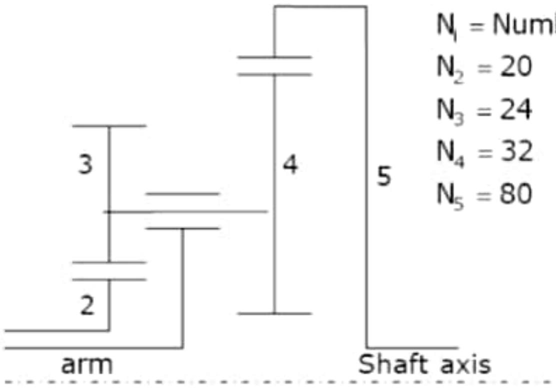
GATE-17. **Z_2 and Z_4 are** **[GATE-2003]**
(a) 64 and 45 (b) 45 and 64 (c) 48 and 60 (d) 60 and 48

GATE-18. **The centre distance in the second stage is** **[GATE-2003]**
(a) 90 mm (b) 120 mm (c) 160 mm (d) 240mm

Epicyclic gear train

GATE-19. For the epicyclic gear arrangement shown in the figure, ω_2 = 100 rad/s clockwise (CW) and ω_{arm} = 80 rad/s counter clockwise (CCW). The angular velocity ω_5, (in rad/s) is

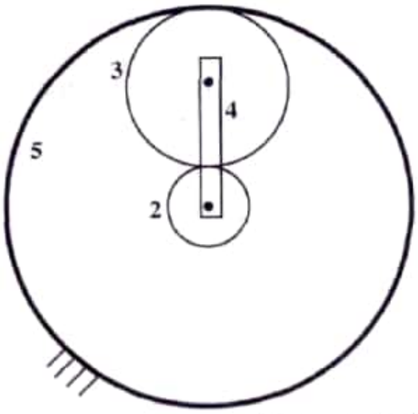
[GATE-2010]



(a) 0 (b) 70 CW (c) 140 CCW (d) 140 CW

GATE-20. An epicyclic gear train is shown schematically in the adjacent figure.
The sun gear 2 on the input shaft is a 20 teeth external gear. The planet gear 3 is a 40 teeth external gear. The ring gear 5 is a 100 teeth internal gear. The ring gear 5 is fixed and the gear 2 is rotating at 60 rpm (ccw = counter-clockwise and cw = clockwise).
The arm 4 attached to the output shaft will rotate at

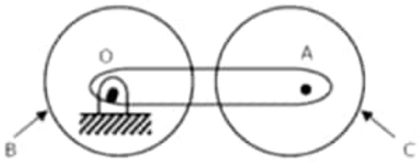
(a) 10 rpm ccw
(b) 10 rpm ccw
(c) 12 rpm cw
(d) 12 rpm ccw



[GATE -2009]

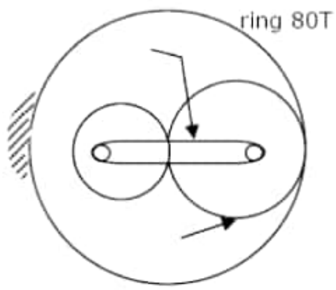
GATE-21 The arm OA of an epicyclic gear train shown in figure revolves counter clockwise about O with an angular velocity of 4 rad/s. Both gears are of same size. Tire angular velocity of gear C, if the sun gear B is fixed, is [GATE-1995]

(a) 4 rad / sec
(b) 8 rad / sec
(c) 10 rad / sec
(d) 12 rad / sec



GATE-22. The sun gear in the figure is driven clockwise at 100 rpm. The ring gear is held stationary.
For the number of teeth shown on the gears, the arm rotates at

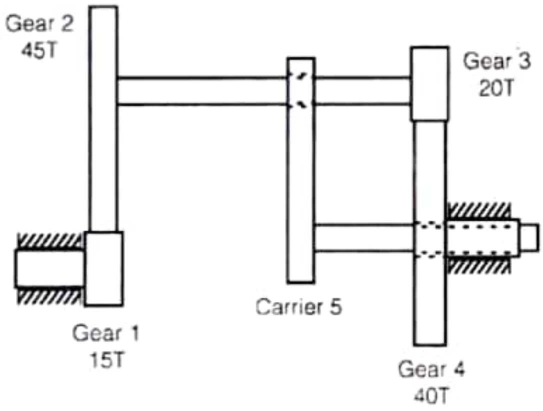
(a) 0 rpm (b) 20 rpm
(c) 33.33 rpm (d) 66.67 rpm



[GATE-2001]

GATE-23. **Two mating spur gears have 40 and 120 teeth respectively. The pinion rotates at 1200 rpm and transmits a torque of 20 Nm. The torque transmitted by the gear is** **[GATE-2004]**
(a) 6.6 Nm (b) 20 Nm (c) 40 Nm (d) 60Nm

Common Data for GATE-24, GATE-25:
A planetary gear train has four gears and one carrier. Angular velocities of the gears are ω_1, ω_2, ω_3, and ω_4 respectively. The carrier rotates with angular velocity ω_5,



GATE-24. **What is the relation between the angular velocities of Gear 1 and Gear 4?** **[GATE-2006]**

GATE-25. **For (ω_1 = 60 rpm clockwise (cw) when looked from the left, what is the angular velocity of the carrier and its direction so that Gear 4 rotates in counter clockwise (ccw) direction at twice the angular velocity of Gear 1 when looked from the left?** **[GATE-2006]**
(a) 130 rpm, cw (b) 223 rpm, ccw
(c) 256 rpm, cw (d) 156 rpm, ccw

Worm Gears

GATE-26. **Large speed reductions (greater than 20) in one stage of a gear train are possible through** **[GATE-2002]**
(a) Spur gearing (b) Worm gearing (c) Bevel gearing (d) Helical gearing

GATE-27. **A 1.5 kW motor is running at 1440 rev/min. It is to be connected to a stirrer running at 36 rev /min. The gearing arrangement suitable for this application is** **[GATE-2000]**
(a) Differential gear (b) helical gear
(c) Spur gear (d) worm gear

GATE-28. **To make a worm drive reversible, it is necessary to increase** **[GATE-1997]**
(a) centre distance (b) worm diameter factor
(c) Number of starts (d) reduction ratio

Previous 20-Years IES Questions

Spur gear

IES-1. **The velocity ratio between pinion and gear in a gear drive is 2.3, the module of teeth is 2.0 mm and sum of number of teeth on pinion and gear is 99. What is the centre distance between pinion and the gear? [IES 2007]**

(a) 49.5 mm (b) 99 mm (c) 148.5 mm (d) 198 mm

IES-2. **Consider the following statements: [IES-2001]**
When two gears are meshing, the clearance is given by the
1. Difference between dedendum of one gear and addendum of the mating gear.
2. Difference between total and the working depth of a gear tooth.
3. Distance between the bottom land of one gear and the top land of the mating gear.
4. Difference between the radii of the base circle and the dedendum circle.
Which of these statements are correct?
(a) 1, 2 and 3 (b) 2, 3 and 4 (c) 1, 3 and 4 (d) 1, 2 and 4

IES-3. **The working surface above the pitch surface of the gear tooth is termed as [IES-1998]**
(a) Addendum (b) dedendum (c) flank (d) face

IES-4. **Match the following** $14\frac{1}{2}^{o}$ **composite system gears**

[IES-1992]

List I	**List II**
A. Dedendum	**1.** $\frac{2}{pd}$
B. Clearance	**2.** $\frac{0.157}{pd}$
C. Working depth	**3.** $\frac{1.157}{pd}$
D. Addendum	**4.** $\frac{1}{pd}$

Code:	A	B	C	D		A	B	C	D
(a)	1	2	3	4	(b)	4	3	2	1
(c)	3	2	1	4	(d)	3	1	2	4

IES-5. Match List I with List II and select the correct answer using the codes given below the lists: **[IES-1993]**

List I (Standard tooth/arms)	**List II (Advantages or disadvantages)**
A. 20° and 25° systems	**1. Results in lower loads on bearing**
B. 14.5° stub-tooth system	**2. Broadest at the base and strongest in bending**
C. 25° Full depth system	**3. Obsolete**
D. 20° Full depth system	**4. Standards for new applications**

Code:	A	B	C	D		A	B	C	D
(a)	4	3	2	1	(b)	3	1	2	4
(c)	3	2	1	4	(d)	4	2	3	1

IES-6. **Assertion (A):** When one body drives another by direct contact, their contact points must have equal components of velocity normal to the surfaces at the point of contact.

Reason (R): Two points in the same body must have the same component of velocity relative to the third body, in the direction of the line joining the two points. **[IES-1993]**
(a) Both A and R are individually true and R is the correct explanation of A
(b) Both A and R are individually true but R is **not** the correct explanation of A
(c) A is true but R is false
(d) A is false but R is true

Classification of Gears

IES-7. Match List I with List II and select the correct answer [IES-1996]

List I	List II
A. Helical gears	**1. Non-interchangeable**
B. Herring bone gears	**2. Zero axial thrust**
C. Worm gears	**3. Quiet motion**
D. Hypoid Gears	**4. Extreme speed reduction**

Codes:	A	B	C	D		A	B	C	D
(a)	1	2	3	4	(b)	3	2	1	4
(c)	3	1	4	2	(d)	3	2	4	1

IES-8. Match List-I (Type of Gears) with List-II (Characteristics) and select the correct answer using the code given below the Lists: [IES-2006]

List-I	List -II
A. Helical gearing	**1. Zero axial thrust**
B. Herringbone gearing	**2. Non-inter-changeable**
C. Worm gearing	**3. Skew shafts**
D. Hypoid gearing	**4. Parallel shafts**

	A	B	C	D		A	B	C	D
(a)	4	1	3	2	(b)	3	2	4	1
(c)	4	2	3	1	(d)	3	1	4	2

IES-9. Match List I with List II and select the correct answer using the code given below the Lists: [IES 2007]

List I	List II
A. Worm gear	**1. Imposes no thrust load on the shaft**
B. Spur gear	**2. To transmit power between two non-intersecting shafts which are perpendicular to each other**
C. Herringbone gear	**3. To transmit power when the shafts are parallel**
D. Spring level gear	**4. To transmit power when the shafts are at right angles to one another**

Code:	**A**	**B**	**C**	**D**		**A**	**B**	**C**	**D**
(a)	1	2	3	4	(b)	2	3	1	4
(c)	1	2	4	3	(d)	2	3	4	1

IES-10. Match List I (Type of Gear/Gear Train) with List II (Different Usage and Drive) and select the correct answer using the code given below the Lists: [IES-2005]

List I	List II
A Epicyclic gear train	**1. Reduces end thrust**
B. Bevel Gear	**2. Low gear ratio**
C. Worm-worm Gear	**3. Drives non-parallel nonintersecting shafts**
D. Herringbone Gear	**4. Drives non-parallel intersecting shafts**
	5. High gear ratio

	A	**B**	**C**	**D**		**A**	**B**	**C**	**D**
(a)	5	4	3	1	(b)	2	3	4	5
(c)	5	3	4	1	(d)	2	4	3	5

IES-11. Which type of gear is used for shaft axes having an offset? [IES-2004]

(a) Mitre gears
(b) Spiral bevel gears
(c) Hypoid gears
(d) Zerol gears

IES-12. The gears employed for connecting two non-intersecting and non-parallel, i.e., non-coplanar shafts are **[IES-2003; 2005]**
(a) Bevel gears (b) Spiral gears (c) Helical gears (d) Mitre gears

IES-13. When two shafts are neither parallel nor intersecting, power can be transmitted by using **[IES-1998]**
(a) A pair of spur gears
(b) a pair of helical gears
(c) An Oldham's coupling
(d) a pair of spiral gears

IES-14. In a single reduction, a large velocity ratio is required. The best transmission is **[IES-1999]**
(a) Spur gear drive
(b) helical gear drive
(c) Bevel gear drive
(d) worm gear drive

IES-15. Which one of the following pairs is not correctly matched? **[IES-1995]**
(a) Positive drive .... Belt drive
(b) High velocity ratio .... Worm gearing
(c) To connect non-parallel and non- intersecting shafts .... Spiral gearing.
(d) Diminished noise and smooth operation .... Helical gears.

Mitres gear

IES-16. Mitre gears **[IES-1992]**
(a) spur-gears with gear ratio 1: 1
(b) Skew gears connecting non-parallel and nonintersecting shafts
(c) Bevel gears transmitting power at more than or less than 90°
(d) Bevel gears in which the angle between the axes is 90° and the speed ratio of the gears is 1: 1

IES-17. Match List-I (Gears) with List-II (Configurations) and select the correct answer using the codes given below the Lists: **[IES-2003]**

List-I (Gears)	List-II (Configurations)
A Spur	1. Connecting two non-parallel or intersecting but coplanar shafts
B. Bevel	2. Connecting two parallel and coplanar shafts with teeth parallel to the axis of the gear wheel
C. Helical	3. Connecting two parallel and coplanar shafts with teeth inclined to the axis of the gear wheel
D. Mitre	4. Connecting two shafts whose axes are mutually perpendicular to each other

Codes:	A	B	C	D		A	B	C	D
(a)	2	4	3	1	(b)	3	1	2	4
(c)	2	1	3	4	(d)	3	4	2	1

Pitch point

IES-18. Gearing contact is which one of the following? **[IES 2007]**
(a) Sliding contact
(b) Sliding contact, only rolling at pitch point
(c) Rolling contact
(d) Rolling and sliding at each point of contact

IES-19. When two spur gears having involute profiles on, their teeth engage, the line of action is tangential to the **[IES-2003]**
(a) Pitch circles
(b) Dedendum circles

(c) Addendum circles (d) Base circles

Pressure angle

IES-20. What is the value of pressure angle generally used for involute gears? **[IES-2006]**

(a) 35° (b) 30° (c) 25 ° (d) 20°

IES-21. Consider the following, modifications regarding avoiding the interference between gears: **[IES-2003]**

1. The centre distance between meshing gears be increased
2. Addendum of the gear be modified
3. Teeth should be undercut slightly at the root
4. Pressure angle should be increased
5. Circular pitch be increased

Which of these are effective in avoiding interference?

(a) 1, 2 and 3 (b) 2, 3, 4 and 5 (c) 1, 4 and 5 (d) 3, 4 and 5

IES-22. An external gear with 60 teeth meshes with a pinion of 20 teeth, module being 6 mm. What is the centre distance in mm? **[IES-2009]**

(a) 120 (b) 180 (c) 240 (d) 300

IES-23. Assertion (A): An involute rack with 20° pressure angle meshes with a pinion of 14.5° pressure angle. **[IES-2002]**

Reason (R): Such a matching is impossible.

(a) Both A and R are individually true and R is the correct explanation of A
(b) Both A and R are individually true but R is **not** the correct explanation of A
(c) A is true but R is false
(d) A is false but R is true

IES-24. Compared to gears with 20^0 pressure angle involute full depth teeth, those with 20^0 pressure angle and stub teeth have **[IES 2007]**

1. Smaller addendum. **2. Smaller dedendum.**
3. Smaller tooth thickness. **4. Greater bending strength.**

Which of the statements given above are correct?

(a) 1, 2 and 3 (b) 1, 2 and 4
(c) 1, 3 and 4 (d) 2, 3 and 4

IES-25. Consider the following statements: **[IES-1999]**

A pinion of $14\frac{1}{2}^o$ **pressure angle and 48 involute teeth has a pitch circle diameter of 28.8 cm. It has**

1. Module of 6 mm **2. Circular pitch of 18 mm**
3. Addendum of 6 mm **4. Diametral pitch of** $\frac{11}{113}$

Which of these statements are correct?

(a) 2 and 3 (b) 1 and 3 (c) 1 and 4 (d) 2 and 4

IES-26. Which of the following statements are correct? **[IES-1996]**

1. For constant velocity ratio transmission between two gears, the common normal at the point of contact must always pass through a fixed point on the line joining the centres of rotation of the gears.
2. For involute gears the pressure angle changes with change in centre distance between gears.

3. The velocity ratio of compound gear train depends upon the number of teeth of the input and output gears only.
4. Epicyclic gear trains involve rotation of at least one gear axis about some other gear axis.
(a) 1, 2 and 3 (b) 1, 3 and 4 (c) 1, 2 and 4 (d) 2, 3 and 4

IES-27. Which one of the following is true for involute gears? [IES-1995]
(a) Interference is inherently absent
(b) Variation in centre distance of shafts increases radial force
(c) A convex flank is always in contact with concave flank
(d) Pressure angle is constant throughout the teeth engagement.

IES-28. In involute gears the pressure angle is [IES-1993]
(a) Dependent on the size of teeth (b) dependent on the size of gears
(c) Always constant (d) always variable

Minimum Number of Teeth

IES-29. Which one of the following statements is correct? [IES-2004]
Certain minimum number of teeth on the involute pinion is necessary in order to
(a) Provide an economical design (b) avoid Interference
(c) Reduce noise in operation (d) overcome fatigue failure of the teeth

IES-30. A certain minimum number of teeth is to be kept for a gear wheel
(a) So that the gear is of a good size [IES-1999]
(b) For better durability
(c) To avoid interference and undercutting
(d) For better strength

IES-31. In full depth $14\frac{1}{2}^{o}$ degree involute system, the smallest number of teeth in a pinion which meshes with rack with out interference is [IES-1992]
(a) 12 (b) 16 (c) 25 (d) 32

IES-33. Match List I with List II and select the correct answer using the codes given below the lists:
List I (Terminology) List II (Relevant terms) [IES-1995]

List I (Terminology)	List II (Relevant terms)
A. Interference	1. Arc of approach, arc of recess, circular pitch
B. Dynamic load on tooth	2. Lewis equation
C. Static load	3. Minimum number of teeth on pinion
D. Contract ratio	4. Inaccuracies in tooth profile

Codes:	A	B	C	D		A	B	C	D
(a)	3	4	1	2	(b)	1	2	3	4
(c)	4	3	2	1	(d)	3	4	2	1

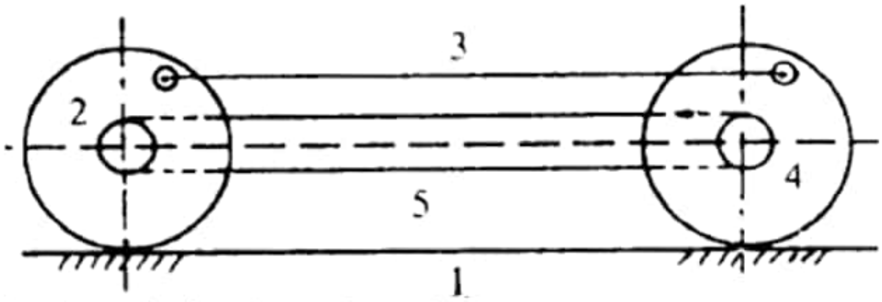
IES-34 **Assertion (A):** When a pair of spur gears of the same material is in mesh, the design is based on pinion. [IES-2002; 1993]
Reason (R): For a pair of gears of the same material in mesh, the 'strength factor' of the pinion is less than that of the gear.
(a) Both A and R are individually true and R is the correct explanation of A
(b) Both A and R are individually true but R is **not** the correct explanation of A
(c) A is true but R is false
(d) A is false but R is true

Cycloidal teeth

IES-35. **The curve traced by a point on the circumference of a circle which rolls along the inside of affixed circle, is known as** **[IES-1992]**

(a) Epicycloid (b) hypocycloid
(c) Cardiod (d) involute

IES-36



In the mechanism shown above, link 3 has **[IES-2004]**

(a) Curvilinear translation and all points in it trace out identical cycloids
(b) Curvilinear translation and all points in it trace out identical involutes
(c) Linear translation & all points in it trace out identical helices
(d) Linear translation & all points in it trace out identical ellipses

IES-37. **A thin circular disc is rolling with a uniform linear speed, along a straight path on a plane surface.** **[IES-1994]**

Consider the following statements in this regard:
1. All points on the disc have the same velocity.
2. The centre of the disc has zero acceleration.
3. The centre of the disc has centrifugal acceleration.
4. The point on the disc making contact with the plane surface has zero acceleration of these statements

(a) 1 and 4 are correct (b) 3 and 4 are correct
(c) 3 alone is correct (d) 2 alone is correct.

Involute teeth

IES-38. **In the case of an involute toothed gear, involute starts from** **[IES-1997]**

(a) Addendum circle (b) dedendum circle
(c) Pitch circle (d) base circle

IES-39. **Consider the following statements:** **[IES-2006]**

1. A stub tooth has a working depth larger than that of a full-depth tooth.
2. The path of contact for involute gears is an arc of a circle.
Which of the statements given above is/are correct?

(a) Only 1 (b) Only 2 (c) Both 1 and 2 (d) Neither 1 nor 2

IES-40. **Consider the following statements regarding the choice of conjugate teeth for the profile of mating gears:** **[IES-1999]**

1. They will transmit the desired motion
2. They are difficult to manufacture.
3. Standardisation is not possible
4. The cost of production is low.
Which of these statements are correct?

(a) 1, 2 and 3 (b) 1, 2 and 4 (c) 2, 3 and 4 (d) 1, 3 and 4

IES-41. **Which one of the following is correct?** **[IES-2008]**

When two teeth profiles of gears are conjugate, the sliding velocity between them
(a) Is always zero, all through the path of contact?
(b) Is zero, at certain points along the path of contact?
(c) Is never zero anywhere on the path of contact?
(d) Can be made zero by proper selection of profiles

Contact ratio

IES-42. **Which one of the following is the correct statement?** **[IES 2007]**
In meshing gears with involute gears teeth, the contact begins at the intersection of the

(a) Line of action and the addendum circle of the driven gear
(b) Line of action and the pitch circle of the driven gear
(c) Dedendum circle of the driver gear and the addendum circle of the driven gear
(d) Addendum circle of the driver gear and the pitch circle of the driven gear

IES-43. **Common contact ratio of a pair of spur pinion and gear is** **[IES-2008]**
(a) Less than 1·0 (b) equal to 1
(c) Between 2 and 3 (d) greater than 3

Interference

IES-44. **Interference between an involute gear and a pinion can be reduced by which of the following?**
1. Increasing the pressure angle of the teeth in the pair, the number of teeth remaining the same.
2. Decreasing the addendum of the gear teeth and increasing the same for the pinion teeth by the corresponding amount. **[IES-2008]**
Select the correct answer using the code given below:
(a) 1 only (b) 2 only (c) Both 1 and 2 (d) Neither 1 nor 2

IES-45. **In gears, interference takes place when** **[IES-1993]**
(a) The tip of a tooth of a mating gear digs into the portion between base and root circles
(b) Gears do not move smoothly in the absence of lubrication
(c) Pitch of the gear is not same
(d) gear teeth are undercut

IES-46. **An involute pinion and gear are in mesh. If both have the same size of addendum, then there will be an interference between the** **[IES-1996]**
(a) Tip of the gear tooth and flank of pinion.
(b) Tip of the pinion and flank of gear.
(c) Flanks of both gear and pinion.
(d) Tips of both gear and pinion.

IES-47. **Interference between the teeth of two meshing involute gears can be reduced or eliminated by** **[IES 2007]**
1. Increasing the addendum of the gear teeth and correspondingly reducing the addendum of the pinion.
2. Reducing the pressure angle of the teeth of the meshing gears.
3. Increasing the centre distance
Which of the statements given above is/are correct?
(a) 1 and 2 (b) 2 and 3
(c) 1 only (d) 3 only

IES-48. **Consider the following statements:** **[IES-2002]**

A 20° stub tooth system is generally preferred in spur gears as it results in
1. Stronger teeth
2. Lesser number of teeth on the pinion
3. Lesser changes of surface fatigue failure
4. Reduction of interference
Which of the above statements are correct?
(a) 1, 2 and 4 (b) 3 and 4 (c) 1 and 3 (d) 1, 2, 3 and 4

IES-49. **Match List-I with List-II and select the correct answer using the codes given below the lists:** **[IES-2001]**

List-I	List-II
A. Undercutting	1. Beam strength
B. Addendum	2. Interference
C. Lewis equation	3. Large speed reduction
D. Worm and wheel	4. Intersecting axes
	5. Module

Codes:	A	B	C	D		A	B	C	D
(a)	2	5	1	3	(b)	1	5	4	3
(c)	1	3	4	5	(d)	2	3	1	5

IES-50. **Which one of the following pairs is correctly matched?** **[IES-1999]**
(a) Governors ... Interference
(b) Gears …….Hunting
(c) Klein's construction.... Acceleration of piston
(d) Cam …….Pinion

IES-51. **Consider the following characteristics:** **[IES-1998]**
1. Small interference **2. Strong tooth.**
3. Low production cost **4. Gear with small number of teeth.**
Those characteristics which are applicable to stub 20° involute system would include
(a) 1 alone (b) 2, 3 and 4 (c) 1, 2 and 3 (d) 1, 2, 3 and 4

IES-52. **The motion transmitted between the teeth of two spur gears in mesh is generally** **[IES-1999]**
(a) Sliding (b) rolling
(c) Rotary (d) party sliding and partly rolling

Beam Strength of Gear Tooth

IES-53. **In heavy-duty gear drives, proper heat treatment of gears is necessary in order to:** **[IES-2006]**
(a) Avoid interference
(b) Prevent noisy operation
(c) Minimize wear of gear teeth
(d) Provide resistance against impact loading on gear teeth

IES-54. **Consider the following statements pertaining to the basic Lewis equation for the strength design of spur gear teeth:** **[IES-2005]**
1. Single pair of teeth participates in power transmission at any instant.
2. The tooth is considered as a cantilever beam of uniform strength.
3. Loading on the teeth is static in nature.
4. Lewis equation takes into account the inaccuracies of the tooth profile.
5. Meshing teeth come in contract suddenly.

Which of the statements given above are correct?
(a) 1, 3, 4 and 5 (b) 1,2, 3 and 4 (c) 1, 2 and 3 (d) 2, 4 and 5

IES-55. **Assertion (A):** The Lewis equation for design of gear tooth predicts the static load capacity of a cantilever beam of uniform strength.
Reason (R): According to law of gears interchangeability is possible only when gears have same pressure angle and same module. **[IES-2008]**
(a) Both A and R are true and R is the correct explanation of A
(b) Both A and R are true but R is NOT the correct explanation of A
(c) A is true but R is false
(d) A is false but R is true

IES-56. **In the formulation of Lewis equation for toothed gearing, it is assumed that tangential tooth load F_t, acts on the** **[IES-1998]**
(a) Pitch point (b) tip of the tooth
(c) Root of the tooth (d) whole face of the tooth

IES-57. **Assertion (A):** The Lewis equation for gear tooth with involute profile predicts the static load capacity of cantilever beam of uniform strength. **[IES-1994]**
Reason (R): For a pair of meshing gears with involute tooth profile, the pressure angle and module must be the same to satisfy the condition of inter-changeability.
(a) Both A and R are individually true and R is the correct explanation of A
(b) Both A and R are individually true but R is **not** the correct explanation of A
(c) A is true but R is false
(d) A is false but R is true

IES-58. **The dynamic load on a gear is due to** **[IES-2002]**
1. Inaccuracies of tooth spacing
2. Irregularities in tooth profile
3. Deflection of the teeth under load
4. Type of service (i.e. intermittent, one shift per day, continuous per day).
Which of the above statements are correct?
(a) 1. 2 and 3 (b) 2, 3 and 4 (c) 1, 3 and 4 (d) 1, 2 and 4

IES-59. **Consider the following statements:**
The form factor of a spur gear tooth depends upon the **[IES-1996]**
1. Number of teeth **2. Pressure angle**
3. Addendum modification coefficient **4. Circular pitch**
Of these correct statements are
(a) 1 and 3 (b) 2 and 4 (c) 1, 2 and 3 (d) 1 and 4

IES-60. **Assertion (A):** If the helix angle of a helical gear is increased, the load carrying capacity of the tooth increases. **[IES-1996]**
Reason (R): The form factor of a helical gear increases with the increasing in the helix angle.
(a) Both A and R are individually true and R is the correct explanation of A
(b) Both A and R are individually true but R is **not** the correct explanation of A
(c) A is true but R is false
(d) A is false but R is true

IES-61. **Match List I with List II and select the correct answer using the codes given below the Lists:** **[IES-2000]**

List I	**List II**
A. Unwin's formula	**1. Bearings**
B. Wahl factor	**2. Rivets**
C. Reynolds's equation	**3. Gears**

D. Lewis form factor **4. Springs**

Code:	A	B	C	D		A	B	C	D
(a)	1	4	2	3	(b)	2	3	1	4
(c)	1	3	2	4	(d)	2	4	1	3

IES-62. **A spur gear transmits 10 kW at a pitch line velocity of 10 m/s; driving gear has a diameter of 1.0 m. Find the tangential force between the driver and the follower, and the transmitted torque respectively.** **[IES-2009]**

(a) 1 kN and 0.5 kN-m (b) 10 kN and 5 kN-m
(c) 0.5 kN and 0.25 kN-m (d) 1 kN and 1 kN-m

Wear Strength of Gear Tooth

IES-63. **The limiting wear load of spur gear is proportional to (where E_p = Young's modulus of pinion material; E_g = Young's modulus of gear material)** **[IES-1997]**

(a) $\left(E_p + E_g\right)^{-1}$ (b) $\left(\dfrac{E_p + E_g}{E_p E_g}\right)$ (c) $\left(1 + \dfrac{E_p}{E_g}\right)$ (d) $\left(1 + \dfrac{E_g}{E_p}\right)$

Gear Lubrication

IES-64. **Match List I (Types of gear failure) with List II (Reasons) and select the correct answer using the codes given below the Lists** **[IES-2004]**

List I	List II
A. Scoring	1. Oil film breakage
B. Pitting	2. Yielding of surface under heavy loads
C. Scuffing	3. Cyclic loads causing high surface stress
D. Plastic flow	4. Insufficient lubrication

	A	B	C	D		A	B	C	D
(a)	2	1	3	4	(b)	2	3	4	1
(c)	4	3	1	2	(d)	4	1	3	2

Simple Gear train

IES-65. **In a simple gear train, if the number of idler gears is odd, then the direction or motion of driven gear will** **[IES-2001]**

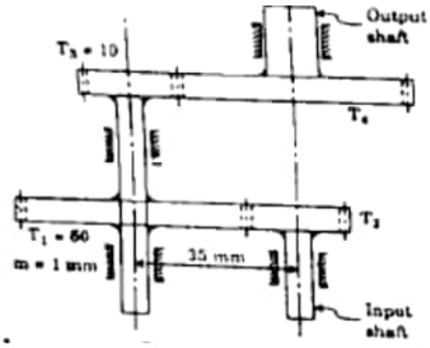
(a) Be same as that of the driving gear
(b) Be opposite to that of the driving gear
(c) Depend upon the number of teeth on the driving gear
(d) Depend upon the total number of teeth on all gears of the train

IES-66. **The gear train usually employed in clocks is a** **[IES-1995]**

(a) Reverted gear train (b) simple gear train
(c) Sun and planet gear (d) differential gear.

IES-67. In the figure shown above, if the speed of the input shaft of the spur gear train is 2400 rpm and the speed of the output shaft is 100 rpm, what is the module of the gear 4?

(a) 1.2 (b) 1.4
(c) 2 (d) 2.5



[IES-2005]

IES-68 In a machine tool gear box, the smallest and largest spindles are 100 rpm and 1120 rpm respectively. If there are 8 speeds in all, the fourth speed will be [IES-2002]

(a) 400 rpm (b) 280 rpm (c) 800 rpm (d) 535 rpm

IES-69. A fixed gear having 200 teeth is in mesh with another gear having 50 teeth. The two gears are connected by an arm. The number of turns made by the smaller gear for one revolution of arm about the centre of the bigger gear is [IES-1996]

(a) $\frac{2}{4}$ (b) 3 (c) 4 (d) 5

Compound gear train

IES-70 The velocity ratio in the case of the compound train of wheels is equal to [IES-2000]

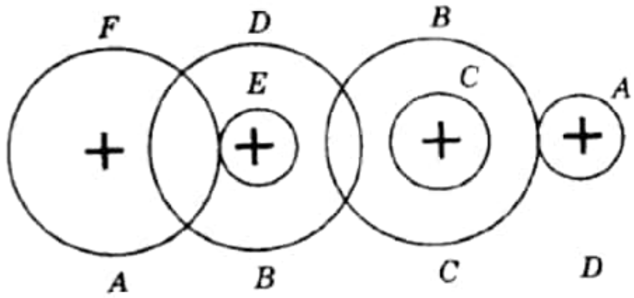
(a) $\frac{\text{No. of teeth on first driver}}{\text{No. of teeth on last follower}}$ (b) $\frac{\text{No. of teeth on last follower}}{\text{No. of teeth on first driver}}$

(c) $\frac{\text{Productof teeth on the drivers}}{\text{Productof teeth on the followers}}$ (d) $\frac{\text{Productof teeth on the followers}}{\text{Productof teeth on the drivers}}$

IES-71. Consider the gear train shown in the given figure and table of gears and their number of teeth.

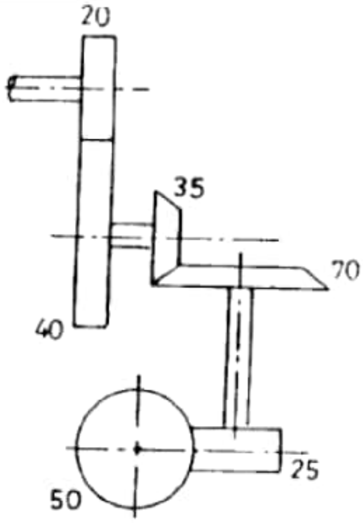
Gear:	A	B	C	D	E	F
No of teeth:	20	50	25	75	26	65

[IES-1999]



Gears BC and DE are moulded on parallel shaft rotating together. If the speed of A is 975 r.p.m., the speed of F will be

IES-72. **A compound train consisting of spur, bevel and spiral gears are shown in the given figure along with the teeth numbers marked against the wheels. Over-all speed ratio of the train is**



(a) 8

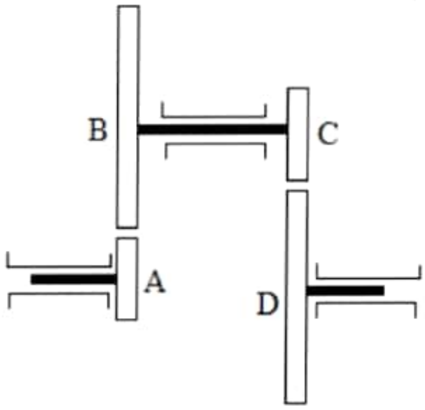
(b) 2

(c) $\frac{1}{2}$

(d) $\frac{1}{8}$

[IES-1996]

IES-73 **In the compound gear train shown in the above figure, gears A and C have equal numbers of teeth and gears B and D have equal numbers of teeth. When A rotates at 800 rpm, D rotates at 200 rpm. The rotational speed of compound gears BC would then be**



(a) 300 rpm
(b) 400rpm
(c) 500 rpm
(d) 600rpm

[IES 2007]

Reverted gear train

IES-74. **Consider the following statements in case of reverted gear train:** [IES-2002]
1. The direction of rotation of the first and the last gear is the same.
2. The direction of rotation of the first and the last gear is opposite.
3. The first and the last gears are on the same shaft.
4. The first and the last gears are on separate but co-axial shafts.

Which of these statements is/are correct?

(a) 1 and 3 (b) 2 and 3 (c) 2 and 4 (d) 1 and 4

IES-75. **A reverted gear train is one in which the output shaft and input shaft**
(a) Rotate in opposite directions (b) are co-axial [IES-1997]
(c) Are at right angles to each other (d) are at an angle to each other

IES-76. **In a reverted gear train, two gears P and Q are meshing, Q - R is a compound gear, and R and S are meshing. The modules of P and R are 4 mm and 5 mm respectively. The numbers of teeth in P, Q and R are 20, 40 and 25 respectively. The number of teeth in S is** [IES-2003]

(a) 23 (b) 35 (c) 50 (d) 53

IES-77. **Two shafts A and B, in the same straight line are geared together through an intermediate parallel shaft. The parameters relating to the gears and pinions are given in the table:** [IES-2003]

Item	Speed	Teeth	PCD	Module
Driving wheel A	N_A	T_A	D_A	m

Driven wheel B	N_B	T_B	D_B	m
Driven wheel C on the intermediate shaft	N_C	T_C	D_C	m
Driving wheel D on the intermediate shaft, in mesh with B	N_D	T_D	D_D	m

(a) $\frac{N_A}{N_B} = \frac{T_C}{T_A} \times \frac{T_B}{T_D}$ (b) $\frac{N_A}{N_B} = \frac{T_A}{T_C} \times \frac{T_D}{T_B}$

(c) $D_A + D_C = D_B + D_D$ (d) $T_A + T_C = T_B + T_D$

IES-78. **A gear having 100 teeth is fixed and another gear having 25 teeth revolves around it, centre lines of both the gears being jointed by an arm. How many revolutions will be made by the gear of 25 teeth for one revolution of arm?**
[IES-2009]

(a) 3 (b) 4 (c) 5 (d) 6

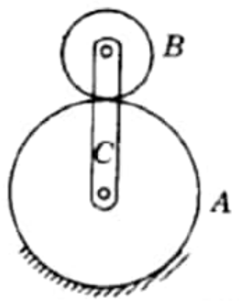
Epicyclic gear train

IES-79. **If the annular wheel of an epicyclic gear train has 100 teeth and the planet wheel has 20 teeth, the number of teeth on the sun wheel is** **[IES-2003]**

(a) 80 (b) 60 (c) 40 (d) 20

IES-80. **In the epicyclic gear train shown in the given figure, A is fixed. A has 100 teeth and B has 20 teeth. If the arm C makes three revolutions, the number of revolutions made by B will be**

(a) 12
(b) 15
(c) 18
(d) 24



[IES-1997]

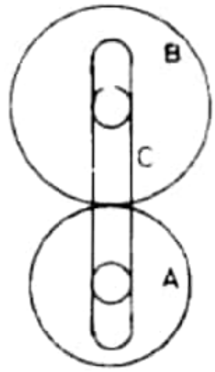
IES-81. **An epicyclic gear train has 3 shafts A, B and C, A is an input shaft running at 100 rpm clockwise. B is an output shaft running at 250 rpm clockwise. Torque on A is 50 kNm (clockwise). C is a fixed shaft. The torque to fix C**

(a) Is 20 kNm anticlockwise **[IES-2002]**
(b) is 30 kNm anticlockwise
(c) Is 30 kNm clockwise
(d) Cannot be determined as the data is insufficient

IES-82. **A single epicyclic gear train is shown in the given figure. Wheel A is stationary. If the number of teeth on A and Bare 120 and 45 respectively, then when B rotates about its own axis at 100 rpm, the speed of C would be**

(a) 20 rpm (b) $27\frac{3}{11}$ rpm

(c) $19\frac{7}{11}$ rpm (d) 100 rpm



[IES-1994]

Terminology of Helical Gears

IES-83. **If α = helix angle, and p_c = circular pitch; then which one of the following correctly expresses the axial pitch of a helical gear?** **[IES 2007]**

(a) pc cosα (b) $\frac{p_c}{\cos\alpha}$ (c) $\frac{p_c}{\tan\alpha}$ (d) $p_c \sin \alpha$

IES-84. **A helical gear has the active face width equal to b, pitch p and helix angle α. What should be the minimum value of b in order that contact IS maintained across the entire active face of the gear?** **[IES-2004]**

(a) $p\cos\alpha$ (b) $p\sec\alpha$ (c) $p\tan\alpha$ (d) $p\cot\alpha$

IES-85. **Assertion (A):** Helical gears are used for transmitting motion and power between intersecting shafts, whereas straight bevel gears arc used for transmitting motion and power between two shafts intersecting each other at 90°. **[IES-2000]**
Reason (R): In helical gears teeth are inclined to axis of the shaft and arc in the form or a helix. Where as in bevel gears, teeth arc tapered both in thickness and height form one end to the other.
(a) Both A and R are individually true and R is the correct explanation of A
(b) Both A and R are individually true but R is **not** the correct explanation of A
(c) A is true but R is false
(d) A is false but R is true

IES-86. **Assertion (A):** Shafts supporting helical gears must have only deep groove ball-bearings.
Reason (R): Helical gears produce axial thrusts. **[IES-1999]**
(a) Both A and R are individually true and R is the correct explanation of A
(b) Both A and R are individually true but R is **not** the correct explanation of A
(c) A is true but R is false
(d) A is false but R is true

IES-87. **Assertion (A):** Crossed helical gears for skew shafts are not used to transmit heavy loads.
Reason (R) The gears have a point contact, and hence are not considered strong.
(a) Both A and R are individually true and R is the correct explanation of A
(b) Both A and R are individually true but R is **not** the correct explanation of A
(c) A is true but R is false
(d) A is false but R is true **[IES-1995]**

Bevel Gears

IES-88. In a differential mechanism, two equal sized bevel wheels A and B are keyed to the two halves of the rear axle of a motor car. The car follows a curved path. Which one of the following statements is correct? [IES-2004]

The wheels A and B will revolve at different speeds and the casing will revolve at a speed which is equal to the

(a) Difference of speeds of A and B
(b) Arithmetic mean of speeds of A and B
(c) Geometric mean of speeds of A and B
(d) Harmonic mean of speeds of A and B

Worm Gears

IES-89. **Assertion (A):** Tapered roller bearings must be used in heavy duty worm gear speed reducers.
Reason (R): Tapered roller bearings are suitable for large radial as well as axial loads.

(a) Both A and R are individually true and R is the correct explanation of A
(b) Both A and R are individually true but R is **not** the correct explanation of A
(c) A is true but R is false
(d) A is false but R is true **[IES-2005]**

IES-90. Consider the following statements in respect of worm gears: [IES-2005]

1. They are used for very high speed reductions.
2. The velocity ratio does not depend on the helix angle of the worm.
3. The axes of worm and gear are generally perpendicular and non-intersecting.

Which of the statements given above are correct?

(a) 1 and 2 (b) 1 and 3 (c) 2 and 3 (d) 1, 2 and 3

IES-91. For a speed ratio of 100 smallest gear box is obtained by using which of the following?

(a) A pair of spur gears **[IES-2008]**
(b) A pair of bevel and a pair of spur gears in compound gear train
(c) A pair of helical and a pair of spur gears in compound gear train
(d) A pair of helical and a pair of worm gears in compound gear train

IES-92. Consider the following statements regarding improvement of efficiency of worm gear drive: [IES-2004]

1. Efficiency can be improved by increasing the spiral angle of worm thread to 45° or more
2. Efficiency can be improved by adopting proper lubrication
3. Efficiency can be improved by adopting worm diameter as small as practicable to reduce sliding between worm-threads and wheel teeth
4. Efficiency can be improved by adopting convex tooth profile both for worm and wheel

Which of the statements given above are correct?

(a) 1, 2 and 3 (b) 1, 2 and 4 (c) 2, 3 and 4 (d) 1, 3 and 4

IES-93. The lead angle of a worm is 22.5 deg. Its helix angle will be [IES-1994]

(a) 22. 5 deg. (b) 45 deg. (c) 67.5 deg. (d) 90°C.

Previous 20-Years IAS Questions

Spur gear

IAS-1. Match List I (Terms) with List II (Definition) and select the correct answer using the codes given below the lists: [IAS-2001]

List I List II

A. Module	1. Radial distance of a tooth from the pitch circle to the top of the tooth
B. Addendum	2. Radial distance of a tooth from the pitch circle to the bottom of the tooth
C. Circular pitch	3. Distance on the circumference of the pitch circle from a point of one tooth to the corresponding point on the next tooth
	4. Ratio of a pitch circle diameter in mm to the number of teeth

Codes:	**A**	**B**	**C**		**A**	**B**	**C**
(a)	4	1	3	(b)	4	2	3
(c)	3	1	2	(d)	3	2	4

IAS-2 Consider the following specifications of gears A, B, C and D: [IAS-2001]

Gears	A	B	C	D
Number of teeth	20	60	20	60
Pressure angle	$14\frac{1}{2}^{o}$	$14\frac{1}{2}^{o}$	20°	$14\frac{1}{2}^{o}$
Module	1	3	3	1
Material	Steel	Brass	Brass	Steel

Which of these gears form the pair of spur gears to achieve a gear ratio of 3?

(a) A and B (b) A and D (c) B and C (d) C and D

IAS-3. If the number of teeth on the wheel rotating at 300 r.p.m. is 90, then the number of teeth on the mating pinion rotating at 1500 r.p. m. is [IAS-2000]

(a) 15 (b) 18 (c) 20 (d) 60

IAS-4. A rack is a gear of [IAS-1998]

(a) Infinite diameter (b) infinite module

(c) zero pressure angle (d) large pitch

Classification of Gears

IAS-5. Assertion (A): While transmitting power between two parallel shafts, the noise generated by a pair of helical gears is less than that of an equivalent pair of spur gears. **[IAS-2000]**

Reason(R): A pair of helical gears has fewer teeth in contact as compared to an equivalent pair of spur gears.

(a) Both A and R are individually true and R is the correct explanation of A

(b) Both A and R are individually true but R is **not** the correct explanation of A

(c) A is true but R is false

(d) A is false but R is true

Pitch point

IAS-6. An imaginary circle which by pure rolling action, gives the same motion as the actual gear, and is called [IAS-2000]

(a) Addendum circle (b) pitch circle

(c) Dedendum circle (d) base circle

Pressure angle

IAS-7. The pressure angle of a spur gear normally varies from [IAS-2000]

(a) 14° to 20° (b) 20° to 25° (c) 30° to 36° (d) 40° to 50°

Minimum Number of Teeth

IAS-8. Minimum number of teeth for involute rack and pinion arrangement for pressure angle of 20° is **[IAS-2001]**

(a) 18 (b) 20 (c) 30 (d) 34

Cycloidal teeth

IAS-9. The tooth profile most commonly used in gear drives for power transmission is

(a) A cycloid (b) An involute (c) An ellipse (d) A parabola **[IAS-1996]**

Contact ratio

IAS-10. Which one of the following statements is correct? **[IAS-2007]**

(a) Increasing the addendum results in a larger value of contact ratio
(b) Decreasing the addendum results in a larger value of contact ratio
(c) Addendum has no effect on contact ratio
(d) Both addendum and base circle diameter have effect on contact ratio

IAS-11. The velocity of sliding of meshing gear teeth is **[IAS-2002]**

(a) $(\omega_1 \times \omega_2)x$ (b) $\frac{\omega_1}{\omega_2}x$ (c) $(\omega_1 + \omega_2)x$ (d) $\frac{(\omega_1 + \omega_2)}{x}$

(Where ω_1 and ω_2 = angular velocities of meshing gears
x = distance between point of contact and the pitch point)

Interference

IAS-12. For spur with gear ratio greater than one, the interference is most likely to occur near the **[IAS-1997]**

(a) Pitch point (b) point of beginning of contact
(c) Point of end of contact (d) root of the tooth

IAS-13. How can interference in involute gears be avoided? **[IAS-2007]**

(a) Varying the centre distance by changing the pressure angle only
(b) Using modified involute or composite system only
(c) Increasing the addendum of small wheel and reducing it for the larger wheel only
(d) Any of the above

IAS-14. Which one of the following statements in respect of involute profiles for gear teeth is not correct? **[IAS-2003]**

(a) Interference occurs in involute profiles,
(b) Involute tooth form is sensitive to change in centre distance between the base circles.
(c) Basic rack for involute profile has straight line form
(d) Pitch circle diameters of two mating involute gears are directly proportional to the base circle diameter

IAS-15. **Assertion (A):** In the case of spur gears, the mating teeth execute pure rolling motion with respect to each other from the commencement of engagement to its termination. **[IAS-2003]**
Reason (R): The involute profiles of the mating teeth are conjugate profiles which obey the law of gearing.

(a) Both A and R are individually true and R is the correct explanation of A
(b) Both A and R are individually true but R is **not** the correct explanation of A
(c) A is true but R is false
(d) A is false but R is true

IAS-16. **Assertion (A):** Gears with involute tooth profile transmit constant velocity ratios between shafts connected by them. **[IAS-1997]**

Reason (R): For involute gears, the common normal at the point of contact between pairs of teeth always passes through the pictch point.

(a) Both A and R are individually true and R is the correct explanation of A
(b) Both A and R are individually true but R is **not** the correct explanation of A
(c) A is true but R is false
(d) A is false but R is true

Compound gear train

IAS-17. **There are six gears A, B, C, D, E, F in a compound train. The numbers of teeth in the gears are 20, 60, 30, 80, 25 and 75 respectively. The ratio of the angular speeds of the driven (F) to the driver (A) of the drive is**

(a) $\frac{1}{24}$ (b) $\frac{1}{8}$ (c) $\frac{4}{15}$ (d) 12 **[IAS-1995]**

Epicyclic gear train

IAS-18. **A fixed gear having 100 teeth meshes with another gear having 25 teeth, the centre lines of both the gears being joined by an arm so as to form an epicyclic gear train. The number of rotations made by the smaller gear for one rotation of the arm is** **[IAS-1995]**

(a) 3 (b) 4 (b) 5 (d) 6

IAS-19. **For an epicyclic gear train, the input torque = 100 Nm. RPM of the input gear is 1000 (clockwise), while that of the output gear is 50 RPM (anticlockwise). What is the magnitude of the holding torque for the gear train?** **[IAS-2007]**

(a) Zero (b) 500 Nm (c) 2100 Nm (d) None of the above

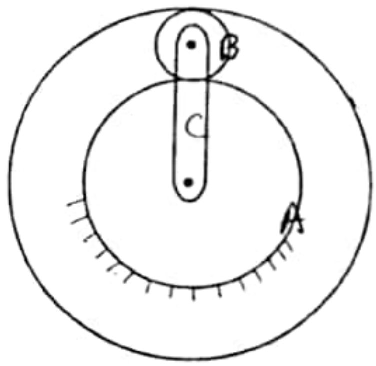
IAS-20. **In the figure shown, the sun wheel has 48 teeth and the planet has 24 teeth. If the sun wheel is fixed, what is the angular velocity ratio between the internal wheel and arm?**

(a) 3.0

(b) 1.5

(c) 2.0

(d) 4.0



[IAS-2004]

IAS-21. **100 kW power is supplied to the machine through a gear box which uses an epicyclic gear train. The power is supplied at 100 rad/s. The speed of the output shaft of the gear box is 10 rad/s in a sense opposite to the input speed. What is the holding torque on the fixed gear of the train?** **[IAS-2004]**

(a) 8 kNm (b) 9 kNm (c) 10 kNm (d) 11 kNm

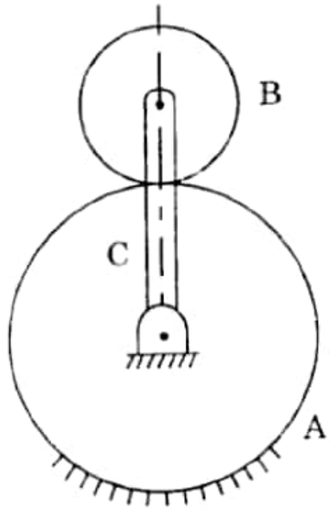
IAS-22. **In the epicyclic gear train shown in the figure, $T_A = 40$, $T_B = 20$. For three revolutions of the arm, the gear B will rotate through**

(a) 6 revolutions

(b) 2.5 revolutions

(c) 3 revolutions

(d) 9 revolutions



[IAS-2003]

Bevel Gears

IAS-23. **Assertion (A):** Spiral bevel gears designed to be used with an offset in their shafts are called 'hypoid gears' **[IAS-2004]**
Reason (R): The pitch surfaces of such gears are hyperboloids of revolution.
(a) Both A and R are individually true and R is the correct explanation of A
(b) Both A and R are individually true but R is **not** the correct explanation of A
(c) A is true but R is false
(d) A is false but R is true

Worm Gears

IAS-24. **If reduction ratio of about 50 is required in a gear drive, then the most appropriate gearing would be** **[IAS-1999]**
(a) spur gears (b) bevel gears
(c) Double helical gears (d) worm and worm wheel

IAS-25. **Speed reduction in a gear box is achieved using a worm and worm wheel. The worm wheel has 30 teeth and a pitch diameter of 210 mm. If the pressure angle of the worm is 20°, what is the axial pitch of the worm?**
(a) 7 mm (b) 22 mm **[IAS-2004]**
(c) 14 mm (d) 63 mm

IAS-26. **A speed reducer unit consists of a double-threaded worm of pitch = 11 mm and a worm wheel of pitch diameter = 84 mm. The ratio of the output torque to the input to ratio is**
(a) 7 6 (b) 12 (c) 24 (d) 42 **[IAS-2002]**

IAS-27. **The maximum efficiency for spiral gears in mesh is given by (Where (θ = shaft angle and ϕ, = friction angle)** **[IAS-1998]**

(a) $\dfrac{1+\cos(\theta-\phi)}{1+\cos(\theta+\phi)}$ (b) $\dfrac{1+\cos(\theta+\phi)}{1+\cos(\theta-\phi)}$

(c) $\dfrac{1-\cos(\theta-\phi)}{1+\cos(\theta+\phi)}$ (d) $\dfrac{1-\cos(\theta+\phi)}{1+\cos(\theta-\phi)}$

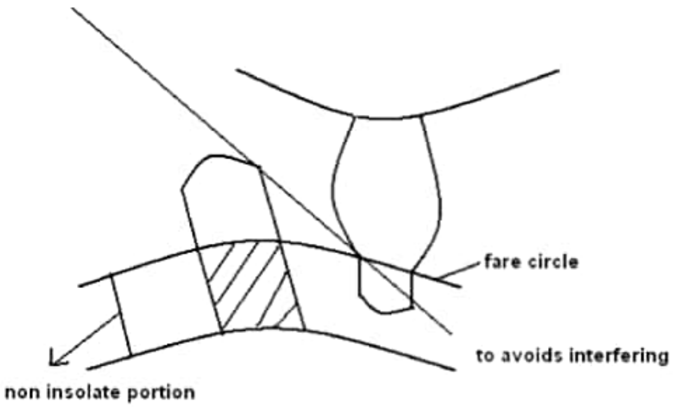
IAS-28. **Assertion (A):** A pair of gears forms a rolling pair. **[IAS-1996]**
Reason (R): The gear drive is a positive drive.

(a) Both A and R are individually true and R is the correct explanation of A
(b) Both A and R are individually true but R is **not** the correct explanation of A
(c) A is true but R is false
(d) A is false but R is true

Answers with Explanation (Objective)

Previous 20-Years GATE Answers

GATE-1. Ans. (a)
GATE-2. Ans. (a)
GATE-3. Ans. (b)
GATE-4. Ans. (a)
GATE-5. Ans. (c)
GATE-6. Ans. (d)



There are several ways to avoid interfering:
i. Increase number of gear teeth
ii. Modified involutes
iii. Modified addendum
iv. Increased centre distance

GATE-7. Ans. (a)

GATE-8. Ans. (a) $\text{Centre distance} = \dfrac{D_1 + D_2}{2} = \dfrac{mT_1 + mT_2}{2} = \dfrac{5(19+37)}{2} = 140\text{mm}$

GATE-9. Ans. (c)
GATE-10. Ans. (a)
GATE-11. Ans. (b)
GATE-12. Ans. (b)
GATE-13. Ans. (c)
GATE-14. Ans. (c)
GATE-15. Ans. (b)

Given : Module m = 2, $\frac{D}{T} = 2$

$\therefore \qquad D = 80 \times 2 = 160$ mm

$2F = 1000$, or $F = 500$ N

Let T_1 be the torque applied by motor.

T_2 be the torque applied by gear.

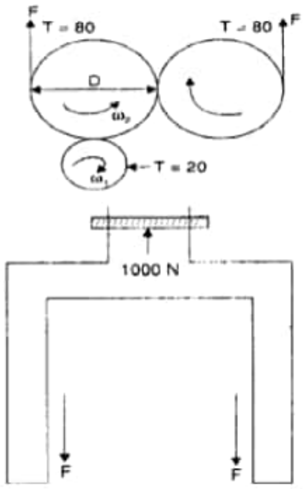
$\therefore$ Power transmission = 80%

Now, $T_1\omega_1 = \frac{2T_2 \times \omega_1}{0.8}$

or $\quad T_1 = \frac{2 \times F \times (D/2)}{0.8} \times \frac{\omega_1}{\omega_2}$

$= 2 \times 500 \times \frac{0.16}{2} \times \frac{1}{0.8} \times \frac{1}{4}$

$= 25$ N – m.



GATE-16. Ans. (c)

$$P\cos\phi = F$$

$\therefore$ Force acting along the line of action,

$$P = \frac{F}{\cos\phi}$$

$$= \frac{500}{\cos 20^\circ}$$

$$= 532\text{N}$$

GATE-17. Ans. (a)

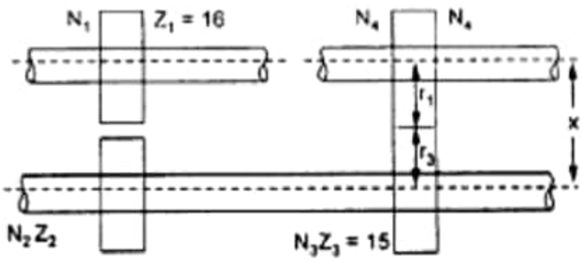
Given, $\frac{N_1}{N_2} = 12, \frac{N_1}{N_2} = 4 = \frac{D_2}{D_1}$

$m_1 = 3,\ m_2 = 4$

Now, $\qquad \frac{D_1}{Z_1} = \frac{D_2}{Z_2}$

$\Rightarrow \qquad \frac{Z_1}{Z_2} = \frac{D_1}{D_2} = \frac{N_2}{N_1} = \frac{1}{4}$

$\Rightarrow \qquad Z_2 = Z_1 \times 4 = 64$



$\Rightarrow \qquad 12 = \frac{D_4}{D_3}$

$\Rightarrow \qquad \frac{D_4}{D_3} = 3$

Also, $\qquad \frac{Z_3}{Z_4} = \frac{D_3}{D_4}$

$\Rightarrow \qquad Z_4 = Z_3\frac{D_4}{D_3} = Z_3 \times 3 = 15 \times 3$

$= 45$

GATE-18. Ans. (b)

Now, $x = r_4 + r_3 = \frac{D_4 + D_3}{2}$

But $\frac{D_4}{Z_4} = \frac{D_3}{Z_3} = 4$

$\Rightarrow$ $D_4 = 180,\ D_3 = 60$

$\therefore$ $x = \frac{180 + 60}{2} = 120mm$

GATE-19. Ans. (c)

	Arm	2	3	4	5
1.	0	$+x$	$\frac{-N_2}{N_3}x$	$\frac{-N_2}{N_3}x$	$\frac{-N_4}{N_5} \times \frac{N_2}{N_3}x$
2.	y	y	y	y	y
	y	x + y	$y - \frac{N_2}{N_3}x$		$y - \frac{N_4}{N_5} \times \frac{N_2}{N_3}x$

$x + y = 100$ (cw)

$y = -80$ (ccw)

$\text{Speed of Gear}\,(W_5) = -80 - \frac{32}{80} \times \frac{20}{24} \times 180 = -140 \quad = 140$ (ccw)

GATE-20. Ans. (a)

GATE-21. Ans. (b)

Explanation

	Arm A	B	C
Fix arm A Give one rotation to B Multiply by x Add y	0 0 y	1 +x X+ y	-1 -x y-x

B is fixed, therefore $x + y = 0$

y = rad/sec(ccw)

$\Rightarrow$ x = -4 rad/sec(cw)

Angular velocity of gear C $= y - x = 4 - (-4) = 8$ rad/s

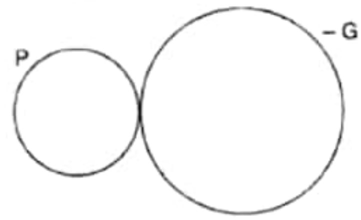
GATE-22. Ans. (b)

Arm	Sun	Planet	Ring
+1	+1	+1	+1
0	$\frac{80}{30} \times \frac{30}{20}$	$-\frac{80}{30}$	-1
1	5	$-\frac{5}{3}$	0

For 5 Revolutions Of Sun, Arm rotates by 1

$\therefore$ for 100 revolutions of Sun, Arm rotates by $\frac{100}{5} = 20$

GATE-23. Ans. (d)



We know $\frac{N_P}{N_G} = \frac{T_G}{T_P}$

where, N_P = speed of pinion, N_G = speed of gear wheel

T_G = number of teeth of gear,

T_P = number of teeth of pinion

$\therefore \quad \frac{1200}{N_G} = \frac{120}{40}$

or $\quad N_G = 400$ r.p.m

Since power transmitted by both gear will be equal

i.e. $\quad T_P\omega_P = T_G\omega_G$

where, T_P = torque transmitted by pinion, T_G = torque transmitted by gear wheel

$$\therefore \quad \frac{20 \times 2\pi \times 1200}{60} = \frac{T_G \times 2\pi \times 400}{60}$$

$\therefore$ torque transmitted by gear, $T_G = 60$N.m.

GATE-24. Ans. (a)

$$\frac{\omega_1 - \omega_5}{\omega_2 - \omega_5} = 3 \qquad \text{(with respect to arm 5 or carrier5)}$$

$$\frac{\omega_3 - \omega_5}{\omega_4 - \omega_5} = 2 \qquad \text{(with respect to carrier5)}$$

As, $\omega_3 - \omega_2$

$$\therefore \quad \frac{\omega_1 - \omega_5}{\omega_4 - \omega_5} = 6$$

GATE-25. Ans. (d)

$\omega_1 = 60$ rpm (Clockwise)

$\omega_4 = 120$ rpm (Counter clock wise)

$$\frac{60 - \omega_5}{-120 - \omega_5} = 6$$

$\therefore \omega_5 = -156$ i.e. counter clockwise

GATE-26. Ans. (b)
GATE-27. Ans. (d) speed reduction = 1440/36 = 40
GATE-28. Ans. (c)

Previous 20-Years IES Answers

IES-1. Ans. (b) Centre distance = $\frac{D_1 + D_2}{2} = \frac{mT_1 + mT_2}{2} = \frac{m}{2}(T_1 + T_2) = \frac{2}{2} \times 99 = 99$mm

IES-41. Ans. (a)
IES-42. Ans. (a)
IES-43. Ans. (c) The ratio of the length of arc of contact to the circular pitch is known as **contact ratio** i.e. number of pairs of teeth in contact. The contact ratio for gears is greater than one. Contact ratio should be at least 1.25. For maximum smoothness and quietness, the contact ratio should be between 1.50 and 2.00. High-speed applications should be designed with a face-contact ratio of 2.00 or higher for best results.
IES-44. Ans. (c)
IES-45. Ans. (a) In gears, interference takes place when the tip of a tooth of a mating gear digs into the portion between base .and root circle.
IES-46. Ans. (a)
IES-47. Ans. (d)
IES-48. Ans. (a)
IES-49. Ans. (a)
IES-50. Ans. (c)
IES-51. Ans. (b) Involute system is very interference prone.
IES-52. Ans. (b)
IES-53. Ans. (c)
IES-54. Ans. (c)
IES-55. Ans. (b) The beam strength of gear teeth is determined from an equation (known as Lewis equation) and the load carrying ability of the toothed gears as determined by this equation gives satisfactory results. In the investigation, Lewis assumed that as the load is being transmitted from one gear to another, it is all given and taken by one tooth, because it is not always safe to assume that the load is distributed among several teeth.

Notes: (*i*) The Lewis equation is applied only to the weaker of the two wheels (*i.e.* pinion or gear).
(*ii*) When both the pinion and the gear are made of the same material, then pinion is the weaker.
(*iii*) When the pinion and the gear are made of different materials, then the product of $(\sigma_w \times y)$ or $(\sigma_o \times y)$ is the deciding factor. The Lewis equation is used to that wheel for which $(\sigma_w \times y)$ or $(\sigma_o \times y)$ is less.

IES-56. Ans. (b)
IES-57. Ans. (c) For a pair of meshing gears with involute tooth profile, the pressure angle and module must be the same to satisfy the condition of inter-changeability it is not correct. Due to law of gearing.
IES-58. Ans. (a)
IES-59. Ans. (c)
IES-60. Ans. (a)
IES-61. Ans. (d)
IES-62. Ans. (a) Power transmitted = Force × Velocity

$$\Rightarrow \quad 10 \times 10^3 = \text{Force} \times 10$$

$$\Rightarrow \quad \text{Force} = \frac{10 \times 10^3}{10} = 1000 \text{ N / m}$$

$$\text{Torque Transmitted} = \text{Force} \times \frac{\text{diameter}}{2}$$

$$= 1000 \times \frac{1}{2} = 1000 \times 0.5$$

$$= 500 \text{ N} - \text{m} = 0.5 \text{ kN} - \text{m}$$

IES-63. Ans. (b)
IES-64. Ans. (b)
IES-65. Ans. (a)
IES-66. Ans. (a)
IES-67. Ans. (b)

$$\frac{mT_2 + mT_1}{2} = 35$$

or $T_2 = 10$

$$N_1 = -N_1 \times \frac{T_2}{T_1} = N_3$$

$$N_4 = \frac{-N_3 T_3}{T_4} = +N_1 \times \frac{T_2}{T_1} \times \frac{T_3}{T_4} \text{ or } 100 = 2400 \times \frac{10}{60} \times \frac{10}{T_4} \text{ or } T_4 = 40$$

$$\frac{m'T_3 + m'T_4}{2} = 35 \text{ or } m' = \frac{70}{(40+10)} = 1.4$$

IES-68. Ans. (b)

IES-69. Ans. (d) 1 + 200/50 = 1 + 4 = 5

IES-70. Ans. (d)

IES-71. Ans. (b) Speed ratio $\frac{N_F}{N_A} = \frac{T_A \times T_C \times T_E}{T_B \times T_D \times T_F} = \frac{20 \times 25 \times 26}{50 \times 75 \times 65} = \frac{4}{75}$ $\quad$ $or\ N_F = 975 \times \frac{4}{75} = 52\, rpm$

IES-72. Ans. (a)

Elements of higher pair like follower in cam is under the action of gravity or spring force .

$$\text{Train value} = \frac{\text{speed of lost driven or follower}}{\text{speed of the first gear}}$$

$$\text{Train value} = \frac{\text{product of no.of teeth no the drives}}{\text{product of no.of teeth on the drives}}$$

$$\frac{\text{speed of the first drive}}{\text{speed of the last driven or follower}}$$

IES-73. Ans. (b) From the figure $r_A + r_B = r_C + r_D$ or $T_A + T_B = T_C + T_D$ and as $N_B + N_C$ it must be $T_B = T_D$ & $T_A = T_C$

Or $\frac{N_B}{N_A} = \frac{N_D}{N_C}$ or $N_C = \sqrt{N_A N_D} = \sqrt{800 \times 200} = 400$ rpm $[\because N_B = N_c]$

IES-74. Ans. (d)

IES-75. Ans. (b)

IES-76. Ans. (a)

Summation of radius will be constant.

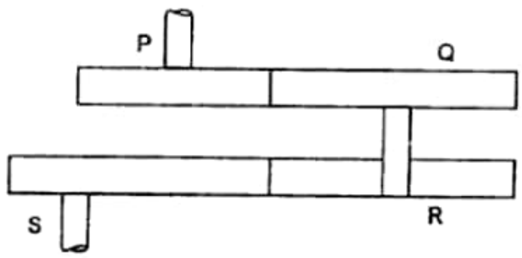
$R_P + R_Q = R_R + R_G$

or $D_P + D_Q = D_R + D_S$

or $m_1(T_P + T_Q) = m_2(T_R + T_S)$

or 4 (20 + 40) = 5 (25 + T_S)

or $T_S = 23$

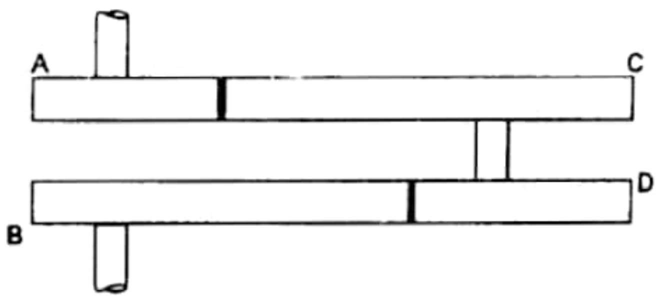


IES-77. Ans. (b)

(i) $D_A + D_C = D_B + D_D$

(ii) $mT_A + mT_C = mT_B + mT_D$

(iii) $\frac{N_A}{N_B} = \frac{N_A}{N_C} \times \frac{N_C}{N_B} = \frac{T_C}{T_A} \times \frac{T_B}{N_D}$



IES-78. Ans. (c)

Arm	N_A	N_B
0	+1	$\frac{-100}{25}$

Multiplying through out by x

0	+x	$\frac{-100}{25}x$
y	y + x	y − 4x

Given that $y + x = 0 \therefore x = -y = -1$

$(\because y = 1)$

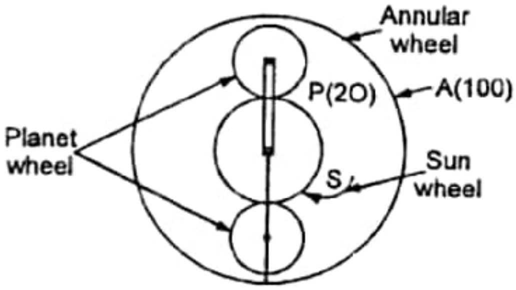
$\therefore \quad N_B = y - 4x = 5$

IES-79. Ans. (b) From geometry

$$2d_p + d_s = d_A$$

$$or\ 2T_p + T_s = T_A$$

$$or\ T_s = T_A - 2T_p = 100 - 2\times 20 = 60$$



IES-80. Ans. (c) For 1 revolution of C,

$$N_B = 1 + \frac{T_A}{T_B} = 1 + \frac{100}{20} = 6 \quad \therefore \text{for 3 revolution, } N_D = 6\times 3 = 18$$

IES-81. Ans. (b)

Now $\omega_1 M_1 - \omega_2 M_2 = 0$

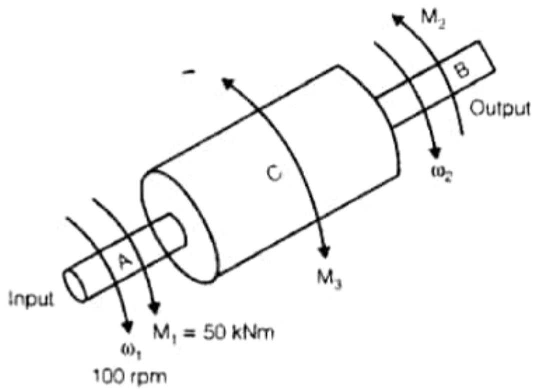
$\therefore \quad M_2 = \frac{100\times 50}{250} = 20$ KNm(anticlockwise)

and $\overrightarrow{M_1} + \overrightarrow{M_2} + \overrightarrow{M_3} = 0$

$50 - 20 + \overrightarrow{M_3} = 0$

$\therefore \quad \overrightarrow{M_3} = -30\text{kNm(clockwise)}$

$= 30\text{kNm(anticlockwise)}$



IES-82. Ans. (c)
IES-83. Ans. (c)
IES-84. Ans. (d) $b \geq \frac{P}{\tan\alpha}$
IES-85. Ans. (d)
IES-86. Ans. (a)
IES-87. Ans. (b)
IES-88. Ans. (d)
IES-89. Ans. (a)
IES-90. Ans. (d)
IES-91. Ans. (d)
IES-92. Ans. (a)

$$\text{Gear } \eta_{\text{wormgear}} = \frac{\tan\lambda}{\tan(\phi_v + \lambda)}$$

$$\tan\phi_v = \pi_v$$

$$\tan\lambda = \frac{z_w.m}{d_W}$$

The face of worm gear is made concave to envelope the worm.

IES-93. Ans. (c) α = Pressure angle $\cong$ lead angle; $\alpha + \beta = 90°$; β = helix angle = 90° - 22.5° = 67.5°

Previous 20-Years IAS Answers

IAS-1. Ans. (a)

IAS-2. Ans. (b)

For a gear pair i) module must be same

(ii) Pressure angle must be same.

IAS-3. Ans. (b)

Peripheral velocity (πDN) = constant. $\pi D_1 N_1 = \pi D_2 N_2$ and D=mT

$$or\ \pi m T_1 N_1 = \pi m T_1 N_1 \ \ or\ T_2 = T_1 \times \frac{N_1}{N_2} = 90 \times \frac{300}{1500} = 18$$

Or you may say speed ratio, $\frac{N_1}{N_2} = \frac{T_2}{T_1}$

IAS-4. Ans. (a)

IAS-5. Ans. (c) In spur gears, the contact between meshing teeth occurs along the entire face width of the tooth, resulting in a sudden application of the load which, in turn, results in impact conditions and generates noise.

In helical gears, the contact between meshing teeth begins with a point on the leading edge of the tooth and gradually extends along the diagonal line across the tooth. There is a gradual pick-up of load by the tooth, resulting in smooth engagement and silence operation.

IAS-6. Ans. (b)

IAS-7. Ans. (a)

IAS-8. Ans. (a) $T_{\min} = \frac{2h_f}{\sin^2\theta} = \frac{2\times 1}{\sin^2 20^o} = 17.1 \quad as > 17 \quad \text{So } T_{\min} = 18$

IAS-9. Ans. (b) It is due to easy manufacturing.

IAS-10. Ans. (d) contact ratio= $\frac{length\ of\ are\ of\ con\tan t}{circular\ pitch}$

$$= \frac{\sqrt{R_{A^2} - R^2\cos^2\theta} + \sqrt{r_{A^2} - r^2\cos^2\theta} - (R+r)\sin\theta}{P_c(\cos\theta)}$$

IAS-11. Ans. (c)

IAS-12. Ans. (d)

IAS-13. Ans. (d)

IAS-14. Ans. (b)

IAS-15. Ans. (a)

IAS-16. Ans. (a)

IAS-17. Ans. (a) The ratio of angular speeds of F to A $= \frac{T_A.T_C.T_E}{T_B.T_D.T_F} = \frac{20\times 30\times 25}{60\times 80\times 75} = \frac{1}{24}$

IAS-18. Ans. (c) Re volution of 25 teeth gear $= 1 + \frac{T_{100}}{T_{25}}$ (for one rotation of arm) $= 1 + \frac{100}{25} = 5$

IAS-19. Ans. (c) $T_i+T_o+T_{arm}=0$ and $T_i\omega_i + T_o\omega_o + T_{arm}\omega_{arm} = 0$

$$\text{Gives, } T_{arm}=T_i\left(\frac{\omega_i}{\omega_o}-1\right)=T_i\left(\frac{N_i}{N_o}-1\right)=100\times\left(\frac{-1000}{50}-1\right)=-2100\ Nm$$

IAS-20. Ans. (a) $\dfrac{N_B-N_C}{N_A-N_C}=-\dfrac{T_A}{T_B}\quad \because N_A=0$

$$\frac{N_B-N_C}{-N_C}=-\frac{48}{24}\qquad or -\frac{N_B}{N_C}+1=-2\qquad or\ \frac{N_B}{N_C}=2+1=3$$

IAS-21. Ans. (b) $T_1+T_2+T_3=0$

$T_1W_1+T_2W_2+T_3W_3=0$

$W_3=0$

$T_1W_1=100kW$, $W_1=100rad/s$

$\because T_1=1\ k\ Nm$

$$\text{Or } T_2=-\frac{T_1W_1}{W_2}=\frac{-100}{(10)}=-10kNm$$

$$T_3=-T_2-T_1=-(-10)-1=9kNm$$

IAS-22. Ans. (d)

IAS-23. Ans. (a)

IAS-24. Ans. (d)

IAS-25. Ans. (b) $m=\dfrac{210}{30}=7\quad and\ P_x=\pi m=\dfrac{22}{7}\times 7=22mm$

Axial pitch = circular pitch of the worm wheel= πm

IAS-26. Ans. (a) $\dfrac{\text{Output torque}}{\text{Input torque}}=\dfrac{\text{pitch diameter of worm wheel}}{\text{pitch of worm}}=\dfrac{84}{11}=7.6$

IAS-27. Ans. (b)

IAS-28. Ans. (d) In rolling pair one link rolls over another fixed link.

TYPES OF BELTS SYSTEMS

1- FLAT BELT SYSTEM 2- GROVED BELT (V) SYSTEM

1) FLAT BELT DRIVES :

In its simplest form a drive of this type consists of an endless belt fitted tightly over two pulleys (driving and driven) transmitting motion from the driving to the receiving pulley by frictional resistance between belt and pulleys. The flexibility of the belt makes it possible to arrange the shafts of the driving and driven pulleys in any manner and to use as many pulleys as necessary.

SELECTION OF BELT DRIVE:

Following are the various important factors upon which the selection of a belt drive depends:

1- Speed of the driving and driven shafts.
2- Speed reduction ratio.
3- Power to be transmitted.
4- Centre distance between the shafts.
5- Positive drive requirements.
6- Shafts layout.
7- Space available.
8- Service conditions.

TYPES OF BELT DRIVES:

1- **Light drives**, these are used to transmit small powers at belt speeds up to about 10 $m/\sec$ as in agricultural machines and small machines tools.
2- **Medium drives**, these are used to transmit medium powers at belt speeds over 10 $m/\sec$ up to 22 $m/\sec$ as in machine tools.
3- **Heavy drives**, these are used to transmit large powers at belt speeds above 22 $m/\sec$ as in compressors and generators.

MATERIALS USED FOR BELTS:

The Martials used for belts and ropes must be strong, flexible, and durable. It must have high coefficient of friction:

1- Leather belts
2- Cotton belts
3- Rubber belts
4- Balata belts

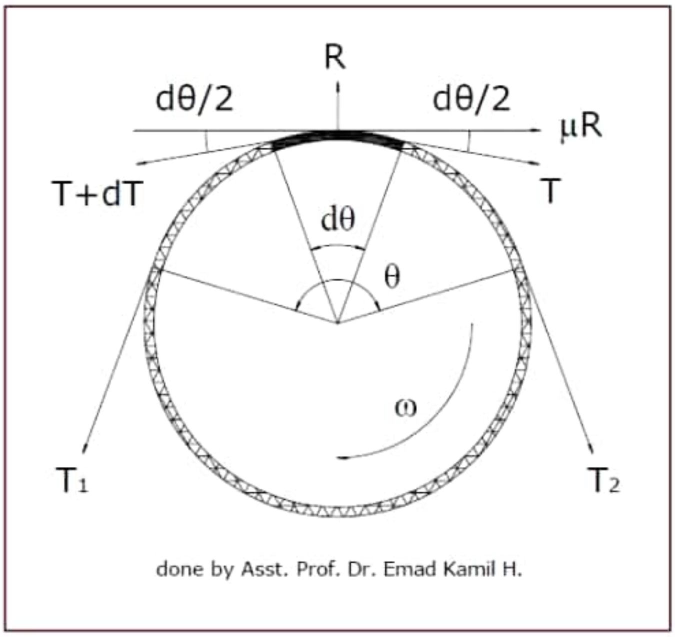
WORKING STRESSES IN BELTS:

The ultimate strength of leather belt varies from 21 to 35 N/mm^2 and a factor of safety may be taken as 8 to 10. However the wear life of a belt is more important than actual strength. It has been shown by experience that under average conditions an allowable stress of 2.8 N/mm^2 or less will give a reasonable belt life. An allowable stress of 1.75 N/mm^2 may be expected to give a belt life of about 15 years.

RATIO OF BELT TENSIONS

Consider a flat belt partly wounded around a pulley so that the angle of lap is θ and let T_1 and T_2 be the tensions in the belt when it is about to l=slip.

If the tensions at the ends of an element subtending an angle dθ at the center are T and T+dT and the reaction between the belt and the pulley is R, then resolving forces radially:



$$(T+dT)\frac{d\theta}{2}+T\frac{d\theta}{2}=R$$

Therefore, neglecting the second order of small quantities

$$Td\theta=R$$

Resolving forces tangentially

$$(T+dT)-T=\mu R$$

$$i.\ e. \qquad dT=\mu R$$

$$dT=\mu T d\theta$$

$$\therefore \frac{dT}{T}=\mu d\theta$$

$$\int_{T_2}^{T_1}\frac{dT}{T}=\int_0^{\theta}\mu d\theta$$

$$\therefore \ln T\Big|_{T_2}^{T_1}=\mu\theta$$

$$\therefore \ln T_1-\ln T_2=\mu\theta$$

$$\therefore \ln\frac{T_1}{T_2}=\mu\theta$$

$$\therefore \frac{T_1}{T_2}=e^{\mu\theta}$$

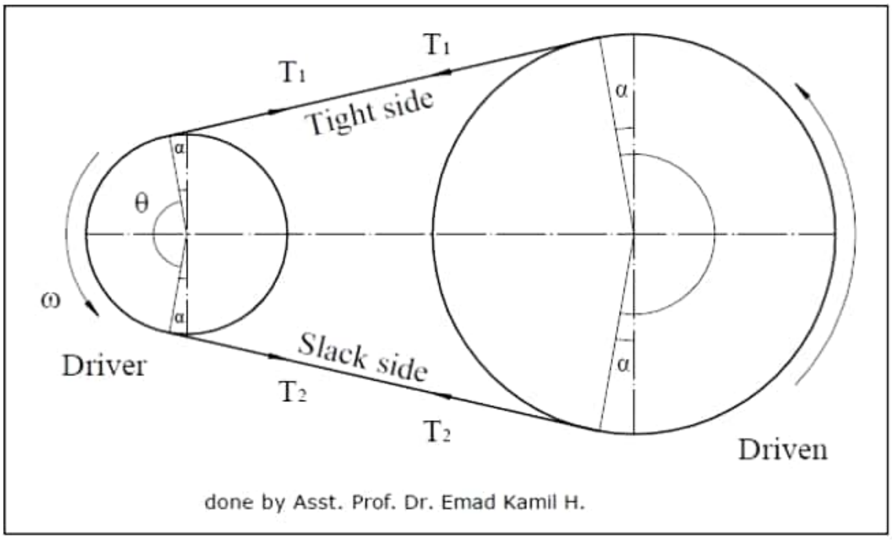
If the belt is used to transmit power between two pulleys T_1 and T_2 are the tight and slack side tensions respectively. If the pulleys are of unequal diameters, then the belt will slip first on the pulley having the smaller angle of lap, i. e. on the smaller pulley.

If v is the linear speed of the belt in $m/\sec$ and T_1, T_2 are in Newton, then,

$$\text{Power transmitted} = (T_1-T_2)*v$$

$$\text{Power} = T_1\left(1-\frac{1}{e^{\mu\theta}}\right)*v$$

H.W. Prove the following formula $\text{Power} = T_1\left(1-\frac{1}{e^{\mu\theta}}\right)*v$



EFFECT OF CENTRIFUGAL ACTION

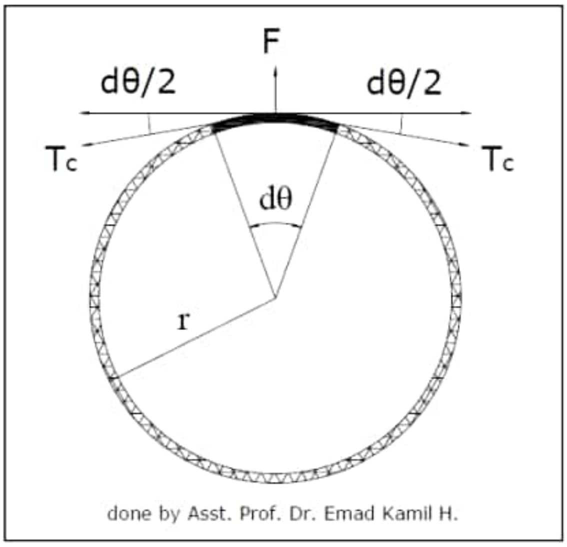
Consider a belt of mass m per unit length wounded around a pulley of radius r. Let the speed of the belt be v and the centrifugal tension T_c. If F is the centrifugal force acting on an element of the belt subtending an angle dθ at the center, then resolving forces radially,

$$F = 2T_c \frac{d\theta}{2}$$

But centrifugal force = $\frac{m * v^2}{r}$ and arc length $S = r * \theta$, taking into your account mass of belt measured by mass per unit length ($\frac{Kg}{m}$)

$$m * r * d\theta * \frac{v^2}{r} = T_c d\theta$$

$$T_c = mv^2$$



This tension is caused by centrifugal force on the belt and is additional to the tension due to the transmission of power. If allowance is made for this additional tension in determining the ratio of the belt tensions,

$$\frac{T_1 - T_c}{T_2 - T_c} = e^{\mu\theta}$$

$T_1 - T_c$ *and* $T_2 - T_c$ are the effective driving tensions and T_1, T_2 are the total tensions in the belt, so the total ordinary power transmitted is:

$$Powe = (T_1 - T_c)\left(1 - \frac{1}{e^{\mu\theta}}\right)v$$

From this equation the power transmitted is a maximum when:

$$\frac{d}{dv}\{(T_1 - T_c)v\} = 0$$

$$\text{but } T_c = mv^2$$

$$\frac{d}{dv}\{T_1 v - mv^3\} = 0$$

$$T_1 - \frac{mv^2}{3} = 0$$

$$\text{or } mv^2 = \frac{T_1}{3}$$

$$\therefore T_c = \frac{T_1}{3}$$

The maximum power is then obtained by substituting this value of T_c and the corresponding value of v in the equation of the power.

INITIAL TENSION

The belt is assembled with an initial tension T_o, when power is being transmitted, the tension in the tight side increases from T_o to T_1 and on the slack side decreases from T_o to T_2. If the belt is assumed to obey Hook's law and its length to remain constant, then the increase in length of the tight side is equal to the decrease in length of the slack side,

$$\Delta T = \Delta T$$

$$T_1 - T_o = T_o - T_2$$

Since the lengths and cross-sectional areas of the belt are the same on each side, then:

$$T_1 + T_2 = 2T_o \quad \text{Or} \quad T_o = \frac{T_1 + T_2}{2}$$

VELOCITY RATIO

When the thickness of the belt is NOT considered, the velocity ratio will be:

$$\frac{N_2}{N_1} = \frac{d_1}{d_2}$$

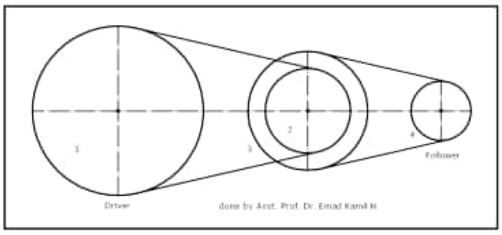
When the thickness of the belt is considered, the velocity ratio will be:

$$\frac{N_2}{N_1} = \frac{d_1 + t}{d_2 + t}$$

In case of compound belt drive, the velocity ratio is given by:

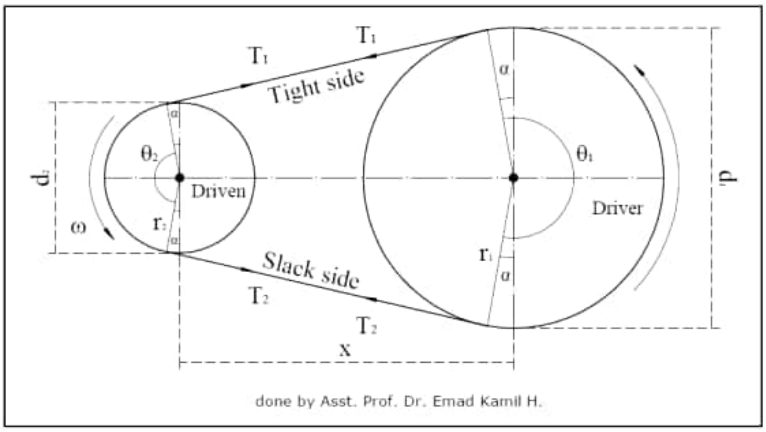
$$\frac{N_4}{N_1} = \frac{N_2}{N_1} * \frac{N_4}{N_3} \qquad (N_2 = N_3)$$

$$\frac{N_4}{N_1} = \frac{d_1 * d_3}{d_2 * d_4} \text{ or } \frac{speed\ of\ last\ driven}{speed\ of\ first\ driver} = \frac{product\ of\ diameters\ of\ drivers}{product\ of\ diameters\ of\ drivens}$$



LENGTH OF THE BELT

A) Open belt drive:



$$\alpha = \sin^{-1}\left(\frac{d_1 - d_2}{2x}\right)$$

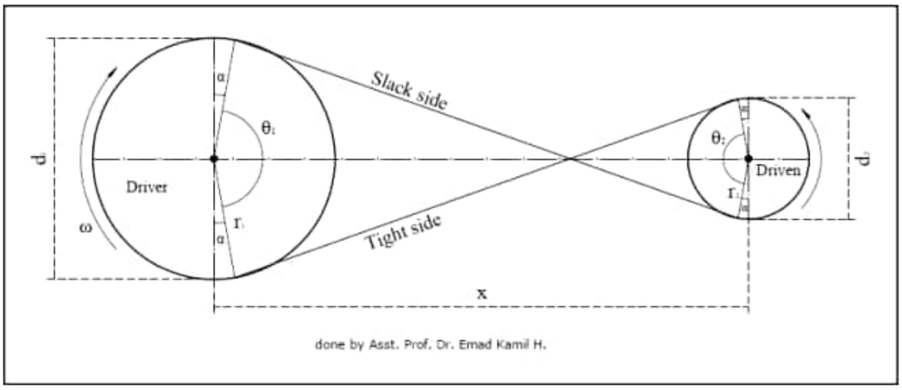
$$\theta_1 = 180 + 2\alpha$$

$$\theta_2 = 180 - 2\alpha$$

$$L = \frac{\pi}{2}(d_1 + d_2) + 2x + \frac{(d_1 - d_2)^2}{4x}$$

$$L = \pi(r_1 + r_2) + 2x + \frac{(r_1 - r_2)^2}{x}$$

B) Cross belt drive:



$$\alpha = \sin^{-1}\left(\frac{d_1 + d_2}{2x}\right)$$

$$\theta_1 = \theta_2 = 180 + 2\alpha$$

$$L = \frac{\pi}{2}(d_1 + d_2) + 2x + \frac{(d_1 + d_2)^2}{4x}$$

$$L = \pi(r_1 + r_2) + 2x + \frac{(r_1 + r_2)^2}{x}$$

SLIP OF THE BELT

If s_1 = slip between driver and belt

s_2 = slip between belt and follower

So the speed ratio equal to:

$$\frac{N_2}{N_1}=\frac{d_1}{d_2}\left[1-\frac{S_1+S_2}{100}\right]$$

$$\frac{N_2}{N_1}=\frac{d_1}{d_2}\left[1-\frac{S}{100}\right] \quad where\ S=S_1=S_2$$

If the thickness of the belt is considered, then:

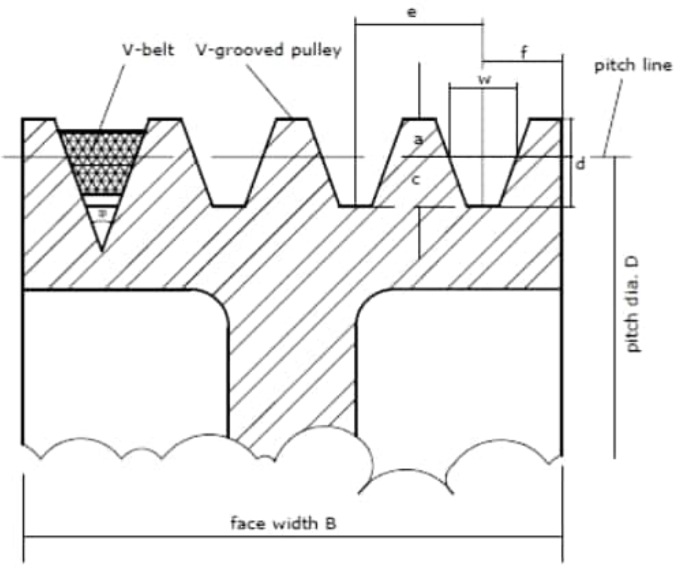
$$\frac{N_2}{N_1}=\frac{d_1+t}{d_2+t}\left[1-\frac{S}{100}\right]$$

Where t is the thickness of the belt.

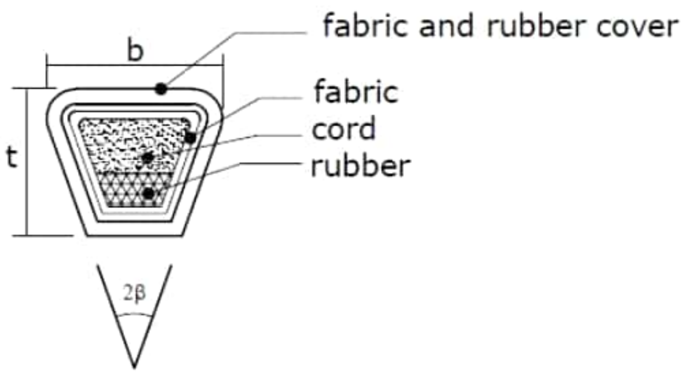
2) GROVED BELT (V) SYSTEM :

The V-belt is mostly used in factories and workshops where a great amount of power is to be transmitted from one pulley to another when the two pulleys are very near to each other. The V-belts are made of fabric and cords moulded in rubber and covered with fabric and cover as shown in figure below.

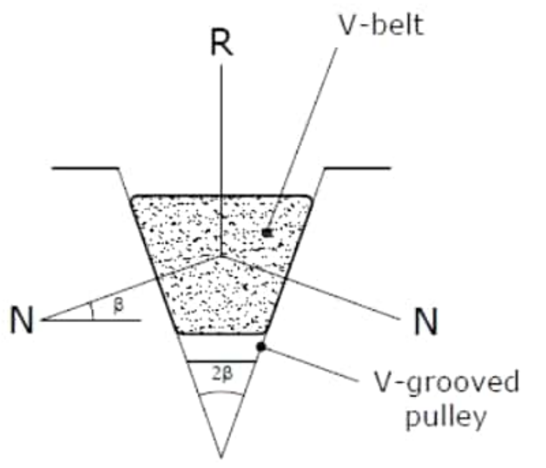
These belts are moulded to a trapezoidal shape and are made endless. The included angle for the V-belt is usually from 30° to 40°. The power is transmitted by the wedging action between the belt and the V-grooved in the pulley or sheave.



Cross-section of a V-grooved pulley
done by Asst. Prof. Dr. Emad Kamil H.



Cross-section of a V-belt
done by Asst. Prof. Dr. Emad Kamil H.



V-belt with a V-grooved pulley
done by Asst. Prof. Dr.Emad Kamil H

In order to increase the power output several V-belts may be operated side by side. It may be noted that in multiple V-belt drive all the belts should stretch at the same rate so that the load is equally divided between them. When one of the set of belts breaks the entire set should be replaced at the same time. If only one belt is replaced then the new one unworn and un-stretched belt will be more tightly stretched and will be move with different velocity.

TYPES OF V-BELTS AND PULLEYS

The V-belts are made of five types i.e. A, B, C, D, and E. The dimensions for the standard V-belts are shown in table 1 below and the dimensions of the standard V-grooved pulleys are shown in table 2 below.

TABLE 1

TYPE OF V-BELT	HORSE POWER RANGE (hp)	MINIMUM PITCH DIAMETER OF PULLEY (mm)	TOP WIDTH b (mm)	THICKNESS t (mm)	MASS PER METER LENGTH (Kg)
A	1-5	75	13	8	0.106
B	3-20	125	17	11	0.189
C	10-100	200	22	14	0.343
D	30-200	355	32	19	0.596
E	40-500	500	38	23	0.866
Note : 1 hp = 745.7 Watts **All above values are obtained from the Germany standard DIN**					

TABLE 2

TYPE OF V-BELT	w (mm)	d (mm)	a (mm)	c (mm)	f (mm)	e (mm)	No. of sheave groove n	Groove angle 2β (degree)
A	11	12	3.3	8.7	10	15	6	32, 34, 38
B	14	15	4.2	10.8	12.5	19	9	32, 34, 38
C	19	20	5.7	14.3	17	25.5	14	34, 36, 38
D	27	28	8.1	19.9	24	37	14	34, 36, 38
E	32	33	9.6	23.4	29	44.5	20	34, 36, 38
Note : Face width $B=(n-1)e+2f$ **All above values are obtained from the Germany standard DIN**								

RATIO OF DRIVING TENSIONS FOR V-BELTS

For V-grooved pulley, the normal force between the belt and the pulley is increased since the radial component of this force must equal R, thus if the semi angle of the groove is β, then:

$$N=\frac{R}{2}\csc\beta$$

Frictional resistance = $2\mu N$

$$2\mu\frac{R}{2}\csc\beta=\mu R\csc\beta$$

The friction force is therefore increased in the ratio (cscβ=1), so that the grooved pulley having a coefficient of friction of μcscβ, hence,

$$\frac{T_1}{T_2}=e^{\mu\theta\csc\beta}$$