BELT DRIVE

* **Flexible Drive :-**
It is having flexible element which provide flexibility to drive

Advantages
- used for long centre distance
- Absorb shock & vibration
- tolerate centre distance due to flexibility
- cheap, low maintenance

Disadvantages:-
- require more space
- velocity ratio - small
- velocity ratio - is not constant

* **Bet Drive :**

Advantages :-
- used for considerable center distance (long)
- smooth & silent operation
- Transmit definite load
- Absorb shock & vibration
- simple design
- low initial cost

Disadvantages :-
- Require more space
- velocity ratio is not constant
- impose heavy load on shaft
- low power transmission & also efficiency is low
- short service life.

* comparison between flat Belt & V-Belt :-

* **Flat Belt**

Advantages
- cheap
- act as clutch.
- simple & inexpensive

- Different velocity ratio can be obtained by shifting mechanism
- work in dusty and abrasive environment
- centre distance - long (15 m)
- efficiency is more than v-belt

Disadvantages :-
- Power transmitting capacity is low
- velocity ratio is lower (4:1)
- large dimension
- noisy
- used for horizontal position power transmission

* V- Belt :-

Advantages
- large power transmitting capacity
- compact
- i = 7:1
- smooth & quite at high speed
- no slip → positive drive
- can be used in vertical position.

Disadvantages :-
- h/diameter = large → so large bending stress
- efficiency - low - creep high
- costlier

* Desirable Properties of Belt
- High coefficient of friction, tensile strength & flexibility
- low wear and low rigidity.

1

Geometric Relationships :-

- Two types of Belt construction
1) open
2) crossed

} Both constructions - shafts are parallel

Open Belt Construction

i) belt proceeds from top of one pulley to top of another without crossing

ii) Driving & driven pulley rotates in same direction.

iii) angle of wrap is smaller so power transmitting capacity is less

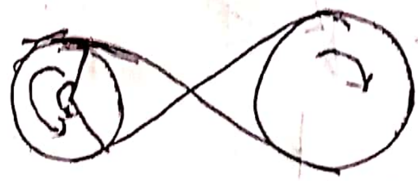
iv) so rubbing & bending so less wear & more life

v) with large centre distance belt whips & vibrates in transverse direction.

- for small centre distn. slip increases

more popular.

Cross Belt Construction



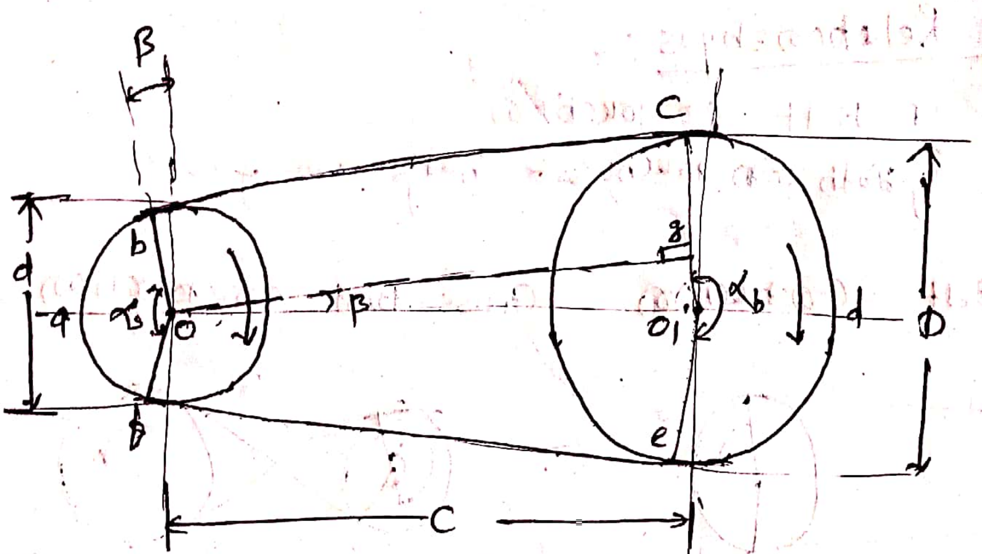
i) Belt proceeds from top of one pulley to bottom of another & cross over itself

ii) driving & driven pulley pulley rotates in opposite direction.

iii) angle of wrap is large so power transmitting capacity is more

iv) Belt rub against each other while crossing & belt has to bend in two diff. plane. so increases wear & reduces life of bearing

v) cross belt drive does not have this limitation.



Open Belt drive is shown in figure.

α_s = wrap angle of small pulley

α_b = wrap angle of big pulley

D = Diameter of big pulley

d = Diameter of small pulley

C = centre distance.

– Draw a line $\overline{og}$ ⊥ to line o_1c

Area ogcb is rectangle.

$$ob = gc$$

– From Δ ogo_1,

$$\sin\beta = \frac{o_1g}{oo_1} = \frac{o_1c - gc}{oo_1} = \frac{o_1c - ob}{oo_1}$$

$$\sin\beta = \frac{(D/2) - (d/2)}{C} = \frac{D-d}{2C}$$

– also. $\alpha_s = (180 - 2\beta)$

$\alpha_b = (180 + 2\beta)$

– Therefore $\alpha_s = 180 - 2\sin^{-1}\left(\frac{D-d}{2C}\right)$

$$\alpha_b = 180 + 2\sin^{-1}\left(\frac{D-d}{2C}\right)$$

- From figure, length of belt (L) is given by

$$L = arc(fab) + \overline{bc} + arc(cde) + \overline{eb}$$

$$= \frac{d}{2}(\alpha_s) + \overline{og} + \frac{D}{2}(\alpha_b) + \overline{og}$$

$$= \frac{d}{2}(\pi - 2\beta) + c\cos\beta + \frac{D}{2}(\pi + 2\beta) + c\cos\beta$$

$$L = \frac{\pi(D+d)}{2} + \beta(D-d) + 2c\cos\beta \quad \text{—— (a)}$$

- for small value of β,

$$\beta = \sin\beta = \left(\frac{D-d}{2c}\right)$$

$$\& \cos\beta = 1 - 2\sin^2\left(\beta/2\right) = 1 - \beta^2/2$$

$$= 1 - \frac{(D-d)^2}{8c^2}$$

- Substituting value of β & $\cos\beta$ in —— (a)

$$L = \frac{\pi}{2}(D+d) + \left(\frac{D-d}{2c}\right)(D-d) + 2c\left[1 - \frac{(D-d)^2}{8c^2}\right]$$

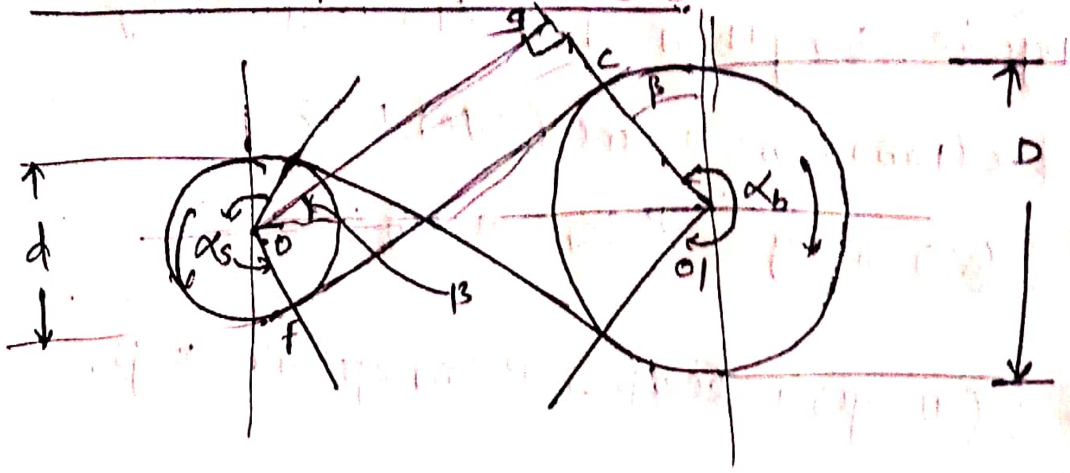
$$= \frac{\pi}{2}(D+d) + \frac{(D-d)^2}{2c} + 2c - \frac{(D-d)^2}{4c}$$

$$\boxed{L = 2c + \frac{\pi(D+d)}{2} + \frac{(D-d)^2}{4c}}$$

for flat Belt drive.

Scanned by CamScanner

For Crossed Belt Drive :-



- Draw line $\overline{og}$ ⊥ to line $\overline{O_1C}$

□ ofcg is rectangle

- from △ ogo_1

$$\sin\beta = \frac{O_1g}{OO_1} = \frac{O_1C + Cg}{OO_1} = \frac{O_1C + OF}{OO_1}$$

$$\sin\beta = \frac{(D/2) + (d/2)}{c} = \left(\frac{D+d}{2c}\right)$$

$$\alpha_s = \alpha_b = 180 + 2\beta$$

$$\alpha_s = \alpha_b = 180 + 2\sin^{-1}\left(\frac{D+d}{2c}\right)$$

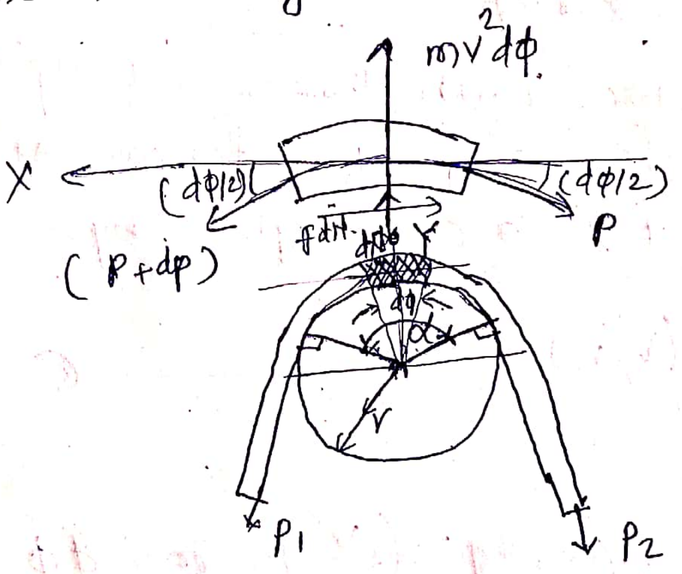
- length of cross belt drive

L =

$$\left[L = 2C + \pi\left(\frac{D+d}{2}\right) + \frac{(D+d)^2}{4C} \right]$$

* Analysis of Belt tensions:-

- The forces acting on the elements of a flat belt are shown in fig.



P_1 = belt tension in tight side (N)
P_2 = belt tension in loose side (N)
m = mass of one meter length of belt (kg/m)
v = belt velocity m/s
f = coeff. of friction
α = angle of wrap (radians)

- Element of belt subtending an angle ($d\phi$) is in equilibrium under the action of the following forces

i) tension (P) & (P+dP) are loose side & tight side respectively.

ii) normal reaction betn surfaces of belt and pulley (dN) & frictional force ($f \times dN$) at the interface

iii) centrifugal force in radially outward direction.

centrifugal force = mass × accln ——— (a)

- length of element is $(r\,d\phi)$ and mass per unit length is m ——— (b)

$\therefore$ mass of element $= m r\,d\phi$

- If a particle revolves with linear velocity 'v' at a radius 'r' from axis of rotation, the normal or centripetal accl^n is given by

$$f = \frac{v^2}{r} \quad \text{——— (c)}$$

- from (a), (b) & (c)

centrifugal force $= (m r\,d\phi)\left(\frac{v^2}{r}\right) = m v^2\,d\phi$.

- considering the equilibrium of forces in X & Y direction.

$$x \to (P + dP)\cos\left(\frac{d\phi}{2}\right) - P\cos\left(\frac{d\phi}{2}\right) - f\,dN = 0. \quad \text{——— (1)}$$

$$Y \to (P + dP)\sin\left(\frac{d\phi}{2}\right) + P\sin\left(\frac{d\phi}{2}\right) - m v^2 d\phi - dN = 0. \quad \text{——— (2)}$$

- for small values of $(d\phi/2)$

$$\cos\left(\frac{d\phi}{2}\right) \approx 1$$

$$\sin(d\phi/2) \approx (d\phi/2)$$

substituting above values in (1).

$$(P + dP) - P - f\,dN = 0.$$

$$dP - f\,dN = 0.$$

$$\boxed{dN = \frac{dP}{f}} \quad \text{——— (3)}$$

Similarly substituting $\sin\left(\frac{d\phi}{2}\right) \simeq \frac{d\phi}{2}$ in (2).

$$(P + dP)\frac{d\phi}{2} + P\left(\frac{d\phi}{2}\right) - mv^2 d\phi - dN = 0.$$

$$P\frac{d\phi}{2} + dP \cdot \frac{d\phi}{2} + P\frac{d\phi}{2} - mv^2 d\phi - dN = 0$$

Neglecting $(dP \times d\phi)$ as second order term is so small.

$$P\,d\phi - mv^2 d\phi - dN = 0.$$

substituting (3) in above eqn.

$$(P - mv^2)\,d\phi - \frac{dP}{f} = 0$$

$$\frac{dP}{(P - mv^2)} = f\,d\phi.$$

Integrating above expression.

$$\int_{P_2}^{P_1} \frac{dP}{(P - mv^2)} = f\int_0^{\alpha} d\phi$$

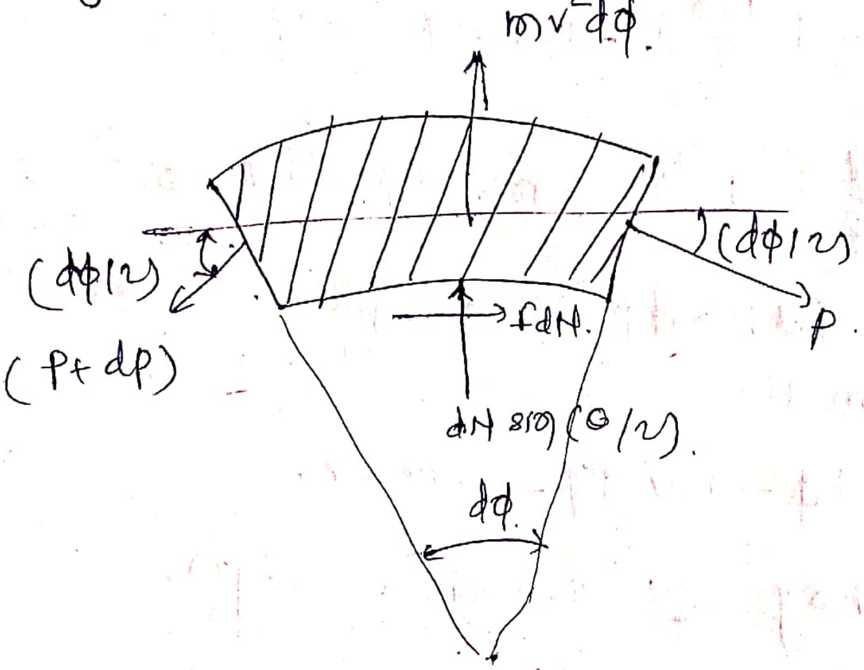
$$\left[\log (P - mv^2)\right]_{P_2}^{P_1} = f\left[\phi\right]_0^{\alpha}$$

$$\log_e(P_1 - mv^2) - \log_e(P_2 - mv^2) = f(\alpha - 0)$$

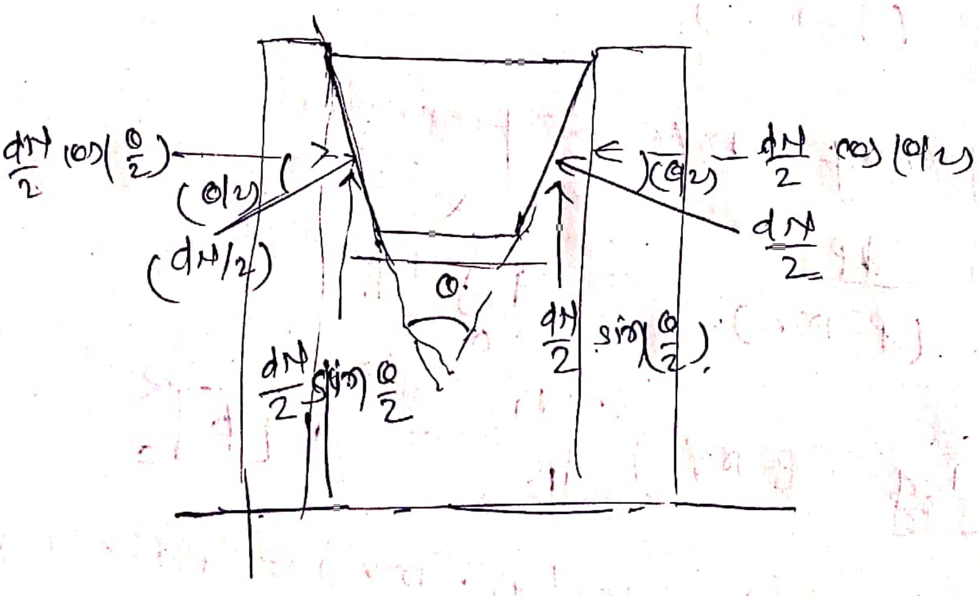
$$\log_e\left[\frac{P_1 - mv^2}{P_2 - mv^2}\right] = f\alpha.$$

$$\boxed{\left[\frac{P_1 - mv^2}{P_2 - mv^2}\right] = e^{f\alpha}}$$

* The forces acting on V-belts are shown in fig.



$$\left[\frac{P_1 - mv^2}{P_2 - mv^2}\right] = e^{\left[f\alpha / \sin(\theta/2)\right]}$$



- The forces acting on element of V-belt are

1) P & (P + dP) = tension on lose & tight side respt.
2) $mv^2 d\phi$ = centrifugal force acting outward.
3) fdN = will be the frictional force, ie. $\frac{1}{2} f dN$ on one side & $\frac{1}{2} f dN$ on other slant surface of belt
4) The normal reaction is inclined to surface.

1

- The normal reaction in x-Y plane is ~~acting~~. its vertical component $= \frac{dN}{2}\sin\left(\frac{\theta}{2}\right) + \frac{dN}{2}\sin\left(\frac{\theta}{2}\right)$

$= dN\sin\left(\frac{\theta}{2}\right)$

- considering equilibrium of forces in horizontal & vertical direction in x-Y Plane.

$\Sigma H = 0: \quad (P+dP)\cos\left(\frac{d\phi}{2}\right) - P\cos\left(\frac{d\phi}{2}\right) - \mu\, dN = 0$ —— (1)

$\Sigma V = 0 \quad (P+dP)\sin\left(\frac{d\phi}{2}\right) + P\sin\left(\frac{d\phi}{2}\right) - mv^2 d\phi - dN\sin\left(\frac{\theta}{2}\right) = 0$ —— (2)

for small values of $\left(\frac{d\phi}{2}\right)$

$\cos\left(\frac{d\phi}{2}\right) \approx 1$

$\sin\left(\frac{d\phi}{2}\right) \approx \left(\frac{d\phi}{2}\right)$

substituting above values in (1)

$(\cancel{P} + dP) - \cancel{P} - \mu\, dN = 0$

$dP = \mu\, dN$

$dN = \frac{dP}{\mu}$ —— (3)

substituting $\sin(d\phi/2) \approx (d\phi/2)$ in eq^n (2)

$(P+dP)\frac{d\phi}{2} + P\left(\frac{d\phi}{2}\right) - mv^2 d\phi - dN\sin\left(\frac{\theta}{2}\right) = 0$

$P\,d\phi + dP\cdot\frac{d\phi}{2} - mv^2 d\phi - dN\sin\left(\frac{\theta}{2}\right) = 0$

$\left(dP\cdot\frac{d\phi}{2}\right) =$ neglecting $\Rightarrow$ terms of second order differential

$$(P - mv^2)\, d\phi - dN \cdot \sin\left(\frac{\theta}{2}\right) = 0$$

$$(P - mv^2)\, d\phi = dN \sin\left(\frac{\theta}{2}\right)$$

substituting eq^n 3 in above equation

$$(P - mv^2)\, d\phi = \frac{dP}{f} \sin\left(\frac{\theta}{2}\right)$$

$$\frac{f\, d\phi}{\sin\left(\frac{\theta}{2}\right)} = \frac{dP}{(P - mv^2)}$$

Integrating above expression.

$$\int_0^{\alpha} \frac{f\, d\phi}{\sin\left(\frac{\theta}{2}\right)} = \int_{P_2}^{P_1} \frac{dP}{(P - mv^2)}$$

$$\frac{f}{\sin(\theta/2)} \left[\phi\right]_0^{\alpha} = \left[\log_e (P - mv^2)\right]_{P_2}^{P_1}$$

$$\frac{f\alpha}{\sin\left(\frac{\theta}{2}\right)} = \log (P_1 - mv^2) - \log (P_2 - mv^2)$$

$$\left[f\alpha / \sin(\theta/2)\right] = \log \left(\frac{P_1 - mv^2}{P_2 - mv^2}\right)$$

$$\boxed{\frac{P_1 - mv^2}{P_2 - mv^2} = e^{f\alpha / \sin(\theta/2)}}$$

comparing expression for flat belt & V-belt

i) f coeff of friction in flat belt & $f/\sin(\theta/2)$ - coeff. of friction in V- belt

ii) for V-belt $\theta = 40$. so

$f / \sin(\theta/2) = f / \sin(40/2) = f/0.342 = 2.92 f$.

- for identical material of belt, pully & coeff of friction of V-belt is 2.92 times of flat belt, so power transmitting capacity is more in V-belt

Condition for Maximum Power :-

- Belt has given initial tension (P_i) for power transmission
- initial tension depends upon length of belt, elasticity of belt, geometry of pulley & centre distance.

Assumption :-

i) length of belt is constant
ii) belt has linear elasticity.

- When the driving pulley begins to rotate, the elongation on tight side is proportional to ($P_1 - P_i$) while contraction on loose side is proportional to ($P_i - P_2$)
- above condition is must for constant length of belt.

$$(P_1 - P_i) = (P_i - P_2)$$

$$P_i = \frac{1}{2}(P_1 + P_2) \qquad \text{———} \quad (1)$$

We know that

$$\frac{P_1 - mv^2}{P_2 - mv^2} = e^{f\alpha}$$

$$\frac{P_2 - mv^2}{P_1 - mv^2} = \frac{1}{e^{f\alpha}} = \frac{e^{-f\alpha}}{1}$$

$$\frac{(P_2 - mv^2) + (P_1 - mv^2)}{(P_2 - mv^2) - (P_1 - mv^2)} = \frac{e^{-f\alpha} + 1}{e^{-f\alpha} - 1}$$

$$\frac{(P_1 + P_2) - 2mv^2}{(P_2 - P_1)} = \frac{e^{-f\alpha} + 1}{e^{-f\alpha} - 1}$$

Substituting (1) in above equation.

$$\frac{2P_i - mv^2}{-(P_1 - P_2)} = \frac{e^{-f\alpha} + 1}{e^{-f\alpha} - 1}$$

$$2P_i - mv^2 = \frac{e^{-f\alpha} + 1}{e^{-f\alpha} - 1} \times -(P_1 - P_2)$$

$$(P_1 - P_2) = 2(P_i - mv^2)(-1)\frac{(e^{-f\alpha} - 1)}{(e^{-f\alpha} + 1)}$$

$$(P_1 - P_2) = 2(P_i - mv^2)\left(\frac{1 - e^{-f\alpha}}{1 + e^{-f\alpha}}\right)$$

We know

$$\text{Power} = (P_1 - P_2)\,v$$

$$= 2(P_i - mv^2)\left(\frac{1 - e^{-f\alpha}}{1 + e^{-f\alpha}}\right)v$$

$$(P) = 2(P_i v - mv^3)\left(\frac{1 - e^{-f\alpha}}{1 + e^{-f\alpha}}\right)$$

Differentiating above eqn w.r.t v & equating it to (0) zero

$$\frac{\partial}{\partial v}(\text{Power}) = 0$$

$$\frac{\partial}{\partial v}\left[2(P_i v - mv^3)\left(\frac{1 - e^{-f\alpha}}{1 + e^{-f\alpha}}\right)\right] = 0$$

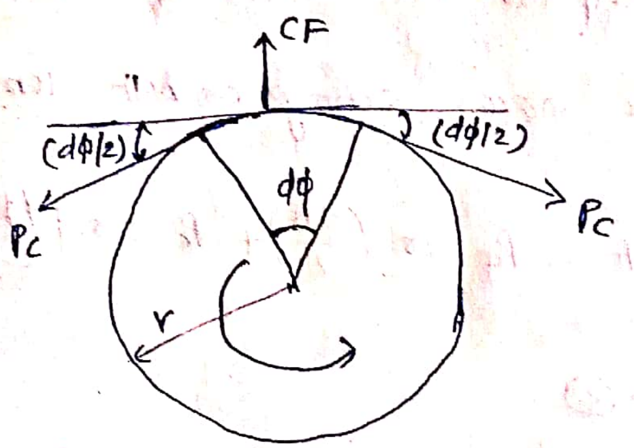
$$2\left[P_i - 3mv^2\right]\left(\frac{1 - e^{-f\alpha}}{1 + e^{-f\alpha}}\right) = 0$$

$$P_i - 3mv^2 = 0$$

$$\boxed{v = \sqrt{\frac{P_i}{3m}}}$$

optimum velocity of the belt for maximum power transmission.

* Condition for maximum Power (Alternative Approach)



- When belt passes over pulley, centrifugal forces due to its own weight tends to lift belt from surface of pulley.
- An element of belt subtending an angle φ at the center of pulley is shown in fig

length of belt element $= r \times \theta = r\,d\phi$

Mass of element $= r\,d\phi \cdot m$

where m = mass per unit length of belt rotating about axis of pulley.

- Acceleration of belt is $\left(\frac{v^2}{r}\right)$

- centrifugal force $= CF =$ mass x acceleration

$$= m r\,d\phi \times \left(\frac{v^2}{r}\right)$$

$$\left[CF = m v^2 d\phi\right] \quad \text{—— (1)}$$

- This centrifugal force induces belt tension P_c, which is equally divided on two sides of belt

- Resolving the forces acting on belt element in vertical position

$$Cf = P_c \sin\left(\frac{d\phi}{2}\right) + P_c \sin\left(\frac{d\phi}{2}\right) = 2P_c \sin\frac{d\phi}{2} \quad -(2)$$

from (1) + (2)

$$mv^2 d\phi = 2 P_c \sin\left(\frac{d\phi}{2}\right)$$

for small value of $d\phi$. $\sin\left(\frac{d\phi}{2}\right) \approx \frac{d\phi}{2}$

$$\therefore mv^2 d\phi = 2P_c\left(\frac{d\phi}{2}\right)$$

$$\boxed{P_c = mv^2}$$

→ Belt can transmit maximum power when following two conditions are simultaneously satisfied.

i) Tension on belt reaches maximum permissible value for the belt cross-section.

ii) The belt on point of slipping ie. maximum friction force is developed in belt.

suppose $b =$ width of belt

$t =$ thickness.

$\sigma =$ max. permissible tensile stress for belt

- Maximum belt tension $P_{max} = bt\sigma$

- since there is tension due to centrifugal force $P_1 = P_{max} - P_c$

We know

$$\frac{P_1}{P_2} = e^{f\alpha}$$

$$P_2 = \frac{P_1}{e^{f\alpha}}$$

Power transmitted by belt is given by

$$\text{Power} = (P_1 - P_2)v = \left(P_1 - \frac{P_1}{e^{f\alpha}}\right)v$$

$$= P_1 v\left[1 - \frac{1}{e^{f\alpha}}\right]$$

$$\text{Power} = P_1 v k \quad - - - \quad k = \left[1 - \frac{1}{e^{f\alpha}}\right] = \text{constant}$$

$$= (P_{max} - P_c)vk$$

$$= (P_{max} - mv^2)vk$$

$$= (P_{max}v - mv^3)k$$

Power transmitted will be maximum when

$$\frac{\partial}{\partial v}(\text{Power}) = 0$$

$$\frac{\partial}{\partial v}\left\{[P_{max}v - mv^3]k\right\} = 0$$

$$P_{max} - 3mv^2 = 0 \quad \longrightarrow (3)$$

$$\boxed{v = \sqrt{\frac{P_{max}}{3m}}}$$ → optimum velocity of belt for maximum power transmission.

from (3)

$$P_{max} - 3mv^2 = 0$$

$$P_{max} - 3P_c = 0 \qquad P_{max} = 3P_c$$

$$\therefore \boxed{P_c = \frac{P_{max}}{3}}$$

$$P_1 = P_{max} - P_c = 3P_c - P_c = 2P_c$$

$$\boxed{P_1 = 2P_c}$$

Conclusion : condition for maximum power transmission

i) maximum permissible tension in belt should be three times tension due to centrifugal force

$(P_{max} = 3P_c)$

ii) Tension in tight side of belt should be twice the tension due to centrifugal force

$(P_1 = 2P_c)$

iii) Belt ~~tension~~ velocity should be

$$V = \sqrt{\frac{P_{max}}{3m}}$$

* Selection of Belt from Manufacturer's catalogue:-

- In practice, while selecting a flat belt from mfg catalogue, the following parameter values should be known to the designer

i) power to be transmitted
ii) input and output speed
iii) centre distance depending upon availability of space
iv) type of load.

- The maximum power transmitted by belt is obtained by multiplying rated power by load correction factor

$$(P_{kW})_{max} = f_a \, (P_{kW})$$

$(P_{kW})_{max}$ = power transmitted by belt for design purpose.
(P_{kW}) = actual power transmitted by belt
f_a = load correction factor.

Type of load	
Normal load	$f_a = 1$
steady load	$f_a = 1.2$
Intermittent load	$f_a = 1.3$
shock load	$f_a = 1.5$

- Power transmitting capacity of belt are calculated for 180° arc of contact.
- actual arc of contact will be different for diff. appl.
- when arc of contact is lesser than 180°, there will be an ~~the~~ additional tension in belt. which is counted by considering arc of contact factor.

$$(P_{kW})_{corrected} = (P_{kW})_{max} \times F_d.$$

→ arc of contact should not lesser than 150°.

→ There are two varities of belt in Dunlop transmission belt.

* HI-SPEED duck belting - general application
  (P_{kw}) → 0.0118 kW per mm width per ply

* FORT duck belting - heavy application.
  (P_{kw}) ⇒ 0.0147 kW. per mm. width per ply.

- optimum belt velocity for these belts
  = 17.8 to 22.9 m/s.

- The above values of power based on two assumption
  arc of contact = 180°.
  belt velocity = 5.08 m/s

* Power rating / load rating of flat belt :-
- power transmitting capacity of the belt per mm width ply at 180° arc of contact.

Procedure :- Flat Belt selection.

i) Optimum belt velocity ⇒ 17.8 to 22.9 m/s
Assume belt velocity 18 m/s (in this range) & calculate.
- diameter of smaller pully

$$v = \frac{\pi d n}{60 \times 10^3}$$

$$d = \frac{v \times 60 \times 10^3}{\pi n}$$

d - n = input speed in rpm for smaller pulley

- Diameter of bigger pully (D)

$$\frac{D}{d} = \frac{\text{speed of smaller pully}}{\text{speed of bigger pully}} = \frac{\text{input speed}}{\text{output speed}}.$$

- Modify values of d & D to nearest preferred diameter for CI & mild steel, pully dia. are
100, 112, 125, 140, 160, 180, 200, 224, 250, 280, 315, 355, 400, 450, 500, 560, 630, 710, 800, 900, 1000.

- Calculate correct velocity for this preferred pully diameter & check whether the actual velocity is in the range of optimum belt velocity (17.8 to 22.9 m/s)

ii) Determine load correction factor 'fa' from table.
('It depends on type of load)
- find Pmax.

$$(P_{kW})_{max} = f_a (P_{kW})$$

iii) Calculate angle of wrap for smaller pully

$$\alpha_s = 180 - 2\sin^{-1}\left(\frac{D-d}{2C}\right)$$

- find arc ~~correction~~ of contact factor 'fd" (from table).

iv) Calculate corrected power.

$$(P_{KW})_{corrected} = (P_{KW})_{max} \times f_d .$$

v) Calculate corrected power rating for belt

- for high speed belt

$$\text{corrected } \cancel{\text{pow}} \; P_{KW} \text{ rating} = \frac{0.0118\, V}{5.08}$$

- For FORT belt.

$$\text{corrected } P_{KW} \text{ rating} = \frac{0.0147\, V}{5.08}$$

V = corrected belt velocity in m/s

vi) Calculate Belt width.

$$(\text{Width} \times \text{No. of ply}) = \frac{\text{corrected Power}}{\text{corrected belt rating}}$$

- calculate width by assuming suitable no. of plies
- Alternative soln – belt where width is near the value of standard width is optimum solution. (from table.
- select standard belt width.

vii) Calculate belt length.

$$L = 2C + \frac{\pi(D+d)}{2} + \frac{(D-d)^2}{4C}$$

Example :-

It is required to select a flat belt drive for a compressor running at 720 rpm, which is driven by a 25 kW, 1440 rpm motor. Space is available for a centre distance of 3 m. The belt is open type

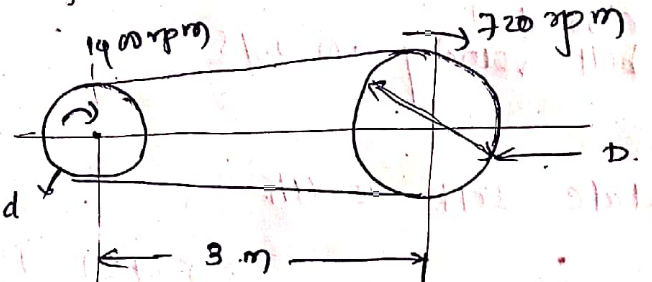
→ $P_{kW} = 25\ kW$, $n_1 = 1440\ rpm$,

$n_2 = 720\ rpm$.

$C = 3000\ mm$.

i) Diameter of pulley

Using velocity of belt 18 m/s.



We know $V = \frac{\pi d n}{60 \times 10^3}$

$$d_1 = \frac{60 \times 10^3 \times V}{\pi n_1} = \frac{60 \times 1000 \times 18}{\pi \times 1440} = 238.73\ mm$$

selecting the preferred pulley diameter of 250 mm

$d = 250\ mm$

$D = d(1440/720) = 250 \times 2 = 500\ mm$ (same as preferred dia.)

Belt Velocity for this dimensions

$$V = \frac{\pi d n_1}{60 \times 10^3} = \frac{\pi \times 250 \times 1440}{60 \times 1000} = 18.85\ m/s$$ which

is in bet^n optimum velocity range.

Scanned by CamScanner

ii) Maximum Power for selection.

load correction factor for compressor is

$$f_a = 1.3$$

$$\text{Maximum } P_{kW} = f_a (P_{kW})$$

$$= 1.3 \times 25$$

$$= 32.5 \text{ kW}.$$

iii) Angle of wrap for smaller pulley

$$\alpha_s = 180 - 2\sin^{-1}\left(\frac{D-d}{2C}\right)$$

$$= 180 - 2\sin^{-1}\left(\frac{500-250}{2 \times 3000}\right)$$

$$= 175.23^\circ C.$$

~~Use~~ for finding ~~index~~ $f_d \Rightarrow$ arc of contact factor at $175.23^\circ C$ of wrap angle, use interpolation.

$$f_d = 1.019.$$

iv) Corrected Power

$$(P_{kW})_{corrected} = (P_{kW})_{max} \times f_d$$

$$= 32.5 \times 1.019$$

$$= 33.12 \text{ kW}.$$

v) corrected Power rating of Belt

for HI speed belt

$$\text{corrected belt rating} = \frac{0.0118(18.85)}{5.08} = 0.0438 \text{ kW}.$$

vi) selection of Belt width.

$$(\text{width} \times \text{no. of plies}) = \frac{\text{corrected Power}}{\text{corrected belt rating}}$$

$$= \frac{33.12}{0.0438}$$

(width × no. of plies) = 756.17

for Belt width using trial & error method

for 4 plies $\quad w = \frac{756.17}{4} = 189.04$ mm

for 5 plies $\quad w = \frac{756.17}{5} = 151.23$ mm

for 6 plies $\quad w = \frac{756.17}{6} = 126.03$ mm

~~for~~ comparing calculated width with preferred width of corresponding plies in standard series. 152 is very close to 151.23 of 5 plies in standard series

So selecting 5 plies & 152 mm width belt.

VII) <u>Belt length</u>

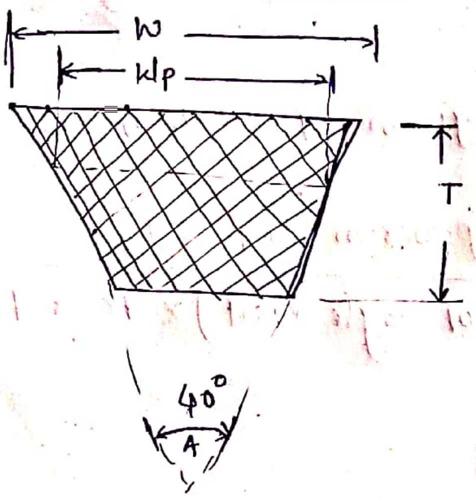
$$L = 2C + \frac{\pi(D+d)}{2} + \frac{(D-d)^2}{4C}$$

$$= 2(3000) + \frac{\pi(500+250)}{2} + \frac{(500-250)^2}{4(3000)}$$

$$= 7183.31 \text{ mm}$$

<u>Belt Dimensions</u> :. 7.2 m length
152 mm width
5 plies belt of HI speed.

* Selection of V- Belts :-



i) Pitch Width : (W_p) - width of the belt at its pitch zone

ii) Nominal ~~pitt~~ top width (W) :- top width of trapezium

iii) Nominal height (T) :- height of trapezium.

iv) Angle of Belt : (A) :- included angle obtained by extending sides of belt (40°)

v) Pitch length (L_p) :- length of pitch line of belt or circumferential length of belt at pitch width.

- six-basic standardised cross-section of belt A, B, C, D, E & Z.
  Z → used for low power transmission
  A, B, C, D & E → widely used for general purpose application

- Designation of Belt :-
  B4430Lp ⇒ V- belt of cross section B & pitch length 4430 mm

- Groove angle for belt = 40° & for pulley (34- 38°)
  This result in wedging action thereby increasing frictional force & consequently transmitted power.

* Selection of cross-section :-

- It depends on power to be transmitted & speed of faster pulley.
- In case of borderline case alternative design calculations are made to determine best solution.

- V-belt dimensions are taken from preferred pitch dia & pitch length. From table.
- No. of belts required for given application is calculated by

$$\text{No. of belts} = \frac{(\text{transmitted power in kW}) \times F_a}{(\text{kW rating of single V-belt}) \times F_c \times F_d}$$

$$= \frac{P \times F_a}{P_r \times F_c \times F_d}.$$

where P = Power to be transmitted.

P_r = Power rating of single V-belt.

F_a = correction factor for service condition

F_c = correction factor for belt length

F_d = correction factor for arc of contact.

Scanned by CamScanner

* Basic Procedure for selection of V-Belts :-

Designer has to select a V-belt from manufacturer's catalogue based on following requirement

i) Type of driving unit.
ii) Type of driven unit
iii) operational hours per day
iv) Power to be transmitted
v) Input and output speeds
vi) centre distance

Basic Steps

① **Correction factor according to service (Fa):-**

Determine correction factor according to service condition ie. type of driving & driven unit, operational hours per day from design data book.

② **Design Power**

Calculate design Power $\left[\text{Design Power} = Fa \times (\text{transmitted Power}) \right]$

③ **Cross-section of Belt**

- Based on design power & speed of faster pulley, the cross-section of belt is selected
- In case of borderline, alternative calculations are made to find out best cross-section.

④ **Diameters of pulley :-**

- Diameter of smaller pulley ie pitch diameter is taken from preferred values (it depends on cross-section of belt)
- Diameter of bigger pulley calculated by

$$D = d \left[\text{input speed} / \text{output speed} \right]$$

- The values of D & d are compared with preferred pitch diameter and standardised diameter is chosen.

⑤ Pitch length of belt (L)

$$L = 2C + \frac{\pi(D+d)}{2} + \frac{(D-d)^2}{4C}$$

⑥ compare calculated L with preferred pitch length L & nearest pitch length is taken from table

⑦ Calculated corrected centre distance C :-

$$L = 2C + \frac{\pi(D+d)}{2} + \frac{(D-d)^2}{4C}$$

It will be quadratic equation in C & after solving 'C'

⑧ Correction factor (Fc)

Find fc from table as it depends upon type of cross. section & pitch length of belt.

⑨ Calculate Arc of contact for small pully : & Fd

$$\alpha_s = 180 - 2\sin^{-1}\left(\frac{D-d}{2C}\right)$$

calculate fd for arc of contact

- It is advisable to use an arc of contact less than 120° for V-belt

⑩ Power rating (Pr)

- Determine power rating (Pr) of single V-belt as it depends on speed of faster shaft, pitch dia. of smaller pulley & speed ratio.

Scanned by CamScanner

(11) No. of belts :-

It depends on design power, power transmitting capacity of belt

$$\text{No. of belts} = \frac{P \times f_a}{P_r \times f_c \times f_d}$$

Example:-

It is required to design a V-belt drive to connect a 7.5 kW 1440 rpm induction motor to a fan, running at approx. 480 rpm, for a service of 24 hr per day. space is available for a center distance of about 1 m.

→ **Given:-** $P_{kW} = 7.5$

$n_1 = 1440$ rpm

$n_2 = 480$ rpm

$C = 1$ m

service $= 24$ hr per day

i) Correction factor (f_a)

Induction motor is driving fan of 7.5 kW for 24 hr per day so from table $f_a = 1.3$

ii) Design Power

Design Power = f_a x (transmitted power)

$= 1.3(7.5)$

$= 9.75$ kW.

iii) Type of cross-section for belt.

For. $P_{kW} = 9.75$ kW & $n = 1440$ rpm

From graph. B-section belt is suitable.

iv) Diameter of pulley

Diameter of smaller pulley from table

$d = 200$ mm for B-section belt

$D = d(1440/480) = 200(3) = 600$ mm

From table $d = 200$ mm & $D = 600$ mm are preferred diameter. so

v) Pitch length of belt :-

$$L = 2C + \frac{\pi(D+d)}{2} + \frac{(D-d)^2}{4C}$$

$$= 2(1000) + \frac{\pi(600+200)}{2} + \frac{(600-200)^2}{4(1000)}$$

$$\boxed{L = 3296.64 \text{ mm}}$$

vi) Preferred pitch length :-

from preferred pitch length for B-section belt is

3200 to 3600 mm

so 3200 is closer to 3296.64 mm

so standard value taken from table L = 3200 mm

vii) correct center distance :-

substituting L = 3200 mm

$$L = 2C + \frac{\pi(D+d)}{2} + \frac{(D-d)^2}{4C}$$

$$3200 = 2C + \frac{\pi(600+200)}{2} + \frac{(600-200)^2}{4C}$$

$$\boxed{C = 950.64 \text{ mm}}$$

viii) correction factor for belt pitch length (Fc)

from table for B-section & L = 3200 mm pitch

$F_c = 1.08$

ix) correction factor for arc of contact (Fd)

$$\alpha_s = 180 - 2\sin^{-1}\left(\frac{D-d}{2C}\right)$$

$$= 180 - 2\sin^{-1}\left(\frac{600-200}{2 \times 950.64}\right)$$

$$= 155.7 \simeq 156^{\circ}$$

From Table for $\alpha = 156°$.

$$\underline{f_d = 0.94}$$

(ix). <u>Power rating of single V-belt</u>

From table for 1440 rpm, 200 mm pulley & B-section.

& speed ratio = 3

$$P_r = 5.9 + 0.46$$
$$= 6.36 \text{ kW}.$$

(x) <u>Number of belts :-</u>

$$\text{No. of belts} = \frac{P \times f_a}{P_r \times F_c \times f_d}$$

$$= \frac{7.5 \times (1.3)}{6.36\ (1.08)(0.94)}$$

$$= 1.51 \simeq 2 \text{ belts}.$$

Scanned by CamScanner

Chain Drive

- chain drive is an intermediate between belt & gear drive in performance.
- It can be used for short as well as long centre distance, also efficiency is in betn gear & belt.
- It consist of endless chain wrapped around two sprocket
- chain is a series of link connected by pin joint & sprocket is a toothed wheel
- used for velocity ratio upto 10:1 & chain velocity upto 25m/s & power transmission upto 100KW

* Advantages of chain Drive compared with belt & gear drive.

i) used for long as well as small center distance
ii) No. of shafts can be driven in same or opposite direction by single chain from single driving sprocket
iii) small overall dimension as compared to belt drive so compact
iv) does not slip and to that extent, it is positive drive
v) Properly lubricated chain drive as efficiency of 96 % to 98 %.
vi) Does not require initial tension so reduces forces on shaft
vii) chains are easy to replace
viii) Atmospheric conditions & temp. does not affect performance of drive.

* Disadvantages :-

i) operate without lubricant film so more wear that will increse pitch of chain & chain may leave sprocket
ii) Not suitable for non parallel shaft
iii) unsuitable where precise motion required due to polygonal effect
   - velocity of chain is not constant so non-uniform speed to driven shaft
iv) requires housing
v) compared to belt drive, it requires precise alignment of shaft
vi) lubrication need
vii) noisy

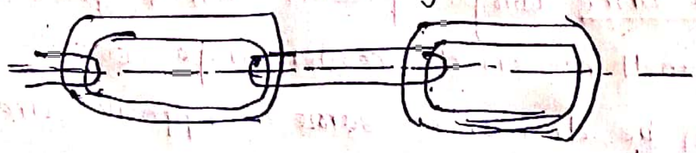
* Different types of chains are their with respect to their purpose.

i) load lifting chain
ii) Hauling chains
iii) Power transmission chains.

① Load lifting chains

- used for suspending, raising or lowering loads in material handling equipments

ex: link chain in low capacity hoists, hand operated cranes etc.



Advantages.

i) good flexibility in all direction
ii) operate with small diameter pulley & drum
iii) simple to design & easy to manufacture
iv) noiseless at low speed (ie. 0.1 m/s)

Disadvantages:-

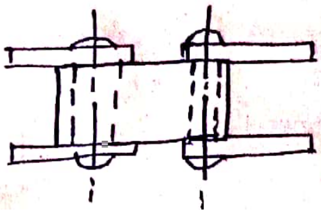
i) heavy
ii) susceptible to jerk & overloads
iii) sudden & complete failure
iv) operate at low speed.

② Hauling chains:-

- used for carrying materials continuously by sliding, pulling or carrying in conveyour.

ex:- Block chain.

- operate at medium velocity upto 2 to 4 m/s
- have long pitch due to long length.
- noisy and wear rapidly



Scanned by CamScanner

③ Power transmission chain :-

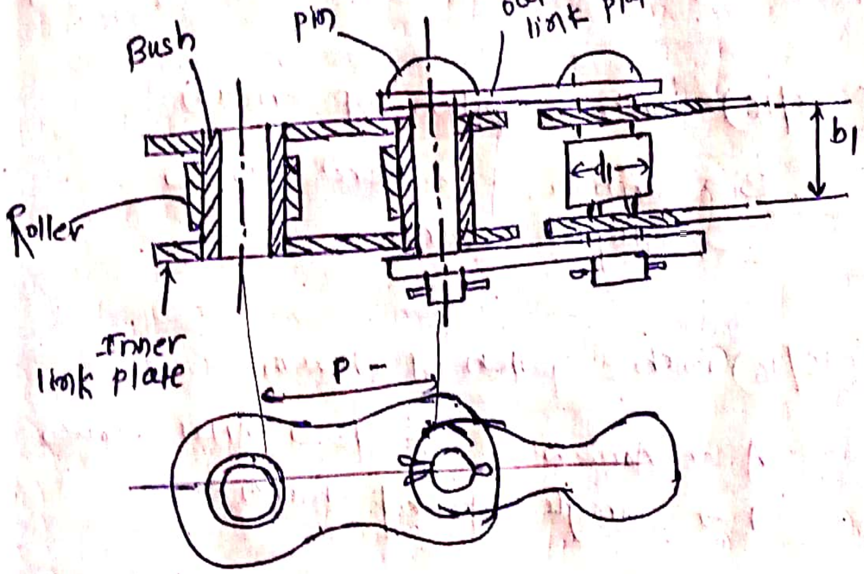
- used for transmitting power.

ex:- Roller chain

* Construction of Roller chain :-

components

i) pin
ii) Bushing
iii) Roller
iv) Inner link Plate
v) Outer link Plate.



- Pin is press fitted to two outer link plate while bush is press fitted to two inner link plate.
- Bush and pin form a swivel joint and outer link is free to swivel with respect to inner link
- Rollers are freely fitted on bush & during engagement turn with the teeth of sprocket wheel. This result in rolling friction instead of sliding friction. & reduces wear & improves efficiency of drive.

Material :-

Inner & outer plate → Medium carbon steel. (Blanked from cold rolled sheets & hardened to 50 HRC)

Pin, bush, Roller → case carbonize alloy steel with 50 HRC.

Pitch → linear distance betn axes of adjacent roller

single or multistrand — multi row of chains.

Breaking load :- maximum tensile load which if applied is result in chain failure.

Designation :-

American standard ANSI series or British standard series

08 B-2 → (08/16) inch ⇒ pitch of chain.

B → British standard series

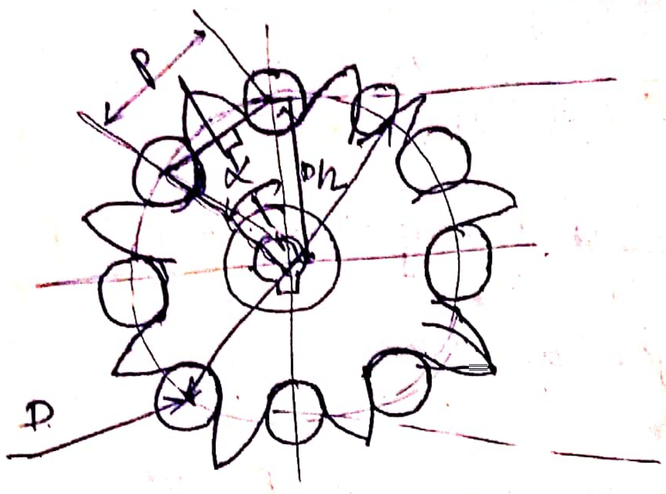
2 → duplex chain.

16 A-1 → (16/16) inch = pitch of chain

A → American standard ANSI series

1 → single strand chain.

Chain length - Geometrical Relationships:-



D = pitch circle Dia. of sprocket
α = pitch angle
p = pitch.
z = no. of teeth on sprocket.
n_1 = speed of driving shaft rpm
n_2 = speed of driven shaft rpm
z_1 = no. of teeth on ~~driving~~ smaller sprocket
z_2 = no. of teeth on ~~driven~~ larger sprocket.

Pitch circle Dia :- Imaginary circle ~~dia of sprocket~~ that passes through centers of link pins. as chain wrapped on sprocket

$$\alpha = \frac{360}{z}.$$

$$\sin(\alpha/2) = \frac{(P/2)}{(D/2)}$$

$$\left[D = \frac{p}{\sin(180/z)} \right]$$

- velocity ratio 'i' of chain drive is given by

$$i = \frac{n_1}{n_2} = \frac{z_2}{z_1}$$

- Angular velocity of sprocket is given by

$$V = \frac{\pi D n}{60 \times 10^3} \quad \ldots \quad \pi D = zp \text{ . substituting}$$

$$V = \frac{zpn}{60 \times 10^3} \qquad V = \text{average velocity in m/s.}$$

- The length of chain is always expressed in terms of no. of links.

$$L = L_n \times p$$

L = length of chain
L_n = no. of links in chain
p = pitch.

Scanned by CamScanner

- The no. of links in the chain is determined by the following approximate relationships

$$L_n = 2(a/p) + \left(\frac{Z_1+Z_2}{2}\right) + \left(\frac{Z_2-Z_1}{2\pi}\right)^2 \times \left(\frac{p}{a}\right)$$

a = centre distance betⁿ axes of driving & driven sprocket
Z_1 = no. of teeth on smaller sprocket
Z_2 = no. of teeth on larger sprocket.

- It is always prefer to have even no of links since chain consist of alternate pairs of inner & outer link plates.
- When chain have ~~additional~~ odd link than additional link (offset link) is provided (which is weaker than main link)
- After selecting exact no. of links, centre to centre distance betⁿ axes of two sprocket is calculated by following formula:

$$a = \frac{p}{4}\left\{\left[L_n - \left(\frac{Z_1+Z_2}{2}\right)\right] + \sqrt{\left[L_n - \left(\frac{Z_1+Z_2}{2}\right)\right]^2 - 8\left[\frac{Z_2-Z_1}{2\pi}\right]^2}\right.$$

- The above centre distance ~~doe~~ formula does not provide any sag.
- In practice small amount of sag is essential for link to take best position sprocket whep. so. centre distance is reduced by margin of 0.0029 to 0.0049 to account for sag.

* Selection of Roller chain from Manufacturer's Catalogue:-

The Power transmitted by roller chain can be expressed by elementary equation.

$$P_{kW} = (P_1 V / 1000) \text{ kW}.$$

P_1 = allowable tension in the chain (N) = F
V = average velocity of chain m/s.

- Allowable tension depends upon no. of factors such as type of chain, pitch of chain link, no. of teeth on smaller sprocket, chain velocity, type of power source & driven machinery & system of lubrication.

- The power ratings are generally given for simple (single strand) roller chain and are based on assumption that there are 17 teeth on driving sprocket.

→ Power rating of chain is determined by following relationship

$$P_{kW} \text{ rating of chain} = \frac{(P_{kW} \text{ to be transmitted}) \times K_s}{K_1 \times K_2}$$

where K_s = service factor
K_1 = multiple strand factor.
K_2 = tooth correction factor

K_s ⇒ It takes into consideration the effect of shocks & vibration on power to be transmitted

K_2 = It account for variation in no. of teeth on driving sprocket as Power rating is generally given for 17 no. of teeth on driving sprocket

K_1 = Power rating is generally given for single strand chain, if no. of strands are more it will compensate for that.

- for satisfactory performance of roller chains, the center distance between sprockets should provide at least a 120° wrap angle on smaller sprocket.

- In Practice, it is recommended that centre distance is between 30 to 50 chain pitches.

$$30p < a < 50p.$$

- Expected service life of these chains is 15000 hr.

- velocity ratio should be kept below 6:1 to get satisfactory performance.