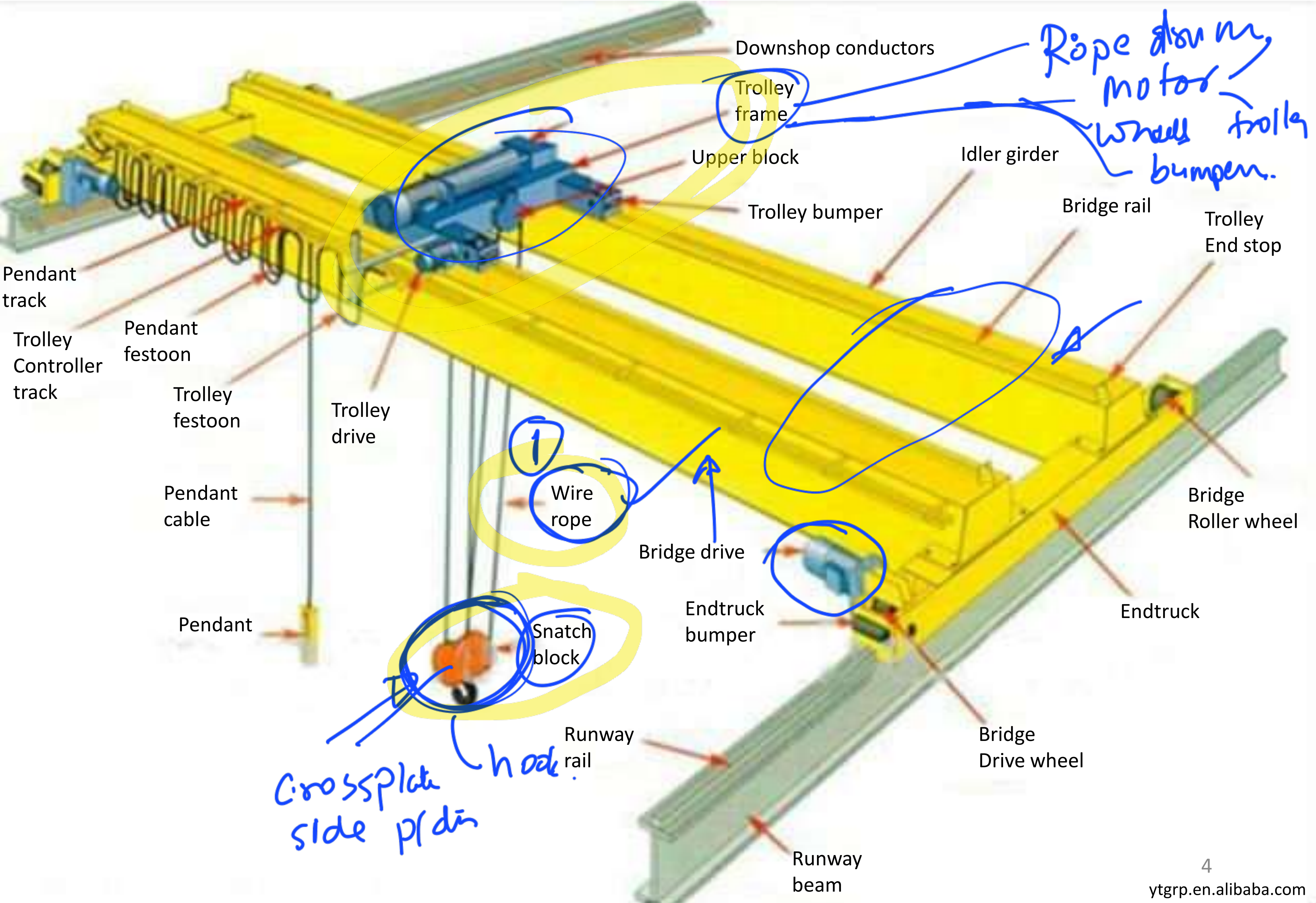
Design of Machines and Mechanical Systems (PC-BTM711)

Session ~~24~~ 25/26

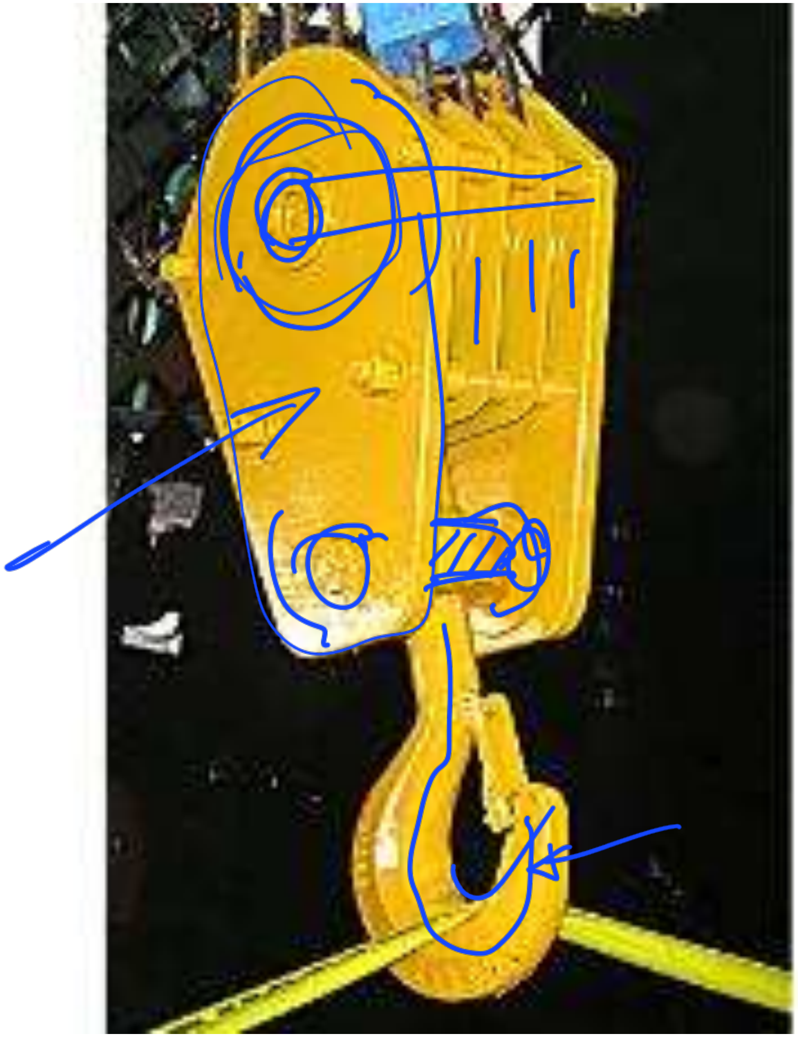
Module 6: Design of EOT Crane

Parts of EOT crane



ytgrp.en.alibaba.com

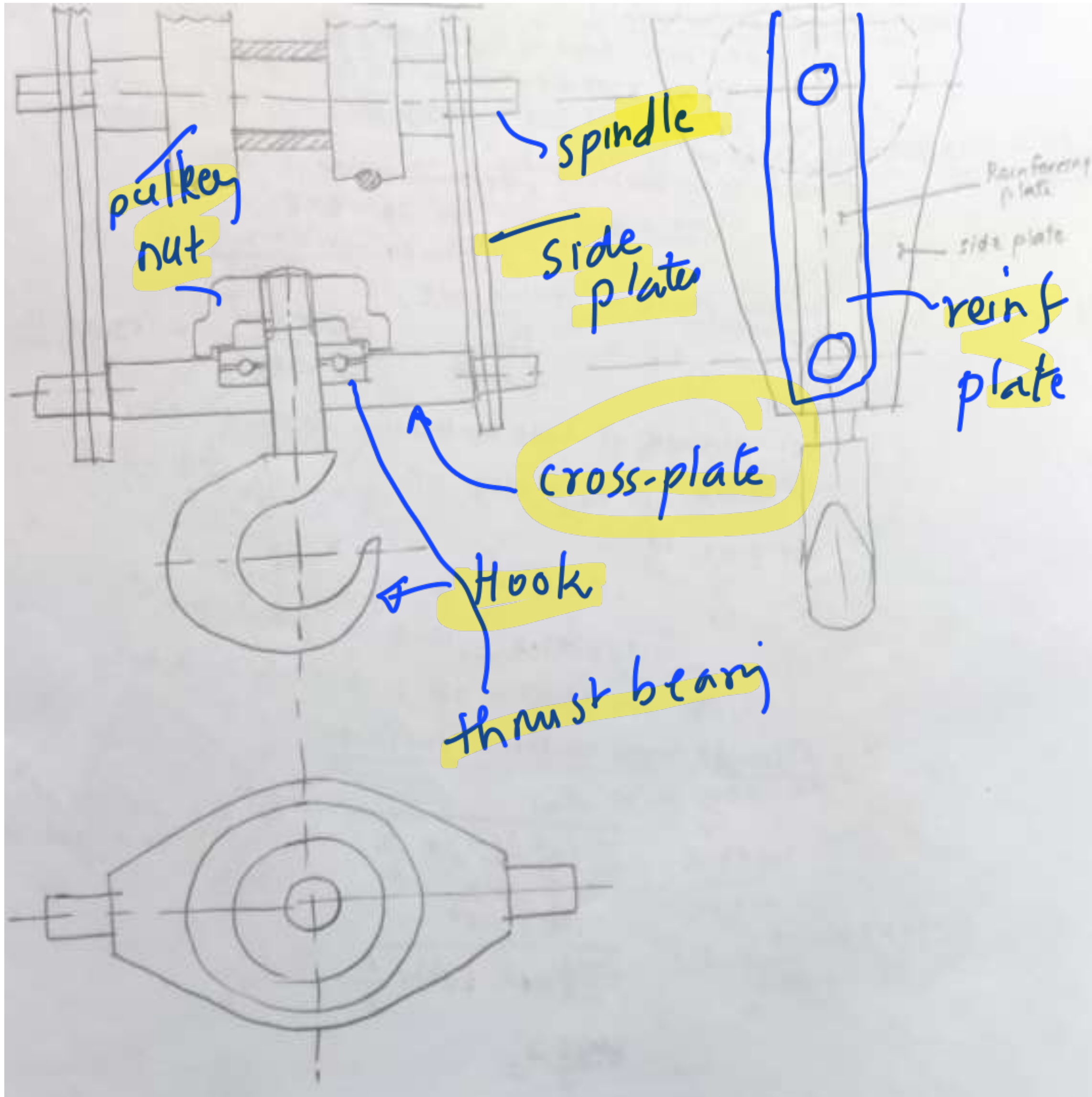
Snatch block



https://suppliers.jimtrade.com/167/166387/snatch_block_assembly_hooks.htm

http://www.eotcranemanufacturers.net/5-ton-snatch-block-780610.html#prod_img

Design of Snatch Block

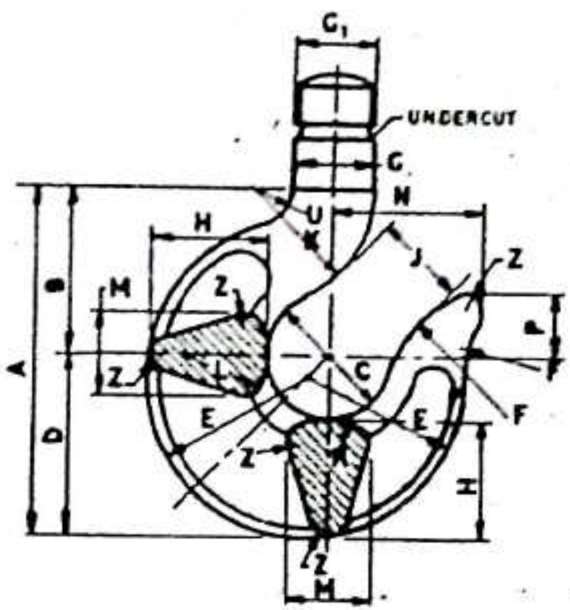


Lifting Hook (IS 3815)

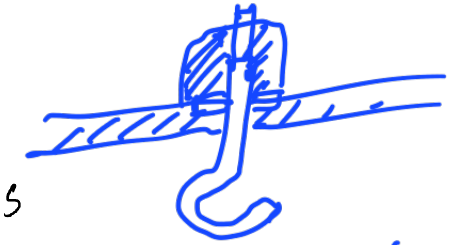
Normal Load for structural parts = 145 kN

14.8 tons

Safe load tonnes	Proof Load tonnes	C		G min	THREAD G_1	THREAD pitch	Thrust bearings on G Series
0.5	1	27	A = 2.75C	15	M14		
		23	B = 1.31C	12	M12		
			D = 1.44C				
1	2	38	E = 1.25C	20	M20		
		33	F = C	20	M18		
			H = 0.93C				511
2	4	53	J = 0.75C	30	M27	COURSE SERIES	
		46	K = 0.92C	25	M24		
			L = 0.7 C				512
3.2	6.4	68	M = 0.6 C	35	M33		
		59	N = 1.2 C	30	M30		
			P = 0.5 C				
5	10	85	R = 0.5 C	45	M42		
		73	U = 0.3 C	40	M30		
			Z = 0.12C				
8	16	107		55	M52		
		93		50	M48		
10	20	119		60	M60		
		104		55	M52		
							513
12 (14.8)	25	134		70	M68		
		116		60	M60		
16	32	151		80	M76	6	
		131		70	M68	6	
20	40	169		85	M80	6	
		147		80	M72	6	
25	50	189		100	M90	6	
		164		85	M80	6	



(A) Hook Design



Hook load = 145 kN = 14.8 tons

From IS 3815, we select hook with SWL = 16 tons

Let Thread size = $\boxed{M76 \times 6p}$

Thread calculations

Normal service load = R_N = 145 kN

Let weight of hook & tackle = $0.1 R_N$ = 14.5 kN

$\therefore$ Load on hook threads = F_h = 145 + 14.5 = $\boxed{159.5 \text{ kN}}$

Thread core diameter (approx), $d_c \approx$ Major dia. $- 2p$

$= 76 - 2 \times 6$

$= 64$ mm

(i) Tensile stress in thread core

$$\sigma_t = \frac{F_h}{\frac{\pi}{4} d_c^2} = \frac{159.5 \times 10^3}{\frac{\pi}{4} \times (64)^2} = 49.6 \ N/mm^2$$

Let strength of hook material = same as 4068

$\therefore F_a = 121.5$ MPa

For pure tension

$$\boxed{1.25 f_t \overset{?}{<} F_a}$$

$\therefore 1.25 \times 49.6 \overset{?}{<} 121.5$

$\therefore \boxed{62.0 < 121.5 \Rightarrow \text{Safe in tension}}$ ✓

(ii) Compressive stresses on thread surface

$$\sigma_c = \frac{F_n}{\frac{\pi}{4}(d_o^2 - d_c^2) \times \left(\frac{l}{p}\right)}$$

1 2

l = length of engaged threads

$$= \frac{159.5 \times 10^3}{\frac{\pi}{4}(71^2 - 64^2) \times \frac{l}{6}}$$

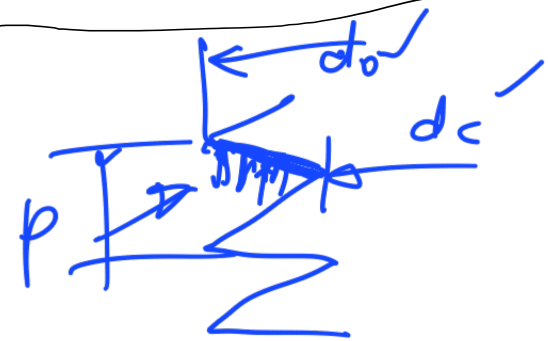
$$= \frac{725.3}{l}$$

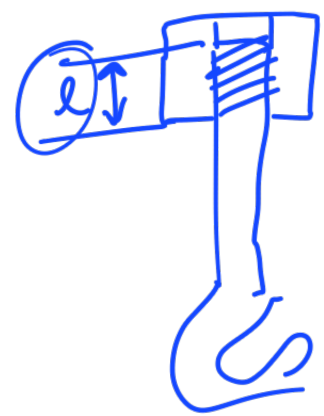
allowable stress

$$\therefore \boxed{f_c < F_a}$$

$$\therefore \frac{725.3}{l} < 121.5$$

$$\therefore \boxed{l = 5.97 \text{ mm}}$$





(iii) Shear stresses in thread

$$f_s = \frac{F_n}{\boxed{\pi\left(\frac{d_o + d_c}{2}\right)} \times \boxed{\frac{l}{2}}}$$

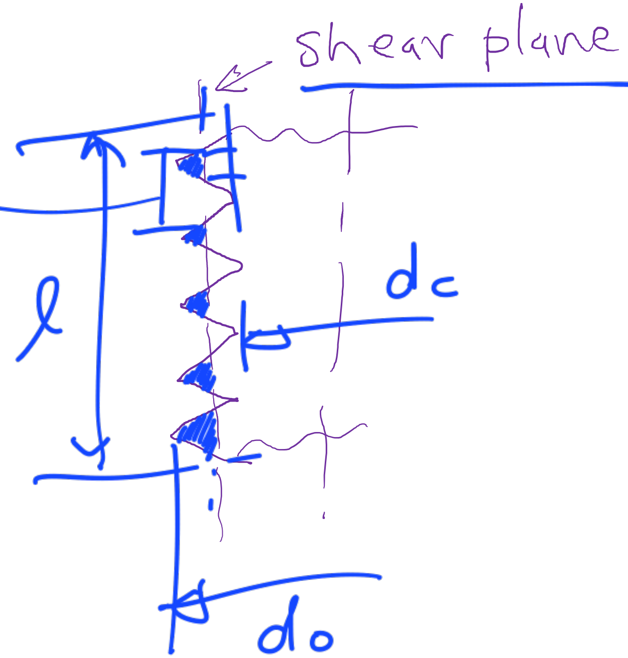
$$= \frac{159.5 \times 10^3}{\pi\left(\frac{76+64}{2}\right) \times \frac{l}{2}}$$

$$= \frac{1450.6}{l}$$

$$\therefore \boxed{\sqrt{3}\, f_s < F_a}$$

$$\therefore \sqrt{3} \times \frac{1450.6}{l} = 121.5$$

$$\therefore l = 20.7 \text{ mm}$$



$\therefore$ $\boxed{\text{selected engaged length of threads} = \text{Height of nut} = \boxed{25 \text{ mm}}}$

QUIZ

Shear Stress in Hook Threads

The nut height considered for calculating the shear stresses in hook threads is:

1) Full height

2) 0.8 x Full height

3) 0.5 x Full height ✓

(B) Thrust bearing selection for hook

Hook shank diameter, $G = 80$ mm

Bearing selected $= 51316$ with $d = 80$ mm

(DDB T15.33)

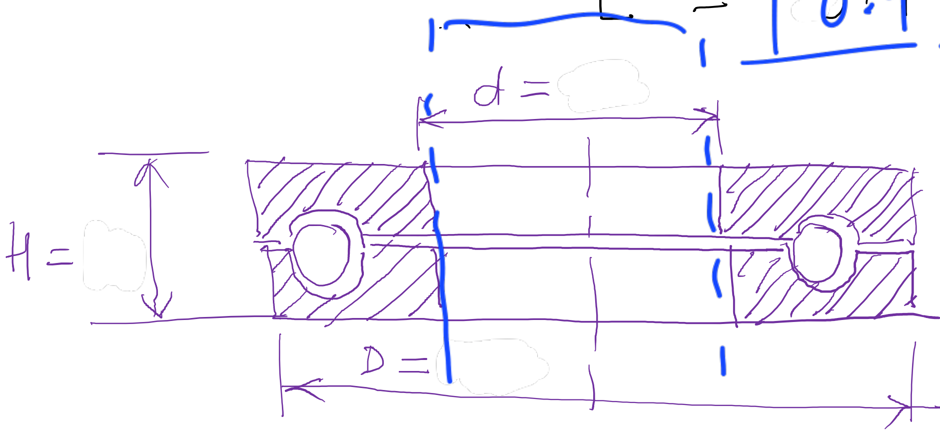
Dynamic capacity, $C = 159$ kN

$$\because \quad C = \left(\frac{L}{L_{10}}\right)^{1/k} \cdot P \qquad k = 3 \text{ for ball bearing}$$

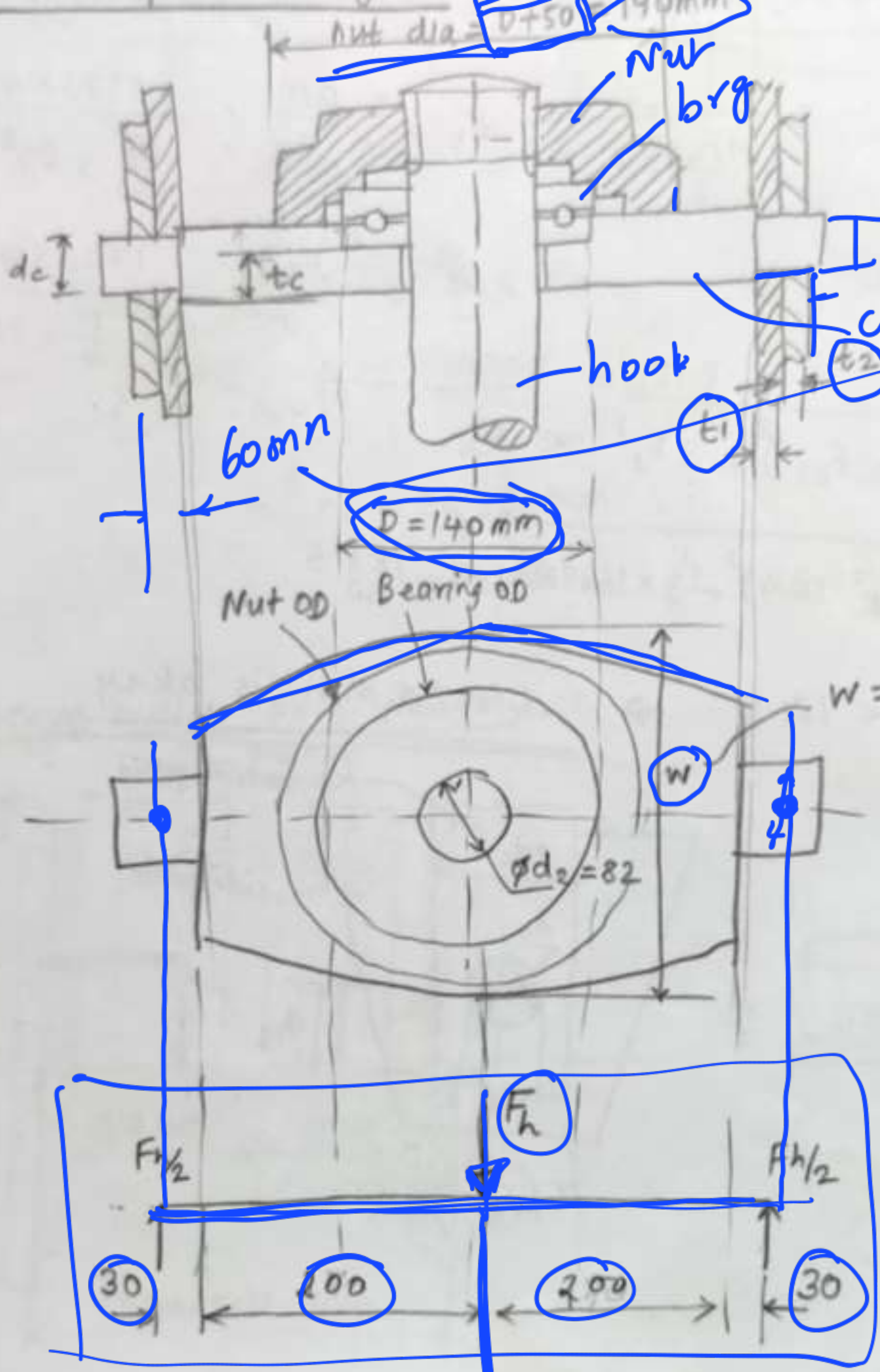
$$\therefore \quad 159 = \left(\frac{L}{1}\right)^{1/3} \times 159.5$$

$$\therefore \quad L = \boxed{0.99} \text{ mr}$$

$\Rightarrow$ okay for EOT service



Cross-plate design



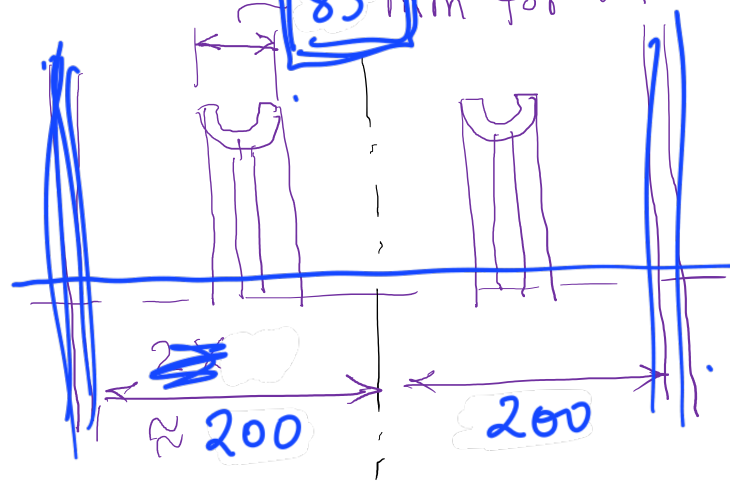
Let $t_1 + t_2 = \underbrace{20 + 40}_{\text{assumed}} = 60$ mm

190 mm

$W = $ nut dia. + 10 = $\boxed{200 \text{ mm}}$

PSG

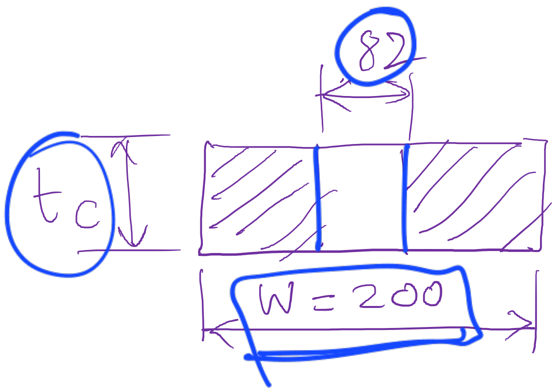
$\boxed{85}$ mm for wire 280



(C) Cross piece design

$$BM_{max} = \frac{F_h}{2} \times 200 = \frac{159.5 \times 10^3}{2} \times 200$$

$$= 1.595 \times 10^7 \; \cancel{\times 10^7} \; N.mm$$



$$Z = \frac{1}{6}(200 - 82) \times t_c^2$$

$$= 19.67 \, t_c^2$$

$$\boxed{f_{bt} < F_a}$$

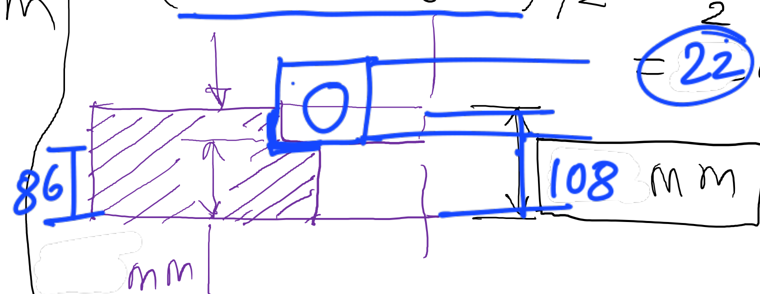
$$\therefore \frac{1.595 \times 10^7}{19.67 \, t_c^2} = 121.5$$

$\therefore \; t_c = \boxed{81.7}$ mm

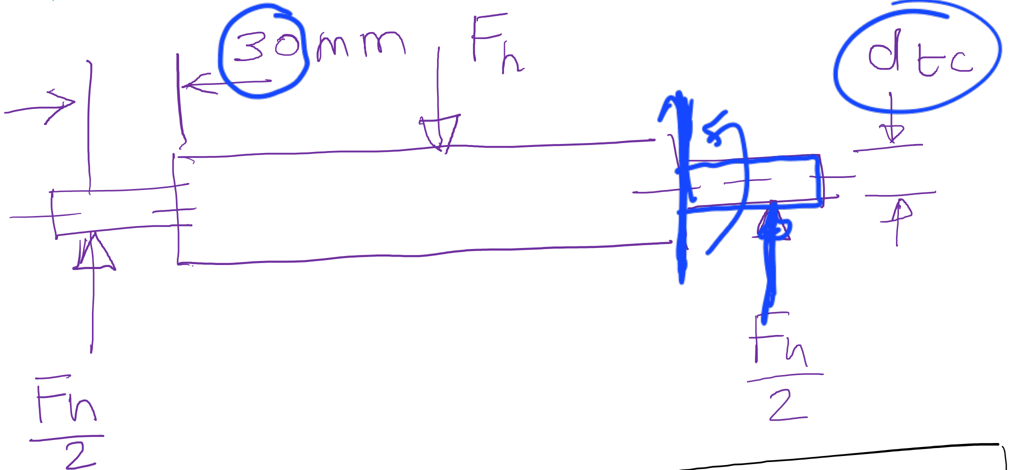
t_c selected $= \boxed{86}$ mm

(Thrust brg. ht)/2 $= \frac{44}{2}$

$= 22$ mm



Cross piece trunion diameter



Let $d_{tc} = 80$ mm

$$B_m = \frac{F_h}{2} \times 30$$

$$= \frac{159.5 \times 10^3}{2} \times 30$$

$$= 2.3925 \times 10^6 \text{ N-mm}$$

$$\sigma_b = \frac{B_m}{Z} = \frac{2.3925 \times 10^6}{\frac{\pi}{32} \times 80^3}$$

$$= 47.6 \text{ N/mm}^2$$

$$\tau = \frac{F_h/2}{A} = \frac{159.5 \times 10^3/2}{\frac{\pi}{4} \times 80^2}$$

$$= 15.87 \text{ N/mm}^2$$

$$\sqrt{(1.25 f_t + f_{bt})^2 + 3 f_s^2} < F_a$$

$$\therefore \sqrt{(1.25 \times 0 + 47.6)^2 + 3 \times 15.87^2} \overset{?}{<} 121.5$$

$\therefore$ $54.97 < 121.5$

$\Rightarrow$ $d_{tc} = 80$ mm is okay.

QUIZ

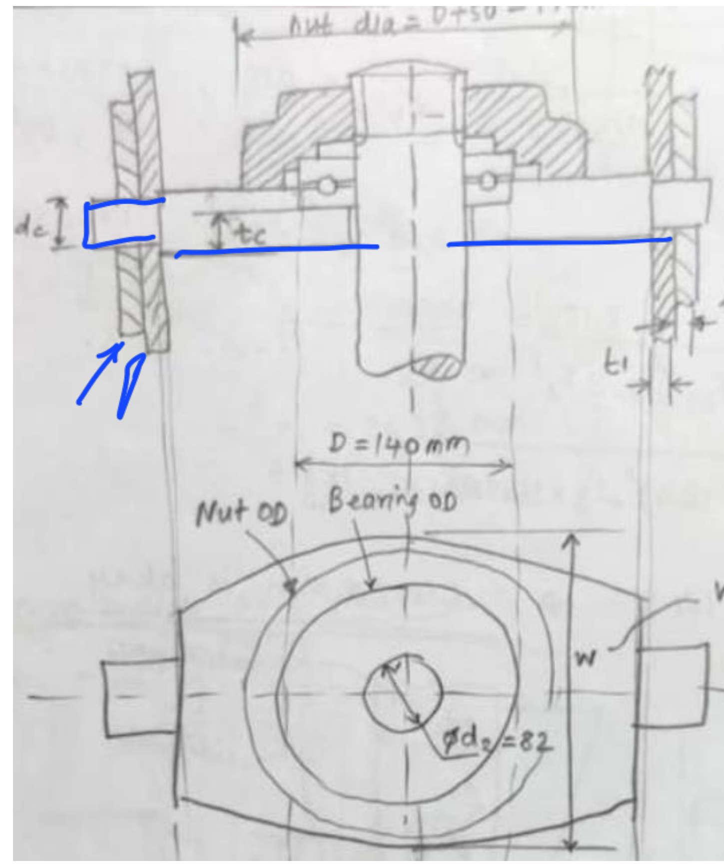
Cross Plate Design

The cross-plate ~~thickness~~ design is based on:

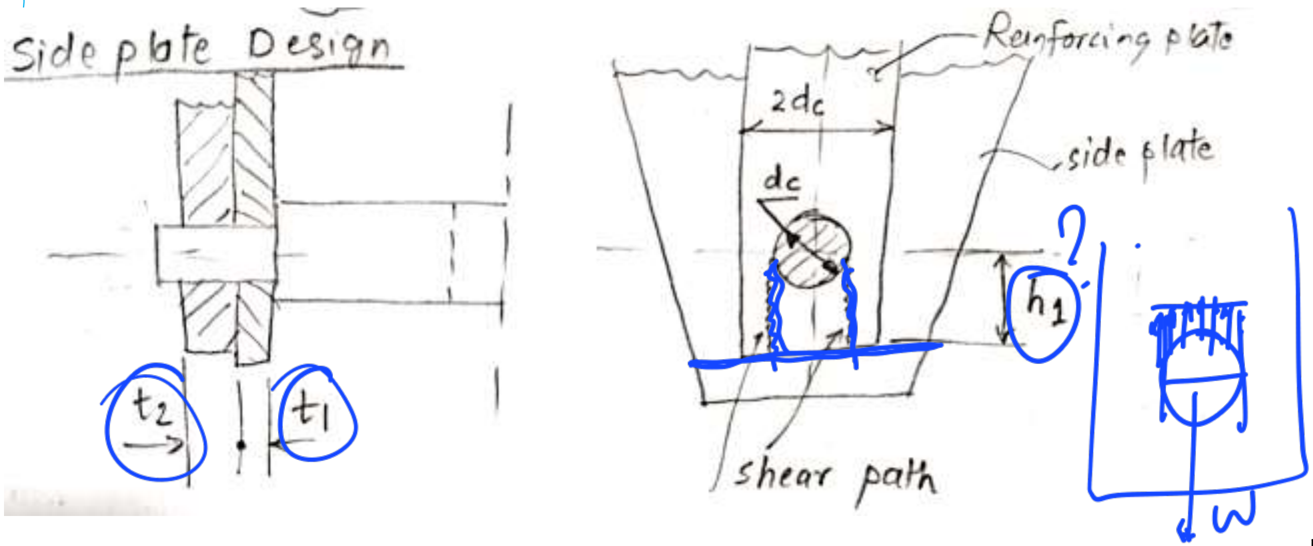
1) Bending stress

2) Shear Stress

3) Both of above



(D) Side plate design



Assumed

$t_2 = 2 t_1$

& $t_1 + t_2 = 60$ mm

$\Rightarrow t_1 = 20$ mm

$t_2 = 40$ mm

Let width of reinforcing plate $= 2 d_{tc} = 160$ mm

(i) Check for bearing pressure

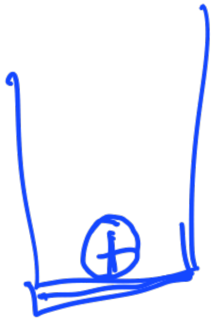
$$\sigma_{bearing} = \frac{(F_h/2)}{d_c \times (t_1 + t_2)} = \frac{159.5 \times 10^3 / 2}{80 \times 60} = 16.61 \ N/mm^2$$

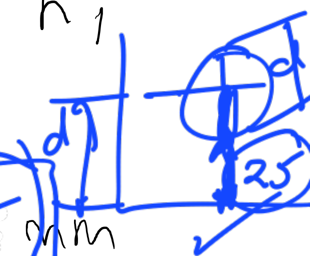
$$\text{Allowable bearing pressure} = \frac{UTS}{FOS} = \frac{600}{1.5} = 400 \ N/mm^2$$

$\because \sigma_{bearing} < \sigma_{allow} \Rightarrow$ safe against bearing.

(ii) Check for shearing of side plate + reinf plate

$$f_s = \frac{F_n/2}{h_1 \times (t_1+t_2)} = \frac{159.5\times10^3/2}{h_1 \times 60} = \frac{1329.2}{h_1}$$



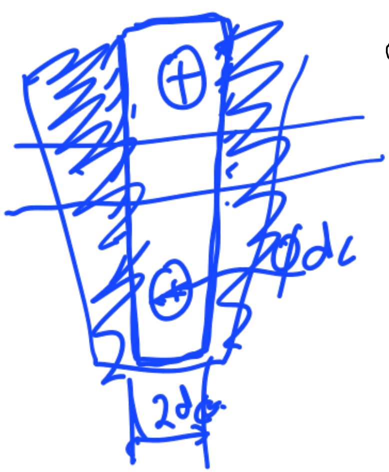


$\therefore \boxed{\sqrt{3} f_s < F_a}$

$\therefore \sqrt{3} \times \frac{1329.2}{h_1} = 121.5 \Rightarrow h_1 = \boxed{18.95}$ mm

h_1 selected $= \frac{d_{tc}}{2} + 25 = \boxed{65 \text{ mm}}$

(iii) Check for tensile failure



$$\sigma_t = \frac{F_n/2}{(2d_c - d_c)\times(t_1+t_2)} = \frac{159.5\times10^3/2}{80 \times 60}$$

160

$= 16.61$ N/mm²

$\therefore \boxed{1.25 f_t < F_a}$

$1.25 \times 16.61 \overset{?}{<} 121.5$

$20.76 < 121.5 \Rightarrow$ safe

QUIZ

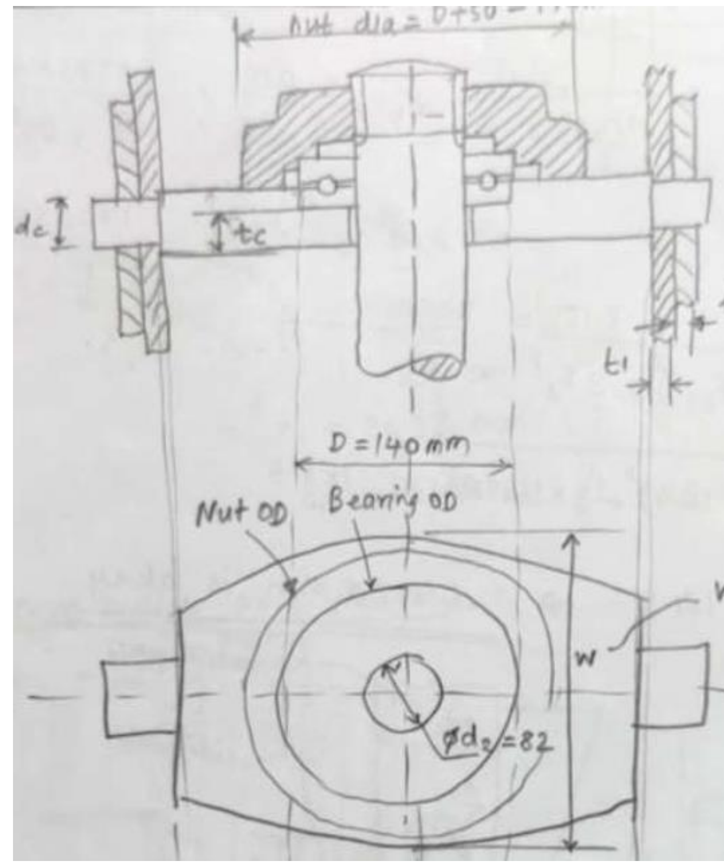
Slide Plate Design

The side-plate thickness is based on:

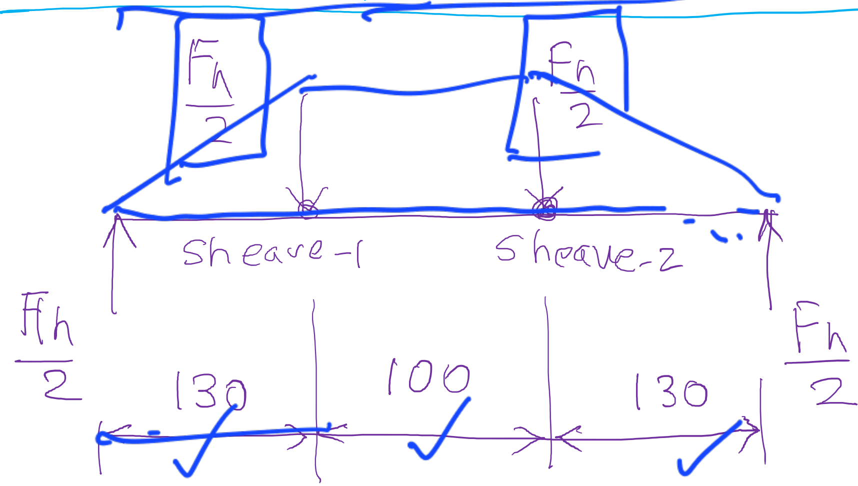
1) Bearing stress

2) Shear Stress + Tensile

3) Both of above ✓



(E) Sheave pulley axle design



$$BM_{max} = \frac{F_n}{2} \times 130$$

$$= \frac{159.5 \times 10^3}{2} \times 130$$

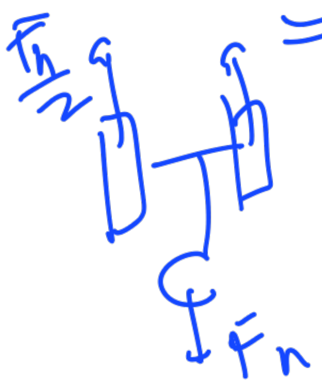
$$= 1.037 \times 10^7 \text{ N-mm}$$

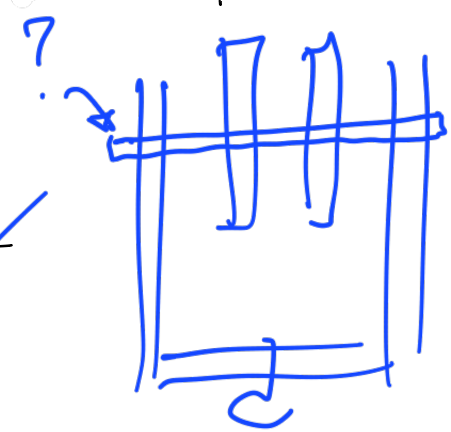
Let sheave pulley axle diameter = $\boxed{100\text{ mm}}$

$$\therefore \sigma_b = \frac{BM}{Z} = \frac{1.037 \times 10^7}{\frac{\pi}{32} \times 100^3} = 105.6 \text{ N/mm}^2$$

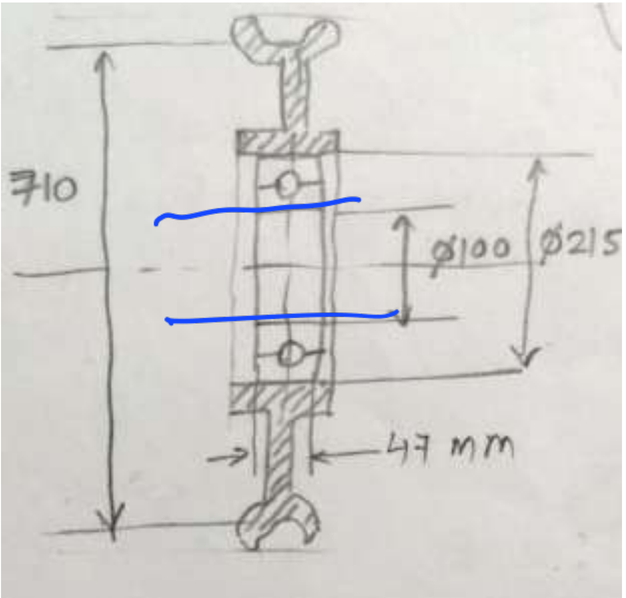
$$\because \boxed{f_{bt} < F_a} \Rightarrow \text{safe}$$

$f_{bt} = 105.6$, $F_a = 121.5$





(F) Pulley bearing selection



$F_r = \frac{F_n}{2} = \frac{159.5}{2} = 79.75$ kN

$F_a = 0$

Type of bearing: deep groove ball brg

$X = 1, \quad Y = 0$

$\therefore P = XVF_r + YF_a$

$V = 1.2$ (outer race rotating)

$\therefore P = 1 \times 1.2 \times 79.75 + 0 = 95.7$ kN

$\boxed{\text{Model selected : 6320}}$ (Life = 1 mr, 90% reliability)

$C = 174$ kN

ID = 100 mm

Trolley Assembly

shoe brakes

gear

l_1

Rope drum

m2

main motor

gear box

gear

motor

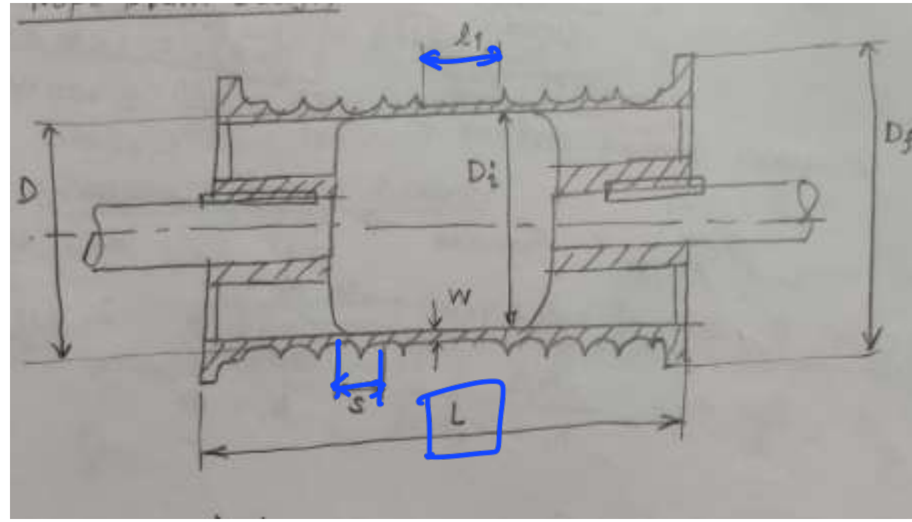
wheels

http://www.travellingoverheadcrane.com/sale-10022333-customized-double-girder-eot-crane-with-heavy-duty-open-winch-trolley-hoist.html

Rope drum design

Length of drum

$$L = \left(\frac{2H \cdot i}{\pi D} + 12\right) S + l_1$$



where,

H = height of load raise = 10,000 mm

i = ratio of pulley system = 2

D = drum diameter = 710 mm ($\boxed{D/d = 23}$) → $D = 28 \times 23 = 644$ mm

(std. rope drum dia. as per IS-3177,
= 200, 250, 315, 400, 500, 630, $\boxed{710}$, 800, 900, 1000, 1250 mm)

S = d + clearance between rope windings (PDB T27.18)
= 28 + 1.5 = 29.5 mm

l_1 = assumed $\boxed{4d}$ ≈ 120 mm

$$\therefore L = \left(\frac{2 \times 10{,}000 \times 2}{\pi \times 710} + 12\right) \times 29.5 + 120 = 1003 \approx \boxed{1010 \text{ mm}}$$

Wall thickness of rope drum - estimate (DDB T 27.16)

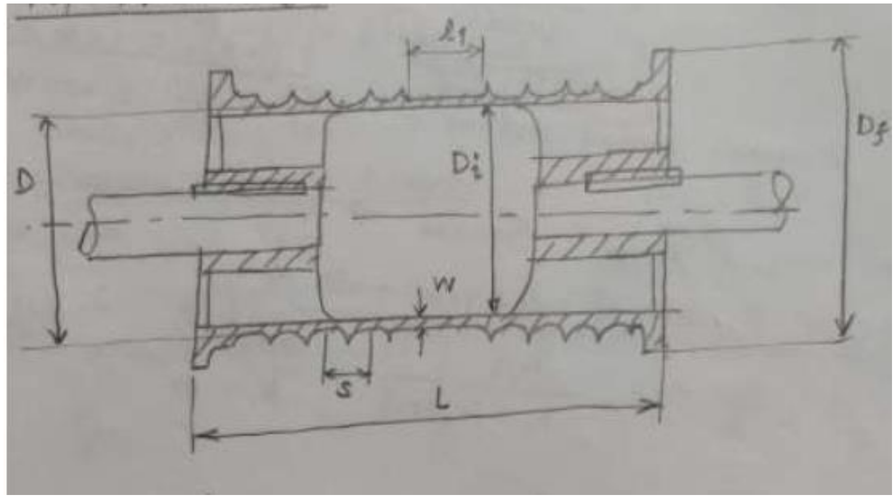
$$w = 0.02D + (6 \text{ to } 10\,mm)$$

$$= 0.02 \times 710 + 10$$

$$= 24.2 \approx \boxed{25\,mm}$$

QUIZ

Rope-Drum Diameter



The rope-drum diameter is based on rope diameter.

1) True

2) False

$\frac{D}{d} = 23$

Load on rope drum (F_{dr})

Weight of rope drum $= \frac{\pi}{4}(D_o^2 - D_i^2) \times L \times \rho$ mm

$D - 2w =$ mm

$= \frac{\pi}{4}(710^2 - 660^2) \times 1010 \times 7.85 \times 10^{-6}$

$= 427$ kg $= 4189$ N

Weight of rope $=$ Length $\times$ (wt. per meter)

$2.97 \frac{kg}{m}$ DDB T27.11

284 rop.

$= \frac{L - l_1}{S} \times \pi D \times \boxed{\text{wt. per meter}}$

meter

$= \frac{1010 - 120}{29.5} \times (\pi \times 0.71) \times 2.97$

$= 200$ kg $= 1962$ kN

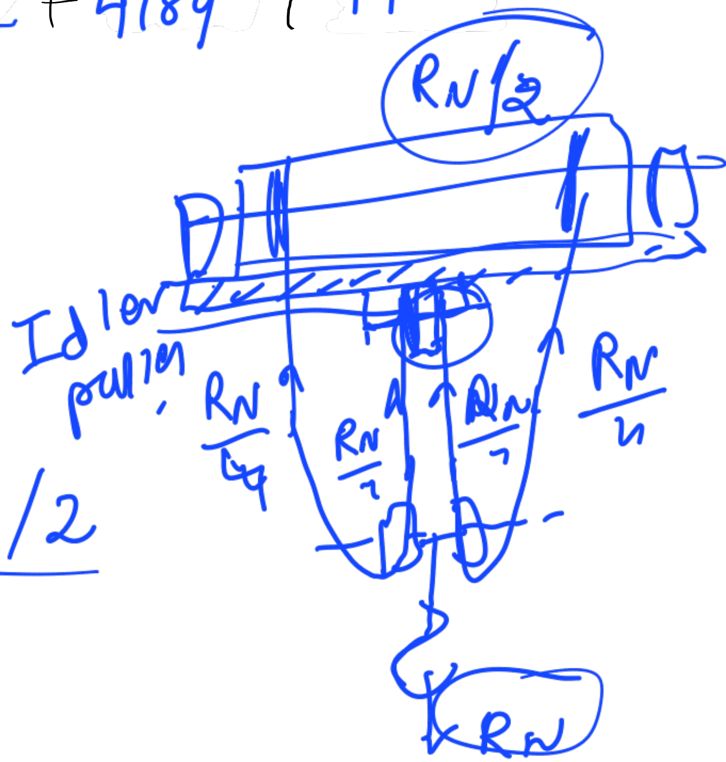
Load on rope drum, F_{dr}

$$= \boxed{\frac{R_N}{2}} + \left(\text{wt. of snatch block}\right) + \boxed{\text{Drum wt.}} + \boxed{\text{Rope wt.}}$$

assumed $\boxed{15\% \text{ of } R_N}$

$$= \frac{145 \times 10^3}{2} + 0.15 \times \left(145 \times 10^3\right) + 4189 + 1962$$

$$\boxed{F_{dr} = 100{,}401 \text{ N}}$$



Torque on rope drum (M_t)

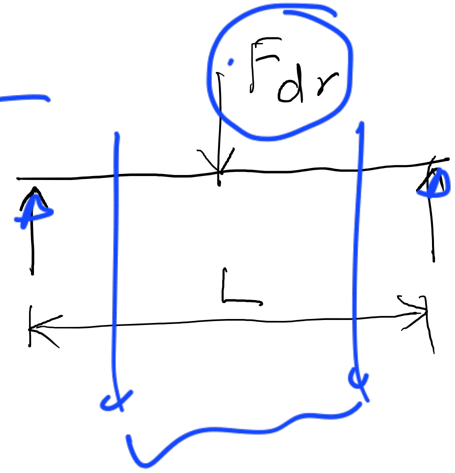
$$M_t = F_{dr}\,(D+d)/2$$

$$= 100{,}401 \times (710 + 28)/2$$

$$= 3.7048 \times 10^7 \text{ N-mm}$$

Bending stress (max) in rope drum

$$\sigma_b = \frac{\left(\frac{F_{dr} L}{4}\right) \times D/2}{\frac{\pi}{64}(D^4 - D_i^4)}$$



$$= \frac{8 F_{dr} L D}{\pi (D^4 - D_i^4)}$$

$$= \frac{8 \times 100401 \times 1010 \times 710}{\pi (710^4 - 660^4)}$$

$$= \underline{\underline{2.85 \ N/mm^2}}$$

Torsional shear stress in drum

$$\tau = \frac{16 M_t D}{\pi (D^4 - D_i^4)} = \frac{16 \times (3.7048 \times 10^7) \times 710}{\pi (710^4 - 660^4)} = \underline{\underline{2.08 \frac{N}{mm^2}}}$$

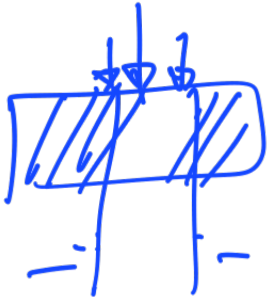
Crushing stress in rope groove of drum

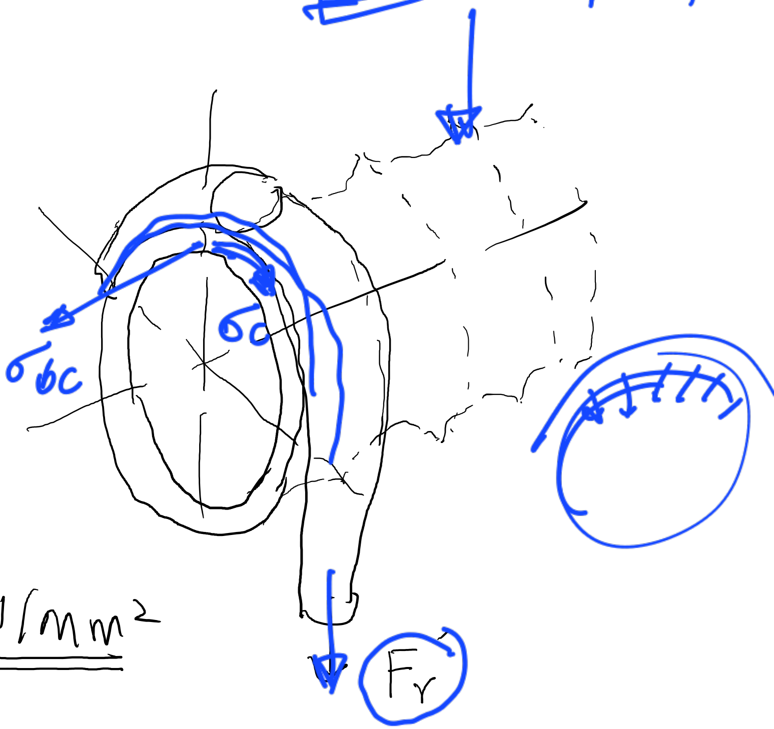
$$\sigma_c = \boxed{\frac{F_r'}{w\,S}}$$

$F_r' \rightarrow$ Tension in rope $\approx \boxed{\frac{F_{dr}}{2}}$

$$= \frac{(100{,}401/2)}{25 \times 29.5}$$

$$= 68.1 \text{ N/mm}^2$$





Total compressive normal stress

$$= \sqrt{\sigma_b^2 + \sigma_c^2}$$

$$= \sqrt{2.85^2 + 68.1^2} = 68.2 \text{ N/mm}^2$$

Combined (compression + bending + shear) stress

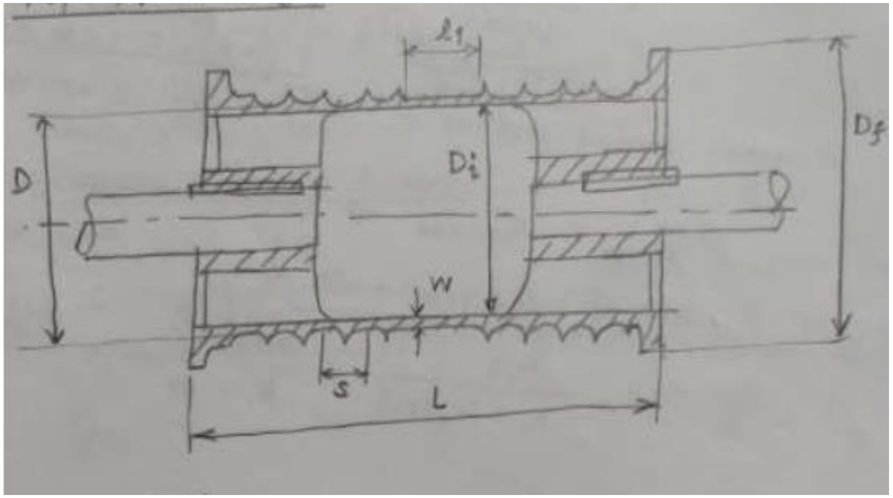
$$\sqrt{(f_c + f_{bc})^2 + 3 f_s^2} < F_a$$

$$\sqrt{(68.2^2 + 3 \times 2.08} \overset{?}{<} 121.5$$

$\Rightarrow 68.3 < 121.5 \Rightarrow$ Safe

QUIZ

Rope-Drum – Load



The fraction of the hook load acting on the rope-drum of 4-fall crane is _______.

1) 100%

2) 50% ✓

Motor for hoisting

Hoisting speed = 10 m/min

$\therefore$ Drum rpm $= \dfrac{\boxed{10\ m/min} \times 2}{\pi \times 0.710} = 8.97$ rpm,

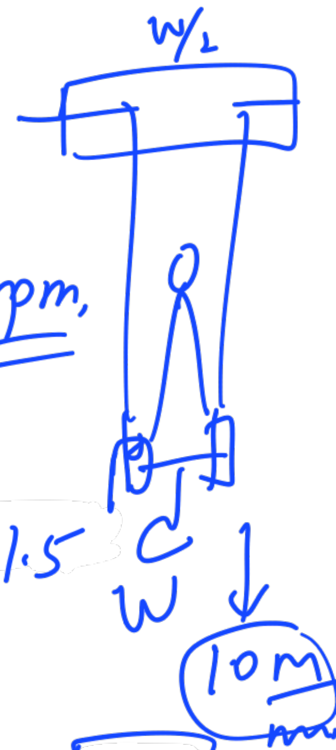
speed ratio

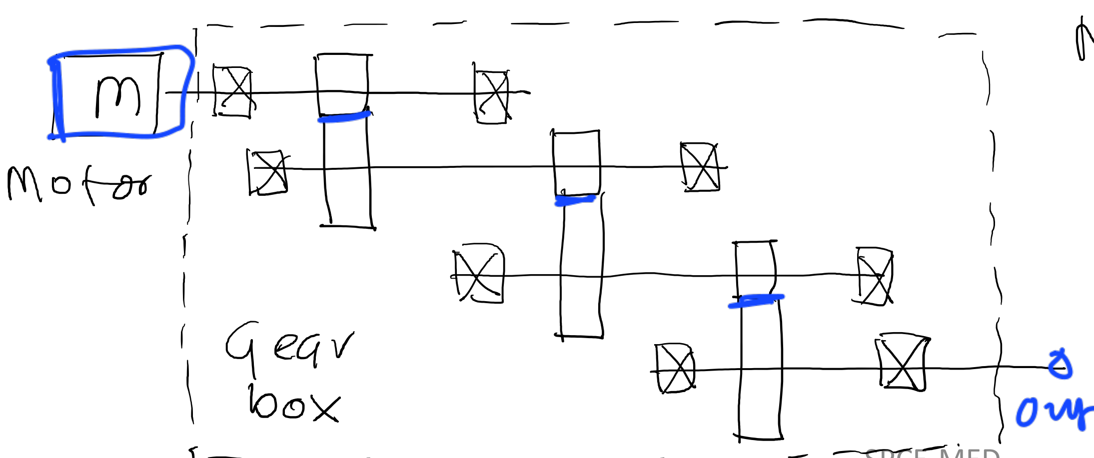
Let motor rpm = 1000

$\therefore$ overall speed ratio $= i_n = \dfrac{1000}{8.97} = 111.5$

We select 3 stage gear box

$\therefore$ Stage speed ratio $= \sqrt[n]{i_n} = \sqrt[3]{111.5} = \boxed{4.8}$





No. of bearings = 8

$\eta_{brg} = (0.99)^8 = 0.923$

$\eta_{gear\ pairs} = (0.99)^3 = 0.97$

$\eta_{pulley\ sys} = 0.94$

$$\text{Motor power} = \frac{\boxed{\text{Hoisting force}} \times \boxed{\text{Hoisting speed}}}{\underline{\eta_{pulley}} \times \underline{\eta_{gear}} \times \underline{\eta_{brg}}}$$

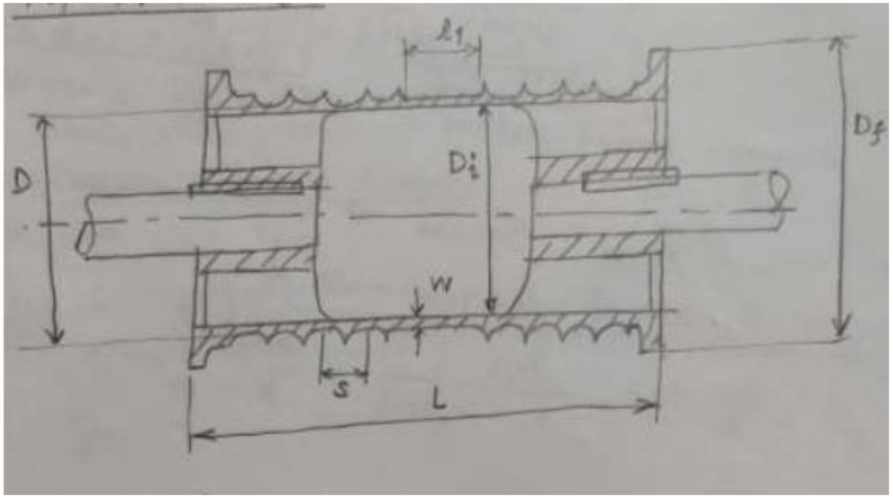
$$= \frac{159.5 \times 10^3 \times (10/60)}{0.94 \times 0.97 \times 0.923}$$

$$= \boxed{\underline{\underline{31.6 \ \text{kW}}}}$$

Select suitable standard motor.

QUIZ

Rope-Drum – Rope Velocity



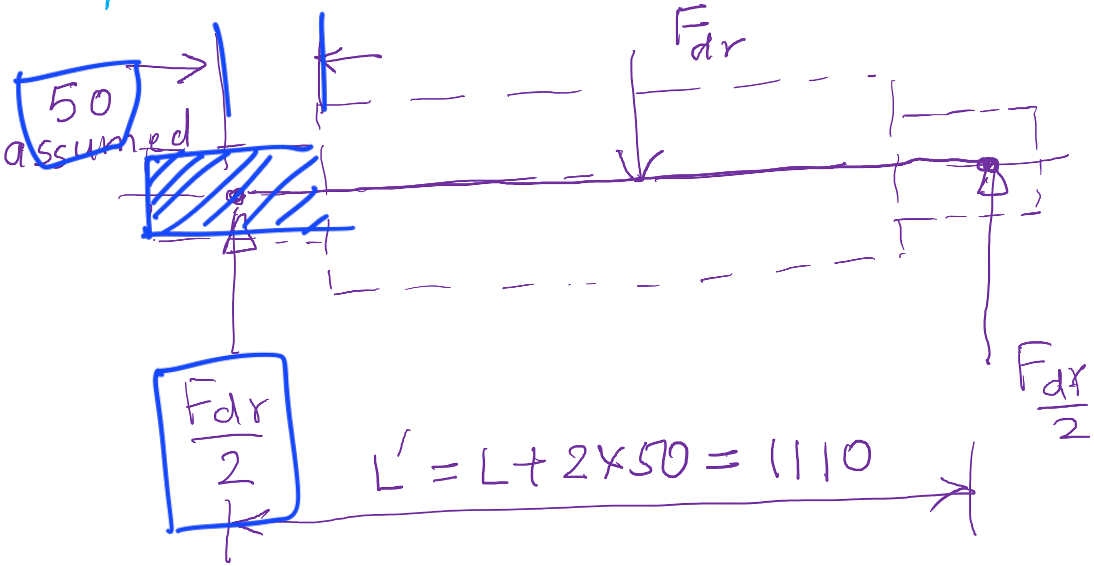
The velocity of rope at the rope drum is ______.

1) Same as hook velocity

2) Smaller than hook velocity

3) Larger than hook velocity ✓

Spindle shaft for rope drum



$$BM = \boxed{\frac{F_{dr}}{2}} \times \boxed{50}$$

$$= \frac{100401}{2} \times 50$$

$$= 2.51 \times 10^6 \text{ N-mm}$$

Let spindle shaft dia. = 150 mm

$$\therefore \sigma_b = \frac{BM}{Z} = \frac{2.51 \times 10^6}{\frac{\pi}{32} \times 150^3} = \underline{\underline{7.58 \text{ N/mm}^2}}$$

Torsional shear stress

$$\tau = \frac{M_t \, r}{J} = \frac{(3.7048 \times 10^7) \times (150/2)}{\left(\frac{\pi}{32} \times 150^4\right)} = \underline{\underline{55.9 \text{ N/mm}^2}}$$

Combined stress check

$$\boxed{\sqrt{(1.25 f_t + f_{bt})^2 + 3\tau^2} < F_a}$$

$$\therefore \sqrt{(1.25 \times 0 + 7.58)^2 + 3 \times 55.9^2} \overset{?}{<} 121.5$$

$$\therefore 97.1 < 121.5 \Rightarrow \text{Safe}$$

Bearing for rope drum

Model selected = 6030, $C = 125$ kN

$d = 150$ mm

Wheel on Rail

frame

Bumpe

Wheels

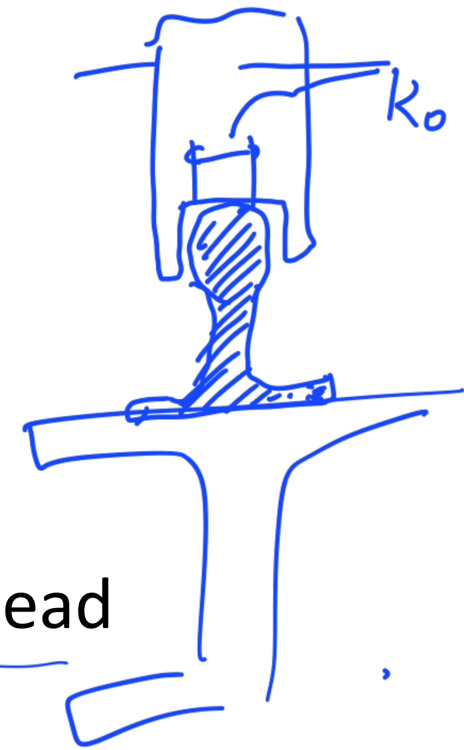
https://www.indiamart.com/proddetail/wheel-assembly-for-long-travel-6649576748.html

Wheel Design

Contact Pressure, $$\boxed{p = \frac{P}{C_1 C_2 D K_0}}$$

- P = wheel load
- C_1 = speed factor (IS-3177)
- C_2 = Life factor
- D = wheel diameter
- $K_0 = K - 2r$ = useful width of rail head

?



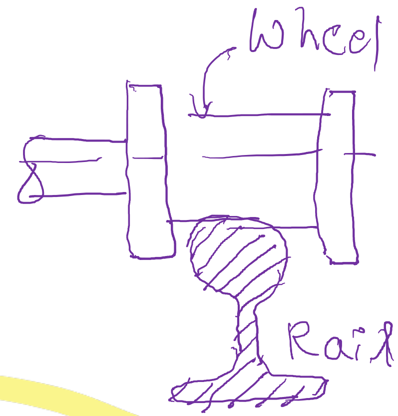
Rotational Speed rpm	Speed factor C_1
200	0.66
160	0.72
125	0.77
112	0.79
100	0.82
90	0.84
80	0.87
71	0.89
63	0.91
56	0.92
50	0.94
45	0.96
40	0.97
35.5	0.99
31.5	1.00
28.0	1.02
25.0	1.03
22.4	1.04
20.0	1.06
18.0	1.07
16.0	1.09

Relative operating period of travel drive, %	C_2
Upto 16	1.25
16 to 25	1.12
25 to 40	1.00
40 to 63	0.90
Over 63	0.80

Design of wheels for trolley

Trolley speed $= V_t = 25$ m/min (given)

(a) Estimation of wheel diameter



Contact pressure $= p = \dfrac{\boxed{P}}{C_1 C_2 D K_0} \leq p_0$

P = wheel load

$= \left[R_N + \boxed{\text{Snatch block wt}} + \boxed{\text{Drum wt}} + \boxed{\text{Rope wt}} + \boxed{\text{Trolley wt}}\right] / 4$

$= [145 \times 10^3 + 0.15 \times (145 \times 10^3) + 4189 + 1962 + 0.1 \times (145 \times 10^3)] / 4$

assumed 15%   assumed 10%

$= 4775.7$ kgf

46850 N

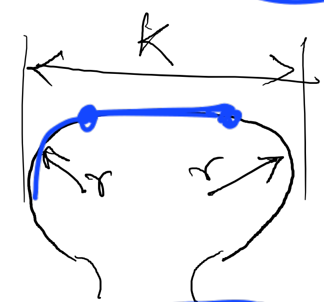
C_1 = speed factor = 0.94 (assumed wheel $\boxed{\text{rpm} = 50}$)

C_2 = Life factor = 0.8 (assumed op. period > 63%)

K_o = useful width of rail head

$= \boxed{k - 2r}$

$= 5.5 - 2 \times 0.5 = 4.5$ ~~mm~~ cm



IS - 3443

32 kg/m rail section

$k = 55$ mm, $r = 5$ mm

P_o = allowable contact pressure

$= \cancel{6.5\ N/mm^2}$ $65\ \frac{kgf}{cm^2}$ (for hardened steel)

$$\cancel{6.5}\ \boxed{65} = \frac{4775.7\ \text{kgf}}{0.94 \times 0.8 \times D \times 4.5\ \text{cm}}$$

$\therefore D = 21.7$ ~~mm~~ cm $\approx \boxed{250\ \text{mm}}$ ← selected standard size

Check rpm of wheel

$$\text{rpm} = \frac{V_{tr}}{\pi \times D_{wm}} = \frac{25}{\pi \times 0.25} = 31.8\ \text{rpm} \Rightarrow C_1 = 1.0$$

$$\Rightarrow p = \frac{4775.5}{1.0 \times 0.8 \times 25 \times 4.5} = 51\ \cancel{N/mm^2}\ kgf/cm^2 < \cancel{6.5}\ 65\ kgf/cm^2 \Rightarrow OK$$

Motor selection for trolley

Resistance to motion = (wheel load × no. of wheels) × μ

$$W = (46850 \times 4) \times 0.02$$

assumed for ball brg

$$= 3748\ N$$

Power for trolley motor

$$kW = \frac{W \times V_{tr}}{\eta} \times \frac{1}{1000}$$

$$= \frac{3748 \times (25/60)}{0.9} \times \frac{1}{1000} = \boxed{1.73\ kW}$$

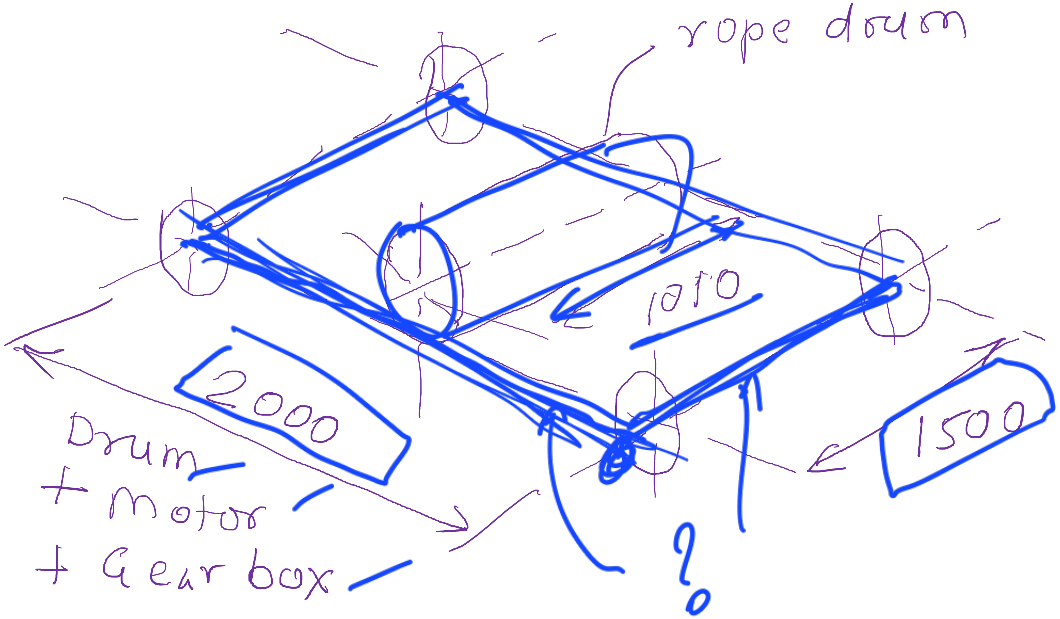
0.9 assumed

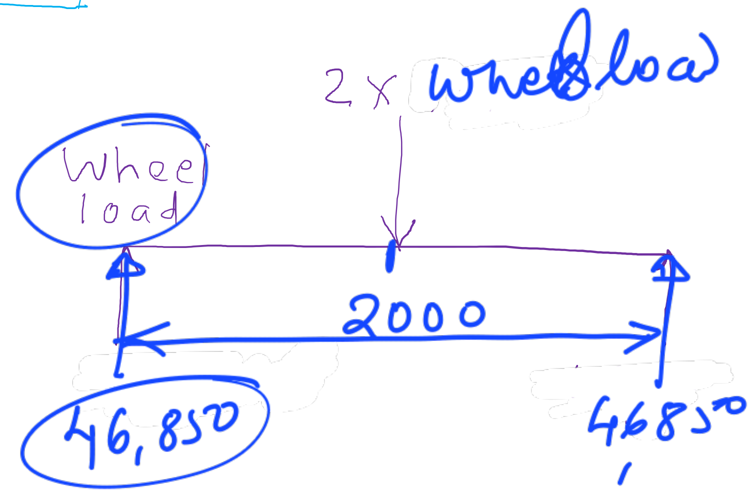
Gear box for trolley

Assuming motor rpm = 1000,

speed ratio = $\frac{1000}{31.8}$ = 31.5 ⇒ 2-stage GB with stage speed ratio = $\sqrt{31.5}$ = 5.6

Structural members for trolley





$$BM = 46{,}850 \times 1000$$
$$= 4.685 \times 10^7 \text{ N mm}$$

$$\sigma_b = \frac{BM}{Z}$$

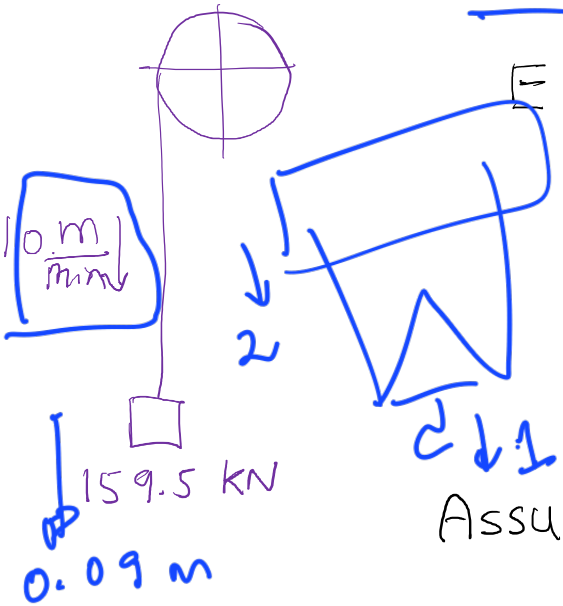
$$\therefore \quad = \frac{4.685 \times 10^7}{Z}$$

$$\Rightarrow Z = 3.856 \times 10^5 \text{ mm}^3 = 385.6 \text{ cm}^3$$

From structural steel section tables,

For ISHB 300, $Z = 4.317 \times 10^5 \text{ mm}^3$ ✓

Hoisting brake design



Energy absorbed by brake

$$E = \text{initial KE} + \text{rotary KE} + \text{PE loss}$$

$$= \frac{1}{2} \times \frac{159.5 \times 10^3}{9.81} \times \left(\frac{10}{60}\right)^2 + \underset{\text{assumed}}{0} + \frac{159.5 \times 10^3}{9.81} \times (9.81) \times \underset{\text{given}}{0.09}$$

$$= \underline{\underline{14,580.8 \text{ J}}}$$

Assumed brake drum diameter = 500 mm

Let θ = angle through which drum rotates during braking

$$\therefore \; \theta \times \text{drum radius} = \boxed{0.09} \times \boxed{2}$$

← speed ratio

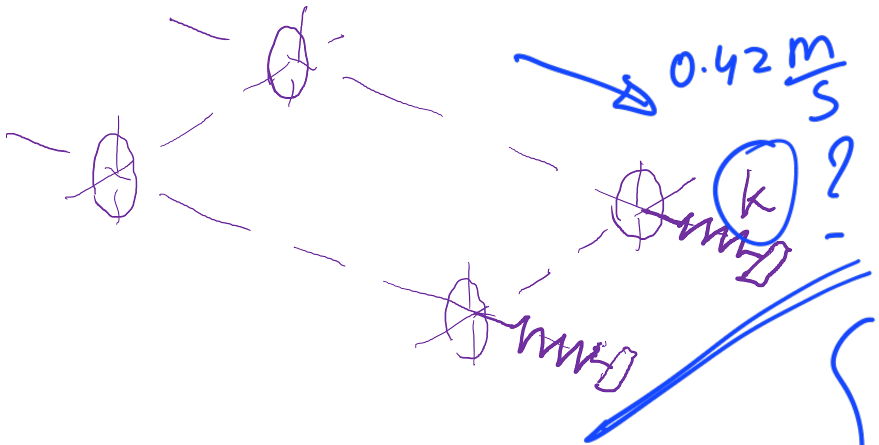
stopping dist

$$\therefore \; \theta \times \left(\frac{0.5}{2}\right) = 0.09 \times 2 \Rightarrow \theta = \underline{\underline{0.72 \text{ rad}}}$$

Torque capacity of brake

$$M_t = \frac{E}{\theta} = \frac{14,580.8 \text{ J}}{0.72} = \boxed{20,251.1 \; \cancel{\text{N-mm}} \; \text{N-m}}$$

Bumpers for trolley



Let number of bumpers = 2

stiffness of bumpers = k (N/m)

Trolley speed = 25 $\frac{m}{min}$ = 0.42 $\frac{m}{s}$

From IS 3177, 8.11.2,

Trolley should be brought to rest from 50% of rated speed with deacceleration not exceeding 5 m/s²

$$v^2 - u^2 = 2as$$

$$\Rightarrow \quad 0^2 - (0.42)^2 = 2 \times 5 \times S$$

$$\Rightarrow \quad S = 4.41 \times 10^{-3} \text{ m}$$

Energy to be absorbed by bumper = Energy absorbed in springs

$\therefore \boxed{\frac{1}{2} m v^2} = \left(\frac{1}{2} k s^2\right) \times 2$ ← no. of bumpers

m = mass of trolley + drum + rope

$= \frac{0.1 \times 145 \times 10^3 + 4189 + 2105.1}{9.81} = 2105.1$ kg ← assumed

$\therefore \frac{1}{2} \times 2105.1 \times (0.42)^2 = \frac{1}{2} \times k \times (4.41 \times 10^{-3})^2 \times 2$

$\therefore \boxed{k = 9.55 \times 10^6 \ N/m}$

$\boxed{k \cdot s = F}$