Research Paper

DESIGN AND SELECTING THE PROPER CONVEYOR-BELT

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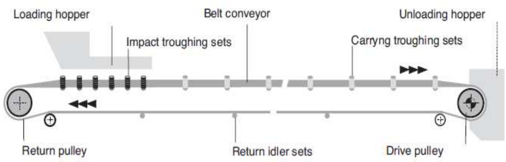
ABSTRACT:

Belt conveyor is the transportation of material from one location to another. Belt conveyor has high load carrying capacity, large length of conveying path, simple design, easy maintenance and high reliability of operation. Belt Conveyor system is also used in material transport in foundry shop like supply and distribution of molding sand, molds and removal of waste. This paper provides to design the conveyor system used for which includes belt speed, belt width, motor selection, belt specification, shaft diameter, pulley, gear box selection, with the help of standard model calculation.

1. INTRODUCTION:

During the project design stage for the transport of raw materials or finished products, the choice of the method must favor the most cost effective solution for the volume of material moved; the plant and its maintenance; its flexibility for adaptation and its ability to carry a variety of loads and even be overloaded at times.

Basic drawing of a belt conveyor



2. THE PARAMETERS FOR DESIGN OF BELT CONVEYOR:

- Belt speed
- Belt width
- Absorbed power
- Gear box selection
- Drive pulley shaft

For designing a conveyor belt, some basic information e.g. the material to be conveyed, its lump size, tonnage per hour, distance over which it is to be carried, incline if any, temperature and other environmental conditions is needed.

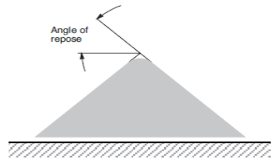
4. DESIGN CALCULATIONS OF CONVEYOR

INPUT DATA

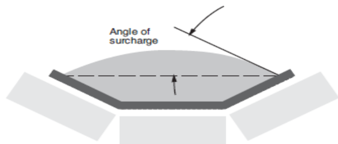
Bulk density (ρ) – 1.7 T/m^3
Size of lump – 0-10 mm
Belt width (B) – 1850 mm
Capacity (C) – 800 - 900 TPH
Lift of the material (H) – 5.112 m
Length between centers (L) – 29m
Belt speed (V) – 1.2 m/s
Troughing angle (λ) – 35^0
Conveyor Inclination – 10.36^0
Take Up Travel – 600 mm
Type of Take up – SCREW

5. DESIGN OF BELT CONVEYOR

The design of the belt conveyor must begin with an evaluation of the characteristics of the conveyed material and in particular the angle of repose and the angle of surcharge. The angle of repose of a material, also known as the “angle of natural friction” is the angle at which the material, when heaped freely onto a horizontal surface takes up to the horizontal plane.



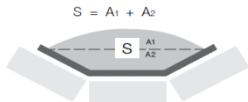
Angle of repose



Angle of surcharge

Angle of surcharge β:

The area of the section “S” may be calculated geometrically adding the area of a circle A1 to that of the trapezoid A2.

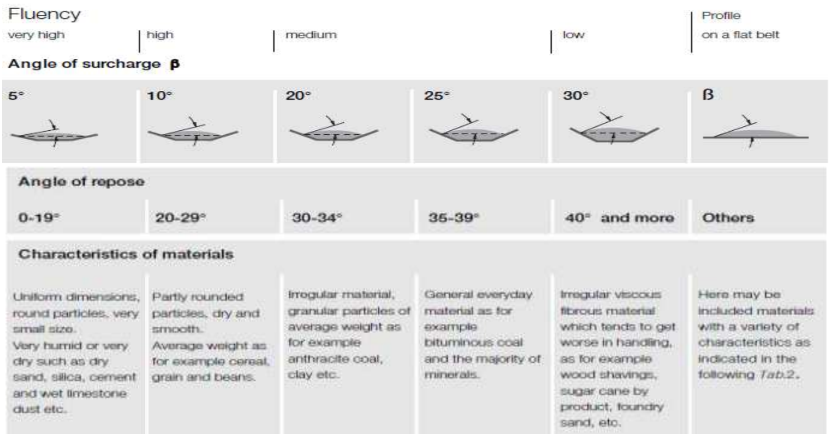


The value of the conveyed volume 1VT may be easily calculated using the formula :

$$S = \frac{I_{VT}}{3600} \; [m^2]$$

where :
IVT = conveyed volume at a conveyor speed of 1 m/s

Angles of surcharge, repose, and material fluency:



Fluency: very high	high	medium		low	Profile on a flat belt
Angle of surcharge β					
5°	10°	20°	25°	30°	β
Angle of repose					
0-19°	20-29°	30-34°	35-39°	40° and more	Others
Characteristics of materials					
Uniform dimensions, round particles, very small size. Very humid or very dry such as dry sand, silica, cement and wet limestone dust etc.	Partly rounded particles, dry and smooth. Average weight as for example cereal, grain and beans.	Irregular material, granular particles of average weight as example anthracite coal, clay etc.	General everyday material as for example bituminous coal and the majority of minerals.	Irregular viscous fibrous material which tends to get worse in handling, as for example wood shavings, sugar cane by product, foundry sand, etc.	Here may be included materials with a variety of characteristics as indicated in the following *Tab.2*.

5.1. Belt speed:

Very high speeds have meant a large increase in the volumes conveyed. Compared with the load in total there is a reduction in the weight of conveyed material per linear meter of conveyor and therefore there is a reduction in the costs of the structure in the troughing set frames and in the belt itself. The physical characteristics of the conveyed material are the determining factor in calculating the belt speed.

With the increase of material lump size, or its abrasiveness, or that of its specific weight, it is necessary to reduce the conveyor belt speed.

The quantity of material per linear meter loaded on the conveyor is given by the formula:

$$q_G = \frac{I_V}{3.6 \times v} \quad [\mathrm{Kg/m}]$$

where:

qG = weight of material per linear meter

Iv= belt load t/h

v = belt speed m/s

qG is used in determining the tangential force Fu.

5.1.1. Maximum speeds advised:

Lumpsize max. dimensions uniform up to mm	mixed up to mm	Belt min.width mm	max.speed A	B	C	D
50	100	400	2.5	2.3	2	1.65
75	150	500				
125	200	650	3	2.75	2.38	2
170	300	800	3.5	3.2	2.75	2.35
250	400	1000	4	3.65	3.15	2.65
350	500	1200				
400	600	1400	4.5	4	3.5	3
450	650	1600				
500	700	1800	5	4.5	3.5	3
550	750	2000				
600	800	2200	6	5	4.5	4

Considering the factors that limit the maximum conveyor speed we may conclude:

When one considers the inclination of the belt leaving the load point; the greater the inclination, the increase in the amount of turbulence as the material rotates on the belt. This phenomenon is a limiting factor in calculating the maximum belt speed in that its effect is to prematurely wear out the belt surface.

The repeated action of abrasion on the belt material, given by numerous loadings onto a particular section of the belt under the load hopper, is directly proportional to the belt speed and inversely proportional to its length.

5.2. Belt width:

The optimum belt speed, the determination of the belt width is largely a function of the quantity of conveyed material which is indicated by the design of conveyed belt.

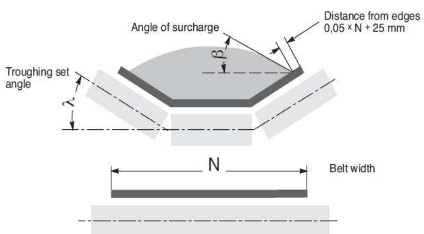
In practice the choice and design of a troughing set is that which meets the required loaded volume, using a belt of minimum width and therefore the most economic.

5.2.1. Calculation of Belt width:

In the following section, the conveyor capacity may be expressed as loaded volume IVT [m3/h] per v= 1 m/sec.

The inclination of the side rollers of a transom (from 20° to 45°) defines the **angle of the troughing.**

Troughing sets at **40° / 45**° are used in special cases, where because of this onerous position the belts must be able to adapt to such an accentuated trough.



All things being equal the width of the belt at the greatest angle corresponds to an increase in the loaded volume IVT. The design of the loaded troughing set is decided also as a function of the capacity of the belt acting as a trough.

In the past the inclination of the side rollers of a troughing set has been 20° . Today the improvements in the structure and materials in the manufacture of conveyor belts allows the use of troughing sets with side rollers inclined at 30° / 35°.

It may be observed however that the belt width must be sufficient to accept and contain the loading of material onto the belt whether it is of mixed large lump size or fine material.

In the calculation of belt dimensions one must take into account the minimum values of belt width as a function of the belt breaking load and the side roller inclination as shown.

5.2.2Minimum belt width:

Breaking load N/mm	Belt width λ= 20/25° mm	λ= 30/35°	λ= 45°
250	400		
315	400	400	450
400	400	400	450
500	450	450	500
630	500	500	600
800	500	600	650
1000	600	650	800
1250	600	800	1000
1600	600	800	1000

5.2.3.Loaded volume IM:

The volumetric load on the belt is given by the formula:

$$I_M = \frac{I_V}{q_S} \quad [\mathrm{m^3/h}]$$

where:

Iv = load capacity of the belt [t/h]

qs = specific weight of the material

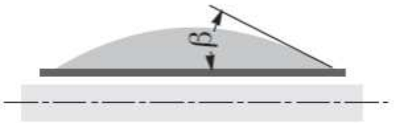
Also defined as:

$$I_{VT} = \frac{I_M}{v} \quad [\mathrm{m^3/h}]$$

Where the loaded volume is expressed relevant to the speed of 1 mtr/sec.

It may be determined from *Tab.* **5**a-b-c-d, that the chosen belt width satisfies the required loaded volume IM as calculated from the project data, in relation to the design of the troughing sets, the roller inclination, the angle of material surcharge and to belt speed.

Loaded volume
with flat roller sets **v = 1 m/s**



One flat roller sets
Loaded volume with flat roller sets v = 1 m/s

Belt width mm	Angle of surcharge β	IVT m³/h λ = 0°
300	5°	3.6
	10°	7.5
	20°	15.4
	25°	20.1
	30°	25.2
400	5°	7.5
	10°	15.1
	20°	31.3
	25°	39.9
	30°	50.0
500	5°	12.6
	10°	25.2
	20°	52.2
	25°	66.6
	30°	83.5
650	5°	22.3
	10°	45.0
	20°	93.2
	25°	119.5
	30°	149.4
800	5°	35.2
	10°	70.9
	20°	146.5
	25°	187.5
	30°	198.3
1000	5°	56.8
	10°	114.4
	20°	235.8
	25°	301.6
	30°	377.2

Loaded volume
with 2 roll troughing sets **v = 1 m/s**

With two roller sets
Loaded with 2 roll troughing sets v = 1 m/s

Belt width mm	Angle of surcharge β	IVT m³/h λ = 20°
300	5°	17.6
	10°	20.5
	20°	28.8
	25°	32.0
	30°	36.3
400	5°	34.5
	10°	41.4
	20°	55.8
	25°	63.7
	30°	72.0
500	5°	57.6
	10°	68.7
	20°	92.8
	25°	105.8
	30°	119.8
650	5°	102.9
	10°	123.1
	20°	165.9
	25°	189.3
	30°	214.5
800	5°	175.6
	10°	192.9
	20°	260.2
	25°	296.6
	30°	336.2
1000	5°	317.1
	10°	310.6
	20°	418.6
	25°	477.3
	30°	541.0

Loaded volume
with 3 roll troughing sets **v = 1 m/s**

with 3 roll troughing sets

Belt width mm	Angle of surcharge β	IVT m³/h λ = 20°	λ = 25°	λ = 30°	λ = 35°	λ = 45°
300	5°	13.3	15.1	17.2	18.7	21.6
	10°	16.9	18.7	20.5	21.6	24.4
	20°	24.4	26.2	27.7	28.8	30.6
	25°	27.7	30.2	31.6	32.4	33.8
	30°	33.4	34.9	36.0	36.3	37.8
400	5°	28.0	32.4	36.6	39.6	45.7
	10°	35.2	29.2	43.2	45.3	51.4
	20°	50.4	54.3	57.2	59.4	66.3
	25°	56.8	62.2	65.1	66.6	69.8
	30°	67.7	70.9	73.4	74.5	77.0
500	5°	47.8	55.8	62.6	68.0	78.4
	10°	60.1	67.3	73.4	78.4	87.4
	20°	85.3	91.8	97.2	101.1	106.9
	25°	96.1	104.7	109.8	112.6	117.7
	30°	114.1	119.1	123.8	126.0	129.6
650	5°	87.8	101.8	114.4	124.9	143.2
	10°	109.4	122.4	134.2	142.9	159.1
	20°	154.4	166.3	176.4	183.6	193.6
	25°	174.2	189.7	198.7	204.4	212.4
	30°	205.5	215.2	223.5	227.8	233.6
800	5°	139.6	162.0	182.1	198.3	227.1
	10°	173.6	194.4	212.7	226.8	252.0
	20°	244.0	262.8	278.2	290.1	306.0
	25°	275.0	299.1	313.2	322.9	334.8
	30°	324.0	339.4	352.4	359.2	367.9
1000	5°	227.1	263.8	296.2	322.9	368.6
	10°	281.1	315.3	345.6	368.6	408.6
	20°	394.9	425.5	450.7	469.8	494.6
	25°	444.9	483.8	506.5	522.0	541.0
	30°	523.4	548.6	569.1	580.6	594.0

Loaded volume
with 5 roll troughing sets **v = 1 m/s**

with 5 roll troughing sets

Belt width mm	Angle of surcharge β	IVT m³/h λ1 30° λ2 60°
800	5°	236.5
	10°	260.2
	20°	313.9
	25°	342.0
	30°	372.9
1000	5°	388.8
	10°	427.3
	20°	510.4
	25°	556.2
	30°	606.2
1200	5°	573.1
	10°	630.0
	20°	751.3
	25°	816.6
	30°	892.4
1400	5°	797.4
	10°	876.6
	20°	1041.4
	25°	1135.0
	30°	1237.3
1600	5°	1075.3
	10°	1181.8
	20°	1371.9
	25°	1495.0
	30°	1629.7
1800	5°	1343.1
	10°	1476.0
	20°	1749.6
	25°	1906.9
	30°	2078.6

5.2.4. Corrects loaded volume in relation to the factors of inclination and feed:

In the case of inclined belts, the values of loaded volume IVT [m3/h] are corrected according to the following:

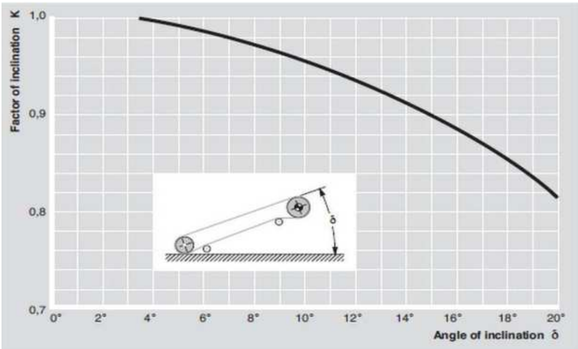
IVM = IVT X K X K1 [m3/h]

Where:
IVM is the loaded volume corrected in relation to the inclination and the irregularity of feeding the conveyor in m3/h with v = 1 m/s.
IVT is the theoretic load in volume for v= 1m/s.
K is the factor of inclination.
K1 is the correction factor given bythe feed irregularity.

The inclination factor K calculated in the design, must take into account the reduction in section for the conveyed material when it is on the incline.

Diagram gives the factor K in function of the angle of conveyor inclination, but only for smooth belts that are flat with no profile.

Factor of inclination K:



In general it is necessary to take into account the nature of the feed to the conveyor, whether it is constant and regular, by introducing a correction factor K1 its value being :

- K1 = 1 regular feed
- K1 = 0.95 irregular feed
- K1 = 0.90 ÷ 0.80 most irregular feed.

If one considers that the load may be corrected by the above factors the effective loaded volume at the required speed is given by:

IM = IVM x v [m3/h]

Given the belt width, one may verify the relationship between the belt width and the maximum lump size of material according to the following:

belt width ≥ max. lump size

5.3. ABSORBED POWER:

P_A = Absorbed power i.e., power required for drive pulley after taking drive pulleys loss into account.

$$= P_{DP} + \frac{(R_{wd} + R_{bd})\, V}{1000} \text{ kW}$$

Where,

R_{wd} = Wrap resistance for drive pulley (230 N)

R_{bd} = Pulley bearing resistance for drive pulley (100 N)

Therefore,

$$P_A = 23.028 + \frac{(230+100)\; x\; 1.2}{1000}$$

$$= 23.028 + 0.396 = 23.424 \text{ kW}$$

5.4. MOTOR POWER

The motor output power (shaft) is given by

$$P_M = \frac{P_A}{\eta}$$

Where,

η = Overall efficiency by taking the power losses of gear-box and couplings into account.= 0.94

Therefore,

$$P_M = \frac{23.424}{0.9}$$

$$= 24.49 \text{ kW}$$

5.5 .MOTOR SELECTION

At present, all the motors are of 1500 rpm.

By referring the catalogue, the selected motor is of 37 kW/1500 rpm (Nominal power).

The shaft diameter of the motor is 60mm.

5.6. GEAR BOX SELECTION:

For gear box selection, we need to calculate the reduction ratio.

$$\text{Reduction ratio} = \frac{Input\ rpm}{Output\ rpm}$$

As the motor is of 1500 rpm,

Input rpm = 1500 rpm

The output rpm is calculated using the formula,

$$V = \frac{\pi\ x\ D\ x\ N}{60}$$

Where,

D = Diameter of driving pulley

= 630 + 12 + 12 = 654 mm [According to IS, 630 mm diameter of driving pulley is suitable for the motor of power which is less than 50 kW & 24 mm (12 + 12) extra diameter is provided due to lagging of the pulley]

$$\Rightarrow 1.2 = \frac{\pi\ x\ 0.654\ x\ N}{60}$$

$$\Rightarrow N = \frac{60\ x 1.2}{\pi\ x\ 0.654} = 35.043\text{rpm}$$

5.7. DESIGN OF SHAFTS:

5.7.1. DRIVE PULLEY SHAFT DESIGN

Considering all the resistances (including wrap & bearing resistances), we get the torque from the following formula

$$P_A = \frac{2\pi NT}{60}$$

$$\Rightarrow T = \frac{60\ x\ P_A}{2\ x\ \pi\ x\ N}$$

$$= \frac{60\ x 23.424}{2\ x\ \pi\ x\ 35.043} \text{ [N=35.043 rpm at output]}$$

= 6383 N-m

= 6.383 KN-m

Tension in the belt = T / r

=6.383/0.327

=19.51 KN

Now, consider

T_E = 19.51 KN

We have,

T_1 = Carrying side belt tension

$$T_1 = T_E \left[\frac{\xi}{e^{\mu\theta} - 1} + 1\right]$$

Where,

ξ = Drive coefficient = 1.66

θ = Angle of wrap= 210^0 = 210 x $\frac{\pi}{180}$ = 3.66 rad

μ = Coefficient of friction between drive pulley and belt = 0.35

$$\Rightarrow T_1 = 19.51 \left[\frac{1.6}{e^{0.35\ x\ 3.66} - 1} + 1\right] = 31.01 \text{ KN}$$

T_2 = Return side belt tension

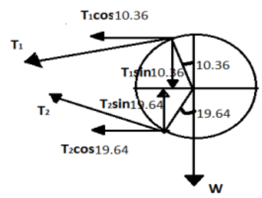
$T_2 = T_1 - T_E$

= 31.01 – 19.198

= 11.812 KN

W = weight of the drive pulley

= 750 Kgs = 7357.5 N = 7.357 KN



Resolving horizontal and vertical components

$F_H = T_1 \cos 10.36^0 + T_2 \cos 19.64$

=30.504+11.124

= 41.6285 KN

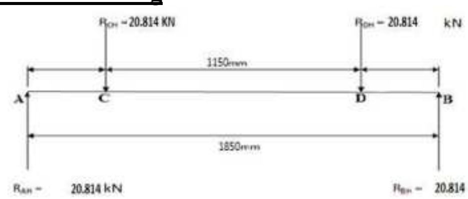
$F_V = T_1 \sin 10.36^0 - T_2 \sin 19.64^0 + W$

= $(31.01 \sin 10.36^0) - (11.812 \sin 19.64^0) + 7.3$

= 5.576 - 3.97 + 7.3

= 8.906KN

Horizontal loading



Point load acting at C, $R_{CH} = F_H/2 = 41.628/2 = 20.814$ KN

Point load acting at D, $R_{DH} = F_H/2 = 41.628/2 = 20.814$ KN

Taking moment at A,

$\sum M_{AH} = 0$

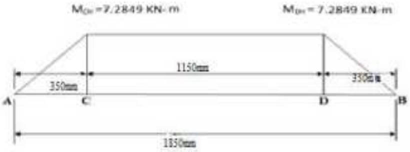
$\Rightarrow (R_{CH} \times 0.35) + (R_{DH} \times (0.35+ 1.150)) - (R_{BH} \times 1.850) = 0$

$\Rightarrow (20.814 \times 0.35)+(20.814 \times 1.5)- (R_{BH} \times 1.850) = 0$

$\Rightarrow R_{BH} = 20.814$ KN

Similarly, $R_{AH} = 20.814$ KN

Horizontal moments



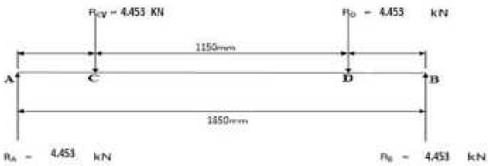
Moment at C,

$M_{CH} = R_{CH} \times 0.35$
$= 20.814 \times 0.35$
$= 7.2849$ kN-m

Moment at D,

$M_{DH} = R_{DH} \times 0.35$
$= 20.814 \times 0.35$
$= 7.2849$ kN-m

Vertical loading



Point load acting at C, $R_{CV} = F_V/2 = 8.906/2 = 4.453$ KN

Point load acting at D, $R_{DV} = F_V/2 = 8.906/2 = 4.453$ KN

Taking moment at A,

$\sum M_{AV} = 0$

$\Rightarrow (R_{CV} \times 0.35)+(R_{DV} \times (0.35+1.15))-(R_{BV} \times 1.85) = 0$

$\Rightarrow (4.453 \times 0.35) + (4.453 \times 1.5) - (R_{BV} \times 1.85) = 0$

$\Rightarrow R_{BV} = 4.453$ KN

Similarly, $R_{AV} = 4.453$ KN

Vertical moments

Moment at C ,

$M_{CV} = R_{CV} \times 0.35$
$= 4.453 \times 0.35$
$= 1.558$ kN-m

Moment at D,

$M_{DV} = R_{DV} \times 0.35$
$= 4.453 \times 0.35$
$= 1.558$ kN-m

Resultant moment at C $= \sqrt{(M_{CV})^2 + (M_{CH})^2}$

$= \sqrt{(7.2849)^2 + (1.558)^2} = 7.44$ kN-m

Resultant moment at D $= \sqrt{(M_{DV})^2 + (M_{DH})^2} = \sqrt{(7.2849)^2 + (1.558)^2} = 7.44$ kN-m

Case-1: Based on Equivalent torque

We know

T= 6278 Nm = 6.278kN-m

$M = M_{CR} = M_{DR} = 7.44$ kN-m

Equivalent torque $T_{eq} = \sqrt{(M \times K_b)^2 + (T \times K_t)^2}$

Where K_b = bending service factor = 1.5

K_t = torque service factor = 1.25

Therefore, $T_{eq} = \sqrt{(7.44 \times 1.5)^2 + (6.278 \times 1.25)^2}$

=13.64kN-m

Allowable shear stress (τ_s) $= \frac{16}{\pi d^3} \times T_{eq}$

$$\Rightarrow d = \sqrt[3]{\frac{16 \times T_{eq}}{\tau_s \times \pi}}$$

$$= \sqrt[3]{\frac{16 \times 13.64 \times 10^3}{45 \times 10^6 \times \pi}}$$

= 0.0115 m

= 115.5 mm

Case 2: Based on Equivalent moment (AT HUB)

Equivalent moment

$$M_{eq} = \frac{1}{2}[(M \times K_b) + T_{eq}]$$

$$= \frac{1}{2}[(M \times K_b) + \sqrt{(M \times K_b)^2 + (T \times K_t)^2}$$

$$= \frac{1}{2}[(7.44 \times 1.5) + 13.64]$$

= 12.4kN-m

Allowable bending stress ($\boldsymbol{\sigma}_b$) $= \frac{32}{\pi d^3} \times M_{eq}$

$$\Rightarrow d = \sqrt[3]{\frac{32 \times M_{eq}}{\sigma_b \times \pi}}$$

$$= \sqrt[3]{\frac{32 \times 12.4 \times 10^3}{90 \times 10^6 \times \pi}}$$

= 0.11 m

= 111.86mm

Case-3: Based on Deflection method

Deflection based diameter

$$d = \sqrt[4]{\frac{W_R \times a \times L \times 16000}{E \times \pi \times \alpha}}$$

Where, W_R = resultant loading

$= \sqrt{(R_{CH})^2 + (R_{cv})^2}$

$= \sqrt{20.814^2 + 4.453^2}$

= 21.28 KN

a = Bearing centre to hub distance (mm)

L = Hub spacing (mm)

E = Young's modulus for shaft (N/mm^2)

α = Allowable deflection (radians)

= 0.0015 rad to 0.0017 rad

= 0.0017 rad (max.)

$$\text{Therefore, } d = \sqrt[4]{\frac{21.28 \times 350 \times 1150 \times 16000}{2 \times 10^5 \times \pi \times 0.0017}}$$

= 106.44 mm

So, selected shaft size is 115 mm at bearing and 120 mm at hub.

5.7.2. TAIL PULLEY SHAFT DESIGN

Tail tension (T_t)

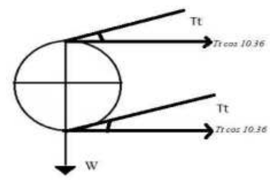
$= T_2 + [f \times L \times g \times (m_B + m_R)] - [H \times g \times m_B]$

$= (11.5812 \times 10^3) + [0.030 \times 29 \times 9.81 \times (20 +10)] - [5.112 \times 9.81 \times 20]$

= 10834.271

= 10.834 KN

Weight of the tail pulley (W) = 520 Kgs = 5101.2 N = 5.1 KN



Resolving horizontal and vertical components

$F_H = T_t \cos 10.36^0 + T_t \cos 10.36^0$

$= 2 \times T_t \times \cos 10.36^0$

$= 2 \times 10.384 \times \cos 10.36^0$

= 21.31 KN

$F_V = W - (T_t \sin 10.36^0 + T_t \sin 10.36^0)$
$= 5.101 - (2 \times T_t \times \sin 10.36^0$
$= 5.101 - (2 \times 10.384 \times \sin 10.36^0)$
$= 1.104$ KN

Horizontal loading

Point load acting at C, $R_{CH} = F_H/2 = 21.31/2 = 10.65$ KN

Point load acting at D, $R_{DH} = F_H/2 = 21.31/2 = 10.65$ KN

Taking moment at A,

$\sum M_{AH} = 0$

$\Rightarrow (R_{CH} \times 0.35) + (R_{DH} \times (0.35 + 1.15)) - (R_{BH} \times 1.85) = 0$

$\Rightarrow (10.65 \times 0.35) + (10.65*1.5) - (R_{BH} \times 1.85) = 0$

$\Rightarrow R_{BH} = 10.65$ KN

Similarly, $R_{AH} = 10.65$ KN

Horizontal moments

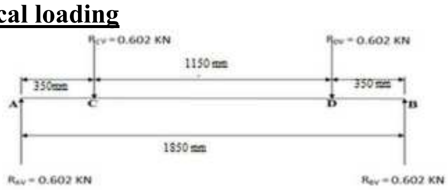
Moment at C,

$M_{CH} = R_{CH} \times 0.35$
$= 10.65 \times 0.35$
$= 3.7275$ KN-m

Moment at D,

$M_{DH} = R_{DH} \times 0.35$
$= 10.65 \times 0.35$
$= 3.7275$ KN-m

Vertical loading



Point load acting at C, $R_{CV} = F_V/2 = 1.204/2 = 0.602$ KN

Point load acting at D, $R_{DV} = F_V/2 = 1.204/2 = 0.602$ KN

Taking moment at A,

$\sum M_{AV} = 0$

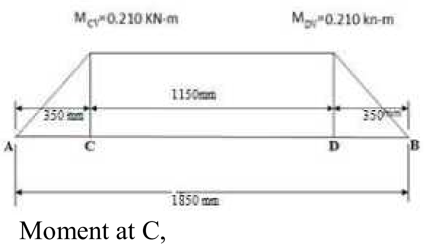
$\Rightarrow (R_{CV} \times 0.35) + (R_{DV} \times (1.5)) - (R_{BV} \times 1.85) = 0$

$\Rightarrow (0.602 \times 0.35) + (0.602 \times 1.5) - (R_{BV} \times 1.85) = 0$

$\Rightarrow R_{BV} = 0.602$ KN

Similarly, $R_{CV} = 0.602$ KN

Vertical moments



Moment at C,

$M_{CV} = R_{CV} \times 0.35$
$= 0.602 \times 0.35$
$= 0.210$ kN-m

Moment at D,

$M_{DV} = R_{DV} \times 0.35$
$= 0.602 \times 0.35$
$= 0.210$ kN-m

Therefore,

Resultant moment at

$C = \sqrt{(M_{CV})^2 + (M_{CH})^2}$
$= \sqrt{(0.210)^2 + (3.7275)^2}$
$= 3.73$ KN-m

Resultant moment at

$D = \sqrt{(M_{DV})^2 + (M_{DH})^2}$
$= \sqrt{(0.210)^2 + (3.7275)^2}$
$= 3.73$ KN-m

Case-1: Based on moment

We know

$M = M_{CR} = M_{DR} = 3.73$ kN-m

Allowable bending stress $(\sigma_b) = \frac{32}{\pi d^3} \times M \times K_b$

$\Rightarrow d = \sqrt[3]{\frac{32 \times M \times K_b}{\sigma_b \times \pi}}$

$= \sqrt[3]{\frac{32 \times 3.73 \times 10^3 \times 1.5}{90 \times 10^6 \times \pi}}$

$= 88.02$ mm

Case-2: Based on Deflection method

Deflection based diameter

$d = \sqrt[4]{\frac{W_R \times a \times L \times 16000}{E \times \pi \times \alpha}}$

Where,

W_R = resultant loading
$= \sqrt{(R_{CH})^2 + (R_{cv})^2}$
$= \sqrt{10.65^2 + 0.602^2}$
$= 10.66$ KN

a = Bearing centre to hub distance (mm)

L = Hub spacing (mm)

E = Young's modulus for shaft (N/mm^2)

α = Allowable deflection (radians)
= 0.0015 rad to 0.0017 rad
= 0.0017 rad (max.)

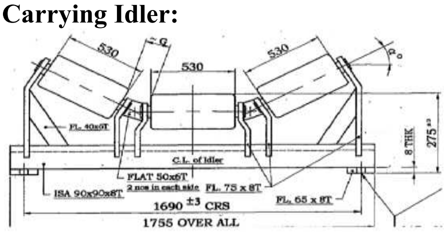
Therefore,

$d = \sqrt[4]{\frac{10.66 \times 350 \times 1150 \times 16000}{2 \times 10^5 \times \pi \times 0.0017}}$
$= 89.53$ mm

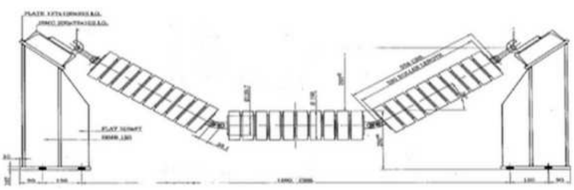
So, selected shaft size is 100 mm at bearing and 110 mm at hub

6. Components of belt conveyor:

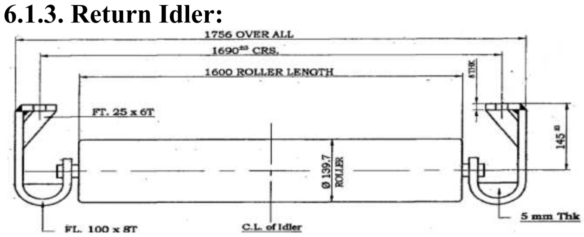
6.1.1. Carrying Idler:



6.1.2. Impact Idler:



6.1.3. Return Idler:



7. RESULTS AND CONCLUSIONS

Absorbed power of the pulley (P_A) = 23.028 kW
Motor power (P_M) =24.49kW
Speed of pulley (N) =1500 rpm at input=35.043 rpm at output
Carrying side belt tension (T_1) = 31.01 KN
Return side belt tension (T_2) = 11.812 KN

Pulleys and shafts:

Pulley	Weight of pulley	Pulley diameter	Shaft diameter
Drive pulley	7357.5 N	630 mm	115 mm at bearing, 120 mm at hub
Tail pulley	5101.2 N	500 mm	100 mm at bearing, 110 mm at hub
Other pulleys	4561.6 N	400 mm	90 mm at bearing, 100 mm at hub

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