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.

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/m³
- 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) – 29 m
- Belt speed (V) – 1.2 m/s
- Troughing angle (λ) – 35°
- Conveyor Inclination – 10.36°
- 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.

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 IVT may be easily calculated using the formula:

$$ S = \frac{IVT}{6000} \left[ \text{m}^2 \right] $$

where:

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

Angles of surcharge, repose, and material fluency:
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.

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:

$$ q_0 = \frac{I_v}{3.6 \times v} \quad [Kg/m] $$

where:
- $q_0$ = weight of material per linear meter
- $I_v$ = belt load t/h
- $v$ = belt speed m/s
$q_0$ is used in determining the tangential force $F_u$.

5.1.1. Maximum speeds advised:

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 $I_vT$ [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.

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.

5.2.2 Minimum belt width:

5.2.3. Loaded volume IM:
The volumetric load on the belt is given by the formula:

$$ l_v = \frac{lv}{qs} \quad [m³/h] $$

where:
- $l_v$ = load capacity of the belt [ t/h ]
- $qs$ = specific weight of the material
Also defined as:

$$ l_v = \frac{lv}{\gamma} \quad [m³/h] $$

Where the loaded volume is expressed relevant to the speed of 1 mtr/sec.
It may be determined from Table 5a-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.
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:

\[ IVM = IVT \times K \times K_1 \ [m^3/h] \]

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 \( v = 1 \) m/s.
- \( K \) is the factor of inclination.
- \( K_1 \) is the correction factor given by the 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
- $K1 = 0.90 \div 0.80$ 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:

$$IM = IVM \times v \ [m^3/h]$$

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

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

Where,

- $P_A$ = Absorbed power i.e., power required for drive pulley after taking drive pulleys loss into account.
- $R_{wd}$ = Wrap resistance for drive pulley (230 N)
- $R_{bd}$ = Pulley bearing resistance for drive pulley (100 N)

Therefore,

$$P_A = 23.028 + \left(\frac{230+100}{1000}\right) \times 2\pi = 23.028 + 0.396 = 23.424 kW$$

5.3. ABSORBED POWER:

The output rpm is calculated using the formula,

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

Where,

- $D = $ Diameter of driving pulley
- $N = $ Motor rpm

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

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

$$⇒ N = \frac{60 \times 1.2}{\pi x 0.654} = 35.043 rpm$$

5.5. DRIVE PULLEY SHAFT DESIGN

5.5.1. DRIVE PULLEY SHAFT DESIGN

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

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

Resolving horizontal and vertical components

$$F_H = T_1 \cos10.36° + T_2 \cos19.64°$$

$$= 30.504 + 11.124 = 41.6285 KN$$

$$F_V = T_1 \sin10.36° - T_2 \sin19.64° + W$$

$$= (31.01 \sin10.36°) - (11.812 \sin19.64°) + 7.3$$

$$= 8.906 KN$$

5.4. MOTOR POWER

The motor output power (shaft) is given by

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

Where,

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

Therefore,

$$P_M = \frac{23.424}{0.9} = 24.94 kW$$

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.

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

As the motor is of 1500 rpm,

Input rpm = 1500 rpm

$\frac{1500}{Output \ rpm} = 37$ kW

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Point load acting at C, \( R_{CH} = F_v/2 = 41.628/2 = 20.814 \text{ KN} \)

Point load acting at D, \( R_{DH} = F_v/2 = 41.628/2 = 20.814 \text{ KN} \)

Taking moment at A,
\[ \sum M_{AH} = 0 \]
\[ \Rightarrow (R_{CH} \times 0.35) + (R_{DH} \times 0.35 + 1.15) - (R_{HH} \times 1.850) = 0 \]
\[ \Rightarrow 20.814 \times 0.35 + (20.814 \times 1.5) - (R_{HH} \times 1.850) = 0 \]
\[ \Rightarrow R_{HH} = 20.814 \text{ KN} \]

Similarly, \( R_{AH} = 20.814 \text{ KN} \)

**Horizontal moments**

Moment at C,
\[ M_{CH} = R_{CH} \times 0.35 = 20.814 \times 0.35 = 7.2849 \text{ kN-m} \]

Moment at D,
\[ M_{DH} = R_{DH} \times 0.35 = 20.814 \times 0.35 = 7.2849 \text{ kN-m} \]

**Vertical loading**

Point load acting at C, \( R_{CV} = F_v/2 = 8.906/2 = 4.453 \text{ KN} \)

Point load acting at D, \( R_{DV} = F_v/2 = 8.906/2 = 4.453 \text{ KN} \)

Taking moment at A,
\[ \sum M_{AV} = 0 \]
\[ \Rightarrow (R_{CV} \times 0.35) + (R_{DV} \times 0.35 + 1.15) - (R_{DV} \times 1.85) = 0 \]
\[ \Rightarrow (4.453 \times 0.35) + (4.453 \times 1.5) - (R_{DV} \times 1.85) = 0 \]
\[ \Rightarrow R_{DV} = 4.453 \text{ KN} \]

Similarly, \( R_{AV} = 4.453 \text{ KN} \)

**Vertical moments**

Moment at C,
\[ M_{CV} = R_{CV} \times 0.35 = 4.453 \times 0.35 = 1.558 \text{ kN-m} \]

Moment at D,
\[ M_{DV} = R_{DV} \times 0.35 = 4.453 \times 0.35 = 1.558 \text{ kN-m} \]

Resultant moment at C = \( \sqrt{(M_{CV})^2 + (M_{CH})^2} = \sqrt{(7.2849)^2 + (1.558)^2} = 7.44 \text{ kN-m} \)

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

**Case-1: Based on Equivalent torque**

We know
\[ T = 6278 \text{ Nm} = 6.278 \text{ kN-m} \]
\[ M = M_{CR} = M_{DR} = 7.44 \text{ kN-m} \]

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

Where, \( K_b \) = bending service factor = 1.5
\( K_c \) = torque service factor = 1.25

Therefore, \( T_{eq} = \sqrt{(7.44 \times 1.5)^2 + (6.278 \times 1.25)^2} = 13.64 \text{ kN-m} \)

Allowable shear stress \( (\tau) = \frac{16}{\pi d^2} \times T_{eq} \)
\[ = \frac{16 \times 13.64 \times 10^3}{40^2} = 0.1115 \text{ m or 115.5 mm} \]

**Case 2: Based on Equivalent moment (AT HUB)**

Equivalent moment
\[ M_{eq} = \frac{1}{2}[(M \times K_b) + T_{eq}] \]
\[ = \frac{1}{2}[(7.44 \times 1.5) + 13.64] \times 10^3 \text{ kN-m} \]
\[ = 12.4 \text{ kN-m} \]

Allowable bending stress \( (\sigma) = \frac{32}{\pi d^2} \times M_{eq} \)
\[ = \frac{32 \times 12.4 \times 10^3}{\pi 115^2} = 0.11 \text{ m or 111.86 mm} \]

**Case-3: Based on Deflection method**

Deflection based diameter
\[ d = \sqrt{\frac{W_R \times a \times L \times 16000}{E \times \pi^2 \times a}} \]

Where, \( W_R \) = resultant loading
\[ = \sqrt{(R_{CH})^2 + (R_{DV})^2} = \sqrt{20.814^2 + 4.453^2} = 21.28 \text{ KN} \]
\( a = \text{ Bearing centre to hub distance (mm)} \)
\( L = \text{ Hub spacing (mm)} \)
\( E = \text{ Young’s modulus for shaft (N/mm)} \)
\( \alpha = \text{ Allowable deflection (radians)} \)
\[ = 0.0015 \text{ rad to 0.0017 rad} \]
\[ = 0.0017 \text{ rad (max.)} \]

Therefore, \( d = \sqrt{\frac{21.28 \times 350 \times 1150 \times 16000}{2 \times 10^6 \times \pi^2 \times 0.0017}} = 106.44 \text{ mm} \)

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

**5.7.2. TAIL PULLEY SHAFT DESIGN**

Tail tension \( (T_i) \)
\[ = T_i - [f \times L \times g \times (m_0 + m_0)] - [H \times g \times m_0] \]
\[ = (11.5812 \times 10^3) + [0.030 \times 29 \times 9.81 \times (20 + 10)] - [5.112 \times 9.81 \times 20] \]
\[ = 10834.271 \text{ N} \]
\[ = 10.834 \text{ KN} \]

Weight of the tail pulley \( (W) = 520 \text{ Kgs} = 5101.2 \text{ N} \)
\[ = 5.1 \text{ KN} \]

**Resolving horizontal and vertical components**

\[ F_H = T_i \cos 30^\circ + T_i \cos 30^\circ \]
\[ = 2 \times 10.384 \times \cos 30^\circ \]
\[ = 21.3 \text{ KN} \]
Horizontal loading

Point load acting at C, \( R_{CH} = \frac{F_l}{2} = 21.31 \div 2 = 10.65 \) KN
Point load acting at D, \( R_{DH} = \frac{F_l}{2} = 21.31 \div 2 = 10.65 \) 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 \times 1.5) - (R_{BH} \times 1.85) = 0 \\
\Rightarrow R_{BH} = 10.65 \) KN
Similarly, \( R_{AH} = 10.65 \) KN

Horizontal moments

Vertical loading

Point load acting at C, \( R_{CV} = \frac{F_l}{2} = 1.204/2 = 0.602 \) KN
Point load acting at D, \( R_{DV} = \frac{F_l}{2} = 1.204/2 = 0.602 \) 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

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 \( (\sigma) = \frac{32}{n^3} \times M \times A_b \)
\[
\Rightarrow d = \frac{32 \times M \times A_b}{\sigma \times \pi} \\
= \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{\frac{W_R \times a \times L \times 16000}{E \times \pi \times a}} \\
Where,
\[
W_R = \text{resultant loading} \\
= \sqrt{(R_{CH})^2 + (R_{CV})^2} \\
= \sqrt{10.65^2 + 0.602^2} \\
= 10.66 \) KN
\[
a = \text{Bearing centre to hub distance (mm)} \\
L = \text{Hub spacing (mm)} \\
E = \text{Young’s modulus for shaft (N/mm²)} \\
\alpha = \text{Allowable deflection (radians)} \\
\Rightarrow = 0.0015 \text{ rad to 0.0017 rad} \\
\Rightarrow = 0.0017 \text{ rad (max.)}
\]
Therefore,
\[
d = \sqrt{\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 ($P_M$) = 24.49 kW
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:**

<table>
<thead>
<tr>
<th>Pulley</th>
<th>Weight of pulley</th>
<th>Pulley diameter</th>
<th>Shaft diameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>Drive pulley</td>
<td>7557.3 N</td>
<td>630 mm</td>
<td>115 mm at bearing, 120 mm at hub</td>
</tr>
<tr>
<td>Tail pulley</td>
<td>5101.2 N</td>
<td>500 mm</td>
<td>100 mm at bearing, 110 mm at hub</td>
</tr>
<tr>
<td>Other pulleys</td>
<td>4561.6 N</td>
<td>400 mm</td>
<td>90 mm at bearing, 100 mm at hub</td>
</tr>
</tbody>
</table>

**REFERENCES**

4. Design of shaft using Concept In design of machine element by Susmitha (K.I. University)
5. Design and its Verification of Belt Conveyor System using concept by Bandlamudi. Raghu Kumar Asst. Professor (K.I. University)
8. Fener Dunlop “Conveyor Handbook” conveyor belting, Australia (June 2009)
9. Mathews “Belt conveyor,” FKI Logistex publication, Cincinnati, Ohio
11. Design Data Book By PSG College of Technology.