Chapter 3: Mechanical Design

3.1 Introduction

A conveyor belt is the carrying medium of a belt conveyor system. It is one of many types of 5conveyor systems. A belt conveyor system consists of two or more pulleys, with an endless loop of carrying medium which is the flat belt that rotates about them. One or both of the pulleys are powered, moving the belt and the material on the belt forward. The powered pulley is called the drive pulley while the unpowered pulley is called the idler pulley. There are two main industrial classes of belt conveyors; those in general material handling such as those moving boxes or food along inside a factory which is implemented in our application as shown in **Fig.3.1**. [51]

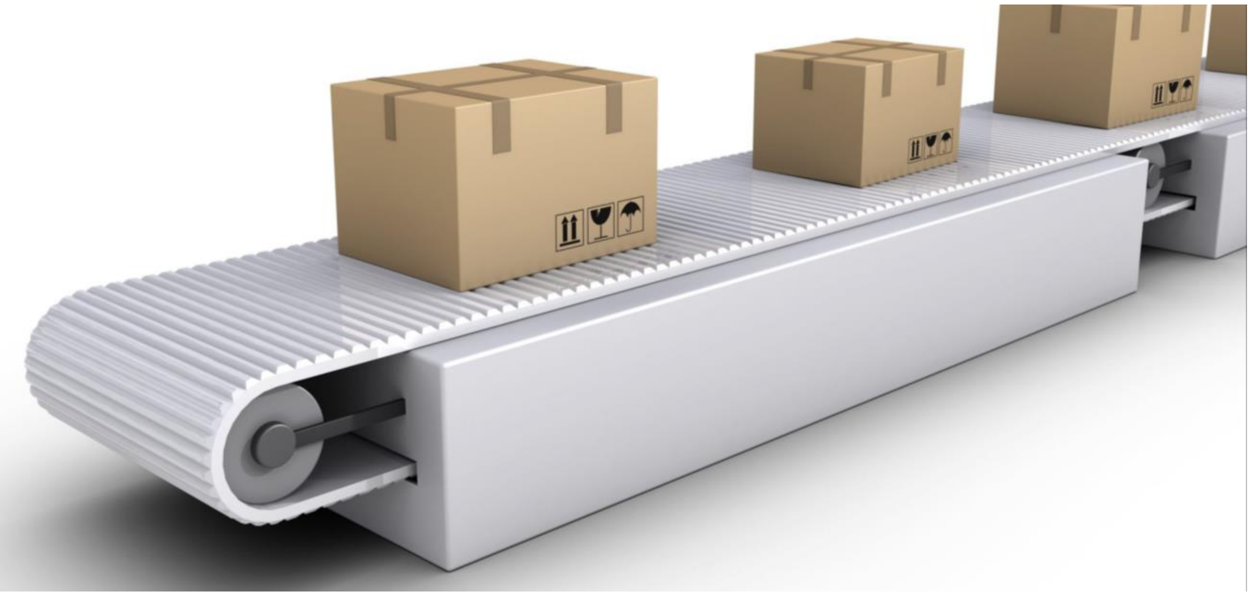

Fig.3.1: light duty conveyor belt

And bulk material handling that transport large volumes of resources and agricultural materials, such as grain, salt, coal, ore, sand, overburden and more, as shown in **Fig.3.2**.

Conveyors are durable and reliable components used in automated distribution and warehousing, as well as manufacturing and production facilities. In combination with computer-controlled pallet handling equipment this allows for more efficient retail, wholesale, and manufacturing distribution. It is considered a labor-saving system that allows large volumes to move rapidly through a process, allowing companies to ship or receive higher volumes with smaller storage space and with less labor expense, which we seek to do in this project as mentioned in **section 1.1**. [51]

Fig.3.2: Conveyor belt transporting coal form a mine

3.2 Power Estimation

Our conveyer system consists of a 3φ induction motor, worm Gearbox with gear ratio of 1:40, four flanged bearing, two main drums, two idler drums, flat belt and slider bed as shown in **Fig.3.3**. Each of the previous components is to be designed in this section to ensure healthy operation and long lifetime. This system is used to move trash to be sorted and extracted into its destination, with dimensions of (2500, 750, 780 mm) representing length, width, and height respectively, and it is set to run at maximum linear speed of 23 cm/s so that both the delta arm and camera can sort as many objects as possible.

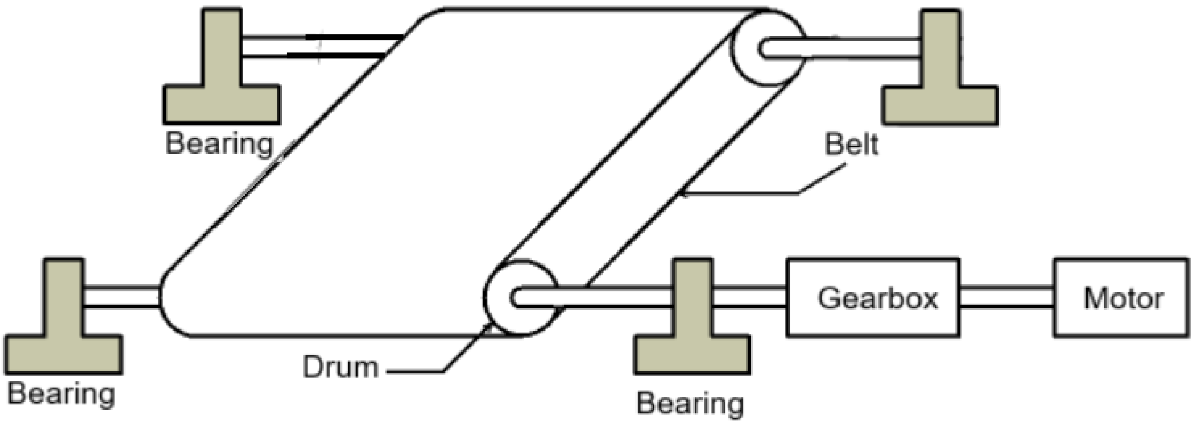

Fig.3.3: Conveyor Belt's components

Total mass of the belt, two drums and the load is necessarily calculated so that power estimation can be done it is also should be mentioned that the head drum is a bit larger than the tail drum so that the contact area of the belt with the head drum gets larger, hence reduce power needed to run the system and also reduce friction losses it should be mentioned also that it is preferred that the head drum pulls the belt not push it to reduce friction losses as shown in **Fig.3.4**. [52]

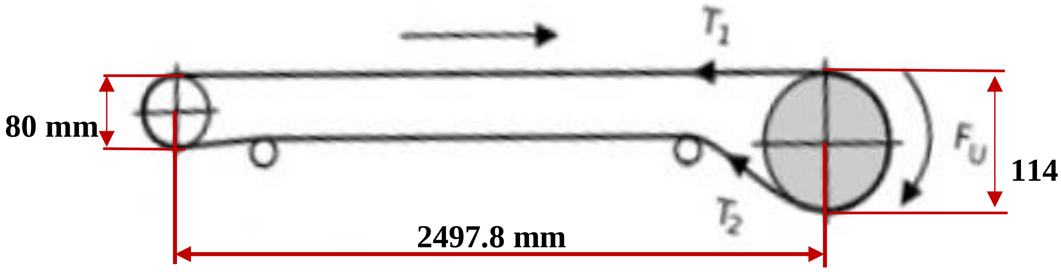

Fig.3.4: cross section view of the belt shape, main drums and idler rollers

Calculating both head and tail drums mass which are both 5mm in thickness and made form carbon steel which has density of 7800 kg/m^3 and its mechanical properties is in Appendix (1)

$$\mathbf{m_{HD} = \rho * V = 7800 * 0.8 * \frac{\pi}{4}(0.114^2 - 0.104^2) = 10.68 \cong 10.7\ kg} \qquad \textbf{(3.1)}$$

$$\mathbf{m_{TD} = \rho * V = 7800 * 0.8 * \frac{\pi}{4}(0.08^2 - 0.07^2) = 7.35 \cong 7.4\ kg} \qquad \textbf{(3.2)}$$

The belt dimensions are (5300*800*2mm) assume the density = 950 kg/m^3 until we choose proper belt from Appendix (2)

$$\mathbf{m_B = \rho * V = 950 * 5.3 * 0.8 * \frac{2}{1000} = 8.056 \cong 8\ kg} \qquad \textbf{(3.3)}$$

Using some estimation, we could assume that the maximum load that the conveyor would carry equals 10kg

$$\mathbf{m_L = 10\ kg} \qquad \textbf{(3.4)}$$

$$\mathbf{m_t = m_L + m_B + m_{HD} + m_{TD} = 10 + 8 + 7.4 + 10.7 = 36.1}\ \boldsymbol{kg} \qquad \textbf{(3.5)}$$

So, the tension force needed to move the conveyer and load to overcome the friction force as shown in Fig (3.5)

$$\mathbf{F_{drive} \geq \mu F_n \geq \mu m_t g} \qquad \textbf{(3.6)}$$

$$\mathbf{F_{drive} \geq \mu * 36.1 * 9.81} \qquad \textbf{(3.7)}$$

Friction coefficient μ equals 0.3 during starting, and equals 0.05 while running so

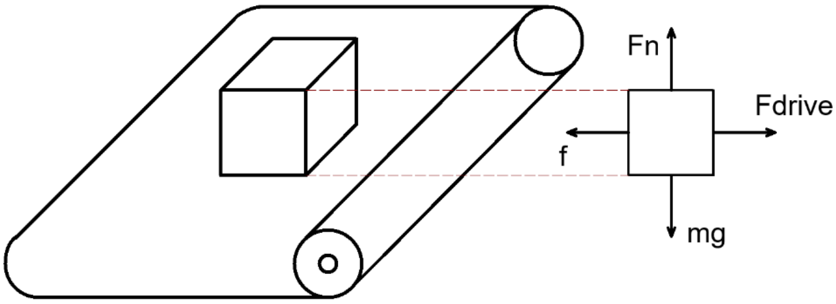

Fig.3.5: Force distribution on the conveyor belt

$$\mathbf{F_{drive\ initial} = 100\ N} \quad \textbf{(3.8)}$$

$$\mathbf{F_{drive\ running} = 16.68\ N} \quad \textbf{(3.9)}$$

So, the total power needed to move the load is

$$\mathbf{Power = F_{drive} * v} \quad \textbf{(3.10)}$$

$$\mathbf{P_{initial} = 100 * 0.3 = 30\ W} \quad \textbf{(3.11)}$$

$$\mathbf{P_{running} = 16.68 * 0.3 = 5\ W} \quad \textbf{(3.12)}$$

3.3 Flat Belt

3.3.1Types and applications

The first type is modular belt conveyors rather than a single, continuous loop, these systems use belting made of individual, interlocked segments. These segments are usually made of hard plastic. Since segments can be replaced individually, modular belt conveyors can be easier to repair than flat belt models. Hard plastic belting can also be easier to wash, and more resilient to sharp, abrasive, or otherwise problematic materials. This type is usually used in food industries and it is shown in **Fig.3.6** (a)

The second type is cleated belt conveyors integrate vertical cleats or barriers along the width of the belt. This can keep loose materials secure during inclines and declines along the length of the conveyor, to stage product for workers, to provide predictable spacing, and more. This type is mainly used in case of vertical or steep conveyers and in palletization also. This belt is made from metal or plastic as shown in **Fig.3.6** (b)

The third and most famous type of flat belt is flexible flat belt conveyor which is the most prevalent and versatile belt conveyor systems in common use. Flat belt conveyors use a series of powered pulleys to move a continuous flat belt of natural or synthetic fabrics such as polyester, nylon, or other materials. Product placed on top of the belt is then carried along from one end of the system to the other. As shown in **Fig.3.6** (c) and this is the type implemented in our application. [53]

(a) (b) (c)

Fig.3.6: Different types of conveyor belt

Belting consists of one or more layers of material. Typically, the belts in general material handling have two layers – the "carcass", which is the under layer of material that provides the belt's linear stability, strength and shape, and the "cover" which is the over layer that provides the belt's carrying or contact surface for the materials being handled. The carcass is often comprised of cotton material or plastic web/mesh. The cover is often various materials based on the belts intended use with the more common being rubber or plastic compounds. Belts are either pinned together at the splice or hot welded to provide a seamless splice.

3.3.2 Belt design

According to the tension needed from the estimation analysis and the available belt material in local markets, we will choose the PVC belt 1M6 V5-V5 from table (3.1). [54]

Material	**Polyvinyl chloride (PVC)**
Area density (W^*)	**2.0 kg/m^2**
Thickness(t)	**1.8 mm**
Admissible tension per width	**6 N/mm**
Coefficient of friction(μ)	**0.5**
Dimensions	**5300* 750 mm**

The mass of the belt then equals

$$\mathbf{m_B = W * \ A = 2 * 0.75 * 5.3 = \ 7.95\ kg} \tag{3.13}$$

Which is very close to the value calculated to estimate the power needed to drive the system in the previous section.

Design of flat belt is basically depending on calculating the needed friction coefficient and ensure that selected belt friction coefficient exceeds it. Friction coefficient is calculated using the belt-drive geometry and friction and the estimated needed power to drive the system.

From **Fig.3.4** the contact angle between the head drum and the belt can be calculated as follows

$$\boldsymbol{\emptyset = \ \pi \pm \sin^{-1}\left(\frac{D-d}{2C}\right)} \tag{3.14}$$

But due to the existence of idler roller this equation is useless and this angle could only be found through measurements from SOLIDWORKS model

$$\boldsymbol{\emptyset = 192°\text{=3.34 rad}} \tag{3.15}$$

Then in order to find the forces acting on the belt the following equation is applied

$$\mathbf{\frac{F_1 - F_c}{F_2 - F_c} = e^{\mu\emptyset} = 5.34} \tag{3.16}$$

Calculating F_c which the hoop tension due to centrifugal force from speed and belt geometry as W is the weight of belt per unit length and V is the maximum linear speed which equals 23 cm/s.

W is obtained by multiplying the area density and width of the belt.

$$\mathbf{W = W^* * 0.75 = 2 * 0.75 = 1.5\ N/m} \tag{3.17}$$

$$\mathbf{F_c = \frac{w * v^2}{g} = \frac{1.5 * (0.23)^2}{9.81} = 0.008909\ N} \tag{3.18}$$

Then torque acting on the drum is calculated by the rated motor power **P** which equals 0.5 hp with rated speed of 1430 **RPM** which was the lowest rating of a 3φ induction motor in the local market, and taking service factor $\mathbf{K_s}$ for light shock from table (3) in Appendix (2) and design factor $\mathbf{n_d}$ for overload 1.1 with the angular speed of drum $\boldsymbol{\omega}$ which is 35.75 RPM due to the effect of gearbox with gear ratio of 1:40.

$$\mathbf{T} = \frac{\mathbf{P\,K_s n_d}}{\boldsymbol{\omega}} = \frac{\mathbf{0.5 * 746 * 1.2 * 1.1}}{\mathbf{35.75 * \frac{2\pi}{60}}} = \mathbf{109.6\ N.m} \qquad \textbf{(3.19)}$$

By getting the torque maximum **T** the belt can withstand, and the head drum diameter **D** the difference between the two belt forces F_1, F_2 can be found using the following equation.

$$\mathbf{F_1 - F_2 = \frac{2T}{d} = \frac{2 * 109.6}{0.114} = 1922.74\ N} \qquad \textbf{(3.20)}$$

Knowing that $\mathbf{F^{`}}$ is the maximum allowed tension per unit width of the belt from table (1) in Appendix (2), the drum width **b** equals 0.8m, pulley correction factor $\mathbf{C_p}$, and velocity correction factor $\mathbf{C_v}$ **from** table (5) in Appendix (2).

$$\mathbf{F_1 = b * F^{`} * C_p * C_v = 0.8 * 6 * 10^3 * 1 * 1 = 4800\ N} \qquad \textbf{(3.21)}$$

Then $\mathbf{F_2}$ can be determined using Eqn (3.20), (3.21)

$$\mathbf{F_2 = F_1 - \frac{2T}{d} = 4800 - 1922.74 = 2877.2N} \qquad \textbf{(3.22)}$$

Then the initial tension in the belt can be also determined as follows

$$\mathbf{F_i = \frac{F_1 + F_2}{2} - F_c = \frac{4800 + 2877.2}{2} - 0.00809 = 3838.6N} \qquad \textbf{(3.23)}$$

Then we ensure that the flat belt can withstand these tensions- in our case it does- after that we check the friction coefficient to make sure that the belt would not slip.

$$\boldsymbol{\mu^{`}} = \frac{\mathbf{1}}{\boldsymbol{\varphi}} * \mathbf{ln}\left(\frac{\mathbf{F_1 - F_c}}{\mathbf{F_2 - F_c}}\right) = \frac{\mathbf{1}}{\mathbf{3.35}} * \mathbf{ln}\left(\frac{\mathbf{4800 - 0.008}}{\mathbf{2877.2 - 0.008}}\right) = \mathbf{0.152 < 0.5} \qquad \textbf{(3.24)}$$

3.4 Slider Bed

3.4.1 Slider bed vs. Rollers

General material handling belt conveyor systems are typically comprised of one of two designs – roller bed belt conveyors, in which the belt is supported and carried by rollers as shown in **Fig.3.7** (a), or slider bed belt conveyors, in which the belt is supported and carried over a continuous metal bed surface or a series of metal pans as shown in **Fig.3.7** (b). Roller bed design allows for longer runs of conveyor due to the lower coefficient of friction experienced between the weight of the product on the belt and the free-turning rollers as compared to the drag created between the belt and the static bed surface and/or metal pans.


(a) (b)

Fig.3.7: (a) Roller bed conveyor belt, (b) Slider bed conveyor belt

Roller bed is usually used in applications where the conveyor must be long as it has low friction coefficient which increases flat belt lifetime, but it is expensive, needs regular maintenance. Slider bed which is basically a plane piece of metal that does not need any maintenance and has very long lifetime, but it has higher friction coefficient which shorten the lifetime of the flat belt. Our main concern in choosing between the two types was vibration and stability; as we deal only with small objects for small distance, if these objects were shifted or moved from its place the arm would not be able to pick and sort it according to the location the camera captured; so our choice was slider bed conveyor belt. [55]

3.4.2 Slider bed design

Consider the bed to have rounded edges, smooth surface, and the minimum distane between belt and bed which is indicaed in Fig (3.8) must be

$$\Delta L_{minimum} = 2 \text{ mm} \quad \textbf{(3.25)}$$

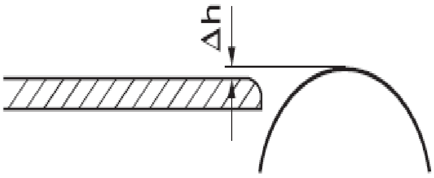

Fig.3.8: Minimum distance between bed and belt

3.5 Roller shaft

Roller shaft is considered to be the main part that its failure causes the whole system to fail so great concern is directed to its stress analysis. Calculating the diameter of the roller shaft is done by two ways the first is to find all stress which will affect the shaft and find the combined stress result of all this, the second is to use ASME equation for solid shaft, then SOLIDWORKS simulation tool is used to verify those two methods. In this design carbon steel in Appendix (1) is used in machining both roller shaft and both main drums.

From analysis the system, we found that the shaft is exposed to 3 kinds of stress direct shear, torsion and bending, so we need to find the 3 stresses and using the combined stress equation in two dimensions to get the maximum normal stress and shear stress.

3.5.1 Combined Stress Calculations

This method is considered static stress analysis although the roller rotates but this static analysis neglects both damping which does not exist in our application and also the inertia of the roller also due to the symmetricity of the drums and rollers so it can be considered stationary. [56]

3.5.1.1 Torsion Stress

To get the torsion, acting torque T must be calculated first, along with the second moment of inertia J, knowing that the used motor power equals 0.5 hp and the rotating speed equals 1420 RPM before reduction at the gearbox which its gear ration equals 1:40.

$$\mathbf{T = \frac{P}{\omega} = \frac{0.5 * 746}{1420 * \frac{2\pi}{60}} * 40 = 100.33\ N.m = 10033N.mm} \tag{3.26}$$

$$\mathbf{J = \frac{\pi\, d^4}{32},\quad r_{max} = \frac{d}{2}} \tag{3.27}$$

$$\mathbf{\tau_t = \frac{T * r}{J} = \frac{10033 * \frac{d}{2}}{\frac{\pi\, d^4}{32}} = \frac{51097.6}{d^3}\ MPa} \tag{3.28}$$

3.5.1.2 Direct shear stress

At first, the force that affect the roller shaft-in the normal direction to its axis- must be calculated by analyzing the conveyer structure it is found that two forces act on the roller which are the tension force in belt and gravitational force, as indicated in **Fig.3.9**, as well as the cross sectional area of the roller shaft should be calculated.

$$\mathbf{A = \frac{\pi\, d^2}{4}} \tag{3.29}$$

To find the resultant force F_t the following equation is used it is obvious that the mass of the roller and drum are needed and that is an unknown so for simplicity and due to the fact that tension force is much larger than the weight force it could be neglected.[56]

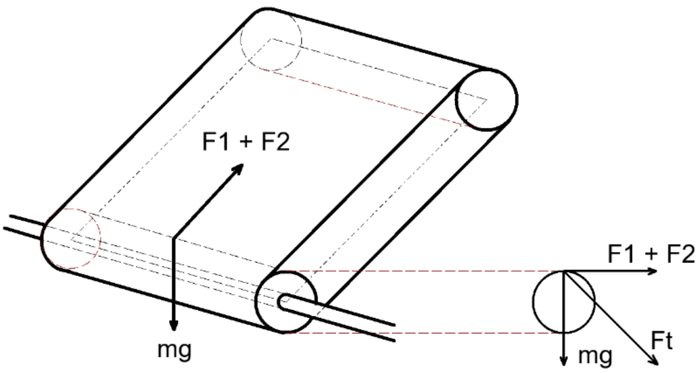

Fig.3.9: Force distribution on the roller shaft

$$\mathbf{F_t} = \sqrt{(\mathbf{m_t\,g})^2 + (\mathbf{F_1} + \mathbf{F_2})^2} = \sqrt{(\mathbf{4800} + \mathbf{2877})^2} = \mathbf{7677\ N} \qquad \mathbf{(3.30)}$$

$$\boldsymbol{\tau} = \frac{\mathbf{F}}{\mathbf{A}} = \frac{\mathbf{F}}{\frac{\boldsymbol{\pi}\,\mathbf{d^2}}{\mathbf{4}}} = \frac{\mathbf{7677}}{\frac{\boldsymbol{\pi}\,\mathbf{d^2}}{\mathbf{4}}} = \frac{\mathbf{9774.6}}{\mathbf{d^2}}\ \mathbf{MPa} \qquad \mathbf{(3.31)}$$

3.5.1.3 Bending stress

To calculate the bending stress, first the reaction forces which causes this stress should be calculated using the resultant force calculated in **Eqn (3.30)** as indicated in **Fig.3.10**. [56]

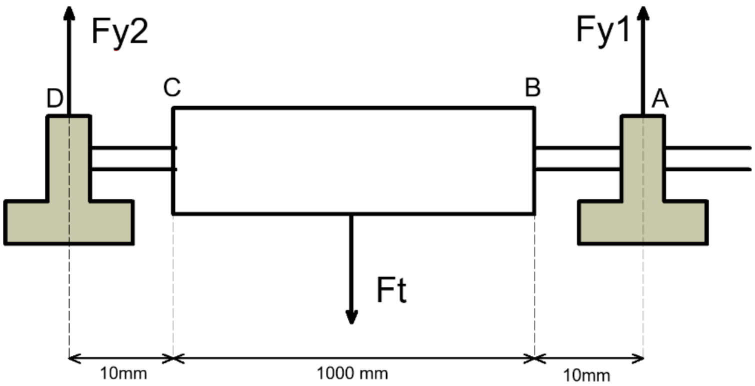

Fig.3.10: Force distribution in the Z-direction

$$\sum \mathbf{F_y} = \mathbf{0} \qquad \mathbf{(3.32)}$$

$$\mathbf{F_t} - \mathbf{F_{y1}} - \mathbf{F_{y2}} = \mathbf{0} \qquad \mathbf{(3.33)}$$

Due to the symmetricity of the roller shaft both the reaction force is equal

$$\mathbf{F_{y1}} = \mathbf{F_{y2}} = \frac{\mathbf{F_t}}{\mathbf{2}} = \frac{\mathbf{7677}}{\mathbf{2}} = \mathbf{3838.5\ N} \qquad \mathbf{(3.34)}$$

It is obvious that the weakest point is where the roller is welded with the drum due to the weight of the drum so the moment is calculated around point **b** as follows

$$\mathbf{M = F * r \ = 3838.5 * 10 = 38385\ N.mm\ (counterclockwise)} \tag{3.35}$$

Then the first moment of inertia of the roller is to be calculated

$$\mathbf{I = \frac{\pi\, d^4}{64}} \tag{3.36}$$

Then bending stress could be calculated using the previous works and $\mathbf{y = \frac{d}{2}}$

$$\mathbf{\sigma_b = \frac{M * y}{I} = \frac{M * \frac{d}{2}}{\frac{\pi\, d^4}{64}} = \frac{390986.4}{d^3}\ MPa} \tag{3.37}$$

3.5.1.4 Combined stress

As there is only one normal stress the simplified combined stress equation can be used

$$\boldsymbol{\sigma_{max} = \frac{\sigma}{2} + \sqrt{\left(\frac{\sigma}{2}\right)^2 + (\tau)^2}} \tag{3.38}$$

$$\boldsymbol{\tau_{max} = \sqrt{\left(\frac{\sigma}{2}\right)^2 + (\tau)^2}} \tag{3.39}$$

First find the diameter from normal stress equation (substitute from Eqn **(3.37), (3.31)**, **(3.28))**

$$\boldsymbol{\sigma_{max} = \frac{195493}{d^3} + \sqrt{\left(\frac{195493}{d^3}\right)^2 + \left(\frac{51098}{d^3} + \frac{9775}{d^2}\right)^2}} \tag{3.40}$$

By simplifying the equation to find d using cross multiplication and set it in polynomial form, as follows

$$\boldsymbol{\sigma_{max} = \frac{\sigma}{2} + \sqrt{\left(\frac{\sigma}{2}\right)^2 + (\tau)^2} = \frac{A}{d^3} + \sqrt{\left(\frac{A}{d^3}\right)^2 + \left(\frac{B}{d^3} + \frac{C}{d^2}\right)^2}} \tag{3.41}$$

$$\boldsymbol{\sigma_{max} = \frac{\sigma_u}{F.O.S} = \frac{282}{2} = 141\ MPa} \tag{3.42}$$

$$\boldsymbol{(\sigma_{max}{}^2) * d^6 - (2\sigma_{max} * A) * d^3 - (C^2) * d^2 - (2B * C) * d - B^2 = 0} \tag{3.43}$$

Substitute With constants in Eqn **(3.43)** we get

$$\mathbf{d^6 - 2773 * d^3 - 4806.1 * d^2 - 50247.2 * d - 131331.7 = 0} \tag{3.44}$$

Second get the diameter from shear stress equation (substitute from Eqn **(3.37), (3.31)**, **(3.28))**

$$\boldsymbol{\tau_{max}} = \sqrt{\left(\frac{195493}{d^3}\right)^2 + \left(\frac{51098}{d^3} + \frac{9775}{d^2}\right)^2} \tag{3.45}$$

By simplifying the equation to find d using cross multiplication and set it in polynomial form, the result will be.

$$\boldsymbol{\tau_{max}} = \frac{\sigma_u}{2 * F.O.S} = \frac{282}{4} = 70.5\ \mathrm{MPa} \tag{3.46}$$

$$d^6 - 19225.4 * d^2 - 201000 * d - 8214993.7 = 0 \tag{3.47}$$

Solving Eqn (3.47), (3.44) using the following piece of **MATLAB** code

```
d1=[1 , 0 , 0 , -2773 , -4806.1 , -50247.2 , -131331.7];
roots(d1)
d2=[1 , 0 , 0 , 0 , -19225.4 , -201000 , -8214993.7];
roots(d2)
```

It is found that the diameter according to normal stress equation solution

$$\mathbf{d = 14.97 \cong 15\ mm} \tag{3.48}$$

The diameter according to shear stress equation solution

$$\mathbf{d = 15.92 \cong 16\ mm} \tag{3.49}$$

3.5.2 ASME equation

By using the ASME equation to find the diameter of shaft taking the advantage of no axial force the equation will be

$$\boldsymbol{\sigma_{max}} = \frac{16}{\pi d^3} \left[(K_m M) + \sqrt{(K_m M)^2 + (K_t T)^2}\right] \tag{3.50}$$

$$\boldsymbol{\tau_{max}} = \frac{16}{\pi d^3} \sqrt{(K_m M)^2 + (K_t T)^2} \tag{3.51}$$

After the bending moment M at point B is calculated, and found the torque T also by choosing a proper value for K_m, K_t from table (4) in Appendix (2). For minor shock$K_m, K_t = 1.25$. We can find the proper diameter for roller shaft. The maximum normal stress is the yield stress over factor of safety.

$$\boldsymbol{\sigma_{max} = \frac{\sigma_y}{F.O.S} = \frac{282}{2} = 141\ MPa} \tag{3.52}$$

$$\boldsymbol{d^3 = \frac{16}{\pi\ \sigma_{max}}\left[(K_m\ M) + \sqrt{(K_m\ M)^2 + (K_t\ T)^2}\right] = 3524.4} \tag{3.53}$$

$$\boldsymbol{d = 15.2 \cong 16\ mm} \tag{3.54}$$

Now to find it from the shear stress equation. According to shear stress theory.

$$\boldsymbol{\tau_{yield} = \frac{\sigma_{yield\ normal}}{2} = 70.5\ MPa} \tag{3.55}$$

$$\boldsymbol{d^3 = \frac{16}{\pi\ \tau_{max}}\sqrt{(K_m\ M)^2 + (K_t\ T)^2} = 3582.63} \tag{3.56}$$

$$\boldsymbol{d = 15.3 \cong 16\ mm} \tag{3.57}$$

So, the specification of solid carbon steel shaft will be 0.82 m long diameter of 25mm as shown in **Fig.3.11**. [57]

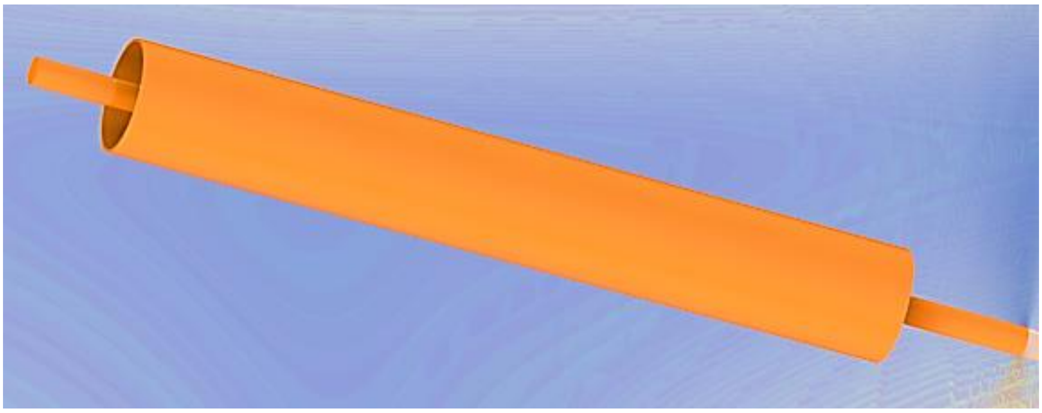

Fig.3.11: Head drum 3D model with the deign dimensions

3.5.3 SOLIDWORKS simulation

3.5.3.1 Model simplification

The model shown in **Fig.3.11** cannot undergo the simulation process as it would give wrong results so some simplifications are to be done to enhance this project. Basically, the simulation is don on the head drum as it suffers the same stress as the tail drum but in addition the motor torque is acting on it and this simulation is done during starting as the friction is maximum. The drum is to be split into two parts along the axis of the drum one that contacts the belt and belt tension is applied on it and the other is the part that contacts free air as shown in **Fig.3.12** while the roller is kept unmodified. [58], [59]

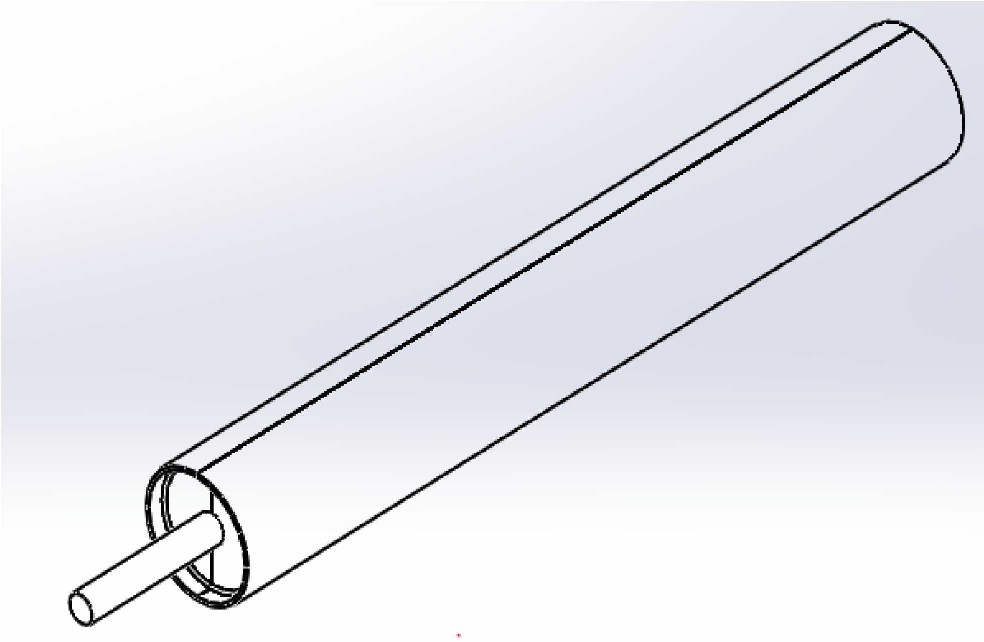

Fig.3.12: Modified drum, roller assembly for simulation

It should be mentioned also that the simulation is done in static mode and to remove conflict mode static mode can be applied when inertia and damping are not taken into consideration but it could be done for both moving and stationary objects. After these materials are added roller is kept carbon steel as the analytical calculations meanwhile the drum is made from annealed steel with $\boldsymbol{\sigma_{yield}} = \mathbf{460\ MPa}$. Then as the roller is starting we fixed both of its endings to simulate the starting process, when adding the acting forces on this assembly there is three main forces the first is the gravitational force, the second is motor torque which equals approximately half the full load torque multiplied by the gear ratio which is 1:40, the third and most important type of stress is the belt tension which affects along the axis of the drum so it is recommended to apply pressure not just point force on the cylinder.

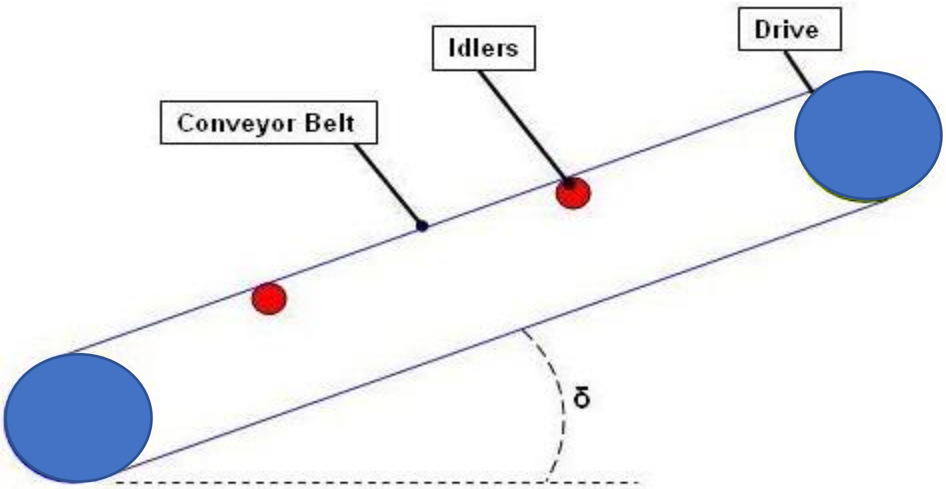

Fig.3.13: Model of a simple conveyor belt

The belt of the conveyor always experiences a tensile load due to the rotation of the electric drive, weight of the conveyed materials, and the idlers. The belt tension at steady state according to the conveyor illustration [60] in **Fig.3.13** equals

$$\mathbf{T_b = [1.37 * \mu * L * g] * [2 * m_i + (2 * m_b + m_m) * \cos(\delta)] + [H * g * m_m]} \quad \textbf{(3.58)}$$

Where,

μ = Coffieceint of friction = 0.3 (from starting).

L = Length of the conveyor = 2.457 m.

g = Acceleration of gravity = 9.8 m/s.

m_i = Idler load (kg/m) which equals

$$\mathbf{m_i = \frac{mass\ of\ a\ set\ of\ idlers}{idlers\ spacing} = \frac{2 * 1}{1.8} = 1.11\ kg/m} \quad \textbf{(3.59)}$$

m_b = Weight of the belt itself = 1.5 kg/m

m_m = Conveyed Load (kg/m)= zero during starting as the machine starts unloaded

δ = Inclination angle of the conveyor in degree = zero (in case of horizontal conveyors).

H = Conveyor belt height = 0.78 m

So, the belt tension is found to be

$$\mathbf{T_b = (1.37 * 0.3 * 2.457 * 9.8) * (2 * 6.4 + (2 * 1.5 + 0) * 1) = 51.65\ N} \quad \textbf{(3.60)}$$

Then the pressure on the drum surface equlas

$$\boldsymbol{P = \frac{T_{bs}}{A} = \frac{1.2 * 51.65}{0.5 * \frac{\pi}{4} * (0.114)^2} = 12.14\ \mathbf{kPa}} \quad \textbf{(3.61)}$$

After applying these forces on the meshing is done and due to the simplicity, symmetricity,and the regularity of the design it does not need any speical meshing method then equivlant stress, which should not exceed the yield strength of the used matrials for both drum and roller shaft. ,and total deformation, which shouldnot excedd 1 mm for good design, are calculated and we got the following results which were very acceptable and verfiy the work done in the previous section using combined stress equation and ASEM equation. [60]

3.5.3.2 Results

3.5.3.2.1 Roller Shaft

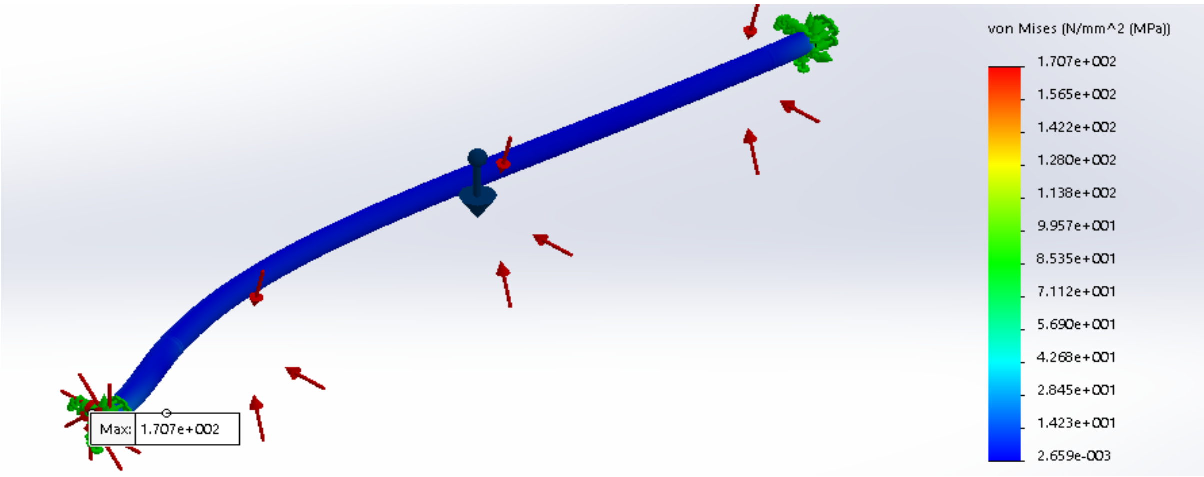

Fig.3.15: Equivlant stress on the roller shaft only

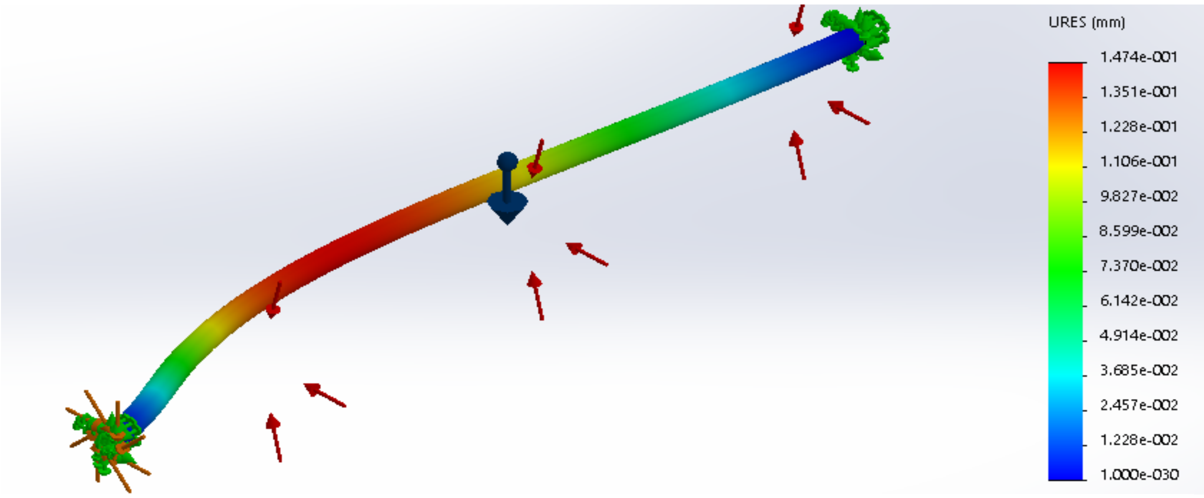

Fig.3.14: Total deformation on the roller shaft only

3.5.3.2.2 Drum

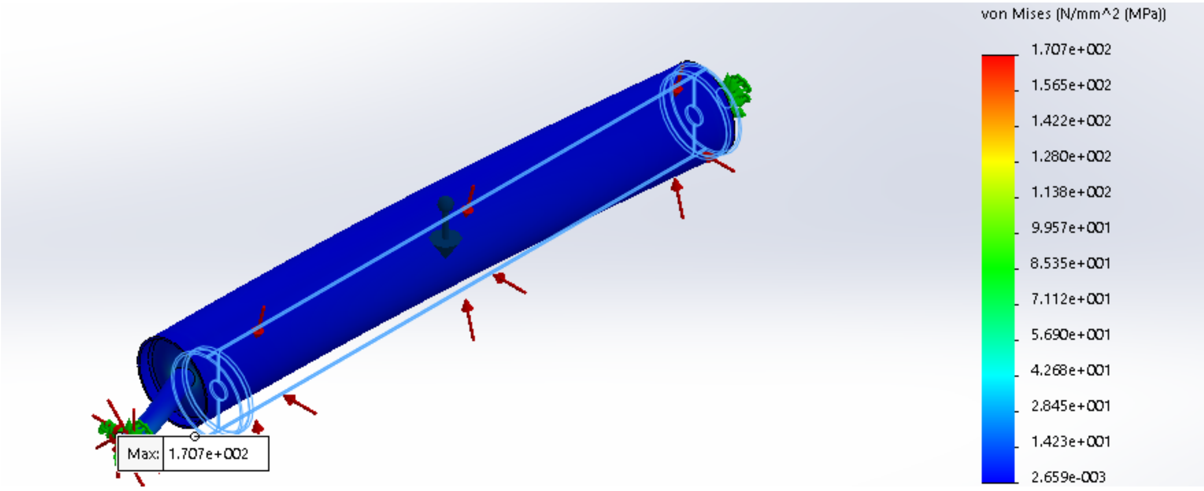

Fig.3.16: Equivlant stress on the roller shaft and drum

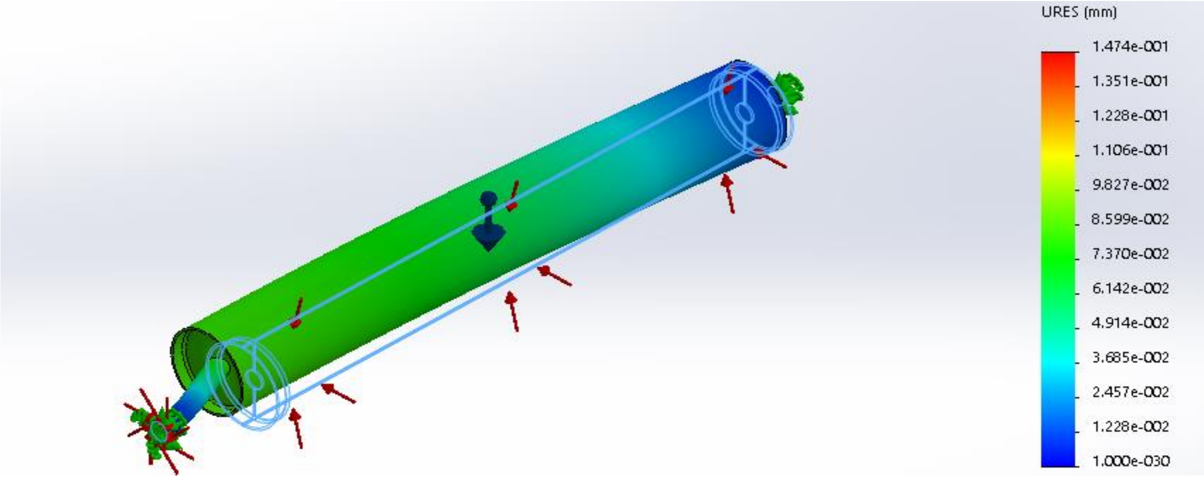

Fig.3.17: Total deformation on the roller shaft and drum

From these results we find that the used materials and diameters can handle the stress and they are not going to fail.

3.6 Drums and idlers

3.6.1 Belt tracking problems

Belt tracking problems such as tilting or divergence of the belt can damage the belt and suspending the process until the belt is recalibrated. The failure cause is usually to be found in the installation itself and may be the result of poorly adjusted pulleys and rollers, or faulty design. To reduce possibilities of failure the following conditions are essential for problem-free belt tracking:

1. The conveyor frame structure must be rigid and stable. It must be able to withstand all the forces acting upon it (belt tension, weight of the conveyed goods, uneven floors, etc.).
2. All pulleys and rollers must be fitted at right angles to the belt running axis.
3. All parts of the installation that come into contact with the belt are to be protected from dirt and soiling and to be cleaned so that any particle doesn't get between drums and the belt which will cause permanent damage to the belt. [61]

Where a belt runs over cylindrical pulleys that are at right angles to its directional path, then the forces acting upon it will be parallel to the running direction of the belt. No tracking forces are exerted on the belt. In fact, this a state of unstable equilibrium and the belt would run off immediately if subjected to the slightest external factors such as off-center loading of product, dirt between belt and pulley. Also if one or both of the two pulleys are not positioned accurately at right angles to the belt running axis. The belt will run off towards the less-tensioned side.as shown in **Fig.3.18**.

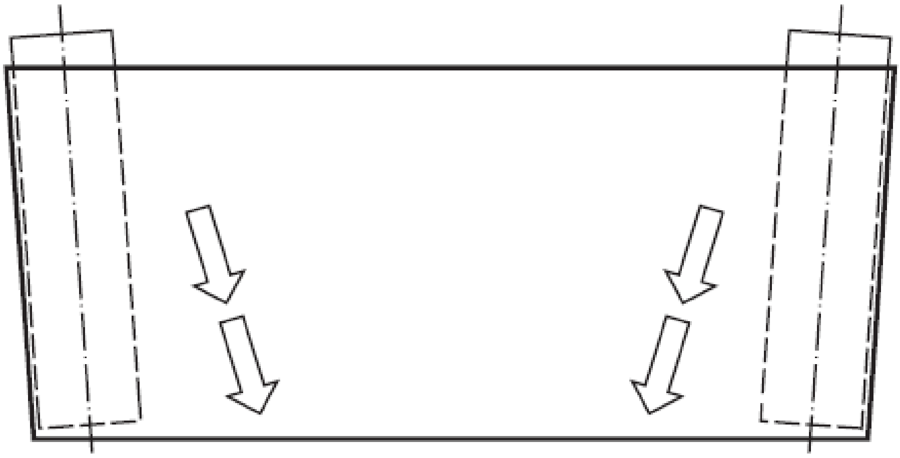

Fig.3.18: The belt tracks to the side with the least tension.

3.6.2 Radially crowned drums

Fabric belt conveyors are normally equipped with at least one, sometimes with several drums with cylindrical conical or radially crowned form. This basic measure is usually sufficient to achieve straight and stable running which is shown in **Fig.3.19**.

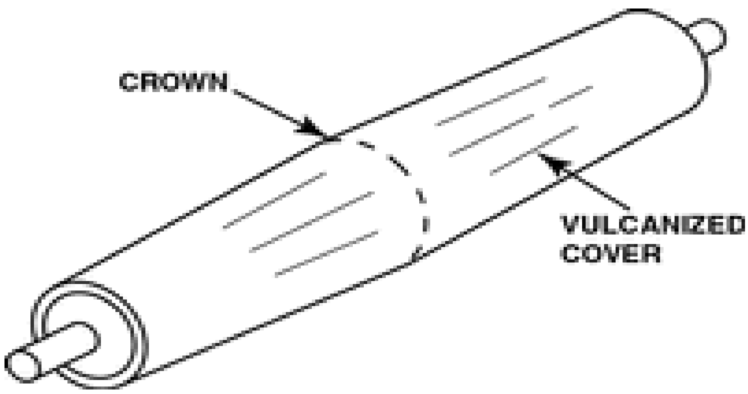

Fig.3.19: radially crowned drum

Drums with this shape exert a self-tracking effect. If there is a variable run-off tendency, or a reversal in running direction, the belt is centered without the need to adjust the axis. Detailed information on cylindrical conical drums. In simple two-drum conveyors with defined running direction it is usually the head pulley that is the driving pulley. It is designed in cylindrical-conical shape, but with conveyor aspect ratio (conveyor length to belt width) in excess of about 5 to 1 and in installations with reversing operations, it is advisable to crown both, head pulley and tail pulley. And that is what is chosen in our application for more safety. [61]

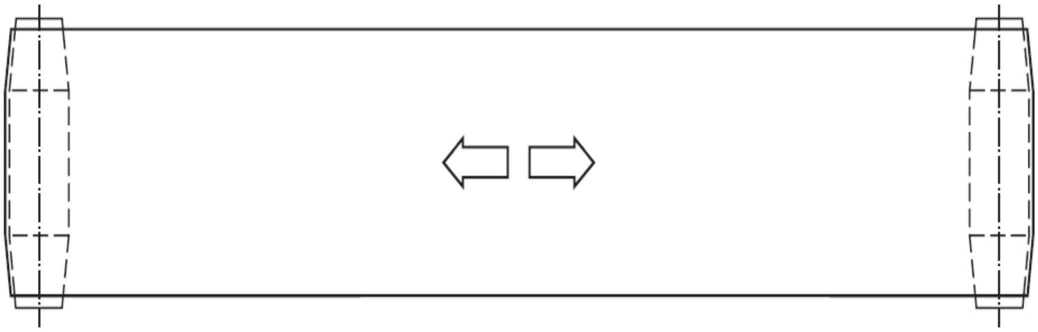

Fig.3.20: Correctly aligned belt between two radially crowned drums

It should also be mentioned that the basic use of cylindrical-conical drums, will not be sufficient. Additional belt tracking measures will be required.

3.6.3 Guiding idlers

Guiding drums, also called control drums or return idlers, are usually cylindrical. Made from rolled steel sheets that are then welded to form the cylinder as they suffer from neglected stress unlike the head and tail drums which are made by drawing as they are subjected to huge stress such as their heavy weight, belt tension, and motor torque. To achieve good tracking, the arc of contact at the return idlers should be minimum 30°. For belts with non-adhesive surface, the tracking effect can be improved with a friction cover of abrasion-resistant rubber. As indicated in Fig (3.21).

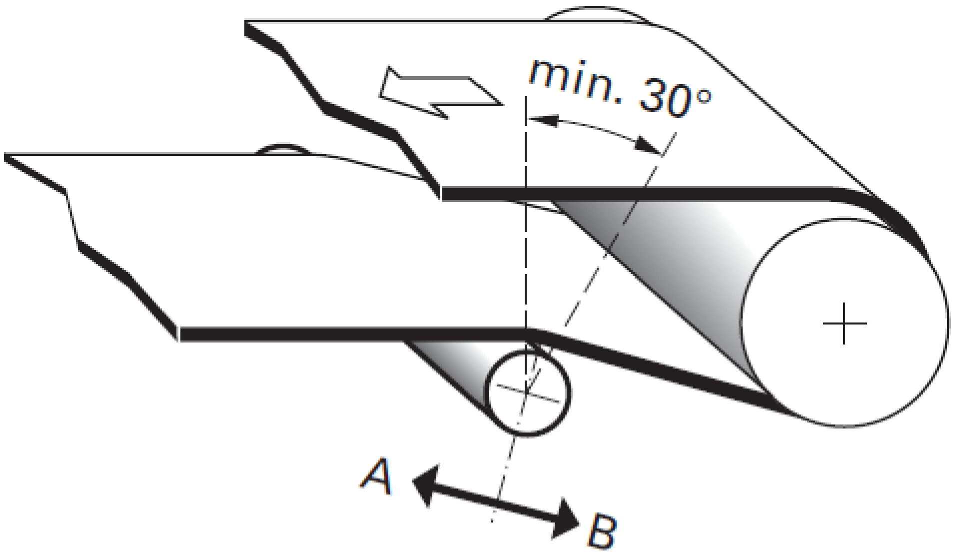

Fig.3.21: Arc of contact at the return idlers

Another advantage of the return idlers that they are used to adjust the tension of the belt as it gets low with time so getting them closer to the main drums would recalibrate belt tension. [61]

3.7 Conveyor motor

3.7.1 Types of motors and applications

There are a number of types of motors that can be selected depending on the work environment such as AC or DC motors, conveyor belts can be small enough to use single phase motors, or large enough for three phase motors. For fixed or constant speed applications, AC motors & gear motors are well suited, while for applications where speed control, higher speeds or maximum torque in a small area may be needed, the AC & brushless DC motor speed control systems can be used. For precise positioning stepper or servo motor packages are ideal.

Motors act on the sprocket of the drive pulley to produce movement of the belt of the conveyor. Generally, there are three categories of motors used for powering conveyors: fixed speed, variable speed, and position control motors.

1. Fixed speed motors provide rotation at a fixed speed. They can be either uni- or bi-directional and are generally easy to use.
2. Variable speed motors have a wide range of speeds. They can be easily adjusted to change the speed based on the type of work being done by the conveyor.
3. Position control motors provide precise position control of the conveyor. They are used to handle complex movements and have a wide variety of functions. They're highly reliable and are available in closed loop, 2-phase, and 5-phase packages. [62]

3.7.2 Motor Sizing

As calculated in the roller shaft section the belt tension during starting equals

$$\boldsymbol{T_{bs}=T_b * K_s = 51.65 * 1.2 = 62\ \mathrm{N}} \tag{3.62}$$

Then the required power can be estimated to be

$$\mathbf{P_b = T_b\ v = 62 * 0.23 = 14.26\ W} \tag{3.63}$$

Due to the fact that the lowest rating in the market was 0.5 hp we had no other choice but to choose it if there were 0.25 hp I would be sufficient also.

The acceleration of the conveyor belt can be calculated as follows

$$\mathbf{A = \frac{[\ T_{bs} - \ T_b]}{L * (2m_i + 2m_b + m_m)} = \frac{0.2 * 51.65}{2.5 * (2 * 1 + 2 * 1.5 + 0)} = 0.82\ m/s^2} \tag{3.64}$$

3.7.3 Gearbox

3.7.3.1 Power Transmission

To transmit power from the motor to the belt there is two common methods which are gearboxes and belts drives. Belt drives are generally more efficient, as belts are quiet during operation, yet require frequent maintenance. Aligning belts and getting the correct tension between the pulleys can be tedious. At high torques and high temperatures you should expect slipping. If it were to fail it is not catastrophic, and only the belt will break. So, it is easily replaced. Therefore, it is used in places with unpredictable loading. It can operate at far distances from input source, setup is quite light, but the main disadvantage is that belt drives need regular inspection which is not affordable in our application so our choice was to implement a gearbox which has zero slipping, and true rotation. It also requires less to no maintenance whatsoever (needs to be oiled frequently), and it is mostly used in short distances such as our application where the motor is hanged on the conveyor frame directly. [63]

3.7.3.2 Worm gearbox

There are endless types of gearbox used in the industrial field, but in conveyor belt the demonstrating two types are worm gearbox which is shown in **Fig.3.22**, and Multi-Stage gearbox which is used in application where high gear ratio is needed, also where speed could vary between different levels mechanically not only one constant speed.

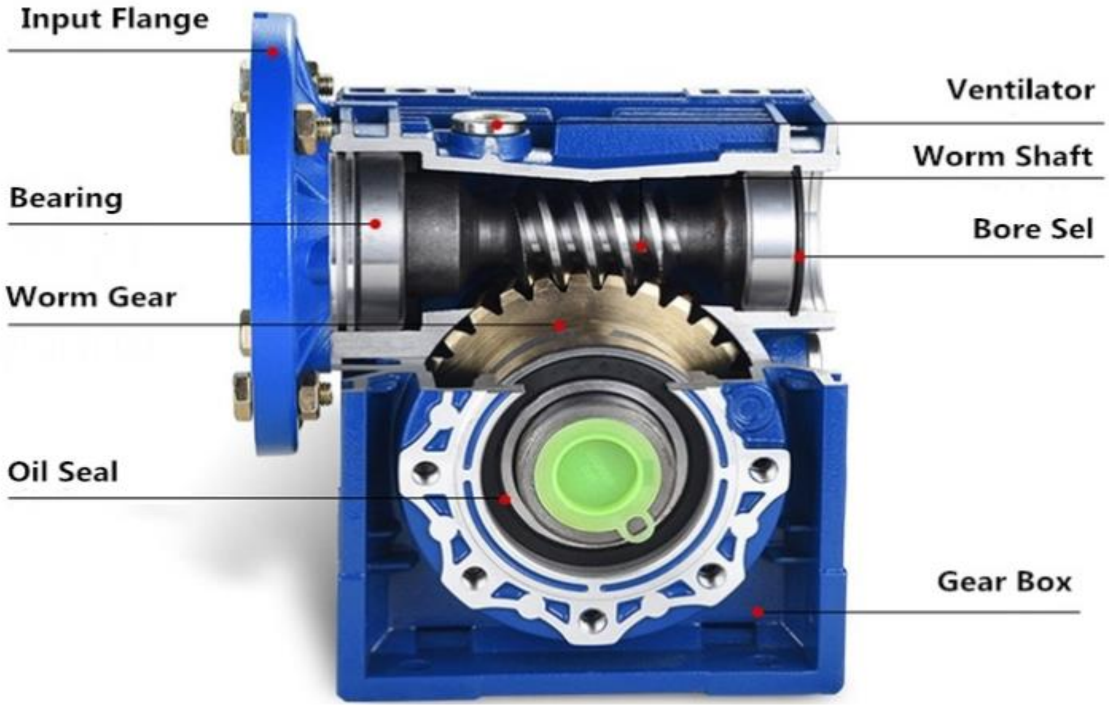

Fig.3.22: Section view of the worm gearbox.

In fact, the gears in this gearbox should be designed so that they can withstand motor torque and load inertia without failure but due to the fact that 0.5 hp motor is oversized, a worm gearbox that fits it would also be oversized. [63] So, we implemented it without design with gear ratio of 1:40 so that it moves the belt with reasonable speed which is calculated as follows:

$$\mathbf{V = \omega * \frac{1}{n} * r = 1430 * \frac{2\pi}{60} * \frac{1}{40} * 0.057 = 23\ cm/s} \tag{3.65}$$

3.8 Conveyor frame

In this section the frame is examined due to the fact that our load is very light, application is small detailed frame design is not necessary but it should be rigid and stable. It must be able to withstand all the forces acting upon it (belt tension, weight of the conveyed goods, uneven floors, etc.). and that is what we took into consideration while designing.

The frame basically consists of 3 sectors with H letter shape that are seprated equalls with 1200 mm distance all the legs and links are made from (40*40*1 mm) carbon steel tubes while the main beam is made from **Auminum profile extrusion** with section of **(40*80 mm)** as indicated in **Fig.3.23** so that it gives us folexibilty and freedom to hang objects n the frame at any position not concerning about holing as it consists of 4 rails that needs only bolt nut to fasten any object any where along the length of the conveyor such as the electrical panel which will be discussed later. [64]

Fig.3.23: 4080 Aluminum profile extrusion

3.8.1 SOLIDWORKS simulation

Fortunately this models needs no simplifications as the simlution tool in SOLIDWORKS provides easy tool to handle frames such as this by converting them into beams and use the basic analytics to solve the model forcertain forces all the user has to is to specify the material as carbon steel for the legs and alumium alloy 6061 for the extrusions, fasten the 6 legs to the ground, apply all the possible forces such as the weights of drums, belt, motor, electrical panel, and slider bed assuming that they all affect at the tips of the beam for maximium bending stress. And finally the gravitaional weight force. As for meshing it is done automatically in case of frames [64] and then the output results are as follow

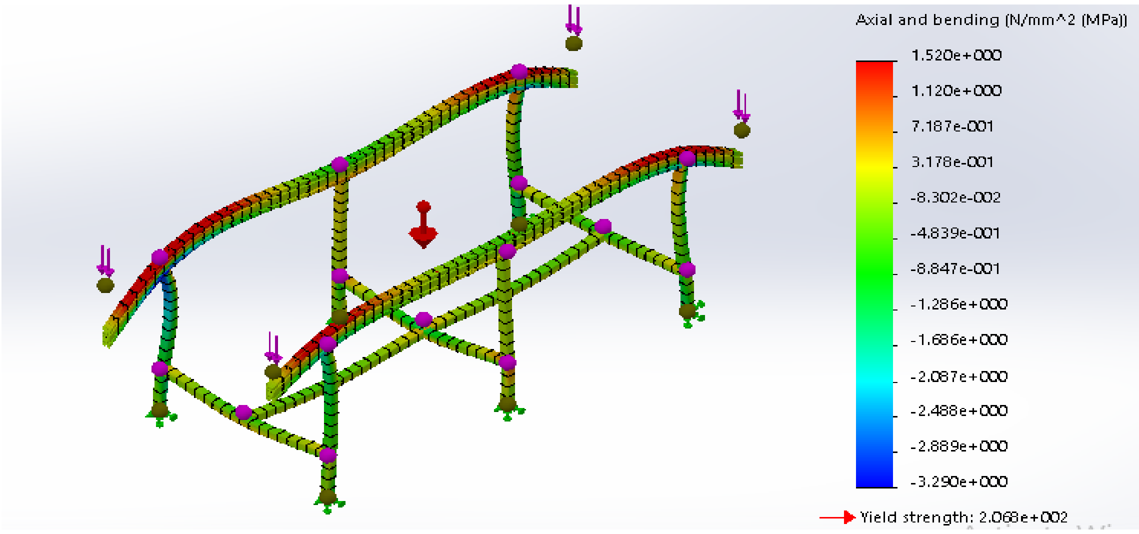

Fig.3.25: Equivlant stress on the conveyor frame.

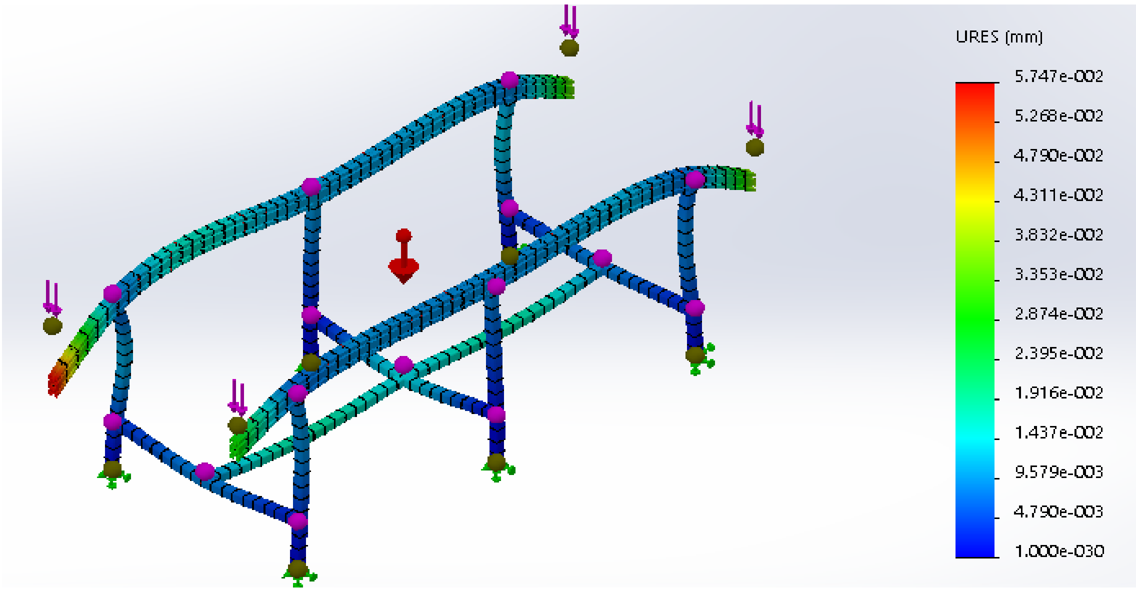

Fig.3.24: Total deformation on the conveyor frame.

Both the stress and the deformation show that the conveyor frame can easily with stand the subjected loads.