AN INVESTIGATION INTO

TENSILE FORCES OF LONG DISTANCE BELT CONVEYOR

A

THESIS

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UNDER THE GUIDANCE OF

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ii

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ABSTRACT

The focus of this thesis is to find belt tension at different locations for long distance belt conveyor under steady state operation. A long belt conveyor, often called as overland conveyor, may follow a complex conveying path and may include horizontal and vertical curves. Belt tensions at different points are not needed in case of a small in-plant belt conveyor but in case of overland conveyors due to the addition of the horizontal and vertical curves, tensions at these curves are needed for design purposes. Further, for a small belt conveyor maximum tension (which is the main criteria for design purposes) mostly occurs at the drive pulley but for overland conveyor it may occur at some other location because of possibilities of multiple inclines and declines in the conveying path. So methods for calculating belt tension of small belt conveyor may not be applicable to long and complex overland conveyor. So, this thesis work deals with a calculation procedure of the belt tension at different locations for an overland conveyor with horizontal and vertical curves and for that purpose a new computer program has been generated. The program was tested with five conveyors of different length and difference in the maximum tension was within ± 9.62% of original industrial data.

NOMENCLATURE

ΔA_e	:	Expected average idler installation deviation	-
A	:	Cross section area	(m^2)
B_C	:	Center idler roll length	(m)
b_t	:	Belt thickness	(m)
B_m	:	Modulus of elasticity of the conveyor belt	(N/m^2/ply)
BW	:	Belt width	(m)
C_{bi}	:	Sliding friction factor between idler material and belt cover	(m)
C_{im}	:	Design factor for frictional resistance due to idler misalignment	-
C_{iw}	:	Torsional load effect	(N-m/m)
Cs	:	Consolidated skirt friction and material property	(N/m^3)
C_{mz}	:	Net material friction loss factor	-
d_{ms}	:	Contact depth of material on skirting	(m)
D_p	:	Diameter of pulley	(m)
D_r	:	Idler roll diameter	(m)
d_s	:	Shaft diameter of pulley	(m)
E_0	:	Belt rubber stiffness property	(N/m^2)
F_{bc}	:	Effective normal force between cleaner and belt	(N/m)
F_D	:	De-training force	(N)
F_S	:	Stabilizing force	(N)
F_{ss}	:	Effective normal force between belt and seal	(N/m)
g	:	Acceleration due to gravity	(m/s^2)

H	:	Lift or drop	(m)
h_b	:	Belt bottom cover thickness	(m)
h_d	:	Height from which the material falls to the belt	(m)
K_{biR}	:	Viscoelastic characteristic of belt cover rubber	-
K_{is}	:	Idler seal torsional resistance	(N-m)
K_{iT}	:	Temperature correction factor	-
K_{iv}	:	Torsional speed effect	(N-m/rpm)
l_d	:	Loading of conveyor	%
L	:	Length of the flight	(m)
n_r	:	No. of idler rolls	-
N_b	:	No. of belt cleaners	-
N_{dp}	:	No. of discharge plows	-
p	:	No. of plies in the belt	-
P_{jn}	:	Belt cover indentation parameter	-
Q_R	:	Rated capacity of the conveyor	(tph)
R	:	Radius of horizontal curve	(m)
r	:	Radius of vertical curve	(m)
R_{mz}	:	Correction between actual sag and catenary sag	-
R_{pn}	:	Vector sum of weight of pulley and tension at pulley	(N)
S_i	:	Idler spacing	(m)
T	:	Belt tension	(N)
T_R	:	Rated belt tension	(N)
ΔT_{am}	:	Tension added in loading to continuously accelerate material	(N)

ΔT_{bc}	:	Tension added due to belt cleaners	(N)
ΔT_{bi}	:	Tension increase from visco-elastic deformation of belt	(N)
ΔT_{dp}	:	Tension added due to discharge plow	(N)
ΔT_{H}	:	Tension change to lift or lower the material	(N)
ΔT_{im}	:	Tension increase from idler misalignment	(N)
ΔT_{is}	:	Change in tension from idler seal friction	(N)
ΔT_{iW}	:	Change in tension from idler load friction	(N)
ΔT_{s}	:	Tension change due to bulk materials sliding on skirtboards	(N)
ΔT_{ss}	:	Tension change due to the belt sliding on skirtboard seal	(N)
ΔT_{m}	:	Tension change due to bulk materials moving between the idlers	(N)
ΔT_{pb}	:	Tension change due to pulley bearings	(N)
ΔT_{px}	:	Tension change due to belt bending on the pulley	(N)
V	:	Belt speed	(m/s)
W	:	Weight per unit length	(N)
w_i	:	Load distribution factor	-
W_{mzn}	:	Belt work needed to move material from one idler to the next	(N-m)
Ws	:	Skirtboard spacing	(m)
Δy_{sn}	:	Average belt sag for n^{th} flight as percentage of the idler spacing	(%)

Greek Symbols

α	:	Horizontal curve super elevation angle	(deg)
β	:	Idler troughing angle	(deg.)
θ_H	:	Angle of horizontal curve	(deg.)

θ_i	:	Angle of incline or decline	(deg.)
θ_{im}	:	Angle of impact of material to the belt	(deg.)
φ_s	:	Material surcharge angle	(deg.)
ρ_m	:	Bulk material density	(kg/m^3)
μ_{ss}	:	Sliding friction coefficient between belt and skirtboard seal	-
μ_{bc}	:	Friction factor between belt cleaner and the belt	-
σ_E	:	Edge stress	(N/m^2)

Subscripts

1	:	Concave curve
2	:	Convex curve
b	:	Belt
c	:	Center idler
i	:	Inner idler
n	:	n^{th} flight
m	:	Bulk material
min	:	Minimum
o	:	Outer idler

TABLE OF CONTENTS

Chapter 1: Introduction and Objectives

1.1 Introduction

Material handling is one of the most important sector of industry. Belt conveyors are beginning to be the most important parts of material handling systems because of their high efficiency of transportation (Xia and Zhang, 2011). Belt conveyors are by far the most widespread transportation system used across the mineral processing and mining industries (Wheeler and Ausling, 2007). A long distance belt conveyor, often referred as an overland conveyor may contain horizontal and vertical turns according to the terrain. The characteristic of adaptability of its conveying terrain has made overland conveyor a very popular conveying system. There are some tough hilly paths where conventional long distance bulk material conveying methods i.e. by truck or train fails but overland conveyor succeeds (Sagheer and Witt, 1994). Beside this overland conveyors is cost wise beneficial as well, an overland conveyor operation is more economical than truck haulage if the conveying distance exceeds 1 km (CEMA, 2007). Further, overland conveyors are far better than the truck transportation system if environmental aspects are considered.

One of the very important parameter while designing any belt conveyor, especially the long distance overland conveyor, is the tension in the belt which has a direct influence on many design issues as follows:

- The size or rating of belt or many conveyor components is governed by the belt tension applied to them (CEMA, 2007).
- Drives and control components must be designed to provide changes in tension needed to cause and control motion (CEMA, 2007).
- Managing minimum and maximum belt tensions is necessary for reliable and efficient conveyor operation (CEMA, 2007).

- The magnitude of the belt tension affects the belt sag and therefore the material movement (CEMA, 2007).

Now, transient behavior during starting and stopping in conveyor belts is often the cause of belt failure in long conveyors and transient analysis is most realistic when the actual static tensions are well defined and accurate (Harrison, 2008). Thus to develop a reliable technology for the industry, along with the study of transient behavior of conveyor belt, the calculation of accurate belt tension during static (steady state) operation is equally important, especially in case of overland conveyor, but there is not much data or modeling methods available in the open literature which concentrates on long distance belt conveyors and horizontal turns.

1.2 Objective

Specific objectives include:

- To develop a new procedure for calculation of belt tension for long distance conveyor (including horizontal and vertical curves) operating at steady state.
- Validate the same with actual industrial data.

Chapter 2: Literature Review

This chapter consists of brief introduction to various types of components used in the belt conveyor and reviews of previous research work done on belt conveyors.

2.1 Main Components of a Belt Conveyor

2.1.1 Belt

The belt forms the supporting and moving surface on which the bulk material moves. Following are the main types of the belt used in the industry (Exton, 2003):

Steel cord Belt

Steel cord Belt is usually used in applications where high tensions are prevalent. Thus in overland conveyors where belt tension is high, steel cord belt is the first preference. Steel cord belts are always vulcanized. The modulus of elasticity for Steel cord belting is high thus there is low stretch in the belt.

Ply or Fabric Belt

Due to lower belt modulus fabric belt is not commonly used in overland conveyor, but mostly used in plant conveyors. Ply belting can easily be joined by use of clip joints but can be vulcanized for permanent installations.

Solid Woven Belt

Due to least belt modulus solid woven belt is almost never used in overland conveyors, but mostly used in mining applications. Solid Woven belting can easily be joined by use of clip joints but can be finger spliced by using a hot vulcanizing method.

2.1.2 Idler

Idlers provide support to the belt and the load being conveyed. Idler rolls are commonly fabricated from steel tube with end disc (bearing housings) welded to the tube ends (CEMA, 2007).

Types of carrying idlers:

- Troughing carrying idlers
- Impact idlers
- Training idlers

Types of return idlers:

- Flat idlers
- Two-roll "vee" return idlers
- Return belt training idlers

2.1.3 Pulley

Increased use of belt conveyors has led industry away from custom-made wood pulleys to present development of standard steel pulleys. Most commonly used type of pulleys are drum pulleys and wing pulleys (CEMA, 2007).

2.1.4 Belt Take-ups

A takeup is used for the following purposes:

- To prevent belt slip at the drive pulley by insuring the proper amount of slack side tension.
- To insure proper belt tension at loading and other points along the conveyor which is necessary to prevent loss of troughing contour of the belt between idlers, thus avoiding spillage of the material from the belt (CEMA, 2007).
- To compensate for changes in belt length due to stretch.

Types of take-ups (Exton, 2003):

- Gravity take up
- Electric Winch take up
- Dynamic take up system
- Eddy Current Winch
- Screw take up
- Hand Winch take up
- Hydraulic winch
- VFD Winch

2.2 Review of Previous Research Work

Page et al. (1993) described the design, construction, commissioning and testing of a 3.2 km long overland belt conveyor in South Africa having 1000 t/h capacity and incorporated two 1,350 m radius horizontal curves. After the static design, dynamic analysis of the overland conveyor was carried out within AAC (Anglo American Corporation of South Africa Limited) using ACSL (Advanced Continuous Simulation Language). Arising from the dynamic analysis, a number of changes were made in the conveyor configuration to solve problems which arose. The preliminary selection of take-up tension of 14 kN was inadequate to overcome belt slip on start-up, thus the take-up tension was increased to a minimum of 20 kN. The static design calculated the conveyor start-up time to be 38 seconds. On simulating the start-up, high peak tensions were produced. These high tensions were reduced by extending the start-up time to 120 seconds.

Sagheer and Witt (1994) described an overland conveyor system used to transport lignite from a mine to the Soma power plant in Turkey. Transportation of coal presented a problem due to hilly region, transportation by trucks was costly and unsuitable because of the mountainous road. Conveyor system was required to be 8.5 km long having descent of nearly 350 m and capacity of 2000 t/h. Several variants of conveyor routes were considered and plotted in CAD. Finally, simulation program was used to calculate the belt tensions, power requirements and to predict behavior of the belt within horizontal and vertical curves.

Gallagher (2000) presented a new belt with lower friction between belt and idler, which reduced the rubber indentation in belt and thus reduced the resistance occurs due to this effect. A number of belt constructions were tested on a dynamic test machine at university of Hannover. A new belt with different rubber composition was manufactured by Goodyear using the theoretical and experimental data. The new belt was tested on two overland conveyors, first a 23000 ft long overland conveyor which was previously using 54" Flexsteel 2100 belt and second 6810 ft long overland conveyor which was using ST 3500 72" x 5/8" x1/4" belt. Keeping all other parameters constant it was found that after replacing the belt there was a decrease of 6% rolling resistance and thus 15% decrease in power consumption for the first conveyor and 10.3% decrease of power for second overland conveyor.

Lodewijks (2001) presented an overview of work done on mathematical analysis of dynamics of the belt conveyor systems. There are models for dynamic behaviour of belt conveyors and even for long distance conveyors but some models only study the dynamic behaviour in the axial or longitudinal direction of the belt. These models do not study dynamic behaviour in the transverse or vertical direction and thus do not take into account the effect of belt sag which is often the reason of breakdowns in the overland conveyor systems. Further, author predicted that in future there will be full three-dimensional models which can accurately study dynamic behaviour of the belt in horizontal curves, but size of three dimensional models could be 10 to 20 times more than the size of two dimensional models. Thus these models could be very complicated and would demand a lot of computation power.

Gilbert (2003) discussed the most common types of belts used in the long belt conveyors which are solid woven belt, plied fabric belt and steel cord belt. In underground coal mine discipline due to confined space, smaller diameter pulleys are required which favors the solid woven belt and also when cost of the conveyor is considered the solid woven belt is preferred, but it has a very low modulus of thus it is highly stretchable. Plied fabric belt is more expensive than Solid woven belt but usually has a longer life due to less wear and also modulus of elasticity is roughly twice than the solid woven belt which results in a stiffer and predictable behavior. Steel cord belt is mostly used for long conveyors that generate massive belt tensions and also it has modulus of elasticity twenty five time that of solid woven belt which makes steel cord belts extremely controllable and predictable.

Wheeler et al. (2004) presented a method for the calculation of flexure resistance of bulk solid materials transported on the belt conveyors. Theoretical methods had been presented for the prediction of pressure distribution caused by the bulk solid interactions and approximations had been developed for the longitudinal and transverse components of flexure resistance. To verify the theoretical method a series of experiments were done in which total main resistance force was measured with the help of an instrumented idler set, from which flexure resistance was calculated by subtracting remaining components from the main resistance. The results were in agreement with the analysis of bulk solid flexure resistance analysis.

Alspaugh (2004) discussed about the main factors responsible for power consumption in a belt conveyor. Power consumption in a belt conveyor of 400 m length and 12 m lift was found to be lift 43%, material flexure 21%, rubber indentation 11%, alignment 9%, idler resistance 6% and miscellaneous 10%, while power consumption for an overland conveyor of 19.1km length and 3m lift was, rubber indentation 48%, idler resistance 26%, alignment 17%, material flexure 4%, lift 1% and miscellaneous 4%. Thus for overland conveyor maximum power consumption is due to rubber indentation and idler resistance, but these can vary from conveyor to conveyor because of different number and length of horizontal and vertical curves.

Paul and Shortt (2007) discussed the maximum belt speed of idlers. In standard like SANS 1313 the maximum belt speed for idlers is 5 m/s but authors said that it is possible to run idlers at a speed greater than 5 m/s. Theoretical analysis showed that carrying idlers for certain instances cannot go more than 5 m/s but return idlers may operate at higher speeds. Further in case of idler failure the common belief is that the cause of failure was either due to bearing failure or shaft deflection. This happens because of forces arising due to unbalanced rotating mass, but result from analysis showed that while this unbalanced force does affect the maximum achievable belt speed it was not the cause of idler failure.

Lill (2007) discussed the design of pulley for belt conveyors. Most of the pulley manufacturers use traditional in-house methods or rules-of thumb for designing pulley which sometime leads to over-stressing of pulley components especially at welds. Finite element methods could give much better results, but most of the finite element programs can only use a three dimensional model for pulley which is time consuming to make and solve because many designs have to be considered before an optimized design comes. Only few FEA programs could model 3-D axi-symmetric structures as pulleys by using a two dimensional profile only. This greatly reduces the time for analysis. Considering cost and time needed for finite element analysis it is only practical if there are standardized pulley designs.

Harrison (2008) described the dynamic simulation of the conveyor belts. A computer simulation program was run on 2 km long overland conveyor having capacity of 2000 t/h and speed of 3 m/s. Start time and stop time was found to be 21.8 s and 6.6 s respectively. During starting and stopping there was an elastic wave which moved from head pulley to tail pulley, speed of this was found from tail start time delay, which was 0.9 s, so speed of wave-front came out to be 2220 m/s. Further, on stopping, the belt tension approached zero towards the tail on the carry side. Therefore Harrison suggested that the design would need an increase in the take-up pre-tension.

White (2009) discussed the advantages and disadvantages of higher belt speed in overland conveyors. Cost analysis of a 6000 m long overland conveyor was done at different speed from 4 m/s to 12 m/s and it was shown that the high speed conveying was more cost effective design over twenty year period. Another big advantage of high speed conveying is lesser indentation rolling resistance because of lesser material load on the belt. Disadvantages include difficulty in the design of idlers, its support and transfer points (chutes), increased belt wear and more installer power. Furthermore, due to higher belt speed, any defect in belt can be magnified, thus the belt needs to be accurately manufactured and improved quality splice joints should be used.

Wiid et al. (2009) presented a case study about comparing constant speed and variable speed operation for six belt conveyors at a coal fired power plant in South Africa. The artificial friction coefficient favored the constant speed operation, but still due to lower belt speed energy consumption was lower in for variable speed operation. Idler performance and life for the constant and variable speed operation was almost similar. Pulley performance was better with variable speed operation. Belt life was also considerably more for variable speed because of less wear of the belt at lower speeds, due to all these advantages, variable speed operation was chosen with the help of variable speed drive.

Wheeler and Munzenberger (2009) presented a theoretical pseudo 3D approach to predict the effect of the belt carcass properties on indentation rolling resistance for steel cord conveyor belts. Analysis of stress distribution in the conveyor belt based on the stress propagation showed increase in the bottom cover thickness. Using a two dimensional visco-elastic model (finite element) the indentation rolling resistance for steel cord belt was determined. The results showed that the indentation rolling resistance would increase mainly due to presence of steel cords and was affected by the diameter of steel cables. Further, for same loading conditions, there was higher peak stress level for smaller diameter cables, but due to lesser thickness of insulation layer the values for indentation rolling resistance were lower.

Frittella and Curry (2009) described the process for selecting the idlers. Load on idlers was found out by factors such as: mass of bulk material and belt, self mass of rollers, idler misalignment, dynamic effect, belt geometry (curves). Then load on individual roll was calculated by using a burden factor and idler roll was selected on the basis of three factors: bearing life, shaft bending stress and shaft deflection. Finally, idler base was selected mainly on the bases of maximum bending stress. Furthermore, authors compared idlers of CEMA and SANS 1313 standards and reached to the conclusion that CEMA does not take shaft deflection of idler into account which may lead to reduced roller life.

Gerard and O'Rourke (2009) discussed the key elements to consider when designing overland conveyors to ensure long trouble free life, low maintenance and cost effective operation. Conveyor temperature affected the power consumption, a lot of different rubber cover materials were shown in the paper with varying temperatures and almost every material consumed more power at lesser temperature with the exception of a few. Horizontal curves should never be negotiated by using physical restraints, which increase wear rates by adding drag to the system and thus reducing component life. Belt selection should be based on splice instead of belt breaking strength. Also, larger diameter idler rolls reduce power consumption by reducing indentation losses of the rubber cover of the belt. Further, authors added whenever possible wide idler spacing should be used as is used in Curragh North overland conveyor, which has 5 m carry side and 10 m return side idler spacing for most of its length.

Paul (2011) discussed idler configuration for overland conveyors to reduce the total cost of ownership. The three highest cost items of operating a conveyor is typically power consumption, the conveyor belt and then idlers. Average conveyor operating cost was calculated for a series of 4 conveyors and it was found that the power consumption cost was 51%, belt replacement cost was 40% and idler cost was 3%. Now the idler configuration has an impact on the belt life of the conveyor. Very deep troughed idlers, incorrectly designed loading areas and incorrect transition distances are all factors that may accelerate fatigue in belt carcass, resulting in a significant increase in cost. Different Idler arrangements were considered and it was found that the most cost effective arrangement of idlers would be to use series 30 idlers of 152 mm diameter, troughed at 20°. This however might not be practical. Further, author recommended that to decrease cost of rollers and cost of power consumption heavier idlers at increased idler centers should be used.

Brink et al. (2011) presented a case study of 890 m long troughed belt conveyor. In case of power failure or emergency the stopping time of the belt was just 3 seconds, due to such a short time there was a 30% speed difference between the belt at head and tail pulley and also there was unacceptable level of 12% maximum belt sag. Four design options were tested out which adding flywheel to each of the drive gave the best result. Stopping time of the belt was increased from three to eight seconds and there was negligible belt speed difference between head and tail pulley, also maximum tension was reduced by roughly 11% and maximum belt sag reduced to 2.6%, but this was just one case study, results could be different for a complex and long conveyor which includes horizontal turns.

Nel and human (2011) discussed the optimum idler troughing profile in regard to the life of idlers and belt. Theoretical cost analysis of belt conveyors showed that for longer conveyors the percentage contribution of belt and idlers is significantly greater than the shorter conveyors. So, for overland conveyors belt and idler profile needs careful attention. A 35° troughed conveyor at 100% of the belt loading is assumed to have roughly 67% of the load on the centre roll, but this figure goes high as the troughing angle is increased. Now idlers at the loading point during transition distance has more troughing angle and thus has increased loading at central idler. If the idler configuration is not optimum especially during the transition distance, the belt carcass can rapidly fatigue.

Lodewijks et al. (2011) discussed the effect of belt speed control on power utilization of belt conveyors with the help of two case studies. In first case study belt speed of three different belt conveyors was reduced from 4.5 m/s to 2.75 m/s by increasing loading of the conveyor and thus keeping the mass flow rate same as before. The power saving for individual conveyor was 10.9%, 15.36% and 7.2%, but it is only possible if the conveyor is not already operating at its maximum volumetric capacity. In second case study belt speed of a single conveyor was varied between 4.5 m/s and 2.11 m/s as per the fluctuating material feed and keeping the conveyor at maximum loading, which gave power saving of 6.1%.

Zamiralova and Lodewijks (2012) presented a detailed method for the calculation of indentation rolling resistance on pipe belt conveyors. A model was generated for viscoelastic behavior of the rubber of belt. The model gave an expression for indentation rolling resistance factor. Simulations for the rolling friction coefficient for 25%, 50% and 75% filling ratio of the pipe conveyor was done with belt speed variation from 0.2 m/s to 10 m/s. The results showed that with increase of the load the indentation factor kept on decreasing. Rolling friction coefficient for trough belt conveyor and for pipe belt conveyor having 25% filling ratio with the same loading was compared and the results showed that the indentation rolling resistance friction factor was more for pipe belt conveyor than the open trough belt conveyor.

Chapter 3: Tension Calculation Procedure and its Computer Program

This chapter includes the static tension calculation procedure including the horizontal and vertical curves in the belt conveyor and coding for a new computer program generated to solve the tension calculation procedure. Further, chapter includes various parameters required for the computer program, guidelines for using the program, features of the program and recommendations for the design of the belt conveyor.

3.1 Tension Calculation Procedure

This is a static (steady state operation) method for finding belt tensions at different locations in a belt conveyor. In this method conveyor is considered as a series of discrete flights or segments as shown in figure 3.1. Major advantage of dividing the conveyor in various flights is that the tension at the end or start of a flight can be found out, which is necessary in case of horizontal or vertical curve. Tension at the end of the flight is basically the algebraic sum of various changes in the tension during current flight and tension at the end of the previous flight.

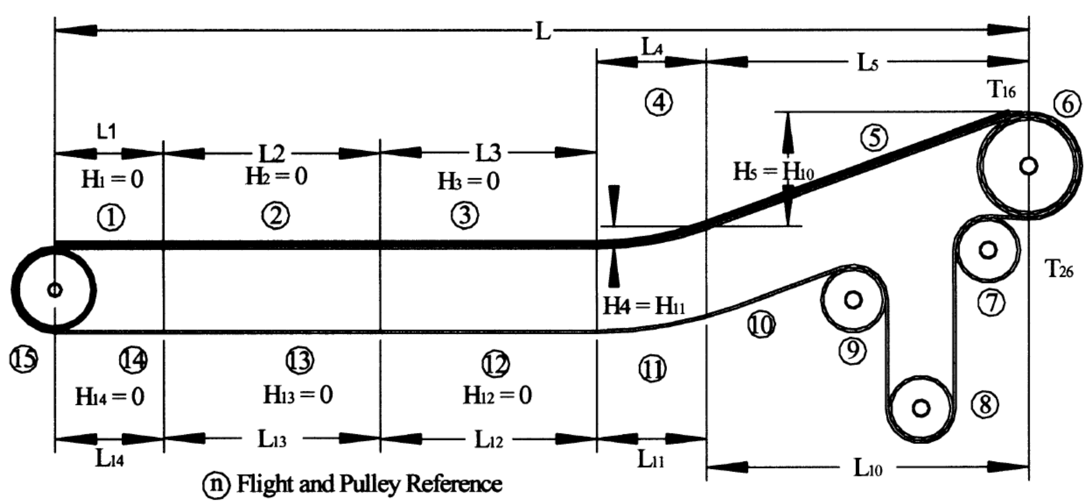

Fig. 3.1 A Typical Belt Conveyor Arrangement (CEMA, 2007)

Figure 3.1 shows a typical belt conveyor divided into sections called flights. Here each pulley is considered as separate flight. Flight 15 is tail pulley, flight 6 is head pulley, flight 7 and 9 are bend pulleys and flight 8 is take-up pulley. L_n is the length of the n^{th} flight and H_n is the lift of the n^{th} flight.

This method is applicable to the following range of parameters (CEMA, 2007):

- Maximum belt speed of 7.6 m/s
- Any length of the conveyor
- Single or multiple freely flowing load points
- Inclined, declined and/or horizontal flights with horizontal or vertical curves
- Any belt profile
- Operating temperatures between -25°F and 120°F
- Maximum Belt width of 2.44 m
- Maximum Idler spacing of 3 m
- Maximum Angle of Repose of 45°

This much range of parameters makes this method very suitable for long distance belt conveyors.

Now the tension added at n^{th} flight is given as (CEMA, 2007):

$\Delta T_n = \sum\Delta T_{Energy\ n} + \sum\Delta T_{Main\ n} + \sum T_{Point\ n}$

Where:

$\sum\Delta T_{Energy\ n} = \Delta T_{Hn} + \Delta T_{amn}$

$\sum\Delta T_{Main\ n} = \Delta T_{sn} + \Delta T_{ssn} + \Delta T_{isn} + \Delta T_{iWn} + \Delta T_{bin} + \Delta T_{mn} + \Delta T_{mzn}$

$\sum \Delta T_{Point\ n} = \Delta T_{pxn} + \Delta T_{pbn} + \Delta T_{bcn} + \Delta T_{dpn}$

Where:

ΔT_n (lbf) = Total change in belt tension to cause steady belt speed

ΔT_{Hn} (lbf) = Change in belt tension to lift or lower the material and belt

ΔT_{amn} (lbf) = Tension added in loading to continuously accelerate material to belt speed

ΔT_{ssn} (lbf) = Tension change due to the belt sliding on skirtboard seal

ΔT_{isn} (lbf) = Change in tension from idler seal friction

ΔT_{iWn} (lbf) = Change in tension from idler load friction

ΔT_{bin} (lbf) = Tension increase from visco-elastic deformation of belt

ΔT_{mn} (lbf) = Tension increase from idler misalignment

ΔT_{sn} (lbf) = Tension change due to bulk materials sliding on skirtboards

ΔT_{imn} (lbf) = Tension change due to bulk materials moving between the idlers

ΔT_{pxn} (lbf) = Tension change due to belt bending on the pulley

ΔT_{pbn} (lbf) = Tension change due to pulley bearings

ΔT_{bcn} (lbf) = Tension added due to belt cleaners

ΔT_{dpn} (lbf) = Tension added due to discharge plow

Tension change to lower or lift the material and belt

Tension change due to potential energy change in the bulk material can be calculated as (CEMA, 2007):

$$\Delta T_{Hn} = H_n \times (W_b + W_m)$$

$$H_n = \tan(\theta_{in}) \times L_n$$

Where:

W_b (lbf/ft) = Weight of belt per unit length

W_m (lbf/ft) = Weight of material per unit length

L_n (ft) = Length of the n^{th} flight of the conveyor along direction of belt

H_n (ft) = Lift or drop over the length L_n

θ_{in} (deg) = Average angle of incline or decline in direction of movement over length L_n

Now, to find W_m following relation can be used:

$$W_m = \frac{Q_R \times 2204.6 \times l_d}{V \times 60 \times 100}$$

Where:

Q_R (metric tph) = Rated capacity of the conveyor i.e. capacity of conveyor at 100% loading

l_d (%) = Loading of the conveyor

V (fpm) = Belt speed

Tension added to continuously accelerate material to belt speed

At the loading point force must provided for the acceleration of the bulk material to match its speed to the belt. This accelerating force is supplied by an increase in tension in the belt at the loading point and is given by the following relations (CEMA, 2007):

$$\Delta T_{amn} = \frac{Q_R \times 2204.6 \times l_d}{g \times 60 \times 100}(V - V_0 \times \sin(\theta_{im}))$$

$$V_0 = \sqrt{2 \times g \times h_d}$$

Where:

g (ft/min^2) = Acceleration due to gravity

θ_{im} (deg) = Angle of impact of material to the belt relative to the belt direction

h_d (ft) = Height from which the material falls to the belt

Tension change due to Bulk Materials Sliding on Skirtboard

Skirtboard is used to prevent the material spillage loss, however pressure of material against the skirtboard offers resistance to the motion of the belt, this resistance can be reflected as additional tension requirement and can be calculated using the following equations (CEMA, 2007):

$$\Delta T_{sn} = C_s \times d_{ms}^2 \times L_n$$

$$d_{ms} = \frac{A_m \times 144 - 0.25(W_s^2 - (.37 \times BW + .25)^2) \times \tan(\beta) - 0.25W_s^2\left(\frac{\varphi_s}{\sin(\varphi_s)^2} - \cot(\varphi_s)\right)}{W_s}$$

Also, $A_m = \dfrac{Q_R \times 2204.6 \times l_d}{V \times \rho_m \times 60 \times 100}$

Where:

C_s (lbf/ft-in^2) = Consolidated skirt friction and material property

d_{ms} (in) = Contact depth of material on skirting

A_m (in^2) = Material cross section area

W_S (in) = Skirtboard spacing

BW (in) = Belt width

B_C (in) = Center idler roll length

β (deg) = Idler troughing angle

φ_s = Material surcharge angle (degrees when used with trig. function and in radians when used alone

ρ_m (lbf/ft^3) = Density of the bulk material

Now value of C_S for some commonly used bulk materials can be found from following table:

Table 3.1: Value of Cs for various materials (CEMA, 2007)

Material	C_S Factor
Alumina (dry and pulverized)	0.121
Ash (dry)	0.057
Cement (Portland, dry)	0.212
Cement (clinker)	0.123
Coal (anthracite)	0.054
Coal (bituminous, mined)	0.075
Flour (wheat)	0.027
Limestone (pulverized, dry)	0.128

Tension change due to the belt sliding on skirtboard seal

Skirtboard is fastened to the conveyor with the help of skirtboard seal. The seal rides on the belt and due to friction between seal and the belt, extra tension has to be provided to maintain the same belt speed, which can be found from the following equations (CEMA, 2007):

$$\Delta T_{ssn} = C_{ss} \times L_n$$

$$C_{ss} = 2 \times \mu_{ss} \times F_{ss}$$

Where:

μ_{ss} = Sliding friction coefficient between belt and seal rubber

F_{ss} (lbf/ft) = Effective normal force between belt and seal

Typical values are μ_{ss} = 1 and F_{ss} = 3 lbf/ft.

Tension change from Idler Seal Friction (CEMA, 2007)

The idlers resist rotation due to mechanisms internal to the idler roll and thus resist the belt movement. This resistance is independent of the loading. Tension needed to overcome this resistance can be found as:

$$\Delta T_{isn} = \left[\left(\frac{3.82 \times V}{D_r} - 500\right) K_{iv} + K_{is}\right] \times \frac{1}{\frac{D_r}{2}} \times \frac{K_{iT} \times n_r}{S_i} \times L_n$$

Where:

K_{iV} (in x lbf/rpm) = Torsional speed effect

K_{is} (in x lbf) = Seal torsional resistance per roll at 500 rpm

K_{iT} = Temperature correction factor

n_r = number of rolls per idler set

D_r (in) = Idler roll diameter

S_i (ft) = Idler spacing

Now, $K_{iT} = 1.114 \times 10^{-8} \times T_F^4 - 3.763 \times 10^{-6} \times T_F^3 + 4.458 \times 10^{-4} \times T_F^2 - 2.136 \times 10^{-2} \times T_F + 1.333$

Where:

T_F (^oF) = Ambient operating temperature

Values of K_{iV} and K_{is} can be found from following table:

Table 3.2: Values of K_{iV} and K_{is} for different idlers (CEMA, 2007)

Idler Series	**K_{is} (in x lbf)**	**K_{iV} (in x lbf/rpm)**
B	3	0.004
C	3.25	0.004
D	4	0.003

E	7.25	0.003

Tension change from Idler Load Friction

Along with load independent idler friction resistance, the resistance of idler to rotation is also affected by the bulk material load and the type of bearing used. The resulting tension added can be calculated as (CEMA, 2007):

$$\Delta T_{iWn} = \frac{C_{iW} \times (W_b + W_m)}{\frac{D_r}{2}} \times L_n$$

Where:

C_{iw} (in x lbf/lbf) = Torsional load effect

Value of C_{iw} can be found from following table:

Table 3.3: Value of C_{iw} for different idlers and bearings (CEMA, 2007)

Idler Series	Taper Roller	Deep Groove Ball
B	0.00155	0.00125
C	0.0017	0.00145
D	0.0017	0.00185
E	0.0029	0.00255

Tension increase from visco-elastic deformation of belt cover (CEMA, 2007)

As the belt moves over an idler the bottom belt cover is squeezed between the belt carcass and the idler roll and it deforms under this pressure. As the belt regains its shape a portion of energy is absorbed by the belt cover as heat. This is also called rubber indentation loss and the tension added due to this loss of energy can be calculated from the following equations:

$$\Delta T_{bin} = K_{biR} \times P_{jn} \times (W_b + W_m) \times w_i \times L_n$$

$$P_{jn} = \left[\frac{(W_b + W_m) \times S_i \times h_b}{E_0 \left(\frac{D_r}{2}\right)^2 \times BW}\right]^{\frac{1}{3}}$$

Where:

K_{biR} = Viscoelastic characteristic of belt cover rubber

P_{jn} = Cover indentation parameter

w_i = Load distribution factor

h_b (in) = Belt bottom cover thickness

E_0 (psi) = Rubber stiffness property

Value of w_i can be found from following table:

Table 3.4: Value of w_i at various troughing angles (CEMA, 2007)

Troughing Angle	20 deg.	35 deg.	45 deg.	Flat
$\mathbf{w_i}$	1.265	1.406	1.465	1

Now, $K_{biR} = b_0 + b_1 \times [1 + \tanh[b_2 + b_3 \times (\log(V) + aT_{ex})]]$

$$aT_{ex} = a_0 + a_1 \times T_F + a_2 \times T_F^2 + a_3 \times T_F^3 + a_4 \times T_F^4 + a_5 \times T_F^5$$

Further, values of constant coefficients, a_n and b_n at E_0 = 1644 psi can be found from following table:

Table 3.5: a_n and b_n values for fabric and steel cable belts (CEMA, 2007)

	Fabric Belts		**Steel Cable Belts**	
n	$\mathbf{a_n}$	$\mathbf{b_n}$	$\mathbf{a_n}$	$\mathbf{b_n}$
0	-2.56×10^{-2}	0.072	-2.56×10^{-2}	0.140

1	-5.74×10^{-2}	0.029	-5.74×10^{-2}	0.029
2	1.06×10^{-4}	-1.75	1.06×10^{-4}	-1.75
3	-2.16×10^{-6}	1	-2.61×10^{-6}	1
4	3.2×10^{-8}		3.2×10^{-8}	
5	-1.03×10^{-10}		-1.03×10^{-10}	

Tension increase from Idler Misalignment

Idlers axes are often considered perpendicular to the direction of the belt travel. However, unless installed very precisely, in most of the cases a small misalignment angle exists which causes a very small transverse slip between belt and the idler. Tension change due to this idler misalignment can be calculated from the following equations (CEMA, 2007):

$$\Delta T_{imn} = C_{im} \times L_n (W_b + W_m)$$

$$C_{im} = \frac{C_{bi} \times \Delta A_e}{BW + 9}$$

Where:

C_{im} = Design Factor for frictional resistance due to idler misalignment

C_{bi} (in) = Sliding Friction Factor between idler material and belt cover

ΔA_e = Expected average idler installation deviation

Now, C_{bi} = 0.5 for steel roll on rubber belt cover

C_{bi} = 0.75 for rubber roll on rubber belt cover

Tension change due to material trampling loss (CEMA, 2007)

Due to the weight of the bulk material there is belt sag between two consecutive idler sets. Now as the material moves from one idler to another due to the belt sag the individual particles of material collides with each other, resulting in loss of energy due to internal friction. It is also called material trampling loss and tension added due to this loss can be calculated as:

$$\Delta T_{mzn} = \frac{W_{mzn} \times L_n}{S_i^2}$$

$$W_{mzn} = \frac{1}{12^4} \times d_m^3 \times \rho_m \times C_{mz} \times BW \times \left[\exp\left(W_b + W_m \times \frac{S_i}{T_n}\right) - 1\right] \times \exp\left[\frac{-1}{2} \times (W_b + W_m) \times \frac{S_i}{T_n}\right] \times R_{mz}$$

Where:

W_{mzn} (lbf-ft) = Belt work needed to move material from one idler to the next

d_m (in) = maximum material depth at center of the belt

C_{mz} = Net material friction loss factor

T_n (lbf) = Average tension in n^{th} flight

R_{mz} = Correction between actual sag and catenary sag

To find average tension in the flight (T_n), first a value of Tn is assumed then iterations are needed until assumed value of T_n comes equal to calculated value of T_n.

For Troughed Fabric Carcass Belts:

$$R_{mz} = \frac{1}{12} \times \exp\left[4.181 - 1.572 \times \left(\frac{BW}{S_i \times 12}\right)^{1.5} - 1.0827 \times \Delta y_{sn}^{0.5}\right]$$

For Troughed Steel Cable Belts:

$$R_{mz} = \frac{1}{12} \times \exp\left[4.966 - 4.071 \times \left(\frac{BW}{S_i \times 12}\right)^{0.5} - (1.062 \times 10^{-2}) \times \Delta y_{sn}^{-1}\right]$$

For all Flat Belt Conveying: $R_{mz} = \frac{1}{12}$

Where:

Δy_{sn} (%) = Average belt sag for nth flight as a percentage of the idler spacing

Now, $\Delta y_{sn} = \dfrac{S_i \times (W_b + W_m)}{8 \times T_n} \times 100$

Value of C_{mz} can be found from the following table:

Table 3.6: Value of C_{mz} at various angle of repose (CEMA, 2007)

Material Flowability	Angle of Repose (deg)	C_{mz}
Very Free Flowing	0 to 19	1.5
Average Flowing	20 to 25	2.1
Average Flowing	26 to 29	2.5
Average Flowing	30 to 34	3.3
Average Flowing	35 to 39	4.2
Sluggish	40 to 45	5.7

Tension added due to belt cleaners

Belt cleaners add resistance to the motion of the belt as they scrape on belt. The tension addition due to the resistance can be calculated as:

$\Delta T_{bcn} = N_b \times BW \times \mu_{bc} \times F_{bc}$ (CEMA, 2007)

Where:

N_b = No. of belt cleaners

μ_{bc} = Friction factor between belt cleaner and the belt

F_{bc} (lbf/in) = Effective normal force between cleaner and belt

Common values for μ_{bc} and F_{bc} are 1.0 and 5.0 lbf/in respectively.

Tension added due to belt discharge plows (CEMA, 2007)

Discharge plows are used for discharging material off the sides of the belt. This results in addition belt tension requirement to overcome the change in kinetic energy of the discharged material and frictional resistance between discharge plow and the belt, which can be calculated as:

$$\Delta T_{dpn} = 8 \times BW$$

If more than one discharge plow is used then following relation should be used:

$$\Delta T_{dpn} = N_{dp} \times 0.6 \times 8 \times BW$$

Where:

N_{dp} = No. of discharge plows

Resistance due to Pulleys (CEMA, 2007)

Total resistance over a pulley can be found as:

$$\Delta T_{pn} = \Delta T_{pxn} + \Delta T_{pbn}$$

Where:

ΔT_{pxn} (lbf) = Resistance due to belt slip and bending

ΔT_{pbn} (lbf) Pulley bearing rotating resistance

Now, for Fabric Belts:

$$\Delta T_{pxn} = 9 \times BW\left(0.8 + .01\frac{T_n}{BW}\right) \times \frac{b_t}{D_{pn}}$$

For Steel Cable Belts:

$$\Delta T_{pxn} = 12 \times BW\left(1.142 + .01\frac{T_n}{BW}\right) \times \frac{b_t}{D_{pn}}$$

Where:

T_n (lbf) = Average Tension in the n^{th} flight

b_t (in) = Belt thickness

D_{pn} (in) = Diameter of n^{th} pulley

Further, $\Delta T_{pbn} = 0.01 \times \frac{d_{sn}}{D_p} \times R_{pn}$

Where:

d_{sn} (in) = Shaft diameter of n^{th} pulley

R_{pn} (lbf) = Vector sum of tension and weight of n^{th} pulley

Horizontal Curves

Horizontal curve is any circular arc in the conveyor profile in the horizontal plane. It is mainly used in long conveyors to take advantage of desirable terrain or avoid undesirable terrain like populated areas. Horizontal curves are negotiated by use of idlers banked transversely to resist the natural tendency of the tensioned belt to run in a straight line (Gerard et al.**,** 2009).

For tension calculation and designing of horizontal curve, it is considered a separate flight, length (L_H) of which can be found out by using the relation:

$$L_H = \frac{R \times \theta_H \times \pi}{180}$$

Where R (ft) is the radius of the horizontal curve and θ_H is angle of horizontal curve i.e. angle between the lines joining center of the curve and starting and ending points of the curve.

During horizontal turn a destabilizing force called de-training force acts on belt towards inside side of the curve, which results in problems including belt misalignment, belt cover wear, belt edge deterioration, accelerated idler roll wear, material shifting and even material spillage. The de-training force (F_D) can be expressed as (CEMA, 2007):

$$F_D = \frac{S_i \times T_H}{R}$$

Where T_H (lbf) is tension in the belt at relevant section and S_i (ft) is the idler spacing during the curve.

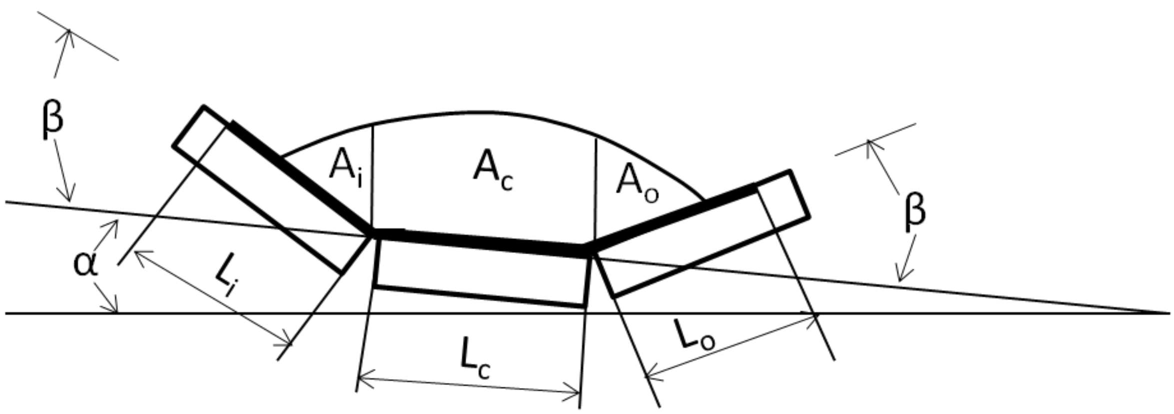

Fig. 3.2 Idler profile during a horizontal curve

The distribution of forces between each of the three individual rolls can be proportioned and defined by the following equations (CEMA, 2007):

$$F_{Di} = \frac{L_i}{BW} \times F_D$$

$$F_{Do} = \frac{L_o}{BW} \times F_D$$

$$F_{Dc} = \frac{L_c}{BW} \times F_D$$

Where L_i (in), L_o (in), L_c (in) are the length of the inner, outer and central idler roll respectively in contact with the belt during the horizontal turn

Further, normal and parallel components of the de-training force for individual roll in an idler set can be calculated from the following equations:

Normal Forces (CEMA, 2007):

$$F_{DNi} = F_{Di} \times \text{Sin}(\beta + \alpha)$$

$$F_{DNo} = F_{Do} \times \text{Sin}(\beta - \alpha)$$

$$F_{DNc} = F_{Dc} \times \text{Sin}(\alpha)$$

Where α (deg.) is horizontal curve super elevation angle

Parallel Forces (CEMA, 2007):

$$F_{DPi} = F_{Di} \times \text{Cos}(\beta + \alpha)$$

$$F_{DPo} = F_{Do} \times \text{Cos}(\beta - \alpha)$$

$$F_{DPc} = F_{Dc} \times \text{Cos}(\alpha)$$

Tests and practical experience have shown that the total parallel force is an accurate quantification of the total destabilizing or de-training force. Therefore total de-training force can be expressed as (CEMA, 2007):

$$F_{DP} = F_{DPi} + F_{DPo} + F_{DPc}$$

The counteracting or stabilizing force is generated by the weight of the belt and material plus from friction between the belt and idler rolls. The normal force generated by the weight of the belt (F_{Sb}) to each individual roll is given as (CEMA, 2007):

$$F_{Sb} = \frac{S_i \times W_b}{BW}\left[\left(L_i \times \text{Sin}(\beta + \alpha)\right) + \left(L_c \times \text{Sin}(\alpha)\right) - \left(L_o \times \text{Sin}(\beta - \alpha)\right)\right]$$

The force normal to the idler rolls due to weight of the material (F_{Sm}) can be expressed by the equation (CEMA, 2007):

$$F_{Sm} = S_i \times \rho\left[\left(A_i \times \mathrm{Sin}(\beta + \alpha)\right) + \left(A_c \times \mathrm{Sin}(\alpha)\right) - \left(A_o \times \mathrm{Sin}(\beta - \alpha)\right)\right]$$

Where A_i (ft^2), A_c (ft^2), A_o (ft^2) are the material cross sectional area reacting on inner, central and outer roll respectively and ρ (lb/ft^3) is the bulk density of the material.

The restoring or stabilizing force from friction between the belt and the rolls of the idler is the parallel force and can be determined by following equation (CEMA, 2007):

$$\begin{aligned} F_{Sf} = \mu_i &\left[\left(\left(\frac{L_i}{BW} W_b\right) + (S_i \times A_i \times \rho)\right) \mathrm{Cos}(\beta + \alpha) + F_{DNi}\right] \\ &+ \mu_c\left[\left(\left(\frac{L_c}{BW} W_b\right) + (S_i \times A_c \times \rho)\right) \mathrm{Cos}(\alpha) + F_{DNc}\right] \\ &- \mu_o\left[\left(\left(\frac{L_o}{BW} W_b\right) + (S_i \times A_o \times \rho)\right) \mathrm{Cos}(\beta - \alpha) + F_{DNo}\right] \end{aligned}$$

Where μ_i, μ_c, μ_o are the appropriate friction factor between the belt and the inner, central and outer roll.

Total stabilizing force (F_{ST}) can be expressed as:

$$F_{ST} = F_{Sb} + F_{Sm} + F_{Sf}$$

Vertical Curves

Vertical curves in belt conveyors are used to connect two tangent portions which are on different slopes. They are of two basically different types: concave vertical curves, where the belt is not restrained from lifting off the idlers; and convex vertical curves, where the belt is restrained by the idlers (CEMA, 2007).

Concave vertical curve (CEMA, 2007)

If the center of the curvature of the vertical curve lies above the belt it is called concave vertical curve. The minimum radius of the curve to prevent the belt from lifting off the idlers is given by the equation:

$$r_{1min} = \frac{1.11\ T_V}{W_b} \tag{3.1}$$

Where T_V (lbf) is the tension at the beginning of the vertical curve

However, two undesirable conditions may exist. First involves the possibility that tension at the center of the belt may exceed the allowable tension in the belt. The second is if the tension in the belt is too low there is a possibility of the belt edges to buckle.

Following formula for the minimum radius of the curve is used to prevent exceeding the tension at the center of the belt beyond the rated tension of the belt:

$$r_{1min} = \frac{B_m \times BW^2 \times p}{T_R - T_V} \times \frac{\mathrm{Sin}(\beta)}{54} \tag{2.2}$$

Where T_R (lbf) is the rated belt tension

B_m (lbf/in-width/ply) is modulus of elasticity of the conveyor belt

Following formula for the minimum radius of the curve is used to make sure that at the belt edges tension is sufficiently high to avoid zero tension:

$$r_{1min} = \frac{B_m \times BW^2 \times p}{T_V - (\sigma_E \times BW)} \times \frac{\mathrm{Sin}(\beta)}{54} \tag{2.3}$$

Where:

r_{1min} (ft) is curve radius

p is number of plies in the belt

σ_E (lbf/in) is Edge stress

For steel cord belt, $\sigma_E = 75 - 1.5 \times \frac{T_V}{BW}$

For fabric belt, $\sigma_E = 30$

Now the largest calculated value of minimum radius of curve using equation (2.1), (2.2) and (2.3) should be used for design purposes.

Convex vertical curves (CEMA, 2007)

In a convex vertical curve the center of the curvature of the curve lies below the belt. In case of a convex curve the belt edges are more stressed than the center of the belt because belt edges have higher radius of curvature than belt center. Minimum radius of curve to prevent overstress of belt edges can be found from equation:

$$r_{2min} = \frac{B_m \times BW^2 \times p}{T_R - T_V} \times \frac{\sin(\beta)}{54} \tag{2.4}$$

In the low belt tension zone during the convex curve, tensile stress may become less than zero at the center of the belt, which may result in spillage of bulk material of buckling in the belt. Minimum radius of curve to prevent these undesirable scenarios can be found from following equation:

$$r_{2min} = \frac{B_m \times BW^2 \times p}{T_V - 30 \times BW} \times \frac{\sin(\beta)}{54} \tag{2.5}$$

Again the largest calculated value of minimum radius of convex curve using equation (2.4) and (2.5) should be used for design purposes.

This tension calculation procedure is long, complex and requires iterations so it is nearly impossible to solve it manually. So a new computer program has been generated in MATLAB to solve it in an efficient manner.

3.2 MATLAB script of the Program

Following is the MATLAB script of the written program:

```
n=input('Enter no. of flights: ');
Cs=0; % preallocation (for skirtboard)
mis=0; % preallocation (misalignment)
r=0; % preallocation (carrying or return side)
drive=0; % preallocation (Drive)
pulley=0; % preallocation (pulley)
Hturn=0; % preallocation (Horizontal turn)
Vcurve=0; % preallocation (Vertical curve)
takeup=input('Enter the Take-up Tension (lbf): ');
for i=1:n
    if drive==1&&r==0
        rs=input('Is this flight return side? (y/n): ','s');
        if rs=='y'||rs=='Y'
            r=1;
        else drive=0;
        end
    end
    fprintf('Details of flight %1.0f: \n',i);
    hort=input('Is there Horizontal turn in the flight? (y/n): 
','s');
    if hort=='y'||hort=='Y'
        HT(i)=1;
    else HT(i)=0;
    end
```

```
        if HT(i)==1
            R=input('Enter Radius of the horizontal turn (ft): ');
            hta=input('Enter Horizontal curve angle (deg): ');
            ea=input('Enter Super Elevation Angle for horizontal
turn (deg.): ');
            Lf(i)=R*hta*pi/180;
        end
        if HT(i)==1&&Hturn==0
            Li=input('Enter Lenght of inside Idler in contact with
belt (in): ');
            Lo=input('Enter Lenght of outside Idler in contact with
belt (in): ');
            Lc=input('Enter Lenght of center Idler in contact with
belt (in): ');
            Hturn=1;
        end
        if HT(i)==0
    Lf(i)=input('Enter Length of the flight (ft): ');
        end
if i==1
Q=input('Enter Rated capacity of the conveyor (tph): ');
V=input('Enter Belt speed (fpm): ');
Wb=input('Enter Weight of the belt (lbf/ft): ');
loading=input('Enter loading of conveyor (%): ');
Wm=Q*2204.6*loading/(V*60*100);
end
if Lf(i)==0
    Lh(i)=0;
else
Lh(i)=input('Enter Length of the conveyor having lift or drop in the
current flight (ft): ');
end
if Lh(i)~=0
As(i)=input('Enter Angle of incline(+) or decline(-) for the same
length (deg): ');
```

```
else As(i)=0;
end
if r==1
    Wm=0;
end
Th(i)=tan(As(i)*pi/180)*Lh(i)*(Wb+Wm);  % change in belt tension due
to lift or drop
ThN(i)=4.44822*Th(i); % for changing into S.I. units
if Th(i)~=0
fprintf('Change in belt tension due to lift or drop = %2.3f lbf =
%2.3f N\n',Th(i),ThN(i))
end
if i==1
g= 115920;  % acceleration due to gravity(ft/min2)
Ai=input('Enter Angle of impact of material to the belt relative to
the belt direction (deg): '); % Angle b/w normal of belt and chute
wall or natural trajectory
if Ai~=0
    hd=input('Enter Height from which material falls on the belt
(ft): ');
    Tam(i)=Q*loading*2204.6*(V-
sqrt(2*g*hd)*sin(Ai*pi/180))/(60*100*g); % Tension added at loading
point to continously accelerate material to belt speed
else
    Tam(i)=Q*loading*2204.6*V/(60*g*100);
end
TamN(i)=4.44822*Tam(i);
fprintf('Tension added at loading point to continously accelerate
material to belt speed = %2.3f lbf = %2.3f N\n',Tam(i),TamN(i))
else
    Tam(i)=0;
end
if Lh(i)~=0
vcurv=input('Is there vertical curve in the flight? (y/n): ','s');
if vcurv=='Y'||vcurv=='y'
```

```
    vu=input('Type of vertical curve:\nPress 1 for concave curve
\nPress 2 for convex curve: ');
    if i==1
        Tver=takeup;
    else Tver=CT(i-1);
    end
end
if vcurv=='Y'||vcurv=='y'&&Vcurve==0
    plies=input('Enter no. of plies in the belt :');
    BulkM=input('Enter Modulus of Elasticity of the conveyor belt
(lbf/in-width/ply) :');
    Trated=input('Enter Rated Belt Tension (lbf):');
    vcurve=1;
end
end
if r==0&&Lf(i)~=0  % to check if it is carrying side because there
in no skirtboard in return side
Ls(i)=input('Enter Length of Skirtboard(ft): ');
if Ls(i)~=0
    Tss(i)= 2*1*3*Ls(i); % Tension increase due to Skirtboard Seal
Friction
                      % 2 is for both sides, 1 is frictional coeff.
for rubber seal, 3 lbf/ft is normal effective force for rubber seal
    TssN(i)=4.44822*Tss(i);
fprintf('Tension increase due to Skirtboard Seal Friction = %2.3f
lbf = %2.3f N\n',Tss(i),TssN(i))
else
    Tss(i)=0;
end
else Tss(i)=0;
    Ls(i)=0;
end
if i==1
Dr=input('Enter Idler roll diameter (in): ');
nr=input('Enter no. of rolls per idler set: ');
```

```
Tc=input('Enter Ambient operating temperature(°C): ');
Is=input('Enter Idler Series (B,C,D,E): ','s');
if Is=='B'||Is=='b'
    Kis=3;
    Kiv=.004;
elseif Is=='C'||Is=='c'
    Kis=3.25;
    Kiv=.004;
    elseif Is=='D'||Is=='d'
        Kis=4;
        Kiv=.004;
        elseif Is=='E'||Is=='e'
        Kis=7.25;
        Kiv=.003;
end
end
if r==1
    nr=input('Enter no. of rolls per idler set: '); % Because return
side may have different no. of rolls per idler set
end
if Lf(i)~=0
Si(i)=input('Enter Spacing of idler sets for current flight (ft):
');
Kit=1.114*10^-8*(Tc*9/5+32)^4-3.763*10^-6*(Tc*9/5+32)^3+4.458*10^-
4*(Tc*9/5+32)^2-2.136*10^-2*(Tc*9/5+32)+1.333;      %     Temperature
correction factor
Tis(i)=((3.82*V/Dr-500)*Kiv+Kis)*2*Kit*nr*Lf(i)/Dr/Si(i);           %
Resisitance due to Idler Seal Drag
TisN(i)=4.44822*Tis(i);
fprintf('Tension change due to Idler Seal Drag = %2.3f lbf = %2.3f
N\n',Tis(i),TisN(i))
else Tis(i)=0;
end
if i==1
```

```
bu=input('Type of idler bearing used:\nPress 1 for Taper Roller
\nPress 2 for Deep Groove Ball: ');
if Is=='B'||Is=='b'
    if bu==1
        Ciw=.00155;
    else Ciw=.00125;
    end
elseif Is=='C'||Is=='c'
    if bu==1
        Ciw=.0017;
    else Ciw=.00145;
    end
elseif Is=='D'||Is=='d'
    if bu==1
        Ciw=.0017;
    else Ciw=.00185;
    end
elseif Is=='E'||Is=='e'
    if bu==1
        Ciw=.0029;
    else Ciw=.00255;
    end
end
end
if Lf(i)~=0
Tiw(i)=Ciw*(Wb+Wm)*2*Lf(i)/Dr;  % Resistance due to Idler Bearing
Losses
TiwN(i)=4.44822*Tiw(i);
fprintf('Tension change due to Idler Bearing Losses = %2.3f lbf =
%2.3f N\n',Tiw(i),TiwN(i))
else Tiw(i)=0;
end
if i==1
hb=input('Enter Belt bottom cover thickness(in): ');
Ta=input('Enter Troughing Angle(0,20,35,45 deg): ');
```

```
if Ta==0
    wi=1.14;
elseif Ta==20
    wi=1.28;
elseif Ta==35
    wi=1.36;
elseif Ta==45
    wi=1.39;
end
BW=input('Enter Belt Width(in): ');
E=1644; % rubber stiffness property(psi)
belt=input('Type of belt: \nPress 1 for Fabric Belt \nPress 2 for
Steel Cable Belt: ');
if belt==1
    a=[-2.56e-02,-5.74e-02,1.06e-04,-2.61e-06,3.20e-08,-1.03e-10];
    b=[.072,.029,-1.750,1];
end
if belt==2
    a=[-2.56e-02,-5.74e-02,1.06e-04,-2.61e-06,3.20e-08,-1.03e-10];
    b=[.14,.029,-1.75,1];
end
Texp=a(1)+a(2)*(Tc*9/5+32)+a(3)*(Tc*9/5+32)^2+a(4)*(Tc*9/5+32)^3+a(5
)*(Tc*9/5+32)^4+a(6)*(Tc*9/5+32)^5;
KbiR=b(1)+b(2)*(1+tanh(b(3)+b(4)*(log10(V)+Texp)));  % viscoelastic
characterstic of belt cover rubber
end
if Lf(i)~=0&&r==0
Pj=((Wb+Wm)*Si(i)*hb*4/(E*Dr*Dr*BW))^(1/3);
Tbi(i)=KbiR*Pj*(Wb+Wm)*wi*Lf(i);  % Tension increase due to rubber
indentation losses
TbiN(i)=4.44822*Tbi(i);
fprintf('Tension increase due to rubber indentation losses = %2.3f
lbf = %2.3f N\n',Tbi(i),TbiN(i))
else Tbi(i)=0;
end
```

```
if i==1
  misa=input('Is there idler misalignment (y/n): ','s');
  if misa=='y'||misa=='Y'
  mis=1;
  end
if mis==1
Aei=input('Expected average idler installation deviation (in): \n0.375 for permanent rigid structure \n0.5 when installed without alignment measurement \n0.75 in. when mounted on independent, imprecise footings \n1.5 for movable or unstable footing, roof hung and other difficult installation conditions: ');
Ae=Aei+.1;
eim=Ae/(BW+9);
IM=input('Idler material: \nPress 1 for Steel \nPress 2 for Rubber: ');
if IM==1
    Cbi=0.5;
end
if IM==2
    Cbi=0.75;
end
end
end
if mis==1&&Lf(i)~=0
Tim(i)=Cbi*eim*Lf(i)*(Wb+Wm); % Tension loss from idler misalignment
TimN(i)=4.44822*Tim(i);
fprintf('Tension loss from idler misalignment = %2.3f lbf = %2.3f N\n',Tim(i),TimN(i))
else Tim(i)=0;
end
if i==1
    densm=input('Enter Density of the bulk material (lbf/ft3): ');
    Am=Q*2204.6*loading/(V*densm*60*100);
    msa=input('Enter Material surcharge angle (deg): ');
end
```

```
if Ls(i)~=0
    if Cs==0
    mat=input('Material to be conveyed: \nPress 1 for Alumina(pulverized & dry)\nPress 2 for Bauxite \nPress 3 for Cement(portland) \nPress 4 for Cement clinker \nPress 5 for Coal Ash \nPress 6 for Coal(anthracite) \nPress 7 for Coal(Bituminous)\nPress 8 for Wheat \nPress 9 for Iron ore \nPress 10 for Limestone \nPress 11 for Phosphate \nPress 12 for Sugar \nPress 13 to directly enter value of friction factor between skirtboard and bulk material : ');
    Ws=input('Enter Skirtboard spacing (in): ');
    dms=(Am*144-.25*(Ws^2-(.371*BW+.25)^2)*tan(Ta*pi/180)-0.25*Ws^2*(msa*pi/180/sin(msa*pi/180)^2-cot(msa*pi/180)))/Ws;
    if mat==1
        Cs=.121;
    elseif mat==2
        Cs=.188;
    elseif mat==3
        Cs=.212;
    elseif mat==4
        Cs=.123;
    elseif mat==5
        Cs=.057;
    elseif mat==6
        Cs=.054;
    elseif mat==7
        Cs=.075;
    elseif mat==8
        Cs=.043;
    elseif mat==9
        Cs=.276;
    elseif mat==10
        Cs=.128;
    elseif mat==11
        Cs=.018;
    elseif mat==12
```

```
        Cs=.034;
    elseif mat==13
        Cs=input('Enter friction factor between skirtboard and bulk 
material: ');
    end
    end
    Ts(i)=Cs*dms^2*100*Ls(i);  % Tension change due to material 
sliding on skirtboard
    TsN(i)=4.44822*Ts(i);
    fprintf('Tension change due to material sliding on skirtboard = 
%2.3f lbf = %2.3f N\n',Ts(i),TsN(i))
else Ts(i)=0;
end
if Lf(i)~=0
beltcln=input('Enter No. of belt cleaners in the current flight: ');
    Tbc(i)=5*BW*beltcln;
    TbcN(i)=4.44822*Tbc(i);
    if Tbc(i)~=0
fprintf('Tension added due to belt cleaners = %2.3f lbf = %2.3f 
N\n',Tbc(i),TbcN(i))
    end
else
    Tbc(i)=0;
end
if Lf(i)~=0
    beltplow=input('Enter No. of belt discharge plows in the current 
flight: ');
end
if beltplow==1
    Tbp(i)=8*BW;
    TbpN(i)=4.44822*Tbp(i);
    fprintf('Tension added due to belt discharge plows = %2.3f lbf = 
%2.3f N\n',Tbp(i),TbpN(i))
elseif beltplow>1
    Tbp(i)=8*BW*beltplow*0.6;
```

```
    TbpN(i)=4.44822*Tbp(i);
fprintf('Tension added due to belt discharge plows = %2.3f lbf =
%2.3f N\n',Tbp(i),TbpN(i))
else
    TbpN(i)=0;
end
if i==1
bc=(0.371*BW+0.25)/BW;
cona=cos(Ta*pi/180)^2/sin(msa*pi/180)^2*(msa*pi/180-
sin(msa*pi/180)*cos(msa*pi/180))+cos(Ta*pi/180)*sin(Ta*pi/180);
conb=bc*sin(Ta*pi/180)+bc*cos(Ta*pi/180)/sin(msa*pi/180)^2*(msa*pi/1
80-sin(msa*pi/180)*cos(msa*pi/180));
conc=-144*Am/BW^2+.25*bc^2/sin(msa*pi/180)^2*(msa*pi/180-
sin(msa*pi/180)*cos(msa*pi/180));
bwmc=(-conb+(conb^2-4*cona*conc)^.5)/(2*cona);
dm=(bwmc*sin(Ta*pi/180)+(bc/2/sin(msa*pi/180)+cos(Ta*pi/180)*bwmc/si
n(msa*pi/180))*(1-cos(msa*pi/180)))*BW; % maximum depth of material
profile
aor=input('Enter Angle of Repose (deg): ');
if aor<=19
    Cmz=1.5;
elseif aor<=25
    Cmz=2.1;
elseif aor<=29
    Cmz=2.5;
elseif aor<=34
    Cmz=3.3;
elseif aor<=39
    Cmz=4.2;
else Cmz=5.7;
end
end
total(i)=Th(i)+Tam(i)+Tss(i)+Tis(i)+Tiw(i)+Tbi(i)+Tim(i)+Ts(i)+Tbc(i
);
if i==1
```

```
for Tassume=takeup:takeup+total(i)
BS(i)=Si(i)*(Wb+Wm)*100/8/Tassume;
if belt==1
    Rmz=1/12*exp(4.181-1.572*(BW/Si(i)/12)^1.5-0.010827*BS(i)^0.5);
end
if belt==2
    Rmz=1/12*exp(4.966-4.071*(BW/Si(i)/12)^0.5-1.062*10^-2*BS(i)^-
1);
end
Wmz=1/12^4*dm^3*densm*Cmz*BW*(exp((Wb+Wm)*Si(i)/Tassume)-1)*exp(-
1/2*(Wb+Wm)*Si(i)/Tassume)*Rmz;  % Belt work required to cause
material movement from one idler to the next
Tmz(i)=Wmz*Lf(i)/Si(i)^2; % Tension loss from internal movements in
the bulk material
Tcheck(i)=takeup+Th(i)+Tam(i)+Tss(i)+Tis(i)+Tiw(i)+Tbi(i)+Tim(i)+Ts(
i)+Tbc(i)+Tmz(i); % Take up is added because it is first flight
Tact(i)=Tcheck(i)/2;
if Tact(i)<(Tassume+2)&&Tact(i)>(Tassume-2)
    break
end
end
TmzN(i)=4.44822*Tmz(i);
fprintf('Tension loss from internal movements in the bulk material =
%2.3f lbf = %2.3f N\n',Tmz(i),TmzN(i))
elseif i>1&&r==0&&Lf(i)~=0
for Tassume=CT(i-1):(CT(i-1)+total(i))
BS(i)=Si(i)*(Wb+Wm)*100/8/Tassume;
if belt==1
    Rmz=exp(4.181-1.572*(BW/Si(i)/12)^1.5-0.010827*BS(i)^0.5);
end
if belt==2
    Rmz=1/12*exp(4.966-4.071*(BW/Si(i)/12)^0.5-1.062*10^-2*BS(i)^-
1);
end
```

```
Wmz=1/12^5*dm^3*densm*Cmz*BW*(exp((Wb+Wm)*Si(i)/Tassume)-1)*exp(-
1/2*(Wb+Wm)*Si(i)/Tassume)*Rmz;  % Belt work required to cause
material movement from one idler to the next
Tmz(i)=Wmz*Lf(i)/Si(i)^2; % Tension loss from internal movements in
the bulk material
Tcheck(i)=CT(i-
1)+Th(i)+Tam(i)+Tss(i)+Tis(i)+Tiw(i)+Tbi(i)+Tim(i)+Ts(i)+Tbc(i)+Tmz(
i);
Tact(i)=Tcheck(i)/2;
if Tact(i)<(Tassume+2)&&Tact(i)>(Tassume-2)
    break
end
end
TmzN(i)=4.44822*Tmz(i);
fprintf('Tension loss from internal movements in the bulk material =
%2.3f lbf = %2.3f N\n',Tmz(i),TmzN(i))
else
    Tmz(i)=0;
end
Ttotal(i)=Th(i)+Tam(i)+Tss(i)+Tis(i)+Tiw(i)+Tbi(i)+Tim(i)+Ts(i)+Tbc(
i)+Tmz(i);  % Total tension increase in current flight
TtotalN(i)=4.44822*Ttotal(i);
if Lf(i)==0
    drive=drive+1;
f(i)=input('Enter Coefficient of friction between pulley surface and
belt surface: ');
aow(i)=input('Enter Angle of wrap on pulley(deg): ');
Ttotal(i)=(CT(i-1)/exp(f(i)*aow(i)*pi/180))-CT(i-1);
TtotalN(i)=4.44822*Ttotal(i);
end
fprintf('Total Tension increase in flight %1.0f is = %2.3f lbf =
%2.3f N\n',i,Ttotal(i),TtotalN(i))
CT(i)=takeup;   % preallocation
for j=1:i
 CT(i)=CT(i)+Ttotal(j);
```

```
end
CTN(i)=CT(i)*4.44822;
if HT(i)==1
    if i==1
        Thoravg=takeup+Ttotal(i)/2;
    else
    Thoravg=CT(i-1)+CT(i)/2;
    end
Fm=Thoravg*Si(i)/R;
Fmi=Li*Fm/BW;
Fmc=Lc*Fm/BW;
Fmo=Lo*Fm/BW;
Fmni=Fmi*sin((Ta+ea)*pi/180);
Fmnc=Fmc*sin(ea*pi/180);
Fmno=Fmo*sin((Ta-ea)*pi/180);
Fmpi=Fmi*cos((Ta+ea)*pi/180);
Fmpc=Fmc*cos(ea*pi/180);
Fmpo=Fmo*cos((Ta-ea)*pi/180);
Fmt=Fmpi+Fmpc+Fmpo;
FmtN=Fmt*4.44822;
fprintf('Total Destabilizing or Motivating Force during the
horizontal turn is = %2.3f lbf = %2.3f N\n',Fmt,FmtN)
Fsb=(Wb*Si(i)/BW)*((Li*sin((Ta+ea)*pi/180))+(Lc*sin(ea*pi/180))-
(Lo*sin((Ta-ea)*pi/180)));
FsbN=Fsb*4.44822;
fprintf('Stabilizing Force generated due to weight of the belt is =
%2.3f lbf = %2.3f N\n',Fsb,FsbN)
Aci=input('Enter Meterial cross section area above inside idler
(ft2): ');
Acc=input('Enter Meterial cross section area above center idler
(ft2): ');
Aco=input('Enter Meterial cross section area above outside idler
(ft2): ');
Fsm=(Si(i)*densm)*((Aci*sin((Ta+ea)*pi/180))+(Acc*sin(ea*pi/180))-
(Aco*sin((Ta-ea)*pi/180)));
```

```
FsmN=Fsm*4.44822;
fprintf('Stabilizing Force generated due to weight of the bulk
material is = %2.3f lbf = %2.3f N\n',Fsm,FsmN)
ui=input('Enter friction factor between belt and inside idler : ');
uc=input('Enter friction factor between belt and center idler : ');
uo=input('Enter friction factor between belt and outside idler : ');
Fsf=ui*(((Li*Wb/BW)+(Aci*Si(i)*densm))*cos((Ta+ea)*pi/180)+Fmni)+uc*
(((Lc*Wb/BW)+(Aci*Si(i)*densm))*cos(ea*pi/180)+Fmnc)+uo*(((Lo*Wb/BW)
+(Aco*Si(i)*densm))*cos((Ta-ea)*pi/180)+Fmno);
FsfN=Fsf*4.44822;
fprintf('Stabilizing Force generated due to friction between belt
and idlers is = %2.3f lbf = %2.3f N\n',Fsf,FsfN)
TFs=Fsb+Fsm+Fsf;
TFsN=TFs*4.44822;
fprintf('Total Stabilizing Force generated is = %2.3f lbf = %2.3f
N\n',TFs,TFsN)
end
if vcurv=='Y'||vcurv=='y'
    if vu==1
    Rcave1=1.11*Tver/Wb;
    if belt==1
        EdgeS=30;
    else EdgeS=75-1.5*Tver/BW;
    end
    Rcave2=BulkM*BW^2*plies*sin(Ta*pi/180)/(Tver-EdgeS*BW*54);
    Rcave3=BulkM*BW^2*plies*sin(Ta*pi/180)/((Trated-Tver)*54);
    if Rcave1>=Rcave2&&Rcave1>=Rcave3
        fprintf('Minimum Radius of the vertical curve is = %2.3f
ft\n',Rcave1)
    elseif Rcave2>=Rcave1&&Rcave2>=Rcave3
        fprintf('Minimum Radius of the vertical curve is = %2.3f
ft\n',Rcave2)
    else
        fprintf('Minimum Radius of the vertical curve is = %2.3f
ft\n',Rcave3)
```

```
    end
elseif vu==2
    Rvex1=BulkM*BW^2*plies*sin(Ta*pi/180)/((Trated-Tver)*54);
    Rvex2=BulkM*BW^2*plies*sin(Ta*pi/180)/(Tver-30*BW*54);
    if Rvex1>Rvex2
        fprintf('Minimum Radius of the vertical curve is = %2.3f
ft\n',Rvex1)
    else
        fprintf('Minimum Radius of the vertical curve is = %2.3f
ft\n',Rvex2)
    end
    end
end
if Lf(i)==0&&pulley==0
    pulley=1;
    ACT(i)=CT(i);
    ACT(i-1)=CT(i-1);
    beltt=input('Enter Belt Thickness (in): ');
    nd=input('Enter no. of Drive pulleys: ');
    sdd=input('Enter Shaft Diameter of Drive pulley (in): ');
    pdd=input('Enter Pulley Diameter of Drive pulley (in): ');
    pwd=input('Enter Weight of Drive pulley (lbf): ');
    nsb=input('Enter no. of Snub/Bend pulleys: ');
    sdsb=input('Enter Shaft Diameter of Snub/Bend pulleys (in): ');
    pdsb=input('Enter Pulley Diameter of Snub/Bend pulleys (in): ');
    pwsb=input('Enter Weight of Snub/Bend pulleys (lbf): ');
    for iteration=1:50
    Tavg=(ACT(i)+ACT(i-1))/2;
    if belt==1
        Twd=nd*9*BW*(0.8+0.01*Tavg/BW)*beltt/pdd;   %   Drive   pulley
wrap resistance
    else
        Twd=nd*12*BW*(1.142+0.01*Tavg/BW)*beltt/pdd;
    end
    Tbd=0.01*sdd*(Tavg+pwd)/pdd; % Drive Pulley bearing resistance
```

```
    Td=Tbd+Twd; % Total Drive pulley resistance
    if belt==1
        Twsb=nsb*9*BW*(0.8+0.01*Tavg/BW)*beltt/pdsb;    %   Snub/Bend
pulleys wrap Tension
    else
        Twsb=nsb*12*BW*(1.142+0.01*Tavg/BW)*beltt/pdsb;
    end
    Tbsb=0.01*sdsb*(Tavg+pwsb)/pdsb;  %  Snub/Bend  pulleys  bearing
Tension
    Tsb=Tbsb+Twsb; % Total Snub/Bend pulleys Tension
    Ttp=Td+Tsb; % Total Resistance due to pulleys
    ACT(i-1)=CT(i-1)+Ttp;
    ATtotal(i)=(CT(i-1)/exp(f(i)*aow(i)*pi/180))-CT(i-1);
    ACT(i)=ACT(i-1)+ATtotal(i);
    end
    CT(i-1)=ACT(i-1);
    CT(i)=ACT(i-1)+ATtotal(i);
    CTN(i-1)=4.44288*CT(i-1);
    CTN(i)=4.44822*CT(i);
    TtpN=4.44822*Ttp;
    fprintf('Total Resistance due to pulleys is = %2.3f lbf = %2.3f
N\n',Ttp,TtpN)
    fprintf('Actual Total Tension till the end of flight %1.0f is =
%2.3f lbf = %2.3f N\n',i-1,CT(i-1),CTN(i-1))
    fprintf('Actual Total Tension till the end of flight %1.0f is =
%2.3f lbf = %2.3f N\n',i,CT(i),CTN(i))
end
if Lf(i)~=0
fprintf('Total Tension till the end of flight %1.0f is = %2.3f lbf =
%2.3f N\n',i,CT(i),CTN(i))
end
if i==n
    maximum=1;
  for z=1:i-1
      if CT(z+1)>=CT(z)
```

```
        maximum=z+1;
      end
  end
  fprintf('Maximum Tension in the conveyor is = %2.3f lbf = %2.3f
N\n',CT(maximum),CTN(maximum))
end
end
```

3.3 Parameters Required

To calculate the belt tension using the computer program, following parameters are required:

1. Profile of the conveyor path
2. Weight of the belt per unit length
3. Percentage of loading of conveyor
4. Rated capacity of the conveyor
5. Angle of impact of material to the belt
6. Speed of belt
7. Material falling height at loading point
8. Length of Skirtboard
9. Skirtboard spacing
10. Idler Series
11. Number of idler roll per idler set
12. Idler roll diameter
13. Spacing of idler sets
14. Type of idler bearing used
15. Idler Material
16. Idler misalignment
17. Belt cover thickness
18. Troughing angle
19. Belt width
20. Type of belt used (Fabric or Steel)
21. Density and type of bulk material
22. Material surcharge angle
23. Ambient operating temperature
24. Angle of repose
25. Angle of wrap on pulley
26. Take up tension
27. No. of drive, snub, bend pulleys
28. Pulleys shaft diameter

29. Radius of horizontal curve
30. Angle of horizontal curve
31. Horizontal curve super elevation angle
32. No. of plies in the belt
33. Rated belt tension
34. Pulleys outer diameter
35. Friction factor between belt and inner, central and outer idler roll
36. Coefficient of friction between pulley and belt

3.4 Guidelines for using computer program

- The conveyor is broken into segments called flights to find tension at specific points. Horizontal and vertical curves are considered separate flights. Head pulley along with snub/bend pulleys is considered a separate flight. For pulleys length of the flight is entered zero in the program. Before running the program, number of flights is decided based on the profile of the conveyor.
- Program is valid for troughing angles 0°, 20°, 35° and 45°.
- Program only asks for most parameters in the first flight. For rest of the flights only flight specific parameters like length of the flight, length of skirtboard in the flight, no. of belt cleaners in the flight, idler spacing etc. need to be entered in the program.
- Twelve most commonly used bulk materials are added in the program to automatically enter friction factor between skirtboard and bulk material, if bulk material required is not one of the twelve that are provided in the program then friction factor between skirtboard and bulk material can directly be entered in the program.
- Resistance effect of take-up pulley is neglected.

- Sometimes in the program data is needed to be entered in the form of yes or no, the program displays the question along with (y/n), y should be pressed for yes and n should be pressed for no. Both small and capital letters are acceptable.
- Most of the input data needs to be entered in English units, but all output results are displayed in both English units and S.I. units.

3.5 Features of the computer program

- All the tables and data needed to calculate various components of the tension are incorporated within the program itself.
- The program calculates the weight of bulk material per unit length (which is needed for the calculation of almost all the tension contributing forces) based on rated capacity of conveyor, speed of the belt and percentage loading of the conveyor.
- To find tension increase due to material trampling losses average tension in the flight is needed. To find average tension in the flight and correct material trampling losses, first all the other tension contributors are found for the current flight then a loop is started where average tension is assumed which is equal to the tension at the start of the flight. Using this assumed average tension, material trampling losses are calculated and after having all the tension contributors for the current flight, average tension is calculated. If calculated average tension is within ±2 lbf of the assumed average tension, the loop is broken and the calculated material trampling losses are correct, if not then another iteration is run increasing the assumption of average tension by 1 and the whole procedure is repeated.

- For conveyors involving horizontal turn, the program calculates and displays the destabilizing or de-training force during the horizontal turn. Further, it calculates the stabilizing force generated by weight of the belt and material plus from friction between the belt and idler rolls. Further, for horizontal turn average tension in the flight is needed for the calculation of destabilizing force, which is found by dividing the increase in the tension in the horizontal turn by two and adding the tension at the start of the horizontal turn.
- For conveyors having vertical curves, the program calculates and displays the minimum radius of the curve needed to avoid undesirable scenarios like buckling in the belt and possible spillage of the bulk material.
- After calculating belt tension at various points, the program compares all of them and displays the maximum tension in the belt.

3.6 Recommendations

- If the calculated maximum tension is coming out to be more than the belt rated tension or there is a need of maximum tension reduction for reducing the size or rating of various components, intermediate drive can be used. Use of intermediate drive can reduce the maximum tension in the belt by up to 40% **(**Alspaugh, 2004).
- Keeping the belt sag in check is important for reliable and efficient operation of the conveyor. Generally the maximum allowed belt sag is up to 2%, if the belt sag is coming out to be more than the allowed limit then either the tension can be increased by increasing take-up tension or idler spacing can be reduced in the affected flight.

- For horizontal curve total stabilizing and total destabilizing forces are calculated. If destabilizing forces are more than stabilizing forces there could be material spillage and in the worst case scenario, belt could come off the idlers. To prevent this there are three solutions. First, tension during the horizontal curve could be reduced, either by reducing the take-up tension or by using intermediate drive before the curve. Second, idler spacing during the curve could be reduced. Third, radius of the curve could be increased.

Chapter 4: Validation of the Computer Program

Validation of the thesis work is necessary to check if the tension calculation procedure and its programming is accurate or not. For that purpose validation of the computer program is done by using the Industrial data for five belt conveyors of different lengths, provided by Development Consultants Pvt. Ltd. (DCPL). The conveyors whose data is provided are already successfully operational in a coal based power plant. The parameter used for the validation of the thesis work is maximum belt tension in the conveyor, which is responsible for rating or sizing of many conveyor components such as the belt, drive etc. Data for all five conveyors along with the calculated maximum tension by the consultancy is as follows:

4.1 Conveyor 1

Table 4.1: Conveyor 1 Industrial data

Sr. no.	Parameter	Value (S.I. units)	Value (English units)
1	Length along belt	409.528 m	1343.25 ft
2	Lift	48.975 m	160.638 ft
3	Angle of incline	6.82°	6.82°
4	Belt width	1.2 m	47.23 in
5	Skirt width	0.8 m	31.48 in
6	Capacity	800 tph	800 tph
7	Belt speed	2.5 m/s	492 fpm
8	Bulk Density	0.8 t/m^3	49.96 lbf/ft^3
9	Belt weight	182.47 N/m	12.49 lbf/ft
10	Carry idler spacing	1.2 m	3.936 ft
11	Drive pulley wrap angle	210°	210°

12	Coefficient friction (pulley/belt)	0.35	0.35
13	Belt thickness	0.020 m	0.787 in
14	No. of belt scrappers/cleaners	3	3
15	Length of skirtboard	32 m	104.96 ft
16	Idler diameter	0.152 m	6 in
17	No. of rolls per idler set	3	3
18	Troughing angle	35°	35°
19	Material surcharge angle	20°	20°
20	Angle of repose	33°	33°
21	Idler misalignment	0	0
22	No. of drive pulleys	1	1
23	Diameter of drive pulley	0.63 m	24.8 in
24	Shaft diameter of drive pulley	0.2 m	7.87 in
25	Weight of drive pulley	16000 N	3596.94 N
26	No. of snub/bend pulleys	3	3
27	Diameter of snub/bend pulley	0.4 m	15.75 in
28	Shaft dia. of snub/bend pulley	0.125 m	4.92 in
29	Weight of snub/bend pulley	8000 N	1798.47 N
30	Take-up tension	35859 N	8076 lbf
31	Tight side tension (calculated)	107702 N	24125 lbf
32	Slack side tension (calculated)	33950.5 N	7604.9 lbf
33	Type of belt used	Fabric belt	
34	Type of idler bearing used	Ball type bearing	
35	Bulk material	Coal (Bituminous)	

4.2 Conveyor 2

Table 4.2: Conveyor 2 Industrial data

Sr. no.	Parameter	Value (S.I. units)	Value (English units)
1	Length along belt	52.46 m	172.12 ft
2	Lift	7.4 m	24.27 ft
3	Angle of incline	8.03°	8.03°
4	Belt width	1.2 m	47.23 in
5	Skirt width	0.8 m	31.48 in
6	Capacity	800 tph	800 tph
7	Belt speed	2.5 m/s	492 fpm
8	Bulk Density	0.8 t/m^3	49.96 lbf/ft^3
9	Belt weight	164.8 N/m	11.25 lbf/ft
10	Carry idler spacing	1.2 m	3.936 ft
11	Drive pulley wrap angle	210°	210°
12	Coefficient friction (pulley/belt)	0.35	0.35
13	Belt thickness	0.020 m	0.787 in
14	No. of belt scrappers/cleaners	3	3
15	Length of skirtboard	16 m	52.49 ft
16	Idler diameter	0.152 m	6 in
17	No. of rolls per idler set	3	3
18	Troughing angle	35°	35°

19	Material surcharge angle	20°	20°
20	Angle of repose	33°	33°
21	Idler misalignment	0	0
22	No. of drive pulleys	1	1
23	Diameter of drive pulley	0.5 m	19.68 in
24	Shaft diameter of drive pulley	0.125 m	4.92 in
25	Weight of drive pulley	10000 N	2240 lbf
26	No. of snub/bend pulleys	3	3
27	Diameter of snub/bend pulley	0.315 m	12.40
28	Shaft dia. of snub/bend pulley	0.1 m	3.93 in
29	Weight of snub/bend pulley	7000 N	1568 lbf
30	Take-up tension	10852 N	2430.8 lbf
31	Tight side tension (calculated)	23846 N	5341.5 lbf
32	Slack side tension (calculated)	8710 N	1951 lbf
33	Type of belt used	Fabric belt	
34	Type of idler bearing used	Ball type bearing	
35	Bulk material	Coal (Bituminous)	

4.3 Conveyor 3

Table 4.3: Conveyor 3 Industrial data

Sr. no.	Parameter	Value (S.I. units)	Value (English units)
1	Length along belt	54.52 m	178.87 ft
2	Lift	6.3 m	20.66 ft

3	Angle of incline	6.59°	6.59°
4	Belt width	1.2 m	47.23 in
5	Skirt width	0.8 m	31.48 in
6	Capacity	800 tph	800 tph
7	Belt speed	2.5 m/s	492 fpm
8	Bulk Density	0.8 t/m^3	49.96 lbf/ft^3
9	Belt weight	164.8 N/m	11.25 lbf/ft
10	Carry idler spacing	1.2 m	3.936 ft
11	Drive pulley wrap angle	210°	210°
12	Coefficient friction (pulley/belt)	0.35	0.35
13	Belt thickness	0.020 m	0.787 in
14	No. of belt scrappers/cleaners	3	3
15	Length of skirtboard	12 m	39.37 ft
16	Idler diameter	0.152 m	6 in
17	No. of rolls per idler set	3	3
18	Troughing angle	35°	35°
19	Material surcharge angle	20°	20°
20	Angle of repose	33°	33°
21	Idler misalignment	0	0
22	No. of drive pulleys	1	1
23	Diameter of drive pulley	0.5 m	19.68 in
24	Shaft diameter of drive pulley	0.125 m	4.92 in
25	Weight of drive pulley	10000 N	2240 lbf

26	No. of snub/bend pulleys	3	3
27	Diameter of snub/bend pulley	0.315 m	12.40
28	Shaft dia. of snub/bend pulley	0.1 m	3.93 in
29	Weight of snub/bend pulley	7000 N	1568 lbf
30	Take-up tension	9925 N	2223.2 lbf
31	Tight side tension (calculated)	21831 N	4890.14 lbf
32	Slack side tension (calculated)	7974 N	1786.17 lbf
33	Type of belt used	Fabric belt	
34	Type of idler bearing used	Ball type bearing	
35	Bulk material	Coal (Bituminous)	

4.4 Conveyor 4

Table 4.4: Conveyor 4 Industrial data

Sr. no.	Parameter	Value (S.I. units)	Value (English units)
1	Length along belt	286.434 m	939.74 ft
2	Lift	51.3 m	168.26 ft
3	Angle of incline	10.15°	10.15°
4	Belt width	1.2 m	47.23 in
5	Skirt width	0.8 m	31.48 in
6	Capacity	800 tph	800 tph
7	Belt speed	2.5 m/s	492 fpm
8	Bulk Density	0.8 t/m^3	49.96 lbf/ft^3
9	Belt weight	182.47 N/m	12.49 lbf/ft

10	Carry idler spacing	1.2 m	3.936 ft
11	Drive pulley wrap angle	210°	210°
12	Coefficient friction (pulley/belt)	0.35	0.35
13	Belt thickness	0.020 m	0.787 in
14	No. of belt scrappers/cleaners	3	3
15	Length of skirtboard	8 m	26.24 ft
16	Idler diameter	0.152 m	6 in
17	No. of rolls per idler set	3	3
18	Troughing angle	35°	35°
19	Material surcharge angle	20°	20°
20	Angle of repose	33°	33°
21	Idler misalignment	0	0
22	No. of drive pulleys	1	1
23	Diameter of drive pulley	0.63 m	24.8 in
24	Shaft diameter of drive pulley	0.2 m	7.87 in
25	Weight of drive pulley	16000 N	3596.94 N
26	No. of snub/bend pulleys	3	3
27	Diameter of snub/bend pulley	0.4 m	15.75 in
28	Shaft dia. of snub/bend pulley	0.125 m	4.92 in
29	Weight of snub/bend pulley	8000 N	1798.47 N
30	Take-up tension	29836 N	8076 lbf
31	Tight side tension (calculated)	99051 N	24125 lbf
32	Slack side tension (calculated)	31224 N	7604.9 lbf

33	Type of belt used	Fabric belt
34	Type of idler bearing used	Ball type bearing
35	Bulk material	Coal (Bituminous)

4.5 Conveyor 5

Table 4.5: Conveyor 5 Industrial data

Sr. no.	Parameter	Value (S.I. units)	Value (English units)
1	Length along belt	217 m	711.94 ft
2	Lift	13.8 m	45.26 ft
3	Angle of incline	3.64°	3.64°
4	Belt width	1.2 m	47.23 in
5	Skirt width	0.8 m	31.48 in
6	Capacity	800 tph	800 tph
7	Belt speed	2.5 m/s	492 fpm
8	Bulk Density	0.8 t/m^3	49.96 lbf/ft^3
9	Belt weight	168.73 N/m	11.52 lbf/ft
10	Carry idler spacing	1.2 m	3.936 ft
11	Drive pulley wrap angle	210°	210°
12	Coefficient friction (pulley/belt)	0.35	0.35
13	Belt thickness	0.020 m	0.787 in
14	No. of belt scrappers/cleaners	3	3
15	Length of skirtboard	8 m	26.24 ft

16	Idler diameter	0.152 m	6 in
17	No. of rolls per idler set	3	3
18	Troughing angle	35°	35°
19	Material surcharge angle	20°	20°
20	Angle of repose	33°	33°
21	Idler misalignment	0	0
22	No. of drive pulleys	1	1
23	Diameter of drive pulley	0.63 m	24.8 in
24	Shaft diameter of drive pulley	0.16 m	6.26 in
25	Weight of drive pulley	16000 N	3596.94 N
26	No. of snub/bend pulleys	3	3
27	Diameter of snub/bend pulley	0.4 m	15.75 in
28	Shaft dia. of snub/bend pulley	0.125 m	4.92 in
29	Weight of snub/bend pulley	8000 N	1798.47 N
30	Take-up tension	19762 N	4426.2 lbf
31	Tight side tension (calculated)	46973 N	10521.9 lbf
32	Slack side tension (calculated)	17157 N	3843.17 lbf
33	Type of belt used	Fabric belt	
34	Type of idler bearing used	Ball type bearing	
35	Bulk material	Coal (Bituminous)	

4.6 Assumptions made during validation

- Loading of conveyor = 100%
- Angle of impact of material to the belt relative to the belt direction = 0°
- Ambient operating temperature = 25°C
- CEMA Idler Series = D
- Belt bottom cover thickness = 3mm

All the assumptions made are standard belt conveyor data.

4.7 Validation

As said earlier, parameter chosen for validation of the thesis work is maximum tension in the conveyor. Conveyor 1 is 1343 ft long, so is has been split into 3 flights. Flight 1 is 700 ft long, flight 2 is 643 ft long and flight 3 is head pulley and corresponding bend pulleys. Flight 1 and 2 are taken separately to check that the program gives the tension at the end of flight 1 as well. Using assumptions and parameters of conveyors provided in the table 4.1, following is the execution of the computer program:

```
Enter no. of flights: 3

Enter the Take-up Tension (lbf): 8076

Details of flight 1:

Is there Horizontal turn in the flight? (y/n): n

Enter Length of the flight (ft): 700

Enter Rated capacity of the conveyor (tph): 800

Enter Belt speed (fpm): 492

Enter Weight of the belt (lbf/ft): 12.498
```

Enter loading of conveyor (%): 100

Enter Length of the conveyor having lift or drop in the current flight (ft): 700

Enter Angle of incline(+) or decline(-) for the same length (deg): 6.82

Change in belt tension due to lift or drop = 6048.045 lbf = 26903.034 N

Enter Angle of impact of material to the belt relative to the belt direction (deg): 0

Tension added at loading point to continously accelerate material to belt speed = 124.760 lbf = 554.960 N

Is there vertical curve in the flight? (y/n): n

Enter Length of Skirtboard(ft): 104.96

Tension increase due to Skirtboard Seal Friction = 629.760 lbf = 2801.311 N

Enter Idler roll diameter (in): 6

Enter no. of rolls per idler set: 3

Enter Ambient operating temperature(°C): 25

Enter Idler Series (B,C,D,E): D

Enter Spacing of idler sets for current flight (ft): 3.936

Tension change due to Idler Seal Drag = 581.475 lbf = 2586.527 N

Type of idler bearing used:

Press 1 for Taper Roller

Press 2 for Deep Groove Ball: 2

Tension change due to Idler Bearing Losses = 31.185 lbf = 138.718 N

Enter Belt bottom cover thickness(in): .118

Enter Troughing Angle(0,20,35,45 deg): 35

Enter Belt Width(in): 47.232

Type of belt:

Press 1 for Fabric Belt

Press 2 for Steel Cable Belt: 1

Tension increase due to rubber indentation losses = 180.208 lbf = 801.606 N

Is there idler misalignment (y/n): n

Enter Density of the bulk material (lbf/ft3): 49.96

Enter Material surcharge angle (deg): 20

Material to be conveyed:

Press 1 for Alumina(pulverized & dry)

Press 2 for Bauxite

Press 3 for Cement(portland)

Press 4 for Cement clinker

Press 5 for Coal Ash

Press 6 for Coal(anthracite)

Press 7 for Coal(Bituminous)

Press 8 for Wheat

Press 9 for Iron ore

Press 10 for Limestone

Press 11 for Phosphate

Press 12 for Sugar

Press 13 to directly enter value of friction factor between skirtboard and bulk material : 7

Enter Skirtboard spacing (in): 31.48

Tension change due to material sliding on skirtboard = 16.674 lbf = 74.171 N

Enter No. of belt cleaners in the current flight: 0

Enter No. of belt discharge plows in the current flight: 0

Enter Angle of Repose (deg): 33

Tension loss from internal movements in the bulk material = 142.508 lbf = 633.908 N

Total Tension increase in flight 1 is = 7754.615 lbf = 34494.235 N

Total Tension till the end of flight 1 is = 15830.615 lbf = 70418.060 N

Details of flight 2:

Is there Horizontal turn in the flight? (y/n): n

Enter Length of the flight (ft): 643.2

Enter Length of the conveyor having lift or drop in the current flight (ft): 643.25

Enter Angle of incline(+) or decline(-) for the same length (deg): 6.82

Change in belt tension due to lift or drop = 5557.721 lbf = 24721.967 N

Is there vertical curve in the flight? (y/n): n

Enter Length of Skirtboard(ft): 0

Enter Spacing of idler sets for current flight (ft): 3.936

Tension change due to Idler Seal Drag = 534.292 lbf = 2376.648 N

Tension change due to Idler Bearing Losses = 28.655 lbf = 127.462 N

Tension increase due to rubber indentation losses = 165.586 lbf = 736.562 N

Enter No. of belt cleaners in the current flight: 3

Tension added due to belt cleaners = 708.480 lbf = 3151.475 N

Enter No. of belt discharge plows in the current flight: 0

Tension loss from internal movements in the bulk material = 90.081 lbf = 400.699 N

Total Tension increase in flight 2 is = 7084.814 lbf = 31514.812 N

Total Tension till the end of flight 2 is = 22915.429 lbf = 101932.872 N

Details of flight 3:

Is there Horizontal turn in the flight? (y/n): n

Enter Length of the flight (ft): 0

```
Enter Coefficient of friction between pulley surface and belt surface: .35
Enter Angle of wrap on pulley(deg): 210
Total Tension increase in flight 3 is = -16562.008 lbf = -73671.456 N
Enter Belt Thickness (in): .787
Enter no. of Drive pulleys: 1
Enter Shaft Diameter of Drive pulley (in): 7.87
Enter Pulley Diameter of Drive pulley (in): 24.8
Enter Weight of Drive pulley (lbf): 3596.94
Enter no. of Snub/Bend pulleys: 3
Enter Shaft Diameter of Snub/Bend pulleys (in): 4.92
Enter Pulley Diameter of Snub/Bend pulleys (in): 15.75
Enter Weight of Snub/Bend pulleys (lbf): 1798.47
Total Resistance due to pulleys is = 419.698 lbf = 1866.910 N
Actual Total Tension till the end of flight 2 is = 23335.128 lbf = 103675.172 N
Actual Total Tension till the end of flight 3 is = 6773.120 lbf = 30128.326 N
Maximum Tension in the conveyor is = 23335.128 lbf = 103675.172 N
```

See Appendix for rest of the conveyors.

As seen from the execution of the program, program calculated and displayed the tension at the end of flight 1 as 70418 N, so the program is capable of finding the tension at specific location in the conveyor profile. Further, program calculated the maximum tension in the conveyor as 103675.17 N, while the calculated maximum tension in the data provided by DCPL in table 4.1 is 107702 N. The percentage error (E) in calculation of maximum tension by the program can be found out by following relation:

$$E = \left|\frac{T_{ind} - T_{cp}}{T_{ind}}\right| \times 100$$

Where: T_{ind} (N) is the maximum tension in the conveyor as per industrial data

T_{cp} (N) is the calculated maximum tension in the conveyor using computer program

Therefore for conveyor 1 the percentage error is:

$$E = \left|\frac{107702 - 103675.17}{107702}\right| \times 100 \quad = 3.73\%$$

Similarly percentage error is calculated for other four conveyors and has been shown in the following table:

Table 4.6: Comparison of Industrial data and output of the code for maximum tension

Conveyor No.	**Length of conveyor**	$\mathbf{T_{ind}}$	$\mathbf{T_{cp}}$	**Error (E)**
1	409.528 m	107702 N	103675.17 N	3.73 %
2	52.46 m	23846 N	26140 N	9.62 %
3	54.52 m	21831 N	23830.39 N	9.15 %
4	286.434 m	99051 N	96022.65 N	3.05 %
5	217 m	46973 N	44975.93 N	4.25%

It can be seen from the above table that maximum error is in conveyor 2 which is 9.62%. Since the maximum error among all five conveyors is within ±10%, it suggests that the tension calculation procedure and its programming is within acceptable limits of accuracy.

Error in conveyor 1, 4 and 5 are 3.73%, 3.05% and 4.25% respectively, which are quite less. Since these conveyors are relatively longer conveyors, it suggests that the method is more accurate for the longer belt conveyors. Error in conveyors 2 and 3, which are shorter conveyors, is due to two reasons. First, the industrial method does not take weight of the belt into account while calculating tension increase due to lift of the material, but method presented in the thesis does take weight of the belt into account. So, the increase in tension due to lift of the material is more in case of the method presented in the thesis. This difference is enlarged in case of shorter conveyor because tension increase due to lift of material is much more dominant in case of shorter conveyors. Second, method presented in thesis is more concerned towards longer conveyors, so tension loss factors, depending on the length of the conveyor like rubber indentation losses, idler seal friction, material trampling loss, idler load friction etc. are accurately defined but factors independent of the length of the conveyors like material acceleration loss, tension loss due to belt cleaners etc. could be a little overdesigned. So, error in shorter conveyors is more where conveyor length independent factors are more dominant.

Chapter 5: Conclusion and Future Scope of Work

5.1 Conclusion

A new computer program has been generated which can find belt tension at different points on a belt conveyor for different loading conditions during steady state operation. The program can be used for belt conveyor of any length, including horizontal and vertical curves thus it makes the program most suitable for long distance belt conveyor. The program is tested with five industrial conveyors of different lengths and maximum belt tension calculated by the program is matched with maximum tension as provided in industrial data and error is calculated. For shorter conveyors error was relatively large, 9.62% and 9.15% while for relatively longer conveyors error was much smaller, 3.73%, 3.05% and 4.25% which suggests that the method and its programming is more accurate for longer belt conveyors.

5.2 Future Scope of Work

In static analysis the belt is assumed to be a rigid body, which is acceptable for the steady state operation of the conveyor, but during transient events like starting of the belt, the torque transmitted to the belt through the drive pulley creates a stress wave which gradually starts the moving of the belt as the wave propagates along the belt, specially is case of long conveyors. There is no model in open literature that accurately takes elasticity of belt into account during starting and stopping. So, further studies are required to make an accurate mathematical model which accurately takes elasticity of belt into account during starting and stopping.

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Appendix

Following are the executions of the computer code for Industrial data of conveyor 2 to conveyor 5

Conveyor 2

```
Enter no. of flights: 2
Enter the Take-up Tension (lbf): 2430.8
Details of flight 1:
Is there Horizontal turn in the flight? (y/n): n
Enter Length of the flight (ft): 172.12
Enter Rated capacity of the conveyor (tph): 800
Enter Belt speed (fpm): 492
Enter Weight of the belt (lbf/ft): 11.25
Enter loading of conveyor (%): 100
Enter Length of the conveyor having lift or drop in the
current flight (ft): 172.12
Enter Angle of incline(+) or decline(-) for the same length
(deg): 8.03
Change in belt tension due to lift or drop = 1723.892 lbf =
7668.253 N
Enter Angle of impact of material to the belt relative to the
belt direction (deg): 0
```

Tension added at loading point to continously accelerate material to belt speed = 124.760 lbf = 554.960 N

Is there vertical curve in the flight? (y/n): n

Enter Length of Skirtboard(ft): 52.49

Tension increase due to Skirtboard Seal Friction = 314.940 lbf = 1400.922 N

Enter Idler roll diameter (in): 6

Enter no. of rolls per idler set: 3

Enter Ambient operating temperature(°C): 25

Enter Idler Series (B,C,D,E): D

Enter Spacing of idler sets for current flight (ft): 3.936

Tension change due to Idler Seal Drag = 142.976 lbf = 635.990 N

Type of idler bearing used:

Press 1 for Taper Roller

Press 2 for Deep Groove Ball: 2

Tension change due to Idler Bearing Losses = 7.535 lbf = 33.519 N

Enter Belt bottom cover thickness(in): 0.118

Enter Troughing Angle(0,20,35,45 deg): 35

Enter Belt Width(in): 47.232

Type of belt:

Press 1 for Fabric Belt

Press 2 for Steel Cable Belt: 1

Tension increase due to rubber indentation losses = 43.293 lbf = 192.577 N

Is there idler misalignment (y/n): n

Enter Density of the bulk material (lbf/ft3): 49.96

Enter Material surcharge angle (deg): 20

Material to be conveyed:

Press 1 for Alumina(pulverized & dry)

Press 2 for Bauxite

Press 3 for Cement(portland)

Press 4 for Cement clinker

Press 5 for Coal Ash

Press 6 for Coal(anthracite)

Press 7 for Coal(Bituminous)

Press 8 for Wheat

Press 9 for Iron ore

Press 10 for Limestone

Press 11 for Phosphate

Press 12 for Sugar

Press 13 to directly enter value of friction factor between skirtboard and bulk material : 7

Enter Skirtboard spacing (in): 31.48

Tension change due to material sliding on skirtboard = 8.339 lbf = 37.093 N

Enter No. of belt cleaners in the current flight: 3

Tension added due to belt cleaners = 708.480 lbf = 3151.475 N

Enter No. of belt discharge plows in the current flight: 0

Enter Angle of Repose (deg): 33

Tension loss from internal movements in the bulk material = 188.610 lbf = 838.981 N

Total Tension increase in flight 1 is = 3262.826 lbf = 14513.769 N

Total Tension till the end of flight 1 is = 5693.626 lbf = 25326.503 N

Details of flight 2:

Is there Horizontal turn in the flight? (y/n): n

Enter Length of the flight (ft): 0

Enter Coefficient of friction between pulley surface and belt surface: .35

Enter Angle of wrap on pulley(deg): 210

Total Tension increase in flight 2 is = -4115.039 lbf = -18304.599 N

Enter Belt Thickness (in): .787

Enter no. of Drive pulleys: 1

Enter Shaft Diameter of Drive pulley (in): 4.92

Enter Pulley Diameter of Drive pulley (in): 19.68

Enter Weight of Drive pulley (lbf): 2240

Enter no. of Snub/Bend pulleys: 3

Enter Shaft Diameter of Snub/Bend pulleys (in): 3.93

Enter Pulley Diameter of Snub/Bend pulleys (in): 12.40

Enter Weight of Snub/Bend pulleys (lbf): 1568

Total Resistance due to pulleys is = 189.945 lbf = 844.918 N

Actual Total Tension till the end of flight 1 is = 5883.572 lbf = 26140.003 N

Actual Total Tension till the end of flight 2 is = 1768.533 lbf = 7866.822 N

Maximum Tension in the conveyor is = 5883.572 lbf = 26140.003 N

Conveyor 3

Enter no. of flights: 2

Enter the Take-up Tension (lbf): 2223.2

Details of flight 1:

Is there Horizontal turn in the flight? (y/n): n

Enter Length of the flight (ft): 178.87

Enter Rated capacity of the conveyor (tph): 800

Enter Belt speed (fpm): 492

Enter Weight of the belt (lbf/ft): 11.25

Enter loading of conveyor (%): 100

Enter Length of the conveyor having lift or drop in the current flight (ft): 178.87

Enter Angle of incline(+) or decline(-) for the same length (deg): 6.59

Change in belt tension due to lift or drop = 1467.069 lbf = 6525.847 N

Enter Angle of impact of material to the belt relative to the belt direction (deg): 0

Tension added at loading point to continously accelerate material to belt speed = 124.760 lbf = 554.960 N

Is there vertical curve in the flight? (y/n): n

Enter Length of Skirtboard(ft): 39.37

Tension increase due to Skirtboard Seal Friction = 236.220 lbf = 1050.759 N

Enter Idler roll diameter (in): 6

Enter no. of rolls per idler set: 3

Enter Ambient operating temperature(°C): 25

Enter Idler Series (B,C,D,E): D

Enter Spacing of idler sets for current flight (ft): 3.936

Tension change due to Idler Seal Drag = 148.583 lbf = 660.931 N

Type of idler bearing used:

Press 1 for Taper Roller

Press 2 for Deep Groove Ball: 2

Tension change due to Idler Bearing Losses = 7.831 lbf = 34.834 N

Enter Belt bottom cover thickness(in): .118

Enter Troughing Angle(0,20,35,45 deg): 35

Enter Belt Width(in): 47.232

Type of belt:

Press 1 for Fabric Belt

Press 2 for Steel Cable Belt: 1

Tension increase due to rubber indentation losses = 44.991 lbf = 200.129 N

Is there idler misalignment (y/n): n

Enter Density of the bulk material (lbf/ft3): 49.96

Enter Material surcharge angle (deg): 20

Material to be conveyed:

Press 1 for Alumina(pulverized & dry)

Press 2 for Bauxite

Press 3 for Cement(portland)

Press 4 for Cement clinker

Press 5 for Coal Ash

Press 6 for Coal(anthracite)

Press 7 for Coal(Bituminous)

Press 8 for Wheat

Press 9 for Iron ore

Press 10 for Limestone

Press 11 for Phosphate

Press 12 for Sugar

Press 13 to directly enter value of friction factor between skirtboard and bulk material : 7

Enter Skirtboard spacing (in): 31.48

Tension change due to material sliding on skirtboard = 6.254 lbf = 27.821 N

Enter No. of belt cleaners in the current flight: 3

Tension added due to belt cleaners = 708.480 lbf = 3151.475 N

Enter No. of belt discharge plows in the current flight: 0

Enter Angle of Repose (deg): 33

Tension loss from internal movements in the bulk material = 215.243 lbf = 957.446 N

Total Tension increase in flight 1 is = 2959.431 lbf = 13164.202 N

Total Tension till the end of flight 1 is = 5182.631 lbf = 23053.485 N

Details of flight 2:

Is there Horizontal turn in the flight? (y/n): n

Enter Length of the flight (ft): 0

Enter Coefficient of friction between pulley surface and belt surface: 0.35

Enter Angle of wrap on pulley(deg): 210

Total Tension increase in flight 2 is = -3745.720 lbf = -16661.787 N

Enter Belt Thickness (in): .787

Enter no. of Drive pulleys: 1

Enter Shaft Diameter of Drive pulley (in): 4.92

Enter Pulley Diameter of Drive pulley (in): 19.68

Enter Weight of Drive pulley (lbf): 2240

Enter no. of Snub/Bend pulleys: 3

Enter Shaft Diameter of Snub/Bend pulleys (in): 3.93

Enter Pulley Diameter of Snub/Bend pulleys (in): 12.40

Enter Weight of Snub/Bend pulleys (lbf): 1568

Total Resistance due to pulleys is = 181.095 lbf = 805.549 N

Actual Total Tension till the end of flight 1 is = 5363.726 lbf = 23830.392 N

Actual Total Tension till the end of flight 2 is = 1618.006 lbf = 7197.247 N

Maximum Tension in the conveyor is = 5363.726 lbf = 23830.392 N

Conveyor 4

Enter no. of flights: 3

Enter the Take-up Tension (lbf): 6719.8

Details of flight 1:

Is there Horizontal turn in the flight? (y/n): n

Enter Length of the flight (ft): 500

Enter Rated capacity of the conveyor (tph): 800

Enter Belt speed (fpm): 492

Enter Weight of the belt (lbf/ft): 12.49

Enter loading of conveyor (%): 100

Enter Length of the conveyor having lift or drop in the current flight (ft): 500

Enter Angle of incline(+) or decline(-) for the same length (deg): 10.15

Change in belt tension due to lift or drop = 6466.053 lbf = 28762.427 N

Enter Angle of impact of material to the belt relative to the belt direction (deg): 0

Tension added at loading point to continously accelerate material to belt speed = 124.760 lbf = 554.960 N

Is there vertical curve in the flight? (y/n): n

Enter Length of Skirtboard(ft): 26.24

Tension increase due to Skirtboard Seal Friction = 157.440 lbf = 700.328 N

Enter Idler roll diameter (in): 6

Enter no. of rolls per idler set: 3

Enter Ambient operating temperature(°C): 25

Enter Idler Series (B,C,D,E): D

Enter Spacing of idler sets for current flight (ft): 3.936

Tension change due to Idler Seal Drag = 415.339 lbf = 1847.519 N

Type of idler bearing used:

Press 1 for Taper Roller

Press 2 for Deep Groove Ball: 2

Tension change due to Idler Bearing Losses = 22.273 lbf = 99.073 N

Enter Belt bottom cover thickness(in): .118

Enter Troughing Angle(0,20,35,45 deg): 35

Enter Belt Width(in): 47.232

Type of belt:

Press 1 for Fabric Belt

Press 2 for Steel Cable Belt: 1

Tension increase due to rubber indentation losses = 128.701 lbf = 572.491 N

Is there idler misalignment (y/n): n

Enter Density of the bulk material (lbf/ft3): 49.96

Enter Material surcharge angle (deg): 20

Material to be conveyed:

Press 1 for Alumina(pulverized & dry)

Press 2 for Bauxite

Press 3 for Cement(portland)

Press 4 for Cement clinker

Press 5 for Coal Ash

Press 6 for Coal(anthracite)

Press 7 for Coal(Bituminous)

Press 8 for Wheat

Press 9 for Iron ore

Press 10 for Limestone

Press 11 for Phosphate

Press 12 for Sugar

Press 13 to directly enter value of friction factor between skirtboard and bulk material : 7

Enter Skirtboard spacing (in): 31.48

Tension change due to material sliding on skirtboard = 4.169 lbf = 18.543 N

Enter No. of belt cleaners in the current flight: 1

Tension added due to belt cleaners = 236.160 lbf = 1050.492 N

Enter No. of belt discharge plows in the current flight: 0

Enter Angle of Repose (deg): 33

Tension loss from internal movements in the bulk material = 219.843 lbf = 977.909 N

Total Tension increase in flight 1 is = 7774.737 lbf = 34583.741 N

Total Tension till the end of flight 1 is = 14494.537 lbf = 64474.890 N

Details of flight 2:

Is there Horizontal turn in the flight? (y/n): n

Enter Length of the flight (ft): 439.74

Enter Length of the conveyor having lift or drop in the current flight (ft): 439.74

Enter Angle of incline(+) or decline(-) for the same length (deg): 10.15

Change in belt tension due to lift or drop = 5686.765 lbf = 25295.980 N

Is there vertical curve in the flight? (y/n): n

Enter Length of Skirtboard(ft): 0

Enter Spacing of idler sets for current flight (ft): 3.936

Tension change due to Idler Seal Drag = 365.282 lbf = 1624.856 N

Tension change due to Idler Bearing Losses = 19.588 lbf = 87.133 N

Tension increase due to rubber indentation losses = 113.190 lbf = 503.495 N

Enter No. of belt cleaners in the current flight: 2

Tension added due to belt cleaners = 472.320 lbf = 2100.983 N

Enter No. of belt discharge plows in the current flight: 0

Tension loss from internal movements in the bulk material = 66.439 lbf = 295.536 N

Total Tension increase in flight 2 is = 6723.584 lbf = 29907.982 N

Total Tension till the end of flight 2 is = 21218.121 lbf = 94382.872 N

Details of flight 3:

Is there Horizontal turn in the flight? (y/n): n

Enter Length of the flight (ft): 0

Enter Coefficient of friction between pulley surface and belt surface: .35

Enter Angle of wrap on pulley(deg): 210

Total Tension increase in flight 3 is = -15335.288 lbf = -68214.733 N

Enter Belt Thickness (in): .787

Enter no. of Drive pulleys: 1

Enter Shaft Diameter of Drive pulley (in): 7.87

Enter Pulley Diameter of Drive pulley (in): 24.8

Enter Weight of Drive pulley (lbf): 3596.94

Enter no. of Snub/Bend pulleys: 3

Enter Shaft Diameter of Snub/Bend pulleys (in): 4.92

Enter Pulley Diameter of Snub/Bend pulleys (in): 15.75

Enter Weight of Snub/Bend pulleys (lbf): 1798.47

Total Resistance due to pulleys is = 394.584 lbf = 1755.196 N

Actual Total Tension till the end of flight 2 is = 21612.705 lbf = 96022.656 N

Actual Total Tension till the end of flight 3 is = 6277.418 lbf = 27923.335 N

Maximum Tension in the conveyor is = 21612.705 lbf = 96022.656 N

Conveyor 5

Enter no. of flights: 2

Enter the Take-up Tension (lbf): 4426.2

Details of flight 1:

Is there Horizontal turn in the flight? (y/n): n

Enter Length of the flight (ft): 711.94

Enter Rated capacity of the conveyor (tph): 800

Enter Belt speed (fpm): 492

Enter Weight of the belt (lbf/ft): 11.52

Enter loading of conveyor (%): 100

Enter Length of the conveyor having lift or drop in the current flight (ft): 711.94

Enter Angle of incline(+) or decline(-) for the same length (deg): 3.64

Change in belt tension due to lift or drop = 3227.638 lbf = 14357.244 N

Enter Angle of impact of material to the belt relative to the belt direction (deg): 0

Tension added at loading point to continously accelerate material to belt speed = 124.760 lbf = 554.960 N

Is there vertical curve in the flight? (y/n): n

Enter Length of Skirtboard(ft): 26.24

Tension increase due to Skirtboard Seal Friction = 157.440 lbf = 700.328 N

Enter Idler roll diameter (in): 6

Enter no. of rolls per idler set: 3

Enter Ambient operating temperature(°C): 25

Enter Idler Series (B,C,D,E): D

Enter Spacing of idler sets for current flight (ft): 3.936

Tension change due to Idler Seal Drag = 591.393 lbf = 2630.645 N

Type of idler bearing used:

Press 1 for Taper Roller

Press 2 for Deep Groove Ball: 2

Tension change due to Idler Bearing Losses = 31.288 lbf = 139.174 N

Enter Belt bottom cover thickness(in): .118

Enter Troughing Angle(0,20,35,45 deg): 35

Enter Belt Width(in): 47.232

Type of belt:

Press 1 for Fabric Belt

Press 2 for Steel Cable Belt: 1

Tension increase due to rubber indentation losses = 179.981 lbf = 800.597 N

Is there idler misalignment (y/n): n

Enter Density of the bulk material (lbf/ft3): 49.96

Enter Material surcharge angle (deg): 20

Material to be conveyed:

Press 1 for Alumina(pulverized & dry)

Press 2 for Bauxite

Press 3 for Cement(portland)

Press 4 for Cement clinker

Press 5 for Coal Ash

Press 6 for Coal(anthracite)

Press 7 for Coal(Bituminous)

Press 8 for Wheat

Press 9 for Iron ore

Press 10 for Limestone

Press 11 for Phosphate

Press 12 for Sugar

Press 13 to directly enter value of friction factor between skirtboard and bulk material : 7

Enter Skirtboard spacing (in): 31.48

Tension change due to material sliding on skirtboard = 4.169 lbf = 18.543 N

Enter No. of belt cleaners in the current flight: 3

Tension added due to belt cleaners = 708.480 lbf = 3151.475 N

Enter No. of belt discharge plows in the current flight: 0

Enter Angle of Repose (deg): 33

Tension loss from internal movements in the bulk material = 451.381 lbf = 2007.844 N

Total Tension increase in flight 1 is = 5476.530 lbf = 24360.809 N

Total Tension till the end of flight 1 is = 9902.730 lbf = 44049.521 N

Details of flight 2:

Is there Horizontal turn in the flight? (y/n): n

Enter Length of the flight (ft): 0

Enter Coefficient of friction between pulley surface and belt surface: .35

Enter Angle of wrap on pulley(deg): 210

Total Tension increase in flight 2 is = -7157.147 lbf = -31836.563 N

Enter Belt Thickness (in): .787

Enter no. of Drive pulleys: 1

Enter Shaft Diameter of Drive pulley (in): 6.26

Enter Pulley Diameter of Drive pulley (in): 24.8

Enter Weight of Drive pulley (lbf): 3596.94

Enter no. of Snub/Bend pulleys: 3

Enter Shaft Diameter of Snub/Bend pulleys (in): 4.92

Enter Pulley Diameter of Snub/Bend pulleys (in): 15.75

Enter Weight of Snub/Bend pulleys (lbf): 1798.47

Total Resistance due to pulleys is = 220.418 lbf = 980.470 N

Actual Total Tension till the end of flight 1 is = 10123.148 lbf = 44975.933 N

Actual Total Tension till the end of flight 2 is = 2966.001 lbf = 13193.427 N

Maximum Tension in the conveyor is = 10123.148 lbf = 44975.933 N