

CALCULATION OF CORRECT RADII FOR VERTICAL CURVES IN TROUGHED BELT CONVEYORS

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INTRODUCTION

The profile of a belt conveyor often changes direction. In the case of a troughed conveyor cross section, the inner portion of the belt will travel a different distance compared to the outer edges. It is necessary that the change of direction does not induce undesirable effects in the belt.

This paper develops the theory of stresses induced in a troughed belt where the angle of inclination changes. It provides methods of calculating the theoretical line that the belt should follow and discusses the challenges facing the building of a conveyor in order for the belt to follow the theoretical line. Practical details of the conveyor layout are not included.

The equations for calculating vertical curve radii listed in four commonly used reference works are compared to the equations derived in this paper. The reference works are Conveyor Equipment Manufacturer's Association (CEMA) edition 5, CEMA edition 6, Mechanical Handling Engineers Association (MHEA) and Conveyor Manufacturer's Association (CMA) diploma course notes.

The paper addresses common issues that result in the incorrect sizing and layout of conveyors having changes in vertical direction.

SYMBOLS USED IN THIS PAPER

α	=	idler wing roll angle (degrees)
γ	=	change of angle factor
θ	=	change in angle at a vertical curve (degrees)
B	=	belt mass (kg/m)
E	=	belt modulus (kN/m)
f_z	=	combined curve bedding and belt mass loss factors
g	=	gravitational constant (m/s^2)
I_e	=	Spacing of the idlers in a convex curve (m)
R_{11}	=	minimum radius at concave curve to prevent belt lifting (m)
R_{12}	=	minimum radius at concave curve to prevent overstress in the centre of the belt (m)
R_{13}	=	minimum radius at concave curve to prevent belt edges from buckling (m)
R_{21}	=	minimum radius at a convex curve to prevent buckling in the centre of the belt (m)
R_{22}	=	minimum radius at a convex curve to prevent overstress of the belt edges (m)

R_{23}	=	minimum radius at a convex curve to limit change of angle at an idler to 30 minutes (m)
R_a	=	concave curve radius (m)
R_e	=	convex curve radius (m)
s	=	length of arc (m)
t_r	=	maximum recommended belt tension (kN/m)
T_0	=	lowest tension (N)
T_{ac}	=	tension at the lowest tangent point of a concave curve (kN)
T_c	=	average tension in a curve (kN)
T_{ec}	=	tension at a convex curve (kN)
T_{min}	=	minimum allowable belt tension at convex curve to prevent buckling (kN)
T_r	=	full width rated tension of the belt (kN)
W	=	belt width (m)

VERTICAL CURVES

There are two types of vertical curve. A convex vertical curve joins two sections of conveyor that have decreasing angle of inclination. The second type of vertical curve is a concave curve and this joins two sections of conveyor that have increasing angle of inclination.

This paper discusses only three roll troughed belt conveyors because it is the trough that imposes any degree of complexity to the design of a curve. The same principles can be extended to five roll troughed idlers.

At any particular position along the conveyor, the belt tension continuously changes with different operating criteria and loading conditions. However, under the same operating criteria and loading conditions, the tension at any point along the conveyor will always be the same.

The first step in design of vertical curves is to calculate the tension at the curve for sets of conditions that represent the extremes. Since the curve geometry is not known, the position used for the calculations is the intersection point of the imaginary lines extended from the sections of conveyor joined by the curve. (Figure 1).

One of the common errors in designing vertical curves is incorrectly estimating the extremes in tension at the intersection point. The upper limit of tension normally occurs when the fully loaded conveyor is started, but if the section preceding the curve has a negative inclination this may not be the case. The tension at any point along the conveyor is the sum of the effective tension at the point plus the adjusted slack side tension. Adjusted slack side tension is the tension applied by the take-up adjusted by an amount to drive the belt between the take-up and the point under consideration, plus any belt slope tension.

Having established the extremes of tension at the curve, the radius of curvature must be chosen to satisfy these criteria.

1. Concave curves

- 1.1 The belt does not lift out of the conveyor structure
- 1.2 The centre of the belt is not overstressed
- 1.3 The belt edges do not buckle due to loss of tension.

2. Convex curves

- 2.1 The centre of the belt does not buckle due to loss of tension
- 2.2 The belt edges are not overstressed.

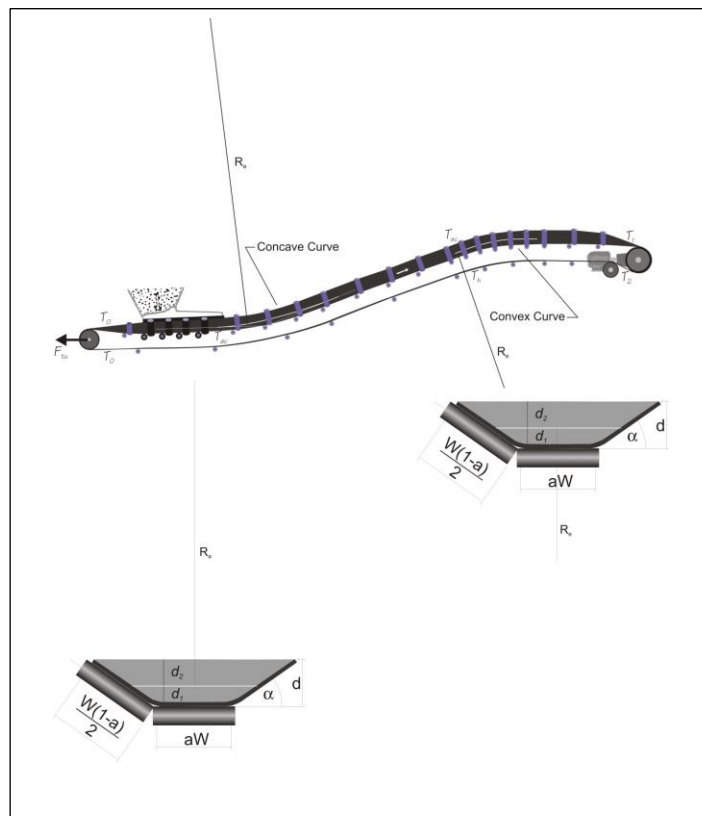


Figure 1. Arrangement of vertical curves

In a concave curve the belt sits on the idlers under its own weight and the weight of the material it is conveying. Tension in the belt tends to pull it out of the idlers. When the belt lifts out of the idlers the trough shape flattens, the belt can foul with other structure or chutes and belt tracking provided by the troughed idlers is lost. All of these are undesirable as they can be detrimental to efficient operation. When the belt is unsupported it will naturally follow part of a catenary curve.

Consider the arbitrary catenary curve shown in Figure 2. Suppose the section from A to P represents a portion of belt that is hanging under its own weight and the tensions T_0 and T at each end of the portion of belt. Since the belt is in a state of equilibrium, the forces must all balance. If the unit mass of the belt is B and the length of the belt between A and P is s , the vertical forces are Bsg and $T\sin\theta$. Horizontal forces are T_0 and $T\cos\theta$.

So

$$T_0 = T \cos \theta$$

And

$$Bsg = T \sin \theta$$

Or

$$\frac{Bsg}{T_0} = \frac{T \sin \theta}{T \cos \theta}$$

From which

$$T_0 = \frac{Bsg}{\tan \theta} \tag{1}$$

When the radius is large and the change in angle small, the portion of the catenary curve is closely represented by the arc of a circle. The arc length for the circle is $R\theta$ and hence s can be replaced by $R\theta$.

The arc length s is also given by

$$s = b \tan \theta$$

Where $b = \frac{T_0}{Bg}$

So equation 1 can be written in terms of a radius R as follows

$$R = \frac{T_0 \tan \theta}{Bg\theta}$$

Introducing a change in angle factor $\gamma = \frac{\tan \theta}{g\theta}$ and using the specific symbol for tension at the beginning of the curve, the equation for determining the minimum radius to prevent belt lifting off the conveyor carry idlers in a concave curve is

$$R_{11} = \frac{1000\gamma T_{ac}}{B} \tag{2}$$

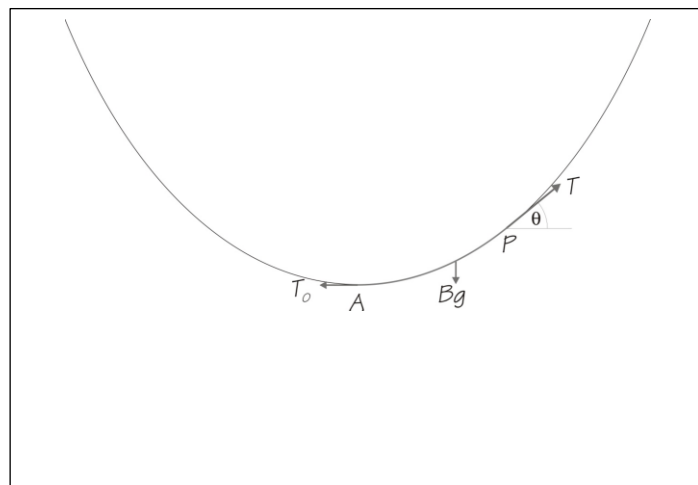


Figure 1. Catenary curve

Since the change in angle is only known if the radius is known, it would be convenient to use a constant angle factor. The largest change in angle that would be encountered at a vertical curve is 30 degrees, for which the angle factor $\frac{\tan \theta}{g\theta} = 0.112$.

Thus, Equation 2 can be re-written as

$$R_{11} = \frac{112T_{ac}}{B}$$

To calculate the difference in tension at belt edges and centre of the trough, the tension gradient in the belt must be understood. The belt, when troughed by three-roll idlers is formed into a shape that roughly approximates a trapezoid. There exists a centroid plane parallel to the bottom of the trough that intersects with the sloped belt edges.

The position of this plane is d_1 from the bottom of the trapezoid shape and d_2 from the top.

$$d_1 = \frac{1}{W} W(1-a) \frac{W}{4} (1-a) \sin \alpha$$

For $a = 1/3$, that is three equal roll trough idlers arranged so that the centre of the belt and the two raised belt edges are all of equal length equal to $W/3$.

$$d_1 = \frac{W}{9} \sin \alpha$$

In terms of the depth of the trough d

$$d = \frac{W}{3} \sin \alpha$$

So

$$d_1 = \frac{d}{3}$$

This means that at a position one third of the depth above the centre of the belt, the belt tension is equal to the calculated tension. Below the centroid plane, the belt must travel a greater distance thereby inducing additional tension. The length of belt along the centroid plane (neutral axis) between the two tangent points is $R_a \theta$. The distance along the bottom of the trough is $(R_a + \frac{d}{3}) \theta$. The difference in length is $\frac{d}{3} \theta$ and therefore the strain $\epsilon = \frac{d}{3R_a}$.

The additional force imposed is $\Delta T = \epsilon E$

That is

$$\Delta T = \frac{dE}{3R}$$

or substituting for d

$$R_a = \frac{W \cdot E \sin \alpha}{9\Delta T}$$

The additional force must be limited to the difference between the rated belt tension and the tension at the curve (in the neutral plane). So the radius to prevent overstressing the centre of the belt in a concave curve is

$$R_{12} = \frac{W \cdot E \sin \alpha}{9 \left(t_r - \frac{T_c}{W} \right)}$$

In the concave curve the belt edges travel through a shorter distance than the centre. Therefore belt edges in the concave curve are at a lower tension than at the neutral plane. The reduction in force is the result of compressive strain which in this case is $\epsilon = \frac{2d}{3R_a}$ associated with a force of

$$\Delta T = \epsilon \cdot E = \frac{2W \cdot E \sin \alpha}{9R_a}$$

So the radius to prevent belt edge buckling is

$$R_{13} = \frac{W \cdot E \sin \alpha}{4.5 \left(\frac{T_{ac}}{W} - T_{min} \right)}$$

A very similar situation applies in a convex curve. However, here there is no possibility of belt lift and the tension gradient goes from lowest in the centre of the belt to highest at the belt edges.

The strain at the belt edges located at two thirds the trough depth above the neutral plane is

$$\epsilon = \frac{2h}{3R}$$

And hence the additional tension at the belt edge is

$$\Delta T = \epsilon E = \frac{2W \cdot E \sin \alpha}{9R}$$

And hence

$$R_{21} = \frac{W \cdot E \sin \alpha}{9 \left(t_r - \frac{T_{ec}}{W} \right)}$$

To prevent buckling in the centre of the belt

$$R_{22} = \frac{W \cdot E \sin \alpha}{4.5 \left(\frac{T_{ec}}{W} - T_{min} \right)}$$

As a consequence of the radial forces, the belt is pulled into the idlers at the convex curve. Instead of following a curved path, the belt will tend to follow a path represented by a series of straight lines joining the tops of the idler rolls, as depicted in Figure 3. In the case of three-roll offset idlers in the convex curve, there are two bend points for each idler set; the centre roll and the wing roll on each side. Figure 4 shows the profile that is adopted when offset idlers are used.

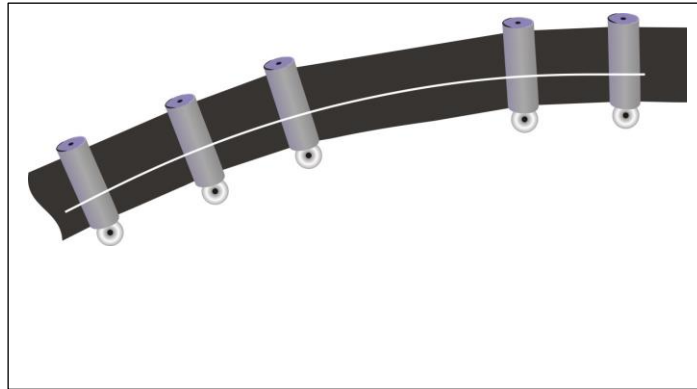


Figure 3. Convex curve detail

The change in angle at any idler should be no more than 30 minutes and only in-line idlers should be used. To keep the change in angle below 30 minutes requires either very close idler spacing or a very large radius. Based on dimensions of the idlers and practical considerations, a minimum spacing between idlers in the convex curve is determined. This leads to a third, more practical, limit to the radius of a convex curve

$$R_{23} = 114I_e$$

The equations for determining the minimum radius of a vertical curve are given in every conveyor handbook. As examples, the equations from four commonly used reference manuals are listed in Appendix 1. They are stated exactly as they are written in the quoted reference. At first glance there appear to be differences in the equations for calculating vertical curve radii. These equations, when simplified and expressed in the same units of measure, are listed below and compared to the derived equations.

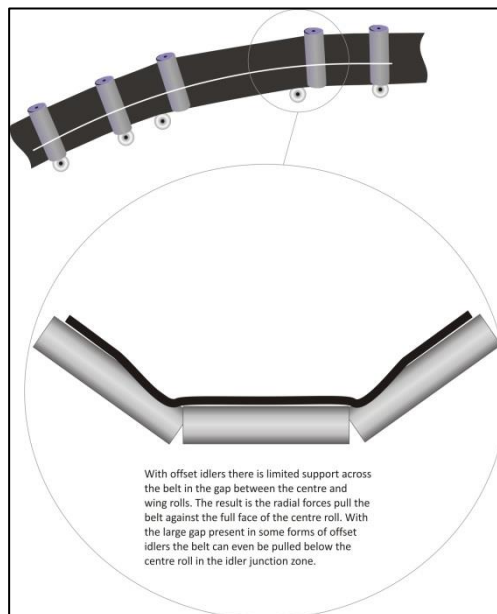


Figure 4. Offset idlers in convex curve

COMPARISON OF EQUATIONS FOR CONCAVE CURVES

Minimum radius to prevent belt lifting from the idlers

Derived

$$R_{11} = \frac{1000\gamma T_{ac}}{B}$$

CEMA 5	CEMA 6	MHEA	CMA
$R_{11} = \frac{113T_{ac}}{B}$	$R_{11} = \frac{113T_{ac}}{B}$	$R_{11} = \frac{113T_{ac}}{B}$	$R_{11} = \frac{123 \cdot f_z \cdot T_{ac}}{B}$

Minimum radius to prevent over stress of centre of the belt

Derived

$$R_{12} = \frac{W \cdot E \cdot \sin \alpha}{9 \left(t_r - \frac{T_c}{W} \right)}$$

CEMA 5	CEMA 6	MHEA	CMA
$R_{12} = \frac{W \cdot E \cdot \sin \alpha}{9 \left(t_r - \frac{T_c}{W} \right)}$	$R_{12} = \frac{W \cdot E \cdot \sin \alpha}{9 \left(t_r - \frac{T_c}{W} \right)}$	$R_{12} = \frac{W \cdot E \cdot \sin \alpha}{9 \left(t_r - \frac{T_c}{W} \right)}$	$R_{12} = \frac{W \cdot E \cdot \sin \alpha}{9 \left(0.1T_b - \frac{T_c}{W} \right)}$ For textile reinforced belts or $R_{12} = \frac{W \cdot E \cdot \sin \alpha}{4.5 \left(0.1T_b - \frac{T_c}{W} \right)}$ For steel cord reinforced belts

Minimum radius to prevent buckling of the belt edges

Derived

$$R_{13} = \frac{W \cdot E \sin \alpha}{4.5 \left(\frac{T_{ac}}{W} - T_{min} \right)}$$

CEMA 5	CEMA 6	MHEA	CMA
$R_{13} = \frac{W \cdot E \cdot \sin \alpha}{4.5 \left(\frac{T_c}{W} - 5.26W \right)}$ For textile reinforced belts Or $R_{13} = \frac{W \cdot E \cdot \sin \alpha}{11.25 \left(\frac{T_c}{W} - 5.26W \right)}$ For steel cord reinforced belts	$R_{13} = \frac{W \cdot E \cdot \sin \alpha}{4.5 \left(\frac{T_c}{W} - 5.26W \right)}$ For textile reinforced belts Or $R_{13} = \frac{W \cdot E \cdot \sin \alpha}{4.5 \left(\frac{T_c}{W} - 1.5T_c \right)}$ For steel cord reinforced belts	$R_{13} = \frac{W \cdot E \sin \alpha}{4.5 \left(\frac{T_c}{W} - 4.4W \right)}$	$R_{13} = \frac{W^2 \cdot E \sin \alpha}{4.5T_{ac}}$ For textile reinforced belts Or $R_{13} = \frac{W^2 \cdot E \sin \alpha}{12T_{ac}}$ For steel cord reinforced belts

COMPARISON OF EQUATIONS FOR CONVEX CURVES

Minimum radius to prevent buckling of the centre of the belt

CEMA 5	CEMA 6	MHEA	CMA
$R_{21} = \frac{W \cdot E \cdot \sin \alpha}{9(T_c - 5.26W)}$	$R_{21} = \frac{W \cdot E \cdot \sin \alpha}{9(T_c - 5.26W)}$	$R_{21} = \frac{f_t \cdot W \cdot E \cdot \sin \alpha}{1.125(T_c - 4.4W)}$	$R_{21} = \frac{W \cdot E \cdot \sin \alpha}{9(T_c - 0.05T_r)}$

Minimum radius to prevent overstress at the belt edges

CEMA 5	CEMA 6	MHEA	CMA
$R_{22} = \frac{W \cdot E \cdot \sin \alpha}{4.5(T_r - T_c)}$	$R_{22} = \frac{W \cdot E \cdot \sin \alpha}{4.5(T_r - T_c)}$	$R_{22} = \frac{f_t \cdot W \cdot E \cdot \sin \alpha}{2.25T_r}$	$R_{22} = \frac{W \cdot E \cdot \sin \alpha}{4.5(1.15T_r - T_c)}$

CONCLUSION

The CEMA equations in both the 5th and 6th editions are consistent with the derived equations. In the case of concave curves the MHEA equations for calculating minimum radii are also consistent with the derived equations. However, the MHEA equations for convex curves differ from derived. This is particularly so for minimum radius to prevent buckling. Using the MHEA equation will result in a radius of between six and four times greater than would be calculated by the derived equation. In the case of calculating a radius required to prevent overstress, the MHEA equation is the same as the derived equation when the tension factor of 2 is used, provided the rated belt tension used is the difference between maximum permissible belt tension and the tension in the curve. Likewise, when the lowest tension factor of 1.3 is used, the calculated minimum radius will be 1.6 times greater than when using the derived equation.

APPENDIX 1

EQUATIONS FOR CALCULATION OF VERTICAL CURVE RADII FROM FOUR REFERENCE SOURCES

Concave Curves

1. Minimum radius to prevent belt lifting from the idlers.

- 1.1 CEMA 5 equation (p. 227)
and CEMA 6 equation (p. 235)

$$r_1 = \frac{1.11T_c}{W_b}$$

Where

T_c = belt tension at point c (lbf)
 W_b = weight of belt (lbf/ft)

The metric equivalent is

$$r_1 = \frac{1.11T_c}{gB}$$

Where

T_c = belt tension at point c (N)
 g = acceleration of gravity (m/s²)
 B = belt mass (kg/m)

- 1.2 MHEA equation (Page 142)

$$R = \frac{1.11T_c}{gB_m}$$

- 1.3 CMA diploma course notes (pp. 6-2)

$$R = \frac{T_x \times 10^3 \cdot f_c \cdot f_b \cdot f_m \cdot f_s}{g \cdot B}$$

Where

T_x = tension at the lower tangent point (kN)
 B = belt mass (kg/m)
 g = acceleration of gravity (m/s²)
 f_b = curve bedding factor (value of 1.1)
 f_c = starting factor
 f_m = mass loss factor (value of 1.1 to allow for 10% loss of belt mass)
 f_s = slope factor $f_s = \frac{1}{\cos^2 \alpha}$
 α = change in slope at the curve.

2. Radius to prevent overstress of centre of the belt

2.1 CEMA5 equation (p. 278)

$$r_1 = \frac{(Factor\ B)(b^2)(B_m)(p)}{T_r - T_c}$$

Where

b = belt width (in)

$$Factor\ B = \frac{\sin \alpha}{12 \times 9}$$

B_m = elastic modulus of belt (lbf/in/ply)

p = number of plies

α = trough angle (degree)

2.2 CEMA 6 equation (p. 236)

$$r_1 = \frac{B_m \times BW^2 \times p}{T_r - T_c} \times \frac{\sin \theta}{48} \times \left(1 - \frac{l_c}{BW}\right)^2$$

Where

BW = belt width (in)

B_m = elastic modulus of belt (lbf/in/ply)

p = number of plies

θ = trough angle (degree)

$$l_c = \frac{BW}{3}$$

T_r = rated belt tension (lbf)

T_c = tension at the curve (lbf)

2.3 MHEA equation (p. 142)

$$R_2 = \frac{(\sin b)(w)(Mod)}{9 \times (T_a - T_c)}$$

Where

w = belt width (m)

Mod = Modulus of elasticity of the belt (kN/m)

T_c = Tension in the belt at the curve (kN)

T_a = Maximum allowable tension for the belt (kN)

b = troughing angle (degree)

2.4 CMA equation (p. 6-3)

$$R = \frac{E \cdot W \cdot \sin \beta}{f_g \cdot \left[(0.1 \cdot T_b) - \left(\frac{T_x \times 10^3}{W} \right) \right]}$$

Where

T_b = belt class (kN/m)

f_g = belt factor = 9000 for textile reinforced belts

= 4500 for steel cord reinforced belts

W = belt width (mm)

β = idler wing roll angle (degrees)

E = Belt modulus (kN/m)

T_x = Tension at the curve (kN)

3. Radius to prevent buckling of the belt edges

3.1 CEMA 5 equation (p. 244)

$$r_1 = \frac{(Factor\ A)(b^2)(B_m)(p)}{f_b(T_c - 30b)}$$

Where

b = belt width (in)

$$Factor\ A = \frac{\sin \alpha}{12 \times 4.5}$$

B_m = elastic modulus of belt (lbf/in/ply)

p = number of plies

α = trough angle (degree)

T_c = tension at the curve (lbf)

f_b = belt factor = 1 for textile reinforced belting

= 2.5 for steel cord reinforced belting

3.2 CEMA 6 equation (p. 236)

$$r_1 = \frac{B_m \times BW^2 \times p}{T_c - \sigma_a \times BW} \times \frac{\sin \theta}{48} \times \left(1 - \left(\frac{l_c}{BW} \right)^2 \right)$$

Where

BW = belt width (in)

B_m = elastic modulus of belt (lbf/in/ply)

p = number of plies

θ = trough angle (degree)

$$l_c = \frac{BW}{3}$$

T_c = tension at the curve (lbf)

σ_a = maximum allowable edge stress (lbf/in)

= $75 - 1.5 \times \frac{T_c}{BW}$ for steel

cord reinforced belts

= 30 for textile reinforced

belts

3.3 MHEA equation (p. 142)

$$R_2 = \frac{(\sin b)(w)(Mod)}{4.5 \times (T_c - 4.4w)}$$

Where

w = belt width (m)

Mod = Modulus of elasticity of the belt (kN/m)

T_c = Tension in the belt at the curve (kN)

T_a = Maximum allowable tension for the belt (kN)

b = troughing angle (degree)

3.4 CMA equation (p. 6-3)

$$R = \frac{W \cdot (E \cdot W \cdot \sin \beta)}{f_d \cdot T_x \cdot 10^6}$$

Where

T_b = belt class (kN/m)

f_d = belt factor = 4.5 for textile reinforced belts
= 12 for steel cord reinforced belts

W = belt width (mm)

β = idler wing roll angle (degrees)

E = Belt modulus (kN/m)

T_x = Tension at the curve (kN)

Convex Curves

4. Radius to prevent buckling of centre of the belt

4.1 CEMA 5 equation (p. 278)

$$r_1 = \frac{(Factor D)b^2(B_m)(p)}{T_c - 30b}$$

Where

b = belt width (in)

$Factor D = \frac{\sin \alpha}{12 \times 9}$

B_m = elastic modulus of belt (lbs/in/ply)

p = number of plies

α = trough angle (degree)

T_c = tension at the curve (lbs)

4.2 CEMA 6 equation 9.14 (p. 240)

$$r_2 = \frac{(B_m)(BW^2)(p)}{T_c - 30 \times BW} \times \frac{\sin(\theta)}{48} \times \left(1 - \frac{l_c}{BW}\right)^2$$

Where

BW = belt width (in)

B_m = elastic modulus of belt (lbf/in/ply)

p = number of plies

θ = trough angle (degree)

$l_c = \frac{BW}{3}$

T_c = tension at the curve (lbf)

4.3 MHEA formula 5 (p. 146)

$$R_2 = \frac{(\sin b)(w)(Mod)(F)}{1.125 \times (T_c - 4.4w)}$$

Where

w = belt width (m)

Mod = Modulus of elasticity of the belt (kN/m)

T_c = Tension in the belt at the curve (kN)

b = troughing angle (degree)

F = Factor between 1.3 and 2 depending on percentage of maximum allowable tension that exists at the curve.

4.4 CMA equation 14 (p. 6-9)

$$R = \frac{E \cdot W \cdot \sin \beta}{9000 \left[\frac{(1000 \cdot T_c)}{W} - \left(\frac{0.05 \cdot T_b}{F_x} \right) \right]}$$

Where

T_b = belt class (kN/m)

F_x = belt safety factor = 10 for textile reinforced belts
= 6.67 for steel cord reinforced belts

W = belt width (mm)

β = idler wing roll angle (degrees)

E = Belt modulus (kN/m)

T_c = Tension at the curve (kN)

5. Radius to prevent overstress of the belt edges

5.1 CEMA 5 equation 5 (p. 254)

$$r_2 = \frac{(Factor C)b^2(B_m)(p)}{T_r - T_c}$$

Where

b = belt width (in)

$Factor C = \frac{\sin \alpha}{12 \times 4.5}$

B_m = elastic modulus of belt (lbs/in/ply)

p = number of plies

α = trough angle (degree)

T_c = belt tension at the curve (lbs)

T_r = rated belt tension (lbs)

5.2 CEMA 6 equation 9.15 (p. 240)

$$r_2 = \frac{(B_m)(BW^2)(p)}{T_r - T_c} \times \frac{\sin(\theta)}{48} \times \left(1 - \left(\frac{l_c}{BW} \right)^2 \right)$$

Where

BW = belt width (in)
 B_m = elastic modulus of belt (lbf/in/ply)
 p = number of plies
 θ = trough angle (degree)
 $l_c = \frac{BW}{3}$
 T_r = rated belt tension (lbf)
 T_c = tension at the curve (lbf)

5.3 MHEA equation (p. 146)

$$R_1 = \frac{(\sin b)(w)(Mod)(F)}{2.25 \times T_a}$$

Where

w = belt width (m)
 Mod = Modulus of elasticity of the belt (kN/m)
 T_a = Maximum allowable tension for the belt (kN)
 b = troughing angle (degree)
 F = Factor between 1.3 and 2 depending on per cent of maximum allowable tension that exists at the curve.

5.4 CMA equation 13 (p. 6-9)

$$R = \frac{E \cdot W \cdot \sin \beta}{4500 \left[\left(\frac{1.15 \cdot T_b}{F_x} \right) - \left(\frac{1000 T_c}{W} \right) \right]}$$

Where

T_b = belt class (kN/m)
 W = belt width (mm)
 β = idler wing roll angle (degrees)
 E = Belt modulus (kN/m)
 T_c = Tension at the curve (kN)
 F_x = belt safety factor = 10 for textile reinforced belts
= 6.67 for steel cord reinforced belts

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