Local Stresses in Belt Turnovers in Conveyor Belts

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Belt turnovers are commonly used in conveyors to rotate the belt so that the clean side of the belt contacts the return idlers. The belt turnover length has traditionally been sized on only the twisting stresses. However, belt turnovers are also subjected to bending stresses due to the weight of the belt. Although bending stresses have in the past been ignored, these stresses are significant and should not be neglected. This paper quantifies and presents a method for calculated twisting and bending stresses and also the belt sag.

INTRODUCTION

Belt turnovers are commonly used in conveyors to rotate the belt so that the clean side of the belt contacts the return idlers. This prevents contamination of the return strand idlers. The benefits of turnovers include cleaner return idler rolls, longer life of the return idlers, and reduced belt vibrations.

Turnovers can be a cause of early belt failure if incorrectly designed. The edge stresses are significantly increased and if they are too high will cause splice or cable failure. Center belt stresses are decreased and can cause buckling if the stresses are negative. Buckling can result in cable failure and cover delamination. Further, both of these negative effects can occur simultaneously in a turnover. It is imperative that the turnover be correctly designed in acceptable operating limits to prevent these problems.

The belt turnover length has traditionally been sized only on the twisting stresses. However, belt turnovers are also subjected to bending stresses due to the weight of the belt and long turnover span. Although bending stresses have in the past been ignored, these stresses are significant and cannot be neglected. This paper quantifies and presents a method for calculating twisting and bending stresses and also the belt sag in turnovers.

The purpose of this paper is to:

1. Set forth good turnover design limits.
2. Show that bending stresses in turnovers must be quantified and considered during design.
3. Present calculation methods to determine twisting and bending stresses in flat helix turnover.
4. Show the effect of quarter point support rolls in flat helix turnovers.

Previous work in turnovers includes determination of the strains distribution including the effects of rubber shear deformation (Oehmen, 1979) and the description of the Mordstein turnover (Mordstein, 1961).

The turnover calculation methods presented in this paper are somewhat complex and require programming for the engineer to be able to use them. Conveyor Dynamics, Inc. has incorporated the turnover calculations methods presented in this paper into BELTSTAT v7.0, which is available to the general public.

TURNOVER TYPES AND DESIGN CRITERIA

There are two basic types of turnovers, which are: 1) the Mordstein turnover, and 2) the flat helix turnover.

The Mordstein Turnover

The Mordstein turnover bends the belt in the transverse axis over a circular roller. The result is that belt has a semi-circular cross section in the turnover. The semi-circular cross section reduces the twisting stresses and can have shorter turnover lengths than the flat helix turnover.

However, the Mordstein turnover is difficult to design correctly and has resulted in various belt failures. The Mordstein turnover requires a spherical roller. The spherical roller has a differential velocity profile across its face. Therefore, as the belt contacts the spherical roller part of the belt slides in a nonuniform manner. This sliding can result in early roller and/or belt failure. Mordstein turnovers are infrequently used today. Figure 1 shows a Mordstein turnover.

This paper deals only with flat helix turnover and not the Mordstein turnover.

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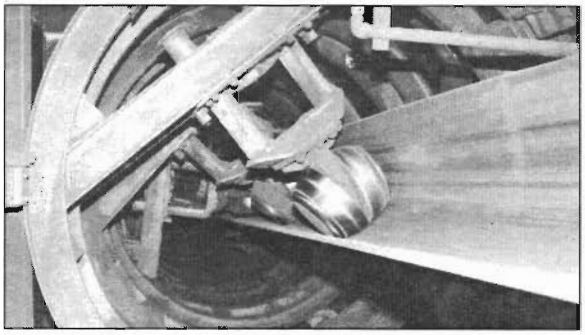

FIGURE 1 Mordstein turnover

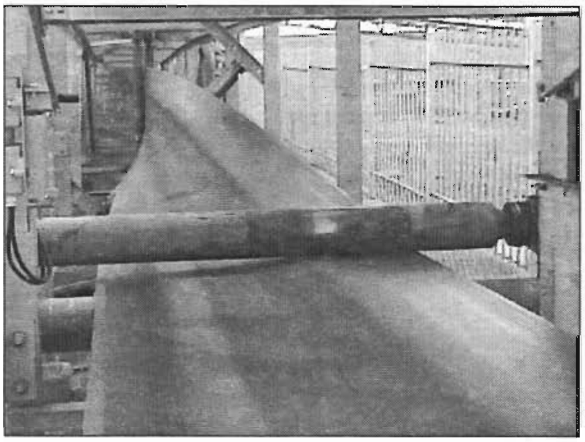

FIGURE 2 Flat helix turnover with quarter point rolls

Flat Helix Turnover

The flat helix turnover places rollers at the center of the turnover length and forces the belt to have a flat cross section. Figure 2 shows a flat helix turnover with quarter point support rolls.

Flat helix turnovers are commonly built in the two following configurations, which are: 1) turnover with support rolls at the turnover quarter points, and 2) turnover with no support rolls. Figures 3 and 4 show these configurations.

The flat helix turnover without support rolls allows the belt to sag freely. This is a simple design, however, the belt sag can be substantial because of the large belt span of the turnover.

The flat helix turnover with quarter point support rolls utilizes a second set of rollers to support the belt. The quarter point support rollers are placed at the quarter and three quarter locations in the turnover and are at a 45° angle.

Turnover Design Criteria

Acceptable turnover design includes the following:

1. Edge stresses must not exceed the acceptable limit. The recommended maximum edge stress is 115% of the rated belt tension per Goodyear's Handbook of Conveyor & Elevator Belting (Goodyear, 1975).
2. Center stresses must be noncompressive to prevent buckling. It is recommended that center stresses be at least 5 N/mm. If the center belt stresses are negative then there is potential for buckling in the belt's center. Buckling can result in cable failure or cover delamination. Figure 5 shows a flat helix turnover with center buckling problems. Note how the belt does not maintain a flat cross section in the turnover.
3. Belt sag should be controlled to acceptable limits. The maximum recommended sag is 1% of the turnover length.
4. Flat helix turnovers require vertical middle guide rolls located at the turnover middle point to maintain proper belt form. These vertical rolls also help in belt training and help prevent belt flapping which can occur due to wind loads (Goodyear, 1975).

The design criteria set forth above is valid for both steel cord and fabric belts. The calculation methods in this paper are likewise valid for both types of belts.

IMPORTANCE OF BENDING STRESSES

Historically, bending stresses in belt turnovers have been ignored. This is likely due to the complex nature of the equations. However, bending stresses should not be ignored as they can significantly affect stresses, especially in heavy wide belts.

To illustrate the significance of bending stresses, an example is used. The belt specifications in the example are:

Width	1,830 mm
Rating	ST-4000 N/mm
Weight	82.6 kg/m
Elasticity	286 kN/mm

Figure 6 shows the edge and center stresses in the belt caused by twisting. The results in this figure ignore bending stresses. The following conclusions are made:

1. Edge stresses due to twisting decrease with increasing length.
2. Center stresses due to twisting increase with increasing length.
3. Twisting stresses (edge and center) are not a function of belt tensions.

If bending stresses are ignored, it (incorrectly) can be argued that it is better to increase the turnover length to avoid high edge stresses or compressive center stresses.

However, bending stresses increase as the turnover length increases. Figure 7 shows the vertical and horizontal bending stresses for a turnover with quarter point support rollers. This figure does not include stresses from twisting or the nominal belt tensile stresses. This figure shows that the bending stresses are important and cannot be ignored.

Since twisting stresses decrease and bending stresses increase with increasing length, there is an optimal length for a belt turnover. Figures 8 and 9 show the minimum (located near the center) and maximum (located at the bottom edge) stresses for the example turnover. All relevant component stresses (belt tension, twisting, and bending) are included in the figures.

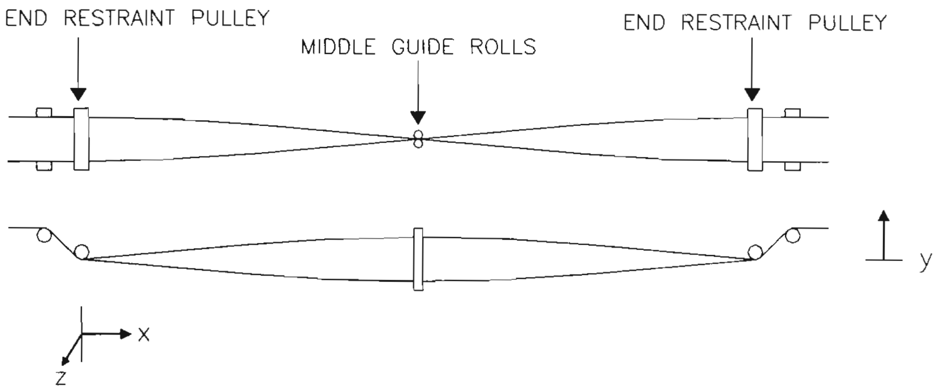

FIGURE 3 Turnover without quarter point support rolls

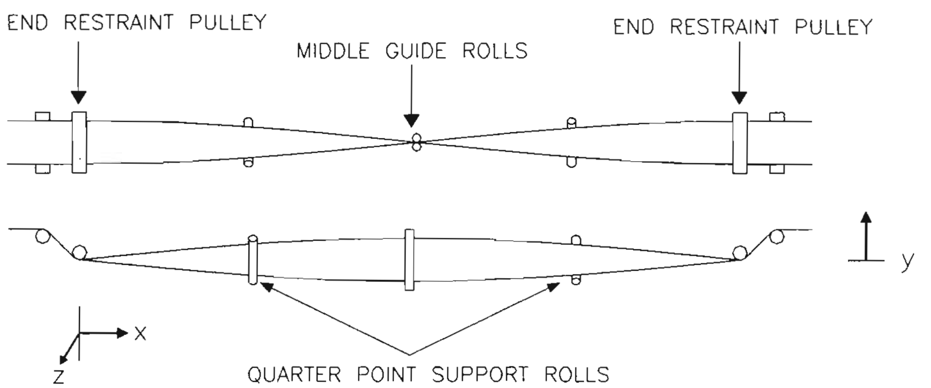

FIGURE 4 Turnover with quarter point support rolls

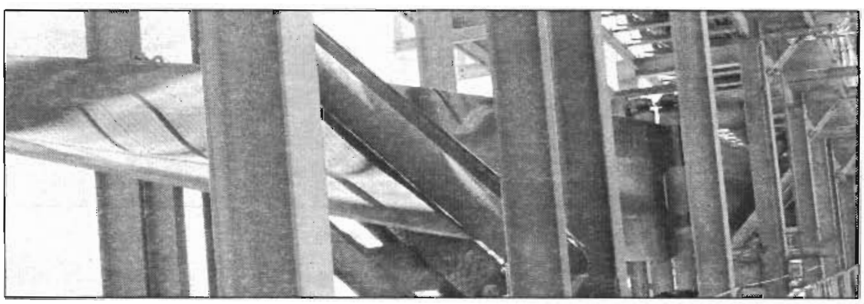

FIGURE 5 Turnover with center buckling problems

Figure 8 shows that there is a minimum required belt tension for the turnover. For the example turnover, the minimum required belt tension is 275 kN. Any belt tension below this value will result in compressive stresses at the belt center. For the example belt, the optimal length is 60 meters (length/belt width = 32.5).

QUARTER POINT SUPPORT ROLLS

Belt turnovers typically have a length of 12 to 40 times the belt width (CEMA, 1997 and Goodyear, 1975). This large span can result in significant belt sag. To reduce belt sag, quarter point support rollers are placed at the quarter and three quarter point locations in the turnover as shown in Figure 4.

The support roll forces the belt to move along the path of the roller plane that is placed at a ±45° angle to the middle vertical support roller. The quarter point support roll applies a force in both the vertical and horizontal planes (Figure 10). The horizontal forces from the rolls apply their forces in opposite direction. The vertical forces are opposite to gravity. The magnitude of the forces depends on the belt tension, weight, elasticity, and turnover length.

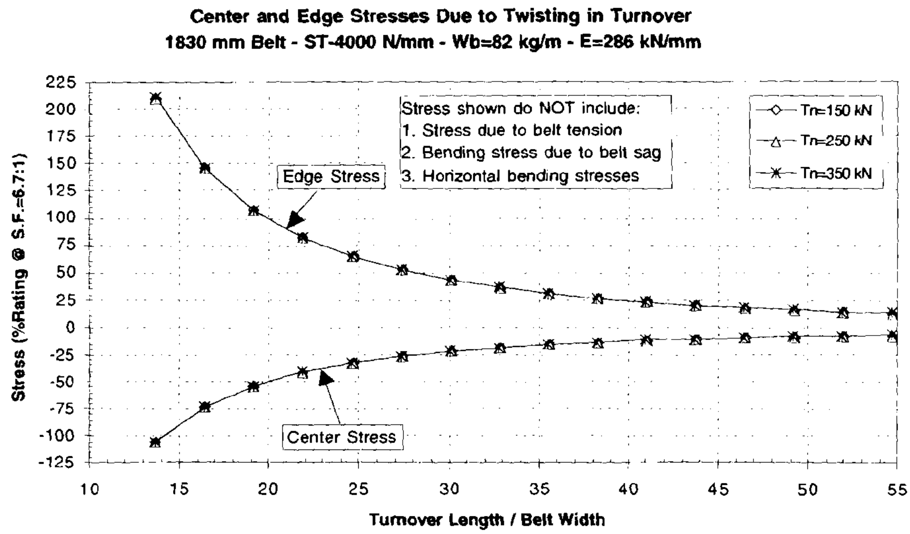

FIGURE 6 Edge and center stresses due to belt twist

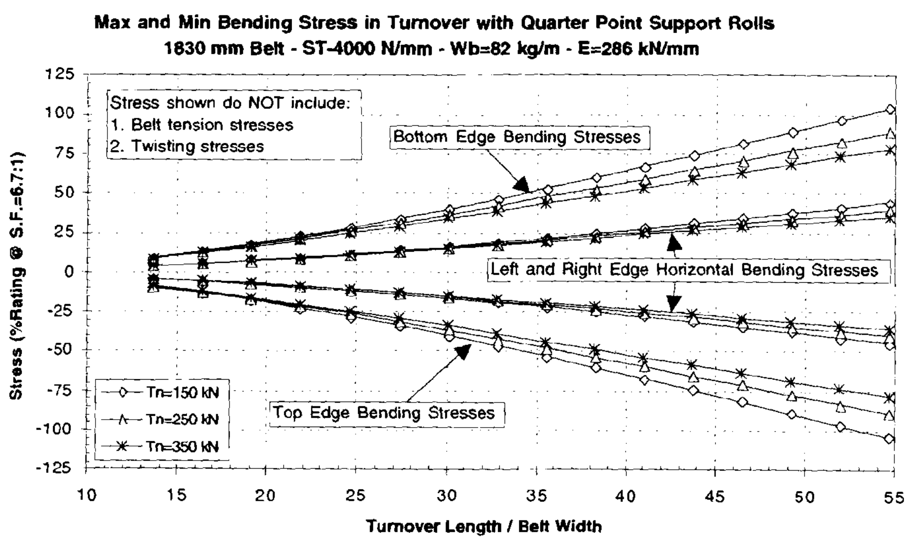

FIGURE 7 Bending stresses

The quarter point support rolls are very effective at reducing belt sag. For the belt to sag, it must also displace in the horizontal plane. Therefore, the horizontal stiffness of the belt acts against the belt sag. Figure 11 shows the belt sag for the example turnover with and without quarter point support rolls. The belt sag is reduced from 1.1% (673 mm) sag to 0.2% (114 mm) sag at low tension. This is a very significant reduction in sag. Also, if the tensions vary in the turnover, then the quarter point support rolls stabilize the belt sag. Without the turnover, the sag varies from 673 mm to 452 mm (total range = 221 mm) for the given tension range. However, the sag in the turnover with quarter point support rolls varies a total of 20 mm.

Finally, the quarter point support rolls reduce the magnitude of stresses in the turnover enabling the placement of a turnover at either higher or lower belt tensions. The support rolls reduce belt tension by limiting the bending stresses in the vertical plane. Figure 12 shows both the maximum and minimum stresses in turnovers with and without quarter point support rolls.

For the selected turnover length of 60 meters, the minimum required belt tension is 290 kN and 320 for turnovers with and without support rolls respectively. At the maximum tension of 550 kN, the edge stress safety factor is 5.5:1 and 5.9:1 for turnovers with and without support rolls respectively.

Quarter point support rolls are recommended because they:

1. Significantly reduce belt sag
2. Significantly reduce the belt sag range in turnovers with a large tension range
3. Reduce the maximum edge stresses

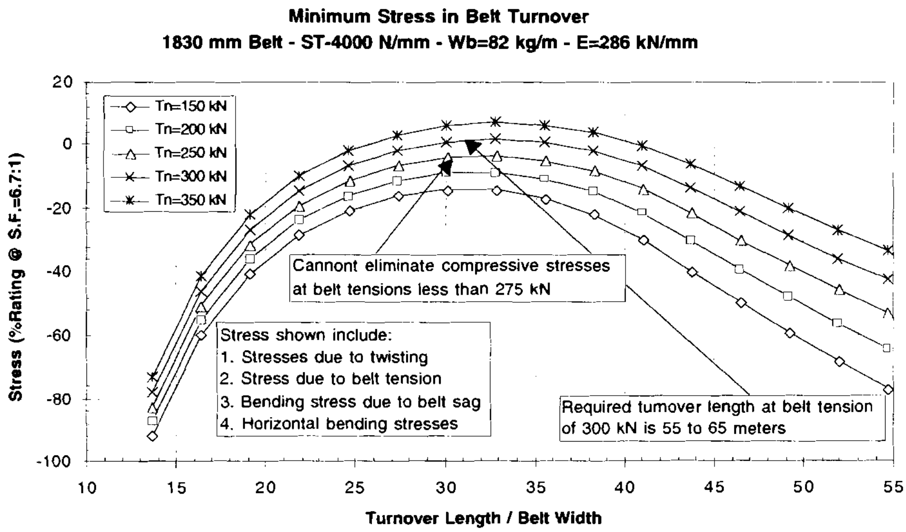

FIGURE 8 Minimum stress in example turnover

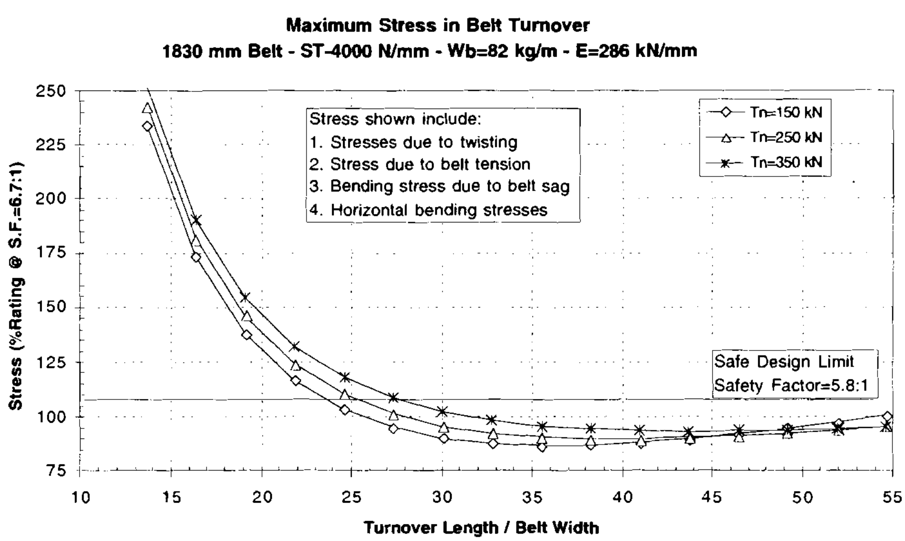

FIGURE 9 Maximum stress in example turnover

4. Increase the minimum center stresses and therefore reduce buckling potential

CALCULATION METHODS

Turnover Twist Stresses

As a belt travels through a belt turnover, the edge stresses increase and the center stresses decrease. The change in stresses are calculated with the following assumptions:

1. The belt is treated as a homogeneous material.
2. The belt is treated as an isotropic material. This assumption is not correct, however, the error is small. The belt is actually an orthotropic material. However, since the longitudinal belt tension is very high, orthotropic material properties would have little effect on the outcome of the stresses and sag.
3. The integration of the belt stresses across its width is equal to the nominal belt tension:

$$T_n = \int_{-bw/2}^{bw/2} \sigma_t \cdot dy \qquad \text{EQ. 1}$$

where: T_n is the nominal belt tension
bw is the belt width
σ_t is the local stress in the belt
dy is an incremental width of the belt

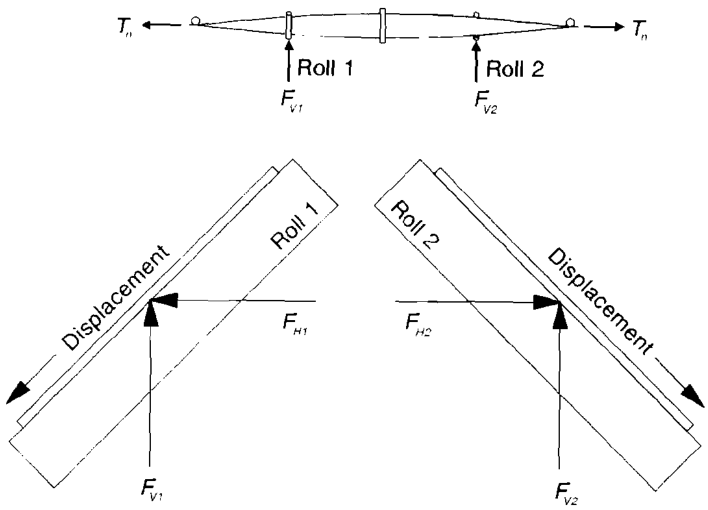

FIGURE 10 Resultant forces from quarter point support rolls

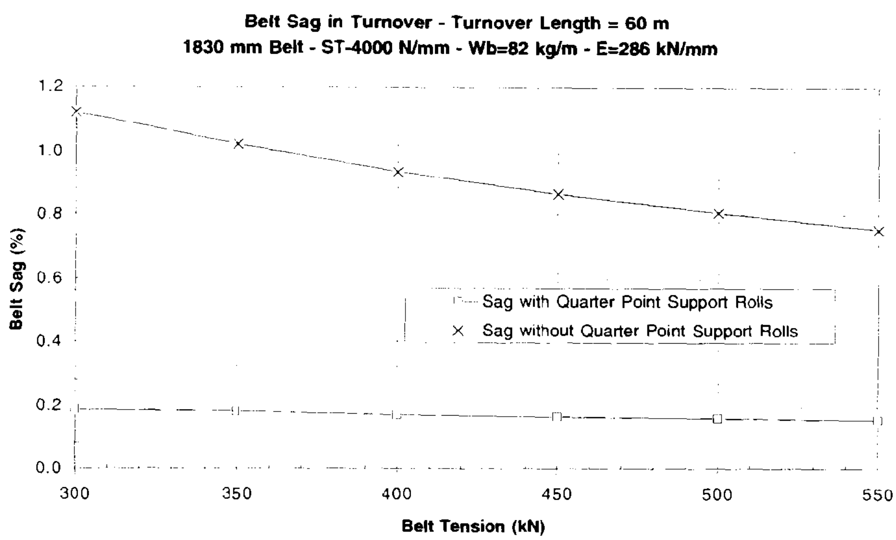

FIGURE 11 Belt sag comparison

4. Shear deformation in the turnover is set to be zero. This assumption implies that the longitudinal displacements across the width of the belt are equal. This is equivalent to taking a belt and cutting it to the length of the turnover then clamping the edge with a rigid body and twisting the belt. This forces a stress discontinuity at the end of the turnover.

   This assumption results in higher edge stresses and lower quarter point stresses than those that really occur, and therefore the calculations are conservative. In the belt there are shear deformations that allow differential displacement across the width of the belt. Figure 13 illustrates the effect of this assumption.

Oehmen presented a method to determine edge and center strains including shear deformation effects (Oehmen, 1979). His method requires the shear deformation material properties of the belt. Oehmen method results in lower edge stresses and higher center stresses. For the maximum stress (located at the bottom edge), the difference in Oehmen's method compared to the one presented here is 1% to 8% of the belt's breaking stress. For the minimum stress (located near the belt's center), the difference is less than 3% of the belt's breaking stress. The difference between the two methods is greatest at low belt tensions and short turnover distances.

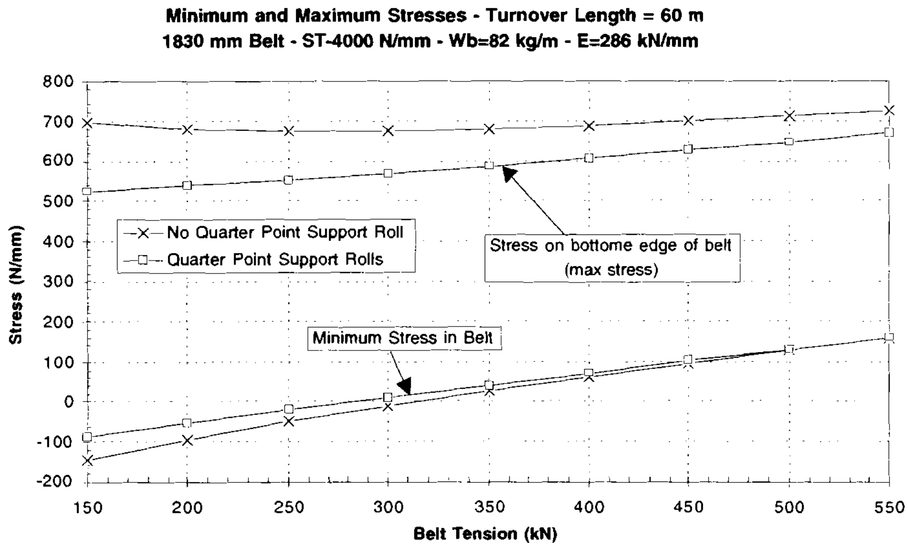

FIGURE 12 Maximum and minimum stress comparison

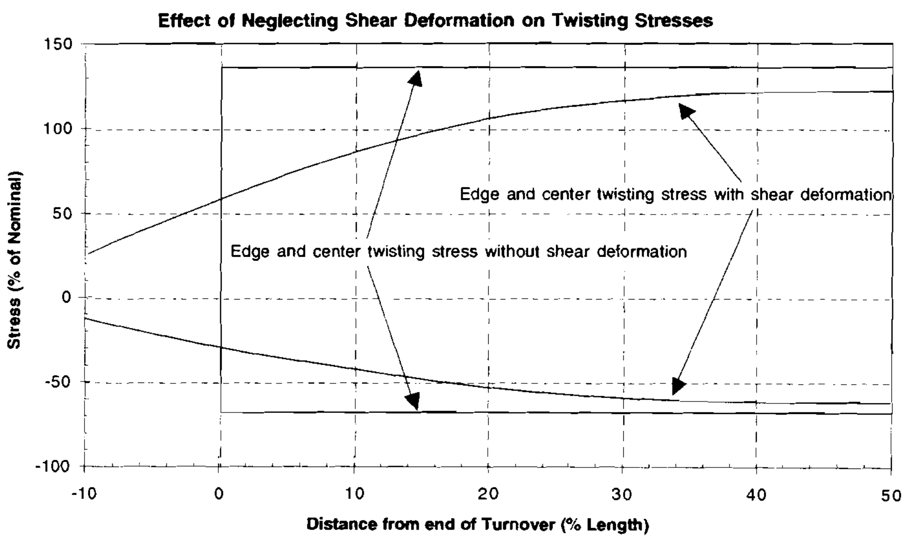

FIGURE 13 Shear deformation effects

5. Bending stresses and displacements in both the horizontal and vertical planes are determined with beam theory.
6. Twisting stresses are calculated by defining the belt's shape in the turnover and then determining the strains. The path of a belt segment in the belt is defined as:

$$\hat{p}(t) = \frac{L}{\pi} \cdot t \cdot \hat{i} + r \cdot \sin(t) \cdot \hat{j} + r \cdot \cos(t) \cdot \hat{k} \qquad \text{EQ. 2}$$

where:

$t = \frac{\pi \cdot x}{L}$

$\hat{i}, \hat{j}, \hat{k}$ = unit vectors in x, y, and z directions

L = length of turnover

r = distance from belt center line to belt segment

$\hat{p}(t)$ = vector path of belt segment

The twisting stresses are determined from this equation. The resulting stress in the turnover (σ_{TO}) is:

$$\sigma_{TO} = \frac{T_n}{bw} + E \cdot \left[\sqrt{\left(\frac{r\pi}{L}\right)^2 + 1} - \frac{1}{2}\sqrt{\left(\frac{bw \cdot \pi}{2L}\right)^2 + 1} - \frac{L}{\pi \cdot bw}\ln\left(\frac{bw \cdot \pi}{2L} + \sqrt{\left(\frac{bw \cdot \pi}{2L}\right)^2 + 1}\right)\right] \qquad \text{EQ. 3}$$

where E is the belt modulus (in N/mm). If the turnover length is at least seven times the belt width then at the belt center line (r = 0):

$$\sigma_{TO,\,r=0} \cong \frac{T_n}{bw} - \frac{E}{3} \cdot \left[\sqrt{\left(\frac{bw \cdot \pi}{2L}\right)^2 + 1} - 1 \right] \quad \textbf{EQ. 4}$$

At the belt's edge (r=bw/2):

$$\sigma_{TO,\,r=bw/2} \cong \frac{T_n}{bw} + \frac{2 \cdot E}{3} \cdot \left[\sqrt{\left(\frac{bw \cdot \pi}{2L}\right)^2 + 1} - 1 \right] \quad \textbf{EQ. 5}$$

This approximation has less than 0.5% error.

This equations define the stress in the turnover at a transverse distance r from the belt's centerline. This equation does *not* include the stresses caused by bending in either the horizontal or vertical planes. These equations will underestimate the stresses in the turnover. Bending stresses must be included in the turnover stress analysis.

Bending Stresses in Belt Turnovers

The following sections present a method to determine the bending stresses and finally to calculate the belt sag in the turnover. The bending stresses are determined according to beam theory. The difficulty in the beam equations is that the bending moment of inertia change as a function of length due to the twisting. Therefore, the equations must be solved numerically as will be shown.

Moment of Inertia. The belt's moment of inertia is not constant in the turnover. The moment of inertia is calculated as a function of the belt orientation along the length of the turnover. The moments of inertia about z and y-axis are:

$$I_{ZZ} = \frac{bw \cdot t_{belt} \cdot (t_{belt}^2 + bw^2)}{24} + \frac{bw \cdot t_{belt} \cdot (t_{belt}^2 - bw^2)}{24} \cdot \cos\left(\frac{2 \cdot \pi \cdot x}{L}\right) \quad \textbf{EQ. 6}$$

$$I_{YY} = \frac{bw \cdot t_{belt} \cdot (t_{belt}^2 + bw^2)}{24} + \frac{bw \cdot t_{belt} \cdot (bw^2 - t_{belt}^2)}{24} \cdot \cos\left(\frac{2 \cdot \pi \cdot x}{L}\right) \quad \textbf{EQ. 7}$$

where:

t_{belt} = effective belt thickness in bending
I_{ZZ} = bending moment of inertia about z-axis
I_{YY} = bending moment of inertia about y-axis

The effective belt thickness is the corrected thickness to get the right bending properties in the x-y plane outside of the turnover. If the real belt thickness is used in the above equation, the bending moments will be overestimated. Belts have little bending resistance and the belt thickness should be adjusted accordingly. The effective belt thickness can be determined from the experimental bending stiffness of the belt. However if unknown, the approximate effective belt thickness (t_{belt}) is:

$$t_{belt} < 0.521 \cdot \sqrt[3]{\frac{NC \cdot d_c}{bw}}$$

where NC is the number of cables in the belt and d_c is the cable diameter. This formula assumes that the bending resistance is from the cables only.

Bending Stresses

The belt sags in the turnover resulting in bending stresses in the vertical plane. The bending stresses are a function of the bending moment, moment of inertia, and the distance from the neutral axis. The bending stress in the vertical plane (σ_V) is:

$$\sigma_v = \frac{M_Z \cdot y}{I_{ZZ}} \quad \textbf{EQ. 8}$$

where: M_Z is the bending moment about the z-axis.

If the turnover is equipped with quarter point support rolls, there will be belt displacements in the horizontal plane also (see Figure 10). The support rolls are at a 45 degree angle from the vertical and therefore apply a force in both the vertical and horizontal planes. The bending stress in the horizontal plane (σ_H) is:

$$\sigma_H = \frac{M_Y \cdot z}{I_{YY}} \quad \textbf{EQ. 9}$$

where: M_Y is the bending moment about the y-axis.

The bending stress must be numerically calculated. Both the bending moment (M_Y, M_Z) and the moment of inertia (I_{YY}, I_{ZZ}) are not linear. Also, the bending moment is different for turnovers with quarter point support rolls and those without support rolls. The bending moment must be solved directly from the differential equations. Figure 14 shows the force diagram in the vertical plane.

For x = 0 to L/4

$$M_Z = -M_V + \left(\frac{wb \cdot L}{2} - F_v\right) \cdot x - \frac{wb \cdot x^2}{2} - T \cdot y \quad \textbf{EQ. 10}$$

For x = L/4 to L/2

$$M_Z = -M_V - \frac{F_v \cdot L}{4} + \frac{wb \cdot L \cdot x}{2} - \frac{wb \cdot x^2}{2} - T \cdot y \quad \textbf{EQ. 11}$$

where:

M_V = bending moment at x = 0
R_V = vertical force applied by end rolls
F_V = vertical force applied by quarter point support rolls
wb = belt weight (force/length)
T = belt tension

In the above equations, the variables M_V and F_V are not known. Variable F_V will be set to zero if there are no quarter point support rolls. These variables are found by solving the differential equation:

$$y'' = -\frac{M_Z}{E \cdot I_{ZZ}} \quad \textbf{EQ. 12}$$

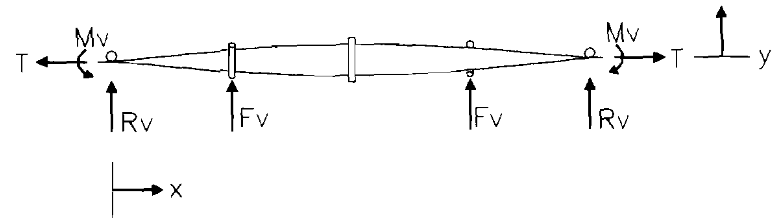

FIGURE 14 Forces in vertical plane

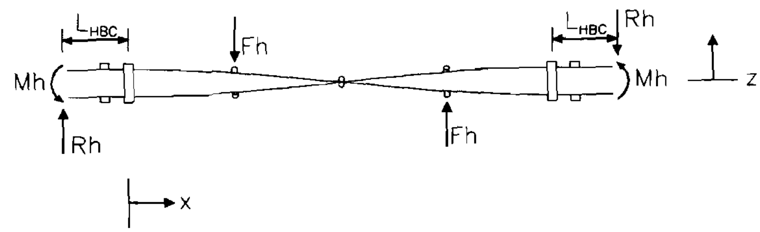

FIGURE 15 Force diagram of belt turnover in horizontal plane

The boundary conditions in the vertical plane are:

1. at x = 0, y = 0, y′ = 0
2. at x = $L/4$, y = z (if quarter point support rolls are present)
3. at x = $L/2$, y′ = 0, y″ = 0

The differential equation cannot be solved in a closed form solution because it is nonlinear. The unknown variables and belt displacement are determined by solving the differential equation numerically. The differential equation is solved using a method such as the Runge-Kutta-Nyström method. Since the variables M_V and F_V are not known they are initially estimated. The differential equation is solved and M_V and F_V are estimated until the boundary conditions are met.

Bending Stresses in Horizontal Plane. If the turnover has quarter point support rolls, there will be curvature in the horizontal plane as well as the vertical plane. Again, the bending stress must be numerically calculated. Both the bending moment (M_Y) and the moment of inertia (I_{YY}) are not linear. The bending moment must be solved directly from differential equations. If there are no quarter point support rollers then the horizontal stresses are zero. Figure 15 shows the forces acting in the horizontal plane.

Figures 14 and 15 show that the quarter point support rolls apply a force in both the vertical and horizontal directions. In the vertical plane, the rolls apply an upward force on the belt and therefore reduce belt sag. However in the horizontal plane, the quarter point support rolls apply a force opposite to each other. The boundary conditions of the belt require that the horizontal displacement is zero at the ends of the turnover. The Horizontal B.C. Length (L_{HBC}) is the location at which the displacement in the horizontal plane is forced to be zero. This variable boundary condition length is included in the analysis to better simulate the real boundary conditions in the turnover. This length must be zero or greater. Increasing the length L_{HBC} will decrease the horizontal bending stresses.

The force equations in the horizontal plane are:

For $x = -L_{HBC}$ to $L/4$

$$M_Y = -M_H + R_H \cdot (L_{HBC} + x) - T \cdot z \qquad \textbf{EQ. 13}$$

For x = L/4 to L/2

$$M_Y = -M_H + R_H \cdot (L_{HBC} + x) - F_H \cdot \left(x - \frac{L}{4}\right) - T \cdot z \qquad \textbf{EQ. 14}$$

where:

$$M_H = R_H \cdot \left(L_{HBC} + \frac{L}{2}\right) - F_H \cdot \frac{L}{4} \qquad \textbf{EQ. 15}$$

The variables are:

M_H = bending moment at x = $-L_{HBC}$
R_H = horizontal force applied by end rolls
F_H = horizontal force applied by quarter point support rolls
L_{HBC} = location at which z is forced to equal 0

In the above equations, the variables M_V and F_V are not known. These variables are found by solving the differential equation:

$$z'' = -\frac{M_Y}{E \cdot I_{YY}} \qquad \textbf{EQ. 16}$$

The boundary conditions in the horizontal plane are:

1. at x = $-L_{HBC}$, z = 0, z′ = 0
2. at x = $L/4$, z=y (if quarter point support rolls are present)
3. at x = $L/2$, z = 0, z″ = 0, M_Y = 0

As in the vertical plane, the differential equation cannot be solved directly and requires a numerical solution. The differential equation in the horizontal plane is solved with the same methodology as the differential equation in the vertical plane.

The second boundary condition in both the horizontal and vertical planes require that $z = y$. This means that the vertical and horizontal differential equation must be solved simultaneously.

CONCLUSIONS

The following items were reviewed in this paper:

1. Acceptable turnover design requirements are reviewed. Turnovers must be designed to prevent buckling in the center of the belt and the edge stresses must not exceed the limiting safety factor. Turnovers that do not meet these criteria subject the belt to early failure.
2. A calculation method is outlined to determine the stress field of the belt in the turnovers. This calculation method can be applied to flat helix turnover with and without quarter point support rolls. The calculation method includes the effects of bending stresses created by the belt's weight in the turnover.
3. Bending stresses cannot be ignored in the turnover. They can be a significant factor in determining the belt length, and minimum and maximum stresses.
4. Quarter point support rolls (located at the quarter and three quarter point locations) in turnovers significantly reduce belt sag and the turnover stresses. The quarter point support rolls allow the tensions in the turnover to be lower or higher as the case may require.

Conveyor Dynamics, Inc. has incorporated the turnover calculations methods presented in this paper into BELTSTAT v7.0. An evaluation copy of BELTSTAT may be obtained at www.conveyor-dyanamics.com.

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