BELT TURNOVER DESIGN USING FINITE ELEMENT ANALYSIS

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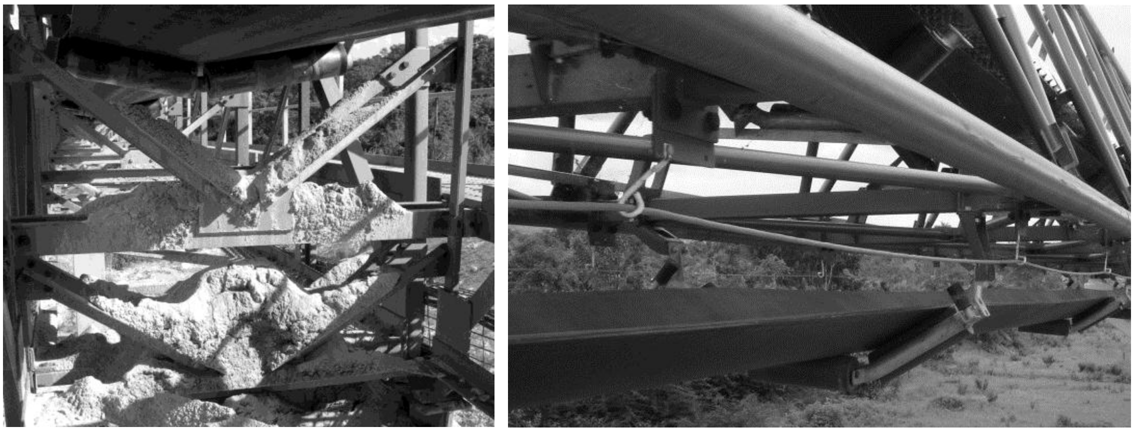
An analytical method for determination of local stresses and sag of belt helix turnovers has previously been published by the author [1]. In this paper a finite element model is developed to validate the original analytical equations and better understand turnover stresses. A comparison is made between the analytical method and the finite element analysis.

1. INTRODUCTION

This paper is a follow up to a previous paper entitled "Local Stresses in Belt Turnovers in Conveyor Belts" [1]. In that paper the author presented a practical approach to the analysis and design of helix turnovers. An analytical method was given which could be used to determine belt stresses and sag in a turnover. Subsequently, a finite element model has been developed to further understand the belt turnover behaviour including stresses and sag. The finite element model is used to validate the original analytical equations and to evaluate non-standard shapes for belt turnovers.

Belt turnovers are often used on overland conveyors to rotate the belt 180 degrees on the return strand. This keeps the clean side of the belt in contact with the return idlers. Turnovers prevent material carry back which in turn results in cleaner idlers and reduced maintenance.

Figure 1 shows a clear example of how turnovers can significantly improve the cleanliness of the return side of the belt. The images show two conveyors at the same limestone mine. The first conveyor feeds the second. Both conveyors are elevated and therefore reducing conveyor maintenance is desirable. The first conveyor does not have turnovers whereas the second conveyor does. The amount of material carry back on the first conveyor is significant along the entire length of the conveyor (which is 800 m). The second conveyor does not have any material carry back and looks virtually brand new even after several years in operation.



Conveyor #1 – No turnover **Conveyor #2 – Turnovers installed**

Figure 1: Turnover makes a big difference in cleanliness

When a turnover is installed on a conveyor, it is essential that it be designed correctly. A proper turnover design takes into account belt stresses, sag and alignment.

Belt stresses in a turnover arise from the combination of:

1. nominal belt tension
2. twisting stresses
3. bending stresses due to belt sag

Although these stresses are described in detail in reference 1, a brief summary is given below.

Twisting the belt causes the outer edge of the belt to stretch and the middle portion to be compressed. Figure 2 shows the resulting stresses from twisting and nominal tension only. The outer edges are in tension whereas the center portion is in relative compression.

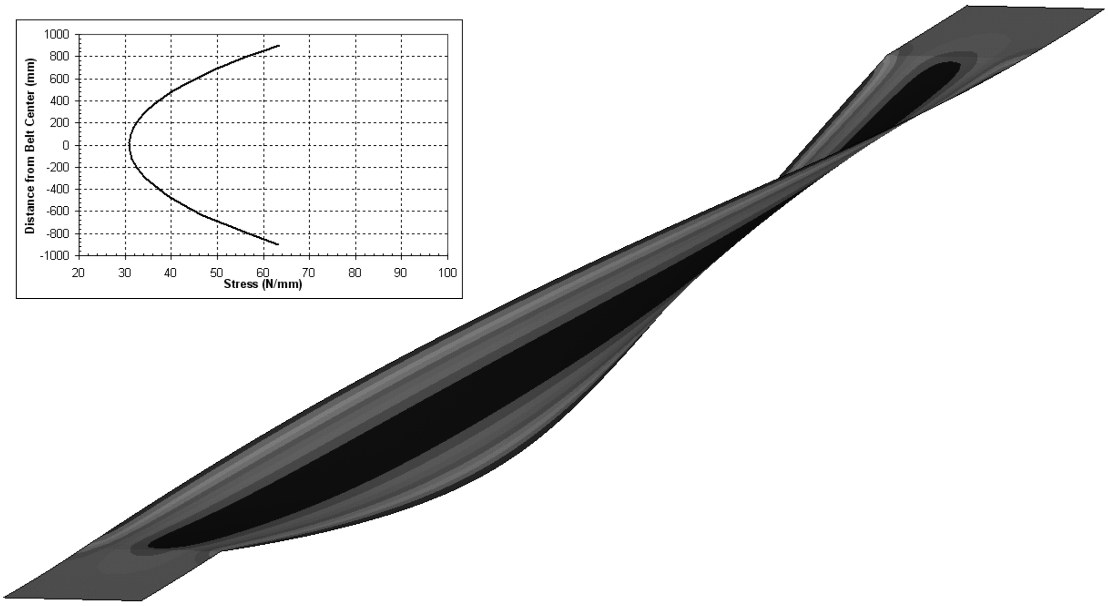


Figure 2: Stresses from twisting

Typical belt turnover lengths range from 11 to 50 times the belt width. The belt will sag in this span. Further, in a helix turnover the belt is vertical at the midpoint where the bending moment is greatest. The design of the turnover must account for these bending stresses. The bending moment increases the stress on the bottom edge and reduces the stress at the top edge.

The resulting stress in the turnover is therefore a complex function of nominal, twisting and bending stresses. The maximum stress occurs at the bottom edge. The minimum stress occurs near the center. The exact location of the minimum stress is dependent on the magnitude of bending stress compared to the twisting stress. Figure 3 shows the resulting stress pattern in a belt turnover.

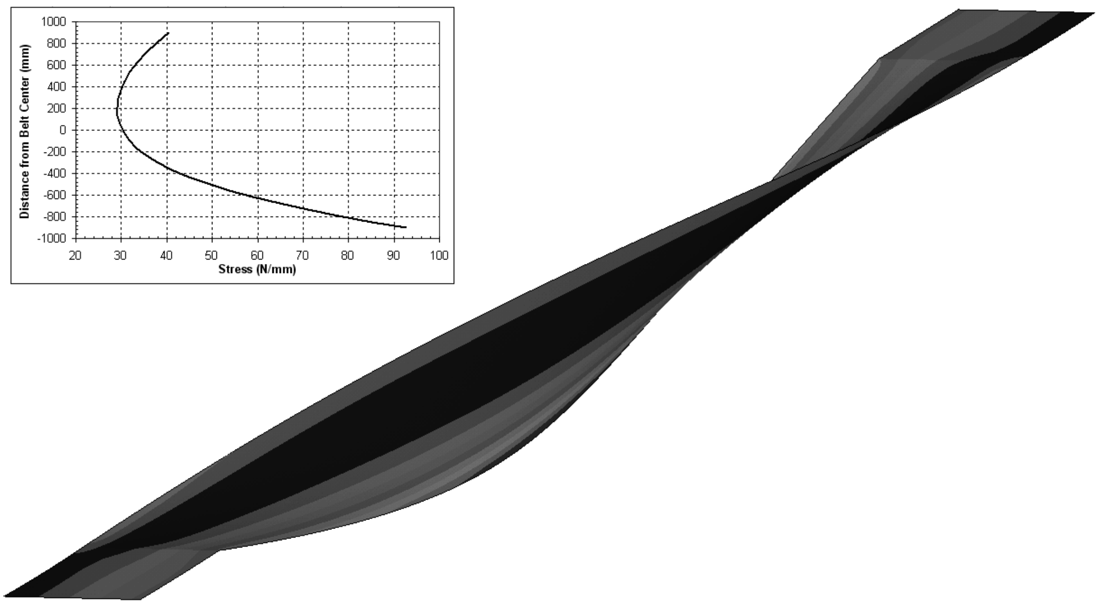


Figure 3: Resulting stresses from nominal tension, twisting, and sag

Figure 4 shows the resulting belt sag in the turnover.

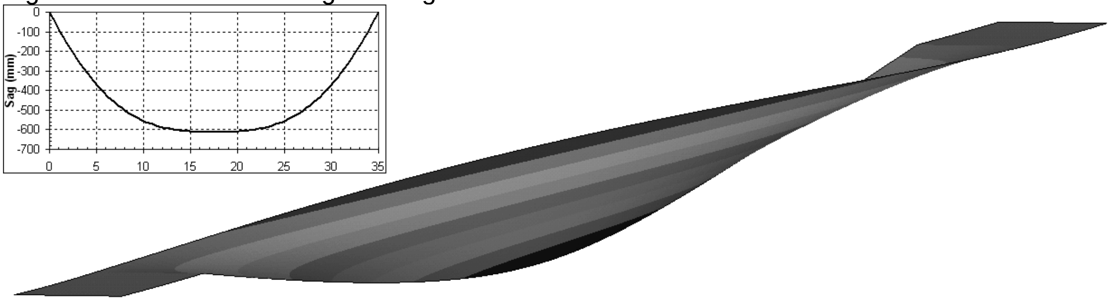


Figure 4: Vertical displacement (sag) in turnover

1.1 DESIGN CRITERIA

The stress design criterion of a turnover is the same as a convex curve. A convex curve causes higher stresses in the outer edges of the belt, and lower stresses in the middle. Turnovers also have this stress state and additionally must consider belt sag and alignment.

Acceptable turnover design includes the following [1]:

1. Edge stresses must not exceed the acceptable limit. The recommended maximum edge stress is 115% (or allowable edge stress = 1.15 * allowable nominal stress) of the rated belt tension [2].
2. Center stresses must be non-compressive 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.
3. Belt sag should be controlled to acceptable limits. The maximum recommended sag for steady state running 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 [2].

The design criteria set forth above is valid for both steel cord and fabric belts.

2 SUMMARY OF ANALYTICAL METHODS

This section will discuss the historical analytical methods available to determine turnover length.

2.1 DIN 22101 METHOD

Several belting manufacturers [3,4,5] simply list the required turnover length as a function of belt width. This is based on the DIN 22101 specification. The turnover length requirements are summarised for three types of turnover, which are shown in Figure 5:

Unguided Turnovers (Simple helix turnover with no support or guide rolls)
Unguided turnovers do not have any support rolls in the turnover. Unguided turnovers may be used for fabric belts with a maximum width of 1200 mm. The required length of an unguided turnover is 10 times the belt width.

Guided Turnover (Simple helix turnover with guide roll at center point)
Guided turnover have a vertical support rolls at the turnover center. Guided turnovers may be used on both fabric and steel cord belts with a maximum width of 1600 mm. The required length of a guided turnover is 12.5 times the belt width for fabric belts and is 22 times the belt width for steel cord belt widths.

Supported Turnover (Mordstein turnover):
Supported or Mordstein turnovers may be used with belts up to 2400 mm. The required length of a guided turnover is 10 times the belt width for fabric belts and is 15 times the belt width for steel cord belt widths.

The design guide specifies that these lengths are only valid for turnovers in low tension zones. Lengths for high tension zones are not given. The DIN method does not give guidance for helix turnovers with support rolls at the quarter points.

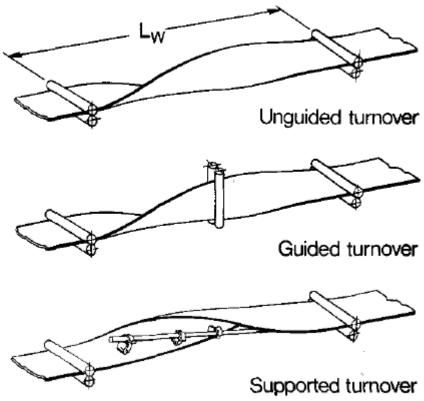


Figure 5: DIN Turnovers [4,5]

2.2 GOODYEAR METHOD

The Goodyear Handbook of Conveyor and Elevator Belting [2], Section 12-4 list requirements for belt turnovers. The Goodyear method takes into account both belt tensions and modulus for determination of turnover length. The method first calculates permissible ΔT_t, which is the difference between edge and center tension at the turnover. To calculate ΔT_t:

1. Calculate T_t in kN/m/ply (belt tension at turnover location)
2. Determine T_r in kN/m/ply (belt rating)
3. Find E in kN/m/ply (belt elastic modulus)
4. Permissible ΔT_t is:
   a. $\Delta T_t = 1.8T_t$ when $T_t \leq 0.5T_r$
   b. $\Delta T_t = 1.8(T_r - T_t)$ when $T_t > 0.5T_r$
5. Calculate ΔT_t /E

Once ΔT_t is calculated, the turnover length is looked up on a graph. The chart can be estimated by the following equation:

$$L_{TO} = BW \cdot (\Delta T_t / E)^{-0.517}$$, where 10·BW ≥ L_{TO} ≥ 45·BW (chart range)

where L_{TO} is the required length. For steel cord belts, the Goodyear method results in a turnover length approximately 50% to 60% longer than the DIN method. The Goodyear fabric belt length is approximately the same as the DIN method.

2.3 ANALYTICAL METHOD

The analytical method [1] solves both the twisting and bending stresses in the turnover.

At the belt center line (r=0), the equation for twisting stress is:

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

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

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

This equation is derived by assuming a helix displacement shape and then solves for the resulting strains and stresses.

Bending stresses are calculated by assuming the belt is a beam and then solving the beam's differential equations. The moment of inertia is assumed to be a function of location within the turnover. The moments of inertia about the z and y axis are:

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

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

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 bending moment in the beam is due to the belt mass and tension (T·y, which is often called stress stiffening). Figure 6 shows the beam with the gravity and belt tension forces.

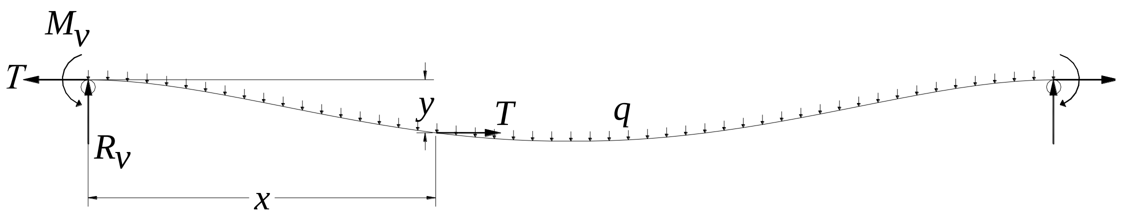


Figure 6: Turnover beam

The differential equation for this beam is:

$$y'' = \frac{1}{E \cdot I_{ZZ}} \left[M_v - \frac{qLx}{2} + \frac{qx^2}{2} + Ty \right]$$

The beam equation must be solved numerically as the above equation is non-linear due to the stress stiffening effect and varying moment of inertia. Once the differential equation is solved for displacements, the bending moment and beam stresses are:

$$M_Z = -y'' \cdot E \cdot I_{ZZ}$$

If quarter point rolls are installed in the turnover, then the vertical and horizontal force at the quarter point must be included in the solution. This is detailed in reference [1].

The above analytical method is included in AC-Tek's Sidewinder software for the design of conveyor belts. In the software, both simple helix and helix turnover with support rolls at the ¼ and ¾ positions can be solved per the analytical method. This software has been used for the analytical calculation results presented in this paper.

2.4 OPTIMAL TENSION AND LENGTH

As explained above, the stresses in a turnover are controlled by both twisting and bending. To reduce twisting stresses one must increase the length of the turnover. However, to reduce bending stresses the turnover length must be reduced. Due to these two competing factors, there is an optimal turnover length where twisting and bending stresses are minimised. In fact, there are actually two optimal lengths: one for the minimum stress (in which the center stress is maximised), and one for the maximum stress (in which the edge stress is minimised). These two optimal lengths usually do not coincide.

The selected turnover length should be the shortest length possible where minimum stress, maximum stress, and belt sag are at acceptable levels.

Figure 7 shows the optimization procedure using the Sidewinder software. To illustrate the optimal length selection, a 1200 mm ST-3500 N/mm belt will be used. Minimum and maximum belt stresses are plotted as a function of the turnover length. Each line shows the stress for a specific belt tension which may occur during steady state running, or starting and stopping.

This figure shows the minimum stress as a function of turnover length and belt tension. This figure illustrates that there is an “optimal” length for the turnover for minimum stresses. Figure 8 shows the safety factor of the edge stress for the same belt. Figure 9 shows the belt sag.

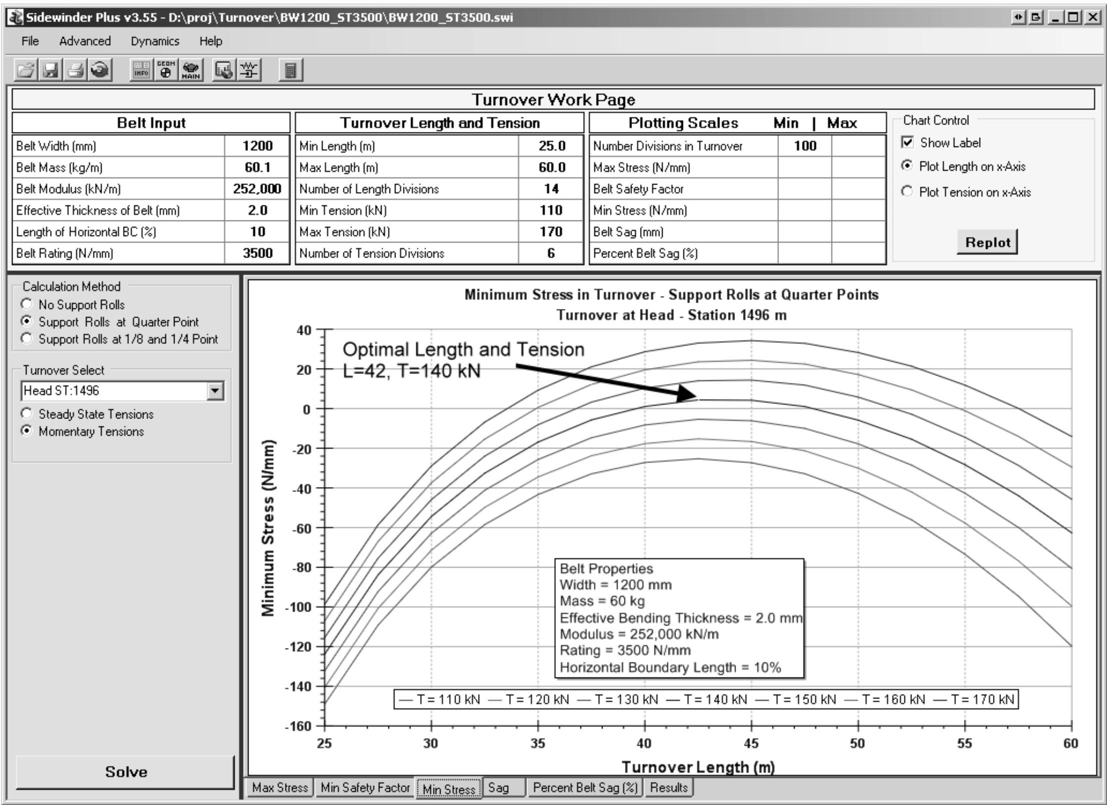


Figure 7: Optimizing turnover length for minimum stress in Sidewinder

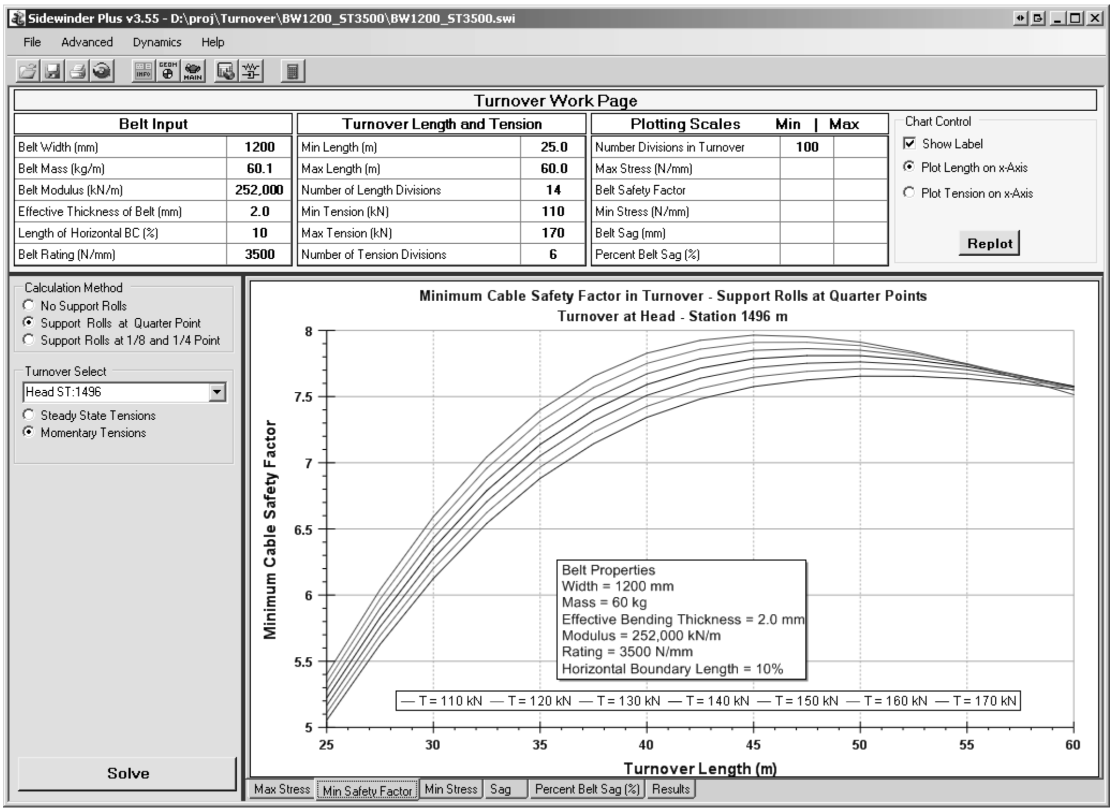


Figure 8: Safety factor of edge stress for 1200 mm ST-3500 N/mm belt

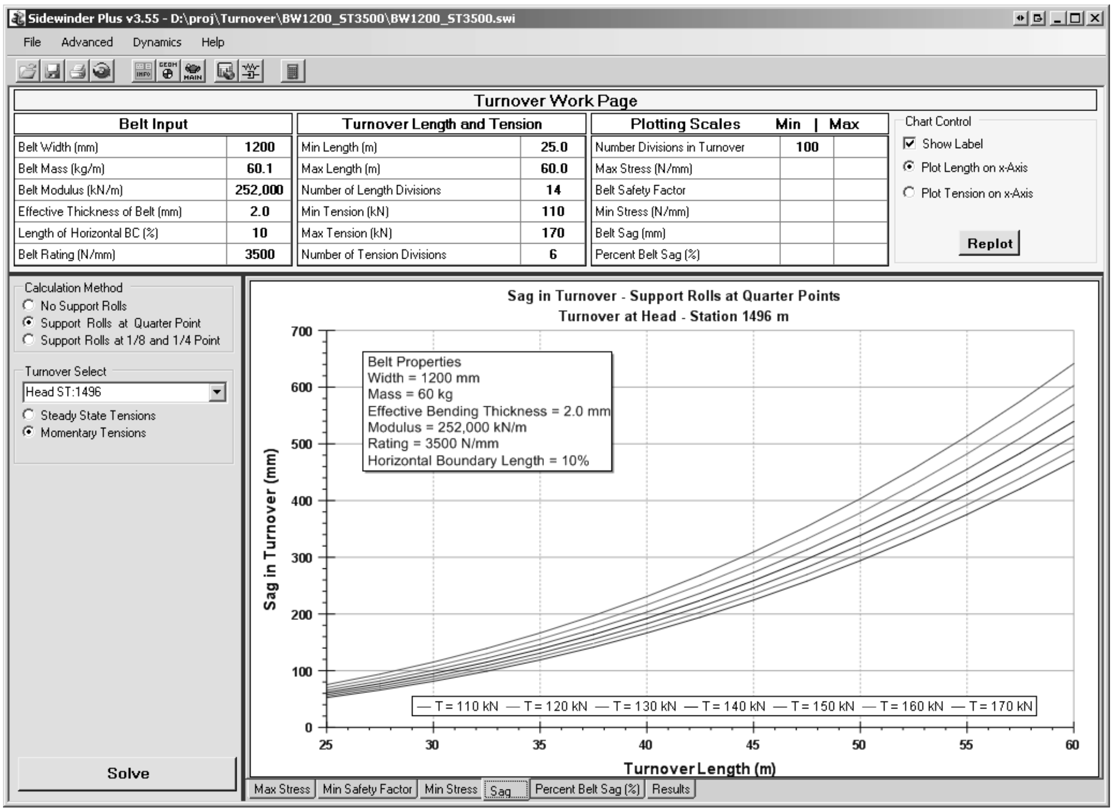


Figure 9: Sag for 1200 mm ST-3500 N/mm belt

Recall that increasing the turnover length reduces the twisting stresses (lower maximum stress and higher minimum stress). Therefore if bending stresses were not significant, then it would be best to increase the turnover length to reduce stresses. The above figures show that from 25 m to about 45 m, the twisting stresses dominate. As the length increases from 25 to 45 m, the stresses are becoming less because the twisting stresses are reducing. However, as the length increases above approximately 45 m, the stresses start to increase again. At lengths greater than 45 m, the bending stresses dominate.

As previously mentioned, the optimal turnover length for minimum stress may not be the same length for maximum stresses. For a belt tension of 140 kN, the optimal length for minimum stress is 42 m, however, for maximum stress the optimal length is 47 m.

Regardless of the optimal length, it is always desirable to make the turnover as short as possible. If the required safety factor is 6.7 / 1.15 = 5.8, then the safety factor plot shows that any length greater than 28 m is acceptable for maximum stresses. So for this turnover, it is the minimum stress that controls the turnover length. To prevent negative stress (or compression), the belt tension must be 140 kN and the length 42 m (Figure 7).

2.5 COMPARISON OF METHODS

Three methods (DIN, Goodyear, and Analytical) for determination of turnover lengths have been reviewed. This section will describe the results, and show how they compare to one another.

For comparison sake, three belt widths were considered. This included a narrow belt width (800 mm), a medium belt width (1200 mm), and wide belt width (1800 mm). Each belt is analyzed as both steel cord and fabric. Further, both low and medium-high belt ratings were analyzed. Tables 1 and 2 summarise the steel cord and fabric belts analysed. These same belt widths and ratings are also analysed with the finite element method.

Table 1: Summary of Steel Cord Belts

Belt Width (mm)	Rating (N/mm)	Covers (mm)	Belt Mass (kg/m)	Modulus (kN/m)	Turnover Length (m)	Belt Tension (kN)
800	ST-1200	7 x 5	19.6	86,400	27	41
1200	ST-1200	7 x 5	29.4	86,400	40	60
1800	ST-1200	7 x 5	44	86,400	52	74
1200	ST-3500	14 x 8	60.1	252,000	42	140
1800	ST-3500	14 x 8	90.2	252,000	58	235

Table 2: Summary of Fabric Belts

Belt Width (mm)	Rating (N/mm)	Covers (mm)	Belt Mass (kg/m)	Modulus (kN/m)	Turnover Length (m)	Belt Tension (kN)
800	EP-1200	7 x 5	16.9	10,060	10	41
1200	EP-1200	7 x 5	25.4	10,060	15	60
1800	EP-1200	7 x 5	38	10,060	23	75
1200	EP-3150	14 x 8	48.9	23,740	25	45
1800	EP-3150	14 x 8	73.3	23,740	34	90

Table 3 compares the required lengths for the three methods for steel cord belts. For the analytical method, the turnover is assumed to have quarter point support rolls. The required length of the DIN method is approximately 65% of the analytical method. The DIN method is not recommended for steel cord belts as it will typically result in high edge stresses and compressive stresses in the belt center. For narrow belts, the Goodyear method results in lengths similar to the analytical method. However for wide belts, the Goodyear method overestimates the required turnover length. Also, the Goodyear method required a longer length for the ST-1200 N/mm 1800 mm wide belt than it did for the ST-3500 N/mm 1800 mm wide belt. This does not seem to make practical sense from an engineering viewpoint.

Neither the DIN method nor the Goodyear methods have a procedure to determine minimum tension to prevent negative stress or buckling.

Table 3: Required lengths for steel cord belts					
Belt Width (mm)	Rating (N/mm)	Belt Tension (kN)	DIN	Goodyear	Analytical
800	ST-1200	41	17.6	27.5	27
1200	ST-1200	60	26.4	41.8	40
1800	ST-1200	74	---	69.4	52
1200	ST-3500	140	26.4	46.9	42
1800	ST-3500	235	---	66.4	58

Table 4 lists the required lengths for fabric belts. For narrow and low tension belts, the three methods give similar results. However, for the EP-3150 N/mm 1200 mm wide belt, the DIN method length is lower than the analytical or Goodyear methods. For fabric belts, the analytical and Goodyear methods are similar.

Table 4: Required lengths of fabric belts					
Belt Width (mm)	Rating (N/mm)	Belt Tension (kN)	DIN	Goodyear	Analytical
800	EP-1200	41	10	9.0	9
1200	EP-1200	60	15	13.7	15
1800	EP-1200	75	---	22.7	23
1200	EP-3150	45	15	24.9	23
1800	EP-3150	90	---	32.1	33

3 FINITE ELEMENT MODEL

The finite element model was completed in ANSYS. Non linear and orthotropic shell elements with stress stiffening capability were used to model the belt. A non linear model is required due to the large deformations and stress stiffening effects in the turnover. Stress stiffening is required because belt tensions increase the bending stiffness. The factor $+\frac{Ty}{E \cdot I_{ZZ}}$ is the stress stiffening portion in the differential equation.

The finite element model has the following aspects:

1. The model is parametric so that any geometry and tension can easily be modeled.

2. Different standard and non-standard geometries can be modeled, including:
    a. Helix turnover with no quarter point support rolls
    b. Helix turnover with quarter point support rolls
    c. Helix turnover with support rolls at the ⅛, ¼, ¾, and ⅞ locations
    d. Sandvik U-Turnover (belt is forced into sideways U shape to reduce required clearance)
    e. Mordstein turnover (this is the DIN standard “supported turnover” – See Figure 5)

3. Additional length can be added past the turnover length. Figure 10 shows the extra boundary length in the turnover. This length can be varied to see how belt stresses vary outside of the turnover.

Figure 10: Extra boundary length (L_{BC}) in turnover

4. Support rolls are placed at the end points and at other optional points (see 2, above). Contact elements are placed between the support rolls and belt. Contact elements prevent the belt from displacing past the support roll.

5. The belt is assumed to be homogeneous with orthotropic material properties. The moduli in both the longitudinal and transverse direction can be varied to accurately and realistically model the belt.

6. The bending stiffness of the belt can be set to the appropriate value.

4 FEA RESULTS AND COMPARISON TO ANALYTICAL

The purpose of the finite element model is two fold: first to compare and validate the analytical model, and second to enable analysis of nonstandard geometric shapes that would be very difficult to analyse with analytical models.

The following sections compare the results of the analytical model with the finite element model. Also shown are some interesting results of turnovers and non-standard geometries.

4.1 TWISTING STRESS

The first point of interest was to see how well the assumed displacement shape of the analytical method and resulting stresses of the helix twist matched the resulting finite element model. The finite element model showed that analytical model twisting stresses were quite accurate. Table 5 lists the twisting stresses for both analytical and FEA models. The difference between the two models is only a few percent.

Table 5: Twisting Stress

Belt		Turnover		Analytical Results		FEA Model Results		Error	
BW (mm)	Rating (N/mm)	Length (m)	Tension (kN)	Center (N/mm)	Edge (N/mm)	Center (N/mm)	Edge (N/mm)	Center (%)	Edge (%)
800	ST-1200	27	41	20.1	113.5	20	114	0.4	-0.4
800	EP-1200	10	41	24.9	104.0	25	103	-0.5	0.9
1200	ST-1200	40	60	18.0	113.9	18	114	0.2	-0.1
1200	EP-1200	15	60	23.6	102.7	24	102	-1.6	0.7
1200	ST-3500	42	140	32.1	285.7	31	285	3.6	0.2
1200	EP-3150	25	45	15.0	82.4	15	82	0.1	0.5
1800	ST-1200	52	74	-1.4	126.2	-1.4	125	2.2	1.0
1800	EP-1200	23	75	16.4	92.1	17	92	3.5	1.0
1800	ST-3500	58	235	30.8	330.0	31	329	-0.6	0.3
1800	EP-3150	34	90	22.5	104.4	22	104	2.2	0.4

Figure 11 shows the twisting stresses (bending stresses not included) for the 1800 mm ST-3500 N/mm belt turnover.

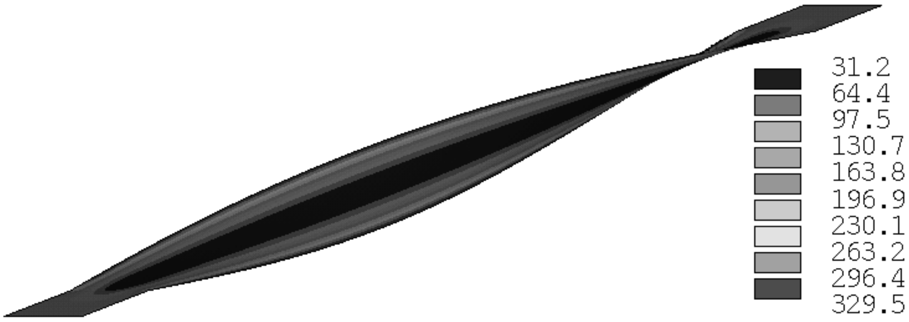


Figure 11: Twisting stress (1800 mm, ST-3500 N/mm)

4.2 HELIX WITH NO SUPPORT ROLLS

The simplest of turnover is a helix turnover with no support rolls. This type of turnover is typically used for fabric belts which can have a short length in which sag is not expected to be large.

The simple helix with no support rolls was modelled for each belt width and rating. Table 6 summarises the minimum and maximum stress results of the fabric belts for both analytical and finite element methods. For the fabric belts, the difference between the analytical and finite element models is less than 5%.

Table 6: Comparison of Simple Helix Turnover Stresses – Fabric Belts

Belt		Turnover		Analytical		Finite Element		% Error Maximum Stress
BW (mm)	Rating (N/mm)	Length (m)	Tension (kN)	Min Stress (N/mm)	Max Stress (N/mm)	Min Stress (N/mm)	Max Stress (N/mm)	
800	EP-1200	10	41	25	112	25	111	0.5
1200	EP-1200	15	60	23	114	24	114	0.2
1800	EP-1200	23	75	15	112	16	113	-0.9
1200	EP-3150	25	45	-1	148	3	149	-0.5
1800	EP-3150	34	90	4	182	6	186	-2.2

Table 7 summarises the minimum and maximum stresses for the steel cord belts. The maximum stress of the analytical method is typically less than the finite element method. For belt widths less than 1200 mm, the stress of the analytical method is within 6% of the finite element method. However for the 1800 mm belt, the analytical method can underestimate the maximum stress by up to 23%. The largest error occurs at lower tensions.

Table 7: Comparison of Simple Helix Turnover Stresses – Steel Cord Belts

Belt		Turnover		Analytical		Finite Element		% Error Maximum Stress
BW (mm)	Rating (N/mm)	Length (m)	Tension (kN)	Max Stress (N/mm)	Min Stress (N/mm)	Min Stress (N/mm)	Max Stress (N/mm)	
800	ST-1200	27	41	183	7	4	176	4.1
1200	ST-1200	40	60	219	-11	9	233	-6.2
1800	ST-1200	52	74	274	-44	-8	354	-22.7
1200	ST-3500	42	140	537	-30	-4	560	-4.0
1800	ST-3500	58	235	658	-59	-10	760	-13.4

Figure 12 shows the longitudinal stress in the 1200 mm ST-1200 N/mm belt turnover.

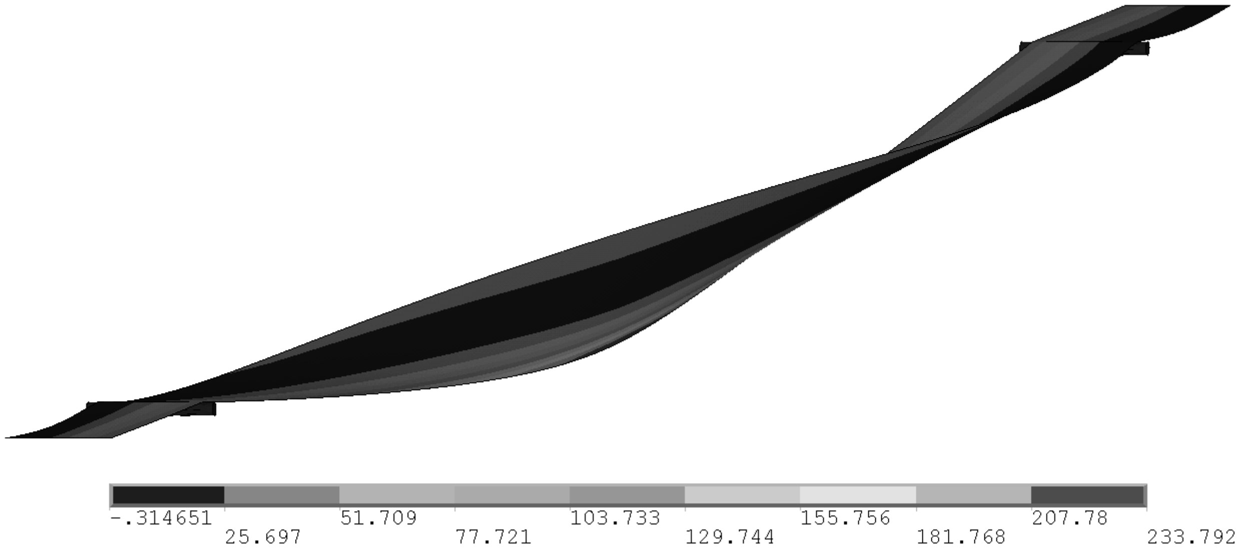


Figure 12: Stress (N/mm) for 1200 mm, ST-1200 N/mm belt

The analytical method underestimates belt sag up to 20% for the fabric belts. Table 8 lists the belt sag for fabric belts. The underestimation of the analytical method is even higher for steel cord belts where the sag is 45% less than FEA (Table 9).

The main reason the analytical method underestimates belt sag is due to the assumption that the belt is a beam for which the moment of inertia can be estimated by the twisted geometry. In reality, the belt is not a beam and this assumption results in excessive stiffness and therefore underestimates belt sag. This is also the reason why the stresses are somewhat underestimated. Of course this error will be greatest for belts that are wide with low belt ratings.

The belt sag can be better estimated by the following equation for simple helix turnovers:

$$Sag = (1-k) \cdot y_{Analytical} + k \cdot y_{Catenary}$$

Where:

$y_{Analtical}$ = turnover sag from analytical method

$y_{Catenary}$ = catenary sag, $y_{Catenary} = \dfrac{w_b \cdot L^2}{8 \cdot T}$

k = correction factor = 0.50 for simple helix turnover

Tables 8 and 9 also show the corrected sag and corresponding error. This equation will normally result in a conservative value (i.e. overestimate the sag) for sag except in cases of long turnovers with wide belts and low tensions. However, in such cases the turnover should be built with support rolls at the ¼ and ¾ points to reduce sag.

Table 8: Comparison of Simple Helix Turnover Sag – Fabric Belts								
BW (mm)	Rating (N/mm)	Length (m)	Tension (kN)	Sag (mm)	Corrected Sag (mm)	FEA Sag (mm)	% Error Sag	% Error Corrected Sag
800	EP-1200	10	41	31	41	37	-15	12
1200	EP-1200	15	60	70	93	83	-16	13
1800	EP-1200	23	75	189	245	225	-16	9
1200	EP-3150	25	45	481	657	599	-20	10
1800	EP-3150	34	90	692	923	845	-18	9

Table 9: Comparison of Simple Helix Turnover Sag – Steel Cord Belts								
BW (mm)	Rating (N/mm)	Length (m)	Tension (kN)	Sag (mm)	Corrected Sag (mm)	FEA Sag (mm)	% Error Sag	% Error Corrected Sag
800	ST-1200	27	41	245	336	295	-17	14
1200	ST-1200	40	60	541	751	729	-26	3
1800	ST-1200	52	74	896	1434	1624	-45	-12
1200	ST-3500	42	140	497	713	673	-26	6
1800	ST-3500	58	235	827	1205	1225	-32	-2

Figure 13 shows displacement in the 1200 mm ST-1200 N/mm belt turnover.

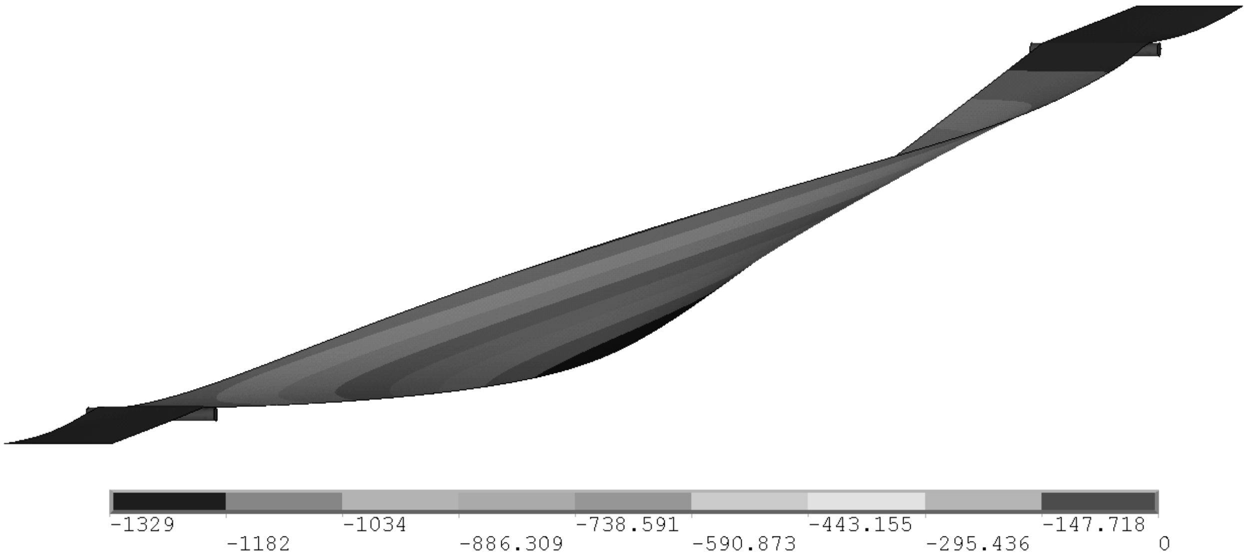


Figure 13: Belt displacement for 1200 mm, ST-1200 N/mm belt

4.3 HELIX WITH SUPPORT ROLLS

Helix turnovers with support rolls at the ¼ and ¾ point locations are normally used on turnover with long spans. The support rolls reduce both belt sag and stresses. Figure 14 shows a turnover with support rolls at the quarter points.

Figure 14: Turnover with quarter point support rolls

The helix turnover with support rolls was modelled for each belt width and rating. Table 10 summarises the minimum and maximum stress results of the fabric belts for both analytical and finite element methods. For the fabric belts, the difference between the analytical and finite element models is less than 5%.

Table 10: Comparison of Simple Helix Turnover Stresses – Fabric Belts

Belt		Turnover		Analytical		Finite Element		% Error Maximum Stress
BW (mm)	Rating (N/mm)	Length (m)	Tension (kN)	Min Stress (N/mm)	Max Stress (N/mm)	Min Stress (N/mm)	Max Stress (N/mm)	
800	EP-1200	10	41	25	111	24	110	0.7
1200	EP-1200	15	60	23	113	23	111	1.8
1800	EP-1200	23	75	15	109	15	109	0.3
1200	EP-3150	25	45	3	139	9	136	2.3
1800	EP-3150	34	90	8	173	17	169	2.3

Table 11 summarises the minimum and maximum stress results of the steel cord belts. The maximum stress of the analytical method is typically less than the finite element method. The maximum analytical method error is 18%. For narrow belts the error of the analytical method is not large. However, for wide belts and longer turnover lengths, the error increases.

Table 11: Comparison of Simple Helix Turnover Stresses – Steel Cord Belts

Belt		Turnover		Analytical		Finite Element		% Error Maximum Stress
BW (mm)	Rating (N/mm)	Length (m)	Tension (kN)	Min Stress (N/mm)	Max Stress (N/mm)	Min Stress (N/mm)	Max Stress (N/mm)	
800	ST-1200	27	41	14	162	16	156	3.7
1200	ST-1200	40	60	5	186	11	193	-3.8
1800	ST-1200	52	74	-19	220	-5	266	-17.4
1200	ST-3500	42	140	4	455	14	468	-2.8
1800	ST-3500	58	235	-9	549	15	602	-8.9

Figure 15 shows the approximate error in the analytical method. This chart assumes the turnover length has been optimised for the minimum tension (Section 2.3). Also, the chart assumes that the belt tension results in a minimum stress that is approximately -20 to +10 N/mm. As the belt tension increases from this point, the error will be less. Also if the turnover length is less than the optimal length for minimum tension, the error will less. However, if the turnover length is longer than the optimal length, the error may be higher.

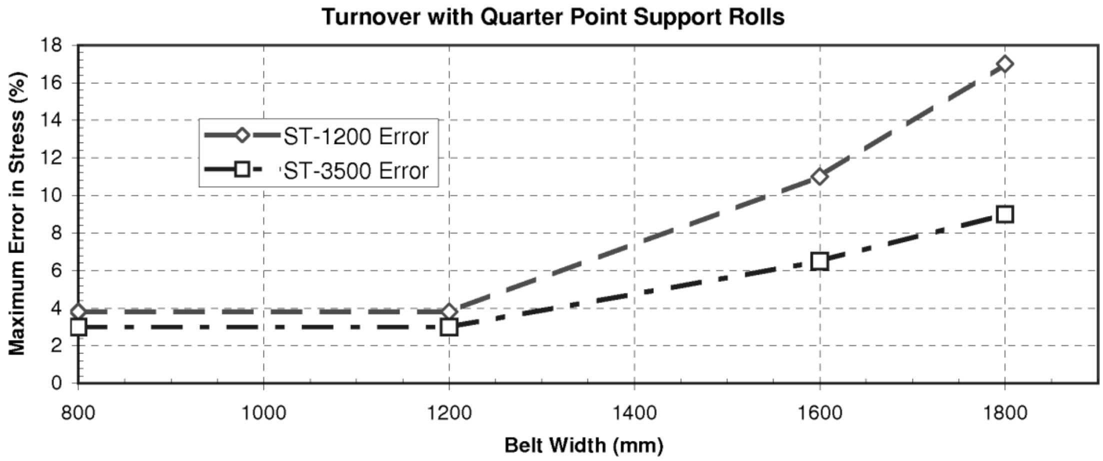


Figure 15: Error estimate for analytical method

Figure 16 shows the longitudinal stress in the 1200 mm ST-1200 N/mm belt turnover.

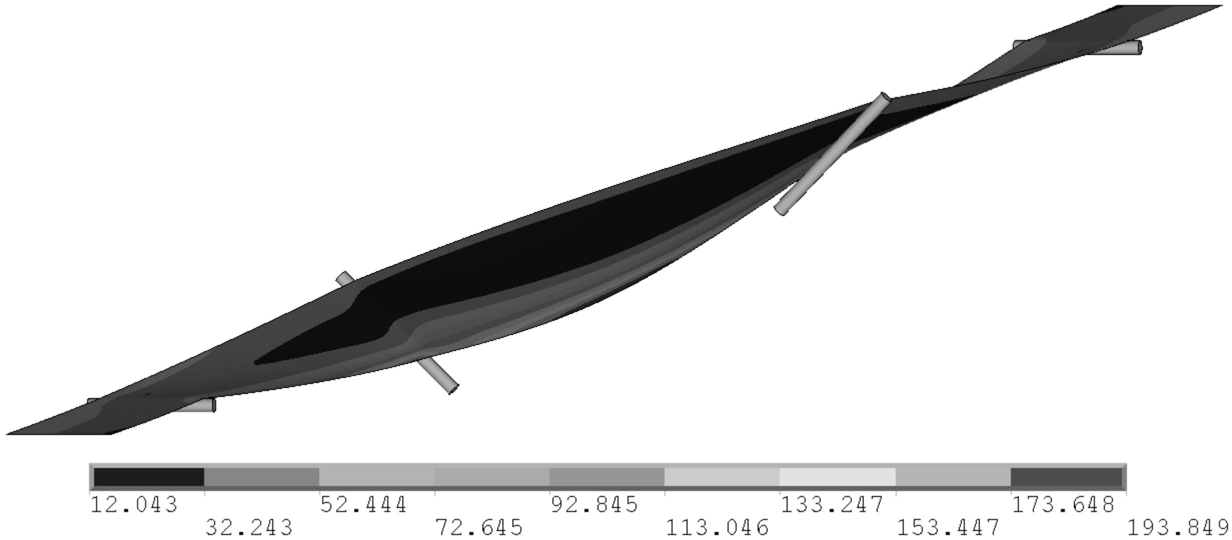


Figure 16: Stress (N/mm) for 1200 mm, ST-1200 N/mm belt

The analytical method underestimates belt sag up to 70%. Tables 12 and 13 list the belt sag for fabric and steel cord belts. The belt sag can be better estimated by the following equation for helix turnovers with quarter point support rolls:

$$Sag = (1-k)\cdot y_{Analytical} + k \cdot y_{Catenary}$$

Where k = 0.15 for helix turnover with support rolls at quarter points.

Tables 12 and 13 also show the corrected sag and corresponding error. The above equation will reduce the sag error to less than 10%.

Table 12: Comparison of Simple Helix Turnover Sag – Fabric Belts

BW (mm)	Rating (N/mm)	Length (m)	Tension (kN)	Sag (mm)	Corrected Sag (mm)	FEA Sag (mm)	% Error Sag	% Error Corrected Sag
800	EP-1200	10	41	4	11	11	-62	5
1200	EP-1200	15	60	10	26	27	-63	-4
1800	EP-1200	23	75	25	71	76	-67	-6
1200	EP-3150	25	45	65	180	190	-66	-5
1800	EP-3150	34	90	97	255	278	-65	-8

Table 13: Comparison of Simple Helix Turnover Sag – Steel Cord Belts								
BW (mm)	Rating (N/mm)	Length (m)	Tension (kN)	Sag (mm)	Corrected Sag (mm)	FEA Sag (mm)	% Error Sag	% Error Corrected Sag
800	ST-1200	27	41	48	105	97	-51	8
1200	ST-1200	40	60	104	232	223	-53	4
1800	ST-1200	52	74	149	423	440	-66	-4
1200	ST-3500	42	140	92	217	204	-55	7
1800	ST-3500	58	235	151	366	355	-58	3

Figure 17 shows displacement in the 1200 mm ST-1200 N/mm belt turnover.

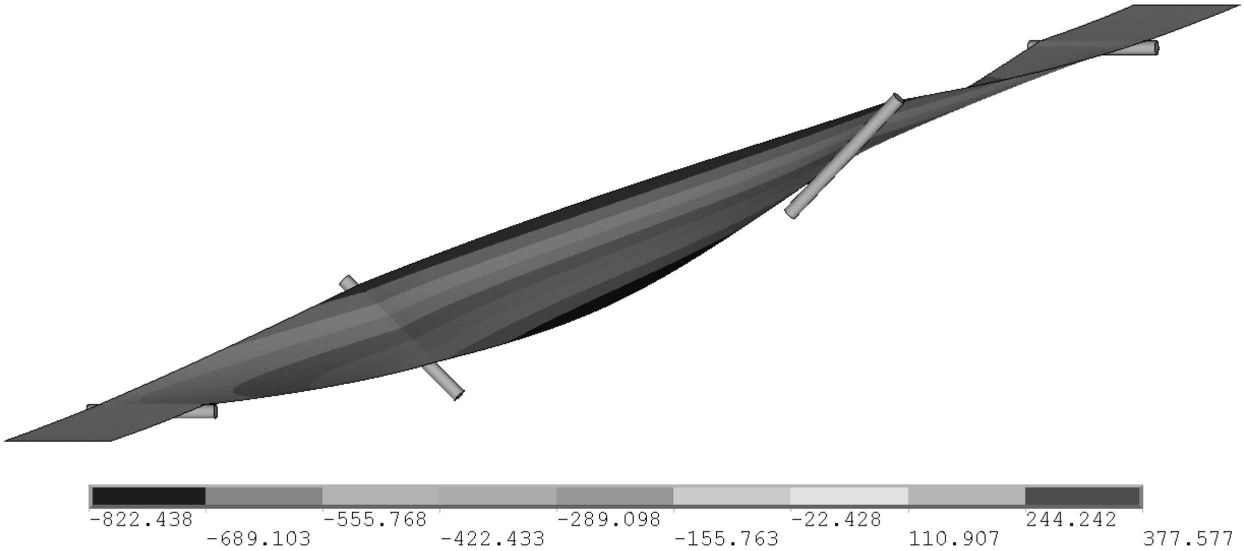


Figure 17: Belt displacement for 1200 mm, ST-1200 N/mm belt

4.4 OPTIMAL LENGTH

As discussed in section 2.3 there is an optimal length for both the minimum and maximum stresses. Figure 18 shows the minimum stress plotted against turnover length for an 1800 mm ST-3500 N/mm belt with a tension of 235 kN. The chart shows both the analytical and FEA results. The finite element minimum stress is approximately 20 N/mm higher than the analytical method. However, the curve shape is the same so the analytical method has correlation to FE and can be used to determine the length of the turnover. The optimal length for this turnover is 55 to 60 m.

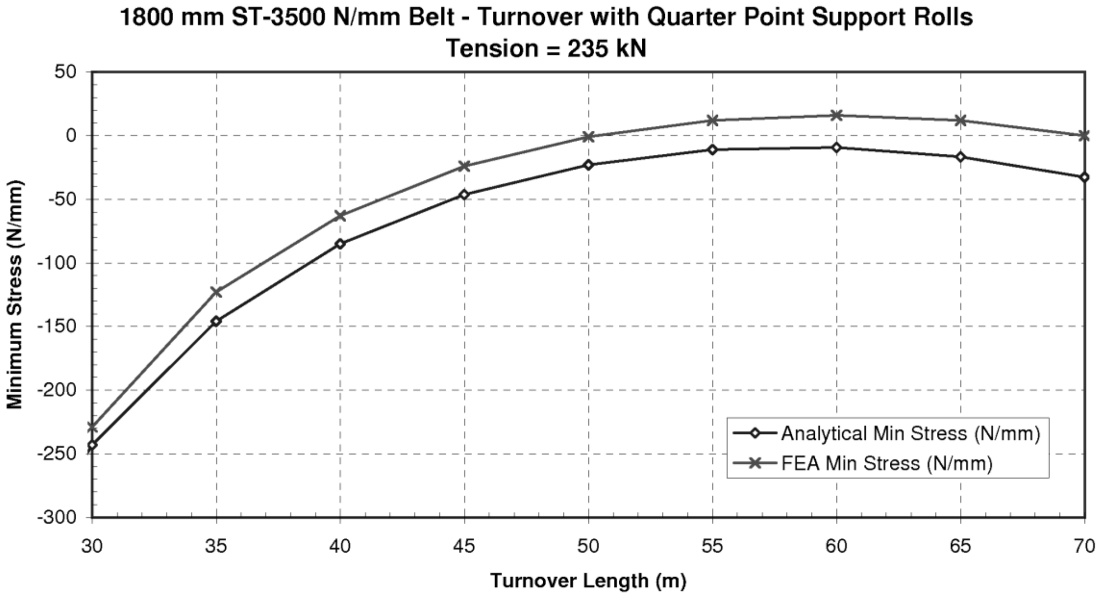


Figure 18: Minimum stress for 1800 mm ST-3500 N/mm belt

The optimal length for maximum stress is normally not the same as the optimal length for minimum stress. Figure 19 shows the maximum stress as a function of turnover length. Again, the maximum stress of the finite element method is slightly higher than the analytical method. However, the shape of the curve is the same. The optimal length based on minimising belt stresses is 60 m.

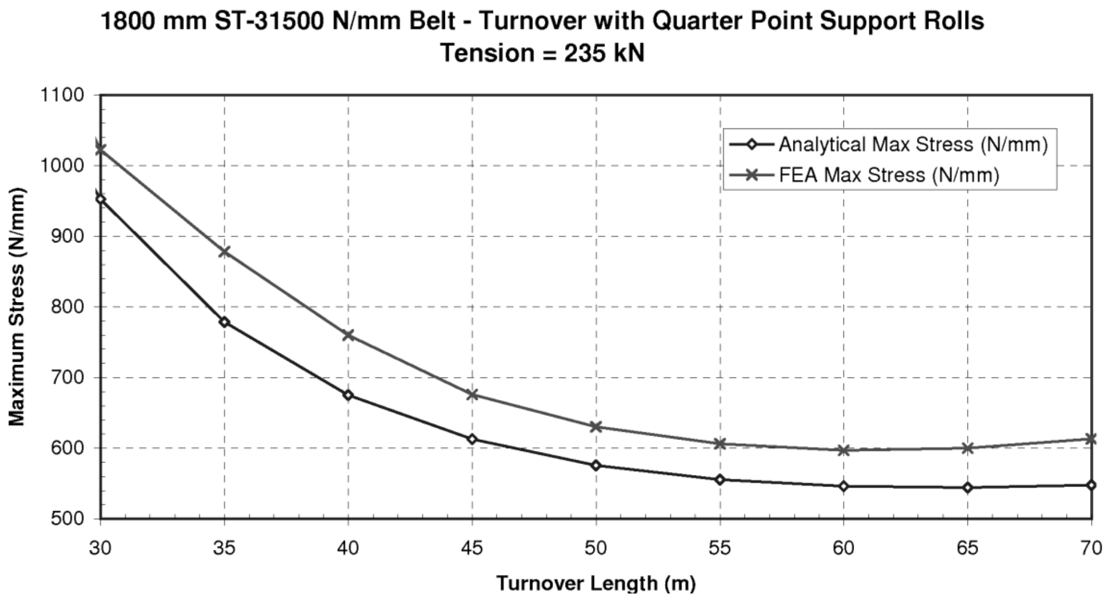


Figure 19: Maximum stress for 1800 mm ST-3500 N/mm belt

5 TURNOVER GEOMETRY RECOMMENDATION

Designers and draftsman often do not fully understand the requirements for turnovers and often neglect to include sufficient clearance in the turnover structure. Figure 20 shows an existing turnover where the quarter point support rolls do not have sufficient length. The belt edge is not on the support roll and is wrapped around the rollers edge. Additionally, the structure had to be cut at the top edge.

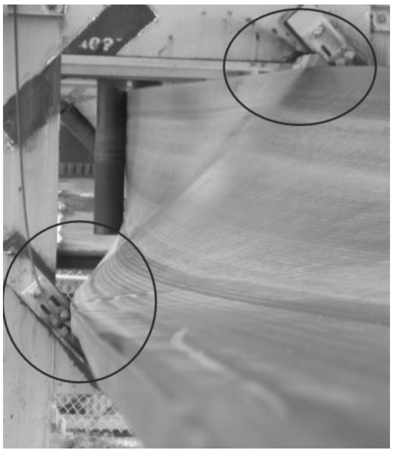


Figure 20: Quarter point support roll with insufficient length

It is recommended that the middle support rolls and quarter point support rolls have a minimum face width 1.5 and 1.35 times the belt width respectively. This face width allows sufficient clearance for the belt sag during low tension dynamic conditions (starting or stopping) and belt maintenance.

Also, it is recommended that the quarter point support roll be adjustable by ±5 degrees. This adjustability will help in obtaining good alignment and belt tracking.

The support rolls should be placed such that top edge of the belt is near the top edge of the support rolls when there is zero belt sag in the turnover. This is required so the belt is supported as the belt sags.

Small safety rolls should be placed at the bottom of the support rolls. These safety rolls will ensure that the belt is not damaged if the belt sag becomes excessive. This can occur during maintenance when the take-up is released.

It is recommended that the turnover pulleys have an additional face width of 100 mm above the standard pulley face width. This is to allow some lateral displacement at the turnover.

Figure 21 shows these recommendations.

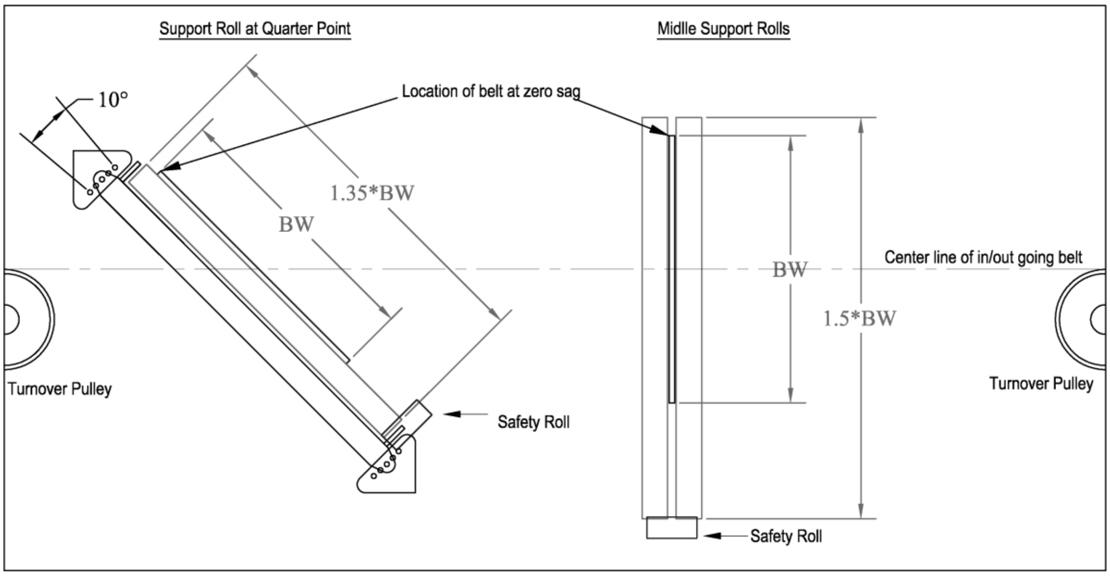


Figure 21: Turnover support roll recommendation

6 CONCLUSION

The author previously published an analytical method for the stress analysis of belt turnovers. Subsequently, a finite element method has been completed to further study the stresses and belt sag in turnovers. This paper reviewed the results of the finite element analysis and made comparisons to the analytical method contained in the Sidewinder conveyor design software. Good correlation was found for fabric belts. The correlation is also good for narrow and medium width steel cord belts. However, there is some divergence for wide belts, particularly at lower belt tensions which may typically occur at the tail. A modified belt sag equation was presented for the analytical method to more accurately match the finite element model.

The paper also reviews published methods for determination of turnover lengths, including the DIN method and the method published in the Goodyear Red book. These published methods are generally acceptable for fabric belts. However, it is recommended to use the analytical method to determine the turnover length for steel cord belts. The finite element method results in the best optimised turnover length for all belt types and classes. It is recommended for wide belt widths and high strength steel cord belts. It is also required for non-helix turnovers, and nonstandard turnover geometries.

Finally, recommendations are given for good design practices of belt turnovers.

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