WRINKLING OF WIDE WEBS

By

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ABSTRACT

Wrinkling remains one of the top causes of converting process waste. Though wrinkles created by roller misalignment and web bending (shear wrinkles) are well-understood, nearly all the published research and modeling has been based on narrow web experiments (less than 15-in wide). Other causes of wrinkling, such as wrinkling due to roller deflection or uneven nipping, have had little or no published results. Similar to wrinkling, most published research on spreading and anti-wrinkle rollers has been based on narrow web experiments.

This paper will review the results from wide web wrinkling due to roller misalignment for various materials and compare the experimental wrinkling results to isolated-span shear wrinkle models. Also, we will share results from over ten different roller designs (including various concave roller profiles) demonstrating their relative effectiveness in preventing shear wrinkles and their ability to spread a web slit down its center. Lastly, during our shear wrinkles trials we unintentionally created tracking or gathering type wrinkles from a deflecting rollers at the top of a long, vertical span. This led to an experiment investigation to eliminate these wrinkles and an upgraded model to predict these wrinkles caused by roller deflection in the negative bow direction.

NOMENCLATURE

- B Length between roller bearings, m
- C Web width, m
- D Roller outer diameter, m
- E Roller material Young's modulus, Pa
- E_w Web Young's modulus, Pa
- F Roller face length, m
- h Roller wall thickness, m
- I Roller bending stiffness, m^4
- L L1 span length, m

R Roller outer radius, m
t Web thickness, m
T Web tension, N
w_c Total distributed load (tension and gravity), corrected for orientation, N/m
w_g Distributed load due to roller weight, N/m
w_t Distributed load due to web tension, N/m
dir Rotation direction: 0 – clockwise, 1 – counter clockwise
α_{in} Entering span angle, degrees
α_{out} Exiting span angle, degrees
β Orientation of tension and gravity load vectors, degrees
γ Orientation of vector sum of tension/gravity load vectors, degrees
$\varepsilon_{d,c}$ Roller deflection web strain due to roller weight and web tension
θ Roller wrap angle, degrees
μ Web-to-roller coefficient of friction, --
ρ Roller shell density, kg/m^3

INTRODUCTION

The knowledge and modeling of wrinkling webs, particularly shear wrinkles created from misaligned rollers, has grown tremendously over the last 25 years, largely due the excellent work on Dr. Keith Good at Oklahoma State University's Web Handling Research Center (OSU WHRC). Much of the initial wrinkling research was performed at 3M Company's web handling development lab where Dr. Good worked closely with Douglas Kedl. Subsequently, many more experiments have been completed and presented through the sponsored research at the OSU WHRC, working to verify advances in wrinkle models. However, most all of this work has been completed using limited range of materials and widths, mostly polyester (PET) and newsprint (with some more recent work using a lower modulus non-woven) and almost all have been narrow webs (less than 0.3m or 12-in wide).

The goal of this work is to: 1) compare the isolated-span shear wrinkle models to wider web, and 2) repeat the anti-wrinkle and spreading tests, as defined by Swanson [4], on wider webs. This work was completed at the Media Conveyance Facility at Optimation Technology Inc. in Rochester, NY as part of our Wrinkling-Spreading Workshops in 2009 through 2011.

During our work with wide webs, in two cases, running wide (>1m), 12 micron aluminum and polyesters, we found stationary, near centerline wrinkles forming on large wrap angle, small diameter rollers at the end of long, vertical spans. This led to an experiment investigation to eliminate these wrinkles and an upgraded model to predict these wrinkles caused by roller deflection in the negative bow direction.

WRINKLING OF WIDE WEBS

A common way to characterize shear wrinkling sensitivity is the shear wrinkle plot. The shear wrinkle plot expresses the transition from wrinkle-free to wrinkled web handling as a function of web tension (or strain) and parallelism of two rollers in an isolated span. Roller parallelism (also called misalignment or tram error) is expressed in mradians (or mm/m or mils/in). Converting equipment is commonly specified to hold alignment to better than 0.2 mradians (2 mils/ft is 0.17 mradians).

The isolated-span shear wrinkle model assumes all stresses and lateral shifting within a span are independent of upstream pre-entering or downstream spans and that the boundary conditions include the web exiting and entering parallel to the rotation of both the upstream and downstream rollers.

The shear wrinkle plots are contour plots showing two transitions from wrinkle-free and wrinkling conditions.

The first transition, commonly called Regime I, predicts the buckling of the span, what is commonly called troughing. In Regime I the critical angle to form troughing increases with tension or strain due to tension stiffening. The two Regime I theory curves in the following figures are created from isolated-span shear wrinkle models [3] to predict when trough buckles would form in the upstream span (the dashed lines) and applying the criteria that the web wrinkles at two-times the misalignment of troughing (the solid lines) [6].

The second transition, Regime II, is a traction limited effect where decreasing tension reduces web-to-roller traction to a point where the roller-applied forces will no longer be able to exceed the web critical buckling stress. The vertical line is the model-predicted Regime II transition, expressing the tension or strain condition where the web-roller traction is considered insufficient to hold the web in a buckled shape.

Figure 1 shows experimental wrinkle results vs. the isolated-span shear wrinkle theory for 1.23m by 12 micron (48.5-in by 0.48-mil) polyester. The experimental data points denote the threshold roller misalignment required to create a shear wrinkle. The web runs wrinkle-free at any misalignment below this value and continues to wrinkle at any misalignment above this value. This highly shear wrinkle sensitive thin, wide web followed quite closely to theory, wrinkling at a condition slightly higher than expected. (Unless otherwise notes, all shear wrinkling spans were 1.2m (50-in) and speeds were low to avoid air lubrication, <15m/min or <50 fpm).

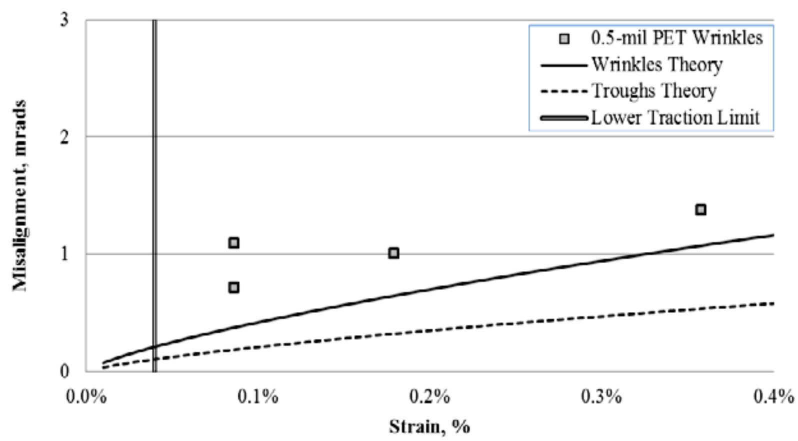

Figure 1 – Shear Wrinkling 12 micron by 1.23m Polyester - Results vs. Theory

Figure 2 plots our shear wrinkle results of wide, thin PET (1.23m by 12 micron) compared to narrow shear wrinkling results [3, 4].

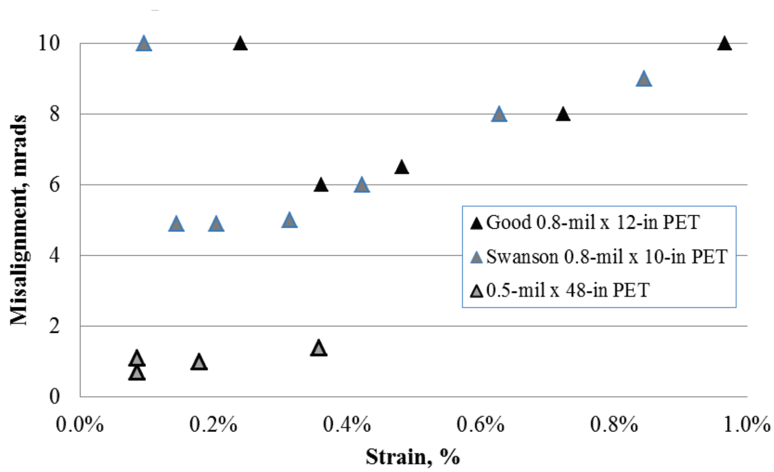

Figure 2 – Comparison of Narrow and Wide Polyester Shear Wrinkles

Our next experiments were with a PET-PE laminate with a thickness of 35 micron (1.4-mil) and width of 1.23m (48.5-in). From a lateral bending view, where the force to bend the web is proportional to the thickness times the width cubed, this web is 2x thicker and 4x wider than the webs studied by Good and Swanson [3, 4].

The wrinkling data of the 35 micron (1.4-mil thick), 1.23m (48.5-in) wide PET-PE laminate has a good (slightly high again) fit to theory under higher tensions, but a poor fit at lower tensions and strain, revealing a process window much greater than isolated-span shear wrinkle theory would predict.

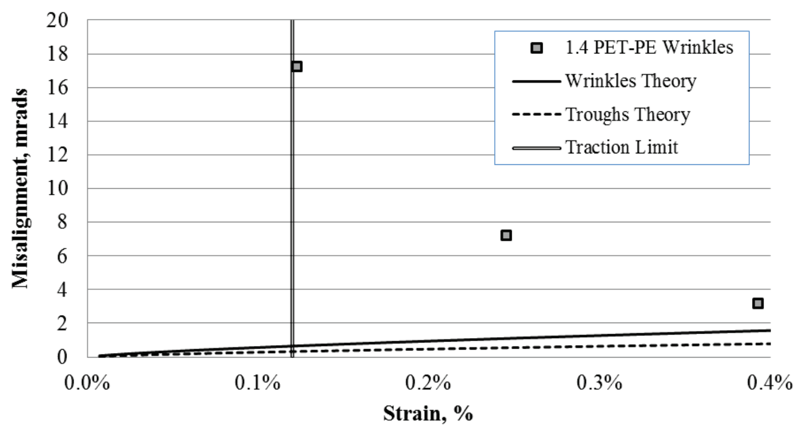

Figure 3 – Shear Wrinkles in 35 micron Polyethylene-Polyester Web

In shear wrinkling copper foil (0.4-mils thick, 15.7-in wide) we have a material that is not far off the thickness and width of traditional wrinkling research webs, but has an elastic modulus that is over 15x higher. At high tensions and strains, the experimental wrinkling results approach theory (again slightly less sensitive than expected), but also again show a wrinkle-free process window as strain, tension, and friction drop.

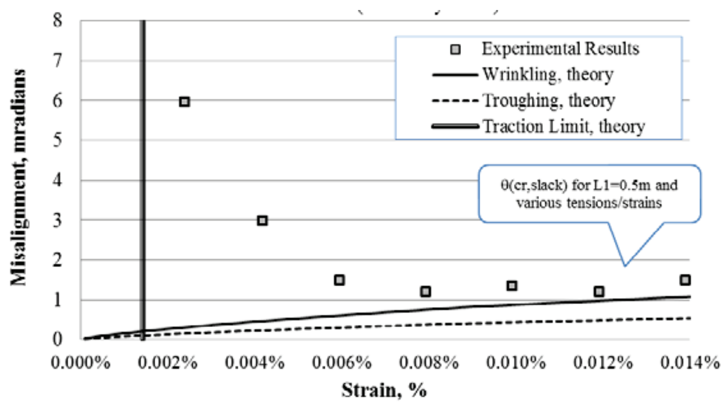

Figure 4 – Shear Wrinkles in Copper Foils - Results vs. Theory

In shear wrinkling of 3-mil paper, we have the poorest fit of experiment results to theory (Figure 5). The 3-mil paper is more wrinkle sensitive than models predict. This could be due to anisotropic mechanical properties in the paper, but that effect, also covered by shear wrinkle models, is not enough to predict this difference.

The wide paper trials had more surprises in store. In the shorter span, isolated span models would predict increased wrinkle sensitivity, but instead we see a decrease in sensitivity to roller misalignment. When the wrap angle was reduced to 20-degrees, no wrinkles would form. These results point to a strong likelihood that the isolated-span model is insufficient to predict the behavior of wide, stiff webs. The combination of span stiffness, moment transfer, buckling past the critical slack edge tram angle, and limited lateral forces relative to width all combine to make it difficult to create the compressive stresses required for troughing and wrinkling.

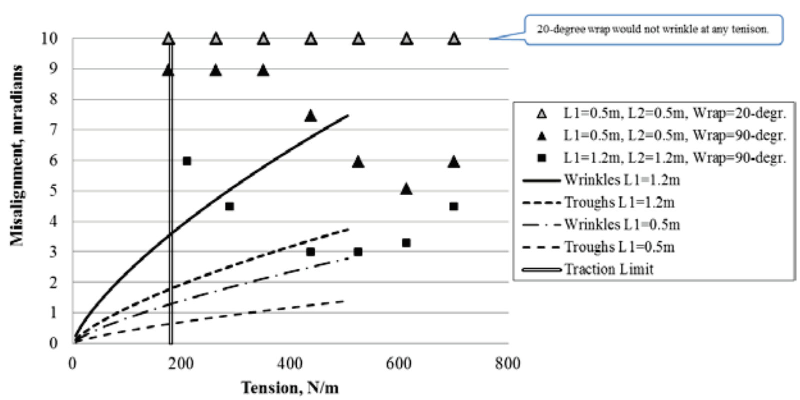

Figure 5 – Shear Wrinkles in 75 micron Paper - Results vs. Theory

Figure 6 shows the shear wrinkling results for our polyester, aluminum, and copper webs and the 20 micron narrow polyester data of Good and Swanson, plotted against tension and misalignment angle. It is difficult to draw conclusions from this graph except that thinner webs will have lower critical misalignment.

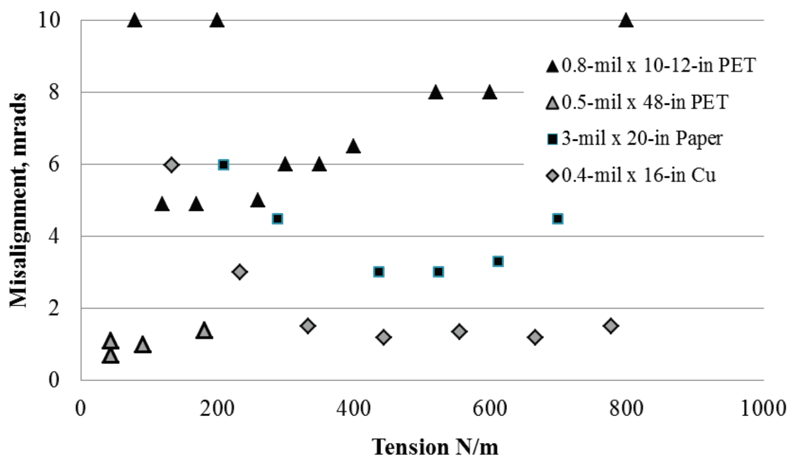

Figure 6 – Summary of Shear Wrinkles vs. Tension

In Figure 7, looking at the same experimental results now plotted against strain instead of tension, the patterns of shear wrinkling is more apparent. Clearly, in all webs, the Regime I and II transitions to wrinkle-free handling are apparent. Most clear is the Regime I transition to a wrinkle-free zone.

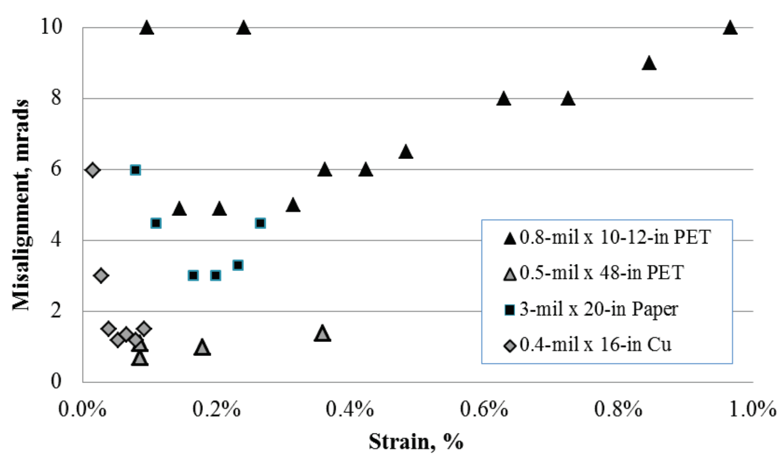

Figure 7 – Summary of Shear Wrinkles vs. Strain

ANTI-WRINKLING AND SPREADING OF WIDE WEBS

The landmark paper on understanding and comparing spreader or anti-wrinkle rollers was published by Ron Swanson of 3M Co. in 1997 [4]. In his work, Swanson characterized ten different rollers, using a standard cylindrical roller as a benchmark and nine other roller designs or commercial products, most widely considered to be either anti-wrinkle or spreader rollers. Swanson compared the nine 'special' rollers to the standard cylindrical roller in two simple tests. In Test 1, Swanson measured the misalignment required to create a shear wrinkle, where a roller was considered a true ‘anti-wrinkle’ roller if it would postpone when shear wrinkles to greater misalignment or indefinitely. In Test 2, Swanson measured the gap that formed at the special roller when a razor blade split the web in half upstream (see Figure 8). A non-deflecting cylindrical roller will not spread the web, but an effective spreader roller will cause the two web halves to move away from the center forming a small gap. However, again, Swanson’s work was also on narrow web (0.25m or 10-in wide polyester).

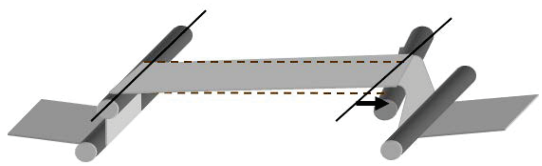

Figure 8 – Schematics of Anti-Wrinkle (Test 1) and Spreading (Test 2) Experiments

Swanson completed all his anti-wrinkling and spreading test using 20 micron by 254mm wide in a 610mm entry span polyester (0.79-mil by 10-in wide, 24-in span), running at 15m/min (50 fpm). The anti-wrinkle tests were run over a tension range of 2000 to 44000 kPa (40 to 875 N/m or 0.25 to 5.0 pli). The spreading gap tests were run at 18000 kPa (2 pli). Though shown in a table, Swanson did not comment on the relationship between the two tests. Figure 9 shows these two test results, anti-wrinkle and spreading, clearly showing that a greater gap in the spreading test correlates to improved resistance to shear wrinkles induced by roller misalignment.

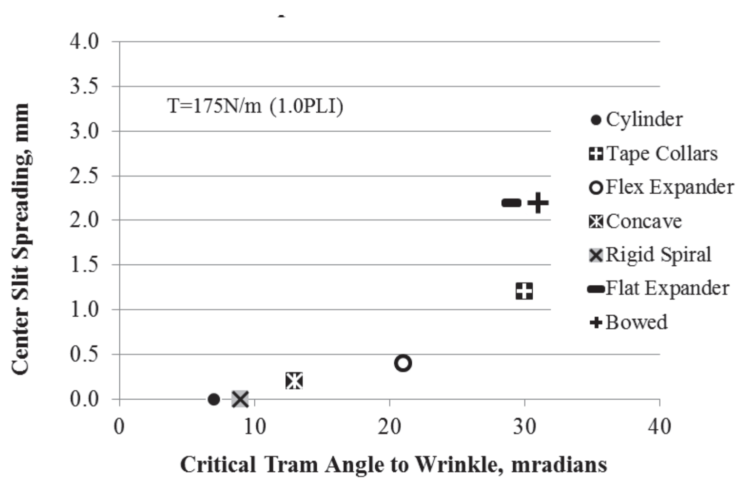

Figure 9 – Anti-Wrinkle vs. Spreading Test Results per Swanson

Our anti-wrinkle and spreading performance trials were completed using 12 micron by 1.2m polyester with a 1.25m entry span (0.5 mil by 48.5-in wide, 50-in span length), running at two tensions or 80 and 175 N/m (0.5 PLI and 1.0 PLI) and speeds below 15m/min (<50fpm) to prevent air lubrication.

Figure 10 shows all of our results of center slit spreading gap (in mm) vs. shear wrinkle generating angle (in mradians). This graph shows the relationship of the anti-wrinkle test relative to the spreading test, separating the data by tension, but not distinguishing between roller designs. Clearly, the more a roller spreads the web, the better the anti-wrinkle performance will be.

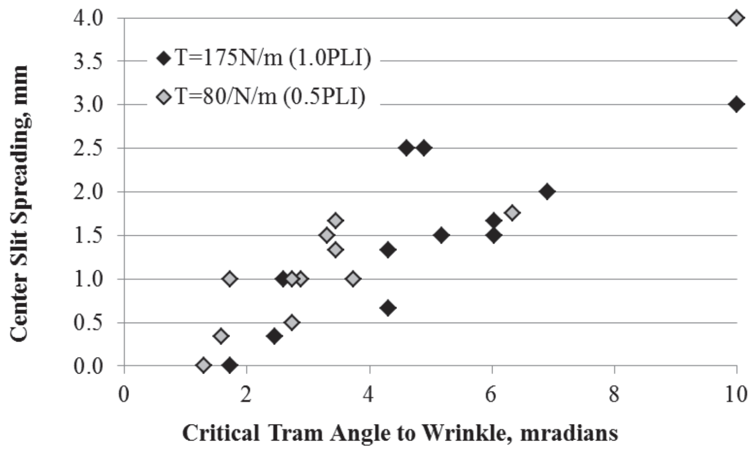

Figure 10 – Wide Web Anti-Wrinkle vs. Spreading Test Results Summary

Figure 11 shows the 80N/m data for the seven types of rollers (the concave rollers are not divided into their different percent diameter variations, only separating hourglass from bowtie concave profiles).

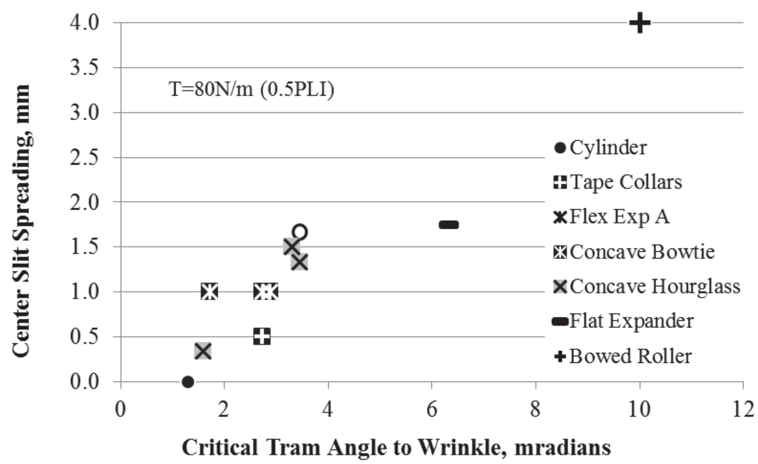

Figure 11 – Anti-Wrinkle vs. Spreading Test Results by Roller, T = 80 N/m

Figure 12 shows the 175N/m data for the seven types of rollers (again the concave roller data is grouped into the two categories of hourglass from bowtie concave profiles).

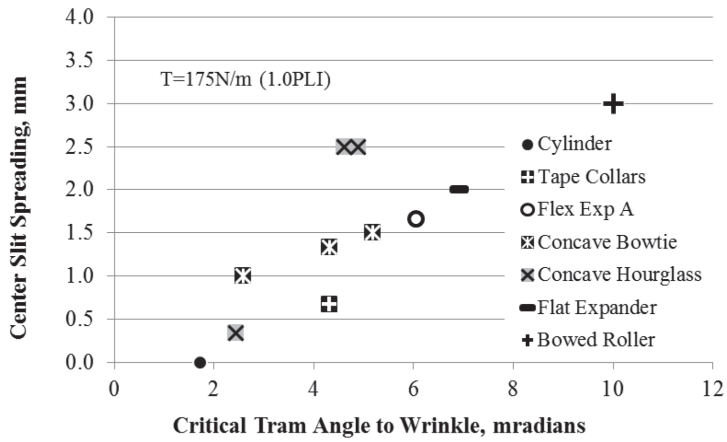

Figure 12 – Anti-Wrinkle vs. Spreading Test Results by Roller, T = 175 N/m

Figures 13 and 14 show the details of the spreading and anti-wrinkle properties of the concave roller profiles, now identified by both profile shape and percent diameter variation. The test tensions of 80 and 175 N/m tension created average strains of 0.2 and 0.4%, respectively.

Figure 13 shows the results of the three bowtie profiles. The machined radial change in the three rollers were 0.1%, 0.5%, and 1.0% or 125, 615, and 1250 micron on a nominal radius of 125mm (2.5, 12.5, and 25 mils over a 2.5-in radius). The bowtie concave profiles all had a center flat width of 1000mm with the taper outside the flat on each side for 225mm. Since the web width was less than the combined width of the center flat and taper widths, the web did not see the full radial variation of the machined profile, but closer to 50% of the taper, making in the bowtie effective radial variations 0.05%, 2.5%, and 0.5%. The bowtie concave rollers had better spreading with higher tension and improved performance with increased radial variations.

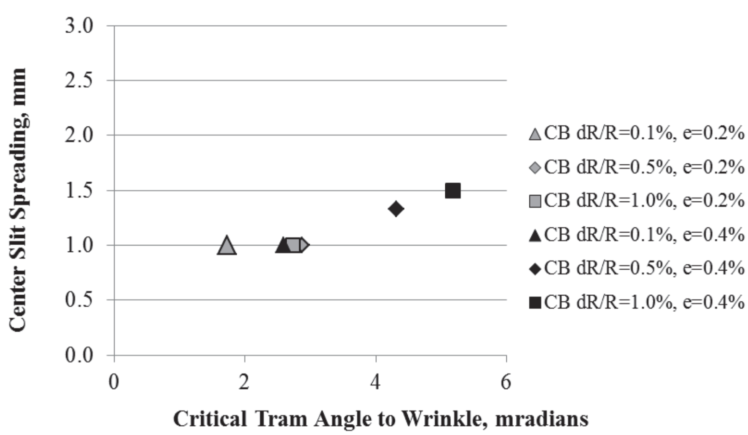

Figure 13 – Bowtie Concave Roller Anti-Wrinkles vs. Spreading Performance

Figure 14 show the results of the three hourglass profiles. The machined radial change in the three rollers were also 0.1%, 0.5%, and 1.0% or 125, 615, and 1250 micron on a nominal radius of 125mm (2.5, 12.5, and 25 mils over a 2.5-in radius). In the hourglass profile, the tapers started at the roller centerline and increased linearly for 736mm (29-in). Since the web width was less than the width of the tapered region, the web did not see the full radial variation of the machined profile, but closer to 80% of the taper, making the hourglass rollers' effective radial variations 0.08%, 0.4%, and 0.8%. As with all hourglass vs. bowtie concave profiles, since more width of web runs on the larger radial regions, much less radial variation is needed to create slackness at the web's center. Thus during our trials, at both the 80 and 175 N/m tension, the 0.5% and 1.0% radial change rollers were under a slack center condition. The greater effective radial change may explain the stronger anti-wrinkle and spreading performance of hourglass vs bowtie concave rollers. The slack center condition may explain why there was little benefit in increasing concavity of the hourglass profile from 0.5% to 1.0%. As with the bowtie roller, high tension created more spreading and anti-wrinkle performance (aided, assuredly, by Regime I benefits).

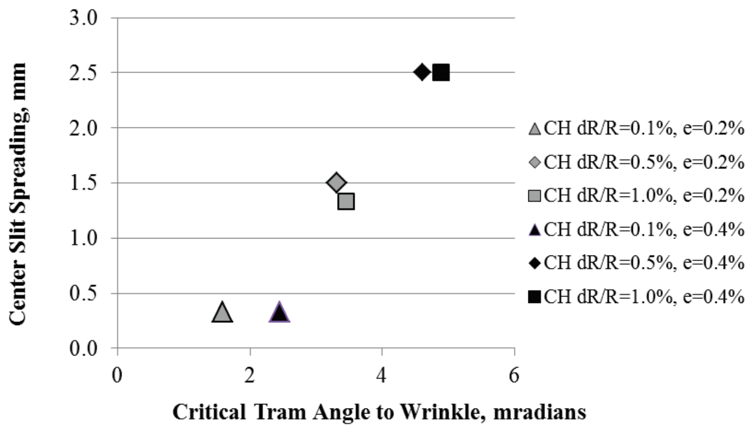

Figure 14 – Hourglass Concave Roller Anti-Wrinkles vs. Spreading Performance

SUMMARY OF ANTI-WRINKLE AND SPREADER ROLLER OPTIONS

Bowed-Axis Rollers

+ The best anti-wrinkle and spreading roller, able to compensate for baggy edges or baggy center. Best with adjustable bow option. Able to evenly spread slit strands (best using two in a displacement guiding geometry). Effective for narrow and wide webs.

- Too many fixed bow rollers (save expenses, but eliminates a primary optimization variable). Most are rubber covered (therefore temperature limited). Commonly over-bowed (leading to excessive wear and shortened cover life). May need to have bow and bow orientation changes with product change, if not input roll change. High drag. Poorly understood. Curved shape is difficult to level and tram.

Flat Expander

+ Second best anti-wrinkle and spreading roller. Always adjustable (no known fixed flat expanders on the market). Straight cylindrical surface allows level and tramming.

- Rubber balloons at high speeds, leading to a crowned profile and wrinkling. Not suitable for slit strand spread (since in wide constructions the center rubber covering does not spread). High drag and poorly understood. Narrow web may have little or no spreading on wide flat expanding rollers.

Flexible Spreader

+ Third best anti-wrinkle and spreading roller.

- Thin webs will crease if grooves are too wide.

Concave Roller

+ May be third best spreader when tailored to the product and process. Best non-rubber option (except for rarely metal segmented bowed-axis rollers), making them insensitive to temperature and most wear resistant option. Much less expensive than first three options.

- Difficult to design for wide range of widths, thicknesses, moduli, and tensions.

Tape Collars on Standard Cylindrical Roller

+ Least expensive, quickest, proven solution. Can be used to prototype concave roller options.

- Tape may shed over time, contaminating product or process. Tape from narrow product must be removed when changing to wide products.

Cylindrical Rollers

+ Lowest cost option. Optimize wrap, spans, traction, drag, and alignment. 99% of rollers are cylindrical and wrinkle free.

ROLLER DEFLECTION WRINKLE MODEL.

Wrinkling on negative bowed rollers, deflecting from the combined effects of tension and gravity (as a function of wrap angle, rotation direction, and gravity orientation) can cause the web to track or gather towards their center, creating compressive stresses beyond their buckling limit. Beyond the pre-roller compressive stresses, additional lateral compressive stresses may develop during contact with the roller. Maximum compressive stresses will be limited by traction available to apply lateral force on the web.

One assumption that is critical to applying the model to long spans is the assumption that the upstream length over which the tracking is active is equal to the actual entry span length, L1, if it is less than the web width and equal to the web width if the entry span length is larger than the web width. This assumption is believed to be an improvement over previous models of negative bow wrinkling [2] where the entry span length is used to calculate maximum lateral strain. This assumption is an approximate application of Saint Venant's principle which states that localized stress distributions at the ends of beams will redistribute to uniform distributions as the distance away from the end becomes greater than one beam width.

In the negative bowed roller case, the deflections and forces at the deflected roller are parabolic in nature and these types of loads can't persist too far upstream – especially if a beam is not buckled. If a beam is buckles (e.g. a thin web), the effect may go further upstream but at some point, there won't be enough lateral compressive stress to buckle the web and troughs will not be present.

Beyond a first principles analysis, post buckling finite element analysis is an option to see when this assumption creates significant modeling deviations.

A model for web wrinkling due to web deflection is developed based on the analysis from two references and experimental data from more recent work in the Optimation Media Conveyance Facility.

Shelton [2] developed a wrinkle model based on application of the right angle rule of web tracking onto a roller deflected by web tension.

Duvall [5] showed that Shelton's theory of right angle tracking onto a deflected roller was not adequate to predict the onset of wrinkling for the cases they studied (polyester film, 6.25 inch wide, 2 mil thick, 54-in. entry span length). These experiments indicated that wrinkling was better predicted by assuming that the web conforms to the deflected roller and thereby assumes in-plane strain equal to the bending strain on the roller at the bisector of the wrap angle.

The formation of the wrinkles also assumes that there is adequate traction force present from belt wrap pressure to sustain the buckling force.

Subsequent evaluation indicates that strain due to web steering can be important, should not be neglected, but is very probably a strong function of the incoming span length-to-width ratio. In the model presented herein, the effect of deflection due to gravity is also rigorously accounted for.

The relationships for this theory are presented as follows: (a) roller deflection web strain due to web tension, (b) roller deflection web strain due to both tension and gravity, (c) roller tracking web strain due to both tension and gravity, (d) critical web buckling strain, (e) comparison between applied and buckling strains, (f) computation of available traction force, and (g) the wrinkling criteria.

Analysis

Figure 15 shows the geometry of a roller and load balance from uniform web tension.

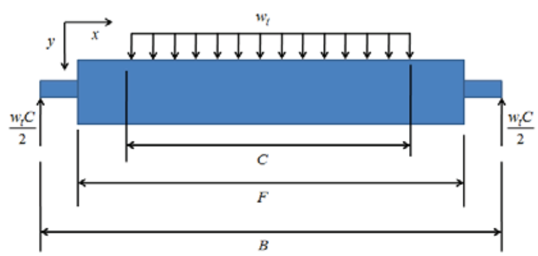

Figure 15 – Roller Geometry and Load Balance

Figure 16 show the balance of moments in the loaded roller. Figure 17 shows the moment applied to the roller as a function of lateral position, x.

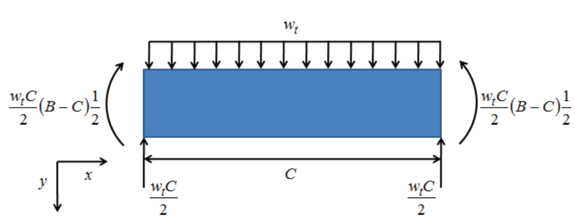

Figure 16 – Roller Geometry and Moment in Web Contact Region

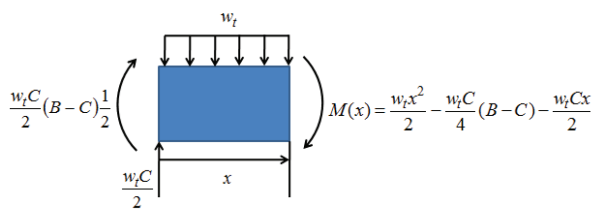

Figure 17 – Moment as a Function of Lateral Position

Strain at Top of Deflecting Roller

The maximum web strain accounting for the arbitrary wrap of the web on the roller can now be found from:

$$\varepsilon(x) = -\frac{MC_f}{EI} \qquad \{1\}$$

Where C_f = -D/2 and by substitution:

$$\varepsilon(x) = \frac{D}{2EI}\left\{\frac{w_t x^2}{2} - \frac{w_t C}{4}(B-C) - \frac{w_t Cx}{2}\right\} \qquad \{2\}$$

The strain is greatest at x = C/2:

$$\varepsilon_{\max} = \frac{Dw_t C}{16EI}\{C - 2B\} \qquad \{3\}$$

To simplify the derivation, Shelton [2] uses the average value of roller strain after assuming a parabolic roller deflection such that the roller deflection at the center and edge of the web is equal to that from the beam deflection equation. With this assumption, there is:

$$y^a(x) = K_1 x(C - x) \qquad \{4\}$$

where after further analysis, K_1 is found to be:

$$K_1 = \frac{w_t C}{96EI}(12B - 7C) \qquad \{5\}$$

from which the roller strain is:

$$\varepsilon_a = \frac{D}{2}\frac{d^2 y}{dx^2} = \frac{Dw_t C}{96EI}(7C - 12B) \qquad \{6\}$$

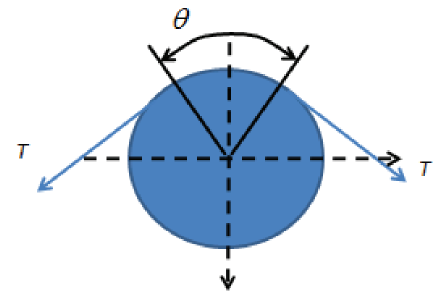

Figure 18 – Vector Load of Incoming and Outgoing Tension

The expression relating the tension load vector to the machine tension is found from Figure 18:

$$w_t = \frac{2T}{C} sin \frac{\theta}{2} \qquad \{7\}$$

The maximum web strain accounting for the arbitrary wrap of the web on the roller can now be found from:

$$\varepsilon_{d,t} = \varepsilon^a - \varepsilon^a cos \frac{\theta}{2} = \frac{D w_t C}{96EI}(7C - 12B)\left(1 - cos \frac{\theta}{2}\right) \qquad \{8\}$$

The effect of roller weight is added by modifying the distributed load in the previous equations to include the effect of weight accounting for the relative orientation between the gravity and tension vectors. The derivation is simplified by making the assumption that the weight density is distributed over the web width in the same fashion as web tension. There are two general cases (clockwise and counter clockwise rotation) and several sub cases for each general case.

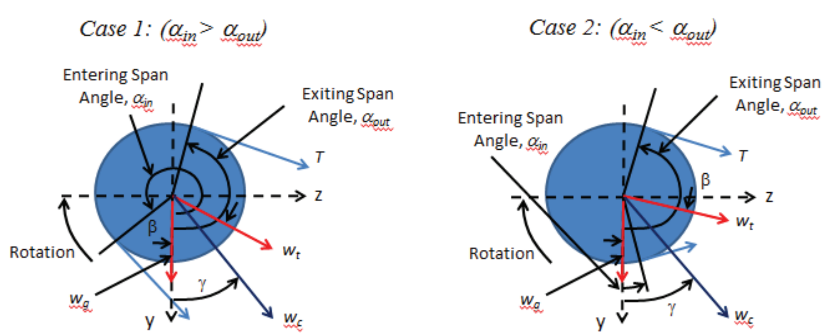

Figure 19 – Roller Wrap Geometry – Clockwise Rotation

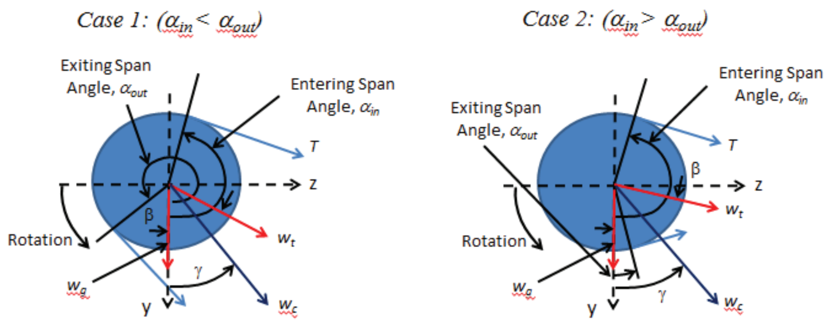

Figure 20 – Roller Wrap Geometry – Counter-Clockwise Rotation

From Shelton [2], the following equation defines the result for kinematic strain due to web steering onto the deflected roller where the combined distributed load is acts in the direction of the gravity load (e.g., the web wrap is 180° and the incoming and outgoing spans are directed down):

$$\varepsilon_{s,c} = -2K_1 L = \frac{w_c C^2 L}{4EI}\left\{\frac{7}{12} - \frac{B}{C}\right\} \tag{9}$$

where L equals the length of the L1 span and w_c is the combined distributed load due to web tension and gravity as determined in the previous section. This solution assumes no spreading due to the effects of the compressive forces on the roller. Further, the strain is constant across the width of the web consistent with the earlier assumption. This leads to slope being a linear function of lateral position and strain being constant across the width.

This result will be additive with the strain computed in the previous section. One further correction is required due to the relative angle between the roller deflection vector and the entrance tangent point. The following results apply depending on clockwise (eqn. 10) and counter clockwise rotations (eqn.11):

$$\varepsilon_{s,cw} = \frac{w_c C^2 L}{4EI}\left\{\frac{7}{12} - \frac{B}{C}\right\} sin(\alpha_{in} - \gamma_m) \tag{10}$$

$$\varepsilon_{s,ccw} = \frac{w_c C^2 L}{4EI}\left\{\frac{7}{12} - \frac{B}{C}\right\} sin(\gamma_m - \alpha_{in}) \tag{11}$$

The critical buckling strain [1] is given by:

$$\varepsilon_{y,critical} = -\frac{0.605t}{R} \tag{12}$$

When the combined web strains due to roller deflection and steering are more negative than the buckling strain, the web is predicted to wrinkle if:

$$-\varepsilon_{d,c} - \varepsilon_{s,c} > \frac{0.605t}{R} \tag{13}$$

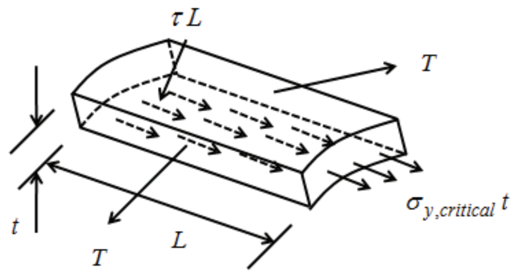

Figure 21 – Schematic of Lateral Traction Limits

From Figure 21, which shows a sector of the web wrapped on the roller:

$$\tau = \frac{T\mu}{CR} \tag{14}$$

From the buckling strain using web Young's modulus:

$$\sigma_{y,critial} = -0.605\frac{E_w t}{R} \tag{15}$$

From a balance of forces:

$$\sigma_{y,critical}\, t + \tau\, L_t = 0 \tag{16}$$

$$L_t = -\frac{\sigma_{y,critial}\, t}{\tau} \tag{17}$$

By substitution of the two equations from the previous page, the following result is obtained:

$$L_t = \frac{0.605 E_w t^2 C}{\mu T} \tag{18}$$

Wrinkling is predicted to occur is two criteria are satisfied. First, the applied strains must be greater than the wrinkling strain and second, the slip distance must be less than one half the web width.

There are four possible combinations:

Case 1	both equations are satisfied	wrinkling will occur
Case 2	the first equation is satisfied but the second is not	wrinkling does not occur because of frictional limitations
Case 3	the first equation is not satisfied but the second is	wrinkling does not occur because there is insufficient roller deflection
Case 4	both equations are not satisfied	wrinkling does not occur because of insufficient roller deflection and friction

Table 1 – Negative Bow Wrinkle Criteria Plot

Case 3 is preferred since traction is high and roller deflection is small. Case 4 is somewhat worse in that even though there are no wrinkles, there is the possibility of scratches due to slippage. Case 2 is even more severe in that there can be sliding and if friction goes up, wrinkles. Case 1 is the worst in that wrinkles are formed.

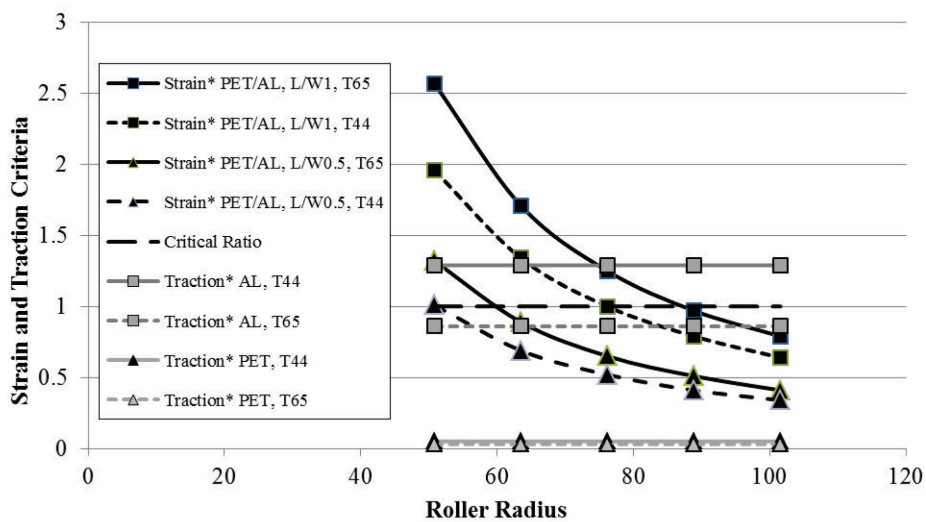

Figure 22 – Negative Bow Wrinkle Criteria Plot – Aluminum and Polyester

Figure 22 shows the negative bow wrinkle criteria plots vs. roller diameter for 12 micron by 1.2m wide (0.48-mil by 48-in) polyester and aluminum webs with entry span length/width ratios of 1.0 and 0.5. The critical ratio is 1 for either wrinkle factor. To predict wrinkles, the strain ratio should be above the critical ratio, the case where strains would be above the buckling limit, and the traction ratio should be below the critical ratio, the case where there is traction force require to exceed the buckling limit is below the available traction.

Example Problem: Negative bow from gravity and tension create wrinkles in 12 micron by 1.2m wide aluminum at 65N/m width (0.38 PLI) or 80N total tension (18 lbs) on a 50mm radius roller (indicated by the circled data points in Figure 23). The roller deflection is sufficient to create both lateral compression and the traction required to exceed the buckling limit.

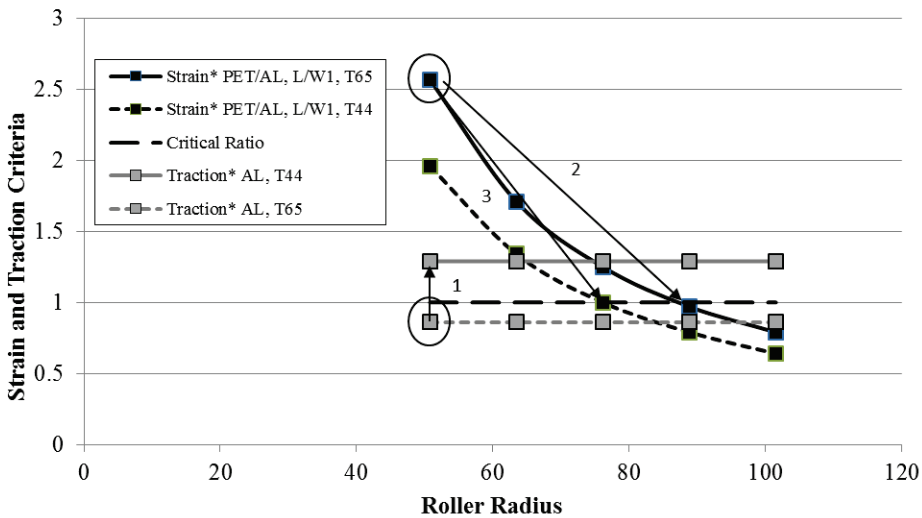

Figure 23 – Negative Bow Wrinkle Solution Options

Solution 1: Lower tension to 44N/m or 53N (0.25 PLI or 12 lbs). Dropping tension increases the critical traction width to greater than web width and applied stresses are too low to create buckling and wrinkles. (This solution was proven during experimental trials.)

Solution 2: Increase roller diameter from 50mm to 88mm. Larger diameter roller will reduce lateral strain roller deflection from both pre-roller tracking and on-roller compression, keeping lateral strain below buckling criteria. (This solution was proven during experimental trials.)

Solution 3: Increase roller diameter from 50mm to 75mm and reduce entry span length from 1.2m to 0.6m. Larger diameter roller combined with shorter entry span will reduce lateral strain roller deflection from both pre-roller tracking and on-roller compression, keeping lateral strain below buckling criteria.

In addition to these solutions, other remedies include: reducing the roller wrap angle, increasing the roller stiffness by increasing wall thickness or elastic modulus, lowering the web-roller coefficient of traction by roller material change or lubrication, or replace the roller with an effective spreader roller. A crowned roller is not a spreader and should not be considered a viable remedy to a deflecting roller at the top of a vertical span. While a crown may improve the roller's top side level, combining two wrinkle mechanisms will only make wrinkles worse.

CONCLUSIONS

- Isolated-span shear wrinkle models show good agreement with experimental results at higher tension and when laboratory conditions are adjusted to minimize moment transfer.

- The transition zone from Regime I to II, including the effects of moment transfer and slack edges, create real world problems in using the isolated-span shear wrinkle model to predict wrinkles, but have the benefit of opening up the wrinkle-free process window.

- Evaluation of anti-wrinkle and spreader roller performance with wide webs yield similar results to Swanson 1997.

- Flat expanders and bowed-axis rollers again proved to be the most effective spreading and anti-wrinkle rollers.

- Wrinkles from negative bow can be predicted by estimating strains from pre-entry tracking and on-roller compression combined with traction limits of applied load.

- Negative bow wrinkling can be reduced by many variables, including larger diameter rollers, lower tension, wrap angle orientation, web and roller widths, and traction controlling factors (speed, tension, radius, web and roller roughness and porosity, and air or liquid viscosity).

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DISCUSSION II *Leaders*: **K. Good & J. Shelton,** Oklahoma State University, USA

Name & Affiliation	Question
Jim Dobbs, 3M Company	It strikes me that the finite element analysis is truly coming into its own. You get results unlike 10-12 years ago that seem to reflect what webs actually do. The results look right, they seem right, they are beautiful and they take only 2 days to calculate a few spans. Doug Kedl and I tried to do this with a rubber band model, a string model, and an elastic model. These modeling efforts have gone on to figure out what that moment transfer span interaction is. Keith Good has done some work beyond that with his closed form simple solution. It strikes me that we need to know how a web behaves throughout a web line. We see explicit analyses that may require two days to study a few spans. We see other alternatives in Jerry Brown's work. My experience is that webs are getting wider, thinner and have higher tensile moduli for a lot of applications. If span interaction governs the day then we have people are out there trying to determine how good do I have to build this machine to prevent moment transfer? What do I do as a practical web handling solution?
Name & Affiliation	**Answer**
Keith Good, Oklahoma State University	Regarding the moment transfer work presented this morning: Nothing was presented there that invalidated the moment transfer model that was developed earlier in closed form. The explicit analysis gave lateral deformation results that were very close to results from the closed form theory. What we were really trying to show with the explicit analysis is how much more you can find out because of the output that was available to you. The model for the critical moment, M_r, used in the closed form theory is crude but it works pretty well. Perhaps the greatest benefit of explicit analysis is that it allows us to work *n* span problems and perhaps simulate entire web lines. Other types of analysis require us to assume boundary conditions whereas with explicit analysis it is possible to have moment interaction in *n* spans, perhaps throughout an entire process machine. The results we presented came from single core computations. Neal Michal presented results using explicit analysis on an accumulator with several spans. I was trying to estimate how many cores and days that must have taken with that many degrees of freedom. Would you care to comment on that Neal Michal?
Name & Affiliation	**Comment**
Neal Michal, Kimberly Clark	It took a long, long time.

Name & Affiliation	Question
Tim Walker, TJWalker & Associates	Keith regarding your paper on the concave roller: You showed that compressive CMD stresses could exist in the exit span of the concave roller. You showed tensile CMD spreading stresses on the concave roller that were largest near the exit. Concave rollers are designed to take a web that wants to trough and pull it taught in the CMD. Isn't this happening even though your highest CMD stresses are at the exit of the roller?
Name & Affiliation	**Answer**
Keith Good, Oklahoma State University	Yes. The web is entering the concave roller and it is spreading the web taught in the entry span as we near the concave roller. A question that is oft asked is how long that spreading effect will last? The explicit results allow us to explore this and see that the concave roller that spread the entry span is inducing troughs in the exit span. Thus the spreading effect is short lived. Will there be a day when we can simulate from unwind to rewind and everything in between? I think that is coming. Can multi-physics analyses be run that would allow air entrainment between webs and rollers such that we do not have to assign constant coefficients of friction? I think the possibilities are endless.
Name & Affiliation	**Comment**
Dave Roisum, Finishing Technologies, Inc.	Until that day when we can afford to model every single roller in the plant, we have to think about safety factors and regard alignment as being a fundamental that we need to set allowable guidelines for the industry. It is a little bit all over the map, but from the data that has been collected experimentally and analytically, it appears you put yourself at risk at one milliradian. Possibly you put yourself at further risk if you run multiple materials, if you do not allow for imperfect webs, if you don't allow for multiple rollers or for tension variations, etc. Maybe we ought to consider classes of alignment: Class B could be set at one milliradian and would be good for tolerant situations, rubber and some nonwovens, etc. Class A for paper or polyester might be set at 0.1 milliradians. There maybe some situations where you might need yet greater alignment. There are guidelines for roller deflection that have been pretty much standardized in the industry. Maybe we should help do the same things for alignment that could help us until we could model every single roller in the line. If we don't do it, who else will?
Name & Affiliation	**Comment**
Tim Walker, TJWalker & Associates	I'll make a couple of comments that are related to accumulators. The tension transients during the accumulator dispensing phases are quite interesting. I expect that is the bigger wrinkling source. This source is the pulsing of width changes resulting from the Poisson

effect combined with tension surges in the web in accumulating. The tension becomes high during the accel case and low during the decel case and there are major web width changes that can relates to scratching and wrinkling. Lateral scratching can be induced by web width change. In the photographic industry, they saw these angled scratches that would have a machine direction component but a lateral component could be 5-10 times greater with a width change over a driven roll, going from low tension to high tension. In the micro-slip zone, you are accommodating the machine direction slip and a width change in the CMD as well. If the tension is changing from high to low on the roller a wrinkle may be produced. If the tension is changing from low tension to high tension on the roller, it will neck in and create a diagonal scratch. The scratches would transition to MD scratches in the middle of the web.

Name & Affiliation	**Comment**
Neal Michal, Kimberly Clark	2 years ago I talked about the tension differential through an accumulator. This work brought up a question that I want to address to Keith and to others: The material was a 17 gsm polypropylene nonwoven with a Poisson ratio of 2.2. I understand that more spreading will reduce the probability for wrinkles. Could be that our material models will not allow Poisson ratio to exceed 1/2 that keep us from finding other sources that cause wrinkles? How do we handle webs where Maxwell's laws are not applicable? I often tell Balaji Kandadai that Maxwell's laws are for wimps because they do not apply for our materials. I knew when I said that you would cringe. We input a Poisson ratio of 0.4 so the models will behave, but does that cast doubts on the results?
Name & Affiliation	**Answer**
Keith Good, Oklahoma State University	The material behaviors that are allowed in elasticity and finite element codes assume that for isotropic materials that that Poisson's ratio is less than 0.5. For orthotropic webs the limits change. The material properties have factors like $1/(1-\nu_{12}\nu_{21})$ where 1 and 2 are the MD and CMD directions, respectively. Now one of the Poisson ratios could be 0.3 while the other could be as high as 3.33 before either a division by zero error or if greater than 3.33 a violation of physical reality would occur by driving the factor negative. An example of violation of physical reality would be to exert a force on a body to the right but it responds by moving left. Orthotropic homogenous materials should obey Maxwell's Laws which dictate the in-plane moduli and Poisson ratios are dependent on one another. Homogenous is a key word here because non-wovens are full of voids. A basic study of nonwovens where all in-plane moduli and Poisson ratios are measured would be interesting Neal. Then we document the

applicability of Maxwell's relations to non-wovens if possible. If the answer is no then the modeling of non-wovens will require fiber by fiber analysis including bond points. So let us hope the answer is yes.

Marko, I wanted to get you to talk about winder vibration: Many people don't have winders where they have drum support. They have winders with supported cores and often a lay-on or rider roll. They may witness a lot of dynamic bounce problems of the rider roller or possible combined vibration of the winding roll and the rider roll. You discussed damping this morning. The form of damping you discussed related to the bearing blocks and trying to damp motion there. I thought you had more to say about damping and damping materials that maybe you could say a few words about.

Name & Affiliation	**Comment**
Marko Jorkama, Metso Paper	Concerning damping materials: We have been studying what is available in the market. The best one found so far came from Finland. It was developed in a cooperative project with VTT company.
Name & Affiliation	**Question**
Keith Good, Oklahoma State University	Where are you using these damping materials? Are you covering rollers with them?
Name & Affiliation	**Answer**
Marko Jorkama, Metso Paper	We are using them in the bearing housings, so the bearings are supported flexibly. The damping faces are still relatively stiff so under the weight of the paper rolls, the deformation is small during winding, about 0.5 mm. There are 2 drums that are sinking the same amount so there will be no problems with the misalignment.
Name & Affiliation	**Question**
Keith Good, Oklahoma State University	Let's recast the problem. Let's say we have a two drum winder and we have a nip roller on top. You've isolated the nip roller as the vibration problem. Maybe you can't change the stiffness and you need to do something about damping up. Would you propose a similar solution there? Would you insert damping material between the support for the nip roller and its base?
Name & Affiliation	**Answer**
Marko Jorkama, Metso Paper	I would consider vibration absorbers in that case.
Name & Affiliation	**Comment**
Doug Offerhaus, Catalyst Paper	I just wanted to follow up on Dave Roisum's comments regarding alignment tolerances. In the industry we would certainly appreciate better guidelines for alignment tolerances. So far, we've had to rely primarily on the tolerances provided by the OEM's. I would suspect that a group like this could provide better information. The papers given this afternoon I found very beneficial because

they provided better insight into the effects of roll misalignment. I especially appreciated Neal's paper on accumulator wrinkles and the effect of misalignments. I've had the privilege of working on a lot of reel stands over the years in customer press rooms and now I have a better understanding of what is happening.

Name & Affiliation	**Question**
Günther Brandenburg, Technische Universität München	I am interested in the following problem. I pointed out that I am not a specialist in wrinkling, but my question would be that if you steer a roller, is there an optimum pivoting point where the tendency of wrinkling goes to a minimum?
Name & Affiliation	**Answer**
John Shelton, Oklahoma State University	There have been some recent installations of utilizing the tension in the web and self weight of rollers to cause roller deflections to achieve spreading. This requires orientation of the spans to optimize the effect of the roller deflection to achieve spreading. There are lots of things we've not done because we didn't know well enough how to do them.
Name & Affiliation	**Answer**
Keith Good, Oklahoma State University	In some cases, roller deflection can be used to your advantage and in other cases it might result in wrinkling the web. In terms of misalignment, I don't think it really matters where the pivot point is.
Name & Affiliation	**Answer**
Tim Walker, TJWalker & Associates	The web sees the absolute misalignment and the lateral motion of the roller. In a dynamic case, it might make a wrinkle. If you carry the web laterally very quickly, it doesn't matter what the alignment is, that will create a wrinkle.
Name & Affiliation	**Question**
Günther Brandenburg, Technische Universität München	I meant the following: If you steer a roller in order to influence the edge position of the web is there an optimum point where the tendency for wrinkling is a minimum, not misalignment, but steering in full tension?
Name & Affiliation	**Answer**
Tim Walker, TJWalker & Associates	The term steering as I used here is used to describe a downstream misaligned roller alignment relative to an upstream roller. That misalignment, which can be dynamic, causes the web to move left or right. The resulting lateral web motion has components due to the lateral translation of the roller and an angle change component. The web is sensitive to both components regarding wrinkling. If you translate the roller too quickly you will induce a wrinkle from dynamic internal shear. If you pivot the roller slowly, a wrinkle will result from the steady state shear due to normal entry. When you misalign a roller at moderate speed the two effects compile and a wrinkle will form at a lower misalignment than it would have if you misaligned the roller slowly. Offset pivot guides don't have this problem. They displace the web laterally and by twisting

Name & Affiliation	Comment
	the entry span to the guide rollers. No internal shear is produced.
Name & Affiliation	**Comment**
John Shelton, Oklahoma State University	These are sometimes called displacement guides.
Name & Affiliation	**Comment**
Kevin Cole, Optimation Technology, Inc.	Today has been a really good day. I almost view it as a competition between a couple of different very interesting methodologies for attacking the lateral dynamics, winding and all sorts of problems. I've always been intrigued by Jerry Brown's papers. I have a sense that there is a lot there and a lot of capability and potential. As I hear the results of the explicit finite element, it's like that seems to solve it all, too. My question is an observation - how do you decide how to pick a winner? Which one is the winner, which one has the best potential to get to all the things we are trying to get to, or are they in fact the same method, just slightly different ways of doing things and in the end the same method of solving all the equations and not making assumptions a priori. Ultimately, we can't afford multiple methodologies as we get to the higher complexities. That is my struggle right now and my challenge to the community.
Name & Affiliation	**Answer**
Keith Good, Oklahoma State University	Consider what Jerry said at the end of his paper concerning friction. Here are some equations, but he was unsure how to implement them into his model at this point. If he's able to do it, there will be considerable savings in solution times in use of his model. With a working friction model Jerry won't have to assume kinematic boundary conditions either. I don't think we know the answer yet, Kevin. Explicit analyses are nice from the standpoint that only basic boundary conditions are set, web tension at one web end and web velocity at the other. What happens to the web at the rollers is dictated entirely by the friction forces that form and stick and slip behaviors can exist anywhere in the contact of a web and a roller. Probably one of the more noble uses of explicit analyses is the exploration of kinetic and kinematic boundary conditions between webs and rollers. Added to that is the desire that companies to solve problems such as web handling problems using commercial codes rather than standalone codes where the programmer has made assumptions that may be invalid for the problem at hand.
Name & Affiliation	**Comment**
Ron Lynch, Procter & Gamble	A benefit to the explicit finite element codes is the ability to handle imperfection. Perfect webs on perfect machines don't wrinkle or mis-track. Many of the problems we see are driven by imperfections of rollers and webs. These imperfections are hard to deal with when solution methods will not allow the web or roller imperfection to be properly

	defined and solved. I think long term the explicit FE analysis will provide great value in solving that class of problems.
Name & Affiliation	**Answer**
Keith Good, Oklahoma State University	I would agree with that. When we were discussing that, I was not thinking about imperfect webs, but an imperfect web traveling through a web line is a time transient problem, I'm not sure what tool other than explicit analysis which is solving the equations of motion through time can solve such problems.
Name & Affiliation	**Comment**
Kevin Cole, Optimation Technology, Inc.	I've seen recently examples where the troughs are localized. When you coat paper webs you locally distort the paper. You have troughs, then no troughs, then troughs. That problem is the toughest one of all because everything is dynamic. It is moving through the system. We obviously try to exploit that fact that you can view the web line either from the fixed reference or you can ride along on the web; that all works well when things are set up in space and are fixed. But when you are running things through, I think you have to the explicit finite methodology. There is really no other way.
Name & Affiliation	**Comment**
Keith Good, Oklahoma State University	You must at least move to a method where time is allowed to vary, but it's not a fixed web model in space.
Name & Affiliation	**Question**
Ron Swanson, 3M Company	Neal Michal: In Figure 16, you show some wrinkle test curves that don't match well with the theory. I'm curious as to how you performed the test. You are actually tipping one roller with a 180 degree web wraps, is this correct?
Name & Affiliation	**Answer**
Neal Michal, Kimberly Clark	If we were to take this graph from Figure 16 and break it into two different graphs, one for high tension, one for low tension, you will see that the data points for three rolls misaligned, do not line up with the theory curve with the exception of one data point. For the most part, everything is well below that Nike swoosh shaped theory curve. When we misaligned one roller on the high tension case, it did fall right on top of Dr. Good's prediction for misalignment.
Name & Affiliation	**Question**
Ron Swanson, 3M Company	For the cases that don't agree with theory, you misaligned the whole rack?
Name & Affiliation	**Answer**
Neal Michal, Kimberly Clark	Yes, all three rollers were misaligned except for where I made note that we misaligned only one roller.
Name & Affiliation	**Question**
Ron Swanson, 3M Company	On the tests that developed that equation, we were very careful to do a 90 degree wrap so that we have bending in one span and pure twisting in another. So you have bending in the input and output spans. That is very different.

Name & Affiliation	Answer
Neal Michal, Kimberly Clark	All the work I could tell for wrinkling is a single roll, open span, 90 degree only.
Name & Affiliation	**Question**
Ron Swanson, 3M Company	I turn that to Keith and ask what happens when you have a 180 degree wrap. It seems to me that you would get moment transfer that would almost cancel the previous span.
Name & Affiliation	**Answer**
Keith Good, Oklahoma State University	I have been looking at this recently. Consider the guiding of the web, as it moves from an upstream roller, goes over a misaligned roller 180 degrees and finally moves to a downstream roller. Further assume the entrance and exit web spans to the misaligned roller are equal in length. Without analysis you might believe that the web returns to the downstream roller at the same lateral location it left the upstream roller. It does not, it is very different and thus the internal shears that form in accumulator where 3 rollers are misaligned will be very different than the shear formed in the entry span of a misaligned roller where the exit span is in twist. From the standpoint of internal shear they are very different problems.