DESIGN OF CONTOURED ROLLERS FOR WEB SPREADING

by

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ABSTRACT

Contraction of a web in the cross width direction while running over rollers can be traced to runability problems and wound roll defects. These defects may include troughing, baggy lanes, wrinkles and registration difficulties on the machine and creases in the wound roll. Web spreading devices have long been used in industry to restore the moving web to a taught width. At IWEB 4, Swanson [1] presented cases for the effectiveness of ten such devices along with simple models to calculate their ability to spread the web. This presentation extends the previous work by looking at two devices, the parabolic and the "bow tie" shaped rollers, and presents the mathematical derivations used to calculate the spreading ability of each roller. In each case, equations will be presented that, given a reduction in web width, a roller profile may be designed to restore the web to its original width.

NOMENCLATURE

a_0	Parabolic roller profile coefficients, m
a_1	Parabolic roller profile coefficients, 1/m
E	Machine direction modulus of elasticity, Pa
I	Moment of inertia, m^4
L	Span Length, m
M	Bending Moment in web span, N-m
r	Roller radius, m
r_d	Roller radius in cylindrical section of bow tie roller, m
r_e	Roller radius at y=W/2 in bow tie roller, m
t	Thickness, m
T	Web line tension force, N
v	Lateral deflection of 1/2 width web, m

V_{avg}	Average velocity of the web, m/s
$V(y)$	Velocity of the web as a function of y location, m/s
W	Web width, m
y	Distance from center of the web, m
y_c	Distance from 1/2 web centroid in the cross machine direction, m
y_d	Distance from center of roller to intersection of cylindrical and conic section, m
$\varepsilon(y)$	Strain as a function of y location
σ_{md}	Machine direction stress, Pa
σ_{ys}	Material yield stress, Pa
ρ	Equivalent radius of curvature, m
ω	Angular velocity of the roller, rads/s

INTRODUCTION

Width wise contraction of the web is inherent in web processing. In order for a web to travel down the process machine it must be submitted to sufficient external force to cause it to move. This force is usually denoted as the web line tension or machine direction stress. The machine direction stress induces a tensile machine direction strain (Hooke's Law). The Poisson effect will then induce a contracting cross machine direction strain. This cross machine direction strain acts on the web in a manner to reduce the width thereby making the web narrower than its original width.

When this width contraction becomes severe enough, visible indications may appear in the web such as troughing or baggy lanes. In more pronounced cases the troughs may, upon entering a roller with sufficient friction as not to allow lateral slippage, form out-of-plane bands that lift from the roller. Should these bands not be able to support themselves, they can fold over and form wrinkles causing permanent deformation to the web.

Much time and effort has been devoted to characterizing and designing spreading elements to reduce the effects of the cross machine direction contraction. In previous IWEB conferences Roisum [2], Swanson [1], and others [3,4] have given synopsis on differing spreading devices and have derived models for a few of the elements. Still with the general knowledge presented in the literature there lacks a robust design model that the engineer can use to specify the parameters of the spreading element. The research presented here is offered as a tool to aid in calculating the effect and shape of two common spreading elements, the parabolic and "bow tie" spreading rollers and their associated conditions of use.

PARABOLIC CONCAVE ROLLER

To begin analysis of the parabolic concave roller the web is treated as a beam and classical and matrix structural analysis is used to describe its behavior. Since the web and roller are assumed to be symmetrical about the longitudinal center line only one half of the width of the web is shown for analysis purposes in Figure 1. Also it is assumed that the web that has been gathered or troughed can be analyzed as two separate beams of web that will be steered outward by the concave roller. Given the properties of the web and a roller profile the goal is to calculate the lateral deflection imparted to the web by the spreading roller. Assuming normal entry of the web to the roller as shown in Figure 2, matrix structural analysis [5], considering only elastic stiffness, gives the lateral deformation of the beam subject to a bending moment to be:

$$v = \frac{L^2}{6EI} M_{centroid} \tag{1}$$

where the lateral deflection (v) of 1/2 of the web (W/2) is given in terms of span length L, Young's modulus E, moment of inertia I, and the bending moment about the centroid of 1/2 width section. L, E and I are measurable properties of the web. Only $M_{centroid}$ needs to be derived. From classical beam theory [6] the bending moment is given as:

$$M_{centroid} = \int_{-W/4}^{W/4} \sigma_{md} \, y_c \, t \, dy \tag{2}$$

where σ_{md} is the stress distribution , y_c the distance from the centroidal axis , and t the web thickness. The values of y_c and t are measurable and only leave σ_{md} to derive. From Hooke's law the machine direction stress, σ_{md}, can be related to the strain in the web:

$$\sigma_{md} = E \ \varepsilon_{md}(y) \tag{3}$$

The strain imparted to the web can be written in terms of the web's velocity (4). Assuming that the web remains in traction with the roller and moves with the same velocity as the portion of the roller with which it contacts, the web velocity is simply the roller surface velocity which is the roller radius multiplied by the roller's angular velocity.

$$\varepsilon_{md}(y) = \frac{V(y) - V_{avg}}{V_{avg}} \tag{4}$$

Given the parabolic profile (5), the velocity can be written in terms of the profile coefficients and the angular velocity of the roller.

$$r = a_0 + a_1 y^2 \tag{5}$$

$$V(y) = (a_0 + a_1 y^2)\omega \tag{6}$$

In equation (4) the strain is given as a change in velocity in relation to the average velocity of the web. The integrated average of the velocity of the web in contact with the roller is:

$$V_{avg} = \frac{1}{W} \int_{-W/2}^{W/2} (a_0 + a_1 y^2)\omega \, dy = \left(a_0 + \frac{a_1 W^2}{12} \right) \omega \tag{7}$$

Using equations (6) and (7) to solve for the strain in (4) yields:

$$\varepsilon_{md}(y) = \frac{a_1 y^2 - \frac{a_1 W^2}{12}}{a_0 + \frac{a_1 W^2}{12}} \tag{8}$$

Substituting the result of equation (8) into (3) yields the stress across the web width. The coordinate system for the beam derivation in (1) was considered to be one half the web width with the centroidal axis in the center of the half width as shown in Figure1. Since equations (4) through (8) were based on a full width coordinate system the full width y coordinates must be transformed to the half width y_c coordinates. This is done via the substitution:

$$y_c = y + \frac{W}{4} \tag{9}$$

Equation (8) now becomes:

$$\varepsilon_{md}(y) = \frac{a_1\left(y_c + \frac{W}{4}\right)^2 - \frac{a_1 W^2}{12}}{a_0 + \frac{a_1 W^2}{12}} \tag{10}$$

Substituting (10) into (2) and solving for the bending moment yields:

$$M_{centroid} = \frac{E\, a_1\, t\, W^4}{16(12\, a_0 + a_1\, W^2)} \tag{11}$$

Finally inserting equation (11) into (1), the lateral deformation of one half the web width is determined. To get the total spreading effect of the roller multiply the value obtained in (1) by 2 to account for both halves of the web. The closed form of equation (1) resolves to:

$$v = \frac{a_1 L^2 W}{12 a_0 + a_1 W^2} \tag{12}$$

Webs whose length to width ratios approach that of a classical beam (approximately 10) often have reduced lateral deflections due to a characteristic known as tension stiffening. This results in the tension of the web and its geometric properties adding to the rigidity of the beam element. Therefore in addition to the elastic analysis, as given by equation (1), another term is require to account for the geometric stiffness. Again from matrix structural analysis [5] equation (1) becomes:

$$v = \frac{M_{centroid}}{\frac{6EI}{L^2} + \frac{T}{10}} \tag{13}$$

Following the same sequence of steps as given above, shifting the coordinate system, and accounting that only half of the tension is available for half the web span, the lateral deflection for a half web beam is:

$$v = \frac{5 a_1 E L^2 t W^4}{(12 a_0 + a_1 W^2)(4 L^2 T + 5 E t W^3)} \tag{14}$$

where T is the web tension in units of force.

The lateral deflection given by equation (12) or (14) represents only half of the total deflection. If the amount of width reduction of the web is known, (v) can be set to half this value and the equations solved for the coefficients needed to manufacture a parabolic roller that will restore the web to its original width. With the coefficients a_0 and a_1 known the equivalent radius for a parabolic tool swing on a lathe can be approximated as:

$$\rho = \frac{1}{2 a_1} \tag{15}$$

Additional Considerations

In order for the parabolic roller to effectively spread the web, the web must remain in contact with the roller surface. Due to the surface being concave inward, the web will tend to lift off the roller unless enough web tension is applied to remove all machine direction compressive stress. The most likely place for compressive stresses to occur is at y = 0, the center of the roller. To insure that there is enough web tension to prevent negative machine direction stresses in the web at this location, convert the stress given by equation (3) into a tension force by multiplying by the area of the web, W * t, and setting y = 0. In equation form this yields:

$$T \geq \frac{E a_1 W^3 t}{12 a_0 + a_1 W^2} \tag{16}$$

where T is the web tension if units of force. This inequality also specifies the minimum tension required to conform the web to the parabolic roller surface. Even though the web may be in contact with the roller across the roll width the machine direction stresses will be a maximum at the web edge and minimum on the centerline. In the central zone where the web stress is minimal air entrainment can greatly reduce the web to roller traction. This can limit the moment that can be applied to the web through friction from the roller. Thus tension above and beyond that predicted by expression (16) may be required to ensure normal entry of the web to the concave roller.

It is also necessary when designing the roll to make sure that the edges of the web are not stretched to the extent that the yield stress of the material is exceeded. The total stress seen by the web at it's edges must not be greater than it's yield stress or some predetermined threshold stress. The total stress can be calculated by summing the bending stress given by (3) with the web tension. This sum must then be lower than the material's yield stress. The mathematical statement for this condition is:

$$\sigma_{ys} \geq \frac{2\, E\, a_1\, W^2}{12 a_0 + a_1\, W^2} + \frac{T}{W\, t} \tag{17}$$

Further, the traction between the web and the roller must be adequate to sustain the lateral forces and the moment generated by the parabolic roller. If the traction is not adequate, increasing web tension or a surface treatment of the roller to raise the coefficient of friction may be in order. There may also be air entrainment issues at higher web line speeds that reduce the traction capability that need to be considered.

BOW TIE ROLLER

Often in a web process lines rollers with taped edges are used to prevent wrinkles. Just as in the case of the parabolic roller the "raised" taped edges induce a bending moment into the web causing it to spread along its width. A more durable form of the taped roller is the bow tie roller. As seen in Figure 3, this roller has a cylindrical mid section with conical end regions. One of the bow tie roller benefit over the parabolic roller is ease of manufacture. Many machine shops many not have the capability to turn a large radius contour onto a roller but almost all shops are able to turn linear tapers. Depending of the manufacturing facility available and machining costs the bow tie roller may be preferable to the parabolic roller.

In the analysis of the bow tie roller the roller is broken into 2 sections, the cylindrical and conical regions. Equations will be derived for both sections and will follow nearly the same steps as presented in the parabolic roller derivation. In this analysis the tension stiffened form of the deflection equation, (13), will be used to derive the deflection. Later the tension will be set to zero to arrive at the elastic solution.

The total bending moment that is generated in the web by the roller is the sum of the moment on the cylindrical region (denoted as subscript od) and the conical region (de). (For clarity these subscripts refer to the cylindrical section starting at zero and ending at point d, and the conical section starting at point d and ending at the web edge, point e.) This moment, as in the parabolic derivation, is about the neutral axis of the half web width.

$$M = \int_{-W/4}^{y_d - W/4} \sigma_{od}\, y\, t\, dy + \int_{y_d - W/4}^{W/4} \sigma_{de}\, y\, t\, dy \tag{18}$$

The stresses in each section are simply given as:

$$\sigma_{od} = E\, \varepsilon_{od} \tag{19}$$

$$\sigma_{de} = E\, \varepsilon_{de} \tag{20}$$

The strains can be calculated from the velocity variation as given in (4) and again, the velocities are a function of the radius profile of the roller. The radius profile in the cylindrical region (od) is given as:

$$r_{od} = r_d \tag{21}$$

and in the conical region (de) as:

$$r_{de} = \frac{r_d(W - 2y) + 2r_e(y - y_d)}{W - 2y_d} \tag{22}$$

The average velocity can be obtained using the expression:

$$V_{avg} = \frac{2}{W}\left[\int_0^{y_d} r_{od}\, \omega\, dy + \int_{y_d}^{W/2} r_{de}\, \omega\, dy\right] \text{ or} \tag{23}$$

$$V_{avg} = \frac{\left[r_e(W - 2y_d) + r_d(W + 2y_d)\right]\omega}{2W} \tag{24}$$

Expression (24) can be solved for ω and the surface velocity of the roller to be calculated per equation (26).

$$\omega = \frac{2\, W\, V_{avg}}{(r_e(W - 2y_d) + r_d(W + 2y_d))} \tag{25}$$

$$V = r\omega \tag{26}$$

The velocity terms for the cylindrical and conical regions resolve to:

$$V_{od} = \frac{2\, r_d\, W\, V_{avg}}{r_e(W - 2y_d) + r_d(W + 2y_d)} \tag{27}$$

$$V_{de} = \frac{2WV_{avg}[r_d(W - 2y) + 2r_e(y - y_d)]}{(W - 2y_d)[r_e(W - 2y_d) + r_d(W + 2y_d)]} \tag{28}$$

With these velocities and the average velocity per equation (24) the machine direction strain can be determined at any y location across the web width. The stresses can be calculated per expressions (19) and (20) and finally the moment can be determined using expression (18) all of which yields:

$$M = \frac{E\, t\, W(r_e - r_d)(W^2 + 2\, W\, y_d - 8\, y_d^2)}{24[r_e(W - 2y_d) + r_c(W + 2y_d)]} \tag{29}$$

Substituting equation (29) into (13) to arrive at the tension stiffened lateral deflection of one half of the web yields:

$$v_{ts} = \frac{10\, E\, t\, W\, L^2(r_e - r_d)(W^2 + 2\, W\, y_d - 8\, y_d^2)}{3\,(4L^2\, T + 5\, E\, t\, W^3)[r_e(W - 2y_d) + r_d(W + 2y_d)]} \tag{30}$$

Letting T in equation (30) be zero and simplifying gives the relationship for the lateral deformation considering only the elastic behavior.

$$v = \frac{2\, L^2(r_e - r_d)(W^2 + 2\, W\, y_d - 8\, y_d^2)}{3\, W^2[r_e(W - 2y_d) + r_d(W + 2y_d)]} \tag{31}$$

Expressions (30) and (31) will now predict how much one half of the web will move laterally. Again, the total web spreading will be twice this amount. With these tools in hand the engineer can now select y_d, y_e, r_d, r_e for the bow tie roller that will remove the slackness from the half web width. To aid in simplicity, the value of r_d can be specified as the nominal web line roller diameter. Also, it is assumed in the derivations that y_e is the half web width leaving only y_d and r_e as unknowns. Since y_e is located at the web edge the designer must realize that the taper of the roller must be continued for a distance past the web edge or else the web may droop over the edge of the roller and suffer damage.

Additional Considerations

As in the parabolic roller case adequate web to roller contact must be maintained in order for the bow tie roller to effectively spread the web. Sufficient web line tension must be applied to overcome the central zone compressive stresses produced by the bending moment. Converting the stresses at the center of the web to tensile forces, equation (32) gives the minimum web line tension that must be applied to ensure that negative machine direction stresses do not occur in the cylindrical region.

$$T \geq \frac{Wt(r_e - r_d)(2y_d - W)}{W(r_d + r_e) + 2y_d(r_d - r_e)} \tag{32}$$

The stress at the web's edge must not exceed the yield stress of the material. The stresses due to bending and web line tension combine to form the total stress on the edge as shown in equation (33).

$$\sigma_{ys} \geq \frac{E(r_e - r_d)(2y_d + W)}{W(r_d + r_e) + 2y_d(r_d - r_e)} + \frac{T}{Wt} \tag{33}$$

EXPERIMENTAL RESULTS

To determine if indeed the derived equations are valid, an experimental apparatus was constructed as shown in Figure 4. The station consisted of a unwind shaft coupled to an electric brake to provide tension, a test section where the web traversed a known length prior to entering the spreading roller, followed by a rewind station. A tension control system was used to provide constant tension during the tests along with load cells mounted to the upstream roller so that the tension could be accurately measured just prior to the test span. The upstream roller was mounted to linear ways so that the span length could be varied.

In the test section, just after exiting from the upstream roller, the web is slit at the center per the method described by Swanson [1]. This allowed each half of the web to seek its preferred path and lateral deformation under the influenced of the spreading roller. The lateral deformation was then measured and noted along with span length, tension and material properties for comparison with the theoretical analysis. This allowed verification of the algorithms presented on narrow webs where the width reductions are typically small. The rollers shown in Table 1, along with their properties, were available for testing. Table 2 lists the properties of the webs.

Parabolic Concave Roller

The first battery of tests was run using three parabolic rollers. Figure 5 shows the results for a roller that was nominally 55.5 mm (2.2 in.) in diameter at the centerline (Parabolic 2 in Table 1). The equivalent radius of the tool cut was 10.16m (400 in.). The web was a low density polyethylene (LDPE) material 0.152m (6 in.) in width. Equation

(14) was used to provide the theoretical spreading deformations that are plotted as continuous curves as a function of the upstream span length. The experimental values were measured using an engineers scale mounted directly in front of the slit web halves where they contacted the spreading roller. The measurement accuracy was improved using a video camera with a 15X optical zoom. Two observations can be noted from this plot, the effect of tension (as tension increases slit separation decreases) and the agreement between experimental and theoretical data. The critical tension as per equation (16) is 2.19N. The lowest tension in Figure 5 is 8.9N, well above the critical tension, therefore good contact between the web and roller is predicted and is evidenced by the closeness of fit of the data.

Next a roller that was nominally 48.2 mm in diameter (Parabolic 1 in Table 1) was tested with an equivalent tool cut radius of 1.23m (48 in.). The web was LDPE that was 0.073m (2.87 in.) wide and the results are shown in Figure 6. This roller is quite an aggressive spreader due to the relatively small toll cut radius as seen by the magnitude of the slit separations in the plot. These slit separations are on order about 3 times those shown in Figure 5. The results for the slit lengths below 0.2m (resulting in lower slit separations) agree quite well with the plots of equation (14). However the experimental data points above 0.2m slit length which result in quite large slit separations show a regime in which the calculated bending moment may not be valid. The critical tension for this test is 2.3N that is below the lowest tension in the plot, good web to roller contact is expected. The yield strength of the LDPE is about 5.5 MPa. At the lowest web tension the average web stress was 4.8 MPa and thus all data taken at higher tensions resulted in average web stresses in excess of the yield stress and thus use of expression (3) is invalid when predicting the moment.

Next a 23.4 μm (0.00092 in.) thick 0.152m (6 in.) wide polyester (PET) material was run on the Parabolic 2 (refer to Table 1) roller. The results are shown in Figure 7. The PET has 25 times higher machine direction modulus than the LDPE. The higher stiffness will result in the PET web not being as compliant to the roller profile as the LDPE web. In Figure 7 many of the experimental points are well below the theoretical plots. Using equation (16) yields a critical tension of 50.4N. All but the highest tension shown in the plots are relatively equal to or below the critical tension. At these lower tensions the web to roller contact is poor or is not contacting in the central zone and thus the traction needed to sustain the calculated bending moment can not be achieved and the slit separation is not as great as that predicted by equation (14). The yield stress of the PET was 55.1 MPa. At the highest web tension the average web stress was 22.5 MPa and thus the web should have been in the elastic range of stress.

Finally a roller 57.2 mm (2.25 in.) in diameter at the centerline with an equivalent tool cut radius of 22.58 m (889 in.) was tested (Parabolic 3 in Table 1). The PET web 23.4 μm (0.00092 in.) in thickness and 0.152m (6 in.) wide was used. The results, shown in Figure 8, display a similar deviation from theory as was shown in Figure 7. Calculating the critical tension from expression (16) yields 22.3N so all but the highest tension are again relatively equal to or below this value. Figures 7 and 8 demonstrate the necessity to provide good web to roller contact in order for the spreading roller to attain the desired results.

Bow Tie Roller

One bow tie roller was tested (refer to Table 1) using the 0.152m (6 in.) wide LDPE web. The results of the tests are shown in Figure 9. Notice that at low tension values the agreement of the data is satisfactory while at higher tension the values do not conform to those predicted. Equation (32) predicts a minimum tension of 0.22 N in order for negative machine direction stress to be avoided. All of the applied tensions are well above this value so that good web to roller contact is assumed. However most of the experimental data fall below the plots calculated by equation (30). As mentioned before only the lowest tension applied resulted in average stresses below the yield stress of this web and thus the linear analysis cannot be expected to perform well.

The 0.152m (6 in.) wide PET material was tested next and the results are shown in Figure 10. At the two highest tensions excellent agreement is shown between theory and data but there are three points, corresponding to the lower tensions and/or the smaller slit span length that fall well below what is predicted. These three cases are indicative of insufficient web/roller traction required to generate the moments needed to steer the web to normal entry. Equation (32) predicts that 4.97 N is needed to initiate contact. As seen from Figure 10 several times this value of tension is needed to assure sufficient contact. In two of the points the slit separation is negative indicating that the roller becomes a gatherer instead of a spreader. In these cases the web to roller traction was not enough to keep the slit edges of the web separated and one edge lapped onto the top of the other producing a "raised" effect inducing motion in the negative direction.

CONCLUSIONS

Closed form expressions have been presented for two types of spreading rollers. The equations can be used in two ways. First, given a parabolic or bow tie spreading roller the designer can calculate the lateral slackness that will be removed by the roller. Second, given the width reduction in the web (lateral slackness) the designer can specify the roller parameters so that a spreading roller can be made to restore the web to the original width. In each case, in order for the spreading roller to have full effect:

- the web must be under sufficient tension to make full contact with the roller
- the web must not be subject to tensions that induce stresses in excess of the yield stress as (a) the web will form baggy edges and (b) the spreading effect is diminished.

As shown by theory and by experiments these devices can be web spreaders or gatherers and must be used with care. It should also be noted that the effect of these devices is very localized and the spreading effect maybe entirely lost after the web has traveled downstream on the order of one web width (3). These devices are destabilizing and as such they should be used only ahead of coating, laminating, or winding processes etc. where the web must lie flat and several concave rollers placed in series will likely result in disaster.

ACKNOWLEDGEMENTS

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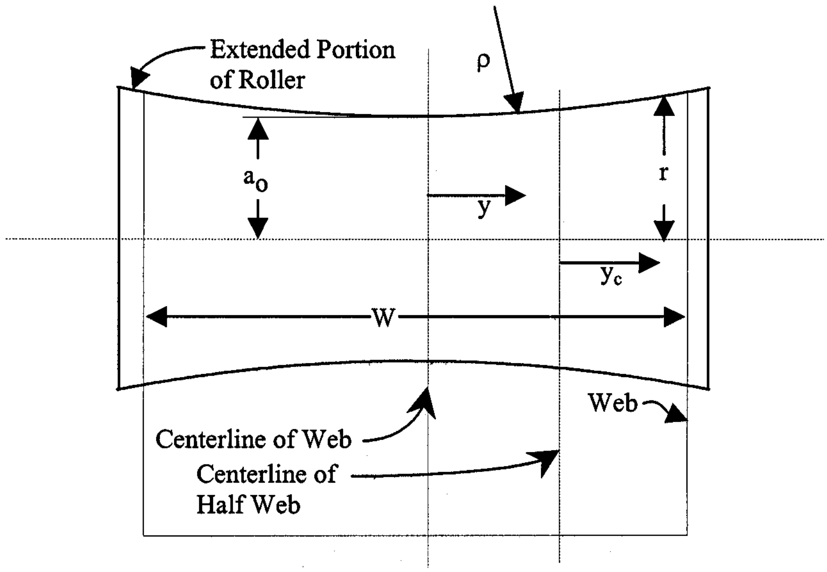

Figure 1 – Geometry of the Parabolic Shaped Roller

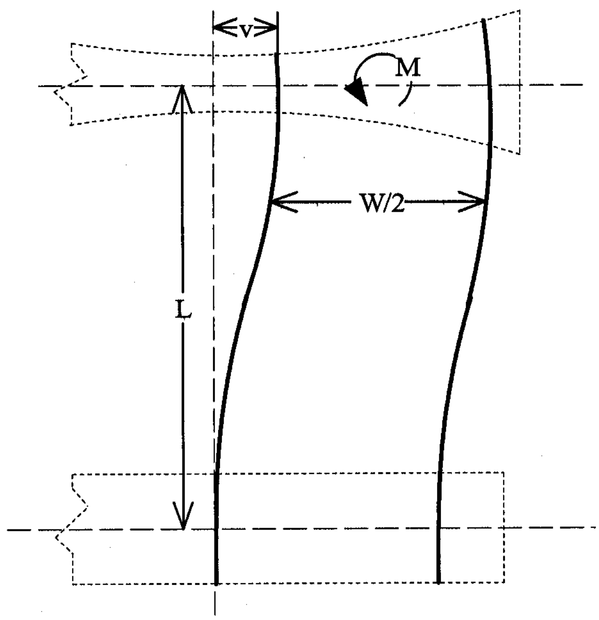

Figure 2 – Half Web Beam Element

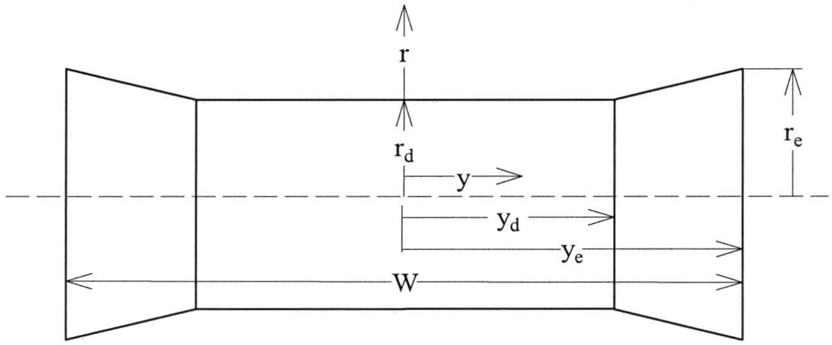

Figure 3 – Geometry of the Bow Tie Shaped Roller

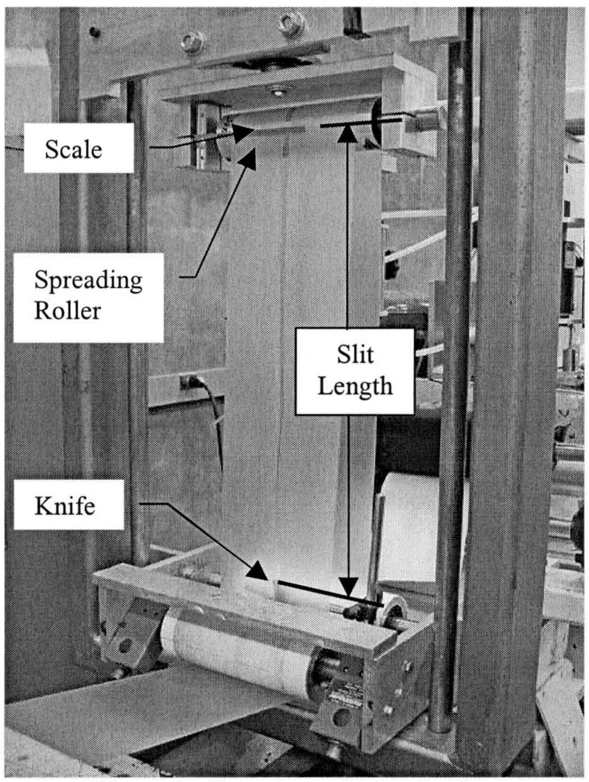

Figure 4 – Experimental Test Section

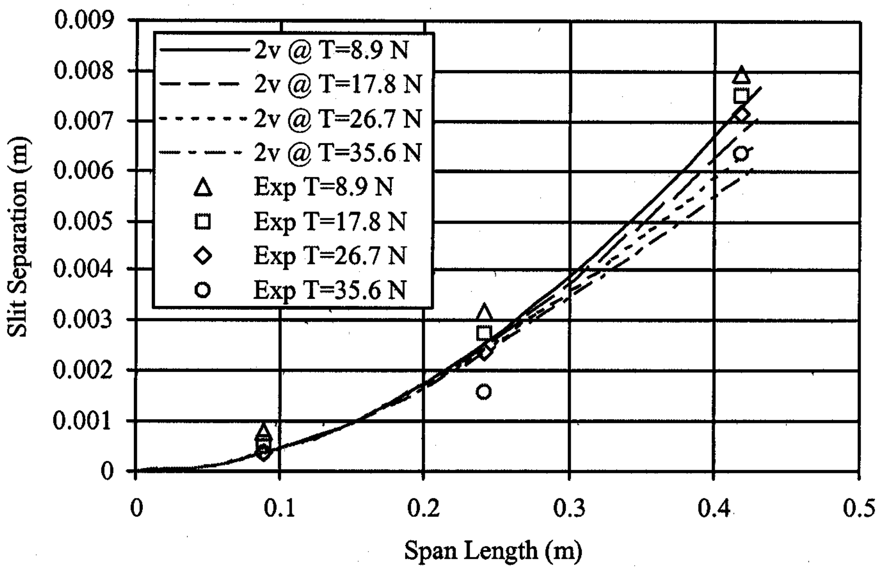

Figure 5 – Parabolic Roller 2, 10.16 m Tool Cut Radius, with 0.152m Wide LDPE Web

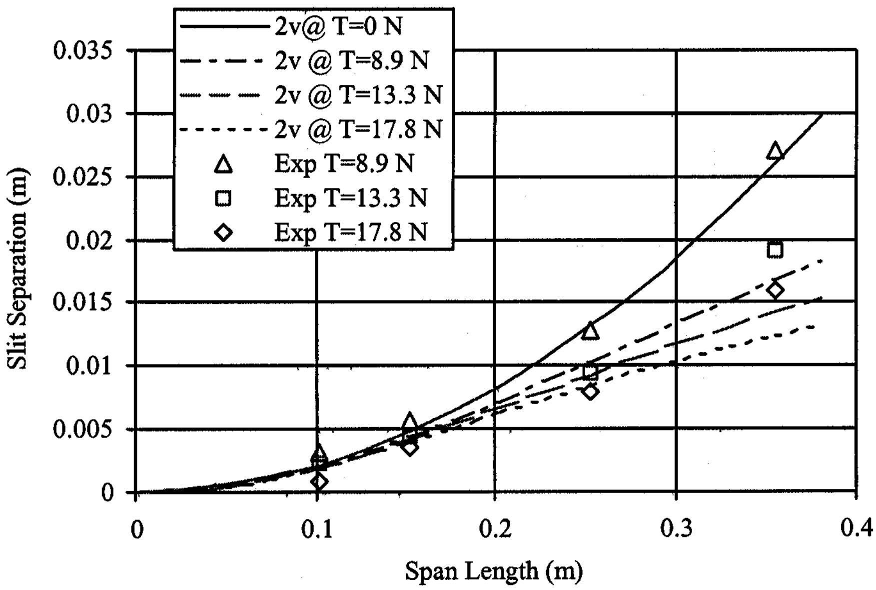

Figure 6 – Parabolic Roller 1, 1.22 m Tool Cut Radius, with 0.073 m Wide LDPE Web

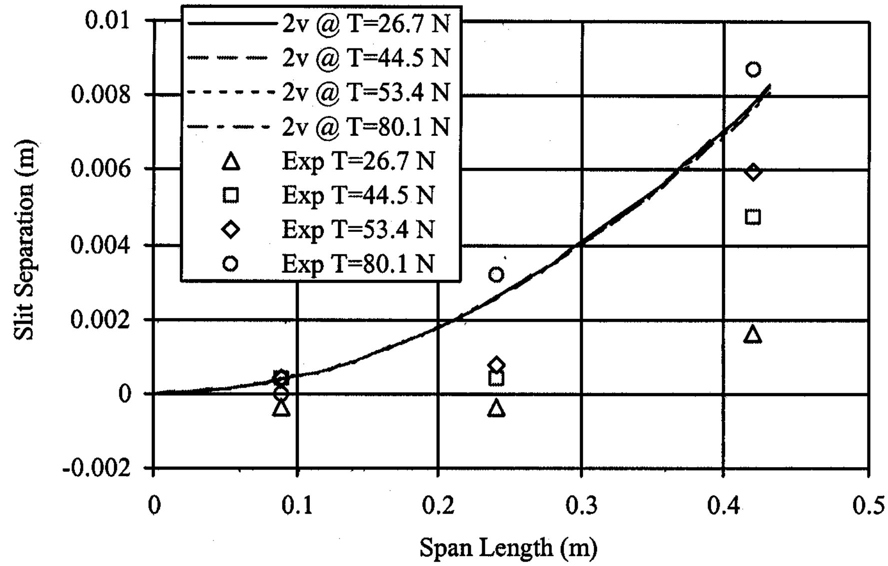

Figure 7 – Parabolic Roller 2, 10.16 m Tool Cut Radius with 0.152m Wide PET Web

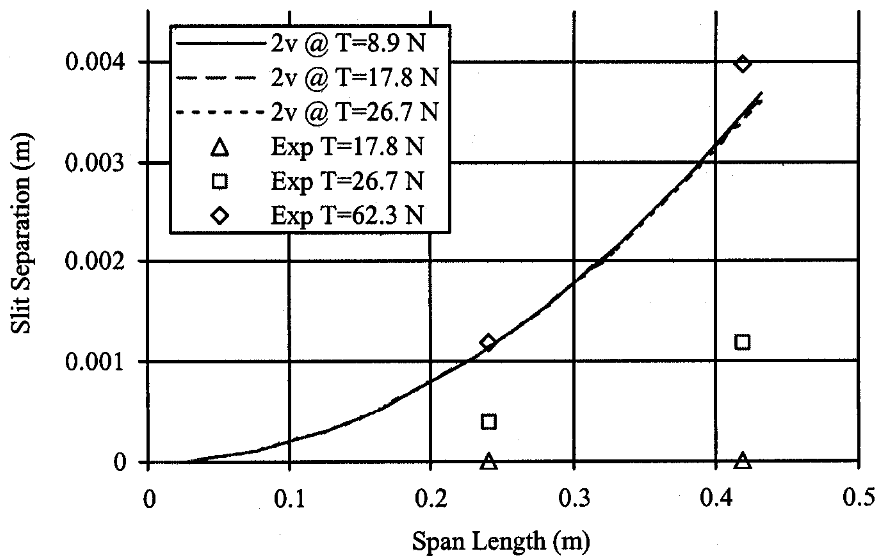

Figure 8 – Parabolic Roller 3, 22.58 m Tool Cut Radius with 0.152m Wide PET Web

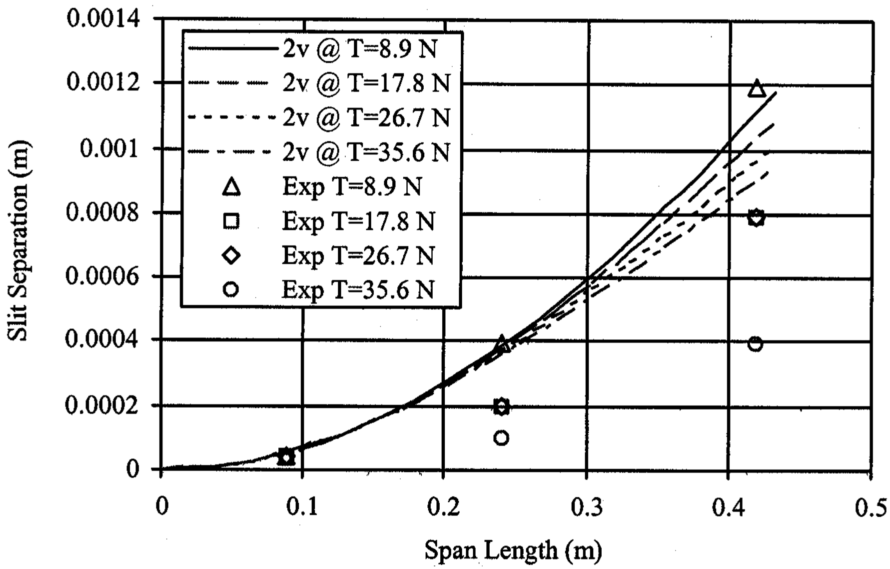

Figure 9 – Bow Tie Roller with 0.152m Wide LDPE Web

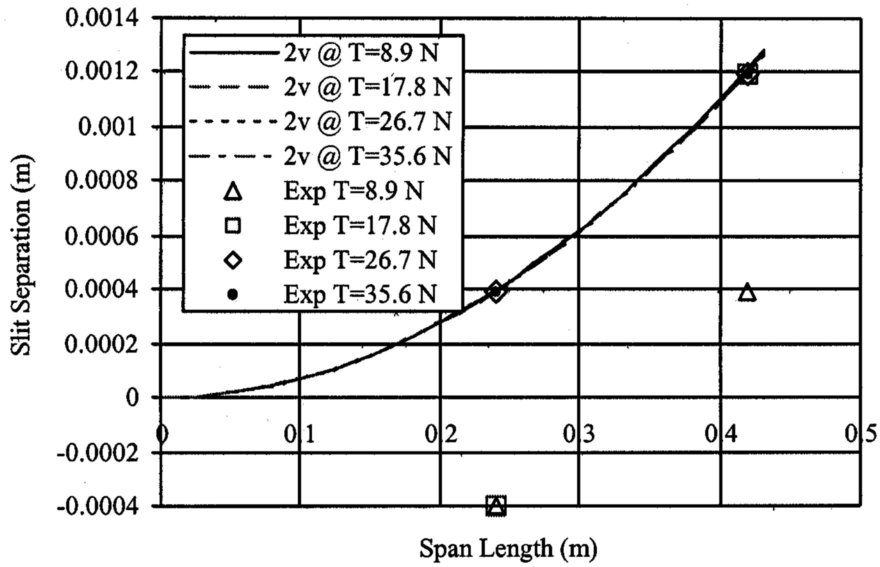

Figure 10 – Bow Tie Roller with 0.152m Wide PET Web

Roller	Eq. Radius, m (in.)	a_0, m (in.)	a_1,1/m (1/in.)	Width, m (in.)
Parabolic 1	1.22 (48)	.02413 (.95)	.410236 (.01042)	.073 (2.88)
Parabolic 2	10.16 (400)	.02776 (1.093)	.0492126 (.00125)	.152 (6.0)
Parabolic 3	22.58 (889)	.02858 (1.125)	.0218976 (.0005562)	.152(6.0)

Roller	r_c	Y_d	r_e	Width (y_e)
Bow Tie	.02822 (1.112)	.0508 (2)	.028302 (1.11425)	.152 (6.0)

Table 1 – Parabolic Roller Properties

Web Material	Caliper	MD Modulus
LDPE	25.4 μm (.001 in.)	165.5 Mpa (24000 psi)
PET	23.4μm (.00092 in.)	4137 Mpa (600000 psi)

Table 2 – Web Material Properties

Name & Affiliation	Question
R. Lucas – GL&V	I have a question on your wrap angle around your spreader. How much wrap angle, what is the criteria for determining how much wrap angle you need in order to have adequate traction to make it effective?
Name & Affiliation	**Answer**
R. Markum – OSU	We did not investigate that. We had 180 degrees in our setup. We didn't look at that in this particular paper.
Name & Affiliation	**Question**
B. Bettendorf – Metso	Have you thought about how your equations will modify when you have in a line no tension profile, not a constant tension, I assume.
Name & Affiliation	**Answer**
R. Markum – OSU	No. We assumed constant tension on the web as it enters the test band.
Name & Affiliation	**Comment**
B. Bettendorf – Metso	Maybe you will, in the future, I hope.
Name & Affiliation	**Question**
G. Homan – Westvaco	If I understood you correctly because of the dramatic difference in the results that if you have a converting line that runs a variety of products, you may want to avoid this type of spreader? Or am I misunderstanding?
Name & Affiliation	**Answer**
R. Markum – OSU	One of the prime reasons for developing models is to allow us to foresee problems before they occur. If you run a variety of products you can use a model such as we have developed to see if one roller configuration will solve all of your spreading needs for your range of webs. This ability depends on your web tension, web/roller traction capability, and the yield stress of your web all of which will vary from web to web. It is not a given that any spreading roll that works on one web will work successfully on the next web. You need to look at each situation because not only do the machine and web material parameters change, but each material may also require different amounts of spreading.
Name & Affiliation	**Additional Statement**
K. Good – OSU	Concave rollers are very effective spreading devices but to some degree all spreaders have their imperfections. You may well find that one contour will solve a large range of your spreading needs just as you might find one bowed roll might serve a large range of your needs. Compare the torque required to turn a bowed roll to a concave roller and you may find that you have to tendency drive the bowed roll. Through time spreaders such as bowed rolls with

	elastomeric covers will degrade and may become a maintenance issue. So in answer to your question, Gary, there is no more reason to avoid the concave roller than any other spreader, in fact it has some advantages.

Name & Affiliation	Comment
B. Feiertag – OSU	In reference to Ron Markum's paper on spreading. You mentioned you might have a baggy lane in the middle of the web. John suggested raising the tension to remove the bagginess. A web with a high Poisson ratio is liable to produce wrinkling in the edge lanes and in that situation the bow tie roll as Ron presented is a good solution. I would, as he mentioned, discourage the idea of simply filling your machine with bow tie rolls. There are strategic places where you can place them occasionally to deal with these problems fairly effectively. Sometimes a mixture of these solutions is preferable so you might combine use of a few concave or bow tie rollers with Dave's suggestion of occasionally using an air floatation device or some such thing as that. Thus more tension may be required to get rid of the length non-uniformity but when combined with some spreading devices to deal with the wrinkling effects that may be associated with the increased tension you will hopefully be able to run the web successfully. That is the challenge, if this were not challenging, it would not be interesting, would it?
Name & Affiliation	**Question**
R. Lucas – GL&V	We have sort of started off with a general subject of spreading, but we really need to divide it up into at least two groups of problems. Are you going to be spreading a single web or are you going to be spreading a multiplicity of slit webs? Because, we could get into totally different discussions depending on which way you chose to go. They have different executions. There are people within this room that would be hard pressed to run a piece of equipment two or three hundred meters per minute and yet there are other people in the room that are running equipment that are going 3000 meters per minute and each of these areas on different webs has their own problems. So it may be a difficult challenge to have a unified discussion here, and we need make a decision. Are we going to talk about slit webs or unslit webs because it really does make a difference?

Name & Affiliation	Answer
K. Cole – Kodak	I would assume the unslit webs tend to be more difficult since you have got a lot more width. They are different challenges. The other way to look at it is that we are running a web that starts wide and becomes narrow, so we are talking about the same problem through the flow. We can talk about both general areas.
Name & Affiliation	**Question**
M. Innala – Metso paper	We can handle both of those areas if we have good mathematical models.
Name & Affiliation	**Answer**
J. Shelton – OSU	I believe, Bob, that the dual bowed rollers that were mentioned today for spreading slit webs should be used for unslit webs also. You have a much more tractable problem, you have geometry instead of a strength in materials problem controlling the spreading. The web should come up perpendicular to the bow and move across to the next bow in either a Z rap or a U rap. The removal of slackness in the unslit web is just the same as the spreading of the slit web. I think that more people should consider dual bowed rollers with the proper configuration, which is similar to a displacement guide. Long bowed rolls have to be driven so this may result in an expensive installation. Another spreading device you might consider that I don't think has been mentioned but that has been used successfully especially with wide webs, is edge nips. Just angled edge nips, which started in the cloth and textile industry, but are used other places.
Name & Affiliation	**Comment**
B. Feiertag – OSU	You can even employ short little nip rollers on a big idler at the edges of the web and that can be quite effective as another technique.
Name & Affiliation	**Question**
K. Cole – Kodak	Does anybody have any comments on weave? We had a few papers on that. How about different air conveyance devices such as flat dryers or loop dryers?
Name & Affiliation	**Comment**
D. Pfeiffer – JDP Innovations	I had an inspiration while Dr. Moretti was presenting the paper on the air flow dryers where there was no place to grab the web there to do an edge guiding function on it. However, if you cut the air plenum system down the middle and you have one pressure on one side and you have a different controllable pressure on the other side, then you could steer the web by steering the amount of air pressure on the left and right sides on the plenum chambers. In that case, if you had the control you would not have to select a low flying web height. As long as it was controllable, you could float the web at a fairly high flying height and have all the advantages that that gives

	you, such as the ability to have higher air flow and more drying affect and so forth. So, that might be a good thing to follow up on if you want to guide a web to through the air flow to oven system.
Name & Affiliation	**Question**
A. Thill – Exxon Mobil	I will tell you that the oven is not intended to guide the web. It is intended to dry the web. If you change the air pressure you are changing the drying.
Name & Affiliation	**Comment**
B. Feiertag – OSU	On that score, and this has been done successfully, it is possible if you are only coating one side to put a steering roll in these air floatation ovens. There are some things you have to be careful about in doing this and if you have interest in this we can discuss this later. My observation on air floatation ovens is that you are pretty safe up to a length to width ratio of the web of about 25 to 1. But I have seen one of these ovens that was 800 feet (244 m) long.
Name & Affiliation	**Question**
G. Homan – Westvaco	I have only attended a couple of the other IWEBs and I cannot say that I have read all the papers that have ever been written on this topic. I am curious as to whether anyone has ever used operator and gear side load cell measurements independently instead of using the total load as a means of identifying any of the defects which may occur on either the operator or gear side of the web. This would seem to be a way to locate bagginess or some other defect. Would this clearly indicate that there is a problem or not?
Name & Affiliation	**Answer**
A. Thill – Exxon Mobil	It is becoming standard in some of the big lines now. You have a reading from both load cells and the tension control reacts to the sum.
Name & Affiliation	**Answer**
J. Dobbs – 3M	The one thing you really will measure is how well centered your web is on that roller.
Name & Affiliation	**Question**
A. Thill – Exxon Mobile	I suppose if you build a machine, you always center the web on the roll. If, for example, you have a bowed roller with the web off center you will steer the web further off center.
Name & Affiliation	**Comment**
J. Dobbs – 3M	I agree that is good in principle, but it does not always work that way.

Name & Affiliation	Comment
A. Thill – Exxon Mobil	Then you better straighten out your equipment.
Name & Affiliation	**Comment**
G. Homan – Westvaco	I have seen some converting operations where the process is running well and then they splice in a new roll. All of a sudden things start to go awry but sometimes it is much later before you begin to figure out that there is a problem and identifying what the problem is. This is something that should have been brought out earlier regarding paper, that unfortunately as you get close to the edge of the deckel your mechanical properties may start rolling off. This can be determined if you collect enough data but if you monitor the operator and gear side load cell signals then maybe that an immediate indication. If you can get rapid feedback to the operator he has the opportunity to take some other action as opposed to waiting until you run into some serious problems and the operator never knows why.
Name & Affiliation	**Question**
R. Lucas – GL&V	There are a number of companies who supply equipment to the paper industry. I can only speak my company, but I am sure that other suppliers would have similar capability. We have slitter sections that have very narrow draws that enable us to use sectional rolls and if we have a very wide machine we have all these sectional rolls and opportunity to install load cells beneath intermediate bearing housings all the way across the machine. It may be a matter of economy as to how many load cells we install. When there are problems running a web that require going in and taking a look for a solution it ends up being a resource for us to take a look at the load distribution from one load cell to the next which may give us a clue that the web is running tight or whatever. So the load cells end up serving a dual function by being a diagnostic tool as well as a feedback for web tension control. I do not know if anybody is using them to take any dynamic corrective action. There have been occasions when people have had rider roll systems installed on winders that have had load cells underneath intermediate bearing housings. This was done to make the geometry of the rider roll change in order to make the nip load uniform beneath the rider roll as part of an ongoing process control. I mentioned this to let you know that there are some things that have been done that are looking at the distribution of nip load across the machine.

Name & Affiliation	Answer
B. Feiertag – OSU	There are several rolls like that on the market and they are fairly widely used in the metals industry. They do control some of the functions on a rolling mill as a result of the data that comes off of those types of rolls.
Name & Affiliation	**Question**
K. Shin – KonKuk University	I have carried out some studies on monitoring and diagnostics on web handling systems. One example was by using the tension signal I could exactly designate the location of the defective roller and also the degree of defect of roller. I carried out the experiment and the logic I found was very accurate. I think we need to develop some kind of system to monitor and diagnose the web handling component or system. Thus you can have an operator predict which component inducing a defect and also you can develop a tolerance system to notify you when you need to change a component due to a defect. You may be able to deal with the defect by using a control action. I think it is time for us to think about these kinds of systems for the web process lines.
Name & Affiliation	**Answer**
J. Shelton – OSU	With viscoelastic materials, Bob (Lucas), the goal is not a uniform nip force. If you have a tapered web the goal is a cylindrical roll. If you make a tapered roll with uniform nip force then a cambered web results and that is not what you want. I picture a rider roll as a device to try make a cylindrical roll out of a web that does not really want to make a cylindrical roll. You attempt to fill in the valleys with air and push it off the mountains to try to wind a cylindrical roll. This is the goal for materials that are viscoelastic to avoid making bagging lanes and cambered webs out of a web that might otherwise be acceptable.
Name & Affiliation	**Question**
D. Pfeiffer – JDP Innovations	I agree John. That is one reason that nip winders or winders with a drum nip, are quite affective in making cylindrical rolls because over those high spots the nip loads will be higher which results in more nip-induced-tension locally, maybe not making a cylindrical roll, but tending in that direction. If you take paper with a bad thickness profile across the width and center wind it without a nip then you get a really bad profile, a non-cylindrical roll, guiding problems and wrinkles.

Name & Affiliation	Answer
J. Shelton – OSU	I had an experience with a man who was supervisor of slitting involving plastics. An operator had taken the pipes connecting the two nip arms out and was winding a beautiful tapered roll. You probably could not visually see the taper, but a conical roll was produced. The supervisor told the operator: "Do not do that, you can make a pretty roll but we will get it back, it will not run flat on the printing press because one side will be stretched relative to the other side."
Name & Affiliation	**Question**
T. Walker – TJWA	I just wanted to say how it has really been neat to come to the IWEB conferences and to see how this sort of technology and this group of people have evolved over a decade or more. The staff here at WHRC, the people sitting in this crowd, several of which may have attended every single IWEB if not most of them, and the people that get out of their seat and get up and do a presentation make this a special group of people. I think it is a combination of the people who delve in theory and the people that are problem solving driven who are obviously talking about problems they had at work or the application of a theory that has been developed. Then there are people that are just the observers, not the model type, but the ones that come in and say, here is what I'm observing in a web line, can you please help me out? And I think this combination is just a neat group that has formed, and you do not necessarily see the end of it. You sort of wonder, in web handling, oh, it's sort of limited little world, maybe after ten years we'll have a handle on most things and we'll taper off what we have to do, but I don't think the list of projects to be done here has stopped either, so I guess I wanted to say thanks and show my appreciation to the people that run this show, the people that come here and attend it, and the people that present, so thank you.

Name & Affiliation	Answer
K. Cole – Kodak	Thank you, Tim. I agree. The comment I would make is everyone is different. Everyone has their unique capability, and I tend to be a little bit more of a modeler, but it really helps to hear experiences from people that are out on equipment and observe problems because just thinking about winding modeling, you can go off on all sorts of tangents with winding modeling. In order to do something that useful you really have to have a good sense for the problems, that's where the experience comes in and that's where people with years of interaction with these types of problems is very invaluable. I think it's coming together, and I agree with what you said, Tim. I think our group is doing well here and we're going in the right direction.
Name & Affiliation	**Question**
P. Bourgin – Ecole Supérieure de Plastirgic	I would like to make additional comments on the need for better understanding of the key elements to improve technology, I think that the both are combined. I would like to give two examples, two challenging problems. I observed that there were two papers dealing with rolling of the nip roll. For instance, I don't know if someone here would be able to answer the question, I have some PET film to be wound, 6 meters wide, what kind of a nip roller should I have? Large diameter, small diameter, soft, hard, which kind of coating, rubber, polyurethane? It means that the functions of the nip roll should be analyzed in order to try to find the answers. Secondly, I came here with a sample of paper, just a fold of paper, or a sample of plastic film. I would like to know if it's possible to run the machine one thousand meters per minute, six meters wide. If I have a pilot line 10 cm wide, can I be sure that I can extrapolate the results to production scale. So, first, I think it's very important to characterize, to come back to the basic properties of the web being wound. In the case of paper, I think it's necessary to know the permeability factor. In the case of plastic film, it is necessary to characterize, not by Ra or Rt, but other parameters to calculate the surface topography, then to include the surface topography description into a comprehensive model.

Name & Affiliation	Answer
K. Cole – Kodak	Obviously, it depends on what your failure modes are and what the problems are that you're concerned about, as to how you answer your first question. So, I don't think we have time to go into all the details, but, there are factors that go into how to optimize pressure roller winding. It depends, like you said, on the film. The properties and so forth.
Name & Affiliation	**Question**
K. Good – OSU	We have internal wound roll defects that we need to worry about. We may have wrinkles coming into the winder and we do not like that either. So, if you were looking entirely from the wrinkling standpoint, you might say I desire a large diameter nip roller here with a large air film so that wrinkles can spread out. This is the John Shelton "big slick roller," I believe John coined that. But, there are many dynamicists sitting amongst us here that would say that nip roll mass is increasing when you increased the nip diameter and now how well is that nip roll going to be following the wound roll at 4000 meters per minute. It may be hitting the wound roll once or twice per revolution at best. So, it becomes a multidimensional problem at that point. Not only winding or wrinkling, it is all interwoven. What about the rubber covers on nips and the fact that rubber speeds up when it is constricted? Do you want additional wound-in-tension in your polyester roll? In many cases you do not because you are afraid of having too much pressure in the roll to begin with. So, these are all multivariable problems.
Name & Affiliation	**Answer**
P. Bourgin – Ecole Supérieure de Plastirgic	I completely agree with Keith's comments. They are based on the analysis of the functions you want to have with your nip roll. I think this is the key way to answer the question. What are the functions of the device I want to operate? Since it's a multivariable problem, a multifunction problem, you have to make a choice. But, you have to be helped by some basic understanding of the fundamentals involved. Spreading issues, winding, tension control, traction and wrinkling, things like that. Then you can make knowledgeable decisions. Problems of weight, problems of vibration induced by too big of a roll, things like that also. I think we have to integrate all these.
Name & Affiliation	**Question**
K. Cole – Kodak	Thank you very much for your comments…