APPLYING HUNDREDS OF YEARS OF SCIENTIFIC AND ENGINEERING KNOWLEDGE IN 50 YEARS OF TIME

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ABSTRACT

Aspects of lateral behavior discussed in this paper include (1) the concept of a web span as a tensioned beam, analyzed for large ratios of L/W, small ratios, and general ratios, and (2) static interaction of two spans and the bistable "jump" phenomenon.

Aspects of transport of a web which are discussed are (1) the importance of J/R^2 for a roller, not the mass moment of inertia J by itself, (2) dimensionless energy parameters which determine the effectiveness of dancers and other elements of transport as attenuators of disturbances to tension, and (3) principles of design of rollers, such as the need for uniform wall thickness and the need for large, thin-walled rollers.

NOMENCLATURE

a	linear acceleration of the web or matching acceleration of the surface of a roller
$C_1, C_2, \ldots$	coefficients of a general solution of the differential equation of a web span
E	x-direction modulus of elasticity of a web (Young's modulus)
f	distributed force per unit width of a web wrapping a roller (force per unit length on the roller)
G	modulus of elasticity in shear
h_0	thickness of the film of air entrained between a web and a roller, dead bar, or winding roll
I	area moment of inertia (bending stiffness)
J	mass moment of inertia of a roller (rotational)
K	$\sqrt{12\varepsilon} / W$ (for a web)
K_b	bearing torque constant, related to size and type of bearing
K_e	$K/[1+n(E/G)\varepsilon]$ (for a web)
L	free length of a web span between rollers
L_B	length between centers of roller-support bearings
L_R	length of the face of a roller

L_T	total length of web between drive rollers
M	mass of a roller –kg
M	moment on a normal section of a web span
m	mass per unit length of a web
N	normal shear (perpendicular to the deflected axis of the web, not necessarily parallel to the y axis)
n	shear factor for short spans of a web (usually 1.17 for homogeneous, isotropic webs)
n	force–newtons
R	outside radii of a roller
T	Tensile force in the web–newtons
t	thickness of the web
U	algebraic sum of the velocities of the web and a roller, dead bar, or surface of a winding roll
V	velocity of the web
V_i	"initial", or steady-state velocity of the web
W	width of the web at operating tension
x	machine-direction distance along the undisturbed direction of a web span
y	distance of a disturbed or guided web from the x axis
y'	dy/dx, the slope of a disturbed web
y''	d^2y/dx^2, related to the curvature or moment in the web
y'''	d^3y/dx^3, related to the normal shear in the web
y^{IV}	d^4y/dx^4, related to the side loading of a web (usually zero)
α	angular acceleration–radians/second
ε	x-direction strain (T/tWE)
θ_r	angle of misalignment of a roller–radians
μ	viscosity of entrained air
ν	Poisson's ratio
π	3.1416
ρ	density (mass per unit volume)
τ	torque
ω	rotational velocity of a roller–radians/second

INTRODUCTION

The vast reservoir of scientific and engineering knowledge which was established from the early years of the Industrial Revolution through recent years would be expected to greatly reduce the time required for quantitative understanding of a new discipline, such as web handling. Myths, misunderstandings, and misapplied analogies and slogans, however, have impeded advancement of knowledge of web handling. The older web-handling industries, notably cold-rolling and processing of sheet metal and continuous making of paper, have advanced primarily by the slow process of trial and error.

Modeling of winding was not successful until the speed and versatility of computers had reached the level achieved about forty years ago; furthermore, general methods of testing of required winding parameters and routine calculating of results have been established only in recent years.

This paper, while not probing into technical depth, summarizes a few advances in understanding of web handling in which the author has contributed in his 48 years of experience in web handling, but does not discuss other aspects of web handling, such as winding, interaction between a web and an air-flotation device, viscoelastic effects, or effects of other nonideal or nonconventional characteristics.

A WEB SPAN AS A TENSIONED BEAM

Prior to the Shelton thesis [1], a tensioned span of a web had been assumed to behave as a string, making sharp angles in leaving one roller and aligning itself in perpendicularity to the next roller. Campbell [2] thus wrote an equation for the response of a web as a string to an upstream disturbance. Sorsen [3] modeled a web as a string to describe the action of a web guide, but the resulting equation described the action of a pivoting guide roller as an integration instead of the correct (as modeled) first-order lag.

Problems with power transmission in industry attracted much experimental and analytical engineering activity in the latter part of the nineteenth century through World War II; however, the level of understanding achieved in power transmission did not migrate to web-handling industries at that time. Examples are (1) Osborne Reynolds [4] described in 1874 the inevitable creep of belts which transport power (or webs with a difference in tension across a roller), (2) Swift [5] in 1928 completed the derivation of the capstan equation for a belt (or web), and (3) Swift [6] in 1932 studied, analytically and experimentally, the crowning of pulleys required for operation with imperfect belts, and with pulleys which were misaligned and imperfect. In the latter analysis, Swift solved a second-order differential equation of lateral behavior of a belt, considered to be a slender tensioned beam, in terms of sinh KL and cosh KL. For large values of L/W, the analysis by Shelton [1] (36 years later) likewise expressed variables in terms of sinh KL and cosh KL, but the use of a fourth-order differential equation broadened the range of application and generality of the analysis (to interacting spans, for example).

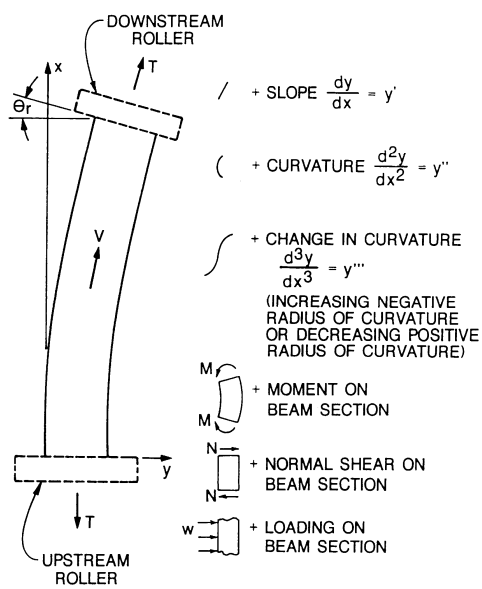

Figure 1 – Sign Conventions for Beam Analysis

The greatest need for the analysis of the Shelton thesis was guiding of steel strip, commonly with long spans of imperfect webs which have moduli of elasticity, in most cases, more than 50 times greater than non-metal webs. For long spans of steel webs, the neglect of lateral bending stiffness by previous analysts became ludicrous; furthermore, a long web span of nearly any material or degree of anisotropy was observed to form a gentle curve between nonparallel rollers, as sketched in Figure 1, with most of the curvature occurring near the upstream end (with the steady-state moment and curvature at the downstream end found to be zero in the thesis research).

The Shelton thesis applied the methods and sign conventions which Timoshenko and Gere [7] used for analysis of beams under both axial compression and lateral loading to analysis of a tensioned web. For large values of L/W, the differential equation for a free span of web with no distributed lateral force was found to be

$$\frac{d^4y}{dx^4} - K^2\frac{d^2y}{dx^2} = 0\,, \tag{\{1\}}$$

wherein

$$K = \frac{\sqrt{12\varepsilon}}{W}\,. \tag{\{2\}}$$

A general solution to equation (1) for either static or dynamic behavior is

$$y = C_1 \sinh Kx + C_2 \cosh Kx + C_3 x + C_4\,. \tag{\{3\}}$$

The coefficients C_1 through C_4 for analyses of specific problems are determined by substitution of four appropriate independent boundary conditions.

The above technique of analysis of a tensioned span of a web is accurate only if L/W is large, as with ordinary beam theory in analysis of beams with no axial stress. Comparisons of results obtained with modified methods indicate that, for engineering purposes, accuracy is usually adequate if L/W of the specific span is greater than 3.5.

Timoshenko and Gere [7] presented a modified method of analysis which is precise for values of L/W from very small to very large, and the Shelton thesis adapted this method to a tensioned span of a web. The primary modification of the method expressed in equations {1}, {2}, and {3} is the replacement of K with an "equivalent" parameter K_e:

$$K_e = \frac{1}{W}\sqrt{\frac{12\varepsilon}{1+n(E/G)\varepsilon}}\,, \tag{\{4\}}$$

where E/G, for homogeneous, isotropic materials, is related to Poisson's ratio ν (which can vary only between 0.0 and 0.5 for homogeneous, isotropic materials) as

$$E/G = 2(1+\nu), \tag{\{5\}}$$

and n varies little from 1.17 as a function of ν for most isotropic materials. For anisotropic web materials or web structures (such as woven or nonwoven fiber webs), E/G needs to be determined with a web sample under tension in a laboratory. The use of K_e modifies the differential equation and its solution only by its substitution for K:

$$\frac{d^4y}{dx^4} - K_e^2 \frac{d^2y}{dx^2} = 0 \qquad \{6\}$$

and

$$y = C_1 \sinh K_e x + C_2 \cosh K_e x + C_3 x + C_4 . \qquad \{7\}$$

Except for the multiple spans in accumulators and spans in drying ovens, values of L/W are commonly less than 3.5. Analysis of interaction and interconnection of multiple spans has proven so complicated that simplification for such values of L/W has proven beneficial since the development of short-span theory by Shelton [8]. For the usual case of no distributed lateral force, the differential equation is

$$\frac{d^4y}{dx^4} = 0 \qquad \{8\}$$

and the general solution is

$$y = C_1 x^3 + C_2 x^2 + C_3 x + C_4 . \qquad \{9\}$$

The coefficients again must be determined for each type of problem by substitution of four independent boundary conditions, then solving the resulting four equations.

Solutions by the short-span theory neglect the effects of tension on the lateral deflection of the web, while the long-span theory expressed in equations {1}, {2}, and {3} neglects the effects of shear on the lateral deflection. The "Timoshenko" beam of equations {4}, {6}, and {7} considers effects on deflection of both tensile and shearing stresses.

Development of beam theory, which allowed reasonably simple and accurate calculations for complex structures, was a steady progression, particularly from the time of Euler (mid-eighteenth century) to early in the twentieth century. Much of this development was simplification of the unwieldy theory of elasticity, and development of methods of breaking a large problem into small parts. Similarly, the present level of understanding of the lateral behavior of a web resulted primarily from simplification of problems and generalization of the results.

Beam theory applies methods of handling many "concentrated" loads with single or multiple supports, where the reaction forces are also usually assumed to be concentrated. Web handling theory requires the solution of a separate differential equation for each span, if the spans are interacting. For the inevitable case of interconnected spans, where the transient lateral conditions at the downstream end of one span are transported across the roller to the upstream end of the next span, Shelton [8] applied methods of control theory to relationships of amplitude ratios and phase angles in response to a sinusoidal upstream disturbance.

An advantage to the solution of the differential equation of a tensioned span as a continuous function of distance along the span is that conditions can be determined at any point within the span, such as the lateral location of the web at the location of a sensor for guiding. Also, slope (dy/dx), moment (related to the second derivative), and normal shear force (related to the third derivative) can be readily determined at any point along the span. A further advantage of keeping the analysis in terms of mathematical functions instead of resorting to numerical methods, such as finite element analysis, is that derivatives can be safely used for determination of best and worst conditions of a given variable.

Interaction of Two Web Spans

The interaction of two web spans was recognized as an important problem approximately 45 years ago, as a steering guide for steel strip in a California mill worsened the upstream error which the guide was attempting to correct. As a historical footnote, previous guiding of metal webs had been limited almost exclusively to lateral shifting of unwinders and rewinders, a daunting task in light of masses of combined rolls and stands (with motors for winding or braking) often exceeding 50,000 kg. Bruce Feiertag, Chief Engineer of Fife Corporation at that time, confirmed the behavior of the web and supervised the successful moving of the steering guide to a location with more entering span. John Shelton was also an employee of Fife at that time, and was devoting considerable analytical and experimental effort toward understanding lateral behavior and control of a web. Collaboration between Feiertag and Shelton resulted in a qualitative understanding of the positive-feedback phenomenon of a steering guide with (1) inadequate length of entering span, (2) a long pre-entering span, and (3) inadequate isolation of tension, allowing the nonuniform distribution of the entering span (caused by lateral bending of the web in the entering span of the guide) to disturb the distribution of tension in the pre-entering span. The early understanding of the cause of interaction proved to be correct: Even though the web at the entering roller slips forward relative to the roller on one side and backward on the other side, the friction near the center of the web remains sufficient to provide the relatively low lateral force necessary to force the web at the slipping roller to be nearly perpendicular to the roller.

The concept of interaction is sketched in Figure 2, and an application of the concept is illustrated in a troublesome bowed roller installation in Figure 3. Figure 4 shows an installation of tandem bowed rollers for spreading a web, or separation of slit webs on a two-drum winder, with no chance of interaction of spans because of the absence of bending moments in the individual slit webs.

After Shelton completed his thesis research relating to non-interacting spans, the method was applied to two interacting spans, requiring eight boundary conditions for finding the eight coefficients of the elastic curves of the two spans. This analysis assumed (without justification) different values of coefficients of friction for lateral slippage and rotational slippage. Despite the nebulous basis for defining the friction at the slipping roller, a computer analysis usually prevented failure of guiding installations by Fife, while allowing installations with interaction which was less than catastrophic, even where an entering span was so short that, in combination with low friction on the slipping roller, the pre-entering span behaved almost as part of the entering span.

As part of a Web Handling Research Center project in 2002, Shelton attacked the problem of characterization of the sharing of frictional forces between the lateral slippage and the rotational slippage within the contact patch between the web and the roller. The basis of this analysis was the fact that frictional forces on each differential area are opposite the direction of relative motion at that point. Solution required the introduction of a new dependent variable, the velocity slippage ratio K_s. Each equation for K_s (for two short spans, for two long spans, and for a short entering span and a long pre-entering span) contained the eight coefficients C_{A1}, C_{A2}, …, C_{B4} as governed by the boundary conditions. In the eight equations specifying the eight initially unknown coefficients, the

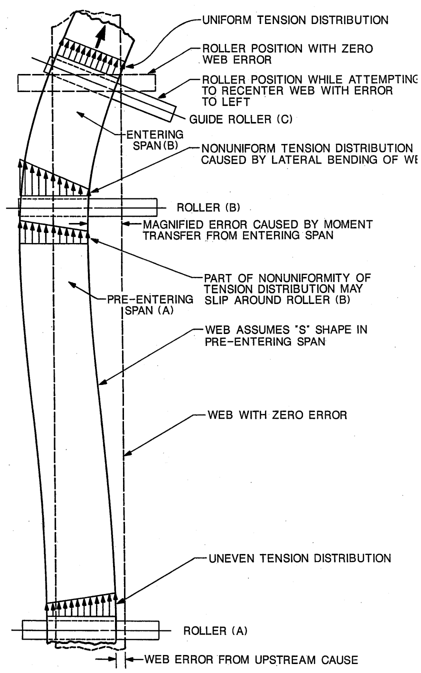

Figure 2 – Pre-Entering Span Steering
Caused by a Web Guide
(Long Pre-Entering Span)

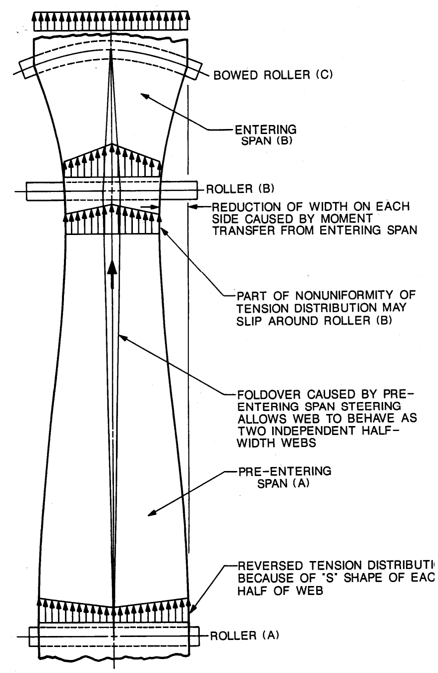

Figure 3 – Pre-Entering Span Foldover
Caused by a Bowed Roller
(Long Pre-Entering Span)

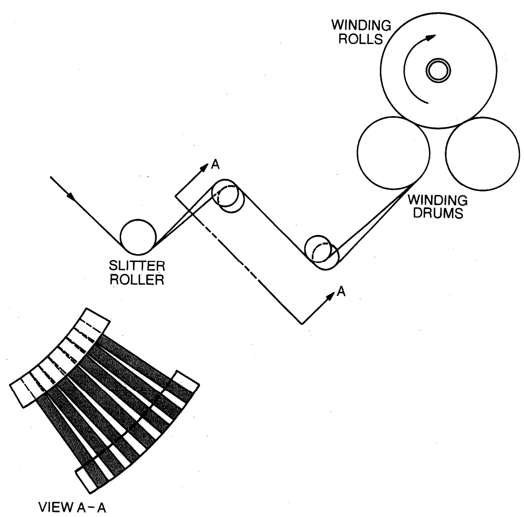

Figure 4 – Separation of Slit Webs on Two-Drum Winder
with Tandem Bowed Rollers

lateral force F_r and the frictional moment M_r at the slipping roller were incorporated as if they were known, with their values calculated from the (known or assumed) independent variables and an assumed value of K_s, with this value of K_s based on experience from previous calculations. A new value of K_s was then calculated from the newly calculated eight coefficients and the closed-form calculations of F_r and M_r, then this new value of K_s was substituted for recalculation. The process of resubstitution of K_s was repeated until the results converged.

No problem of convergence was discovered until January, 2007, when calculations for a very short entering span relative to the pre-entering span resulted in nonconvergence, with nonvalid negative calculated values of K_s occurring with very small changes of K_s from an assumed value of K_s which produced a positive value. The nature of this convergence problem, whether mathematical, physical, or computer modeling, has not been determined; however, such installations should be avoided because of well-known severe problems with interaction. A very short span should not be preceded by a very long span unless isolation is guaranteed by adequate precision of alignment of the roller downstream from the short span and adequate friction of the intermediate roller for isolation of the two spans; also, a short entering span (to the next upstream gripping roller) should never be attempted with a steering guide because of bending stresses and/or shearing distortion.

The work by Shelton [9] was reported in condensed form in the 2005 IWEB proceedings, with the first documentation of the phenomenon of lateral "jump" (based solely on the analysis), the occurrence at the condition of borderline interaction of a different state of operating conditions (lateral position, slope, moment, and lateral force) with an infinitesimal change in the group of independent variables $\theta_r Et/f\mu$. This behavior has been verbally reported from experiments by a WHRC sponsor, but data is not available for publication.

The recognition of interaction of spans as a serious source of lateral disturbances, or an aggravation of existing disturbances, led to further study by Shelton [9] of simple methods of avoidance of interaction, including a specification of precision of alignment of rollers.

Best and Worst Ratios of L/W

Either the "Timoshenko" theory for spans of general length or the Shelton short-span theory can sometimes be used to find the best or worst ratio of L/W for particular desirable results, or for undesirable results to be avoided. The best ratio from one standpoint may be identical to the worst ratio from another standpoint. Those cases of best and worst ratios were not discovered by the author until approximately 30 years after the writing of the Shelton thesis, even though the foundation lay in the "Timoshenko" analysis of the thesis; for example, page 65 of the thesis shows a graph of the "curvature factor" for a web with a misaligned downstream roller, as a function of nT/AG (equal to $n\varepsilon E/G$) and K_eL, which is a function also of L/W. The curves of this Figure show minimum curvature factors at values of K_eL from approximately 0.6 to 1.8.

Although some best and worst ratios of L/W were first discovered by using the "Timoshenko" theory, they were all within the range (of L/W) of reasonable accuracy of the short-span theory, hence are only weakly affected by strain (tension in a given web). For webs which are not greatly anisotropic or inhomogeneous:

(1) The worst ratio for edge slackness as caused by a misaligned upstream or downstream roller is approximately 0.7.
(2) The worst ratio for buckling (potential wrinkling) as caused by a misaligned upstream or downstream roller is approximately 0.4.

(3) The worst ratio of L/W for the entering span of a tapered roller or tapered winding roll for buckling is approximately 0.7; that is, buckling or wrinkling would occur at the smallest angle of taper if L/W were approximately 0.7.
(4) The best ratio of L/W for the entering span of a tapered roller or tapered winding roll for preventing the upstream transfer of the moment caused by the taper is approximately 0.7, for which ratio (with a slight variation with changes in E/G) the moment at the upstream end of the span is zero.

The above four examples of best and worst ratios of L/W will rarely influence design or modification of machinery, because of the multitude of other problems which must be considered. The primary benefit from knowledge of the first three ratios is to counteract the intuitive feeling that buckling and edge slackness (or an uneven distribution of tension for lesser taper or misalignment) would progressively worsen as the length of the pertinent span is decreased. The reason that the buckling and tension distribution improves as short spans are made shorter is that the amount of steering decreases as the span is shortened.

The worst ratio for buckling as the web feeds onto an inadvertently tapered winding roll has been avoided for many years by using a proximity roller (a roller which is servo-controlled to maintain a very short span as the web feeds onto the winding roll and as the winding roll grows). Taper of a winding roll may not be nearly constant across the winding roll, but may be local because of the buildup of increased thickness of a band of printing, for example, where the taper would be a transition between the smaller and larger portions of the roll. Winding of a cylindrical roll by means of a contact roller which is constrained for parallelism with the core, with entrained air filling voids between layers, should be practiced if possible, but porous webs may not trap air effectively, and contact rollers are not always acceptable.

EFFECTIVE INERTIA J/R^2 OF A WEB-HANDLING ROLLER; DIAMETER AND WALL THICKNESS OF ROLLERS

For most web-handling rollers, maximizing acceleration is not a high priority, as most web processes must be run at a constant velocity because of the many cases of limited response of elements of the process. Even starts and completions of rolls which feed into and out of processes are usually accomplished at full speed. Furthermore, slitters and some other machines which are stopped and started frequently may have their acceleration rates limited by the inertia of a fully wound roll. However, zero-speed splicing at unwinders and zero-speed cutting and unloading at winders are prevalent in several industries, notably in processing of metal strip, processing of tire-cord fabric, papermaking, and in some cases of processing of rolls of paper. Intermittent motion of webs in other operations (besides the movie-projection process) may be necessary. Wherever a web accelerates and decelerates, the inertia of idlers (if present) demands careful attention.

One of the simplest dynamic relationships relates angular acceleration α, torque τ, and rotational inertia J as

$$\tau = J\alpha \,, \qquad \{10\}$$

where the angular acceleration could be expressed in units of radians/sec^2, τ in newton-cm, and J in kg-cm^2, if the right-hand side of equation {10} is divided by 100 cm/m to convert to the mks system.

Unfortunately, equation {10} does not relate the proper variables for optimizing the acceleration of a web as it drives idlers while avoiding slackness, excessive tension, or (in some cases) slippage. Rather, the surface of the roller, not the angular rotation, needs to be accelerated by an actual tension difference ΔT, not an abstract torque. These relationships are

$$\alpha = \frac{a}{R} \tag{11}$$

and

$$\tau = (\Delta T)R\,, \tag{12}$$

where *a* is the linear acceleration of the web and matching acceleration of the surface of the roller (in the absence of slippage). Substitution of equations {11} and {12} into equation {10} and rearrangement results in

$$\Delta T = \frac{J}{R^2} a\,, \tag{13}$$

as an appropriate replacement of equation {10} for a roller which is accelerated by the web.

Equation {13} shows that the effective inertia of an idler (relating the acceleration *a* of the surface to the tension difference ΔT across the roller) is J/R^2, not the rotational inertia J alone. This simple equation also provides insight into the reason that analyses of accumulators, dancers, and other components involving acceleration of web-driven rollers always contain J/R^2 in combination.

The term J/R^2 may be important for drive rollers as well as web-driven rollers, leaving little argument (except initial cost) against usage of large rollers, which are advantageous in many ways. The argument for $(1/R^2)$ as a modifier of J for a driven roller (if rotational inertia is an issue, as for a tension-control roller) follows: Inertia is reflected through a gearbox (or other means of reduction) as the square of the ratio; hence, if the gearboxes are chosen for the same maximum web speed regardless of the size of the driven roller, the effective inertia is divided by R^2.

Example of Improvement of J/R^2 by Using Larger Rollers. As an example of the benefit of large rollers, consider a fictitious but representative case of handling tire cord fabric, wherein an accumulator was equipped with solid steel idler rollers of 12 cm diameter. The length L_B (between bearing centers) of the rollers was 180 cm, and the nominal width of the web was W = 140 cm. The tension T was 2000 n (450 lb_f). Neckdown was a problem, with 30 spans in the accumulator. The length L of the spans varied from 1.0 meter for an empty accumulator to 5.0 meters for a full accumulator. The subscript (1) will be used for the original design, and (2) for an improved design utilizing larger rollers. All rollers are steel, with $\rho = 0.00783$ kg/cm^3 and $E = 2.0(10)^7$ n/cm^2. Calculations are:

$$I_1 = (\pi/64)(12\text{ cm})^4 = 1018\text{ cm}^4,$$

$$M_1 = (\pi/4)(12\text{ cm})^2(180\text{ cm})(0.00783\text{ kg/cm}^3) = 159.4\text{ kg (approximately)},$$

$$J_1 = M_1 R_1^2 / 2 = (159.4 \text{ kg}/2)(6.0 \text{ cm})^2 = 2869 \text{ kg} - \text{cm}^2,$$

and

$$J_1 / R_1^2 = 2869 \text{ kg} - \text{cm}^2/(6 \text{ cm})^2 = 79.7 \text{ kg}.$$

Neckdown of the spans of web in the accumulator can be approximately determined by using the nominal width (not the actual width of each span as it suffers neckdown), using equation (12b) from Shelton [10]:

$$\left.\frac{dy}{dx}\right|_{(W/2)} = \frac{fW^3}{8EI}\left(\frac{L_B}{W} - \frac{2}{3}\right), \qquad \{14\}$$

where f (for 180-degree wrap) is twice the distributed tension, or 28.6 n/cm, and the other variables are specified above. The angle of each edge relative to the centerline of the web therefore calculates to be

$$\left.\frac{dy}{dx}\right|_{(W/2)} = \frac{1}{8}\frac{(28.6 \text{ n})}{\text{cm}}\frac{(140 \text{ cm})^3}{1018 \text{ cm}^4}\frac{\text{cm}^2}{2.0(10)^7 \text{ n}}\left(\frac{180 \text{ cm}}{140 \text{ cm}} - \frac{2}{3}\right)$$
$$= 0.000298 \text{ radians, or } 0.0171 \text{ degrees.}$$

This slope of the roller at the edge of the web might, at first glance, be dismissed as negligible, but the total neckdown of both edges of the web would be 2(30 spans)(100 cm)(0.000298) or 1.79 cm for an empty accumulator, and 5.0 times as great (8.94 cm) for a full accumulator. This 6.4 percent neckdown is very difficult to recover with spreading devices while re-establishing a uniform distribution of the cords.

The improved design will be a tubular roller with an outside diameter of 25 cm with a wall thickness of 0.5 cm:

$I_2 = \frac{\pi}{64}\left[(25.0 \text{ cm})^4 - (24.0 \text{ cm})^4\right] = 2889 \text{ cm}^4$, which is an improvement of stiffness by a factor of 2.84; hence, the neckdown with a full accumulator would be reduced from 8.94 cm to 3.15 cm, which probably is still too large.

$$M_2 = \frac{\pi}{4}\left[(25.0 \text{ cm})^2 - (24.0 \text{ cm})^2\right](180 \text{ cm})\left(0.00783 \text{ kg}/\text{cm}^3\right) = 54.2 \text{ kg}$$

(approximately) (34 percent as heavy, for ease of maintenance and reduction of load on the bearings in the upper rollers).

$J_2 / R^2 = (54.2 \text{ kg})(12.25 \text{ cm})^2 / (12.5 \text{ cm})^2 = 52.1 \text{ kg}$, resulting in an improvement by a factor of 1.53 in the acceleration of the rollers by the web.

The above example of improvement of an accumulator for tire-cord fabric is summarized in Table 1.

	Original Roller	Improved Roller	Improvement Factor
Outside Diameter – cm	12.0	25.0	
Inside Diameter – cm	zero (solid)	24.0	
Bending Stiffness – cm^4	1018	2889	**2.84**
Effective Inertia (J/R^2) – kg	79.7	52.1	**1.53** (reduction)
Mass – kg	159.4 (neglecting shafts)	54.2 (neglecting hubs and shafts)	**2.94** (reduction)
Neckdown – cm (full accumulator)	8.94	3.15	**2.84**

Table 1 – Example of Improvement of Rollers for Accumulator for Tire Cord Fabric
[(1) Roller Material: Steel, (2) Original W = 140 cm, (3) L_B = 180 cm,
(4) T = 2000 n, (5) L (full) = 5.0 m, and (6) 30 spans]

Other Improvements by Increasing the Diameter of Rollers. According to theory of rolling-element bearings, the life at a given load is measured in revolutions, not time; hence, life of the bearings for a roller would be twice as long if the diameter were doubled without changing the weight of the roller, by using a thinner wall and by careful design of hubs.

Another advantage of using large rollers also comes from theory of rolling-element bearings: The torque required to rotate a lightly-loaded bearing which has shields, not friction seals (often the case for achieving low friction of bearings for rollers) is proportional to the rotational velocity to the 2/3 power:

$$\tau = K_b \omega^{2/3}, \qquad \{15\}$$

where K_b is a constant for a given size and type of bearing and viscosity of the lubricant. The variables in equation {15}, however, are not the variables of interest for a web-driven roller, as was the case with equation {10}. If τ is again replaced with $(\Delta T)R$ and ω is replaced with its equivalent V/R, equation {15} becomes

$$\Delta T = K_b V^{2/3} / R^{5/3}. \qquad \{16\}$$

An example of application of equation {16}, similar to the example for the life of the bearings, is: If the diameter of the roller is doubled without changing its weight, the tension difference across the roller to maintain its surface velocity equal to a constant web velocity decreases by a factor of $(2)^{5/3} = 3.17$, or to 32 percent of the former requirement.

The torque of bearings which are lightly loaded (as are most idler bearings) has been found to be almost unaffected by changes in load.

Above equations expose a flaw in usage of a popular test for rollers, the “spindown test”, in which an installed idler without threadup of the web is spun above its operating speed by a string or friction wheel, then the time versus the rpm (measured by a noncontact tachometer) is recorded. The intention of this test is to check for bearing drag, such as the idealized case of equation {15}, as a cause of deceleration as expressed by equation {10}. While this test is useful for checking for deteriorating performance of rollers of a given design, it is useless for comparing rollers of different diameters and mass moments of inertia. The results of a spindown test would be improved by the addition of a flywheel, but the goal of low tension difference at a constant velocity as expressed by equation {16} would be unaffected, while the effective inertia J/R^2 of equation {13} would be adversely affected. Similarly, a spindown test would favor a roller with a thick wall, while a roller of the same diameter with a thin wall would be preferred for an accelerating roller (if its stiffness were adequate).

Thin-Wall Rollers:

Significant deflection should be avoided for virtually all rollers for web handling, because (1) deflection of cylindrical rollers under the influence of web tension causes neckdown and possibly wrinkling of a thin web, (2) the natural frequencies caused by the deflection and mass of rollers degrade the measurement and control of tension, and (3) unless the thickness of the wall of a roller is precisely uniform, the variation of deflection as a function of the rotational angle of the roller causes a vibration even if the roller is precisely balanced. In this article, the argument for large rollers with a thin wall is solidified.

Equations for the mass moment of inertia, J, and the area moment of inertia, I, of a cylindrical tube are usually expressed in terms of the fourth power of the outside diameter minus the fourth power of the inside diameter. In such form, the effect of the wall thickness and even the diameter are obscured. Nondimensionalization, expansion of $(1 - t/R)^4$, and simplification result in

$$\frac{J}{R^2} = 2\pi L_R \rho R t \left[1 - \frac{3}{2}\frac{t}{R} + \left(\frac{t}{R}\right)^2 - \frac{1}{4}\left(\frac{t}{R}\right)^3\right] \qquad \{17\}$$

and

$$I = \pi R^3 t \left[1 - \frac{3}{2}\frac{t}{R} + \left(\frac{t}{R}\right)^2 - \frac{1}{4}\left(\frac{t}{R}\right)^3\right]. \qquad \{18\}$$

Similarly, the mass of a cylindrical tube can be expressed as

$$M = 2\pi L_R \rho R t \left[1 - \frac{1}{2}\frac{t}{R}\right]. \qquad \{19\}$$

In equations {17}, {18}, and {19}, the terms in the brackets approach unity as t/R approaches zero; therefore, for a thin-walled roller,

$$\frac{J}{R^2} \approx 2\pi L_R \rho R t, \qquad \{20\}$$

$$I \approx \pi R^3 t, \qquad \{21\}$$

and

$$M \approx 2\pi L_R \rho R t. \qquad \{22\}$$

Equation {22} neglects the mass of the hubs and shafts.

As previously explained, J/R^2 should be small if the web must accelerate idlers, or if angular acceleration of a driven roller is important. If linear acceleration is important, as for a dancer or a contact roller, M should be small. For all rollers, the area moment I should be large. These needs, contrary to common belief, may not be conflicting in a major way because the R^3 relationship in equation {21} points to a large roller with a thin wall, while most rollers in web handling machinery have wall thicknesses far greater than desirable, if the radius were increased.

Table 2 shows great improvement of certain parameters of performance if a roller in a process machine is replaced by a roller with twice the diameter and one half the thickness of its wall, if the original roller could already be classified as "thin walled".

Performance Item	**Equation**	**Improvement Factor**
Effective Inertia J/R^2	{20}	Neutral
Bending Stiffness I	{21}	**4.0**
Mass M of Shell	{22}	Neutral
Life of Bearings	–	**2.0**
Tension Loss across Roller at Constant Velocity	{16}	**3.17** (32% as much)

Table 2 – Improvements of Performance of Thin-Walled Rollers by Doubling Diameter and Halving Wall Thickness (Same Material)

The amount of improvement as documented in Table 2 might not be necessary, and doubling the diameter of rollers might be objectionable because of space limitations. Table 3 examines a change of diameter by a factor of 1.5 while decreasing the wall thickness by a factor of 0.5, reasonable levels of change for many web handling processes.

Performance Item	**Equation**	**Improvement Factor**
Effective Inertia J/R^2	{20}	**1.33** (75% as large)
Bending Stiffness I	{21}	**1.69**
Mass M of Shell	{22}	**1.33** (75% as large)
Life of Bearings	–	**1.5**
Tension Loss across Roller at Constant Velocity	{16}	**1.97** (51% as much)

Table 3 – Improvements of Performance of Thin-Walled Rollers by Increasing Diameter by a Factor of 1.5 while Halving the Wall Thickness (Same Material)

The Need for Uniform Wall Thickness. Another reason that rollers should be large is to allow a stiff boring bar or arm for an internal grinder inside a roller while it is supported in a lathe, for achieving a uniform thickness of the wall. Figure 5 shows one problem which may be encountered when the inside of a roller is not a circular cylinder which is concentric with the outside: The stiffness of the roller varies with the angle of rotation, causing a variation in the deflection created by web tension and/or the weight of the roller, unless the deflection is negligible under all operating conditions (as for the heating drums in a papermaking machine, or for large, short heating drums for machine-direction orientation of a narrow plastic web). Figure 5 shows a cross-section of the shell of a roller which was defectively elliptical, then was machined round on the outside. This roller would have a maximum and minimum deflection twice per revolution; hence, vibration would occur at twice the rotational frequency of this roller even if it were perfectly balanced, particularly if the peak of the first natural frequency were approximately twice the rotational frequency. This point is repeated for emphasis: If a vibration is caused by a variation of deflection because of a variation in stiffness, any amount of effort at balancing (in multiple planes, for example) will not cure the problem.

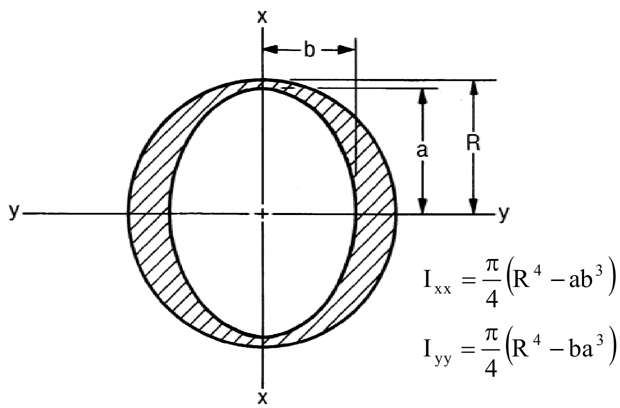

Figure 5 – Roller Machined From Elliptical (Inside) Tube

In processes involving chill rollers (usually with circulating water) or heating drums (with steam, hot oil, etc.), a nonuniform wall thickness has been known to cause variations in the temperature of the surface because of the variation in heat transfer as the wall thickness varies; however, a problem which may be more serious is the resulting warpage of the roller because of the variations in temperature of areas of the wall of the roller. Hence, a roller which was machined only on its outside to be round and straight may be unacceptably distorted in operation, when the web is acting as a heat source or a heat sink, with spotted variations in heat conductivity.

TRANSPORTING A WEB AT VELOCITIES GREATER THAN THE CRITICAL VELOCITY

Shelton [11] examined misconceptions about the causes of tension in a papermaking machine, and found that only one, differential-speed drawing, from seven listed causes actually generated tension in a draw-controlled process line. Other items on the list were indicators of the level of tension, or modified the level of tension in different locations along the machine. The cited paper is an extreme example of misunderstandings which have impeded progress in web handling, as mentioned in the Introduction of this paper.

The 1991 Shelton paper is mentioned here because of the growing variety of unusual webs, including some which are relatively massive but which must be handled at a low level of tension. Low tension in combination with the inexorable rise in velocities of handling of webs as a product matures can be expected to increase the demand for transport of webs at velocities greater than critical; that is, with mV^2 greater than the tension T, meaning that the inertia of the web dominates over the effect of tension. For operation at this supercritical velocity, the web must be captured by porous belts and directed in the path to the next roller and its capturing belt, as practiced in the wet end of a papermaking machine, as shown in Figure 6. Also as in handling of wet paper, pockets of air under pressure or vacuum, caused by the inertia of the air surrounding the web, are likely to need to be controlled with baffles, blowers, and other devices.

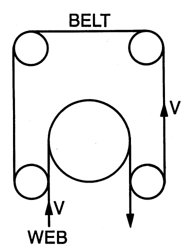

Figure 6 – Belt for Capturing Web at Supercritical Velocity
and Directing it to the Next Roller

ACCUMULATORS

Shelton [12] analyzed the acceleration and deceleration of an accumulator for zero-speed splicing at an unwinder or for zero-speed cutoff, unloading, and starting a new roll at a winder. The other modes of operation (constant velocity of the carriage during filling or emptying, or running of the line at a constant velocity with a stopped carriage) are relatively simple and trouble-free except for problems with non-parallelism of carriage rollers relative to the stationary rollers when location and constraint of the carriage are carelessly designed. The numbering system for rollers used by Shelton [12] is shown in Figure 7. Shelton has resumed work on analysis of accumulators, including accumulators for intermittent stoppage of a section of a process line, but results will not soon be published in open literature.

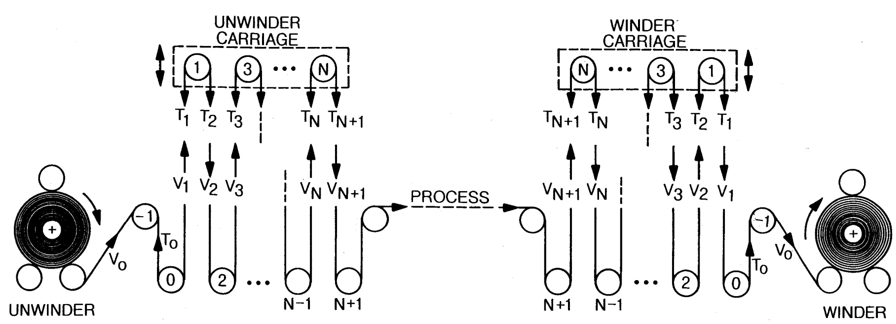

Figure 7 – Accumulators for Unwinder and Winder with Nomenclature
(Two-Drum Unwinder and Two-Drum Winder Shown)

Equations for J/R^2 (to be minimized in an accumulator) and bending stiffness I (to be maximized) are presented earlier in this paper, along with suggestions for the approach to roller design for an accumulator.

DIMENSIONLESS GROUPS OF VARIABLES

Solutions to engineering problems become vastly more difficult to generalize and to qualitatively understand as the number of independent variables increases beyond two or three. One of the most useful groups of variables is Reynolds' number, which relates four variables in a group which indicates the type of flow of a fluid as laminar or turbulent, and further indicates the degree of turbulence, with resulting pressure loss and noise. Therefore, tests do not have to be run with individual variation of the four parameters, with the group of variables applying to gases or liquids of all viscosities, large or small pipes of all shapes, dense or light fluid, and all velocities of flow.

A dimensionless group which has been an enlightenment for web handling for approximately thirty-five years is the Knox-Sweeney equation, which relates four independent variables raised to the power of 2/3. The thickness h_0 of the film of entrained air is then related linearly to the radius R of the roller, roll, or dead bar, making generalized graphs, tables, etc., very simple to construct. A generalized version of the Knox-Sweeney equation (with U the algebraic sum of the velocities of the web and the converging surface of a roller, winding roll, or dead bar of radius R) is

$$\frac{h_0}{R} = 0.643\left[\frac{6\mu U}{T/W}\right]^{2/3}. \qquad \{23\}$$

In web handling, a problem which has been simplified as much as appears to be practical may still involve more than a dozen independent variables. For such problems, even after defining of simple and obvious ratios such as strain (stress/modulus of elasticity), KL, L/W, E/G, J/R^2 (not dimensionless) as previously discussed, other natural groups of variables need to be formulated for simplification of the presentation of results of analysis, as well as for more general experimentation and computerized analysis.

In analysis of transport of a web along with measurement and control of tension, Shelton [13] found that a translational energy parameter $mV_i^2/L_T EtW$ defined the frequency response of a frictionless dancer, while a rotational energy parameter $\left(J/R^2\right)V_i^2/L_T EtW$ indicated the frequency response of cascaded rollers (one or more of which might be on load cells) when the web was not dynamically slipping. A simpler dimensionless group then reported was the roller design parameter J/mR^2, which indicates the degree of efficiency of usage of material in a roller when the surface of a roller needs to accelerate with a web.

In analysis of interaction of two web spans because of a misaligned downstream roller (with circumferential slippage at the next roller upstream from the misaligned roller), Shelton [9] showed that the problem and presentation of the results could be simplified with the five-variable group $\theta_r Et/f\mu$.

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Applying Hundreds of Years of Scientific and Engineering Knowledge in 50 Years of Time	**J. J. Shelton,** Oklahoma State University, USA

Name & Affiliation	**Question**
Dan Perdue, Goss International, Inc.	I have a general question about one of your topics. We now model a web span as a tensioned beam. Do you anticipate there being a need to move to a next level of complexity and model a web span as a plate?
Name & Affiliation	**Answer**
J.J. Shelton, Oklahoma State University	Not for most purposes. Of course, my thesis was on web guiding. I've done a lot of work since then on short spans. Instead of using hyperbolic sines and cosines to represent the lateral deformation you can employ a polynomial and solve the boundary condition problem in terms of a constant times x and x^2, etc. So I can anticipate simpler models for my work for some cases but not a shell or more complicated things. Jerry Brown will talk about cases where a beam is not adequate for modeling, but for lateral behavior you're not really going to benefit much by a more complicated model.
Name & Affiliation	**Question**
Dan Perdue, Goss International, Inc.	The question was directed toward some of the problems that we haven't been able to get a solution to, such as the cambered web.
Name & Affiliation	**Answer**
J.J. Shelton, Oklahoma State University	I just don't know. I've done a lot of work on a cambered webs and I don't know what to do next. It's a difficult problem, Jerry Brown has modeled it with his non-linear theory of elasticity program but he didn't get very close to Ron Swanson's data and Ron's eight or so data points seem to be the best data that we have. I tried to take data in 1969, as I reported previously. I improved the machine in 1971 and took more data. I have performed the analysis with the web modeled as a beam, but didn't come close to the test work that I did in 1969 and 1971. I have applied the theory of minimum potential energy to a cambered web. We need more data on a cambered web and that's very difficult to obtain.
Name & Affiliation	**Question**
Neal Michal, Kimberly Clark	John, you're in the process of writing a book and I thought maybe you could share with the group here where you're at and what your plans are for your book.
Name & Affiliation	**Answer**
J.J. Shelton, Oklahoma State University	I am writing the book in the format of IWEB with a font size of 10 and small figures. I have about 200 pages written, including the figures. I am just trying to summarize what I have learned over the years and what I've done in lateral behavior and control, tensile behavior

and control, span interaction, and other subjects. I don't know exactly when it will become available, I'm essentially working full time at this. I come to OSU one day a week and work with people here, bring written material to get it typed, and I bring sketches to be prettied up for my figures. I particularly don't know when it might be available to other than WHRC sponsors when I finish it. The first thing is to get something on paper. I think the book will be about 400 pages. It will include my work on the interaction of spans, which is essentially written up and is about 60 pages by itself. I have to edit this and try to make it more understandable.

DISCUSSION I

***Leaders*:** K. Hopcus, Fife Corporation, and R. Steves, Mitsubishi Polyester Films

Name & Affiliation	Comment
Ken Hopcus, Fife Corporation	We are open to discussion on any of the papers that were presented this morning.

Name & Affiliation	Comment
Dave Roisum, Finishing Technolgies	We had a couple of papers on telescoping and they seemed to compare constant tension, constant torque, heavy side, and hyperbolic profiles of winding tension, but these are all just mathematical idealisms. We don't have to be restricted to those shapes. Those profiles are convenient for electrical engineers and mathematicians. If you wind several rolls you may find a different winding tension profile is best for preventing telescoping. I have found that a z-shaped tension profile is best for maximum resistance for telescoping. In other words do not start to taper the winding tension immediately and keep it constant. If we observe the edges of rolls of web that were wound previously we can monitor the breakpoints where the roll edge transitions from curved regions near the core where slippage was occurring, to straight sections where slippage was not occurring, and finally back to curved regions near the outer radius where slippage was also occurring. We want to wind at a constant and sufficient tension to prevent slippage until we wind an inch beyond the wound roll radius near the core at the transition between the curved region (telescoping and slippage) and the straight region where no slipping occurs. This will be the first breakpoint in the Z-shaped tension profile at which we begin our taper of winding tension. We don't want to begin a taper right away and we don't want to be loosening anywhere near the core. If we focus only on these mathematical idealisms we are going to miss other ways of constructing a curve that might actually be better. If you study the edge slippage of the wound roll itself, you'll find two breakpoints which help you set the breakpoints for a Z shaped curve in winding tension that provides the optimum resistance to telescoping.

Name & Affiliation	Question
Cal Estrada, DuPont	This is also on the topic of winding tension tapers. We heard a talk from Konkuk University where they employed a heavy-side taper profile somewhere between the linear and hyperbolic profiles. Later we heard a talk from Tokai University where they had produced an optimized taper profile that was way below the hyperbolic profile. There appears to be a substantial discrepancy here. Does anyone know the source?

Name & Affiliation	Answer
Kee Shin, Konkuk University	I don't know whether either of my profiles would be the best since I just studied the hyperbolic and linear profiles. I cannot claim that either of these profiles is best because I just tried to find a test profile between those two – linear and hyperbolic – to optimize the stress distribution as well as telescoping at the same time. I don't know if we can find an optimal tension profile for all the cases.
Name & Affiliation	**Comment**
Keith Good, OSU	First I would make the comment that the validity of the Burns model employed in this research was discussed and recorded at the Eighth International Web Handling Conference. Thus care needs to be taken in drawing conclusions based upon the use of this model. I would also make the comment that often our results and conclusions are affected by the objectives we establish. My point is that telescoping and slippage related defects at the winder are undesirable, but these defects are also undesirable if they occur during unwinding the same roll in a downstream process. Thus if our objective also included the prevention of telescoping and slippage for the wound roll when it is next unwound the optimal winding tension profile would be different.
Name & Affiliation	**Question**
Ken Hopcus, Fife Corporation	Mr. Sieber, a question on your paper this morning. You presented a new method of controlling the lay on roll load. Traditionally devices such as pneumatic or hydraulic cylinders or springs have been used to produce the roll load. You indicated also that your new device has been installed on some machines, one or more machines, and it seems to be working very well. How many machines have you installed it on and what is the approximate cost of putting it onto a machine winding film?
Name & Affiliation	**Question**
Robert Steves, Mitsubishi Polyester Films	Mr. Sieber, I was wondering on your contact roll if you had looked at different contact roll stiffness. I know you said you had considered different wound roll materials, but it was unclear if you considered different contact roll stiffness. Did you study different contact roll stiffness in conjunction with the control system?
Name & Affiliation	**Answer**
B. Sieber, Brückner Maschinenbau GmbH & Co. KG	Yes. It is integral to the simulation. In the simulation, it is very easy to vary all the parameters. You can see there is a satisfying congruence in both the simulation and the reaction of the real system. We can simulate both the changing parameters of the contact roll and as well as C parameters of different materials. In most of the cases, we don't know the related spring constants of the material.
Name & Affiliation	**Question**
Marko Jorkama, Metso	I'd like to ask a question about the damping of the system.

Paper	I think this arrangement is increasing the damping in the system. Did you do any frequency function measurements in order to evaluate the increase in damping?
Name & Affiliation	**Answer**
B. Sieber, Brückner Maschinenbau GmbH & Co. KG	We did simulate this but we did not measure it.
Name & Affiliation	**Question**
Tim Walker, TJ Walker and Associates	Mr. Sieber – I want to clarify that the purpose of the mechatronic damping system is to reduce the bouncing of the contact roll, is this is the primary goal? The number one purpose of your upgrade, of going away from air cylinder approach to linear motors is to reduce bouncing of the contact roll. Is that the main reason you've gone to a new system? What motivated you to go away from air cylinders?
Name & Affiliation	**Answer**
B. Sieber, Brückner Maschinenbau GmbH & Co. KG	I have never witnessed a bouncing roll in an upgraded or in the new line using this device. The most important thing is damping and I indicated that the primary objective is to reduce the bouncing of the contact roll.
Name & Affiliation	**Question**
Tim Walker, TJ Walker and Associates	Dr. Shelton asked about the diameter the roll you can accommodate. Is the linear motor essentially equal to the length of the radius of the roll, so it slides from an upfront position all the way to a rear position? There's no typical design without a carriage and a contact roll would have a limited swing motion and the entire carriage would back off. Is this mechatronic system replacing only the small swing of the contact roll or the entire carriage motion, as well?
Name & Affiliation	**Answer**
Hans-Juergen Kittsteiner, Brückner Maschinenbau GmbH & Co. KG	I am the design engineer on this project. The length of the linear motor is about 800 mm. The big advantage of the linear motor is that compared to the older systems, there is now no mechanical backlash. The older systems all have dampers or cylinders and mechanical dampening requires movement. If there is no movement there is no dampening effect. The linear motors require no movement for dampening and you can adjust the parameters very easily.
Name & Affiliation	**Question**
Tim Walker, TJ Walker and Associates	From the discussion after your presentation it sounds like the contact roll is not maintained in alignment, it's allowed to conform to the winding roll. Does that mean there are spherical mounts on the bearings on the two ends to prevent racking of the system as it does skew?

Name & Affiliation	**Answer**
Hans-Juergen Kittsteiner, Brückner Maschinenbau GmbH & Co. KG	You must not do it. The system can't tolerate it. Air can enter between the secondary and the primary.
Name & Affiliation	**Question**
Keith Good, Oklahoma State University	This question is for Balaji Kandadai: In your presentation you stated you were modeling 13 layers of web with isotropic properties. I know you made some runs where you allowed the web to be orthotropic and then you made a few runs where you allowed it to be orthotropic, state dependant. You told us this morning it took 2-1/2 days to make an isotropic run. What computational times will be required when we start modeling realistic web properties?
Name & Affiliation	**Answer**
B.K. Kandadai, Oklahoma State University	Let's take for example, if we were to wind a linear orthotropic web. When I tried, especially with a nip roller in place, if you were to wind for example 5 layers or so, it will probably take about 10 days using one processor. If you were to include a nonlinear routine, a subroutine such that you can simulate the actual nonlinearity in the radial modulus, 15-20 days would be required to wind 5 laps. For 13 laps it is much more.
Name & Affiliation	**Question**
Kevin Cole, Grid Computing Solutions	Since we are talking about Balaji's paper, I have a question about surface directions. You show surface directions on the top side of the web and on the bottom side for about 5 laps. I am finding it difficult to understand why the slip would alternate between one direction and the other within a continuous one revolution for the winding process. What are your thoughts on that as opposed to just going one way or the other through the whole process?
Name & Affiliation	**Answer**
B.K. Kandadai, Oklahoma State University	Honestly, if we think about this, it is kind of hard to actually make physical sense of the top surface tractions, alone, or the bottom surface tractions alone. Yes, a discrete point you see slip and stick, but I think we have to focus on the net traction and I believe that is the key parameter so when we look at that then you can make some sense as to whether you will get additional wound-in-tension or not. But in terms of relative velocities, how the layer is going to move, whether it is going to be moving forward and then instantaneously moving backward on the bottom surface? Based on the plot that you see, it's definitely hard to imagine it is doing that, but that is what the results showed.
Name & Affiliation	**Question**
Tim Walker, TJ Walker and Associates	In your model, it is a spiral you are modeling instead of a series of rings, so if you are implying slip are you seeing a place where a layer comes down, has a reference point

	relatively below it, and then as 2-3 layers go by the layer will slip relative to the layer below it in the clockwise direction, are you keeping track of that amount?
Name & Affiliation	**Answer**
B.K. Kandadai, Oklahoma State University	Not for every case, but one case I did try to plot surface traction as a function of time. For example, what I would do is after one layer is built, I would look at the surface tractions and after two or three layers of being wound, you look at the surface tractions and compare it to the first time step where you just had one layer alone. I did not see much change in the surface traction, which tells me that the layers are not moving relative to each other – they are pretty much in a locked state. But, I did not do it in every single case, that was just on a single one that I tested.
Name & Affiliation	**Question**
Tim Walker, TJ Walker and Associates	I saw in your references you listed the work by Dr. Jain Cao at Northwestern University. How did your work differ from the work they did there?
Name & Affiliation	**Answer**
B.K. Kandadai, Oklahoma State University	Theirs mainly simulates the deformation of a hot coil. They were dealing with metals and these are huge, heavy rolls. What they were analyzing was if you were to sit a fully wound roll on the floor, how much permanent deformation would you get? So, how much out of roundness would result? The way they did it was without a core, the mandrel was removed, so the problem they were studying was different. They were not analyzing the winding of a roll. They modeled the wound roll as a disk that was divided into several sections. In each section they would specify different modulus, as if to simulate different radial pressure. They were not clear on how to define the radial pressure inside the roll. They would measure the radial pressure and then plug it back into their model and alter their shear modulus values until they obtained the same mass radial pressure profile. Then they would apply a gravity force to simulate the sheet coil deformation. Again, no interlayer slippage was modeled, it was a continuum model.
Name & Affiliation	**Comment**
Jan Erik Olsen, Sintef	I thought about the same paper. I think this is interesting work. I would like to see that nonlinear orthotropic simulation for 200 layers. It is two years until the next conference, hopefully the results will be ready for the next IWEB conference.
Name & Affiliation	**Question**
Marko Jorkama, Metso Paper	Balaji – a question about the web tension in the top layer. We know that the most of the increase of the web happens at the topmost layer, but what about that second layer which is found inside the roll? Did you see any increase in tension in that because at least one of your references talks about that? Kilwa Arola saw quite significant increase in

	tension also in second layer.
Name & Affiliation	**Answer**
B.K. Kandadai, Oklahoma State University	He used a different model, a continuum based model. The way he simulated his work was similar to what I was telling Dr. Olsen, it would be a disk and it is modeled using a continuum element, which is capable of having slip/stick behavior. In this case, he did see an increase of tension. In all of the cases that I simulated, I did not. Again, these are two completely different modeling approaches. So, I don't know. I really can't comment on how he got a tension increase. In other work he did, where he modeled the interlayer slippage in flat bed nip mechanics problem – he saw a little tension increase in the second layer. We have to remember that that's a flatbed case and that the layers underneath the first layer aren't constrained in the same way as it would have been in a wound roll. I did do some flatbed tests, but I'm not sure I can comment on that because that was from WHRC work. I will just say I have seen some increase before. In a wound roll, no but in a flatbed, yes.
Name & Affiliation	**Question**
Dilwyn Jones, Emral Ltd.	I have a question for Mr. Sieber. Your diagram of the moving contact roll didn't show the incoming web. Does it come in parallel to the direction of movement of the contact roll or does it come in at 90 degrees? Did you take the web path into and exiting the contact roll into account in the simulations that you were doing?
Name & Affiliation	**Answer**
B. Sieber, Brückner Maschinenbau GmbH & Co. KG	In the simulation models the incoming web is not modeled. There is no influence of the tension.
Name & Affiliation	**Question**
Tim Walker, TJ Walker and Associates	This question is for Keith Good – I have a question about your laser Doppler velocimeter use. The first thing that jumps to my mind after watching your measurements, is have you used it to look at cross roll variations, have you used that technique to look at the stress variations of web entering and leaving a concave roller? Would the LDVs be applicable for such problems?
Name & Affiliation	**Answer**
Keith Good, Oklahoma State University	I made the comment this morning that, in fact, you could use the LDVs to examine the strain changes between any two spans where, in the case reported, the downstream span was actually the web on the wound roll. There is no reason you couldn't use this to profile strain changes across the web width as well. I have used it to do other things, but I won't comment on what I have done.
Name & Affiliation	**Question**
Ken Hopcus, Fife Corporation	What does the LDV system like that sell for?

Name & Affiliation	**Answer**
Keith Good, Oklahoma State University	10 years ago one LDV would have cost $40,000. You need two of them, so you would have needed $80,000 to measure a draw or strain. The early LDV's were robust; they could continue making measurements if the target slowed to a stop and then reversed direction. Today there are companies that are streamlining the costs of these devices by only allowing velocity measurements in one direction. This is not a significant limitation in a web line where the web is usually transported in a downstream machine direction. If you are willing to just measure velocities going in just one direction, you can buy these today for $6,000-7,000 per LDV.
Name & Affiliation	**Question**
Dilwyn Jones, Emral Ltd.	Keith – I have another question on the laser Doppler work. You're referring to pulses generated, and I'm struggling to understand where those come from and why they are regular. From my understanding you've got a set of interference with the two laser beams and as particles come across you get a change in intensity. But it seems to me you need a regular supply of particles to generate pulses. Could you explain a little bit where they are coming from?
Name & Affiliation	**Answer**
Keith Good, Oklahoma State University	The earliest measurements of this type were called laser Doppler anemometry. These measurements were made in flow fields and one had to inject particles to scatter the laser light which resulted in the interference you speak of. It was assumed that the particle and the local flow velocity were the same and that by measuring the velocity of the particle you could infer the flow velocity. The laser Doppler velocimeter is a somewhat different device in that the previous particles are now asperities on the surface of the web and instead of comparing two beams of transmitted light through a flow field we now compare reflected light from the web surface. The interference that occurs can be related to the velocity of the asperities which are moving at the same speed as the web. That velocity can now be integrated through time to provide a length measurement and after a predetermined length has passed the target area a TTL pulse is output. In my case there were 1000 TTL pulses output for every foot of web that past the measurement site. The change in length that is measured between the two velocimeter target sites helps you to infer a change in strain between the two measurement sites. If you know what the strain in the web was at the upstream measurement site you can deduce the total strain at the downstream site.

Name & Affiliation	Question
Dilwyn Jones, Emral Ltd.	Ok, I think I understand now. I was getting confused between the anemometry you were mentioning and the individual particles that generate pulses which you pick up from the correlation and something which is generated from the software.
Name & Affiliation	**Answer**
Keith Good, Oklahoma State University	As I was alluding to this morning, there are limitations to the method. If you have a surface that was optically flat, it would not work. It has to have some roughness, some specularity. The web must also have adequate opacity so that you can ensure that the reflected laser light is being reflected from the outer surface of the outermost layer. If these two requirements are not satisfied then there is no choice but to employ encoders. Encoders are really used as a last resort because they have to be force loaded to keep them in contact with the winding roll. You will find your results will be affected by the force loading you select.
Name & Affiliation	**Comment**
Dilwyn Jones, Emral Ltd.	I tried to do that measurement 15 years ago on a wound roll and we were never sure that we were measuring the velocity of the outer layer or several layer underneath.
Name & Affiliation	**Question**
Cal Estrada, DuPont	Dr. Good – if you are able to determine wound-in-tension on a fly, could you also determine the internal stresses on the fly? If so, would that give you some sort of probability whether or not you're creating a high quality roll or creating defects?
Name & Affiliation	**Answer**
Keith Good, Oklahoma State University	The answer is yes but must be qualified. You have to incorporate the wound-in-tension measurement that you just made with what you think is a valid winding model. You would have to have both of those things running simultaneously to know what the internal stresses were in the roll. Once you know what those internal stresses are, then you compare some sort of a defect or an objective function, as we were talking about earlier. So with the WIT measurement, a winding model, and defect models running simultaneously yes we should be able to know if we are winding good rolls while we are winding them.
Name & Affiliation	**Question**
Cal Estrada, DuPont	Could you then use the same sensors as a feedback mechanism in case things were going south?
Name & Affiliation	**Answer**
Keith Good, Oklahoma State University	If you are asking if this could be used in the real time quality improvement of wound rolls that are winding, yes.
Name & Affiliation	**Question**
Dilwyn Jones, Emral Ltd.	I have a question for Dr. Boutaous on the optimization of tensions in winding systems. It is really on the types of errors you might have; it seems to be just a deviation on the tension, sort of the wrong slope, or the wrong reference

	value with a given slope. I wondered if you had considered adding random noise to the tension and studying that result. At this stage, you've had some surprising effects that the tangential stress was outside the range between that produced by the maximum value. Would other profiles of the tension, plus error yield more surprising results?
Name & Affiliation	**Answer**
M. Boutaous, INSA de Lyon, Site de Plasturgie	When you have this variation, it's opposite. When we have this variation, you occupy all the range of the gauge, the maximum value of this tension. So, you start the computer at each one layer. Change from the maximum to the minimum value of the tension and in this case, you have all stresses in these limits and they do not exceed the limits.
Name & Affiliation	**Question**
Dilwyn Jones, Emral Ltd.	So the band exists for the limits of that as they would for changing the slope?
Name & Affiliation	**Answer**
M. Boutaous, INSA de Lyon, Site de Plasturgie	Yes.