CONSIDERATIONS IN THE SELECTION OF A DANCER OR LOAD CELL BASED TENSION REGULATION STRATEGY

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ABSTRACT

Two methods are predominantly used to control tension in web lines: a position regulated dancer roller strategy or a regulator based on tension feedback from a load cell. Dancers are most typically used on the unwind or rewind sections; sometimes they are applied on interior machine sections with special requirements, such as a rapidly changing web span lengths. Load cell control has predominantly been used on the interior machine sections, although it is also applied on unwind and rewind sections.

There has been much controversy and debate in the industry on the benefits and limitations of dancers as compared to load cell tension control. Web handling equipment suppliers offer varying and often contradictory reasons for making this selection. At the 5th International Web Handling Conference, J. J. Shelton of the Oklahoma State University published a paper that analyzed the performance of dancers and fixed span tension sensing rollers, and pointed out some limitations of dancers at higher frequencies. Another paper at this conference by J. P. Ries of DuPont suggested that load cells were better at low frequencies, dancers at mid frequencies, and they provided comparable performance at high frequencies. Other literature makes different claims on the drawbacks and benefits of these two methods.

This paper attempts to align and clarify this variety of viewpoints by presenting measured data on an actual webline. Comparisons are made between these measured results and the results predicted by these papers. In addition, this paper also presents some pragmatic considerations in the selection of either dancer or load cell tension regulators.

NOMENCLATURE

A	the span between the controlled roll or roller and the sensing roller
A	the first idler in a series and its entering span (multiple idlers)
B	the span across the sensing roller from the controlled roll or roller
B	the second idler in a series and its entering span (multiple idlers)
c	general numerical constant
C	damping constant of a dancer or load cell
C_n	constant for expressing the undamped natural frequency
C_V	pneumatic flow resistance constant, dimensionless
E	Young's tensile modulus of web in the longitudinal direction, Pa (psi)
F_D	air cylinder force applied to dancer, N
F_S	dancer static (breakaway) friction, N
g	gravitation acceleration, 980 cm/sec^2
J	mass moment of inertia, $kg\text{-}cm^2$
j	the square root of -1 ($\sqrt{-1}$)
K	total spring rate of a dancer or set of load cells, N/cm
K_τ	torque constant of motor (torque/current, N-cm/A)
L	length of a span, cm
L_T	total length of spans between driven rollers, cm
m	effective translational mass of a roll or roller, kg
R	a constant radius, cm
r	a variable radius, cm
s	Laplacian operator (equal to $j\omega$ for frequency response analysis)
T	web tension, N
T_k	torque, N-cm
t	web thickness, cm
V	web velocity, cm/sec
w	web width, cm
y	dancer displacement, cm
Δ	amplitude of the sinusoidal change from the average condition
λ	an incremental length of web, cm
μ	effective coefficient of friction
θ	angle of wrap (radians), angle of rotation
τ	time constant of process

INTRODUCTION

Load cell control and dancer control are two common methods of controlling tension in a web handling machine. Both strategies are widely used; however, dancers are normally employed only in the unwinding and winding sections of a machine, or in internal machine sections that are expected to experience large upsets. Figure 1 shows the general geometrical configuration and nomenclature for both systems.

A load cell consists of an idler roller mounted on generally two beams, each supporting one end of the roller. Strain gauges are attached to each beam, and the resulting deflection in the beams from the tension in the web that is wrapping the roller is measured. The beam deflection is scaled to the equivalent tension. The measured tension is compared to the desired tension setpoint, and (in the most commonly employed control loop strategy) a proportional-integral-derivative (PID) controller calculates a corrective speed trim signal that is summed with the line speed reference and applied as a commanded velocity reference to the drive and motor.

A dancer (Figure 2) consists of a roller that is free to move, generally on a pivot but sometimes on linear slides. The web wraps the roller, optimally by 180 degrees, and an opposite directed variable opposing force is applied, normally by air cylinders. A sensor is used to measure the position of the dancer roll. The PID controller compares the present dancer roll position with the desired position, and the calculated motor trim value is applied to the drive-motor system to bring the dancer roller back to its normal position. Tension setpoint changes are made by adjusting the air pressure to the loading cylinder.

The following equation describes the forces acting on a dancer:

$$2 \cdot \boldsymbol{T} = \boldsymbol{F}_D + \boldsymbol{F}_S + \boldsymbol{K}_1 \boldsymbol{y} + \boldsymbol{C}_1 \frac{\mathrm{dy}}{\mathrm{dt}} + \boldsymbol{mg} \sin(\theta) \tag{1}$$

The web tension force, equal to two times the web tension T, is counter balanced by the sum of the dancer forces: cylinder force F_D, the break away static friction F_S, the dancer spring rate force K_1y, the dancer viscous force C_1dy/dt, plus the dancer gravitational force mg $\sin(\theta)$. F_D may also vary due to the relatively long time constant of pneumatic systems; the dancer cylinder essentially acts adiabaticly over short time periods relative to its time constant. While the load cell roller has the majority of these forces, they are much more idealized than in the dancer roller system. A load cell employs a set of beams, so friction, hysteresis, and other non-idealized forces are minimal. Only deflection, and not rolling, sliding, or other friction generating motions are present. The spring rate is extremely high, so generally the lower web spring rate dominates. Typical maximum displacements are several orders of magnitude lower than a dancer experiences during normal operation.

A dancer is a position regulator. No corrective action is taken by the regulator until the dancer has moved. A slightly larger web exiting velocity than the entering velocity results in increasing strain in this section of the web. This means that the tension is also increasing, until the tension increase exceeds the dancer static friction F_S. The result is a sudden movement by the dancer each time the static friction is exceeded. The control loop senses this change in position, and increases the command signal to the incoming web's drive and motor, increasing the web velocity slightly. Now the tension will fall until it reaches F_S below the current value, at which time the dancer will again move. Note this total change of F_S is balanced by two times the web tension change, meaning a dancer will **always** dither by at least +/- (F_S /2) around the tension setpoint. Careful attention to design details can decrease the level, but can never eliminate this dithering effect.

One of the primary benefits of a dancer is the increased bandwidth it adds to the tension control loop. In a standard load cell zone, the tension sensitivity to velocity changes as a function of the spring constant of this zone, which is Etw/L. This value often approaches 1750 N per cm. Enormous tension transients result from velocity upsets, such as velocity mismatches on spindle transfers due to the error in estimating the incoming roll diameter. This results in missed splices and web breaks. Note that the mechanical mass-spring-damper arrangement of a load cell acts as a low pass filter, and the true transient tension value experienced by the web will be substantially higher than the value indicated by the load cell amplifier. The true peak level is masked by this often overlooked mechanical time constant of the tension transducer. Often, the amplifier will have additional filtering.

In typical production equipment, spindle drives rarely have more than a few hertz velocity bandwidth due to the large roll inertia and the limiting effects of torsional compliance in the drive train. With such low bandwidths, the drive system can only correct tension disturbances in the very low hertz range; therefore the drive and motor can be considered at steady state in the analysis to determine the tension disturbance rejection capability of each system.

The load cell system is effectively a span between two driven rollers containing one or more rollers. The dancer system is a span between two driven rollers containing an (ideally) constantly force loader roller, with additional rollers. The question therefore becomes how much disturbance attenuation does a dancer have relative to a load cell controller, and over what frequency range is it useful? After surveying various dancer analysis in the literature, and making preliminary experimental testing of Shelton's dancer analysis presented at the 1999 IWEB conference, it was decided to perform a more detailed analysis and experimental verification.

Shelton has derived a system of equations in matrix format describing the relationship between the various dependant variables as a function of the independent variables [equation 2 in Ref 1] in an unwinding zone is :

$$\begin{pmatrix} 1 & 0 & 0 & 0 & 0 \\ -1 & 1 & -\left(\dfrac{L_{Ai}s+V_i}{Etw}\right) & 0 & s \\ 0 & -1 & \dfrac{V_i}{Etw} & -\left(\dfrac{L_{Bi}s+V_i}{Etw}\right) & s \\ 0 & -\left(\dfrac{J_1}{R_1^{\,2}}\right)s & -1 & 1 & 0 \\ 0 & 0 & 1 & 1 & m_1 s^2 + C_1 s + K_1 \end{pmatrix} \times \begin{pmatrix} \Delta V_0(s) \\ \Delta V_1(s) \\ \Delta T_A(s) \\ \Delta T_B(s) \\ \Delta Y(s) \end{pmatrix}$$

$$= \begin{pmatrix} \dfrac{V_i}{r_{0i}}\Delta r_0(s) + r_{0i}\Delta\omega_0(s) \\ -\dfrac{V_i}{Etw}\Delta T_0(s) \\ 0 \\ 0 \\ \Delta F(s) \end{pmatrix} \qquad (2)$$

This equation may be solved to express any of the five desired dependant variables as a function of the independent variables on the right side of the equation, as shown in Ref. 2.

Shelton's analysis relied heavily on the concept of dimensionless groups. Various parameters would be grouped together as a single variable, normally having unitless dimensions. This allows analysis and plotting of these very complex relationships for various variables, or groups of variables. In addition, instead of the traditional frequency axis of s, f, or $j\omega$, Shelton chose the dimensionless group $s = jL_T/r_{oi}$. This variable represents the frequency as a function of spindle RPM at a given line speed; i.e. it is scaled spindle RPM with large diameters corresponding to low frequencies which will lie on the left side of the x frequency axis. Figure 3 is the plot of the predicted amplitude ratio of the change in tension in span B vs. span A for a velocity controlled unwind, with frequency scaled by the factor L_T/r_{oi}. The assumption was made that $J_1/m_1r^2=0.75$ for a typical roller having significant portion of its mass near its circumference and is reasonable for typical thin shell constructed rollers. The numbers on the plot are the dimensionless group:

$$\frac{m_1 V_i^2}{r_{oi} Etw} \tag{3}$$

The top left curve is the tension response for a fixed idler roll as would be found in a load cell strategy. Several features are noteworthy. First, at low frequencies, the dancer performs better than the fixed roller, as expected. However, as the parameter $(m_1V_1^2)/(r_{oi}Etw)$ increases, the response exhibits strong resonant peaking, and the dancer performs substantially worse than a fixed idler roller at these frequencies. At high frequencies, both responses fall off at comparable rates (20dB/decade).

TEST PROCEDURE

To perform the experimental verification, modifications were made on a tension control zone of a web handling machine (Figure 4). To simulate the unwinding roll analysis, web passed over the vacuum pull roll located on the right side of the drawing. The web passes over a load cell in zone A, around the dancer roll at 180 degrees of wrap, and then exits by the nipped pull roll. Sinusoidal transfer function analysis is used. A fixed sinusoidal disturbance at a given frequency is applied to disturb the web velocity, and the tension response as a function of this frequency is measured.

In equation (2) there are three sets of independent variables. The first set represents changes in web velocity from the zeroth roller:

$$\frac{V_i}{r_{0i}}\Delta r_0(s) + r_{0i}\Delta\omega_0(s) \tag{4}$$

It consists of two components. The left hand component represents changes in the unwinding roll diameter as a function of frequency, scaled by the angular velocity of the spindle. That is, Δr_0 times the *fixed* spindle velocity ω (V_i/r_{oi}) is the resulting web velocity change. The second term results from changes in the *variable* spindle velocity ω scaled by the radius at that instant, r_{oi}. In the experimental setup, constant diameter driven rollers were used, so the left hand side was set to 0.

The second set of independent parameters is:

$$\frac{V_i}{Etw}\Delta T_0(s) \qquad (5)$$

Delta T_0 is the change in the wound in tension of the unwinding roll, or changes in tension in the incoming span. For experimental purposes, the web was entering from another span with its tension held as constant as possible. This term is therefore set to zero in the analysis for the experiment.

The third independent variable was $\Delta F(s)$, the change in the force applied to the dancer as a function of frequency. For experimental purposes, this was assumed constant and set to 0 in the analysis; however, as noted in [1] this value is dependent on many effects, including pneumatics, friction, and gravity effects from pivot angle.

Equation (2) was solved for the tension in span A, T_A, and the experimental setup used to measure the response curve. Matlab® was then used to predict the theoretical response.

One difficulty in making these measurements is the expected low frequencies of the response. The time required for making frequency response measurements is inversely proportional to the frequency. Additional time is required to allow transients to settle for each change in frequency. Time is also required to optimize the applied amplitude of the disturbance. The net result is measurement times on the order of tens of minutes may be required for a set of measurements. Due to this long time, small steady state velocity differences between the two driven rollers would result in a substantial difference in the amount of web added and removed from the span. Therefore the dancer had to be operated in a closed loop mode to prevent it from 'bottoming out' or pulling taught. To minimize the impact of this effect on the response being measured, the control loop dancer gain was set to extremely low values. The proportional gain was set to its minimum, and the integral gain was set so that it had a net effect over many tens of seconds, but had minimal effect at the 5 seconds corresponding to the lowest 0.2 Hz signal applied. An experiment was also conducted to demonstrate that the control loop had minimal impact on the open loop dancer response.

Figure 5 demonstrates the theoretical effect of the dancer spring rate on the tension response. There are two notable effects. The first is that the amplitude of the second predicted peak is increased with respect to the amplitude of the first peak. Secondly, note there is an increase in amplitude with a 1/f slope at low frequencies. Consider the effect of gravity in equation 1. This term was not included in Shelton's analysis, and normally is negligible for small displacements. However, the applied low frequency web velocity disturbance is integrated into a position error:

$$\Delta L_{span} = \int_{start}^{end} \left(V_{exiting} - V_{entering}\right) dt \qquad (6)$$

Low frequencies result in appreciable web span length excursions, and large dancer displacements. The dancer movement results in the weight of the dancer being added to and subtracted from the force balance equation, with zero effect when the dancer is at its lowest hanging position. Note that this force is directly proportional to the dancer position (for small displacements when the $\sin(\theta) \sim= \theta$). Therefore this effect adds to the spring rate, but with slope 1/f because of the integration described above.

A similar effect results from the change in dancer force due to cylinder pressure changes caused by the dancer moving faster than the pneumatics can react. The cylinder is essentially adiabatic unless the frequency is well below the pneumatic system time constant. Since the pressure times the volume is a constant the resulting changes in force can be expressed as:

$$\Delta F_D \propto \frac{c}{\pi r^2} \Delta L \qquad (7)$$

Note the form is the same as the dancer spring rate. This also suggests that the cylinder volume should be as large as possible, and that a large length to radius, or alternatively air accumulators (buffers) that have minimal flow restrictions to the cylinder. Machine cross tube members make good accumulators, but ensure that pertinent ASME pressure vessel codes are adhered to, as substantially energy can be contained within. Pneumatic E/P and I/P transducers have time constants on the order of a second, but the overall system time constant is dominated by the flow rate, overall system volume, and C_V of the piping. Volume boosters can improve performance. Large diameter piping increase the C_V. The dancer pneumatic design is an area that is often overlooked, resulting in degraded dancer performance. Note that other high bandwidth actuators, such as motors, may be used in place of the slow pneumatics.

The closed loop dancer's disturbance rejection is increased at lower frequencies over the open loop dancer, due to the increased corrective action. The loop's bandwidth is limited, so the disturbance rejection decreases as the frequency increases. The closed loop control has the same slope as Shelton's model at low frequencies, and so would increase the apparent disturbance rejection at low frequencies. It was observed in ALL experiments than the slope always increased at low frequencies, i.e. there actually was less rejection at lower frequencies. This indicates that the closed loop dancer control had minimal impact on the measured transfer function, as its effects were negligible compared to the spring rate mechanisms described above.

DISCUSSION AND ANALYSIS

Figure 5 from Ref. 1 would imply three possible variable groupings that impact the dancer resonance: web velocity, dancer mass, and web dimension and material properties.

The first set of experiments were designed to measure the impact of web velocity, which should show up as a squared effect. An experiment was performed to measure the magnitude of the tension response in span A as a function of the applied web velocity disturbance frequency for four line speeds. A distinct resonant peak is observed in all plots, but this frequency did not change with linespeed. Figure 6 shows the plot of the predicted tension response of span as a function of line speed for these conditions. The predicted response also does not exhibit a dependence on linespeed, although there appears to be evidence of increased damping (reduction in the sharpness of peaks) as a result of linespeed and strain transport. Boulter had previously noted the improved stability and the ability to increase control loop gains at higher web speeds. [Ref. 8]

In hindsight, it can be seen that there is no speed dependence. In (3), when V_i^2 changes, then the x axis frequency L_T/r_{oi} also changes For example, in Figure 3 the peak for $(m1V_i^2)/(L_TEtw)$ equal to 10 occurs on the x axis at L_T/r_{oi} about 0.6, and the peak for $(m1V_i^2)/(L_TEtw)$ equal to 0.001 occurs on the x axis at L_T/r_{oi} about 60. The ratio of velocities on the x axis is 100, and the ratio means the grouped parameters should vary by 100^2; and $0.001 * 100^2$ is indeed 10. The use of dimensionless groups are a great tool to ease the analysis, but care must be used as it can also obscure some behaviors.

An experiment was performed to compare the theoretical response of a single roller to that predicted by the model. To set up the model for a single roller, the spring rate K_1 is set to a typical value for a load cell (1660 N /cm). This ensures the web spring rate would have the dominate effect. The dancer damping rate C_1 was set to zero. With the high spring rate, there should be negligible roll movement and C_1 should not have any effect. The machine test configuration had 2 rollers with 100 cm spans to the driven rollers, and 356 cm of 30.5 cm wide by 0.0025 cm thick polyester web between them. This was not as close to a single roller as was desired, but was deemed suitable for the low frequency verification. Figure 7 shows the theoretical and measured responses. Note there is quite good agreement both in the amplitude and frequency to nearly the upper end of the scale. The theoretical response shows an anti-resonance minimum at 5 Hz and a resonance at 10.7 Hz. The measured response above 5 Hz (not shown due to time limitations) showed a broad minimum 10.5 and narrow peaks at 11.3 and 12 Hz. There appears to be two peaks as expected for two rollers, but as the primary interest was low frequency behavior of the roll this phenomenon was not further investigated this time.

The next set of experiments focused on the impact of the dancer roller mass and inertia, and the web span parameter Etw. Note that two additional idler rollers are required to configure the web path for the dancer, one before and one after the dancer roller. These rollers were not included in (2), and it is expected that an additional resonant frequency would appear in the transfer function for each roller. To help differentiate these frequencies, two dancer rollers were selected with dramatically different masses and inertias. A 30.54 cm wide by 0.0025 cm thick Polyester web was chosen for a very stiff web span, and a 40.6 cm wide by 0.0056 cm thick Polyethylene web to produce a very soft web span.

Figure 8 compares the theoretical vs. the measured frequency response of the tension in span A for a large heavy roller with a relatively stiff Polyester web. The model predicted an initial peak at about 3.8 Hz, and a second peak at 6.2 Hz. The measured response had a peak at 4.5 Hz. There was a hint of a second peak at about 7.5 Hz, although it is very weak.

Figure 9 uses the same conditions as Figure 8, except that a lighter dancer roller has been installed. The model predicted an initial peak at about 10 Hz, and a very strong second peak at 10.7 Hz. The measured response has a broad peak at 10.2 Hz and a strong second peak at 12 Hz. In addition, there are smaller peaks at 4, 10, and 10.7. The signal is in general substantially noisier that the larger roller. The measured peaks are higher than predicted, and the additional peaks are probably due to the additional rollers' resonances. The light dancer in conjunction with the stiff web results in the dancer resonance being near the roller resonances.

Figure 10 compares the responses for the large heavy roller with a relatively soft Polyethylene web. The model predicted a peak at 2.0 Hz, with a hint of a second peak at 3.4 Hz. The measured response has a first peak at 2.8 Hz, and a second peak at 10.5 Hz.

Figure 11 compares the responses for the small light roller with the Polyethylene web. The model predicts a broad peak at 5.6 Hz, and a second peak at 9.2 Hz. The measured response has a first peak at 8.5 Hz, and a smaller second peak at about 12 Hz. There is also a relatively strong third peak at about 17 Hz which may be an idler resonance.

Small changes in the value of the dancer damping coefficient C_1 have a major impact of the height of the resonant peaks. Its actual value is not known, and probably changes depending on the dancer's operating point. While a good design may keep this value low, it certainly never approaches zero, especially when the effects of the pneumatic air cylinder operation are considered. It was demonstrated through the equations that the damping constant C_1 has virtually no effect on the location of the frequency, only on the predicted relative amplitude. Therefore, for the purposes of this paper, modest values ranging from 2.8 down to 0.13 N per cm/sec were selected when plotting the model's predicted response. These values were "adjusted" so that the magnitude of the predicted peaks were comparable to the measured vales. Figure 12 shows the theoretical frequency response of the tension in the A span as a function of the dancer's viscous damping. The great reduction in the peak and broadening of the resonance is typical of other conditions for web properties and dancer rollers that were examined. Note that were the spring rate, damping, and resonance are not present, such as the single roller model, the amplitude agreement between the model and ,measured data is quite good.

The first peak is the result of the resonance of the translational dancer mass coupled with the web spring rate. The second peak is the result of the resonance of the rotational dancer inertia coupled with the web spring rate. Equation (2) assumed no slippage between the web and the roller. Note that for the roller to oscillate at the frequencies shown, large differential tensions are required across the dancer roller:

$$\frac{\Delta T \cdot R_1}{J_1} = \frac{d\omega}{dt} = 2\pi\, f\, A \sin(2\pi\, f\, t) \qquad (5)$$

By the belt equation:

$$\frac{T_{max}}{T_{max}} = e^{\mu\theta} \qquad (6)$$

For the rubber roll to web, this ratio is less than about 3. The absence of the second peak indicates that the web is slipping on the roller. This is confirmed by setting J_1 to zero in the model, which removes the torsional force of the roller inertia on the web. The model does not predict a second peak under these conditions.

CONCLUSIONS

I would fully agree with the assertion by Shelton that the "dynamics of dancers and load-cell rollers, even with several simplifying assumptions, are quite complicated." In particular, during testing, it was very difficulty to arrive at an experimental format that could isolate the response across all frequencies to a very small number of variables.

The experiments support the general shape of the frequency response predicted by Shelton's model. The absence of the second peak during dancer roller slippage is also confirmed. In general, the frequency of the measured peaks were higher than that predicted by the model. The predicted amplitude response could not be accurately verified due to the strong dependence and the inability to accurately characterize the dancer viscous friction and spring rate, although excellent agreement was noted for the fixed roller scenario where these parameters were not present.

The analysis for the dancer was verified, and appears sound. The author has high confidence in the model, and believes that it is a very useful tool in making tension control strategy selections. Due to the dependence on a large number of parameters that are difficult to accurately characterize, a 20% deviation in predicted frequencies may be expected.

A well designed dancer will always perform better at lower frequencies than a load cell control. However, there will always be a resonance at higher frequencies that can dramatically degrade the performance. This resonance is independent of speed, although higher line speed results in greatly increased damping. Lower dancer mass and roller inertia will result in improved performance. Increasing viscous damping can also be helpful, but there is an optimal amount. The dancer resonance is the key issue when selecting a dancer based tension regulation strategy. In winding or unwinding sections the disturbance frequency will vary with roll diameter, and with sufficient damping the low frequency benefits may outweigh the degradation at higher frequencies. In all cases the user must assure that the resonant frequency is either well above the expect operating frequency range, or is sufficiently damped to allow intermittent operation. Dancers are most suitable for short web spans and high web modulus. The dramatic change in performance of a dancer due to spring rate and viscous damping probably explains the large amount of controversy on this topic.

ACKNOWLEDGMENTS

I would like to thank 3M, and in particular my management, for supporting this research. Dr. John Shelton of Oklahoma State University has provided repeated and numerous insights, both on his analysis as well as this problem in general. I would also like to thank Dr. James Dobbs of 3M for numerous suggestions, analytical insight and experimental assistance. Ron Swanson, also of 3M, has provided valuable suggestions and review.

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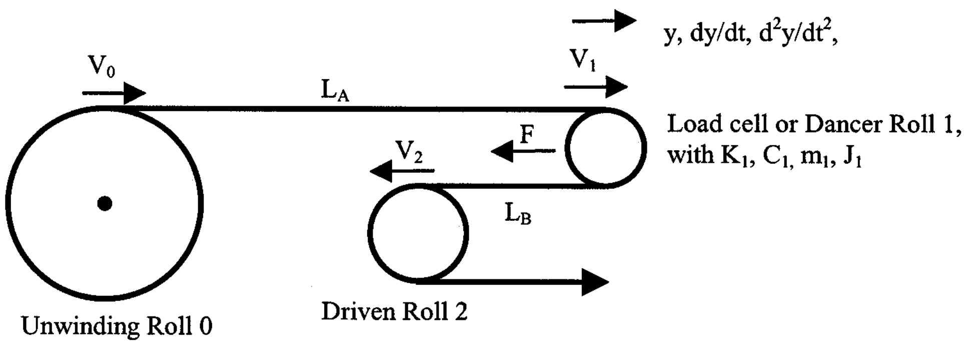

Figure 1, Unwinder or winder with dancer or load cell control

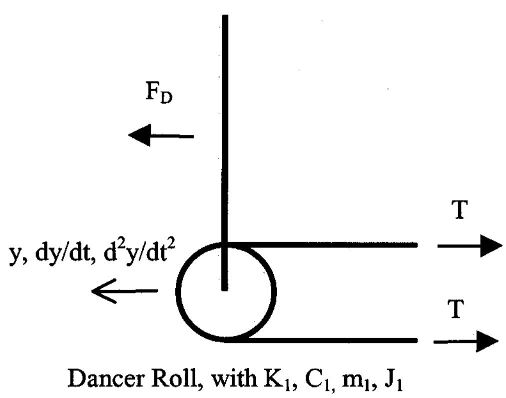

Figure 2, Dancer roller configuration

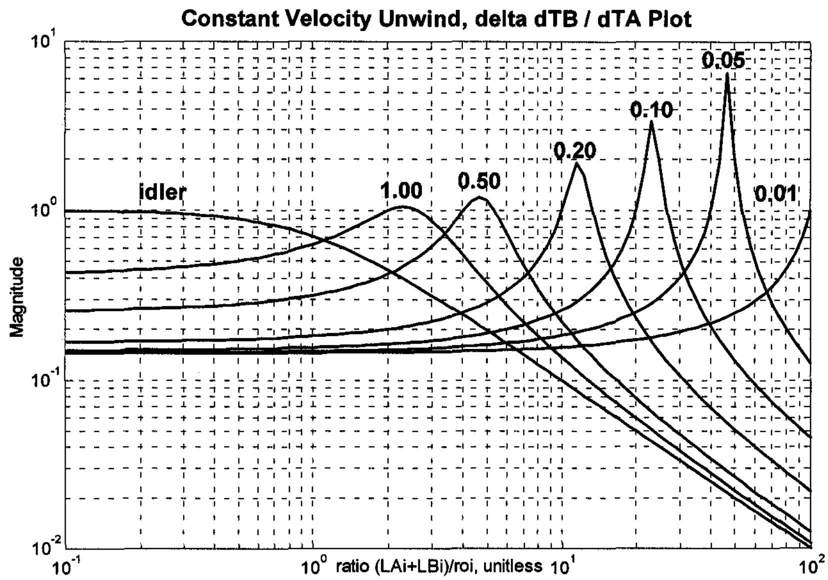

Figure 3 Amplitude Ratio delta T_B/ T_A

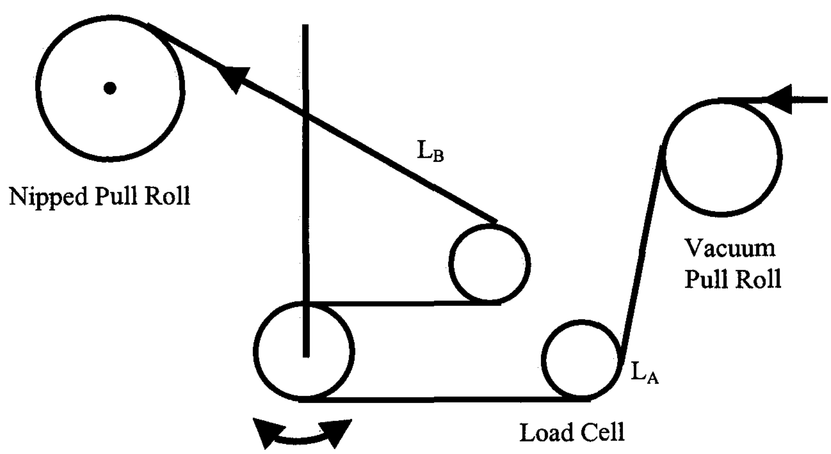

Figure 4, Experimental Setup

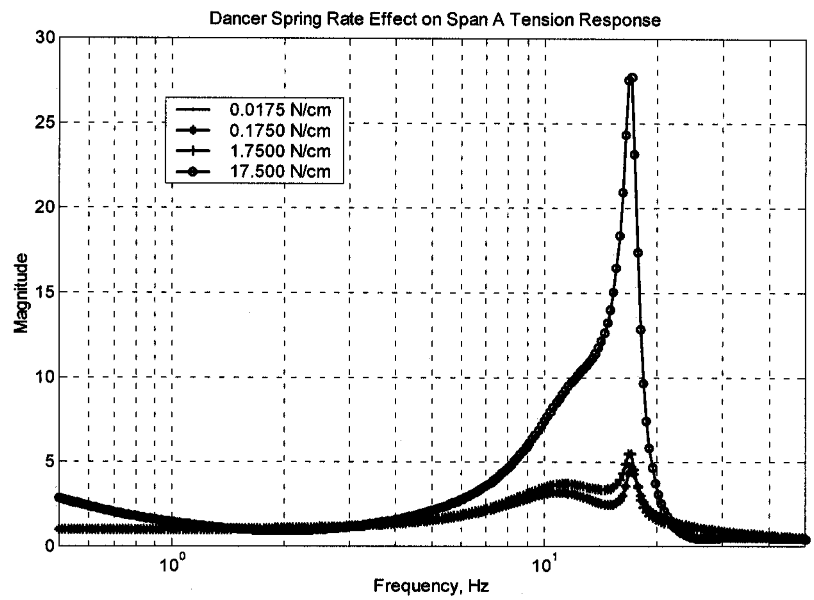

Figure 5, Effect of Dancer Spring Rate Resonant Response

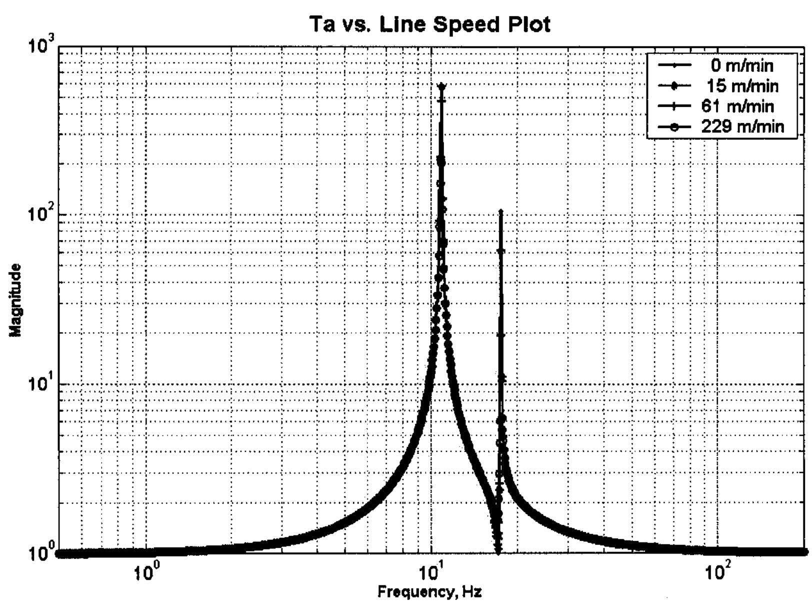

Figure 6, Predicted Response as a Function of Linespeed

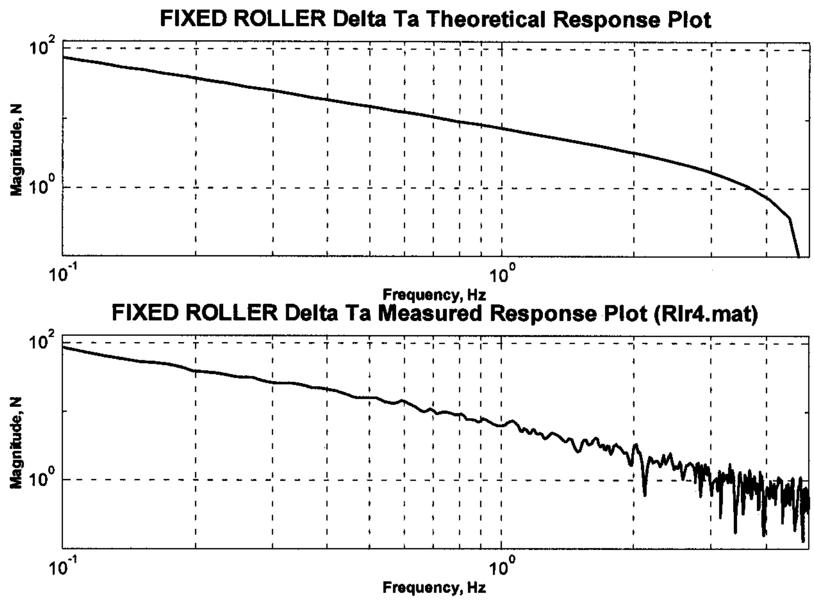

Figure 7, Tension Span T_a Response of a Fixed Roller

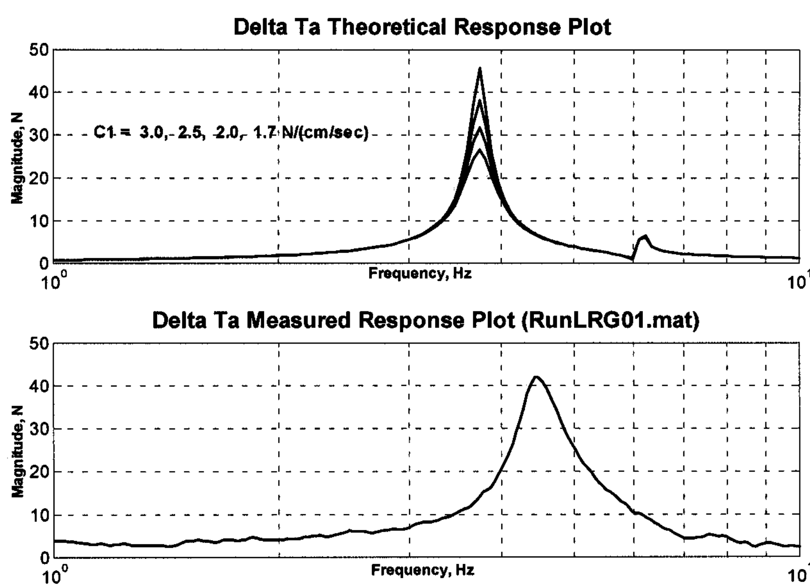

Figure 8, T_a Response, Large Roller with PET Web

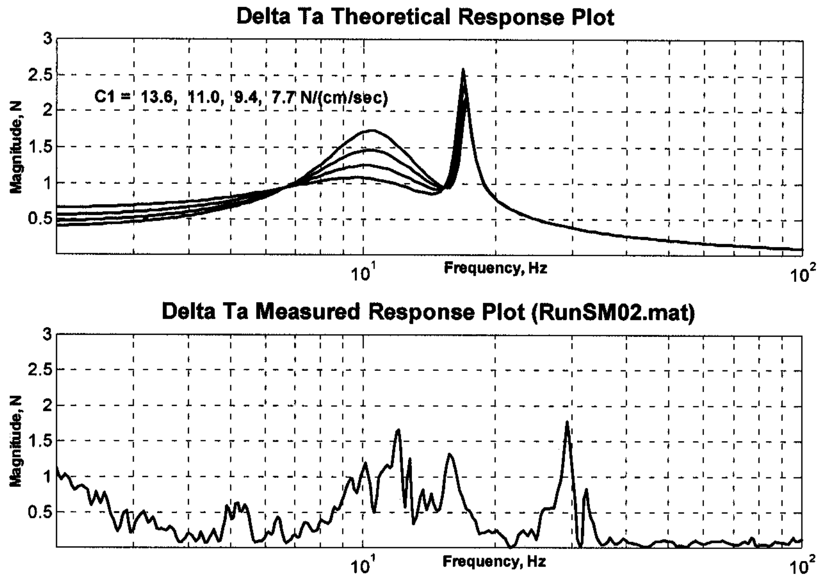

Figure 9, T_a Response, Small Roller with PET Web

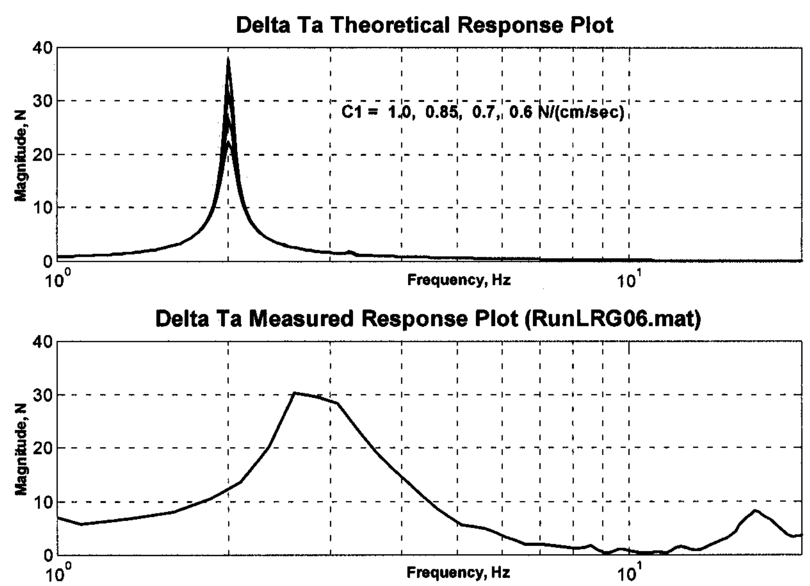

Figure 10, T_a Response, Large Roller with PE Web

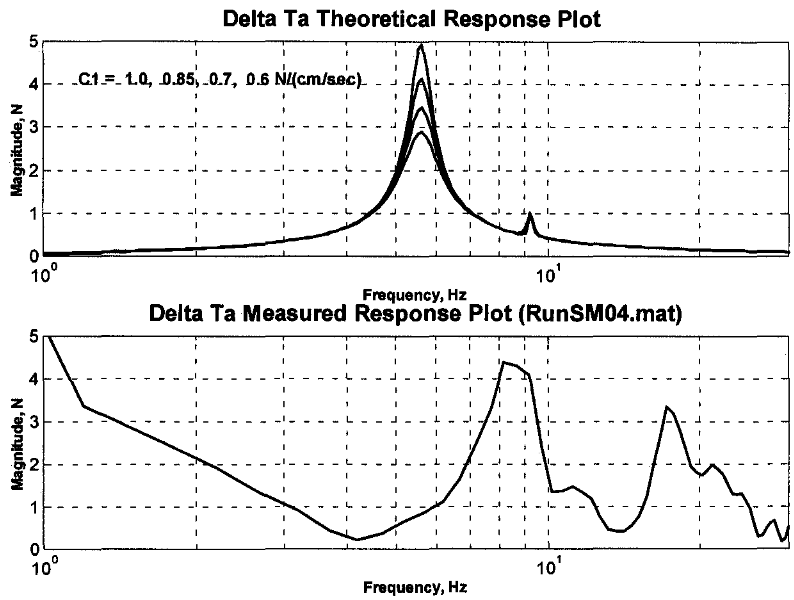

Figure 11, T_a Response, Small Roller with PE Web

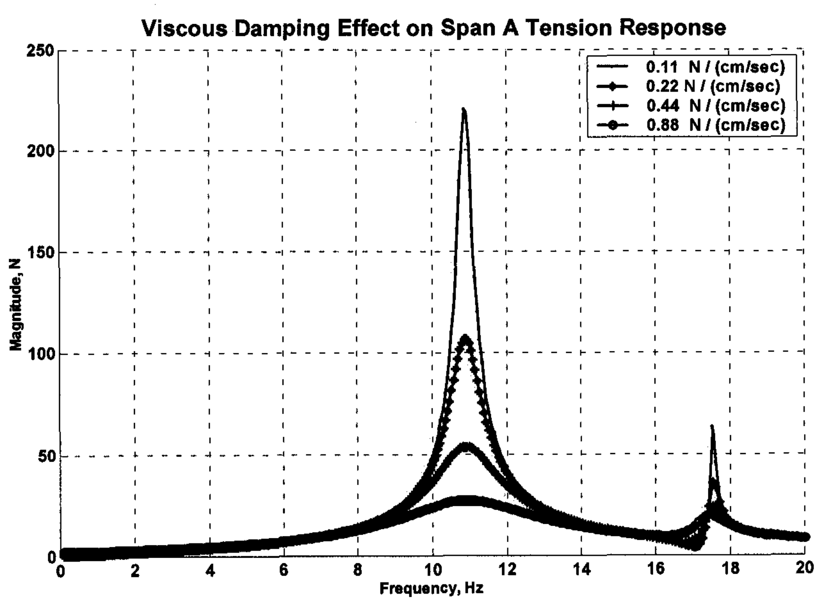

Figure 12, Effect of Viscous Damping on Resonant Response

PARAMETER	VALUE	UNITS	VALUE	UNITS
T_0 operating	67	N	15	lb_F
E PET	3.45	GP	500K	PSI
t PET	0.00254	cm	0.001	mils
w PET	30.5	cm	12	inches
Etw PET	26700	N	6000	lb_F
E PEN	0.324	GP	47K	PSI
t PEN	0.0056	cm	0.0022	mils
w PEN	40.64	cm	16	inches
Etw PEN	7360	N	1654	lb_F
L_A	429	cm	169	inches
L_B	196	cm	77	inches
L_T	625	cm	246	inches
m_1 Small	3.18	kg	7.00	lb_F
J_1 Small	0.0059	$kg\text{-}m^2$	0.052	$inch\text{-}lb_F\text{-}sec^2$
R_1 Small	5.59	cm	2.2	inches
m_1 Large	27.45	kg	60.4	lb_F
J_1 Large	0.26	$kg\text{-}m^2$	2.3	$inch\text{-}lb_F\text{-}sec^2$
R_1 Large	12.7	cm	5	inches
M arms	3.61	kg	7.95	lb_F

Table 1, Experimental Values

Considerations in the Selection of a Dancer or Load Cell Based Tension Regulation Strategy	**D. H. Carlson** – 3M Company, USA

Name & Affiliation	Question
K. Shin – KonKuk University	You compared dancer and load cell feedback systems, but they are two different systems. They have different objectives in the system. The dancer is usually used for disturbance rejection. A load cell is used for measuring web tension. Regarding your summary for the dancer, I have one question first. The dancer is usually generating a higher resonant frequency. If you have a low natural frequency system the dancer can also generate resonance in that low frequency, too. This is my first question, can you comment on that?
Name & Affiliation	**Answer**
D. Carlson – 3M Company	My view is that these are the open-loop disturbance rejection. This focuses on what they do without the control system. The control system will be limited in performance, mostly due to the drives. The bandwidth of the drives is only a few hz. Thus the performance of the system is limited by the bandwidth of the drives. This is typically lower than the frequencies due to the resonance exhibited by a dancer. With proper control system design, I can take advantage of that if I'm designing a load cell system versus a dancer system. There would be some differences in control design. The other consideration for a dancer system is that I would advocate having a load cell, also, because I'm going to need to know that tension. The final point is that the primary design goal in tension control is disturbance rejection because the steady state response always achieves the target. The measure of performance is how well we reject roll out-of-roundness, downstream web coupling, coupling between sections. Does that answer your question?
Name & Affiliation	**Question**
K. Shin – KonKuk University	The dancer can be used to reject the disturbance in open-loop system, but it can also generate resonance if we have a low natural frequency system when you design the system. Also you cannot control the tension with the load cell alone, you need an additional drive to control the tension. What is your point in comparing with the dancer system and load cell?

Name & Affiliation	Answer
D. Carlson – 3M Company	Suppose you have extremely sluggish drives with very poor response. The strategy with a load cell would be to measure the tension versus a command value and based upon the error an adjustment to the torque or to the velocity of the drive would be made. Again, I've got poor dynamics, poor mechanics, and very slow response. Your system is now dominated not so much by the drives but by the geometry and the web properties that exist. I can improve upon these to some degree but the largest improvements will result from an increase in the bandwidth of my actuators.