A COMPARATIVE STUDY ON ACTIVE AND PASSIVE DANCERS USED FOR ATTENUATION OF WEB TENSION DISTURBANCES

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ABSTRACT

A dancer mechanism, used in most of the web process lines, consists of a roller which is either connected to a fixed support by passive elements such as springs and dampers or is force loaded in opposition to the web tension. Dancer mechanisms are commonly used to attenuate web tension disturbances caused by uneven wound rolls, eccentric rollers, mis-alignment of idle rollers, and slacks in webs. A dancer mechanism is also used as a feedback element in a number of web tension control systems. The tension control system is driven by the variations in the position of the dancer mechanism as opposed to the variations in actual tension from the desired tension.

Since a substantial number of web process lines in web handling industries use dancer mechanisms, there is a need for a systematic comparative study of different types of dancer mechanisms and their applicability; the focus of this paper will be on such a study. Active and passive dancers will be compared using analytical models; a representative sample of the most common dancer mechanisms will be considered. The results of this analysis will assist in the selection of the dancer mechanism and its components, design of the dancer mechanism, and the effectiveness of a particular dancer mechanism to reject different types of web tension disturbances. Also, to substantiate the fundamental analysis, results from experiments, for certain situations, will be shown and discussed.

NOMENCLATURE

A	Cross-sectional area of web
B_f	Bearing friction
E	Modulus of elasticity
J	Polar moment of inertia of roller
K_i	Web span spring constant (EA/L_i)
L_i	Length of the i-th web span

M	Mass of the dancer roller
R	Radius of a roller
T_i	Change in tension from the reference
U	Dancer translational velocity input
V_i	Change in web velocity from the reference
X	Change in linear displacement of the dancer roller from the reference
b	Viscous friction coefficient at the dancer roller
k	Spring constant of the spring loading on the dancer mechanism
t_r	Reference web tension
v_r	Reference web velocity
τ	Time
τ_i	Time constant of a web span (L_i/v_r)
α, β, γ, η	System constants $(EA/v_r, J/R^2, B_f/R^2, \beta/\alpha)$

INTRODUCTION

Control of web tension within a small tolerance zone is an important aspect of web handling. The quality of the finished product is strongly affected by the web tension in the machine direction of the web transport system, winders, and processing sections. Even the distribution of tension across the width of the web affects the quality of the final product. Thus, the measurement and control of web tension has been an active area of research. The conventional method for measuring web tension is to measure the load the web causes on a roller [1, 2]. Though various tension measuring devices are reported in [1], the most common strategy used in a tension control system employs (i) a loadcell or (ii) a dancer roller [3, 4]. In the control schemes using a loadcell, an idler roll is mounted, generally on two beams, each supporting one end of the roller. Strain guages attached to each beam measure the deflection in the beam and the tension in the web is inferred from these deflections. The measured tension is compared to the desired tension set point and a PID controller calculates the corrective signal to the drive motor.

In the control scheme using dancer rollers, the dancer roller can be either (a) an active dancer roller or (b) a passive dancer roller. Figure 1 shows a schematic of the dancer subsystem. If the dancer roller is driven by an actuator to place it at a particular position, the mechanism is called an "Active Dancer Mechanism". The actuator positions the dancer roller based on the tension measured by a loadcell mounted on a roller downstream to the dancer roller. This scheme is expected to offer control engineer extra flexibility in controlling tension within tighter tolerance zones [5, 6, 7].

In the case of a passive dancer mechanism, the dancer roller is free to move, generally about a pivot or on linear slides. Instead of an actuator, a counterpressure is used. This pressure opposes the motion of the dancer roller. The control scheme using a passive dancer roller is driven by variations in the displacement of the dancer roller. A variation in the tension causes movement of the dancer roller and this displacement, in turn, causes a change in the tension. Usually, the dancer roller is kept under a counterpressure from a an air cyllinder to avoid continuous motion of the dancer roller for small changes the tension [3]. This counter pressure acts as the tension set point. Additional passive elements such as a spring, or a damper may be used to ensure contact between the dancer roller and the web and also to increase the speed of response of the dancer mechanism [8].

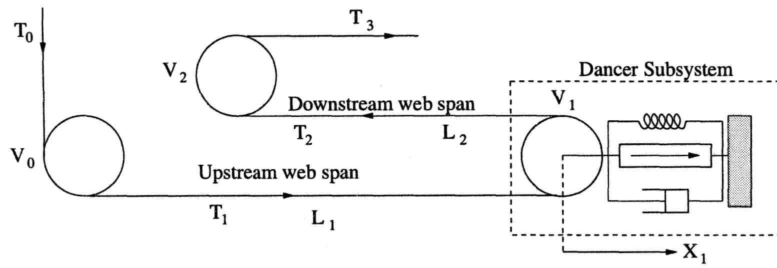


Figure 1. - Dancer Subsystem is an "Active Dancer" when the dancer roller is driven by an actuator otherwise it is a "Passive Dancer".

Extensive literature is available on control schemes (*e.g.,* [9, 8, 10, 11]) using both passive dancer rollers and load cells. A generalized dynamic model for the dancer subsystem is presented in [9]. An example of the inertia compensated dancer subsystem is presented in this work. It is demonstrated that an inertia compensated dancer mechanism acts as a low-pass filter and that a spring-loaded dancer mechanism normally responds to tension variations faster than a free roller. Effects of the PID gains for controller with a dancer mechanism are investigated in [10]. A comparison of the effects of the individual gains in various controllers using P-contol, PD-control, PI-control and PID-control is also presented in this work. Also, published literature is available on the performance of various control schemes using active dancer mechanisms [5, 6, 7, 12, 13]. These and the other representative literature listed in references at the end of this paper did not focus on the relative merits and demerits of the active dancer mechanism and passive dancer mechanism. This paper attempts to compare and contrast the active and passive dancer mechanisms with respect to their tension disturbance attenuation capability.

DYNAMIC MODELS

This section considers the dynamic models of the active dancer and passive dancer subsystems.

Active Dancer Subsystem

A typical active dancer subsystem shown in Figure 1 is considered and the input/output and the state space models are presented. The design of this subsystem is generic in the sense that it can be included as a subsystem anywhere in the web process line where precise tension regulation is required. This system contains web spans adjacent to the dancer roller in upstream and downstream directions and three rollers including the dancer roller. It is asumed that T_0 is the tension disturbance from upstream of the active dancer roller that is to be rejected/attenuated.

The linearlized dynamics of the active dancer system shown in Figure 1 are derived in [5, 6, 7] and are given by

$$\beta \dot{V}_0 = -\gamma V_0 + (T_1 - T_0) \tag{1}$$
$$\tau_1 \dot{T}_1 = -T_1 + T_0 + \alpha(V_1 - V_0) + \frac{\alpha}{\tau_1} X_1 + \alpha U \tag{2}$$
$$\beta \dot{V}_1 = -\gamma V_1 + (T_2 - T_1) \tag{3}$$
$$\tau_2 \dot{T}_2 = -T_2 + T_1 + \alpha(V_2 - V_1) + \alpha \left(\frac{1}{\tau_2} - \frac{1}{\tau_1} \right) X_1 + \alpha U \tag{4}$$
$$\beta \dot{V}_2 = -\gamma V_2 + (T_3 - T_2) \tag{5}$$
$$\dot{X}_1 = U \tag{6}$$

where $\beta = J/R^2$, $\gamma = B_f/R^2$, $\alpha = EA/v_r$, $\tau_1 = L_1/v_r$, and $\tau_2 = L_2/v_r$. Note that the variables in the linearized dynamics given above denote variations of the actual variables around their operating reference values. The input/output model for the active dancer subsystem can be derived by considering the translational velocity of the dancer roller ($\dot{X}_1$) as the control input and tension (T_2) at the roller immediately downstream of the dancer roller as measured output. Taking Laplace transform of equations (1)-(6) and simplifying, we obtain

$$T_2(s) = \frac{D_{ad}(s)}{C_{ad}(s)} U(s) + \frac{A_{ad}(s)}{C_{ad}(s)} T_0(s) + \frac{B_{ad}(s)}{C_{ad}(s)} T_3(s) \tag{7}$$

where the input $U(s)$ is the dancer translational velocity, $\eta = \beta/\alpha$, and

$$A_{ad}(s) = (\eta s + 1)^2, \tag{8}$$
$$B_{ad}(s) = (\eta s(\tau_1 s + 1) + 2), \tag{9}$$
$$C_{ad}(s) = \eta^2 \tau_1 \tau_2 s^4 + \eta^2(\tau_1 + \tau_2)s^3 + \eta(\eta + 2\tau_1 + 2\tau_2)s^2 + 3\eta s + 3 \tag{10}$$
$$D_{ad}(s) = \beta\eta\tau_1 s^3 + \beta\eta \left(1 + \frac{\tau_1}{\tau_2}\right) s^2 + \beta \left(3 + \frac{\eta}{\tau_2}\right) s + \beta \left(\frac{2}{\tau_2} - \frac{1}{\tau_1}\right). \tag{11}$$

Notice that, if $\tau_2 > 2\tau_1$, i.e., $L_2 > 2L_1$, then the constant term of the numerator polynomial, $D_{ad}(s)$, is negative, which results in a right-half-plane zero; the classical Routh criterion can be used to arrive at this conclusion. A detailed analysis is performed in [5, 6, 7] to show that $L_2 < 2L_1$ for effective tension disturbance attenuation.

An intuitive explanation of the effect of span length is given in the following. Assuming that the web is mostly elastic, it is common practice in the web handling community to model a web span as an elastic spring with a spring constant $K_n = E_n A_n / L_n$. The spring constants of the upstream and the downstream web spans to the dancer roller are $K_1 = EA/L_1$ and $K_2 = EA/L_2$, respectively, as shown in Figure 2.

If $L_1 > L_2$, $K_1 < K_2$, then any motion of the dancer roller results in larger tension variations in span 2 than in span 1. Thus, rejection of periodic disturbances from the spans upstream of the dancer roller into the spans downstream of the dancer roller is possible in this case. If $L_1 < L_2/2$, $K_1 > K_2/2$, then periodic dancer motion induces larger tension disturbances into span 1 than it rejects in span 2 due to feedback of tension T_2.

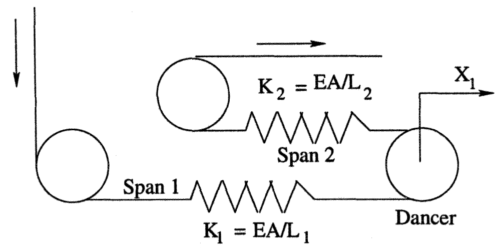


Figure 2. - Interpretation of the effect of span lengths.

Passive Dancer Subsystem

Passive dancers are widely used for tension disturbance attenuation and/or as position feedback elements. Substantial amount of work has been done in the past on passive dancers but a clear understanding of how a passive dancer can or cannot reject tension disturbances is not fully investigated. In this section, a simplified analysis is given to provide a simple understanding of how a passive dancer works and its limitations.

The dynamics of the passive dancer shown in Figure 1 is derived in [7, 14]. These results are presented below

$$T_2(s) = \frac{A_{pd}(s)}{C_{pd}(s)} T_0(s) + \frac{B_{pd}(s)}{C_{pd}(s)} T_3(s) \tag{12}$$

where

$$A_{pd}(s) = \left((\eta s + 1)(s^2 + \bar{b}s + \bar{k}) - \bar{\beta} s \left(s + \frac{1}{\tau_2} - \frac{1}{\tau_1} \right) \right) (\eta s + 1)(s^2 + \bar{b}s + \bar{k}), \tag{13}$$

$$B_{pd}(s) = (s^2 + \bar{b}s + \bar{k}) \left((\eta s(\tau_1 s + 1) + 2)(s^2 + \bar{b}s + \bar{k}) - \bar{\beta} s \left(s + \frac{1}{\tau_1} \right) \right), \tag{14}$$

$$\begin{aligned} C_{pd}(s) = {} & \left((\eta s(\tau_1 s + 1) + 2)(s^2 + \bar{b}s + \bar{k}) + \bar{\beta} s \left(s + \frac{1}{\tau_1} \right) \right) \times \\ & \left((\eta s(\tau_2 s + 1) + 2)(s^2 + \bar{b}s + \bar{k}) + \bar{\beta} s \left(s + \frac{1}{\tau_2} - \frac{1}{\tau_1} \right) \right) \\ & - \left((s^2 + \bar{b}s + \bar{k}) - \bar{\beta} s \left(s + \frac{1}{\tau_1} \right) \right) \left((\eta s + 1)(s^2 + \bar{b}s + \bar{k}) - \bar{\beta} s \left(s + \frac{1}{\tau_2} - \frac{1}{\tau_1} \right) \right). \end{aligned} \tag{15}$$

Equation (12) relates the effects of the tension upstream the passive dancer roller, T_0, and the tension downstream the passive dancer roller, T_3, on the tension T_2. Notice the difference between the dynamics of the active dancer, given by (7), and that of the passive dancer, given by (12). The first term on the right hand side of (7), which indicates the effect of the control effort on the tension T_2, does not appear in (12). The effect of of the passive dancer on the tension T_2 is solely determined by the coefficients in $A_{pd}(s), B_{pd}(s)$, and $C_{pd}(s)$. These coefficients in turn are determined by the mechanical features of the passive dancer (the mass of the dancer roller M, the spring constant k, and the viscous friction constant b), and the properties of the web material (Young's modulus E, the cross-section area of the web A). To illustrate the tension disturbance attenuation features of the passive dancer, we will use a simplified model of the passive dancer in the following.

Consider the simplified dynamics of the passive dancer as shown in Figure 3 ; the web is assumed to be an elastic spring with the upstream spring constant of $K_1 = EA/L_1$ and the downstream spring constant of $K_2 = EA/L_2$. The passive dancer system is

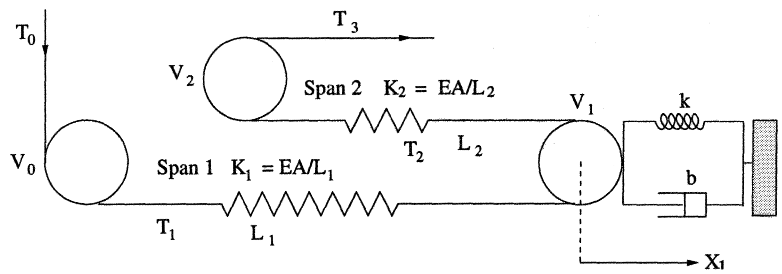


Figure 3. - Simplified dancer system.

taken to be a mass-spring-damper system. With these simplifications, the function of the passive dancer as a tension disturbance attenuator can be represented by the control block diagram shown in Figure 4, where w denotes the tension disturbance and x denotes the displacement of the passive dancer roller around the equilibrium point. Equilibrium

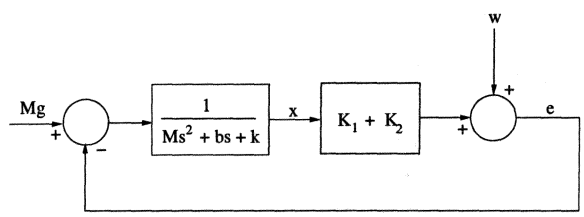


Figure 4. - An interpretation of the action of passive dancer tension control system. Passive dancer mechanism attempts to attenuate the error signal $e = T_1 + T_2$. w is the disturbance acting on the passive dancer roller. x is the displacement of the dancer roller.

point here means the position of the dancer roller when there are no disturbances, that is, when *w(t)=0.* The tension variation around the equilibrium point is given by $e(t) = T_1(t) + T_2(t)$. The passive dancer attempts to minimize $e(t)$ whenever the disturbance *w(t)* appears. In the Laplace domain, the relationship between *w(t)* and $e(t)$ is given by the following:

$$\begin{aligned} E(s) &= \frac{Ms^2+bs+k}{Ms^2+bs+k+K_1+K_2}W(s) + \frac{K_1+K_2}{Ms^2+bs+k+K_1+K_2}Mg \\ &:= G_{passive}(s)W(s) + G_{mass}(s)Mg \end{aligned} \quad (16)$$

where g is the acceleration of gravity, M is the mass of the dancer roller, b is the damping coefficient, k is the spring constant, and $E(s)$ and $W(s)$ are the Laplace transforms of $e(t)$ and *w(t)*, respectively.

The next section discusses salient features of the passive and active dancers with a view to compare these and bring out the benefits and limitations of each of these.

ACTIVE AND PASSIVE DANCERS: BENEFITS AND LIMITATIONS

This section compares the active dancer and the passive dancer based on the model of the active dancer developed in the previous two sections.

The following inferences may be made for the passive dancer:

- At the equilibrium point, the tension is given by

$$T_1 + T_2 = \frac{K_1 + K_2}{k + K_1 + K_2} Mg. \tag{17}$$

  Equation (17) indicates a proportional relationship between the mass of the dancer roller and the web tension. This relationship shows that the mass of the dancer roller cannot be chosen arbitrarily small.
- Define the tension attenuation factor ρ as

$$\rho(\omega) = \left| \frac{Ms^2 + bs + k}{Ms^2 + bs + k + K_1 + K_2} \right|_{s=j\omega}. \tag{18}$$

  From (18), it is observed that, when the tension disturbance is a low frequency signal, the passive dancer can reduce the disturbance to $\rho = k/(k + K_1 + K_2)$ times the amplitude of the original disturbance. Consider two extreme cases: (1) $k \to \infty$. In this case, the passive dancer behaves like a fixed roller. Obviously, $\rho = 1$, no tension disturbance attenuation can be achieved. (2) $k \to 0$ and $b \to 0$. The passive dancer behaves like a floating mass block. In this case, the transfer function between the tension disturbance $W(s)$ and the tension $E(s)$, $G_{passive}(s)$, is given by

$$G_{passive}(s) \approx \frac{Ms^2}{Ms^2 + K_1 + K_2}. \tag{19}$$

  It can be seen that $G_{passive}(s)$ represents an undamped second-order system. For low frequency tension disturbance, $\rho = 0$; this implies that the disturbance is completely attenuated. However, the drawback of the zero-damping is that the dancer will exhibit sustained oscillations even when subjected to a small tension disturbance. Also, in this case, the mass M should be chosen according to $M = (T_1 + T_2)/g$. This means that a change in the reference web tension of the processing line requires a change of the mass of the dancer roller.
- The passive dancer has better tension attenuation capability for high modulus web materials since both K_1 and K_2 are large.
- To make ρ small, that is to improve tension disturbance attenuation, k should be chosen small.
- There exists a resonant frequency at $\sqrt{(k + K_1 + K_2)/M}$. To avoid resonance, the resonant frequency should be high as compared to the frequency of potential tension disturbances. Consequently, the passive dancer should be designed with large k and/or small M. However, to achieve large disturbance rejection, small k is required. Hence, there is a trade off between the ability to attenuate tension disturbances and resonance avoidance.
- The tension disturbance attenuation factor $\rho(\omega)$ is approximately one in the high frequency region and thus the passive dancer does not have any tension disturbance attenuation capability at high frequencies.

Contrary to passive dancer, active dancer can have different controller implementations with the same mechanical structure. However there still are some considerations on designing and application of the active dancer. As already noted, for efficient tension disturbance attenuation using an active dancer, it is necessary to construct the active dancer system such that the downstream span length is smaller than the upstream span length, which means that the active dancer should be installed closer to the downstream roller than to the upstream roller. This statement comes from the analysis of the simple proportional control on the active dancer. The active dancer generates tension disturbance to the upstream web while controlling the downstream web tension. Apart from the consideration of the limitation on the relative lengths of the downstream and upstream web, the following factors should also be considered.

As mentioned before, the control signal appears explicitly in the dynamics of the active dancer system as given in (7). This gives the active dancer the additional flexibility to control the web tension as compared to the passive dancer. A few aspects of the active dancer are listed below:

- The behavior of the controlled web tension strongly depends on the control algorithm $u(t)$, and in turn the controller strongly depends on the quality of the feedback signal. The feedback signals for the control of the tension on a moving web are usually obtained from load cells which support a roller wrapped by the web, or from the position of a force-balanced floating roller ("dancer"). The dynamic response of such devices is severely limited because the mass of rollers, as required for economy and practicality of fabrication and operation [2].
- The web span between the active dancer and the load cell immediately downstream the active dancer should not be too long. This is because the tension feedback comes from this load cell. If the load cell is too far from the dancer, the feedback tension signal has large time delay.
- PID controller has very limited capability to attenuate the tension disturbance. Advanced controller is required in the situation where high quality tension performance is needed. Advanced controller always needs more feedbacks. For the active dancer, both measurements of the dancer actuator velocity and dancer position or either of them should be available.
- The active dancer actuator is also a major consideration. The bandwidth of the actuator should be at least five times above the frequency of the tension disturbance. Otherwise, actuator will not respond to high frequency control signals and exhibits saturation. Actuator saturation will invalidate the control effort from the controller. For the low bandwidth actuator, PID control and advanced controller show similar performance. The load capability is another consideration of the active dancer actuator which affects the cost of the total implementation.
- Active dancer mechanism almost invariably involves a gearing and/or other mechanism to convert the rotary movement of the actuator to the translatory movement of the dancer roller. This mechanism usually introduces nonlinearities such as backlash and friction. Such nonlinearities complicate the controller algorithm, and in extreme cases, may destroy the control effort.

The tension disturbance in the web process line is often determined by the constructional and/or operational features of the web process line itself such as the misalignment of rollers and web velocity. Designing and implementing active dancers needs *a priori* knowledge of the web process line and the consideration of the tradeoff of the cost and and the performance of the active dancer.

A COMPARATIVE EXPERIMENTAL STUDY

To compare the performance of the active dancer and the passive dancer and to validate theoretical analysis presented , experiments were conducted on two web process lines: (i) experimental platform at Web Handling Research Center (WHRC), Oklahoma State University (OSU), and (ii) Web process line at Fife Corporation, Oklahoma City. The experimental platforms are briefly described here followed by the results of the experiments.

OSU Experimental Web Platform

A sketch of the open-architecture experimental web platform together with an active dancer system is shown in Figure 5 and a picture of the complete experimental platform is shown in Figure 6. The web platform consists of an endless web line with a number of rollers, a passive dancer, an active dancer system, and web guides for maintaining lateral position. The term endless web line refers to a web line without unwind and rewind rolls. This type of platform mimics most of the features of a process section of a web process line. Mechanical components used in the platform include sixteen idle rollers, a master

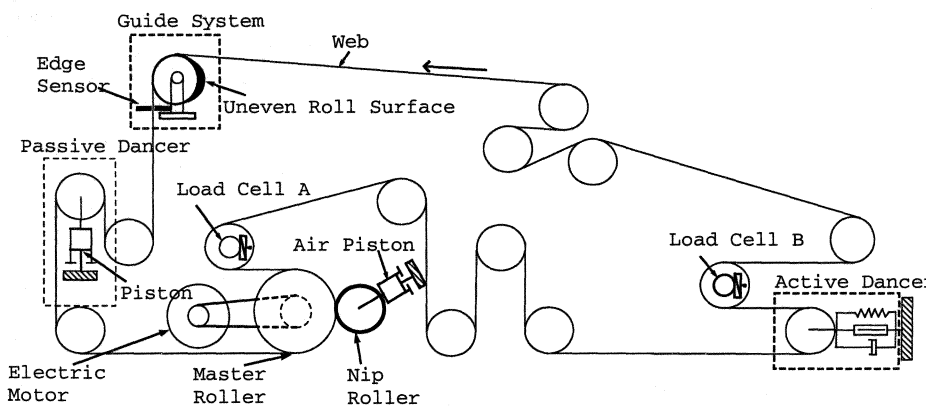


Figure 5. - Sketch of the experimental web platform at OSU.

speed roller with a nip roller, an electric motor, a passive dancer system with a pneumatic cylinder, an active dancer system, a Fife Kamberoller guide, and two load cells to measure the web tension at different locations along web path. The width of each roller is 8 inches and the diameter is 5 inches, except for the master speed roller, which has a diameter of 10 inches. Nip roller is used to reduce slip during start-up. The master speed roller sets the desired transport velocity of the web. Passive dancer system has a Bellofram super cylinder with a bore of 2.3 inches and a stroke of 1.8 inches. Air pressure in the pneumatic cylinder of the passive dancer system is used to set the desired reference web tension. The active dancer system consists of an electro-mechanical actuator and a guide

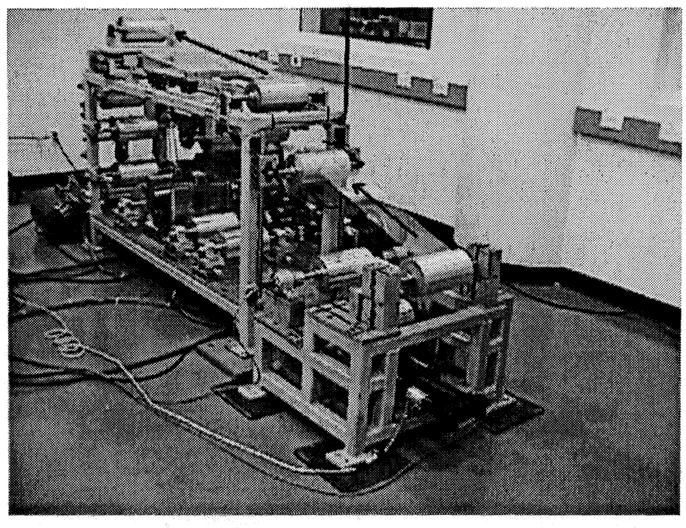


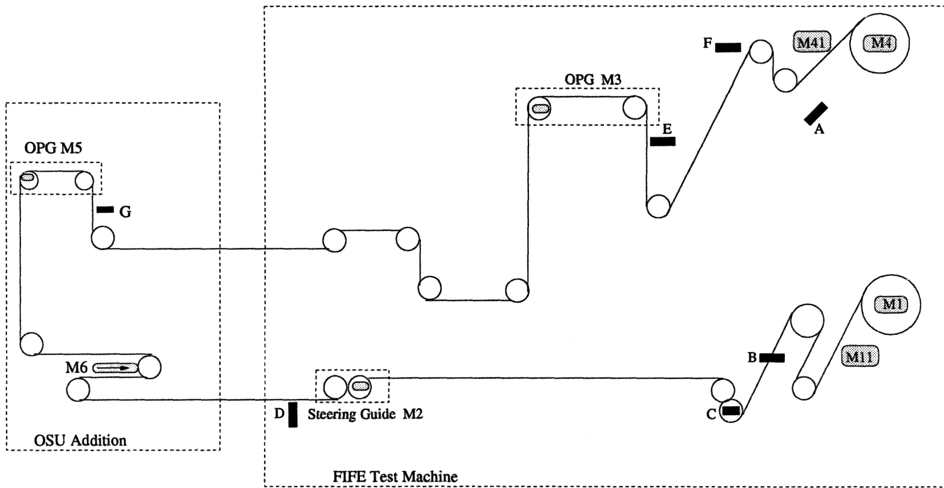
Figure 6. - Picture of the experimental web platform at OSU.

way with the dancer roller mounted on it. The measured signals on the experimental web platform (see sketch in Figure 5) include the velocity of the web, tension from both load cells, velocity of the active dancer actuator, and lateral position of the web downstream of the guide roller.

Periodic tension disturbance upstream of the passive dancer roller is created by introducing an uneven roll surface on the guide roller as shown in Figure 5. The fundamental frequency of the periodic tension disturbance for a given an uneven roll surface increases with the increase in web transport speed and the amplitude of the tension disturbance increases with the eccentricity of roller.

Web Process Line At Fife Corporation

A schematic of the web process line at Fife Corporation is shown in Figure 7 and a picture is shown in Figure 8. This process line has seventeen rollers including the guide rollers. The unwind and rewind rolls are controlled by a brake and a clutch, respectively. Lateral sensors next to the unwind and rewind stands measure the lateral position of the web coming out of the unwind roll and going into the rewind roll. Based on these measurements, the unwind and rewind stands have the capability to displace laterally to provide lateral correction to the web. The unwind roll is braked based on feedback from the load cell located immediately downstream of it. This maintains the web reference tension in the web line. An ultrasonic sensor (Magpower US-2) measures the radius of the rewind roll. A steering guide and a displacement guide within the web line provide lateral correction to the web; edge sensors downstream of the guides measure the lateral position for feedback to the guide motor. The active dancer module used in Fife process line is shown in Figure 9. This module consists of a displacement guide and a active dancer system. Such a module can be included at any section in a given web process line where tension disturbances are to be attenuated. An electro-hydraulic actuator is used in the active dancer system in Fife web process line. This actuator has larger bandwidth



Motors/ Drives	Sensors
A – Magpowr US–2 (Ultrasonic) B – SE–31 Edge Sensor C – Magpowr TS–150 FC Load cell D – SE–17 Infrared Edge sensor E – SE–34 Edge sensor F – SE–26 Edge sensor G – Ultrasonic Sensor	M1 – Unwind Roll Motor M11 – Lateral Motor for Unwind roll M2 – Steering Guide Motor M3 – OPG Motor M4 – Rewind Roll Motor M41 – Lateral Motor for rewind roll M5 – Motor for OPG M6 – Hydraulic Actuator

Figure 7. - Schematic of the web process line at Fife Corporation.

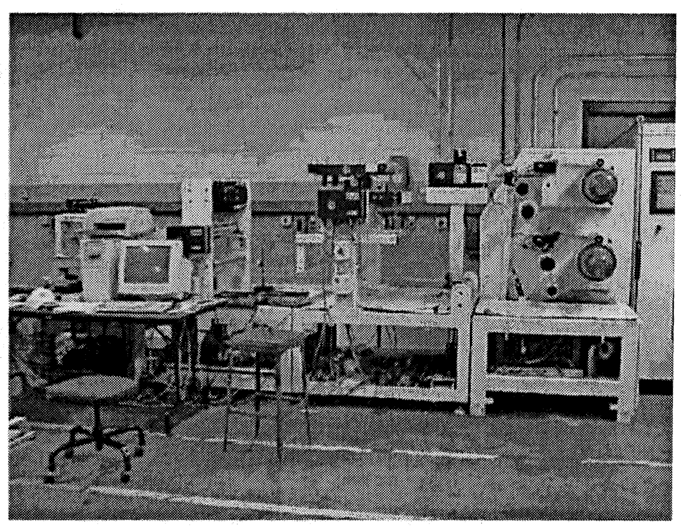


Figure 8. - Picture of the web process line at Fife Corporation.

than that of the electro-mechanical actuator used in OSU experimental platform.

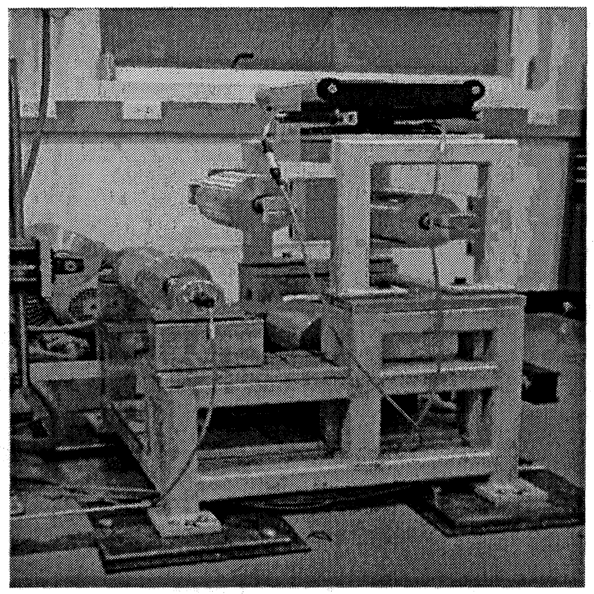


Figure 9. - Active dancer module. This module shows the configuration of active dancer subsystem used in the experiments. The bottom most roller appearing within the structure is the dancer roller. A displacement guide to correct the lateral displacement is also shown in this picture.

Results From OSU Experimental Platform

This section presents experimental results on the OSU experimental platform. The performance of the passive and active dancer when they are subjected to tension disturbances at various frequencies are compared. Three sets of experiments were conducted at different web transport velocities: (i) Both passive dancer and active dancer roller are fixed. (ii) Active dancer roller is fixed and only passive dancer is used for attenuating the tension disturbance. (iii) Passive dancer roller is fixed and only active dancer is used for attenuating the tension disturbance. In each set of experiments, two loadcells (A and B) measure the tension disturbance entering the process line.

Figures 10 to 13 show the results of experiments. Reference tension in all experiments is 8.2 lbf. Figure 10 shows the tension measured by loadcell A at the web speed of 56 fpm. The top two plots show the FFT of the deviation from the mean tension and the tension itself without attenuation by either passive or active dancer. The middle two plots show attenuation obtained by using passive dancer only. The bottom two plots show attenuation obtained by using active dancer only. From the figure, it can be seen that the fundamental frequency of the tension disturbance at 56 fpm is around 0.74 Hz. Passive dancer attenuated the tension disturbance at 0.74 Hz by around 28% and active dancer by around 75%. Figure 11 shows the tension disturbance attenuation indicated by loadcell B. The passive dancer attenuated the tension disturbance by around 32% and active dancer by around 78%. Figures 12 and 13 show the tension measured by loadcells A and

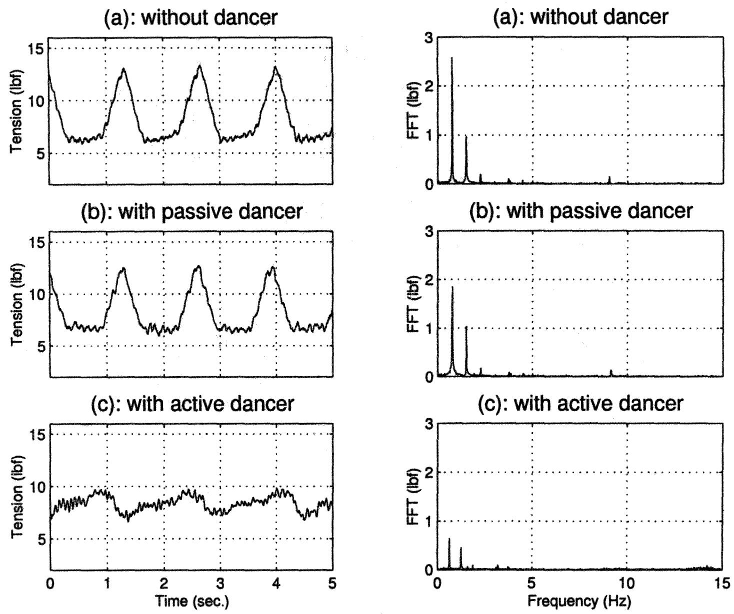


Figure 10. - Tension measured by loadcell A. The fundamental frequency of disturbance is around 0.74 Hz.

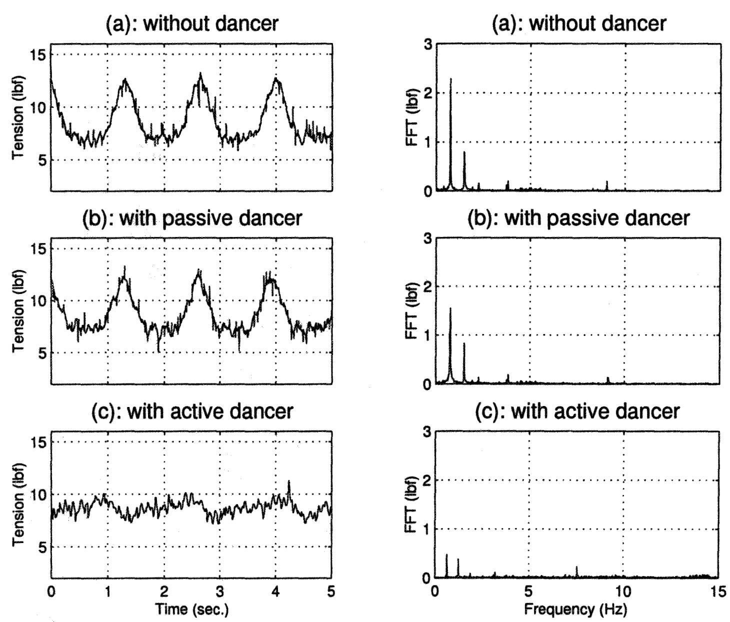


Figure 11. - Tension measured by loadcell B. The fundamental frequency of disturbance is around 0.74 Hz.

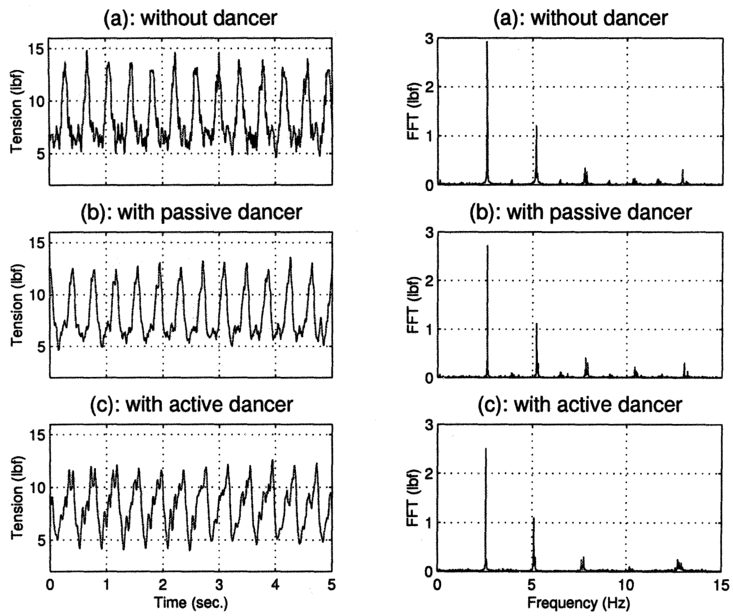


Figure 12. - Tension measured by loadcell A. The fundamental frequency of disturbance is around 2.7 Hz.

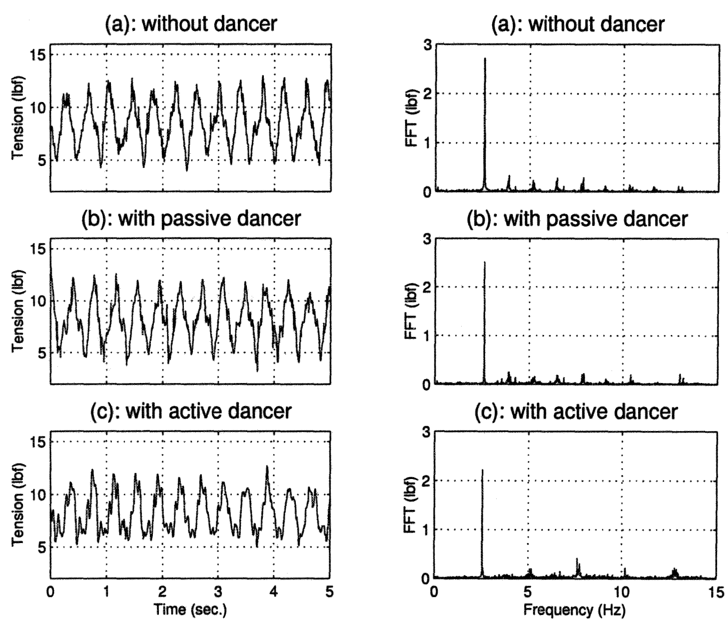


Figure 13. - Tension measured by loadcell B. The fundamental frequency of disturbance is around 2.7 Hz.

B when the web is running at 200 fpm. The fundamental frequencies of the tension disturbance is around 2.7 Hz. From these two figures, it is seen that there is not much tension disturbance attenuation either by passive dancer or active dancer. This verifies the analysis of the passive dancer made in section , that is, passive dancer can attenuate low frequency tension disturbances, but not high frequency tension disturbances. Although active dancer exhibited better result at low frequency (0.74 Hz) than passive dancer, at high frequency, similar performance was not seen. This can be attributed to the fact that the electro-mechanical actuator used in the active dancer has a very low bandwidth (around 7 Hz). Also, because the experimental platform connsists of an endless web line, the tension disturbances propagate to both upstream and downstream diretions of the misaligned roller. This may limit the performance of the active dancer system. It is expected that an actuator with a higher bandwidth can reject tension disturbances to a greater extent. To verify this, experiments were conducted using an electro-hydraulic actuator which has larger bandwidth. The results of the experiments are shown in next section

Results From Fife Web Process Line

Experiments were conducted on Fife web process line at different line speeds and two different reference tensions. The tension disturbances were created by using an uneven surface roller in the process line. The fundamental frequency of the tension disturbance is determined by the web travel speed. The tension signals from loadcell (**G**) located immediately downstream of the active dancer roller were acquired using a data acquisition system. Three controllers were used for controlling the active dancer, namely, (i) a Proportional-Integral controller (PI), (ii) an Internal Model Controller (IMC), and (iii) a Self-Tuning Controller (STC)[7].

Figures 14 and 15 show a representative sample of the performance of the three controllers (PI, IMC and STC) at a reference tension of 15 lbs and web speed of 300 fpm and 700 fpm, respectively. Figure 14 shows the results of the experiments in which the active dancer is switched on at around 10 seconds. Results from all three controllers show good tension disturbance reductions. Similar results can be seen in Figure 15 when the web travel speed was about 700 fpm.

The results of the experiments indicate that the active dancer is able to reject the tension disturbances by around 30–60%. The results of the controllers for the reference tension of 15 and 20 lbs at different web speeds are summarized in Figure 16. More experimental results can be found in [7].

Experimental results from OSU platform and Fife web process line indicate that the active dancer rejected tension disturbances effectively. The bandwidth of the actuator in the active dancer is a very important consideration in designing an active dancer. A higher bandwidth actuator can reject higher frequency tension disturbances. Whereas, the passive dancer is effective only in the region of low frequencies. Also, the active dancer gives the user an additional flexibility in designing advanced controllers to provide better tension regulation.

SUMMARY AND CONCLUSIONS

In this paper, analytical and experimental study is undertaken to compare the performance of passive dancer and active dancer in attenuating tension disturbances.

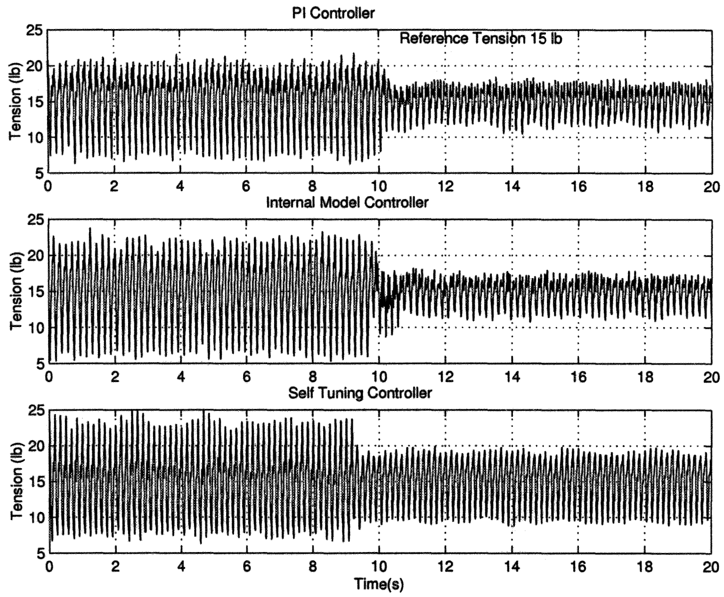


Figure 14. - Tension disturbance attenuation (Fife web process line). The fundamental frequency of the tension disturbance is ≈ 5.5Hz.

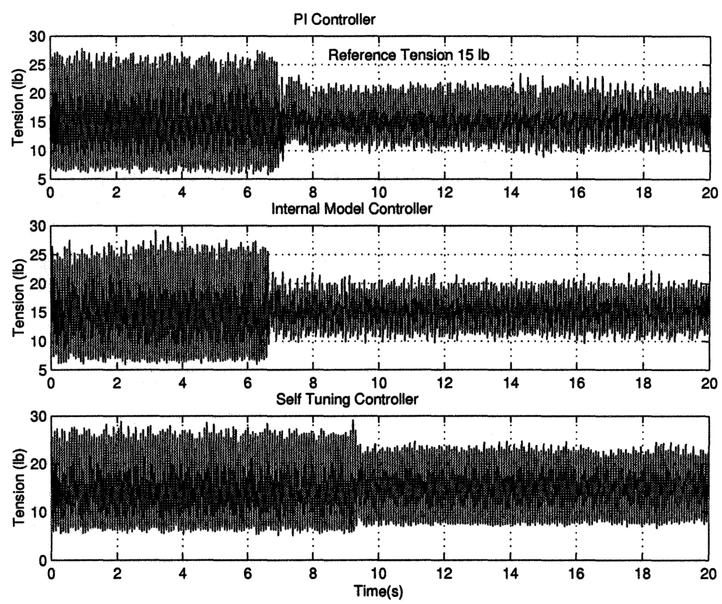


Figure 15. - Tension disturbance attenuation (Fife web process line). The fundamental frequency of the tension disturbance is ≈ 7.5 Hz.

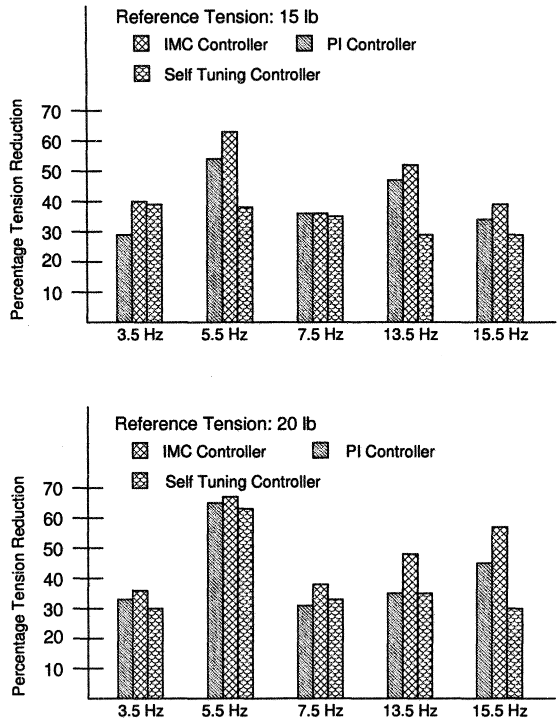


Figure 16. - Summary of tension disturbance attenuation (Fife web process line).

A simplified model of passive dancer is developed and this model predicts that passive dancer is effective in low frequency region. Whereas, the active dancer has the capability of tension regulation over wider frequency regions subject to the limitation of the bandwidth of the actuator.

This paper did not investigate the effect of the sensor (mainly loadcell) characteristics on the tension control. Aspects of the sensor used as the feedback for active dancer controller, such as, response time, the location in the process line, and the bandwidth, are other important considerations that need to be taken into account.

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