THE MECHANICS OF WEB HANDLING

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Preface

The goal of web handling is simple: to move a web as quickly and as reliably as possible through a machine. Web *handling* seeks to preserve a web as contrasted to *manufacturing* which creates it and *converting* which modifies it. Preservation might include, for example, geometries so that the web remains free of breaks, cracks, wrinkles, and unintended size changes. Preservation also might include protecting the web's properties or even its passengers such as coatings and inks.

The goal of preserving a web seems so modest that it has not always had the attention it deserves. This has led to machines which have been designed and operated as much by mythos or trial and error as by science. Training is scarce, literature is scattered, and fallacies abound. For example, many believe spiral grooving on a roller spreads when it in fact tends to contract the web. Similarly, rollers or drives are added to help "support" or "carry" the web through a machine when in fact they may burden the web. Draw or speed controlled drives are believed to *control* web tension when in fact they only *modify* tension.

Components are added to a machine with the full expectation that they will perform their advertised role. However, a guide may not control edge position and may even wrinkle the web if appropriate span lengths and motions are not used. Most spreaders will not spread unless there is sufficient or complete web to roller traction. Finally, while any winder will make a roll, only those that attend to certain details and constraints will make a consistently good roll.

Fortunately, web handling can provide the guidelines and rules for successful movement of a web through the machine to avoid these and many other problems. These rules are based on mechanics or science which underlies the behavior of webs. In some cases, such as winding, the science was developed uniquely for web handling. In other cases, such as avoiding web breaks, tools were borrowed from many disciplines and industries.

With attention to web handling, some machines have obtained truly impressive productivities. For example, paper is manufactured and coated at speeds in excess of 5,000 ft/min and wound at speeds in excess of 10,000 ft/min. Printing presses have run 100 consecutive rolls of paper of 5 miles long each without an interruption from a web break. Routinely, people handle films thinner than 1/10th of the thickness of a human hair without distortion, or materials as delicate as tissue without tearing. Many can hold geometries and positions to 1 part in 1,000 and a few to 1 part in 10,000.

As might be expected, these achievements might be easier for some materials than others. The difference, however, is due to physical properties rather than chemistry. This is why fluffy nonwovens tend to behave similarly to paper tissue though one is made from petroleum and the other is made from trees. This is why film and paper have difficulty with air entrainment problems. Knowing certain easily measured physical properties of the web gives clues as to possible difficulties and is necessary for optimizing a machine for a particular web.

Acknowledgments

This book completes my trilogy which began with the *The Mechanics of Winding* (1994) and *The Mechanics of Rollers* (1996). Those who enjoy my writings will note that I have already begun a book on *Critical Thinking in Converting* which I expect to finish by 2000. Here, I will teach problem solving tools and techniques using examples from the web industries. I am most enthusiastic about this new project.

First, I would like to reiterate my appreciation for TAPPI. If it were not for TAPPI, our knowledge of web handling would be more limited and less accessible. All one has to do is to look at the bibliography listings at the ends of the chapters to see the major role that TAPPI has played. Specifically, I would like to thank Karen Daniel for making the publication of my books as painless as possible.

Second, I again owe a debt to the Web Handling Research Center at Oklahoma State University. Their theoretical and experimental studies have laid the foundations for much of the science of web handling. On a more personal level, I played a small role in helping start the WHRC and at the same time was one of their first products.

Third, I would like to express my gratitude to the following individuals and their companies who have provided technical information and technical review of one or more of the chapters in this book. It is my belief that it is always wise to have a "sanity check" by others, especially when presenting new, controversial, or subjective material.

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25. Brandenburg, G. *New Mathematical Models for Web Tension and Register Error.* Proc. 3rd Int'l IFAC Conf. on Instrumentation, 1976.

26. Puckhaber, Charles F. *Print Registration Control in a Rotogravure Printing Press Combined with Extrusion Coating or Lamination Processes.* Tappi J., vol 78, no 3, pp 144-154, March 1995.

27. Tasch, J.J. *Improved Quality through High-Precision Register Controls.* Newspaper Technology, January 1990.

28. Weiss, Herbert L. *Consistent Web Print Length Requires Tension Controls.* Paper, Film & Foil Converter, vol 66, no 12, pp 68-69, December 1992.

29. Roisum, David R. *The Mechanics of Winding..* TAPPI PRESS, #0101 R-236, 1994.

30. Lucas, Robert G. *Dishing in Winding Rolls of Paper.* Tappi J., vol 60, no 7, pp 121-125, July 1977.

31. Roisum, David R. *Nip Induced Defects of Wound Rolls.* TAPPI Finishing and Converting Conf. Proc., Chicago, pp 89-98, October 2-5, 1994.

32. Roisum, David R. *What causes a roll to telescope?* Converting Magazine, Web Works Column, pp 30, November 1997.

33. Schulze, Peter. *The Multi-Ply Effect (Stair Cut): A Sheeting Problem.* Tappi J., vol 54, no 8, pp 1305-1308, August 1971.

34. Stack, K. D. and LaFleche, J.E. and Benson, R. C. *The Effects of Nip Parameters on Media Transport.* Proc. 3rd Int'l Conf. on Web Handling, Oklahoma State Univ., June 18-21, 1995.

35. Roisum, David R. *How can I Reduce Waste?* Converting Magazine, Web Works column, pp 20, September 1994.

36. Roisum, David R. *How can I Improve Productivity?* Converting Magazine, Web Works column, pp 22, July 1994 and pp 24, August 1994.

37. Roisum, David R. *Waste is not a dirty word.* Converting Magazine, pp 64-66, August 1997.

38. Roisum, David R. *Should I Question the Value of Process Controls?* Converting Magazine, Web Works column, pp 20 & 64, January 1997.

39. Smith, R. Duane. *Roll and Web Defect Terminology.* TAPPI PRESS, 0101R234, Atlanta GA, 1995.

Chapter 4

Tension and Tension Control

In this chapter, we discuss why tension is important, what is an appropriate level of tension, and how can it be controlled. Covered are draw/speed, dancer, load cell, drive torque, and other means of tension control.

Why Web Tension Is Important

More often than not, web handling requires knowledge and control of web stresses at every point in a machine, as well as at all times. The stresses imposed upon a web could be in the MD (machine direction), CD (cross direction), and ZD (z direction), as well as in three shear directions. Of these six, the MD stresses set by tension control is the best understood and most frequently considered direction.

Tension is a very important web manufacturing and process setting. If the tension is too low, the web will tend to flop and **flutter** and will tend to shift and the **path** will wander sideways going through the machine and may lose **traction** on rollers. Conversely, if the tension is too high, portions of the web may be damaged due to local yielding, or **web breaks**[1] may increase if the material is brittle.

Tension also is important because it affects **geometry** such as changes in length,[2] width[3] and registration,[4] curl,[5] and layflat.[6,7] For example, increasing the web line tension will increase the apparent length and decrease the apparent width of the web as measured at that point. Ironically, the web will then appear shorter and wider as measured by the customer. If tension is increased on only one ply of a laminate, the composite web will tend to curl toward that side.

Tension also can affect the tendency toward **wrinkling**.[8] For example, increasing the line tension can decrease the wrinkling resulting from roller misalignment. However, increasing tension can increase wrinkling on very thin or very nonuniform webs, as well as on machines where there is excessive roller deflection.

Web tension also determines in part how much **air** is pumped in between the web and roller[9] which can cause a loss of traction, or into a winding roll[10] which can cause roll defects. At higher speeds, a low tension web will tend to lift off a large diameter roller. This results in ever decreasing web/roller traction as speeds increase beyond as little as a few hundred FPM (ft/min). At speeds beyond a few thousand FPM, rollers often will need to be grooved or heavily textured to provide a passage for the air to improve traction. This is similar to the tread on car tires which is needed to reduce hydroplaning when braking on a wet surface.

Finally, web tension affects **rollers**, which are the foundation of web machines.[11] First, web tension indirectly affects the required roller diameter due to deflection. Second, web tension (as well as wrap angle and coefficient of web/roller friction) determines how much torque can be transmitted into the roller to overcome drag, windage, and inertia for web driven idler rollers or how much torque can be applied by an externally driven roller before the web breaks loose from its roller. Third, a required change in tension from one section of machine to another requires a drive. Fourth, the required precision of tension control determines how many undriven rollers can be used in a section.

As we see from these few examples, tension control is fundamental to many aspects of web handling. Thus, we must ensure that the tension stay within appropriate bounds as the web progresses through a machine. Web tension must be considered carefully at the machine specification and design stages, as well as during process troubleshooting.

In this chapter, we will primarily consider the nominal web tension and how it varies down through a machine due to drive points as well as roller drag and inertia. We will consider dynamics only briefly here. Variations of tension across the width of the web will be covered in other chapters.

Choosing a Process Tension

Web tension is expressed as a lineal force per unit of web width, instead of total tension or other arbitrary values. The appropriate units of tension in the Metric system is kN/m and in the English system is lb/in. MD stress can be calculated from lineal tension simply by dividing by web caliper.

There are several means of determining a process tension setting for a particular web material. These include previous experiences with similar materials, generic charts, and the 10-25% rule. What is most important here is the material rather than the process. In other words, for a given material the optimum tensions for calendering, printing, and winding will be quite similar.

The best guide for tension settings are based on experiences with similar materials. If these operations were successful, they will have struck a good balance between running too loose with the attendant difficulties of keeping the web flat and stable throughout the machine, and too tight with the attendant difficulties of web yielding or breaks.

However, the experience need not necessarily be with the same caliper, gauge, thickness, or weight. A good estimate of a running tension based on a similar material but of a different **thickness** would be

$$(4.1) \qquad T_2 = T_1 \frac{t_2}{t_1}$$

where

T_1 = old tension
T_2 = new tension
t_1 = old thickness
t_2 = new thickness

Lacking any previous experience with a similar material, one can turn to guidelines suggested by machine builders, technical organizations (such as TAPPI for paper), or published in the literature.[12,13] These guidelines will typically be a chart or family of curves of tensions versus thickness/weight for several nominal classes of materials.

Previous experience and generic guidelines work quite well provided that your material has similar MD strength (load to failure), MD stretch (strain at failure), or better yet yield point.

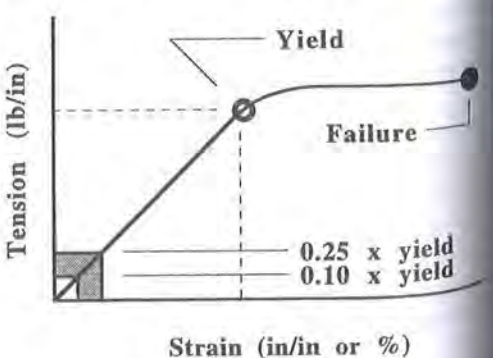
However, even if you've found no similar tabulation for the material you've developed, the 10-25% tension rule will serve well as a starting point for most web materials.[14]

Tension
Run a tension of 10-25% of the MD strength (or, better yet, yield **strength**) of the material.

Speed
Run a speed differential (often called draw) of no more than 10-25% of the MD stretch (or, better yet, yield strain) of the material.

Figure 4.1 shows a generic stress-strain diagram for a ductile material. By web convention, we've plotted strength in force/length instead of the mechanics units of force/area. The initial part of the stress-strain curve is linear and elastic (web returns to previous geometry when unloaded) for many materials. This is the section of the curve where most webs are tensioned as seen in the shaded area. At higher stresses, the ductile webs will yield and brittle webs will break. The near horizontal section of the curve is typical of many polymers which extend without further increases in load. Notice that the yield strength is only slightly lower than the failure strength as measured ubiquitously in test labs. However, the yield strain is much less than the failure strain for ductile materials.

Figure 4.1
10-25% Rule for Tension Settings



Tension Exceptions

Very **lightweight** or low caliper (<0.001") webs often are tensioned at only a few percent of their yield. This lower tension helps avoid a particular type of MD trough wrinkling or tunneling caused by imperfections in the machine and/or material.

Very **heavy and strong** webs are also often tensioned at only a few percent of their yield. Higher tensions simply are not effective or not needed to keep a thick web flat. Also, even at a lower percentage of strength, there is still usually plenty of tension for web/roller traction. Thus, while a more conventional (10-25%) tension would not hurt, a lower tension often is used to keep the equipment costs down (roller diameter for deflection and drive torque capacity).

Very **nonuniform** webs or machines are often best run at low tension. For example, high tensions might induce wrinkling of a puckered web on a crooked roller. Low tension allows the web/roller interface to act like a slip clutch when tension variations exceed available traction.

Low modulus or very ductile (stretchy) materials often are tensioned more than 25% of yield, sometimes even exceeding 60% of yield. The reason they can be tensioned higher is that the very elastic and ductile nature of the materials makes them more forgiving of machine geometry errors and tension control variations. The reason that they are more highly tensioned is often because the machine is too crude to safely run low tensions without risk of the tension momentarily dropping below zero. However, other reasons include to reduce web flutter (tissue) or to decrease the volume of a wound roll to reduce shipping costs.

True **drawing** (permanent stretching or elongation) will require the tension capacity of the machine and drive to be equal to the maximum web strength. However, the process must be controlled by speed differential as dictated by process/product requirements. If drawing is attempted using tension type controls on the plastic portion of the curve, the machine will run away and the web will pull apart. While true drawing operations are uncommon, examples include some metals, food processing, and the forming end of many machines.

However, we may not always be able to achieve good web handling conditions over a wide tension range as implied, for example, by the 10-25% rule. This is because one might have low tension problems as high as 15% and high tension problems as low as 20%. In this case, the working tension range would be only 15-20%. Indeed, it is not uncommon for there to be no sweet spot for the tension setting which avoids all problems.

Often, a reduced web handling tension range is an indicator of poor process health, such as nonuniformities of material and/or machine. For example, a severely misaligned **roller** can cause one side of the web to be floppy while the other is near breaking. Another example is where a high drag roller is near the onset of slippage, even when web tension is near breaking.

One may have process considerations that also further constrain what tensions must be run. For example, tension may need to be altered to achieve a **registration** target. Similarly, a specific tension ratio might be needed on both plies of a **laminate** to reduce curl. Finally, tension might need to be as low as possible to reduce the width variation of the winding of nonwoven products, or **winding** tension might need to be tapered to avoid roll starring. In these examples, we could handle the web (move it through the machine) under a wide variety of tensions, but product and process requirements further restrain the acceptable range of web tensions.

In conclusion, tension must be considered carefully for all web processes. At the design step, one might specify the tension range of the machine to span from 10% of the strength of the weakest web to 25% of the strength of the strongest web unless previous experience indicated otherwise. Then, one would select a midpoint tension for an initial startup value. Finally, a process capability study would determine the actual range of allowable tensions, as well as the problems that are encountered on each end of the tension range. Hopefully, the process is very tolerant of web tension, indicating a true web and machine, as well as a robust process. In the next section, we will discuss tension control precision.

How Close Must Tension Be Held

Since web tension is an important process parameter, it must not be allowed to vary excessively. Certainly, the tension must not be allowed to fall below zero even momentarily or the web might entangle itself on a roller or other part of a machine. Similarly, we could not allow tension to climb uncontrollably above the strength of the web or it would break.

However, as indicated in the previous section, web handling requirements might restrict acceptable tensions to 10-25% of yield. Thus, for example, a nominal tension setpoint might be selected at 15% of yield and allowed to vary between 10 and 20% of yield. This means a tension variation of ± 33% of the setpoint would be acceptable. However, many processes require a tighter control and many machines are easily capable of better tension control precisions.

A common tension control precision guideline is that tension should not vary more than 5% of setpoint during steady state and 10% of setpoint during transients (speed change or toggle of condition). Thus, for example, a machine set at 1.00 pli should hold tensions between 0.95 and 1.05 pli during steady state and between 0.90 and 1.10 pli during a speed change. This tension guideline is based on measurements from fast acting load cells (as opposed to the highly damped readings that are displayed digitally or used by drives).

The tension control precision guideline can be used to size the precision of the mechanical and electrical components of a machine. For example, a load cell or dancer must resolve much less than 5% of the lowest web tension (which may be 0.5% of the lowest web strength) if the system is to achieve a maximum 5% variation. Tension precision guidelines will determine acceptable resolutions and wrap angles for load cells, friction of dancers, and many other aspects of machine design. For example, as we will see in a later section of this chapter, speed control precision must be held to within 1 part in 10,000 for many webs in order to meet the above guideline. This tension control guideline also makes it more challenging to size unwind brakes and motors.

In a few cases, we can give more concrete design rules for MD tension control precision than the general 5% and 10% guidelines given earlier. For example, we could calculate the allowable tension excursion given a width tolerance for a product, MD modulus, and Poisson ratio as indicated in Chapter 3. Similarly, we could calculate the allowable tension excursion to avoid a certain magnitude of curl.

All the tension precision rules given thus far determine how close the average tension across the width of the web must be held at a certain roller position with respect to time. These rules say nothing about how much tension could be allowed to vary down through the machine or across the width.

To address downweb tension variations, a guideline states that tension down through a drive section should not span more than 10% of the setpoint during run and 25% of the setpoint during speed changes. As will be explained later, this determines how many undriven rollers can be used in a section. A section is defined as a series of undriven or slaved rollers bounded on each end by a driven and controlled roller.

Finally, to address cross web variation, a guideline states that tension should not vary more than ± 10% across the width of any roller. This guideline effectively determines maximum permissible roller misalignment and roller diametral variations.

Web bagginess also causes a variation of stresses across and sometimes down a web, even on a perfect machine. Web bagginess is a variation of natural length that will be discussed in a later chapter.

In conclusion, web tension control is an important aspect of web machinery and web processes. Thus, the permissible degree of tension variation must be sized or selected as a kind of quality parameter for the machine. If this value is too loose, the product and process will be excessively variable. If this value is too tight, machine costs will be unnecessarily high. The starting point guidelines given here will be adequate and easily achievable for most web processes.

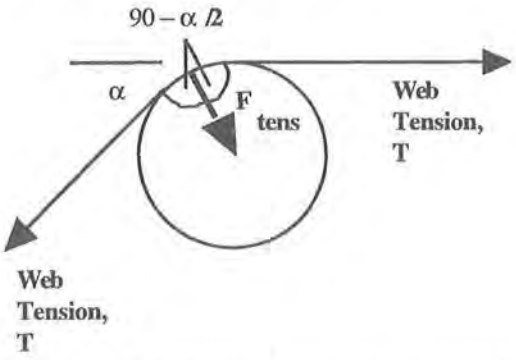
Load Cell Tension

The most direct measurement of web tension is by the use of rollers mounted on load cells.[15,16,17] A load cell is scale which measures a force component produced when a web wraps a roller under tension as seen in Figure 4.2. Most commonly, the load cell is an electronic strain gauge. However, LVDTs (linear variable displacement transformer), spring loaded pneumatic regulators and other force sensors have been used.

In addition to the web tension force, the load cell also must bear the weight of the roller, bearings and mountings as seen in Figure 4.3. Thus, the first rule of load cell sizing is that the **gross** (vector sum of tension and weight) must not exceed the capacity of the cell. Note that some cells have different ratings in the sensing and cross sensing directions. The second rule of load cell sizing is that the **net** force (tension component in the direction of the sensing) should be at least 10 times the accuracy of the cell for the lowest tension product to be run. For example, if we have a 100 lb cell with a 1% accuracy, then we would want the web tension force component in the sensing direction (÷2 for a two cell roller) to be at least 10 lb. To avoid the proverbial problem of trying to "weigh a pea in a dump truck," the following steps will give an improved sensitivity by increasing the ratio of net to load cell capacity.

1. Select the smallest load cell capacity that meets the first rule of gross weight sizing.
2. Wrap the load cell roller as much as possible.
3. Orient the wrap (or cell) to generate a tension force acting as closely as possible to the sensing direction of the cell.
4. For cells with a vertical sensing direction, orient the wrap so that it pulls upward to relieve roll weight and thus not increase the gross load.
5. Use light rollers and mountings. Composites can be very helpful here.
6. In general, the load cell roller must be an undriven idler so that drive forces and torques are not superposed onto the tension signal.
7. The tension must produce sufficient moment on a pivoting cell.

Figure 4.2
Roller Force from Web Tension



Web tension induces a **force on a wrapped roller**, bar, or other element. The direction of the force is oriented with the bisector of the wrap angle. The magnitude of the force is

$$F_{tension} = 2\,T\,W \sin(\alpha/2) \quad (4.2)$$

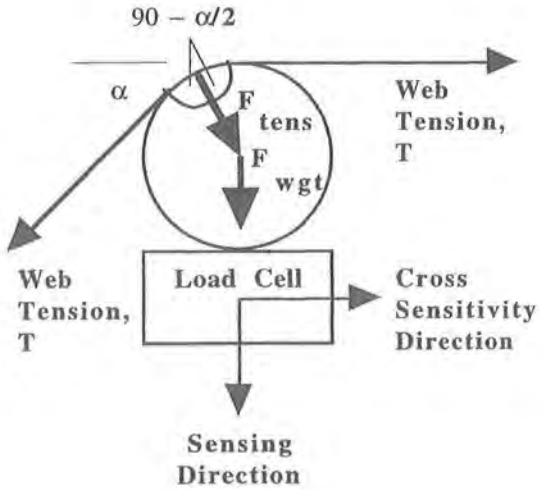
where

$F_{tension}$ = web force (kN) or (lb)
T = web tension (kN/m) or (lb/in or pli)
W = web width (m) or (in)
α = wrap angle (degrees)

For a load cell on each end of a roller, the cell will see a gross load that is the vector sum of

$$\mathbf{F}_{\text{load cell}} = \frac{1}{2}\left(\mathbf{F}_{\text{tension}} + \mathbf{F}_{\text{wgt (roller, brgs, mtgs, etc.)}}\right) \quad (4.3)$$

Figure 4.3
Forces Acting on a Load Cell



Load Cell Sizing Example

Metric Version

Given a 2 load cell system with the orientation of Figure 4.3 with a 45 degree wrap angle and

F_{wgt} = 50 kg = weight of roller etc.
T = 0.2-0.5 kN/m = web tension range
W = 1.5-1.8 m = web width range

Then,

$$F_{\text{tension max}} = 2\,T\,W\sin(\alpha/2)$$

$$= 2 \times 0.5\frac{kN}{m} \times 1.8m \times \sin(45/2)$$

$$= 0.689\ kN\ \frac{102.0\ gf}{N} = 70.3\ kgf$$

$$F_{\text{lc vert max}} = \frac{1}{2}\left(F_{\text{tens max vert}} + F_{\text{wgt vert}}\right)$$

$$= \frac{1}{2}\left(70.3\cos(45/2) + 50\right)kgf$$

$$= 57.5\ kgf$$

Given mechanically overload protected cells available in 20, 50, and 100 kg capacities, the 100 kg cell would be required. Now let's check sensitivity assuming the cells are accurate to 1% of full load.

Then the accuracy of the 100 kg cell would be 0.01 x 100 = 1 kg. Then the tension component in the sensing direction for even a crude application must be at least 10 x 1 = 10 kg.

$$F_{\text{tension min}} = 2\,T\,W\sin(\alpha/2)$$

$$= 2 \times 0.2\frac{kN}{m} \times 1.5m \times \sin(45/2)$$

$$= 0.230\ kN\ \frac{102.0\ gf}{N} = 23.4\ kgf$$

$$F_{\text{tens sense min}} = \frac{1}{2}\left(23.4\cos(45/2)\right)kgf$$

$$= 10.81\ kgf$$

This load cell system just meets the crude sensitivity requirements, but would probably compromise the tension control accuracy of a nice drive.

English Version

Given a 2 load cell system with the orientation of Figure 4.3 with a 45 degree wrap angle and

F_{wgt} = 100 lb = weight of roller etc
T = 1-3 lb/in = web tension range
W = 60-70 in = web width range

Then,

$$F_{\text{tension max}} = 2\,T\,W\sin(\alpha/2)$$

$$= 2 \times 3\frac{lb}{in} \times 70in \times \sin(45/2)$$

$$= 160.7\ lb$$

$$F_{\text{lc vert max}} = \frac{1}{2}\left(F_{\text{tens max vert}} + F_{\text{wgt vert}}\right)$$

$$= \frac{1}{2}\left(160.7\cos(45/2) + 100\right)lb$$

$$= 124.2\ lb$$

Given mechanically overload protected cells available in 50, 100, and 200 lb capacities, the 200 lb cell would be required. Now let's check sensitivity assuming the cells are accurate to 2% of full load.

Then the accuracy of the 200 lb cell would be 0.02 x 200 = 4 lb. Then the tension component in the sensing direction for even a crude application must be at least 10 x 4 = 40 lb.

$$F_{\text{tension min}} = 2\,T\,W\sin(\alpha/2)$$

$$= 2 \times 1\frac{lb}{in} \times 60\ in \times \sin(45/2)$$

$$= 45.9\ lb$$

$$F_{\text{tens sense min}} = \frac{1}{2}\left(45.9\cos(45/2)\right)lb$$

$$= 21.2\ lb$$

Since the tension signal is too low, we would need a 1% accurate cell or would try wrapping the cell nearly 90 degrees, or would wrap pulling up, or could lighten the roller so that we could drop down to the next lower capacity. Then we would reiterate on both the capacity and sensitivity calculations. This typical example illustrates why load cell rollers usually need a heavy wrap angle.

Other Load Cell Considerations

Selecting a Load Cell

While there are many considerations for load cell selection beyond those discussed earlier, we will focus on four that are most commonly overlooked. First and most important, the cell should have a 10X or better mechanical (not electrical) overload protection. Mechanical protection is an internal stop that keeps accidental overloads from excessively deflecting and damaging cells. Unfortunately, some vendors do not provide this feature. Second, have spare cells because they will fail eventually.

Third, high speed applications (>1000 mpm) may require high mechanically damped load cells because roller/cell/framework system resonance is almost inevitable, causing high amplitude imbalance noise to compromise the signal. Unfortunately, I am aware of only one load cell vendor that provides this feature. Fourth, high speed load cell rollers must be balanced well as covered in my roller book.

Amplifier & Display

Raw load cell signals are amplified and then sent to a display and/or controller. Among many other important electrical considerations, the black box should have the following features.

1. Be able to sum 2 or more cells for tension control use.
2. Have a selector for front, back, and difference. Being able to select either a front or a back makes troubleshooting a bad cell much quicker. Being able to select a difference can be used to indicate how level the tension is across the web. This also could be used to automatically position an adjacent guide or squaring roller.
3. Have an adjustable filter to clean up noisy signals coming from web bounce or roller imbalance.
4. Be able to scale the display to engineering units of kN/m or lb/in (pli). An analog meter is much easier for the operator to use than a digital display.
5. Have an additional 0-5 VDC signal terminal to allow hookup of a strip chart recorder or other device for troubleshooting.
6. Lastly, it sometimes is easier to work with a vendor who supplies both load cells and drives (electric motors or mechanical brakes).

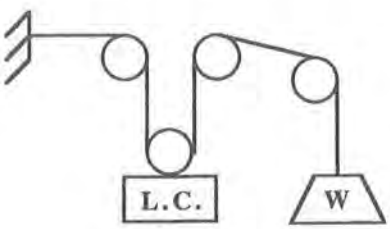
Calibrating Load Cells

Calibrating load cells involves three steps: set zero, set gain, and recheck zero. The zero is adjusted without any load applied to the roller. The gain is calibrated by stringing a rope through the machine upon which weights are hung. The position of the rope is centered on the upstream, load cell, and downstream rollers following the same path as the web shown in Figure 4.4. Weights should be hung to give a load near the maximum of the cells to set gain and half that to crudely check linearity.

Troubleshooting Load Cells

1. See if the load cell display moves when you push on the roller. If not, consult the load cell manufacturer's troubleshooting procedure.
2. Check to see if the zero has moved since the last calibration. An unexplained movement of zero on a single cell is a symptom of the most common load cell problems: it has been mechanically damaged because it was overloaded or struck by something, or the electronics are drifting.
3. Change out a cell with a spare: Did the new cell solve the problem? If not, it might be the amplifier box. Try changing it out.
4. Noisy load cell readings come from two major sources. The first is mechanical vibration. The second is electrical noise. Systems where the amplifier is remote from the cell are most prone to electrical noise.
5. A 4-20 mA signal is much more robust than a 0-5 V signal.

Figure 4.4
Calibrating a Load Cell with a Weighted Rope



Dancer Controlled Tension

The most common design arrangement of a dancer is shown in Figure 4.5. The dancer roller is connected to a pivoting arm whose position is measured by a rotary or linear pot (potentiometer).[18,19] If the tension is high, the dancer is pulled upward and the pot sends a signal to the downstream drive to slow down (or the upstream drive to speed up). If the tension drops, so does the dancer roller.

A minimum tension is provided by the weight of the roller and a portion of the arm. To change tension, pneumatic cylinder pressure might be varied or weights might be added or removed.

Dancers are one of the oldest web tension sensors and are still commonly used today. Their primary advantage is that they can accommodate drives that are sloppy (as were most before digital drives) or for applications which have extremely fast tension transients (such as on flying splices). The dancer is tolerant of fast tension transients and slow drives because of the length of stored material in the loop that can be drawn from or fed into. A less common arrangement of dancer is a festoon as seen in Figure 4.6. This provides even more stored material and is used for zero speed splicing of unwinding or winding rolls.

However, there are several disadvantages and misconceptions with dancers. The first disadvantage is that, while it does respond to changes in tension, it does not display tension as does a load cell. Thus, there is no output of average or instantaneous tension to be used for process design, quality control, or troubleshooting. The best that can be done is to calculate the (average) tension as a function of cylinder pressure or arm counterweight. Even then, however, one is not able to determine tension excursions to gauge the health of the drive control system.

Another concern of the dancer is the common use of pots (variable **potentiometers**) for position sensing. An example of a pot is the volume knob on a radio. It is not durable to continuous turning. Also, the pot can fail in the worst manner possible due to occasional and erratic control glitches as the slider gets dirty.

Figure 4.5
A Pivoting Dancer

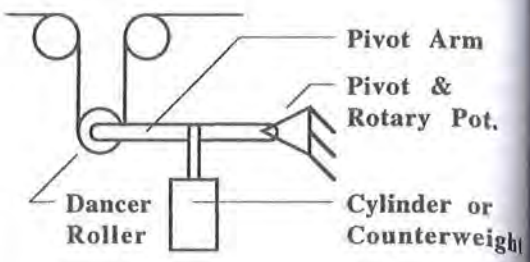
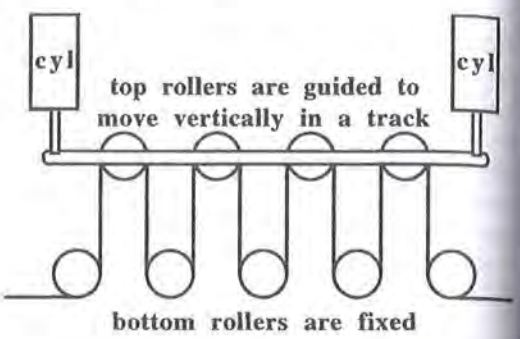
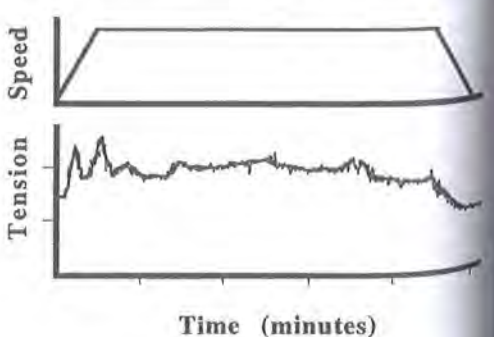


Figure 4.6
A Festoon Dancer



However, the best evaluation of dancer/drive performance is to measure tension excursions after the dancer with a load cell. As seen in Figure 4.7, tension is poorly held, especially during speed changes. Much of the poor performance of dancers is excessive **mechanical friction** of the pivot and cylinder assembly which should be much less than 1/10th of the force of the lightest tension to be run. It also is important to keep dancer **mass** to a minimum to improve response.

Figure 4.7
Tension (Im)Precision of a Dancer



Dancer Design

The first step in dancer design, as it is for load cells, is to size the **roller**. As discussed in more detail my roller book, the nominal diameter is primarily determined by deflection standards which tend to yield slenderness ratios (length/width) of around 15. Then, the minimum roller weight is largely determined because wall thickness and shell material are smaller factors than diameter. Of course, the roller can be made heavier by increasing wall thickness or adding weight.

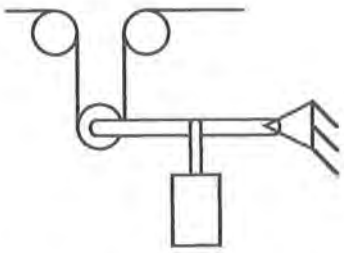
Second, we must choose a dancer **configuration** that minimizes the required cylinder size so that cylinder friction which compromises sensitivity also is reduced. Indeed, the cylinder friction alone often will put a dancer out of spec without even considering pivot/slide friction or regulator hysteresis. The two primary configurations are the horizontal and vertical arrangements as seen in Figure 4.8. The vertical arrangement is suggested for materials whose nominal running tension is similar to the already calculated roller weight. The horizontal dancer is more appropriate for light webs whose tension is much lower than roller weight. Note that accumulators or festoons should never be used as tension sensors because of the friction penalty.

Third, we need to size and position an air cylinder to provide coverage of the required **tension range** so that the minimum tension is achieved with no less than 10 psi, and the maximum tension is achieved with no more than 60 psi. Often, we may need to weight or counterweight the dancer mechanically to get the assembly to balance at about the midpoint of the design tension range. If the tension range is quite large, then we may need to provide adjustable counterweights.

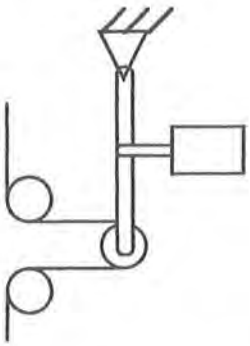
Fourth, we must choose a position **sensor**. Since the dancer is in near constant motion, contacting sensors such as rotary pots don't tend to hold up well and can add additional friction of their own. Thus, noncontacting sensors such as servo-pots, LVDTs, or ultrasonics may be preferred. Optical encoders may not work well if the dancer doesn't stay put long enough to make the read.

Figure 4.8
Dancer Configurations

vertical dancer for heavier webs



horizontal dancer for lighter webs



Finally, we need to **calibrate** the dancer graphing tension versus cylinder pressure. This is done by a weighted rope following the web path much like already described for load cells. With a given weight, the cylinder pressure is raised until the dancer moves in one direction, and lowered until the dancer moves in the other. The average pressure is used to make the chart, while the difference is a good measure of sensor **sensitivity**. The hysteresis of the dancer should be <<10% of the minimum design tension. The process is repeated for several weights corresponding to the design tension range.

The **vacuum dancer** is a design variation for tensions that are light but heavier than can be provided by a catenary. The vacuum dancer uses a lightweight puck instead of a pivoting roller to hold loop geometry. The loop and puck are contained in a vacuum chamber to provide additional tension. Loop length feedback can be provided by photoeyes or ultrasonics. The most common application is for infeed/outfeeds of magnetic tape drives on some mainframe computers.

Dancer Dynamics

A common misconception is that the dancer provides a damping of tension disturbances as might be generated by an out-of-round supply roll, unstable drive, or vibrating roller. While this can happen, the realities show much more mixed results.

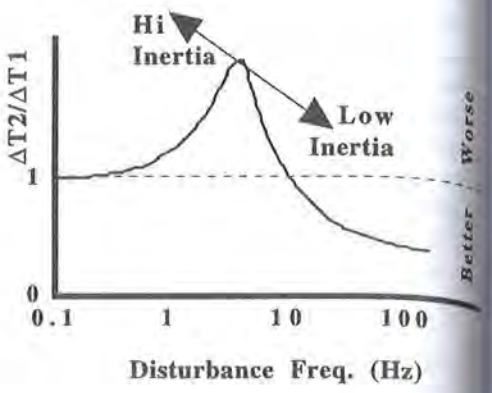
Figure 4.9 shows the **frequency response** of a typical dancer system. This is a plot of the ratio of tension disturbances before and after the dancer as a function of the frequency of the disturbance. A value of one means that the tension disturbance passes through unaltered. A value less than one means the dancer is attenuating or reducing tension fluctuations, while a value greater than one means the dancer is exaggerating or increasing tension fluctuations.

As seen in the figure, the dancer does little to change web tension excursions at low frequencies. However, at intermediate frequencies the dancer system goes into **resonance** and actually magnifies tension disturbances. The position and severity of the peak depends on the dancer's inertia and other dynamic factors. It is only at very high frequencies that the dancer provides a damping or reduction of tension disturbances. Thus, it might be helpful for shock loads coming from a flying splice or rotary die-cutter where many of the frequency components are high.

Those familiar with vibration will note the similarity of the graph to the response of a single spring mass system. Those who think in electrical terms will classify this system as a low pass filter. Though perhaps surprising to some, the dancer has the same frequency response character as an undriven idler roller. Thus, if damping is the only criteria, a high inertia roller may give similar results.

There is an exception to the mixed benefits of the typical dancer. This is the **inertia compensated dancer roller** that was developed and patented[20] (expired) by Martin Automatic Inc. The frequency response shape is still similar but lies mostly on the attenuation side.

Figure 4.9
Dancer Damping of Tension Fluctuations



According to the patent, a complete attenuation of tension disturbances is achieved if

(4.4) $J = M r^2$

where

J = polar inertia of the dancer roll(ers)
M = dancer mass
r = dancer roller outer radius

For a typical single translating or pivoting dancer roller, this will require the addition of a flywheel (rotary inertia) to the ends of the roller. Conversely, multiple roller festoon systems might require the addition of mass (translational inertia) to the moveable carriage.

While a desktop demo and the patent are convincing, there were a few higher order terms left out of the derivation, as was the case in other early models.[21,22] The realities of the inertia compensated dancer roller generally are beneficial, but short of the complete attenuation sometimes claimed. This has been verified both experimentally and analytically[23,24] by several independent researchers. While the inertia compensated design might lay a serious claim to smoothing out tension disturbances, the question must be asked if it is necessary and sufficient. Indeed, even idler rollers can significantly reduce the transmission of tension disturbances downstream. However, inertia of any kind will cause tension upsets during speed changes.

Draw Control

Draw control is the speed regulation of several driven rollers in a process line. While the equipment and controls are quite simple, draw or speed control is **limited to very stretchy materials** such a tissue paper, stretch film, nonwovens, and some textiles. (True drawing operations will permanently stretch the web.)

Figure 4.10 is an example of draw control on a roll-to-roll converting line. Here, the unwind, a converting process (calender, emboss, print, slit, etc.), and a windup are all driven at a controlled surface speed. However, the tach or encoder connected on the back end of a motor measures rpm (revolutions per minute), and must calculate surface speed from the equation for rigid body motion,

(4.5) $V = \omega r$

which can be given in conventional units as:

(4.6a) Metric Units

$V = 0.031416 \times D \times rpm$

(4.6b) English Units

$V = 0.26180 \times D \times rpm$

where the speed V is in mpm (m/min) in the metric system and in fpm (ft/min) in the English system, and while D is the outer diameter of the roll(er) in (cm) for the metric system and (inches) for the English system. The **diameter** of rollers will be measured by a caliper, while the changing diameter of unwinding and rewinding rolls may be measured by a variety means such as ultrasonically.

The draw between any two rollers is expressed as a percent speed change with respect to some reference which should be stated. For example, if the converting unit were the speed reference for the system, the windup draw would be

$\%Draw_{3/2} = 100 \times (V_3 - V_2) / V_2$

Web strains (elongations) can be calculated from speeds or draws provided that a strain is known somewhere. For example,

Figure 4.10
Draw Control in Roll-to-Roll Converting

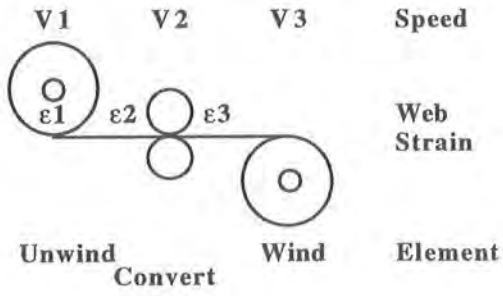
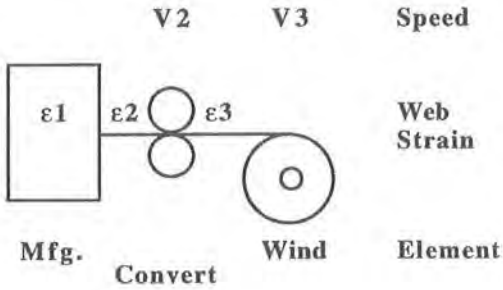


Figure 4.11
Draw Control in a Manufacturing Line



(4.7a) $\varepsilon_2 = \varepsilon_1 + \frac{\%Draw_{2/1}}{100}$ and

(4.7b) $\varepsilon_3 = \varepsilon_2 + \frac{\%Draw_{3/2}}{100}$ etc.

Notice, however, that in the roll-to-roll converting line of Figure 4.10, we would generally not know the strain present in the unwind (ε_1). Similarly in the manufacturing line of Figure 4.11, we would generally not know the strain coming out of the maker (ε_1). Thus, **draw control is not closed loop** on the web because there is no web sensor. This means it is, by itself, unable to measure or compensate for upstream web fluctuations.

Finally, **draw control precision** needs to be sized appropriately. If, for example, a typical quality speed control can achieve a 0.1% accuracy, then anything less than a 1% draw setting would be imprecise (10% variation), which means the MD stretch (or, better yet, yield) should be greater than 10% (minimum tension 10% of breaking value).

Tension Variations in a Draw Zone

Draw control is required in areas where the web is being yielded because the stress-strain curve is relatively flat there. In other words, small changes in tension can cause large changes and often unpredictable changes in anelastic strain (i.e., permanent length changes). Thus draw is more suitable than tension control in stretching and true drawing operations. Examples include the wet end of a paper machine, tenting of plastic film, and some oven applications.

However, draw control also is used widely as a ßcheap but crude form of tension control in the elastic region of a material's response. Here, the objective is to control tension indirectly through draw. Unfortunately, the consistency with which the tension (or **registration**) is held in any particular draw zone is affected by a number of parameters. These factors vividly demonstrate the common fallacy of draw "**tension isolation.**"

One strong factor in the variable effects of draw is the **pre-draw zone tension**. In other words, as we discussed in the previous section, any changes in the upstream tension will be passed down through the next zone.

Another factor is the **percent draw**, which is the nominal setpoint. Note that it is completely acceptable to run zero or slightly negative draws because that only means web tension is maintained or drops slightly in the zone. However, we also need to be concerned with how closely the drive system holds the target speed. Clearly, the speed increments and tolerances need to be <<10% of the web's strain.

Web modulus also affects tension so an increase in modulus will increase the tension at that point. However, if the modulus is increasing because the material is thicker, the average stress will be similar even though tension has changed.

Any **hygrothermal** process that changes the length of the material also changes the tension. For example, if the moisture (paper and some polymers), solvent loading (if it causes the material to swell), or temperature (polymers) increases, the tension will drop and vice versa. This, of course, is in addition to any modulus changes that also will result as a consequence.

Traction must be maintained on draw controlled rollers. If the web slips on a positive draw zone, the tension will not increase as much as calculated and vice versa.

Covered nipping rollers will have an effective radius that increases with increasing nip as discussed in another chapter. Conversely, lowering the **nip loading** on a soft pull roller drive section will effectively decrease the draw and hence tension on that zone.

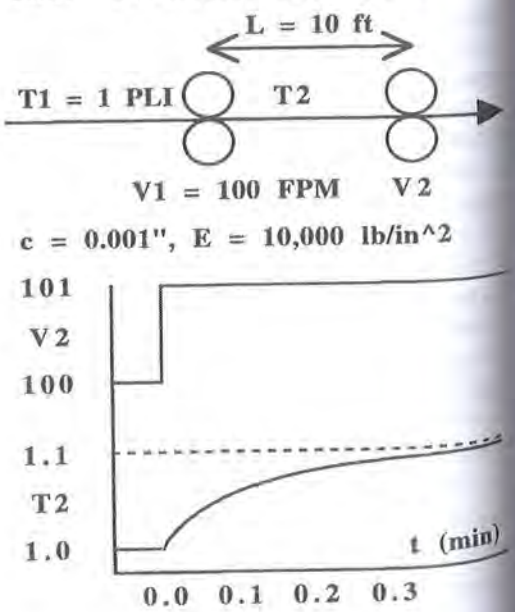
Finally, there is a **time constant** involved in draw control that makes tension react sluggishly to draw changes. For example, if the downstream roller is sped up immediately, the tension does not increase immediately. Rather, it increases as an exponential as shown in Figure 4.12 with a time constant τ = length/speed. Note that similar issues are involved with **registration** (strain) using tension control.

$$T_2 = T_1 + (E\,\Delta\varepsilon c)\left(1-e^{-t/\tau}\right) \quad (4.8a)$$

where

$$\Delta\varepsilon = \frac{V_2 - V_1}{V_1} \quad (4.8b)$$

Figure 4.12
Time Constants and Draw



Catenary (Loop) Control

Some applications require such exceptionally **light tensions** that load cells or dancers may not have suitable resolution. If this material also had **limited stretch**, then draw control also would not be suitable. Fortunately, light tensions can be run using catenary control.

The shape of the sagging arc of a telephone line, clothesline, kite string, or web between two rollers are all the same as seen in Figure 4.13. The shape goes by the fancy mathematical term of catenary. We can use the catenary for light tension applications as well as for a poor man's tension measurement on machines not equipped with load cells or other sensors. The catenary curve as seen below has a curve defined as:

$$(4.9) \quad y = \frac{a}{2}\left(e^{x/a} - e^{-x/a}\right)$$

where x and y are the horizontal and vertical positions respectively and "a" is called the parameter of the catenary. Mathematically, "a" is the distance from the lowest point in the curve to the directrix. Physically, "a" relates weight and tension. For webs, a is calculated as:

$$(4.10) \quad a = \frac{T_h}{w} \text{ where}$$

T_h = horiz'l component of tension (force/width)
w = basis weight (weight/area)

Both rollers on each end of a span must lie tangent to the curve defined above and shown in Figure 4.14. The application of the catenary to calculate tension from the web sag, span, and basis weight as seen in Figure 4.15. This can be simplified for the specific case of a shallow horizontal curve as:

Figure 4.13
A Catenary Sag

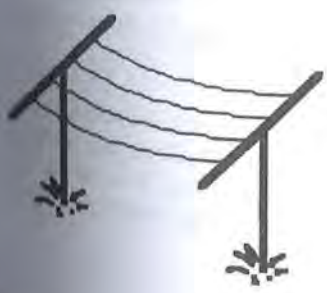
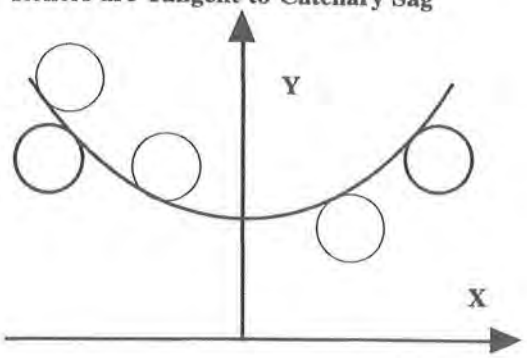


Figure 4.14
Rollers are Tangent to Catenary Sag



$$(4.11) \quad T = \frac{L^2 w}{8 s} \text{ where}$$

L = span length between roller tangents
w = basis weight
s = sag

Example Problem
Given a 20 lb light paper grade sagging 0.25 in. on a horizontal span 50 in. long, what is the tension?

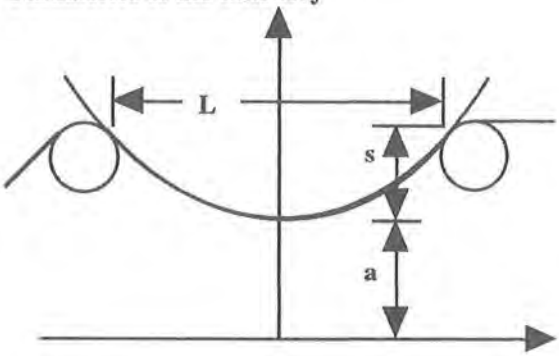
$$T = \frac{L^2 w}{8 s} = \frac{(50 \text{ in})^2 \left(\frac{20 \text{ lb}}{3000 \text{ ft}^2} \frac{\text{ft}^2}{144 \text{ in}^2}\right)}{8 (0.25 \text{ in})}$$

$$T = 0.0579 \text{ lb/in}$$

As an exercise, the reader can verify that this result when plugged into Equations 4.9 and 4.10 will yield a legitimate catenary.

The technique works well on lightly stressed webs with long spans. Sag can be measured with a ruler, optically, or ultrasonically.

Figure 4.15
Parameters of the Catenary



Tension from Motor Readings

Tension is a byproduct of motor torque. This knowledge enables us to approximate web tension from electric motor drive readings. It can be applied to unwinds, windup sections, and rollers. For unwinds and windups, the value will be for absolute tension. If applied to a roller, the calculation is for a tension differential across the roller. However, either measuring or controlling tension via amperage neglects inertial effects due to speed changes. The meter amperage used is the different between values of running with and without web tension.

Need to know:

Amperage from meter during run
Motor amp and power nameplate values
Web width
Speed

Efficiency can be assumed to be 90% for many systems (10% friction and other losses). However, friction can be high on some drive components and with some punky materials rolling on a drum.

English Units

(4.12)

$$P_R = P_{nameplate}\,(\text{Efficiency})\left(\frac{\text{Amps}_{meter}}{\text{Amps}_{nameplate}}\right)$$

where
P_R = power during run (HP)
$P_{nameplate}$ = motor nameplate power (HP)
Efficiency = elect & mech efficiency ≈ 0.90
Amps_{meter} = meter reading during run (amp)
$\text{Amps}_{nameplate}$ = motor nameplate current (amp)

(4.13)

$$T = \frac{P_R \times 33{,}000}{W \times S}$$

where
T = tension (lb/in)
P_R = power during run (HP)
W = web width (in)
S = speed (ft/min)

Example for two-drum winder given:
front drum and rear drum motors
rated at 100 HP and 160A each,
front drum runs at 100A, rear drum at 60A
speed = 6000 fpm, width = 200 inch

$$P_R = (100+100)\,(0.90)\left(\frac{100+60}{160+160}\right) = 90 \text{ HP}$$

$$T = \frac{90 \times 33{,}000}{200 \times 6{,}000} = 2.48 \text{ lb/in}$$

Note: From Equation 4.12 you can also size the tension power requirements (does not include accel/decel or losses). Also, it is better to measure friction than to assume a 90% efficiency as shown here.

Metric Units

(4.14)

$$P_R = P_{nameplate}\,(\text{Efficiency})\left(\frac{\text{Amps}_{meter}}{\text{Amps}_{nameplate}}\right)$$

where
P_R = power during run (kW)
$P_{nameplate}$ = motor nameplate power (kW)
Efficiency = elect & mech efficiency ≈ 0.90
Amps_{meter} = meter reading during run (amp)
$\text{Amps}_{nameplate}$ = motor nameplate current (amp)

(4.15)

$$T = \frac{P_R \times 60}{W \times S}$$

where
T = tension (kN/m)
P_R = power during run (kW)
W = web width (m)
S = speed (m/min)

Example for an unwind given:
motor nameplate rating 150 kW
motor nameplate current 300 amp
motor running current 150 amp
speed = 2000 mpm, width = 5 m

$$P_R = (150)\,(0.90)\left(\frac{150}{300}\right) = 67.5 \text{ kW}$$

$$T = \frac{67.5 \times 60}{5 \times 2{,}000} = 0.41 \text{ kN/m}$$

Note: From Equation 4.14 you can also size the tension power requirements (does not include accel/decel or losses). Also, it is better to measure friction than to assume a 90% efficiency as shown here.

Current Control

Since tension is a result of armature current in the ubiquitous DC motor, we can use it to crudely control web tension. All that need be done is to calculate from the previous section the required current to obtain a specified tension (unwind or windup) or tension differential (intermediate roller). This technique is applied to constant diameter elements such as found on surface winders.

An example usage of this technique is on some large paper machinery unwinds. Here, the unwind tension is provided by large (50+hp) regenerative motors. Normally, the tension is controlled via load cell feedback. However, if the load cells should fail, which happens occasionally in that rugged environment, current control is the fallback to keep the process running. The operator merely switches to current and the former tension rheostat becomes a current controller which he sets to a value similar to what normally was observed.

Current control is open loop on the web and, as such, results in compromised tension control. First and foremost is the large inertia component of tension during speed changes. In current control, the tension will rise during accel and drop during decel for unwinds and vice versa for intermediate rollers and winders. While some of this variation can be reduced by adjusting current automatically, inertia can vary many orders of magnitude on winding and unwinding rolls.

Other problems include drag, friction, and other losses that are usually variable. Large losses will be seen on rolling nips, particularly if one of the elements is soft (covered roller or wound roll) and highly loaded. Other losses include bearing drag, sliding element drag (turnbars, stationary spreaders, etc.), and windage. Finally, there are many motor complexities also that can increase tension uncertainties when using the open loop current control scheme.

Torque Control

Pneumatic brakes, magnetic particle brakes and clutches, slipped core winding shafts, and tendency drives are common converting components. These and similar devices are torque controlled actuators.

For example, the **air pressure** set by a regulator primarily determines the output torque of a pneumatic brake. Similarly, the magnetic particle device has an output torque approximately proportional to current. The slipped core winding shaft torque is determined primarily by axial load. Finally, the tendency drive torque is a reflection of bearing torque.

Unfortunately, none of these devices is very **repeatable**. For example, the pneumatic brake torque also can vary as a function of the friction coefficient of a particular pad, with rpm and with temperature (brakes fade at high temperature). Similar effects are found on the other torque devices as well.

Many plants operate these torque devices in **open loop** on the web. Thus, for example, an operator will set brake torque, and thus indirectly tension, through an air regulator. Unfortunately, the results often will be tension excursions exceeding standards of decency. The solution for poor tension control in this case is quite simply to use load cell (or dancer) feedback to run the devices instead.

Of even graver concern is the use of open loop torque devices on unwinds. The problem is compounded here because the torque must be reduced proportionally as the diameter of the unwind roll decreases. True, the operator can lower the setting manually as the roll is unwound, but what is the quality of the results? Also, there are diameter compensating devices that can reduce the error a bit, but why not simply install load cells instead?

Good practice for web processes:

No friction without feedback!

Bibliography

1. Roisum, David R. *Runnability of Paper.* Tappi J., vol 73, no 1, pp 97-101, January 1990, and no 2, pp 101-106, February 1990.

2. Roisum, David R. *What is the Best Way to Measure Length?* Converting Magazine, June 1995.

3. Roisum, David R. *How Can I Control Width Accurately?* Converting Magazine, June 1995.

4. Puckhaber, Charles F. *Print Registration Control in a Rotogravure Printing Press Combined with Extrusion Coating or Lamination Processes.* Tappi J., vol 78, no 3, pp 144-154, March 1995.

5. Roisum, David R. *How Can I Reduce Laminator Curl?* Converting Magazine, Web Works column, pp 22, March 1995.

6. Roisum, David R. *What Causes a Baggy Web?* Converting Magazine, Web Works column, pp 22, April 1994.

7. Roisum, David R. *How Can I Fix a Baggy Web.* Converting Magazine, Web Works column, pp 20, April 1995.

8. Roisum, David R. *The Mechanics of Wrinkling.* Tappi J., vol 79, no 10, October 1996.

9. Roisum, David R. *The Mechanics of Wrinkling.* Tappi J., vol 79, no 10, October 1996.

10. Roisum, David R. *The Mechanics of Winding.* TAPPI PRESS, Atlanta, GA, 1994.

11. Roisum, David R. *The Mechanics of Rollers.* TAPPI PRESS, Atlanta, GA, 1996.

12. DuBois, G.B. *Selecting Proper Winding Tension for Various Web Substrates.* Tappi J., vol 76, no 5, pp 91-93, May 1993.

13. Jones, J.P. *Determining Tension Values.* Converting Mag., pp 64-72, November 1989.

14. Roisum, David R. *What Tension Should I Run?* Converting Magazine, Web Works column, pp 20, November 1994.

15. Endersby, D. C. and Zucker, E. *Direct Tension Control by Strain Gauges and Disc Brakes for Rewinders.* Pulp and Paper Magazine of Canada, vol 67, no 9, pp 393-398, September 1966.

16. Viney, David. *Measurement and Control of Web Tension in Papermaking.* Paper, vol 198, no 1, pp 26, 28, 30, July 19, 1982.

17. Montalvo, William W. *Unraveling Web Tension Control.* Power Transmission Design, vol 27, no 1, pp 35-37, November 1985.

18. Bak, David J. *Dancer Arm Feeback Regulates Tension Control.* Design News, vol 43, no 6, pp 132-133, April 1987.

19. Weiss, Herbert L. *Getting Better Quality With Dancer Controlled Unwinders.* Package Printing, vol 28, no 12, pp 11-12, 103 and vol 29, no 1, pp 14, 50-52.

20. Martin, John R. *Tension Regulation Apparatus.* U.S. Patent 3,659,767 issued May 2, 1972.

21. Pfeffer, J.F. Jr. *Dancer Roll Assembly.* U.S. Patent, February 21, 1961.

22. Marhauer, H.H. *The Dynamics of Web Tension Measurement.* ISA Transactions, vol 5, no 3, pp 242-247, 1966.

23. Lin, P. and Lan, M. S. *Effects of PID Gains for Controller with Dancer Mechanism on Web Tension.* Proc. of the 2nd Int'l Conf. on Web Handling, Oklahoma State Univ., Stillwater, OK, June 6-9, 1993.

24. Reid, Karl N. and Lin, Ku-Chin. *Dynamic Behavior of Dancer Subsystems in Web Transport Systems.* Proc. of the 2nd Int'l Conf. on Web Handling, Oklahoma State Univ., Stillwater OK, June 6-9, 1993.

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