Getting and Losing Traction

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AIMCAL Web Coating and Handling Conference
October 2012

Introduction:

“Assuming good traction …” is a phrase that is often seen at the beginning of papers on web handling. Authors, including yours truly, use it to dismiss a mare’s nest of problems. The biggest of them is the effect of air lubrication and its interaction with surface roughness of both the roller and the web. Because of it, the web begins to lose traction as soon as it starts moving.

A lot is known about air lubrication. There are even models available for precisely predicting the coefficient of friction of a particular web on a particular roller at a particular speed. But, they usually require numerical data that isn’t available or, alternatively, require a good deal of educated judgment. However, there is still great value in understanding the physical principles of this subject. It can help you troubleshoot problems and predict which direction things will go when you change something.

This paper is not intended as a report on new research into fundamental principles of traction. It will focus is on comparisons of three different types of rollers – a standard roller, a spiral groove roller and a microgroove roller. The review of basic principles is based on the work of others.

The grooved rollers are commercially available designs which were generously donated by Webex (thank you Pete Eggen). The web used throughout the tests is Tedlar donated by DuPont (thank you John Wysokowski). [Historical note: The microgroove design was developed in collaboration with 3M many years ago (thank you Tim Walker)]

The basic plan:

The paper will begin with a description of the test setup

- The belt
- Sensors
- The drive
- The brake and torque measurement
- Bearing friction

Then there will be a review of basic air lubrication principles:

- Static friction of a web wrapping a roller
- Effects of air entrainment
- Venting techniques

Finally, the test results will be presented and discussed.

- Standard roller
- Microgroove roller
- Spiral groove roller

The rollers:

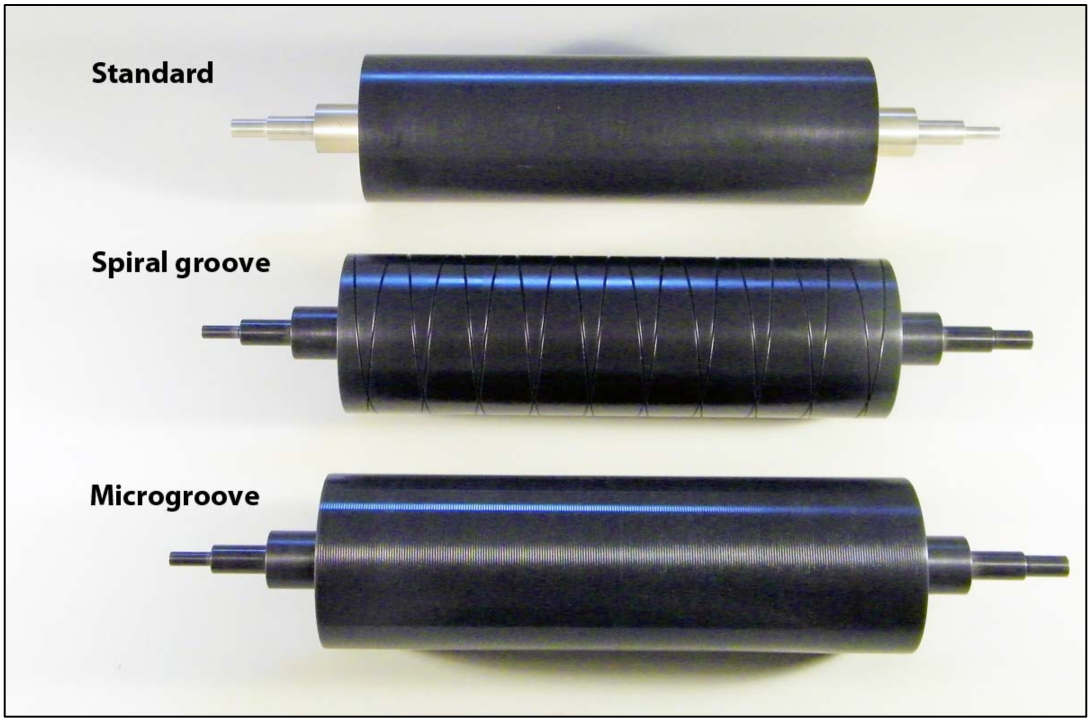


Figure 1
The three test rollers
Live shaft, 3 inch (76 mm) diameter, 10 inch (254 mm) face, hard anodized
Webex supplied the two grooved rollers with commonly used patterns

Standard- machined 32 Ra surface

Spiral groove – Two grooves cut end to end, opposing directions, machined 32 Ra surface

Microgroove – One groove on each half, opposite hand, machined 32 Ra

Each roller weighs 7 Lbf (31 N)

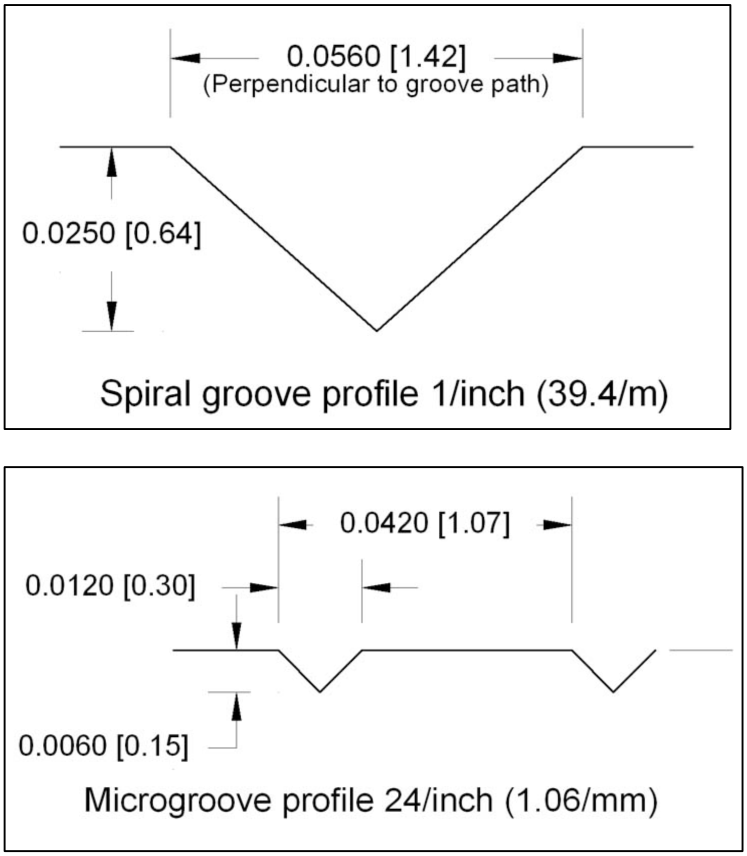


Figure 2
Groove geometries

The machine:

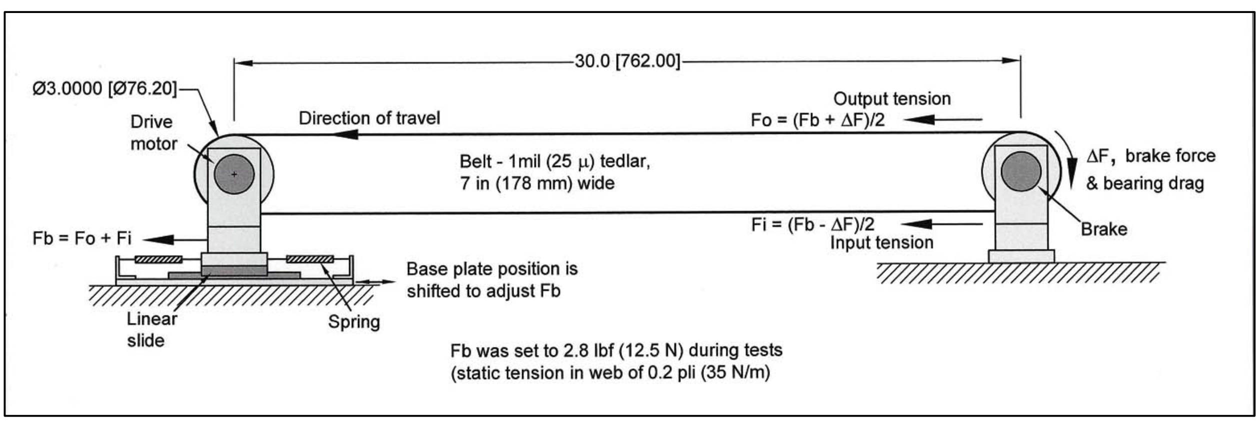


Figure 3
Continuous belt mechanism

Note: To maintain a sense of scale, all torques and tensions are described in terms of the total force in the web or at the roller surface.

The sum of the tensions in the entering and exiting spans is always constant. Thus, tension in the entering span is equal to $F_b/2 - \Delta F/2$. And tension exiting is $F_b/2 + \Delta F/2$.

The location of the sensors relative to the angle of wrap is easily adjusted.

A piece of tape on the left end of the roller serves as a marker which can be used to calculate roller speed as well as serve as a reference to locate the web measurement relative to the roller surface. The splice in the belt serves as a marker for the web location and can also be used to calculate its speed.

Measurement end:

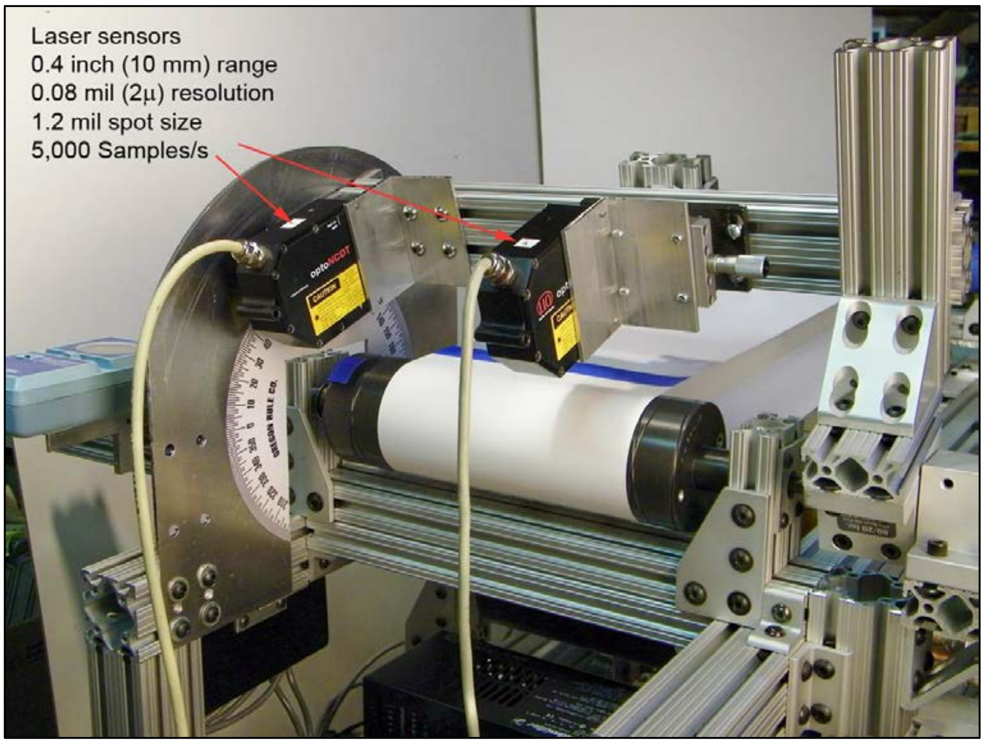


Figure 4

Measurement end of belt showing laser sensor and tape markers

Each sensor produces an analog signal of ± 5 volts which was recorded with a PC scope capable of recording and saving 32,000 samples at a time.

An eddy current brake is used to apply an external torque load to the test roller. An aluminum gear attached to the roller serves as a reaction plate for the brake. Eddy currents are generated by neodymium magnets attached to an identical plate. The magnet plate is prevented from rotating by a force applied through a moment arm on the plate. The force is supplied by a push rod attached to a force gauge. Torque is adjusted by adjusting the distance between the magnet and reaction plate. This is done with a rack and pinion slide. Friction torque in the magnet plate mechanism is negligible because it doesn't rotate and the bearings are quite small. Ranges and accuracy are shown in Figure 5.

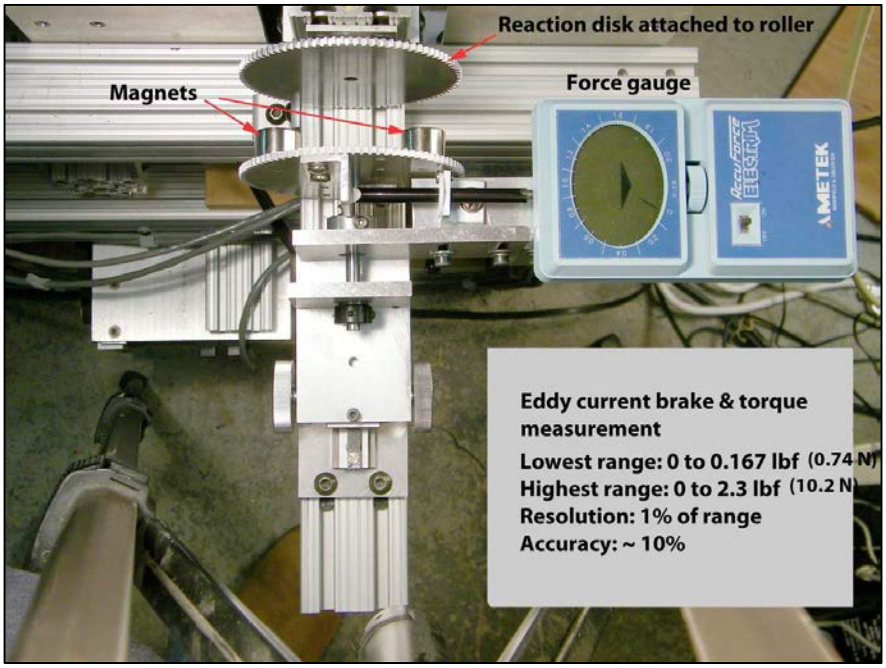


Figure 5
Eddy current brake and torque measurement

The drive end:

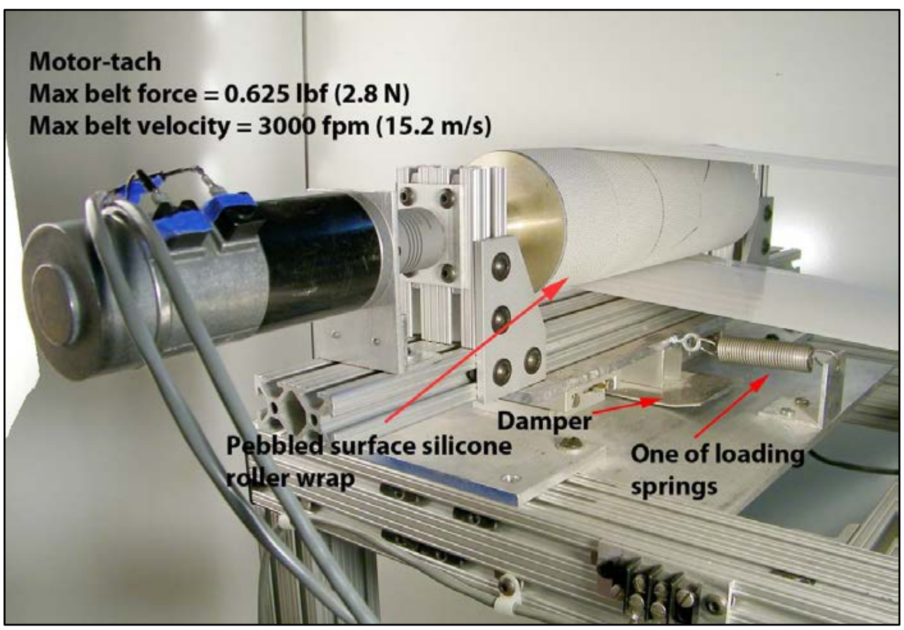


Figure 6
Drive end of belt

The motor is directly coupled to the drive roller. For a maximum web speed of 3000 fpm (15.2 m/s), a maximum motor speed of 3,800 rpm is required.

A pebble surface roller wrap was used to improve traction at high speed.

To aid in keeping the web centered, a slight crown was created by putting two layers of masking tape under a 6 inch (152 mm) wide section of the wrap in the middle. The crown was not much help and it caused wrinkling at tensions more than 0.2 pli.

A flat plate, is attached to the roller assembly to form a damper. It is spaced 1/32 inch (0.8) mm above the mounting plate and the gap is filled with petroleum jelly.

Stick and microslip zones:

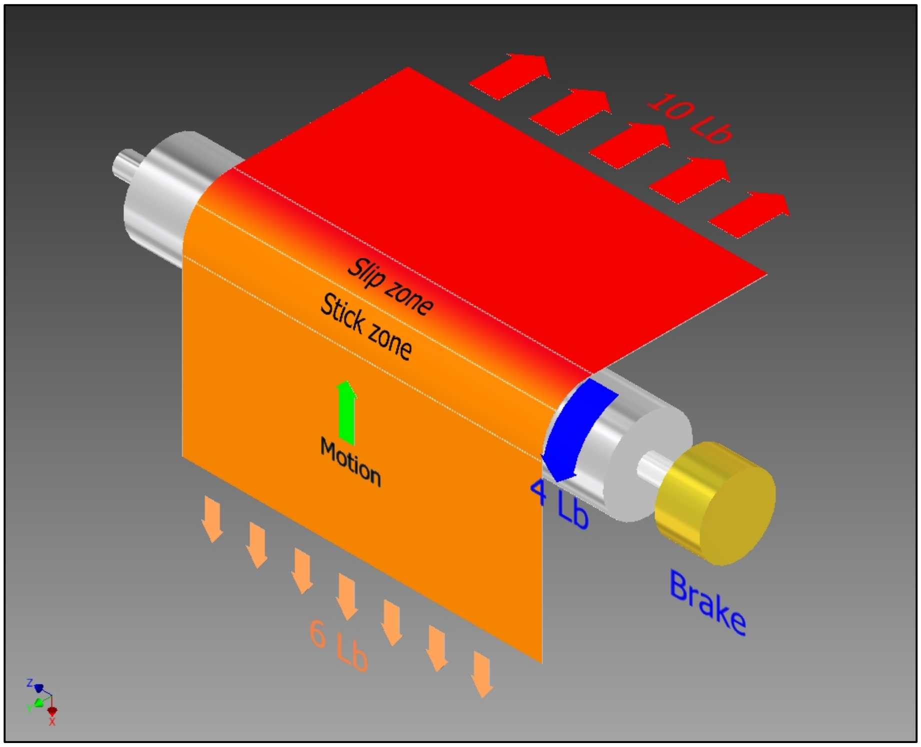


Figure 7

Stick and microslip zones on a braked roller

Even rollers without brakes or drives will have some resistance to turning due to bearing friction (and that friction will increase with speed). This causes a microslip zone to form at the exit of the roller. The term microslip refers to slipping that's caused by stretching of the web on the roller surface. It's generally not large enough to cause scratching. But, it is the zone in which torque is transferred between the web to the roller and it is there that any change in tension occurs. The microslip zone is always at the exit and it will grow as braking force is increased until it completely consumes the wrap angle. At that point, the web will break away from the roller and there will be gross slipping (that can scratch the web).

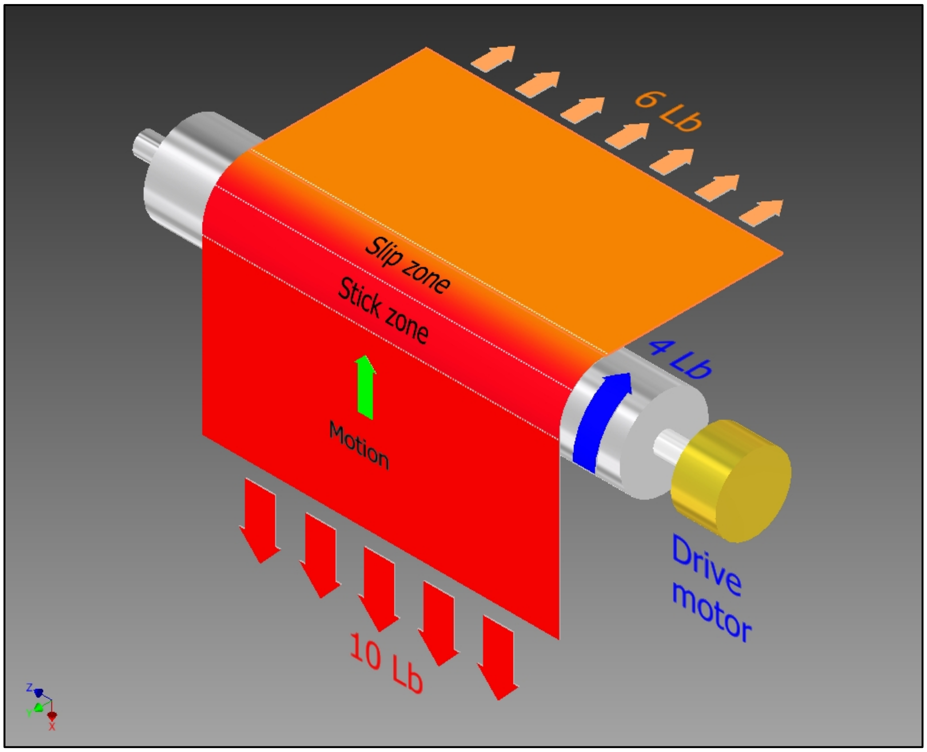
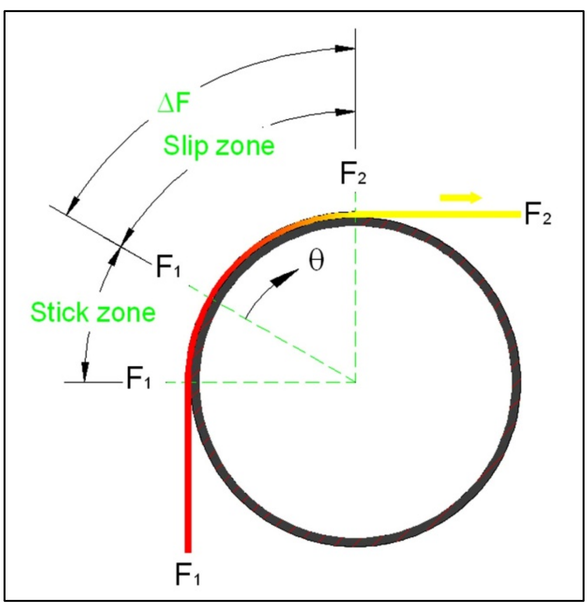


Figure 8
Stick and microslip zones on a driven roller

Capstan equation:



$$\frac{F_2}{F_1} = e^{f\theta}$$

f is the coefficient of friction

Θ is the angle of wrap

Figure 9
The capstan equation

If we know, F_1, F_2 and Θ, we can calculate f .

$$f = \frac{1}{\theta} \ln \frac{F_2}{F_1}$$

Or if we know F_1, F_2 and f, we can calculate Θ.

$$\theta = \frac{1}{f} \ln \frac{F_2}{F_1}$$

Bearing friction:

It is important to note that the torque measurements at the eddy current brake do not include the roller bearing friction. So, the values of ΔF reported here include estimates of bearing friction based on a spin-down test (similar to the one described by Jim Dobbs in "Measurement and Modeling of Bearing Drag in Idler Rollers"). This indicated significant viscous drag in the bearings and was made while the bearings were cold. So, even though the bearings were lubricated with light oil, it is quite likely that the friction changed to some extent depending on how long the belt had been running. The friction model is shown below. Values above 800 ft/min (4.06 m/s) were extrapolated.

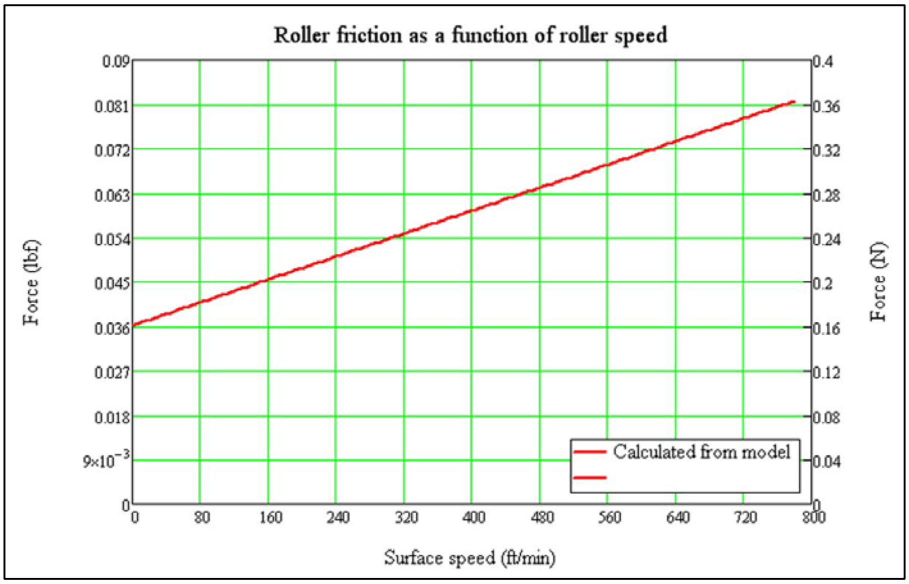


Figure 10
Bearing friction model

The slip curve (web speed vs. roller speed):

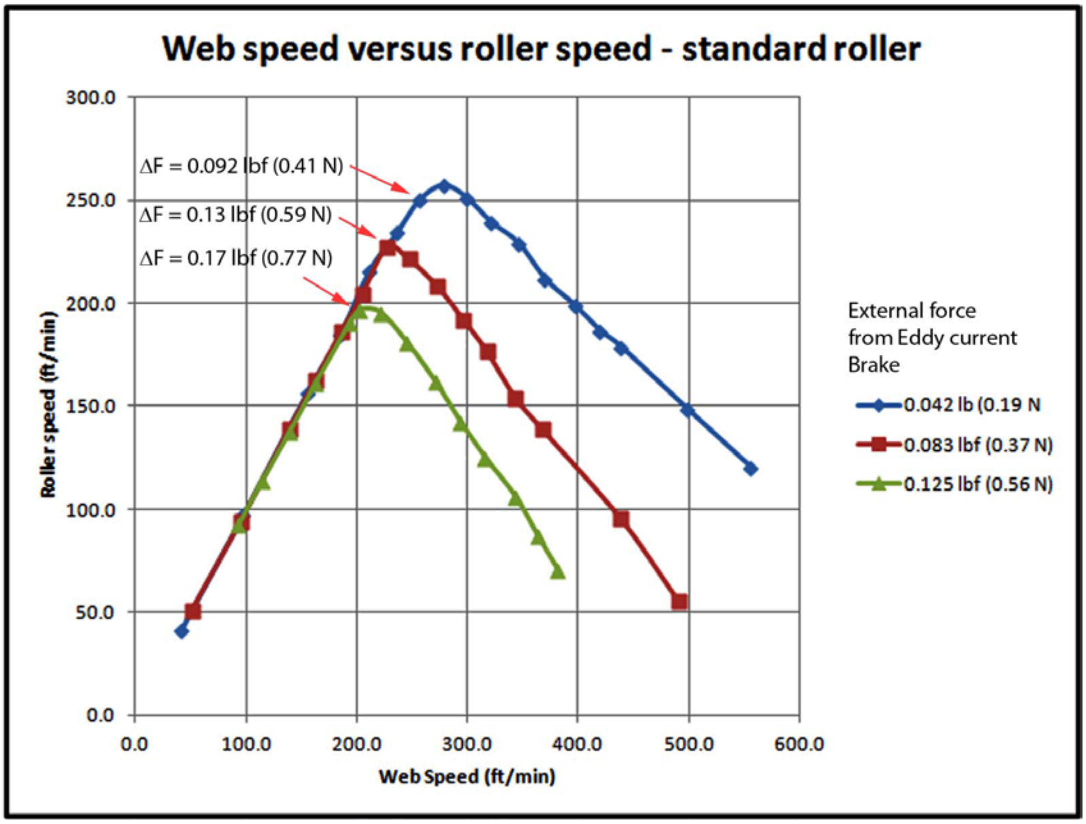


Figure 11

Web speed vs. roller speed 20 degrees from exit of 180 degree wrap

Each curve was produced as follows.

- The drive speed was set, starting at a low value.
- The eddy current brake was then adjusted to set the value of the external torque to a constant value.
- Then, the times for one revolution of both the roller and the belt were recorded.

Each curve shows a breakaway point where the roller velocity peaks and begins declining relative to the web velocity. This point is reached at lower speeds as ΔF is increased. This is an indication of air lubrication.

As the web velocity is increased beyond the breakaway point, the roller velocity drops, reducing the air film thickness until the web reengages the roller to keep it moving at a lower speed. This part of the curve can be erratic since the traction is very small.

Asperities

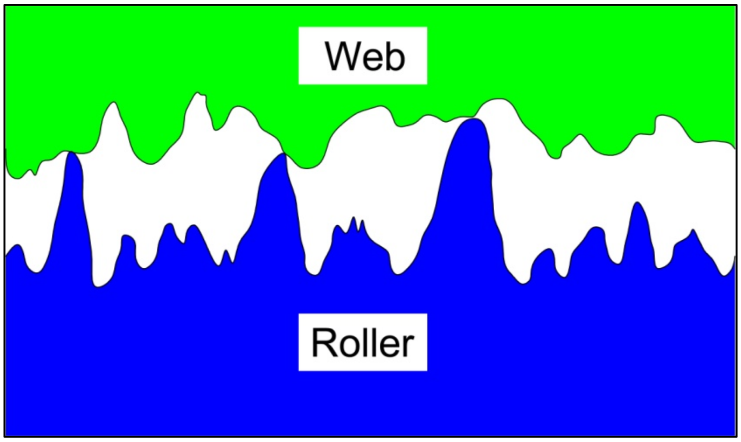


Figure 12
Air space between the web and roller created by asperities (roughness)

Real webs and rollers are never in complete contact. At the microscopic level where air lubrication really matters, only a small percentage of points touch. The contact points are at peaks in the roughness profile called asperities. Up to the point where the web breaks away from the roller, the air flow is through the space created by these asperities.

Air lubrication:

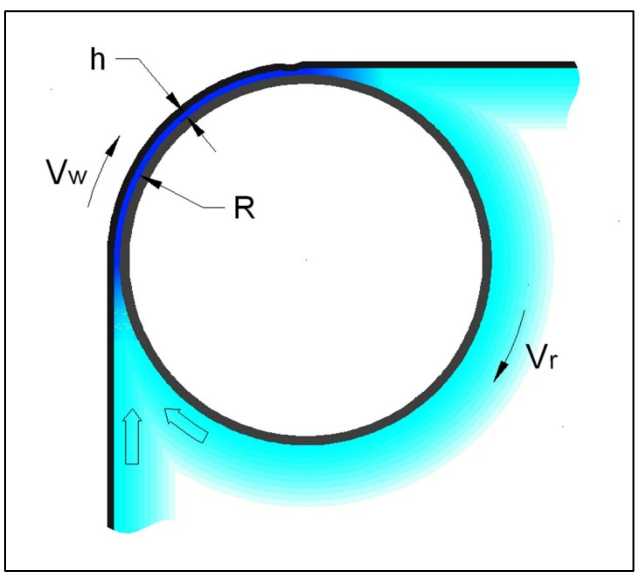


Figure 13
Air lubrication

Air is pumped into gap h by boundary layers that form on the moving surfaces.

At speeds below breakaway, the air flows through spaces created by the asperities.

As web speed increases, air pressure in the asperity space rises. This opposes the pressure due to tension, which, in turn, reduces friction.

At some speed, the air pressure will become high enough that the web will completely lose contact with the roller (the asperities separate).

Except for a small dip near the exit and narrow zones at the edges, the effective thickness of the entrained air film is nearly constant over the entire angle or wrap.

Increasing the roughness of the roller surface increases the asperity space, allowing the web to maintain contact at higher speeds.

The foil bearing equation:

If a web is nonporous, the thickness of the entrained air film, h, can be calculated using the foil bearing equation (sometimes known as the Knox-Sweeney equation).

$$h = 0.643R\left(\frac{6\mu\left(V_w + V_r\right)}{T}\right)^{\frac{2}{3}}$$

R = roller radius

V_w = the web speed

V_r = roller surface speed

T = the web tension (in units of force per unit of web width and including centripetal effects)

μ = the viscosity of air

There is a subtle point about the foil bearing equation that is often misunderstood. The equation is precise only if the web and roller are perfectly smooth. For rough surfaces, h is fixed to an effective value, h_{eff}, that is determined by the volume of air space that isn't occupied by asperities. In that range of speeds, the air pressure in the asperity space opposes the pressure created by web tension; but it isn't high enough to lift the web. So there will be a net pressure pressing the asperities together. Pressure in the asperity space goes up as the web speed goes up until if finally balances all of the web tension and all the traction is lost.

When calculating the pressure effects of a web on a roller, centripetal force must be taken into account. It is usually incorporated in the nominal web tension when doing calculations of the pressure between the web and roller.

$$T = T_w - d\rho V_w^2$$

Where T is the tension adjusted for centripetal force when it is on a roller, T_w is the tension in the web when it is not on a roller, d is thickness, ρ is density and V_w is web speed.

Putting it all together (the friction curve):

When the web slips (breaks away from the roller), the angle Θ in the capstan equation is equal to the wrap angle. So, if we know the entering and exiting tensions it is possible to use slip curves to calculate the effective coefficient of friction. Data from the slip curves can then be used to calculate the effective coefficient of friction in the presence of air lubrication. We would expect it to decline as speed increases.

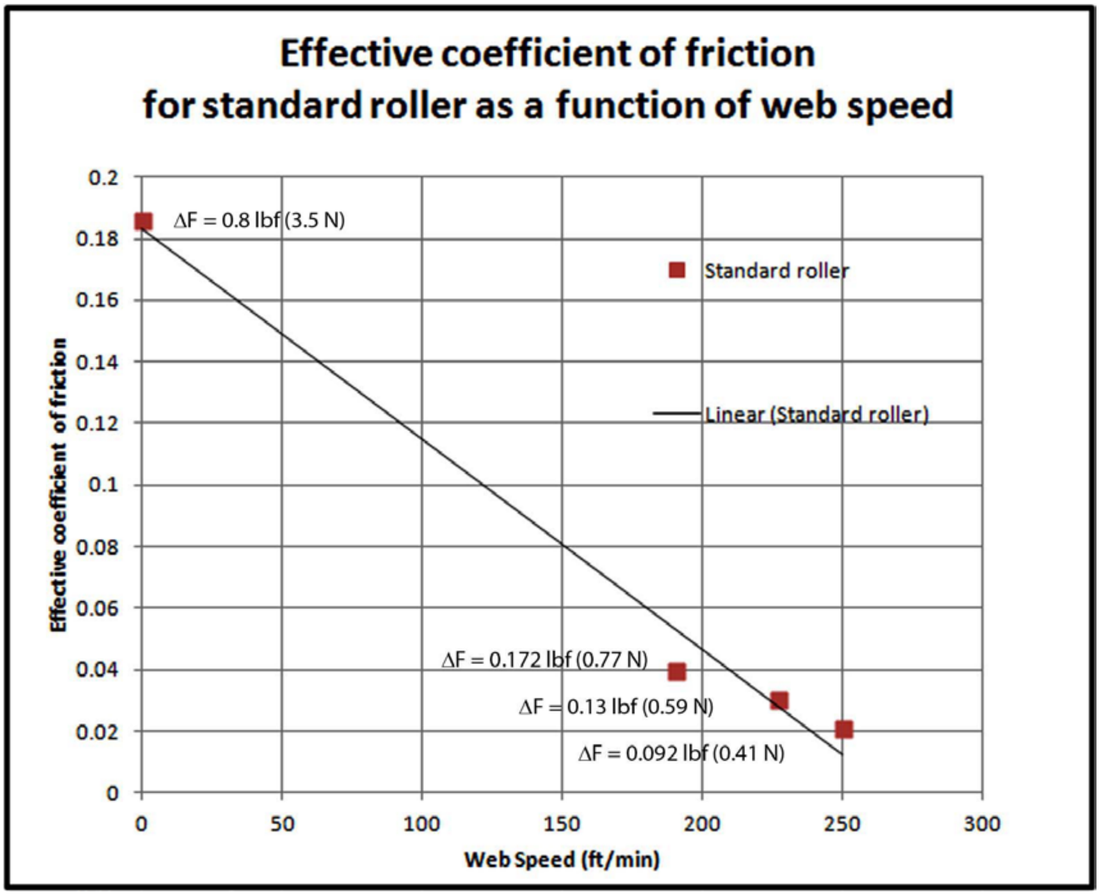


Figure 14
Coefficient of friction as a function of line speed for the standard roller

This graph and others in this paper have only a few data points at the low end plus the static coefficient value at zero speed. These aren't intended to "prove" that the complete curves look like the solid black line. That work has been done by many other researchers. Graphs like this can be found in their papers and dissertations, for example in "Reduction in Web to Roller Traction as a Result of Air Lubrication". In those, many more points are plotted over the relevant speed range. They all show this general shape. The curve may be slightly bowed in some cases. But, it tends to be close to a straight line, like the linear regression curve shown above.

The important thing to realize is that the coefficient of friction begins to drop as soon as the web starts moving and will continue to drop with speed until a point is reached where the web breaks away from the roller and begins to slip.

So, what do you lose when the web slips:

The normal entry rule no longer works. This has two important consequences.

First, (the bad news) the web's location on the roller is no longer determined by its angle of entry. So, it's lateral position will be unstable, changing with tension and other things. The web may also be scratched.

Second, (the good news) it will be harder for wrinkles to form because the normal entry effect is an important factor in turning troughs into wrinkles.

What to do about slipping:

Make the roller surface rougher (bigger asperities). Or create artificial asperities with grooves.

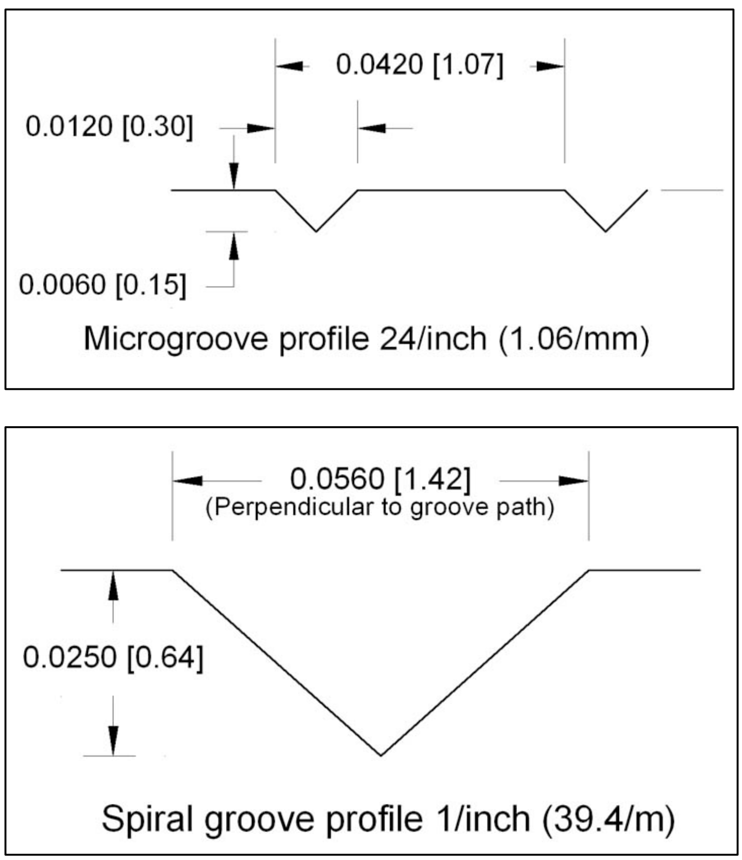


Figure 15
Groove profiles

One way to look at grooves is that they give the air film a place to go so that the web can rest on the flat areas between them.

Grooving like that shown for the microgroove profile is likely to produce good traction in a converting application with typical running speeds and tensions. However, when moving into new process territory, there are things to consider. For grooves to be effective, they must have enough volume to carry away the entrained air. But, a groove that is very deep compared to its width won't work well. Also, the air that is over the lands (the flat portion between the grooves) takes time to be squeezed laterally into the grooves. In some cases, the web will have passed completely through the wrap angle before the web has time to land.

Take for example an 18 inch (457 mm) diameter roller running at 1100 fpm (5.1 m/s) with 0.2 pli (35 N/m) tension. If the web and roller were perfectly smooth, the air gap would be 6 mils (0.15 mm). So roughening the roller surface wass not an option. A surface with asperities of that order would have damaged the web. A special groove profile was needed. To keep manufacturing costs down it needed to be cut as a single helix with a pitch that was as large as possible (the face length was 354 inches). The groove width needed to be small enough that the web wouldn't be pulled down into it and the land width had to be small enough that there would be sufficient time for the web to settle on the surface before it left the roller (the web speed was over 1000 fpm (5.1 m/s). The depth to width ratio had to be kept small enough that the drag of the walls wouldn't spoil the venting effect. The groove we settled on was rectangular, 45 mils (1.14 mm) wide and 30 mils (0.76 mm) deep with 16/inch (6.3/cm).

Comparison of the three rollers:

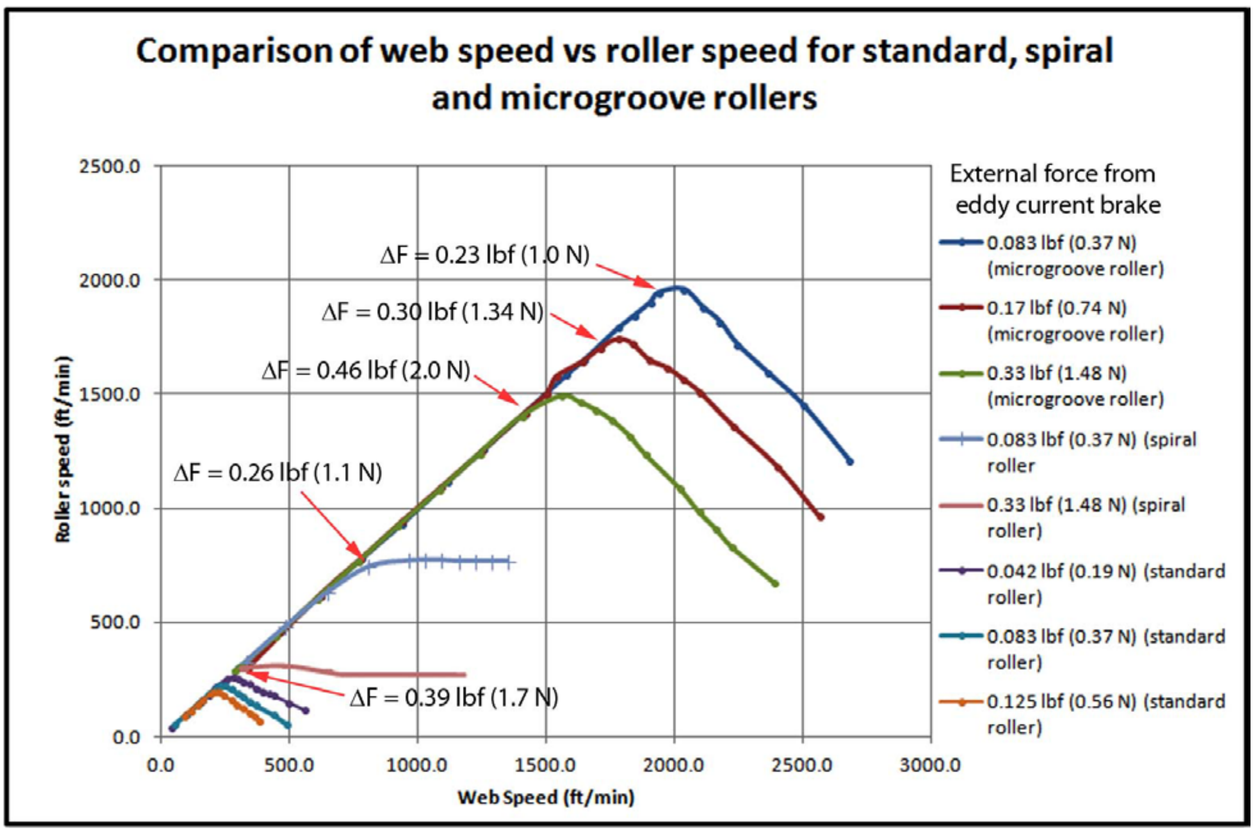


Figure 16

Slip curves for the three rollers

It is clear that the microgroove does an excellent job of venting the entrained air. It holds on until 2000 fpm (10.2 m/s) compared to less than 200 fpm (1.0 m/s) for the standard roller under comparable conditions.

The spiral groove roller also does significantly better than the standard.

Comparison of the friction curves for the three rollers:

The friction curves provide a more meaningful display of the data.

It's worth noting that a person could get a very rough estimate of curves like this by taking two measurements. The zero speed coefficient can be measured by using a force gauge to pull a strip of web over a stationary roller with a known weight on the free end and then using the known forces and wrap angle in the capstan equation (make the force measurement while the web is moving). The other point would correspond to zero friction (actually the bearing friction) and would be measured by noting the speed at which the web starts slipping. Detecting the point of slipping would be the biggest challenge. You need an accurate measurement of line speed. The roller speed might be measured using a strobe light and a piece of tape on the roller surface.

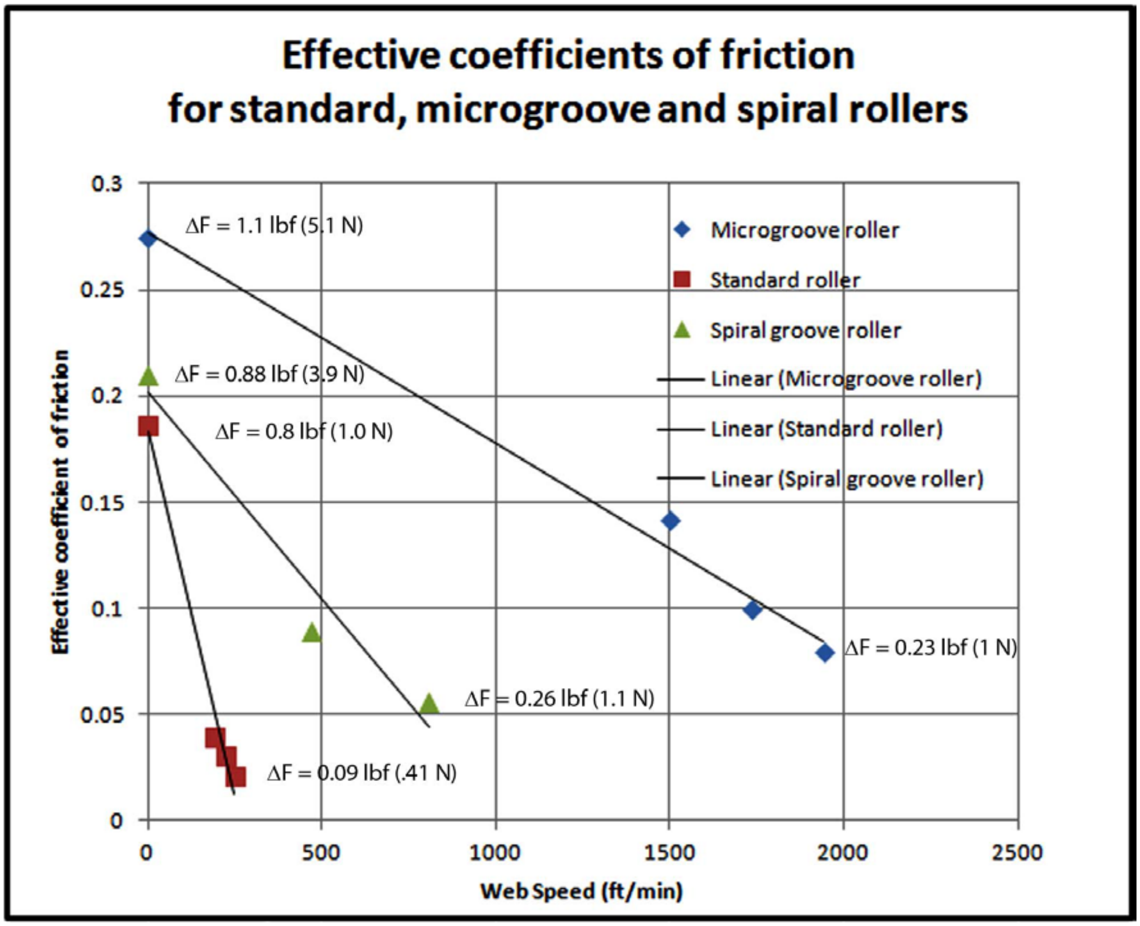


Figure 17

Friction curves for the three rollers

Air gap profiles of the microgroove roller:

At 5,000 samples/sec, and with the web traveling at 2000 fpm (10.2 m/s) the laser sensors are fast enough to resolve a distance of 0.080 inch (2 mm) on the circumference of the roller. At first sight, this doesn't look good enough to make measurements on a groove that is only 0.012 inches (0.152 mm) wide. However, when viewed circumferentially, the groove has a width of 2.714 inches. So, it is quite adequate.

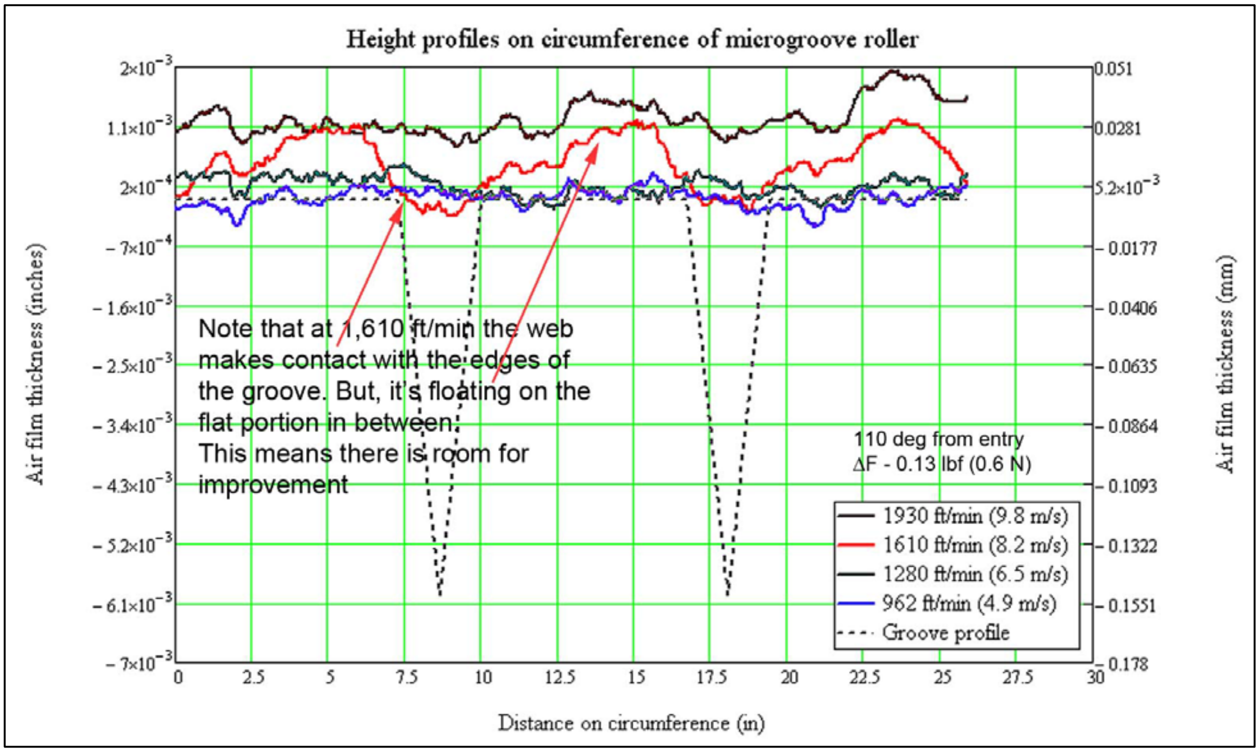


Figure 18

Height profiles on circumference of the microgroove roller at four speeds

At speeds 962 fpm and 1,280 fpm, the web is in full contact with the roller surface and is not pulled into the grooves. At 1,610 fpm the web has lost contact with the lands. But, it appears to have some contact at the edges of the groove. It looks like it has been pulled down into the groove, plugging it so that the rest of the air can't get off the land. At 1,930 fpm the web is completely floating. It is apparent that this roller could be improved by reducing the land width. This could be done several ways.

The groove width and depth could be doubled. Or two grooves of the current design could be cut simultaneously (a double helix) to create 48 grooves per inch.

Conclusion:

The purpose of the foregoing data is only to illustrate “how things go”. Results will be different depending on just about everything.

Roller diameter

Tension

Bearing drag

Web material and roller surface finish

Grooving pattern

If you want to learn more, the following references are good sources. And, of course, there are the AIMCAL training seminars conducted by other consultants here today.

References:

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