EXPERIMENTAL STUDY OF IDLER ROLL SLIPPAGE

by

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ABSTRACT

Idler roll slippage occurs when the roll surface velocity no longer equals the velocity of the web that transports the roll. In a web transport system, slippage will produce scratches in sensitive, coated web or cause process problems such as loss of lateral tracking. This paper presents the results of an experimental study on roll slippage with polyester film and several different roll surfaces at web speeds to 1000 fpm.

A test stand was installed in a laboratory based web transport line. This allowed for the control of tension and the continuous measurement of torque, tensions and speeds. Tests were conducted using four different polyester films. Roll surfaces that were tested consisted of ground metal, rubber covered, grooved and knurled. The effects of wrap angle and outgoing tension were also studied.

Slippage was found to vary continuously from 0.05 to 100% depending on the amount of applied braking torque. The parameter used to quantify the onset of slippage was "effective coefficient of friction". It was determined experimentally when the difference between web velocity and roll velocity was 0.25% of the web velocity. Reasons for this definition are discussed in the paper. The standard belt equation was used to calculate the coefficient from the test results. The change in the coefficient with web velocity, roll surface, wrap angle and film surface are presented. A roughened knurled roll provided the most traction and had fewer tendencies to slip at higher speeds.

NOMENCLATURE

AA	roll roughness (μ inch)
r	roll radius (in)
V_f	film velocity (in/sec)
V_r	roll velocity (in/sec)
T_{in}	incoming tension (lb)
T_{out}	outgoing tension (lb)
μ	coefficient of friction (lb/lb)
μ_{eff}	effective coefficient of friction (lb/lb)
θ	wrap angle (radians)
τ_B	braking torque (in-lb)
τ_{max}	maximum braking torque (in-lb)
Ln	natural log function

BACKGROUND

The ability of an idler roll to maintain the same velocity as the web, which it transports, depends on a number of factors. The most important ones are: roll surface, web velocity, web surface friction drag, wrap angle and tension level. The two surface condition factors are usually lumped into a parameter referred to as the coefficient of friction. An idler roll is said to be slipping when its surface velocity is not equal to the web transport velocity. In a web line, slippage can cause scratches in the web, a loss of lateral tracking stability or web tension variations due to irregular roll motion.

Early studies on belts found that there was a maximum torque that could be transmitted by the belt without slipping. This condition is described by the familiar "belt equation" which relates the incoming belt tension to the outgoing belt tension.

$$T_{out} = T_{in}\, e^{\mu\theta} \tag{1}$$

In the case of an idler roll at constant velocity, T_{out} is always higher than T_{in} and the braking torque is related to the tension drop across the roll by the following:

$$\tau_B = r[\, T_{out} - T_{in}\,] \tag{2}$$

Thus, the maximum braking torque at the point of slippage can be expressed in terms of either T_{in} or T_{out}. For example,

$$\tau_{max} = rT_{out}\,[\, 1 - e^{-\mu\theta}\,] \tag{3}$$

Remember that this is the critical torque at the onset of slippage. For non-slippage, equation (1) does not apply and the relationship between T_{in} and T_{out} is given by equation (2) where the braking torque is known. This early analysis assumed that the coefficient of friction was constant and did not take into account the effect of air entraining or surface conditions.

Since then, more work has been done on the problem of slippage and in particular, the problem of slippage between an idler roll and a moving web. Good and Ducotey

(Ref. 1,2) have done a significant amount of work, which takes both surface effects and air entrainment into account. Their work included both analytical and experimental studies and utilized the earlier work of Knox and Sweeney (Ref. 3) on air entrainment. Knox and Sweeney derived an empirical equation, which described the riding height between a stationary roll and a moving web. Jones and Zahlan (Ref. 4) used finite element analysis to investigate slippage on driven rolls. Additional information on the problem can be found in the foil bearing and lubrication literature, such as those in Ref. 5 and 6.

INTRODUCTION

At the time the authors conducted this study (1984), several things were known about the traction of a web over an idler roll. The first was the concept of the coefficient of friction between a roll surface and a film surface. When the coefficient of friction is known, the belt equation was used to predict the onset of slippage. Knowing the roll radius, wrap angle and outgoing tension, the torque it takes to cause slipping can be predicted using equation (3). This classical theory can be easily described with the slippage versus braking torque curve shown in Figure 1. The curve is a simple backward "L" and the critical torque is τ_{max} from equation (3). The roll will turn at web speed when the braking torque is below τ_{max} (ratio < 1). Above the critical torque, the roll slips completely and its velocity goes to zero. With this background, the test was designed to include rolls of different roughness, film surface types and an adjustable, electromagnetic torque generator to apply the braking torque.

From the earlier work of Sweeney and Knox (Ref. 3) and our experience with film lines, the concept of air entrainment and its negative effect on roll traction were also known. It was postulated that air entrainment reduced the coefficient of friction. Thus, both grooved rolls and a knurled roll were tested because they would allow for the escape of air through the contact region. Low wrap angles were investigated because of their effect on reducing the critical torque for slippage.

EXPERIMENTAL STUDY

An experimental study was outlined with the specific objectives of developing tools to predict the onset of slippage and identifying roll surfaces that would prevent slippage at high speeds. Figure 2 shows a schematic of the test setup and data acquisition. The test stand was located on a laboratory web transport line with a tension controlled process zone between two pull rolls. Thus, the web tension downstream of the test roll at loadcell roll 3 could be controlled during the experiments. An electromagnetic brake was mounted to the test roll shaft to control the amount of braking torque applied to the roll. A torque transducer was used to measure the actual applied torque. A movable idler in front of the test roll was used to set the wrap angle.

Six measurements were taken during each test. Test roll and web and velocities were measured with a precision, encoder-based velocity transducer. Loadcells were used to measure the controlled tension, incoming tension and outgoing tension. The torque from the torque transducer was recorder continuously. Data were collected and recorded continuously as shown in Figure 2. The test variables are outlined in the following table:

Roll surface: flat metal (45, 60, and 135AA), flat rubber (75AA), grooved rubber (1 tpi, 75AA), knurled metal (21 tpi, 77 AA)
Speed setting: 100 to 1000 fpm
Tension setting: 0.5, 1.0, 2.0 pli
Wrap angles: 30, 60, 90, 180 degrees
Film surface: Polyester types A, B, C and D

A preliminary test was designed to measure the inherent drag in the test setup. With a small drive motor, it took less than 0.3 in-lb of torque to turn the brake and torque transducer. This inherent drag would be added to the applied torque. A second, preliminary test was conducted to determine the bearing drag for grease versus oil lubrication. Figure 3 shows the results for speeds to 650 fpm. With grease lubrication, the drag torque continued to increase with speed. After several minutes of running, the warm grease bearings had a reduced drag torque. The combined drag for the oil-lubricated bearings was less than 0.5 in-lb as shown on Figure 3. Therefore, the study was run with oil-lubricated bearings and 0.5 in-lb was added to the applied brake torque.

The test procedure was to set the film speed and then increase the braking torque in small increments until slippage occurred. Figure 4 shows a typical test run for the grooved rubber roll at 500 fpm. The film was type B polyester at 60 degrees of wrap and 1 pli of tension. The film and roll speeds are shown in traces 1 and 2. The expanded scale needed to observe roll speed is shown in trace 7. As the brake torque (trace 3) was increased in steps, the incoming tension (trace 4) decreased and the roll speed started to decrease. The steps were continued until a 1% (5 fpm) decrease in speed was obtained. Notice that the outgoing tension remained constant during the test as desired. This test procedure was continued for each test roll, film surface, speed setting and test condition.

DISCUSSION OF RESULTS

The key measurements were roll velocity, film velocity and braking torque. The parameter used to compare this data was called **% Slip** and was defined as follows:

$$\% \text{ Slip} = [V_f - V_r] / V_f \tag{4}$$

Roll Slippage Results

When % Slip was plotted versus braking torque, an interesting J shaped curve resulted. The J curves in Figure 5 show the effect of roll roughness on slippage. At 400 fpm and 60 degrees of wrap, the higher the roughness the more braking torque it took to produce a significant amount of slippage. The shape of the curves also illustrated an important point. That is, any amount of slippage could be produced with the appropriate amount of braking torque. This result was not expected, as discussed previously. The curves get very steep above 5% and data was difficult to obtain without an extremely precise torque generator. The effects of web velocity and air entrainment on slippage are shown in Figure 6. At speeds below 300 fpm, the torque required to produce slippage remained about the same. However, at the higher line speeds of 400 and 500 fpm it took less braking torque to initiate slipping. Thus, for a 60AA roll with 60 degrees of wrap and 2 pli of tension, air-entraining effects started above 300 fpm.

Effective Coefficient of Friction Results

Plotting all the test data as slippage versus braking torque would result in a large catalog of curves for every possible wrap angle, tension level, roll surface and film type. A single parameter was needed to describe the onset of slippage or traction limit. The chosen parameter was the "effective coefficient of friction" defined by the belt equation. Each roll surface and film combination would have a coefficient, which varied with film speed. However, the coefficient would have to be computed at the point of slippage. This required that the onset of slippage be specified. Slippage was arbitrarily defined to occur when the % Slip value reached 0.25%. This was usually a well-defined point on the J curve and well above the normal slowdown needed to unstretch the incoming web. For example, in our test system with 1 mil thick, 26 inch wide polyester film at 1 pli, a braking torque of 10 in-lb would have slowed the test idler down 0.05% and unstressed the incoming web. This curve is also shown on Figure 6 and represents the slowdown of the idler when the load is applied and no slippage occurs. Change in velocity above this level is actual slippage of the web relative to the roll.

Thus, the "effective coefficient of friction" was defined by the following equation, which results from the combination of the belt equation and the tension-torque relationship.

$$\mu_{eff} = \mathrm{Ln}[(1 - \tau_{max}/rT_{out})^{-1}]/\theta \qquad (5)$$

The torque τ_{max} was evaluated at 0.25% slip. Figure 7 shows change in μ_{eff} for the four different film types at film speeds to 1000 fpm. The curves show a slight difference in μ_{eff} for the film types and a decrease with increasing speeds. If air entrainment were not present, the curves would be horizontal.

Air was attempted to be removed by testing a rough roll (135AA), grooving a rubber roll and knurling a metal roll. The results are shown in Figure 8. For the rough, 135AA roll, although initially high, the coefficient decreased with increasing film speed. The coefficient for the flat rubber roll dropped off rapidly. However, when this roll was grooved the coefficient remained nearly constant. The two knurled metal (77 and 90 AA) rolls, although initially quite smooth, performed well when knurled.

The two other parameters that might affect the friction coefficient were the wrap angle and outgoing tension level, T_{out}. Figure 9 shows the result for the 77 AA, knurled roll at different wrap angles. Thus, regardless of film speed, the friction coefficient (μ_{eff}) did not change significantly. The same was obtained for the flat rubber roll which experienced air-entraining effects. Figure 10 compares μ_{eff} for the 77 AA knurled roll with a 60-degree wrap angle at two tension levels. The higher tension level did increase the coefficient slightly. It was postulated that a higher tension level would pull the film around the knurls more, thus increasing the traction.

CONCLUSIONS

This study started out to be a simple, experimental determination of the effects of film surface and roll surface on idler roll slippage. However, it soon became much more complicated. The normal procedure for measuring a coefficient of friction is to collect data at the point of impending slippage and then to calculate μ using a traction formula like the belt equation for webs on rolls. However, as the braking torque was increased during the test, the idler roll would slow down to a new speed and then track the film at a speed ratio slightly less than 100%. This effect was referred to as the "J" curve by the authors. The conclusion for this observation can only be that the traction force actually increases with slippage until the traction force equals the applied braking torque and equilibrium is again reached. Thus, the roll would seek a higher % Slip where the traction was high enough to offset the braking torque.

Rather than plot all the data as % Slip, a parameter called the "effective coefficient of friction" was defined. It was calculated for each test result when the % Slip factor reached 0.25% using the standard belt equation. This parameter could be used in the future to predict the amount of braking torque that would produce slippage for a given wrap angle and tension level. The coefficient varies with each roll type, film surface and film velocity and must be determined experimentally. From the testing, a minimum value for the "effective coefficient of friction" was concluded to be about 0.1. Below this value, normal idlers with low wrap angles will have a problem tracking the film.

One of the unique features with the test setup was that the test roll was located in a film transport line rather than a continuous loop test stand. This arrangement permitted us to keep the downstream tension constant during each test. This more closely represented an actual web line. Also, new film was always crossing the test roll and previous scratches and scuffs on the film did not influence the measurements.

The effect of entraining air was also observed during the study. It was found that it could be eliminated with grooving and knurling. For these rolls, μ_{eff} would remain nearly constant with film velocity over the test range.

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5. Licht, L. "An Experimental Study of Elastohydrodynamic Lubrication of Foil Bearings," Journal of Lubrication Technology, Jan. 1968.
6. Gross, W. A., editor, <u>Fluid Film Lubrication</u>, John Wiley and Sons, New York, 1980.

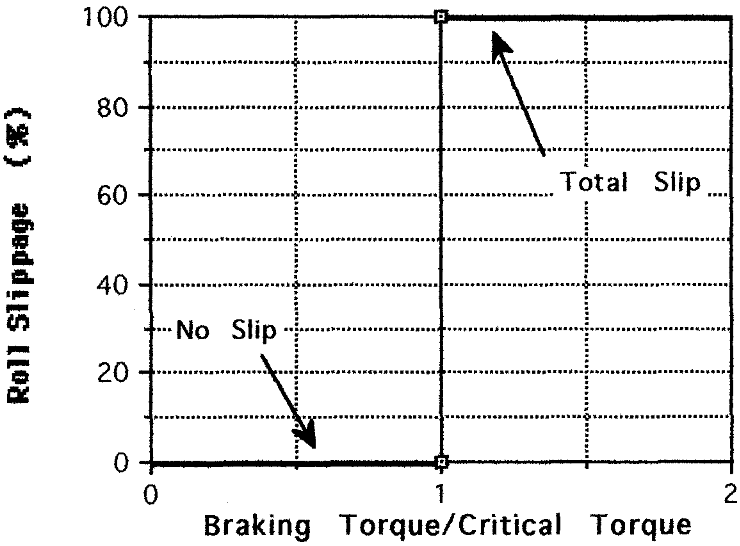

Figure 1. Classical Theory of Roll Slippage

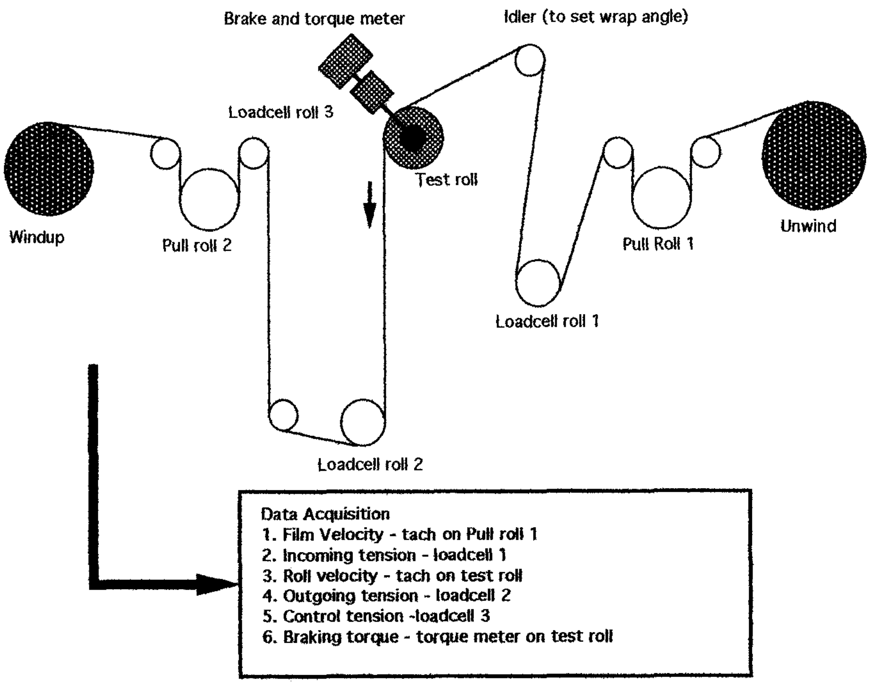

Figure 2. Schematic of Test Stand and Data Acquisition

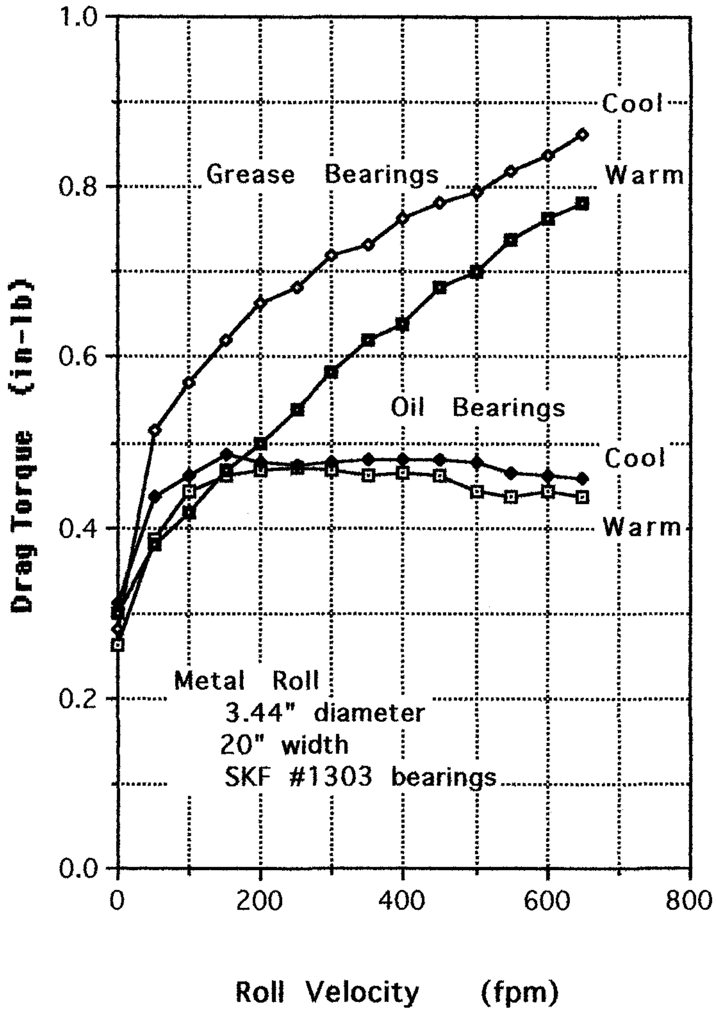

Figure 3. Bearing Drag for the Test Roll

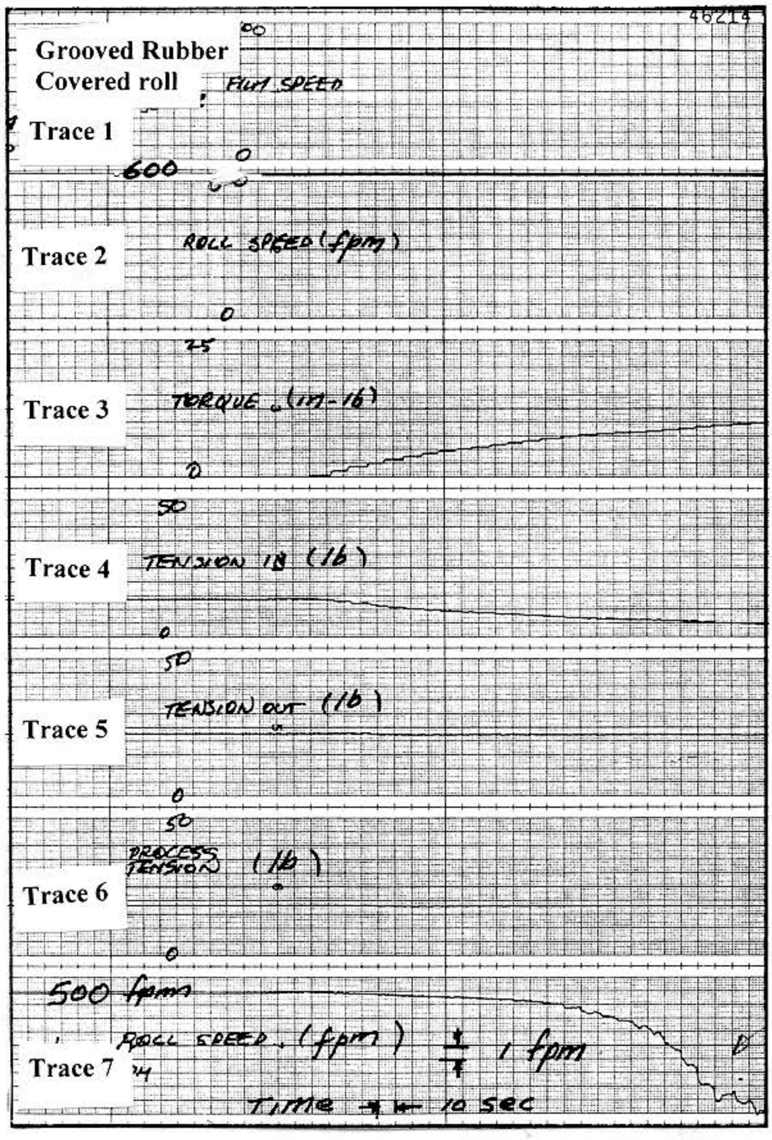

Figure 4. Time Data Recording from Grooved, Rubber Covered Roll, 60 Degree Wrap, 1 pli, 500 fpm and Film B

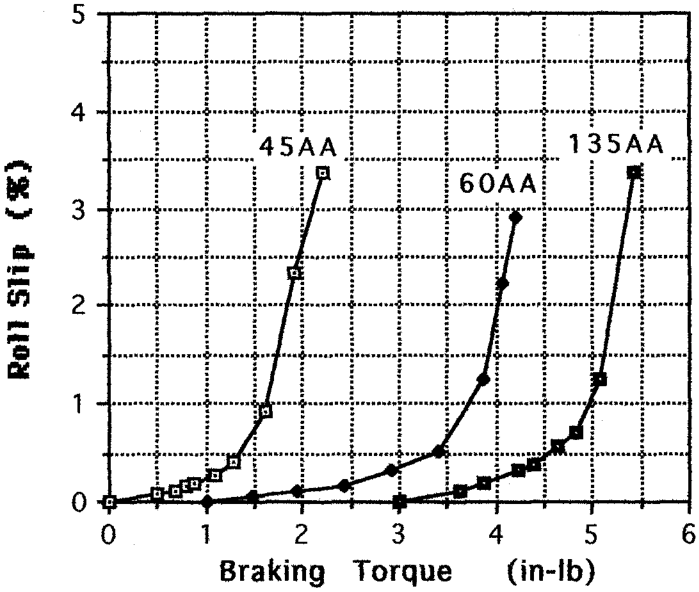

Figure 5. Slippage Curves for Three Metal Rolls at 400 FPM, 1 pli, 60 Degree Wrap Angle, Film A

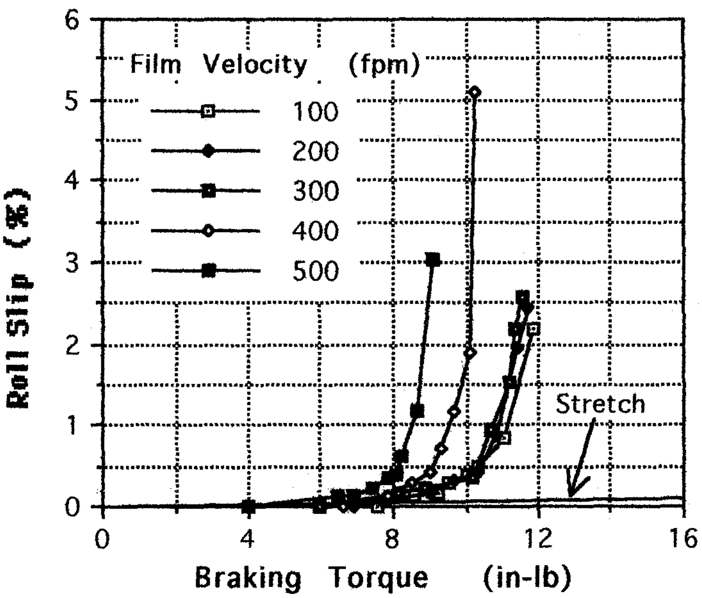

Figure 6. Effect of Film Velocity on Slippage for 60AA Roll, 2 pli, 60 Degree Wrap, Film A

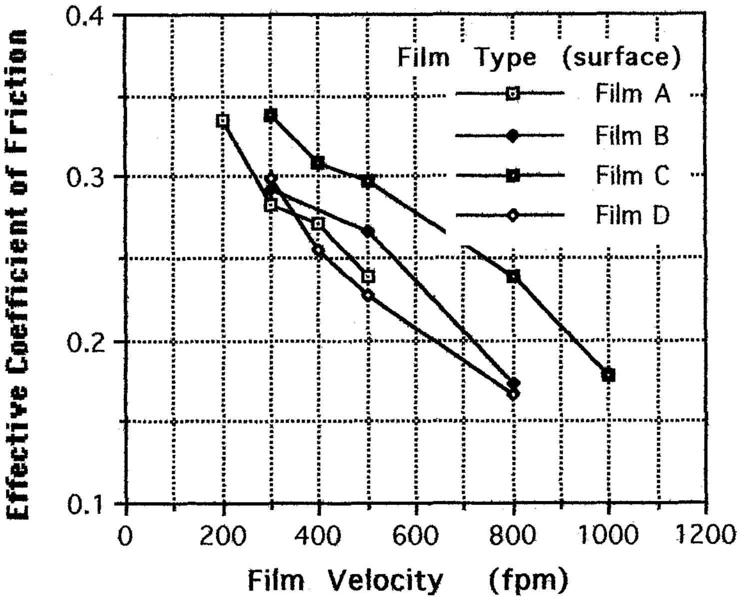

Figure 7. Effect of Film Surface on 135AA Metal Roll, 1 pli, 60 Degree Wrap Angle

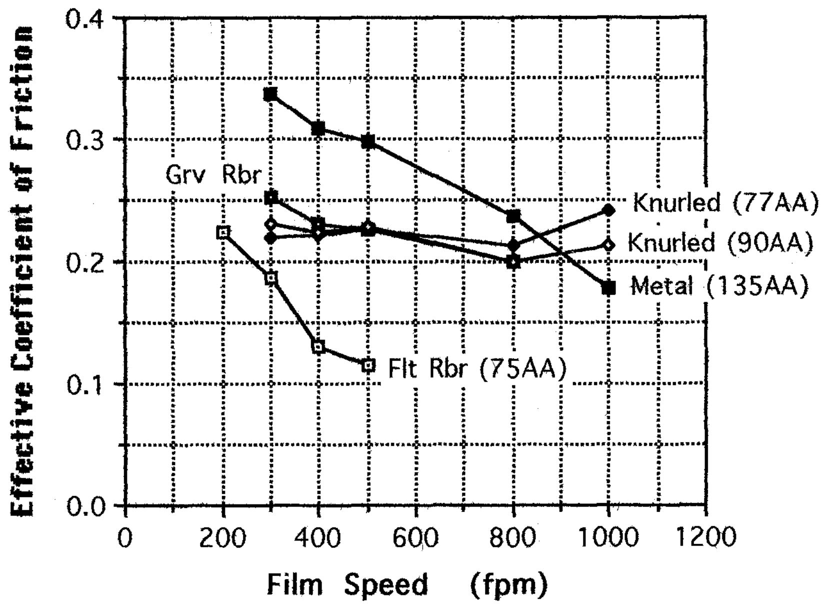

Figure 8. Effect of Roll Surface on Film C, 1 pli, 60 Degree Wrap Angle

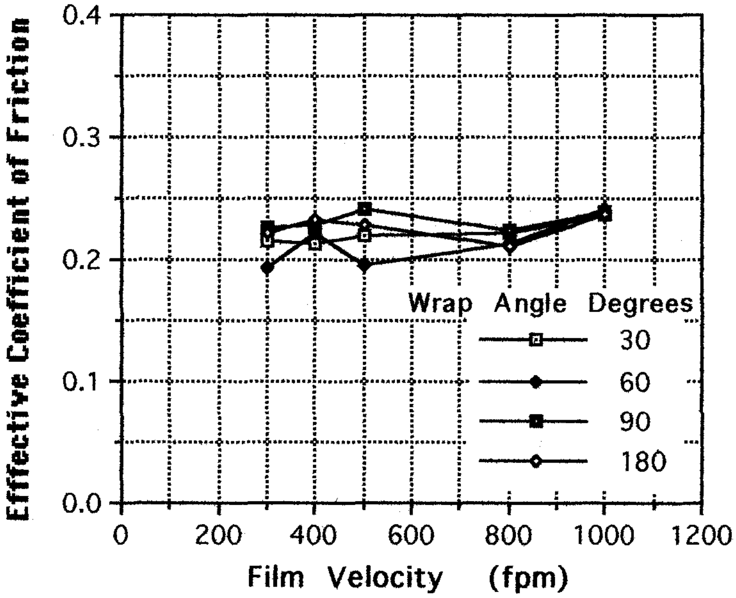

Figure 9. Effect of Wrap Angle on a Knurled Roll (77AA), 1 pli and Film C

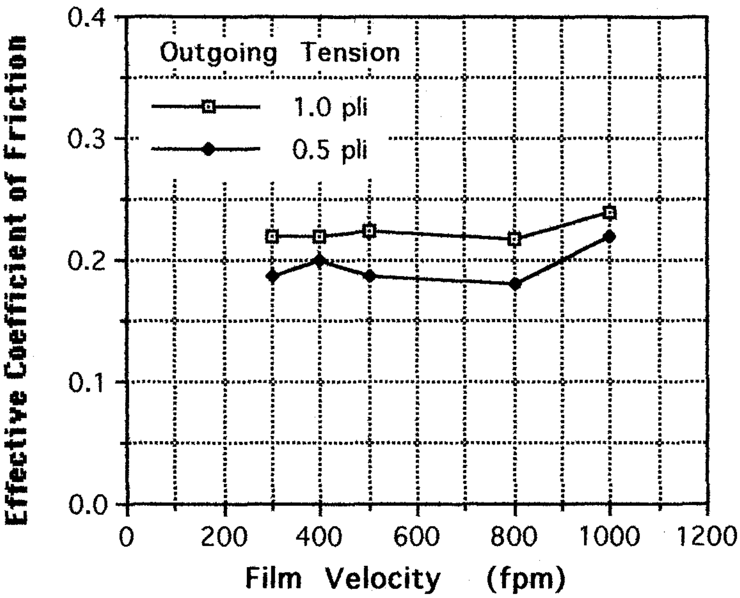

Figure 10. Effect of Outgoing Tension on Knurled Roll (77AA), 60 Degree Wrap, Film C

J. P. Ries And T. J. Farr

Experimental Study of Idler Roll Slippage

6/9/99 Session 4 2:15 - 2:40 p.m.

Question - Pete Werner, Rockwell Automation

I have a suggestion for your classical torque or braking curve, since the actual data does not have the same kind of break point. If your entrance tension to that roll is falling off and exit tension is remaining constant as you apply the break, so therefore you actual average tension across that roll is decreasing. Therefore you are going to begin to slip earlier. Look at the plots on figure 4, trays 4 and 5 are being your input tension that's beginning to fall off because of your torque versus delta tension conservation equation. This explains why it begins to slip earlier.

Answer – J. P. Ries, Dupont

When you run that test you have 3 choices, one is hold the output constant and let the input drop, the other is to hold the input constant, let the output go up; the other is to run a continuous loop and let them both change, they have got to be different, the belt equation predicts T_{in} over T_{out} is going to be μ θ.

Question - Pete Werner, Rockwell Automation

No, I do agree, I'm not criticizing, just explaining why you would begin to develop this slip earlier than the classical one would suggest. The tension has to fall off on the lead tension if your keeping the exit tension fixed.

Answer - J. P. Ries, Dupont

Don't buy it. Even if it doesn't slip T_{in} - T_{out} times the radius has got to equal the torque so you got to have a ΔT. As you increase the torque you've induce a higher ΔT. When it slips nothing happens, now you can no longer control Tin by more torque, it is controlled by the belt equation rather than by T_{in} - T_{out} times R equals torque.

Comment - Keith Good, Oklahoma State University

You made a comment you had a constant coefficient of friction versus wrap angle on one of your later slides. You said its obvious it should be that way, I want to make the comment that if that roughness was high enough that starts to act like a porosity or permeability and then you can get a depletion of the air layer down and then it can effect traction at some point. Traction should be constant with wrap angle as long as air film thickness is not changing. If you had a permeable web, for instance, where the air film is decaying then you would expect the traction to increase with some wrap angle, but you can get the same effect with a very rough roller, because the air can bleed.

Answer - J. P. Ries, Dupont

Yes, that right. Air bleeds out through rough roll, grooved roll, a knurled roll.

Questions - Ed Givens, Proctor and Gamble

Jim I see your percentage slippage curves going up very steeply. Did you take any of those to 100% slippage? They probably would have hit it very quickly at the end.

Answer - J. P. Ries, Dupont
Yes, we got up to 20 or 30%. What happens there is say your slipping at 20% and you want to go to 30, you have got to raise that torque from 12 inch pounds to 12.02 ever so slightly, and experimentally you can't do it. You have to really keep adding torque very slowly to get it to do that.

Questions:
Your saying it was truly was asymptotic and one little bump in torque caused the roller to come to a screeching halt?

Answer - J. P. Ries, Dupont
Yes I would say that.

Questions - David Pfeiffer, JDP Innovations
Your classical belt theory, would say that if you had a elastic belt, a rubber belt or v-belt drive you'd have a really sloped torque speed drive characteristic so by adding a little torque your speed would slow down. Your saying that can't be it cause we're using polyester with a modulus of 4.1 GPa, so that can't be the reason. However the torque transmission is through surface shear contact with belt material so its being transmitted through asperities can lie down be pulled over and otherwise there can be a distortion right at the contact of the quarter percent of speed difference. Although it isn't the volume material of the belt stretching it could be just that elasticity at the points of contact that cause the speed torque curve to have a slop to it instead of an instant break.

Answer - J. P. Ries
I didn't make reference to it being elastic or inelastic; but your in the realm of micro slip, we're only slipping a foot a minute, running 500 and slipping, and the rolls running 499, so that means the first asperity that it runs that first group of asperity coming in to contact that when it got around the film had gone a little bit further, and that scratches, so its micro slipping. I expect to see some micro slip, but I didn't realize that you can micro-slip that all the way up to 10, 15, 20 %. That's all I can explain why you get that, bury that slippage as a function with breaking torque and get any slip you want.

Questions - Jim Dobbs, 3M
I found it pretty useful to rewrite the delta equation in terms of average tension and belt tension. You might try that and that gets around T_{in} or the T_{out} to get a little formulation and that can be used to renormalize your data in order to keep the average tension constant.

Answer - J. P. Ries, Dupont
T_{in} and T_{out} both float.

Questions - Jim Dobbs, 3M
You write the belt equation in terms of $T_{average}$ and ΔT, you look at it that way and re-normalize the data. Based on Keith's comment as far as when you start getting large ΔT across the roller, the float height depends on T. I don't know where all that air goes if it has to leak out or not but its a pretty good bet that whether it is leaking out, compressing or simple contacting the asperity results in higher μ.

Answer - J. P. Ries, Dupont
Right, its starting to lift up, when does it slip when these two velocities not equal. That is why the coefficient of friction drops of because they are starting to separate.