MEASUREMENTS OF THE AXIAL FORCE REQUIRED TO DRIVE AN OSCILLATING ROLLER UNDER A WIDE RANGE OF CONDITIONS

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Abstract: A program of tests is reported on in which the initial objective was to determine the force required to cause a resilient inking roller, running in contact with a metallic roller, to move or slide in the axial direction. The variables investigated included ink film thickness, circumferential roller speed, hardness of the resilient roller covering, pressure setting or stripe of the roller, and viscosity of the ink. The results obtained are directly applicable to the design of mechanisms for driving an inking roller in an axial oscillating motion. Beyond that, the data sheds considerable light on the phenomena taking place in the nips formed by inking rollers and on the properties of typical printing inks in such nips where the rates of shear are in excess of 10,000 reciprocal seconds. Byproducts of this work include a measure of the relationship between radial force and roller stripe for three different roller hardnesses, a comparison of Laray plastic viscosities viz-a-viz actual apparent viscosities in an inking roller nip, and a comparison of actual ink film thicknesses on an inking roller versus those measured with a wet film thickness gauge.

INTRODUCTION

The tests described in this paper were part of a program aimed at developing a self-driven oscillating inking roller, based on a new and novel drive mechanism. The tests were undertaken because no data on the force necessary to move a rotating inked roller in the axial direction could be found in the literature. Force data under a wide variety of conditions of roller settings, hardness, ink film thickness, and speed were obtained in a straightforward manner, on a laboratory inker, salvaged from a 49 inch sheetfed press.

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By themselves, the force measurements would not be of interest to anyone except for that small group of engineers involved in the design of mechanisms for generating oscillatory roller motion.

However in analyzing the data, it was discovered that this type of measurement can provide considerable insight into the conditions existing in the nips or conjunctions formed by rotating inked rollers. Thus, the main object of this paper is to present what is thought to be new information on the conditions existing in ink roller nips.

The main body of this paper consists of four sections. The first contains brief descriptions of the materials, equipment, and procedures used to obtain the data. The second major section presents the experimental data, while the third is devoted to analyses and discussions of the results. The last section summarizes what the authors consider to be the main conclusions of the work.

EXPERIMENTAL EQUIPMENT AND PROCEDURES

Test Rollers

Data was collected using three 2-7/8 inch diameter by 48-7/8 inch long rollers which differed primarily in the hardness of the rubber coverings, as shown in Table I.

Table I Properties of Test Roller Coverings

Number	Hardness (Type A Shore Durometer)	Type of Material	Thickness (inches)
1	25-27	Nitrile-	0.312
2	35	Family of Synthetic Rubbers	0.25
3	30	Unknown	0.25

The hardness of Roller #1 is listed as a range because its durometer changed from 25 at the onset to 27 at the end of the tests. Also, the covering of Roller #3 was cut down to 38-3/4 inches near the end of the program to confirm that the drag of the roller bearings did not introduce an error into the measurements of driving force.

Rig for Measuring Stripe vs. Normal Load

In analyzing the driving force measurements, it became apparent that it would be helpful to know the effect of roller hardness and setting (or stripe) on the radial or normal roller load. Accordingly, the rig shown in Figure 1 was assembled to carry out such measurements.

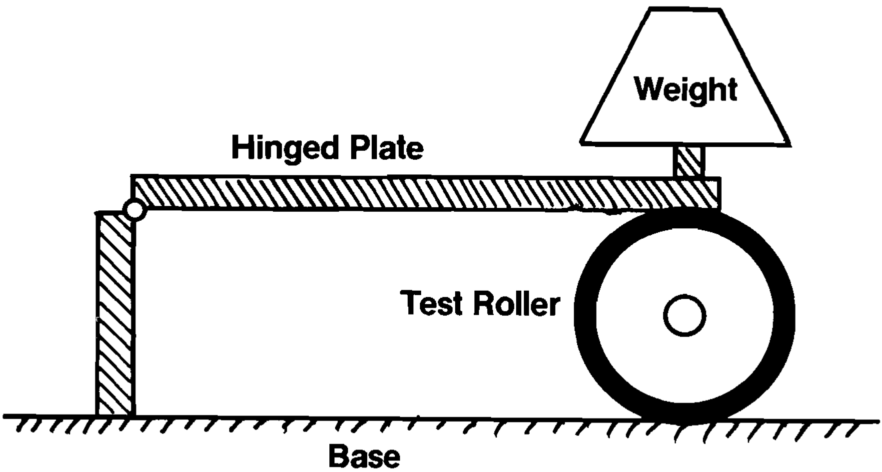


Figure 1 Test Rig for Determining Roller Stripe vs. Normal Load

The procedure followed was to first ink the test roller with a very thin film, about 0.1 mils (0.0001 inches) thick. The weighted six inch wide hinged plate was gently lowered onto the inked roller and allowed to rest for 5-10 seconds. The plate was then swung away and measurements were made of the resultant stripe on the roller, using a caliper with a least count of 1 mil. The stripe was measured at each end and both readings were recorded along with the force exerted by the hinged plate and weight. Although this method is very simple, it is also analogous to the method used to gage stripes during the driving force tests.

Rig for Measuring Driving Force

A complete inking system, salvaged from a 49 inch wide sheetfed press, was used to support and drive the roller under test. A diagram of this inker is shown in an earlier paper (MacPhee, 1976). The total surface area of this roller train, including the test roller, was 7,604 square inches.

The roller to be tested was installed in a position where it came in contact with a single oscillating steel roller, 10.5 inches in diameter. Sufficient end play was introduced into the test roller sleeve type bearing so that the test roller would oscillate in the axial direction, as a result of the friction between it and the steel roller. The set-up used to measure driving force is shown in Figure 2. The procedure followed was to first ink up the rollers to the desired ink film thickness. With the roller train running at the desired surface speed, air pressure to the cylinder was increased until the oscillatory motion of the test roller was just silenced (as determined by motion of the dial indicator). The force exerted by the cylinder was calculated and recorded as the driving force corresponding to the test conditions. This set-up and procedure were selected on the theory that the roller bearings would not introduce error into the measurements because the journals of the silenced roller would not be moving in the axial direction.

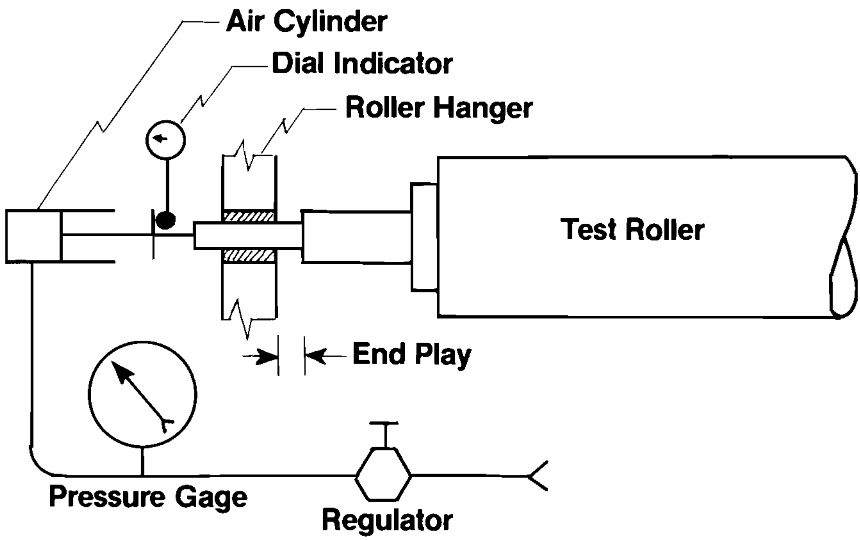


Figure 2 Test Rig for Measuring Axial Driving Force. Pressure gage had a least count of 0.5 pounds per square inch and a maximum reading of 100.

Ink Film Thickness Measurements

A Model S Gardner Wet Film Thickness Gage with a range of 0-4 mils was used to measure ink film thickness on the rollers. However, these measurements were for reference only and were not used to establish the ink film thicknesses recorded during the force measurements. The reason for this is that it had been discovered earlier (MacPhee, 1976) that the Gardner Gage has a systematic error (which was explored further and is discussed below). Accordingly, the desired ink film thicknesses were achieved by weighing out and applying to the roller train the calculated amount of ink needed. For example, to produce an ink film thickness of 0.2 mils*, 25 grams of ink were weighed out on a laboratory balance and then applied to the roller train.

* One mil equals 0.001 inches

Test Fluids

In all, four different inks and two standard oils were used in the course of the program. The inks selected represented four different types and covered a viscosity range of 35 to 450 poise. The two standard oils were included so that data on Newtonian fluids could be collected. Viscosity data were obtained, using a Laray viscometer (Evans, 1989) and these are recorded in Table II along with the other information known about the test fluids.

Table II Fluids Used in Experiments

I.D. Code	Type	Laray Data(1) Yield (dynes/cm^2)	Viscosity (poise)	Tack (4)
AY	News Ink	300	35	9.4
IY	Heatset Ink	800	250	14.8
VD	Sheetfed Ink	1,950	270	Not Measured
AB	Dupli. Ink	3,100	450	"
S2000	Standard Oil	<200	66(2)	"
N4000	Standard Oil	390	127(3)	"

1. Test at 77°F (25°C). Viscosity at 2500 $sec.^{-1}$
2. Viscosity certified by supplier is 55 poise.
3. Viscosity certified by supplier is 106 poise.
4. 1200 rpm, 90°F, 1 minute

TEST RESULTS

All of the data discussed in the subsequent section is included in Appendix A in the form of tables. Table A-1 gives typical measurements of ink film thickness, Table A-2 contains the data obtained on stripe vs. normal load, Tables A-3 thru 6 give the data on driving force at a constant speed (335 feet per minute plus minus 10), and Table A-7 gives the data on driving force vs. speed. The test conditions under which the driving force measurements were made included two different stripes (3/16 and 5/16 inches), the three roller hardnesses listed in Table I, the six fluids given in Table II, and a surface speed range of 96-536 feet per minute.

ANALYSIS AND DISCUSSION

Ink Film Thickness

In an earlier program, it had been discovered that measurements of the single ink film thickness employed (0.2 mils) was approximately twice the theoretical or calculated value (MacPhee, 1976). This phenomenon was explored further in this program and the results are shown in Figure 3. Here it can be seen that the measurements have a built-in offset of about 0.2 mils In other words, the measurements have a constant error on the high side of 0.2 mils

These results were discussed with someone having much greater experience in this area (Voas, 1989). He suggested that the error was due to a failure to follow the manufacturer's recommended procedure as follows:

(i) Hold the gage with its axis at 90 degrees with respect to the direction of travel of the moving surface. Tilt the gage so that only one of the rims makes contact with the moving surface. When the gage rim has attained the same velocity as the moving surface, level the gage so that both rims make uniform contact and let it continue to turn for one complete revolution.

(ii) Lift the gage quickly from the moving surface and read the thickness of the pickup points on the center wheel directly across from the engraved scale, or on rotating scale gages use the indicating pointer to eliminate parallax in reading the rotating scale. If the two readings are not the same, the lowest pickup point represents the correct thickness and the higher pickup point should be

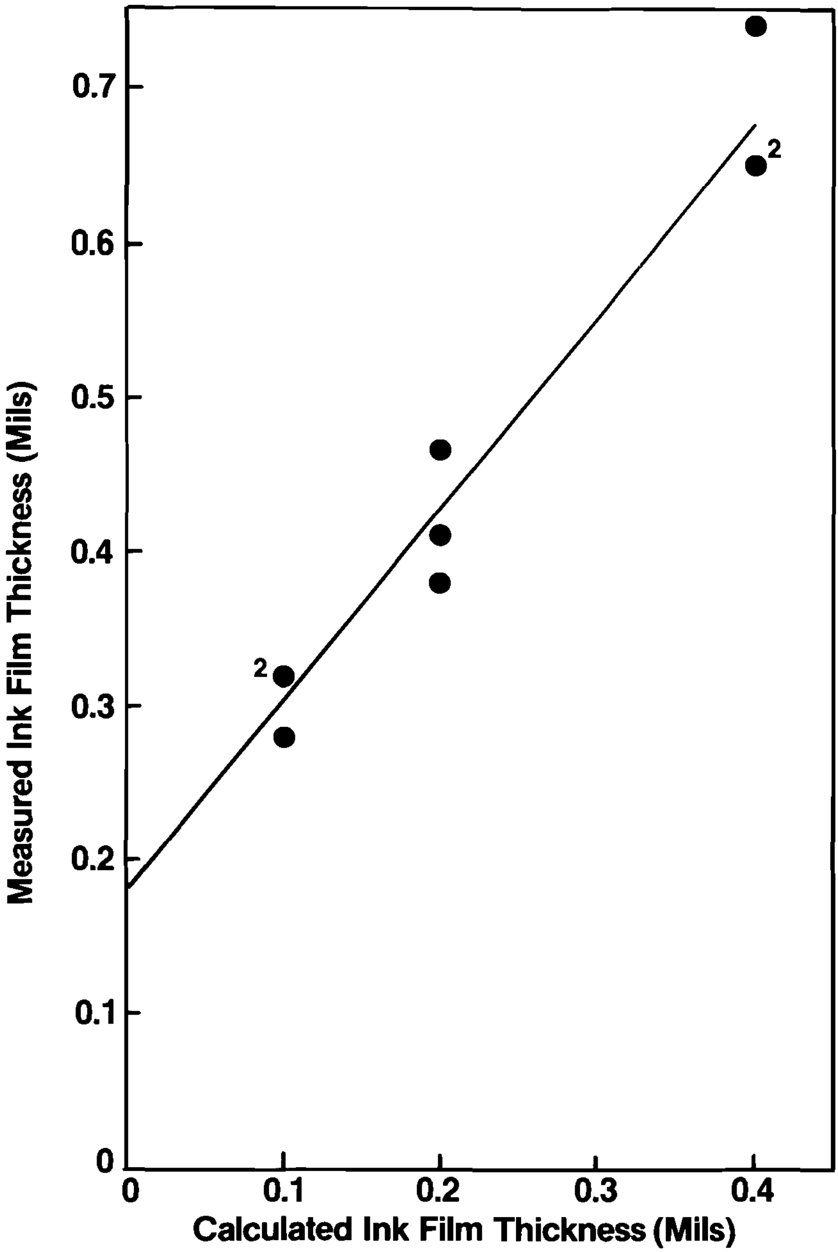


Figure 3 Measured Ink Film Thickness Versus Calculated Values. Numerals indicate number of coincident points.

discarded. Where a pickup point is speckled or otherwise poorly defined, select an arbitrary point 1/3 of the distance from "solid pickup" to "no pickup"

Further investigation revealed that this indeed was the case in that the technician taking the readings was, contrary to (i), allowing the gage to rotate many revolutions before lifting it off the roller surface. It was also discovered that the Model S Gage being used was a general purpose one and that a low inertia Model L was recommended for use on moving surfaces. Use of the recommended procedure and Model L gage did eliminate the systematic error shown in Figure 3. However the inexperienced authors found it very difficult to obtain consistent readings. For that reason, the procedure and gage model used initially is preferred, since the systematic error in readings which result, is much more tolerable than the inconsistencies of the alternative.

Stripe vs. Normal Load

The measurements of stripe vs. normal load are plotted in Figure 4, along with the equations of the best straight line fits obtained from log-log plots. Related data, given in Table III show that the stripe varies very closely with the one-third power of load. Initially this was surprising because the corresponding Hertz equations predict a one-half power relationship (Roark and Young, 1975). However, Deshpande pointed out that Hertz's equations do not apply to a layered structure (such as a rubber covered steel roller) and went on to derive equations that do apply and which were again confirmed by the data in Table III (Deshpande, 1978). Deshpande's results predict a one-third power relationship which can be put in the following form:

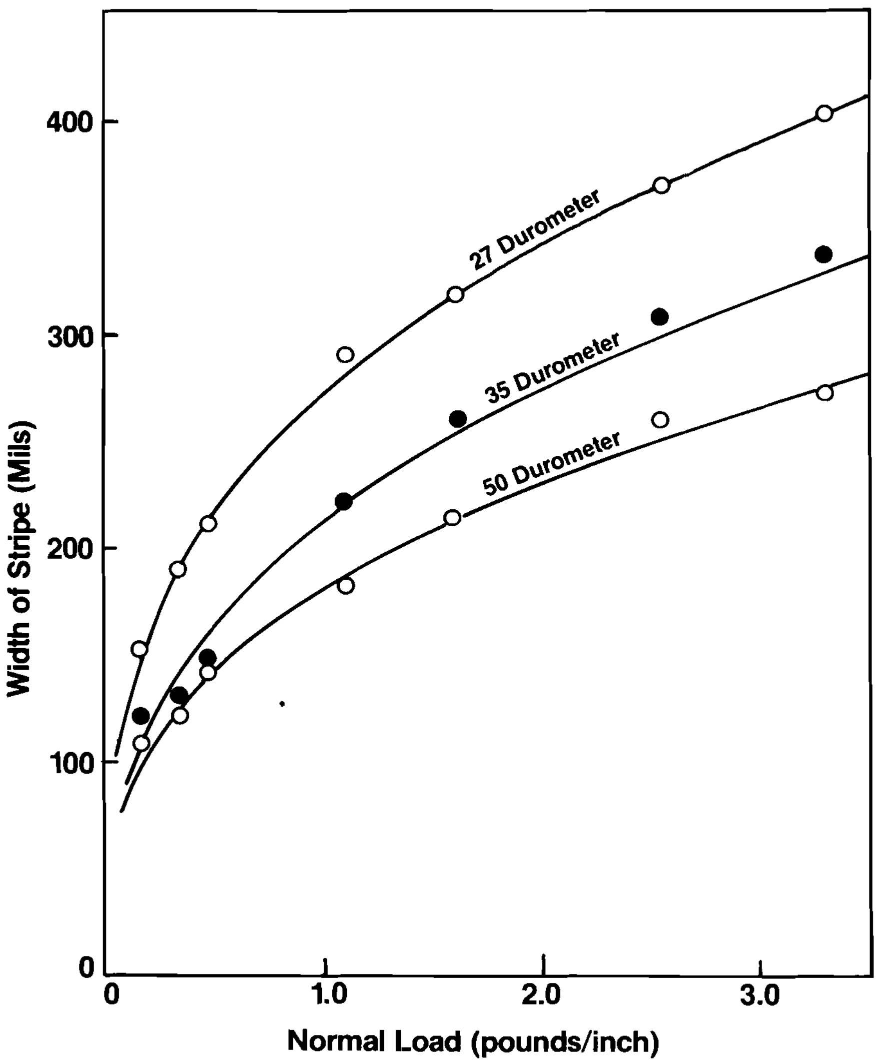


Figure 4 Roller Stripe Width Against a Flat Plate Versus Normal Lead. Curves are plots of best fit equations given in Table III.

$$S = \left(\frac{3.48h\ R\ F}{E}\right)^{1/3} \qquad (1)$$

Where:

h = thickness of cover

E = modulus of elasticity of cover

F = normal force

S = width of stripe

$$R = \frac{R_1\ R_2}{R_1 + R_2} \qquad (2)$$

R_1 = radius of roller 1

R_2 = radius of roller 2

Equations (1) and (2), along with the best fit equations were used to calculate the moduli of elasticity given in Table III. No measure of modulus, using a similar procedure could be found in the literature. As a result there was nothing with which these derived values of modulus could be compared. However the data indicate that a change in hardness from 27 to 50 is equivalent to increasing the elastic modulus by a factor of 2.65, and this compares very well with a ratio of 2.7 for a hardness change from 25 to 50 obtained from a published curve of sheer modulus versus durometer (Eirich, 1978).

The other data in Table III is interesting in that it indicates the following:

(i) Increasing roller stripe from 3/16 to 5/16 inch increases the normal load by a factor of almost five.

(ii) The normal load for what the industry accepts as ideal form roller conditions (25 durometer cover, 3/16 inch stripe on a 3 inch diameter roller) is predicted to be 0.3 pounds per inch of roller length. This is a factor of seventy five lower than the typical normal load in the plate blanket cylinder nip. The latter is based on Tyma's data (Tyma, 1982) and an assumed interference of 0.003 inches.

Table III Calculated Data Obtained From Best Fit Curves of Stripe vs. Normal Load

	27 Durometer	35 Durometer	50 Durometer
Best Fit Curve			
Correlation Coefficient	0.999	0.982	0.994
Equation*	$S=273F^{0.327}$	$S=212F^{0.368}$	$S=182F^{0.346}$
Modulus of Elasticity	77 psi	108 psi	204 psi
Normal Load			
3/16 inch stripe	0.317 lbs/ inch	0.716 lbs/ inch	1.09 lbs/ inch
5/16 inch stripe	1.51 lbs/ inch	2.87 lbs/ inch	4.78 lbs/ inch
Average Pressure			
3/16 inch stripe	1.69 psi	3.82 psi	5.82 psi
5/16 inch stripe	4.83 psi	9.18 psi	15.3 psi

*Where S = stripe width in mils and F = normal force in pounds per inch of roller length.

(iii) Increasing durometer from 27 to 35 increases load by about a factor of two while an increase from 27 to 50 increases load by about a factor of three.

(iv) The pressure to which printing inks are subjected in roller-to-roller nips is very low - just a few pounds per square inch under recommended conditions. Thus, there is no reason to think that ink viscosity will be affected (i.e. increased) by pressure in the nip.

Driving Force

The data on driving force versus ink film thickness, at constant speed for three different inks and the two standard oils, is plotted in Figures 5-9. Once these plots were made, two questions arose as follows:

(i) Why do some of the curves exhibit a maximum, while others simply decay, with increasing ink film thickness?

(ii) What can be deduced from these curves regarding the viscosity exhibited by the test fluids in the nips?

In an endeavor to answer these questions, the first step is to understand that the force necessary to produce relative axial movement of the rubber test roller, with respect to the steel roller, is a measure of the strength, in shear, of the area of contact between the two rollers. With no ink on the rollers, the shear strength is determined by the area of contact and the strength of the adhesive bond between the rubber and metal. However, if the two roller surfaces are completely separated by a film of ink, then the shear strength is determined by the viscosity of the ink, the area of contact, speed, and the ink film thickness.

A Rubber Friction Consider first the case where no ink is present, such that friction forces dominate.

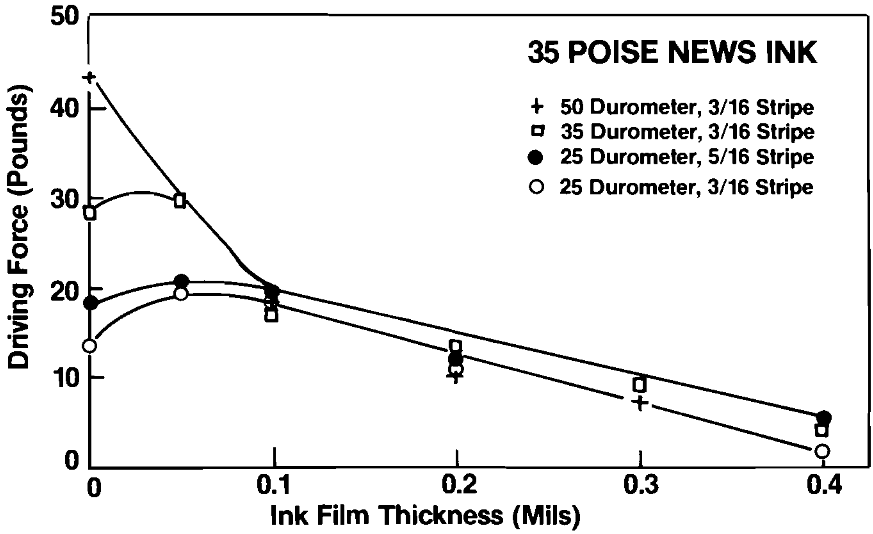


Figure 5 Driving Force Versus Ink Film Thickness at Constant Speed of 335 Feet/Minute - 35 Poise Newsink

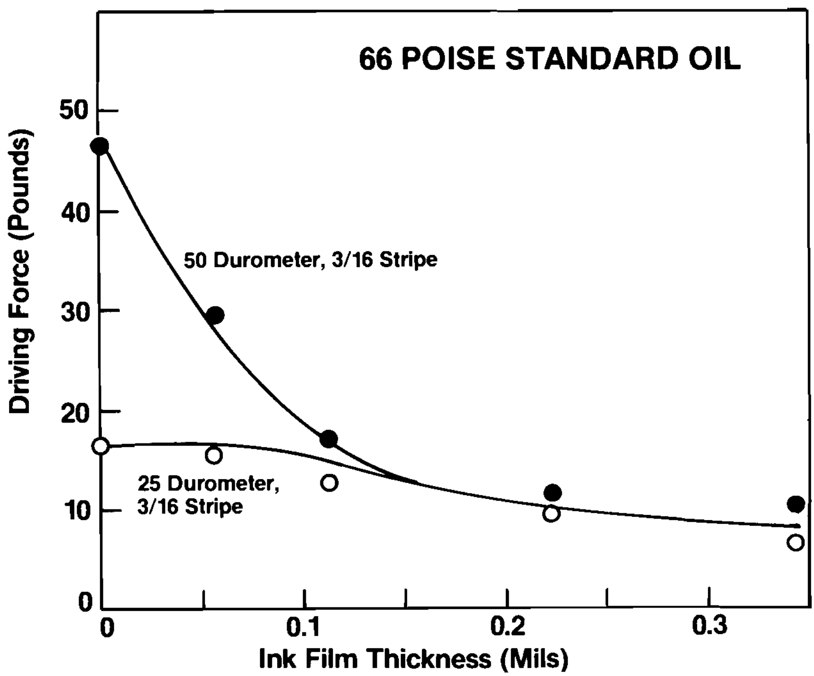


Figure 6 Driving Force Versus Ink Film Thickness at Constant Speed of 335 Feet/Minute - 66 Poise Standard Oil

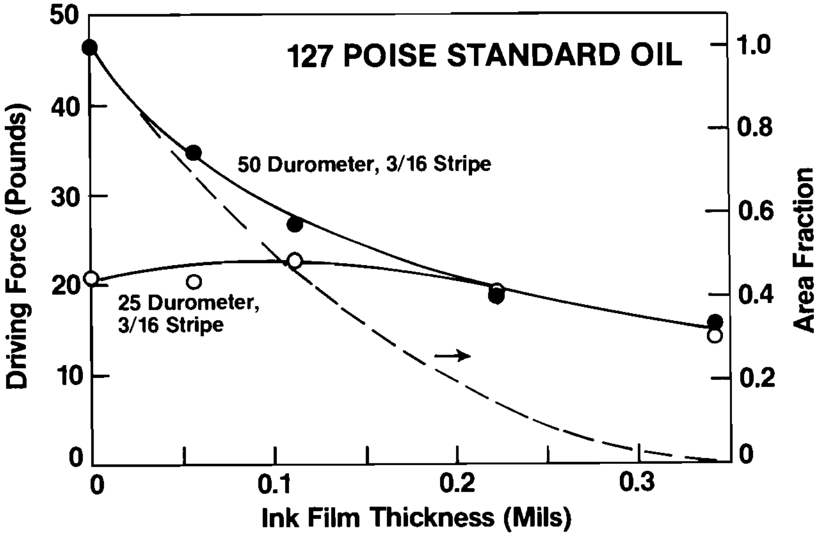


Figure 7 Driving Force Versus Ink Film Thickness at Constant Speed of 335 Feet/Minute - 127 Poise Standard Oil

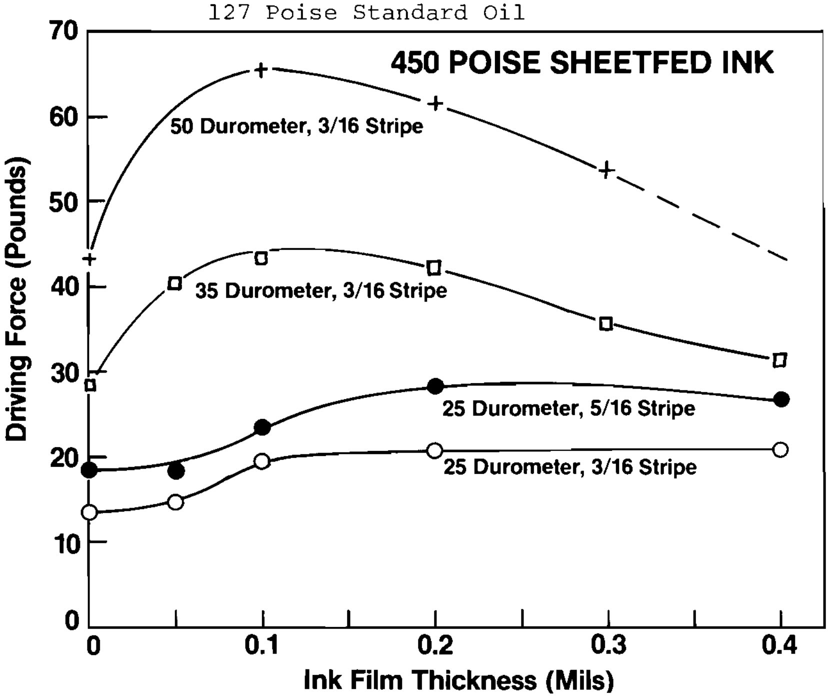


Figure 9 Driving Force Versus Ink Film Thickness at Constant Speed of 335 Feet/Minute 450 Poise Duplicator Ink

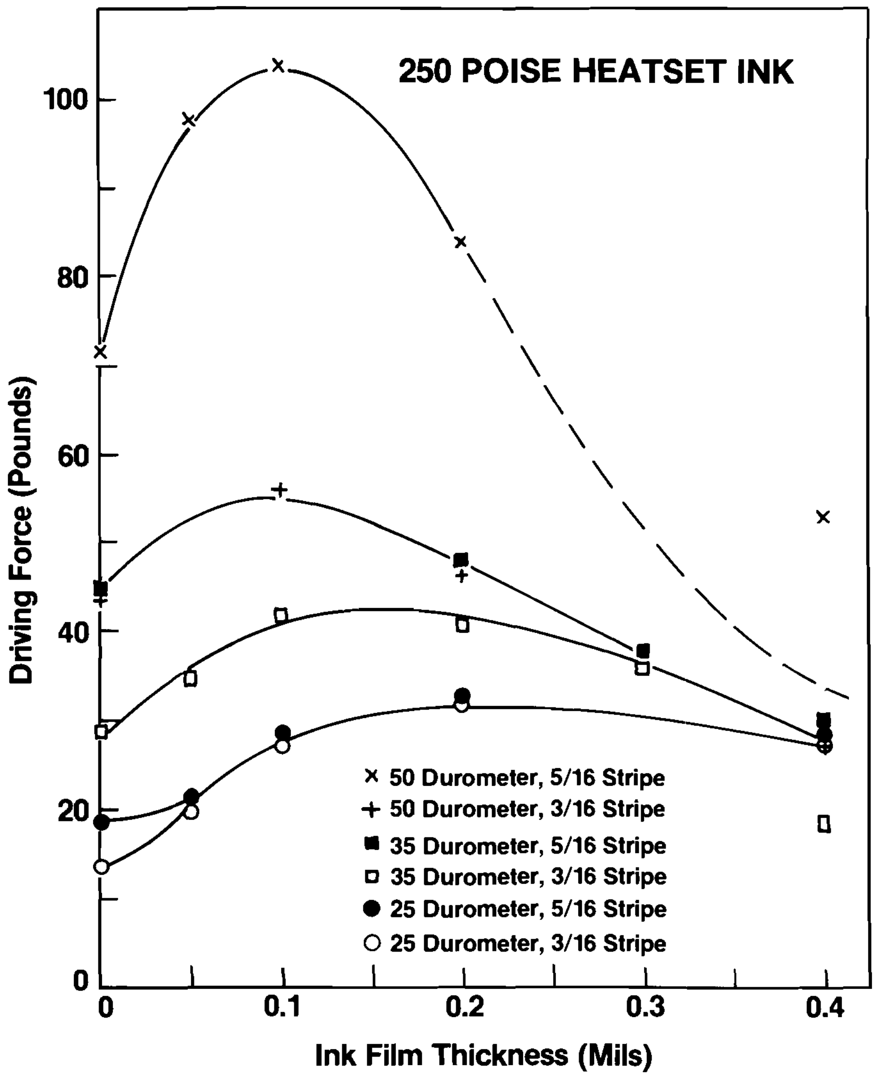


Figure 8 Driving Force Versus Ink Film Thickness at Constant Speed of 335 Feet/Minute - 250 Poise Heatset Ink

Studies of rubber friction (Kummer and Meyer 1960) show that for rubber pressed against a very smooth, hard surface, the friction force will be independent of the normal force if the contact pressure is high enough. Thus, the coefficient of friction, i.e., the ratio of friction force to normal force, decreases with increasing pressure. The explanation for this is that in such a case the actual contact area is equal to one hundred percent of the apparent contact area and thus not a function of the normal force. In contrast, at low pressures, actual contact area is much less and increases with the normal load. Therefore, at low pressures, shear strength increases with normal load, giving rise to a constant coefficient of friction. Because of this phenomenon, the so-called classic laws of friction do not always apply to rubber.

The pressures existing in the test nips (as given in Table III) are high enough for the above theory to predict that the dry friction forces will vary only with contact area (stripe width) and material composition. Examination of the data shows this to be true. For example, increasing the stripe form 3/16 to 5/16 inches should increase driving force by a factor of 1.6. Examination of Figure 8 shows the corresponding ratios to be 1.37, 1.56 and 1.64 for the 25, 35, and 50 durometer rollers respectively. (Table III shows that such a stripe change results in a five-fold increase in normal force; further proof that the classic laws of friction do not apply here.)

In contrast to the effect of stripe width, roller hardness strongly affected driving force as shown in Figures 5-9. For example, for a constant stripe width, driving force almost doubled from 25 to 35 durometer, and increased by 1.5 from 35 to 50 durometer. Because these ratios roughly correspond to the variations in normal force given in Table III, one might argue that here is contradictory evidence that indicates that the classic laws of friction do indeed apply here. However, it seems far more likely that this latter phenomenon can be explained by differences in material composition. Specifically, it is suggested that the decreasing amounts of liquid plasticizer present in the harder rubber compounds account for their higher coefficients of friction.

B Shape of Curves. The conclusion that the rubber roller is in full contact with the mating steel roller under dry conditions is one of the keys needed to explain the shape of the curves in Figures 5-9.

As the second step in the process of explaining the shape of the curves, the total drag force, which is equal to the driving force, will be defined as follows:

$$F_t = F_f + F_v \qquad (3)$$

Where:

F_t = total drag force

F_f = friction component

F_v = viscous component

It was pointed out above that under dry conditions, the friction component governs, while at thick ink films the viscous component is the determining one. This raises the question as to how F_f decreases and F_v increases as ink film thickness increases. Given that the two surfaces are in full contact under dry conditions, it is reasonable to assume that the following relation holds:

$$F_f = (C_1)\ (a) \qquad (4)$$

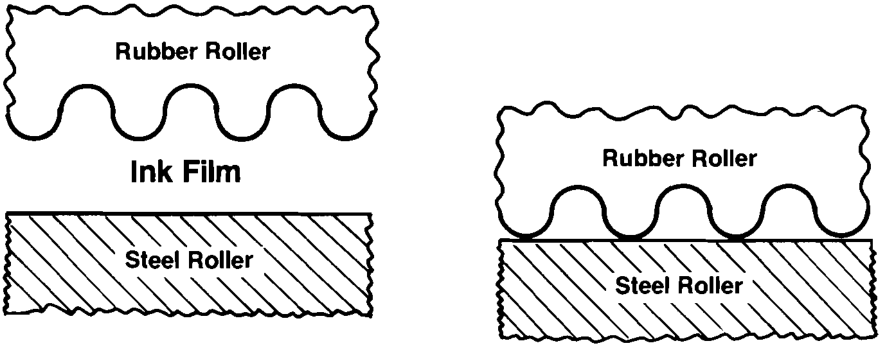
Where:

C_1 = a constant

a = area fraction

The area fraction is defined as the ratio of the actual area of contact to the total area of nip. Furthermore, "a" equals one at zero film thickness and decreases as film thickness increases.

As for the second component of drag, F_v, it would be expected to be inversely proportional to film thickness and to increase as the area of contact decreases, since it acts over an area proportional to (1-a). The model illustrated in Figure 10 can be used to explain a further dependence of F_v on (1-a).



(a) Thick Film Case **(b) Thin Film Case**

Figure 10 Model of Interface between Surfaces of Rubber and Steel Rollers. In thick film case (a) height of rubber surface asperities is small compared to film thickness. In thin film case (b) film is pierced by asperities.

That is, for very thin ink films, as shown in Figure 10(b), the asperties of the rougher rubber surface pierce the film and thus reduce the area of the steel covered by the film. As a result, a second effect of increasing the film thickness is present. That is, because there is a smaller surface area over where the ink is spread, its film thickness, where it is spread, is increased by the factor 1/(1-a). The complete expression for F_v can thus be written:

$$F_v = \frac{C_2 (1-a)^2}{T} \qquad (5)$$

Where:
C_2 = a constant

T = ink film thickness

Combining equations (3), (4), and (5) yields:

$$F_t = (C_1)(a) + \frac{C_2)(1-a)^2}{T} \qquad (6)$$

One problem remains, and that is to determine how the area fraction, "a", varies with ink film thickness. The curves of measured driving force versus film thickness provide a way of estimating this. In particular, consider the curves for the 25 and 50 durometer rollers and standard oil N-4000, as given in Figure 7. Under dry conditions, the two curves differ by a factor of over two, reflecting the higher dry friction force of the harder roller . However the two curves coincide at film thicknesses greater than about 0.25 mils, indicating that the friction component has decayed to a negligible amount at that thickness. Thus, it is reasonable to assume that the area fraction decreases in a manner shown by the dotted curve in Figure 7.

With the development of Equation (6) and the relationship of the area fraction just given, a hypothetical curve of driving force versus film thickness can be calculated. To do this, assume that the driving force is equal to 40 pounds at zero thickness, and again at a thickness of 0.35 mils (i.e $C_1 = 40$ and $C_2 = 14$). If Equation (6) is then used, the calculated curves shown in Figure 11 are obtained.

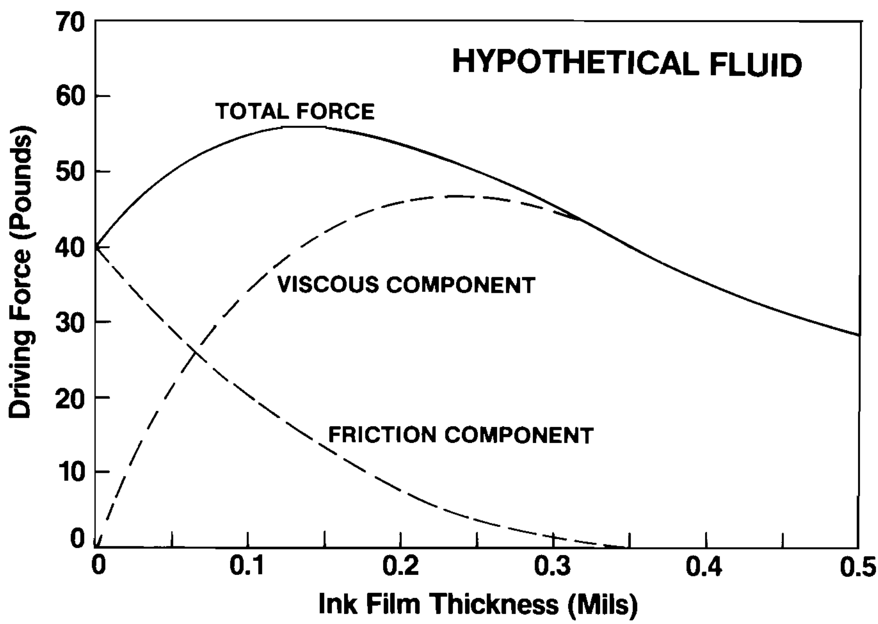


Figure 11 Calculated Curve of Driving Force Versus Ink Film Thickness for Hypothetical Fluid

The curve for total force exhibits a maximum, much like the measured curves in Figures 8 and 9. If, on the other hand, a low value for the viscous component (i.e., a low value of C_2) had been assumed, curves in the form of those in Figures 5-7 would have been obtained.

C Equivalent Viscosity Aside from explaining why the curves in Figures 5-9 are so shaped, the above analysis indicates that data obtained at film thickness less than 0.3 mils should not be utilized to infer equivalent values of viscosity - since the friction components becomes significant at the thinner film thicknesses. However, because the viscous component is fully developed beyond about 0.3 mils, it should be possible to use the definition of dynamic viscosity to calculated equivalent viscosities in the nip. This definition is as follows:

$$\text{Viscosity} = \mu = \frac{\text{Shear Stress}}{\text{Shear Rate}}$$

$$\mu = \frac{F/(LS)}{V/T} = \frac{FT}{VLS} \qquad (7)$$

Where:

F = Measured driving force
T = Film thickness in the nip
V = Roller axial surface speed
L = Length of roller (and stripe)
S = Width of stripe

The only variable in question on the right side of Equation (7) is T, the ink film thickness in the nip. If there were a space between the roller pair just equal to twice the film thickness on the rollers, then there would be no pressure buildup at the nip entrance and the film thickness in the nip would simply be double the film thickness on the rollers. In the actual case, the roller surfaces are initially in contact with each other and thus must be forced apart by the two converging films, which will result in a pressure buildup and some thinning. However this pressure buildup will be very small for the following two reasons:

(i) The data in Table III show that the pressures in the nip are just a few pounds per square inch when the two surfaces are in contact with each other.

(ii) The film thicknesses are not more than one tenth the amount that the rubber cover is compressed (i.e. the compression is about 8 mils for a 3/16 inch stripe and 1/4 inch thick cover). Thus the added compression of the rubber due to the presence of the two converging films can only increase the nip pressure by a slight amount, in which case there can only be a slight amount of thinning.

In view of this, the film thickness in the nip was assumed to be twice the value of the nominal film thickness on the rollers.

In accordance with this assumption and Equation (7), equivalent viscosities were calculated for all of the force data obtained, based on a maximum axial roller velocity of 1.44 inches/sec (7.2 feet/minute). (The corresponding shear rates are 2400 and 1800 reciprocal seconds for 0.3 and 0.4 mils respectively.) These results are included in Tables A-3 thru A-6. The data obtained for film thicknesses equal to or greater than 0.3 mils is also plotted in Figure 12 versus the Laray

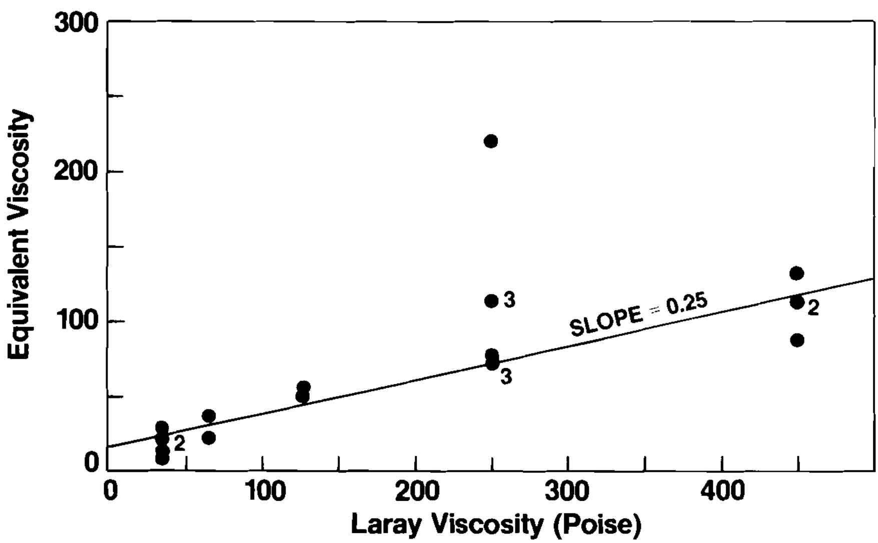


Figure 12 Graph Showing Relationship Between Equivalent Viscosity and Laray Viscosity Data for Constant Speed of 335 feet per minute. Numerals indicate the number of coincident points. Correlation Coefficient of best fit line is 0.89.

viscosities. In calculating the best straight line fit of the data, one measurement of the 250 poise heatset ink (50 durometer, 5/16 inch stripe, 0.4 mil thick film) was ignored. The reasoning was that this ink appeared to have "tacked up," as a result of solvent loss, and thus produced an abnormally high driving force when the last reading was taken, as shown in Figure 8. (If this point is not ignored the only significant effect is to reduce the correlation coefficient to 0.75.)

Examination of the data in Figure 12 reveals that the equivalent viscosities of the test fluids were consistently low, compared to the Laray viscosities, by a factor of two to four, with a greater reduction occurring with the more viscous fluids. It is also interesting to note that the Newtonian standard oils behaved no differently than the non-Newtonian inks. There are only two possible explanations for the reduction in fluid viscosity while in the nip: either the actual film thickness is much greater than assumed, or else there is significant heat being generated within the nip. Since film thickening by a factor of two to four is not plausible, it is far more likely that the viscosity reduction is due to a temperature rise in the nip. This is quite plausible in that it is estimated that a rise to 110 °F is all that would be needed to produce the observed results.

D Speed Effect. Plots of driving force versus roller surface speed are given in Figure 13 for two different inks at two different film thicknesses, and for dry conditions.

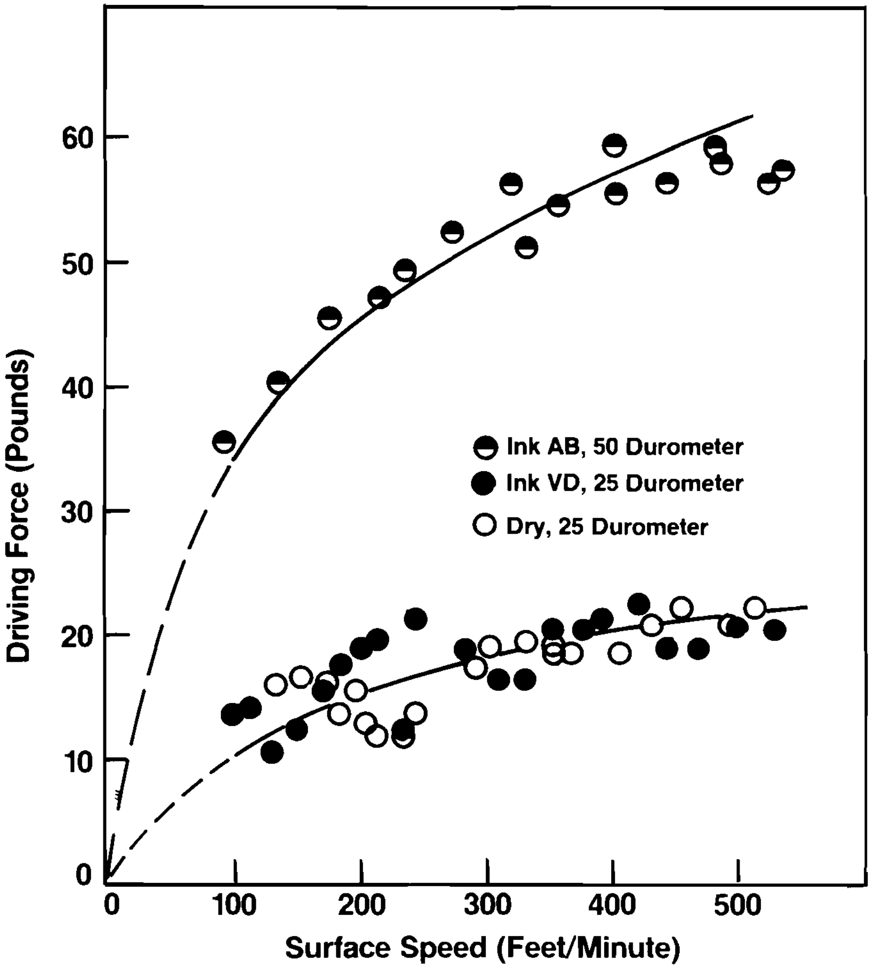


Figure 13 Driving Force versus Press Speed. Rollers were set to 3/16 inch stripe. Ink AB film thickness was 0.2 mils and that of Ink VD was 0.4 mils.

In theory the curve for the thicker ink film (0.4 mils), where friction effects are negligible, should be a straight line. This is because the only variable affecting driving force, as speed is increased, is shear rate. The fact that this curve bends over is a second piece of evidence which suggests that temperature rise in the nip has a great effect on the viscosity exhibited by the ink. That is, as speed is increased, it is quite reasonable to assume that temperature rise in the nip also increases, thus causing the driving force to increase at a lower rate.

CONCLUSIONS

Beyond satisfying the engineering need for data on driving force, these measurements led to the following conclusions which have much pertinence to a better understanding of the lithographic process:

1. Correlations of the measurements of roller stripe versus normal load conform to the appropriate analytical relationship and thereby provide a means for calculating nip pressure and compression for rollers of known diameter, cover hardness, cover thickness, and setting or stripe.

2. The above measurements point up the importance of proper roller care and maintenance in that very high nip pressures will result if rollers are not properly set or not replaced when cover hardness increases. For example, an increase in stripe from 3/16 to 5/16 inches increases pressure by about a factor of three, while an increase in hardness from 25 to 35 durometer doubles the pressure.

3. The above measurements also revealed that the pressures existing in inking roller nips are much lower (i.e. by a factor of about 75) than the corresponding pressure in the plate-blanket cylinder nip.

4. The test results confirm that slippage occurs between the two roller surfaces which come in contact in a nip. (Although not discussed earlier, this conclusion is based on the observation that, with zero ink films, the at rest driving forces were much higher than when the rollers were turning.) For a three inch diameter roller pair with a 3/8 inch thick cover on the rubber roller, this slippage is estimated to be 1.6 percent of circumferential surface speed for a 3/16 inch stripe and 4.3 percent for a 5/16 inch stripe. At a surface speed of 1500 feet per minute and an ink film thickness on the rollers of 0.2 mils, these slippages will produce shear rates of 12,000 and 32,000 reciprocal seconds respectively. No basis could be found for estimates by others (Bisset et al, 1979) that shear rates may be as high as one million reciprocal seconds.

5. The force necessary to drive a roller in the axial direction is equal to the drag force generated in the nip(s). The drag force in turn is made up of two components: the force due to friction between the two mating surfaces, and the force due to the viscous drag produced by the fluid in the nip. The friction force dominates at small ink film thicknesses and, under dry conditions, is a function only of nip area, surface roughness of the rubber, and material composition. At large ink film thicknesses, the viscous force dominates and is a function of fluid viscosity, surface speed, nip area and ink film thickness.

6. Fluids carried into roller nips undergo a tremendous reduction in viscosity - by a factor of 2-4 under the conditions of the measurements made here. The most likely explanation for this is that there is a high rate of local heat generation in the nip. The viscosity reduction observed in these measurements is consistent with a transient temperature rise in the nip from 76 to 110 degrees Fahrenheit.

This phenomenon may well explain why there have been far fewer picking problems encounted, than expected, as web press speeds have increased over the past ten years.

7. No difference was observed between the behavior of the two Newtonian test oils and the non-Newtonian inks which were tested. This would indicate that fluid viscosity in the nip is not significantly affected by shear rate. (One exception to the first statement is that the test oils had a big tendency to mist.)

8. A most important uncertainty about the conditions existing in the nip formed by an inking roller having a resilient covering is the thickness of the ink film. A series of reliable calculations to resolve this uncertainty would constitute a significant addition to the literature on lithography.

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NOTICE

Because of space limitations, Appendix A, consisting of Tables A-1 through A-7 was omitted from the Proceedings. Readers may obtain copies by writing to the senior author (MacPhee) at the address given in the Directory of Members.