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Application of the Bending-Under-Tension Friction Test to Coated Sheet Steels

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Application of the bending-under-tension-test to study the friction behavior of six zinc-based coated sheet steels is considered. The details of a laboratory test system are presented along with an analysis of the methods used to reduce the data. Several theoretical treatments exist in the literature; however, those that incorporate sheet bending have been shown to yield more accurate results. For coated sheet steels, the assumption that the friction coefficient is constant with contact pressure is considered in an analysis of the dependence of friction coefficient on contact pressure.

1. Introduction

EVALUATION of the potential performance characteristics of sheet metals in forming operations is critical to the proper specification of materials and forming parameters. The response of sheet metal in forming operations is controlled by both the bulk deformation behavior of the material, as might be evaluated in a standard tensile test, and the interfacial flow behavior, as controlled by the frictional conditions at the die/metal interface. To evaluate the combined effects of both bulk and interfacial deformation, several simulative forming tests, including the limiting dome height test,[1] cup tests,[2] and other punch-stretch tests,[3,4] have received considerable use, and the results for sheet steels have been shown to correlate with performance in a press shop.[5] These simulative tests evaluate forming with complex imposed stress and strain states, as controlled by surface interactions between the tool and the sheet metal.[6]

With the recent developments in coatings for sheet steels, *e.g.*, zinc-based coatings on automotive sheet steel in which a variety of coatings can be produced on a given substrate, there is significant interest in directly measuring the frictional behavior between the sheet metal and the die materials. Several laboratory tests have been developed to measure friction. In most tests, Coulombic friction is assumed, *i.e.*, the friction coefficient, μ, is constant and described by:

$$\mu = \frac{F_s}{F_n} = \frac{\tau}{p} \qquad [1]$$

where between the sheet and the die, F_s is the shear force, F_n is the normal force, τ is the interfacial shear stress, and p is the average contact pressure. The various laboratory friction tests differ in geometry, method of load application, degree of substrate strain (elastic or plastic), testing speed, and lubrication method.

Friction is a system parameter,[7] not a material property, and as a result depends directly on the measurement procedure. Consequently, it is not appropriate to compare results from different friction tests.[8] In general, the more closely a test approximates the actual forming conditions of interest, the more representative the results of the friction test.[9,10] For example, if one is interested in drawbeads as used in sheet stamping operations, then the drawbead simulator (DBS) friction test may be most accurate.[11-13]

In this article, laboratory-scale friction analysis techniques that involve sheet sliding over cylindrical dies are considered. Although there are several tests that incorporate cylindrical dies, three test methods of specific interest include the drawbead simulator (originally developed by Nine),[11] the tensile strip test,[9,14] and the bending-under-tension test.[15] The bending-under-tension test method has recently received considerable interest for the evaluation of the frictional characteristics of coated sheet steels.[16-19] The bending-under-tension test is the primary focus of this article. An experimental test system is described along with a critical consideration of data analysis methods. Data were obtained on a series of commercially produced coated sheet steels for automotive applications. In the following section, the details of the test methods that incorporate cylindrical dies are reviewed with an emphasis on the assumptions and forms of the equations that have been used to obtain friction coefficients. An analysis that describes data reduction methods along with a consideration of the force due to bending is also presented.

2. Friction Test Methods with Cylindrical Dies

Friction test methods that use cylindrical dies (*i.e.*, rollers) incorporate either a single die (and thus a single bend) or multiple dies with multiple bends and unbends. Two single-bend test methods the "tensile strip test"[9] and the "bending-under-tension test,"[15] are shown schematically in Fig. 1 and 2, respectively. A multiple bend arrangement, the drawbead simulator test,[11] is illustrated in Fig. 3. In these figures, the measured loads, which incorporate forces due to both friction and bending as the strip conforms to the geometry of the test, are indicated, and the displacements imposed in each test are indicated by the dashed arrows. Note that the geometry of the bending-under-tension test is equivalent to one half of the tensile strip test and that the drawbead simulator test is similar to a summation of four 90° bending-under-tension tests. In contrast to simple sliding friction tests on elastically deformed flat sheet, all three test methods involve a plastic component due to bending and may involve additional plastic deformation due to the applied strip loads.

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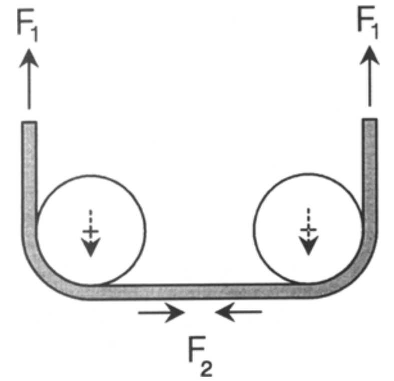

Fig. 1 Schematic of tensile strip test with nonrotating rollers.[9] Displacement of rollers is indicated by dashed arrows.

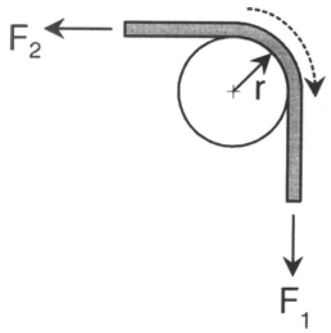

Fig. 2 Schematic of bending-under-tension friction test.[15]

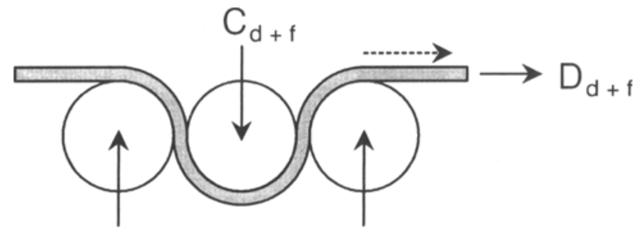

Fig. 3 Schematic of drawbead simulator friction test.[11]

Although the basic geometries illustrated in Fig. 1 to 3 are similar, differences in the assumptions and test procedures have resulted in variations in calculated friction coefficients. Equations that have been used to calculate μ from the test methods are summarized in Table 1 along with the classic "pulley equation." Solutions are based on three primary approaches: (1) an incremental force balance in which the effects of the forces are integrated along the surface of contact (Eq 2, 3, 4, 5, and 8); (2) a system force balance in which the external forces are balanced with a friction force averaged over the arc of contact (Eq 7); and (3) an incremental energy balance in which the energies associated with the displacements due to the applied forces are integrated over the arc of contact (Eq 6). Derivations of the equations are found in the cited references. To understand the test methods, experimental approaches for each are briefly reviewed below.

In the tensile strip test,[9,14] which is typically configured to a standard screw-driven test frame, the two nonrotating rollers are mounted on a rigid assembly attached to the movable crosshead. The imposed pulling force F_1 induces strain in the strip and a corresponding reaction force F_2 in the strip ligament between the rollers, as shown in Fig. 1. The relative displacement rate between the sheet and the roller is low and limited by the deformation rate in the strip. The friction coefficient for one sheet side is calculated by applying the pulley equation as modified to account for a total bend angle of 180°. Contributions due to strip bending are ignored. This test would best simulate friction conditions over a punch nose, because there is little displacement of metal over the tool.[9]

The bending-under-tension test is performed in a two-step process. First, a strip is drawn over a freely turning roller and the force due to bending and unbending, F_b, is determined as the difference between the pulling and back tension forces, F_1^* and F_2^*, respectively. A second strip is then drawn over a fixed roller, and the corresponding pulling and back tension forces F_1 and F_2 are determined. Several equations, each derived based on different sets of assumptions, have been developed to determine μ from the four forces measured during a bending-under-tension test. Based on an integrated force balance solution, Fox *et al.* [21] determined Eq 4, an expression for μ for a general contact angle θ, expressed in radians. Note that Eq 4 is similar to the pulley equation (Eq 2), except that the pulling force is modified to remove the effects of bending. To account for the effects of sheet thickness, Fox *et al.*[21] modified the coefficient in Eq 4 to incorporate the roll radius r and the sheet thickness t, thus producing Eq 5. This equation which was obtained from a force balance, is equivalent to the result of Sulonen *et al.* (Eq 6)[15] developed with an incremental energy balance integrated over a 90° arc. A third solution, Eq 7, developed by Wilson *et al.*,[22] was determined from a macroscopic force balance that assumes the external forces are balanced by a roller force due to the effects of contact pressure and an average friction force. Their result ignores both the effects of bending and sheet thickness.

The drawbead simulator test, shown in Fig. 3, has received wide industrial acceptance and was designed to measure friction under geometrical conditions similar to those observed at drawbeads in a stamping operation.[11] In this test, a strip of sheet metal is drawn through an assembly of rollers, shown schematically in Fig. 3, that resembles a drawbead. To determine μ, the drawing force, D_d, with a set of free rollers is determined as a measure of the force required to impose the bending and unbending strains associated with the roller geometry. With a set of fixed rollers, D_{d+f}, the drawing force to overcome sliding due to friction and bending, and C_{d+f}, the clamping force, are then measured. With the assumption that the contact pressure between the sheet and the rollers is constant throughout, an incremental force balance solution yields Eq 8. In the original form, friction coefficients measured with the drawbead simulator test average the interfacial behavior of both sides of the sheet. With a combination of free and fixed rollers, the drawbead simulator test can be used to evaluate the frictional behavior of a single side, although the contact pressure is fixed by the roller spacing.

Table 1 Equations Used to Calculate Friction Coefficients from Tests with Cylindrical Dies

Reference/summary of approach	Equation	Equation number
Pulley equation[20] Incremental force balance of sliding element on fixed rollers	$\mu = \frac{1}{\theta} \ln \frac{F_1}{F_2}$	2
Tensile strip test (Duncan *et al.*)[9] Strip test with fixed rollers designed to simulate punch nose; based on pulley equation; 90° bend angle	$\mu = \frac{1}{\pi} \ln \frac{F_1}{F_2}$	3
Bending-under-tension test (Fox *et al.*)[21] Incremental force balance with both free and fixed rollers; ignore sheet thickness; variable bend angle θ	$\mu = \frac{1}{\theta} \ln \frac{(F_1 - F_b)}{F_2}$	4
Bending-under-tension test (Fox *et al.*)[21] Incremental force balance with both free and fixed rollers; include sheet thickness; variable bend angle	$\mu = \frac{1}{\theta}\left(\frac{r+0.5t}{r}\right)\ln\left(\frac{F_1 - F_b}{F_2}\right)$	5
Bending-under-tension test (Sulonen *et al.*)[15] Incremental energy balance with both fixed and free rollers; 90° bend angle	$\mu = \frac{2}{\pi}\left(\frac{r+0.5t}{r}\right)\ln\left(\frac{F_1 - F_b}{F_2}\right)$	6
Bending-under-tension test (Wilson *et al.*)[22] System force balance; 90° bend angle	$\mu = \frac{2}{\theta}\left(\frac{F_1 - F_2}{F_1 + F_2}\right)$	7
Drawbead simulator test (Nine)[11] Incremental force balance; four 90° bends	$\mu = \frac{1}{\pi}\left(\frac{D_{d+f} - D_d}{C_{d+f}}\right)$	8

F_1 = Pulling force, fixed roller
F_2 = Back force, fixed roller
F_1^* = Pulling force, free roller
F_2^* = Back force, free roller
F_b = Force due to bending = $F_1^* - F_2^*$
θ = Bend angle in radians
r = Roller radius
t = Sheet thickness
D_{d+f} = Drawbead simulator drawing force with fixed rollers
D_d = Drawbead simulator drawing force with free rollers
C_{d+f} = Drawbead simulator clamping force with fixed rollers

To evaluate the significance of the various equations for determining μ from the bending-under-tension test, consider the calculated μ values summarized in Table 2 for Eq 2 through 7. The calculations were obtained from the following results of Vallance[18] on an electrogalvanized sheet steel (steel EG3 identified in Table 3): $F_1 = 11.34$ kN, $F_2 = 8.16$ kN, $F_b = 0.85$ kN, $\theta = \pi/2$, $r = 12.7$ mm, and $t = 0.76$ mm.

Four primary observations from Table 2 are as follows. The calculations produce essentially two results: ≅0.21 for Eq 2, 3, and 7 and ≅0.16 for Eq 4, 5, and 6. The contribution due to bending is significant, as Eq 2, 3, and 7 ignore F_b, whereas Eq 4, 5, and 6 incorporate bending. Equation 7, a solution based on a system force balance, is essentially equivalent to the result of the pulley equation, Eq 2. Also, the effect of sheet thickness is negligible for the 12.7-mm radius roller, as the result of Eq 4, which ignores thickness, is equivalent to the predictions of Eq 5 and 6. The effect of the sheet thickness will increase with an increase in t or decrease in r.

Even though drawbead simulator test results have been correlated with press shop performance,[5] there are several advantages of the bending-under-tension test for fundamental studies of the friction and deformation behavior of coated steels. First, the bending-under-tension test is simple to configure and measures only the frictional response of one side of the sheet. Second, the average contact pressure can be varied easily, because it is proportional to the controlled back-tension force and is given by:

$$p = \frac{F_1 + F_2}{2wr} \quad [9]$$

where w is the sheet width. With a variation in back force, the degree of substrate plastic strain also can be controlled. The direct measurement of strip forces, in addition to using controllable displacement rates, makes the test preferable to the tensile strip test for several applications.

3. Experimental Friction Test System

A bending-under-tension test system was designed and adapted to a standard MTS servohydraulic 89-kN (20,000-lb) capacity test frame equipped with a 44.5-kN (10,000-lb) actuator. The friction test system, shown schematically in Fig. 4, consists of four major components: structural frame, roller assembly, commercial hydraulic test frame to apply the primary tensile force, and a hydraulic system to control the reaction force.

A structural frame that adapts directly to the MTS load frame holds the roller assembly and provides correct alignment between the two hydraulic systems that control the forces as the strip is pulled over the roller. The frame, constructed from structural steel tubing, can be removed easily and stored as a complete unit.

The roller assembly (Fig. 4b) can be configured with fixed, free, or driven rollers with diameters between 6.35 mm (0.25 in.) and 101.6 mm (4.0 in.). This investigation used a 25.4-mm (1.0-in.) diameter case-hardened steel (60 HRC) roller with a ground and polished surface finish of 0.25 to 0.40 μm rms (10 to 16 μin. rms). A roller bearing steady rest is provided to prevent deflection of the smaller diameter rolls at high contact pressures.

Table 2 Calculated Friction Coefficients for the Bending-Under-Tension Test

Eq No.	Calculated coefficient (μ)
2, 3	0.210
4	0.165
5, 6	0.160
7	0.208

The assembly is designed to accommodate a 51 mm (2 in.) maximum width sheet metal strip with a contact angle of 90°. Each roller is mounted on an adjustable base plate, fixed at 45° to the two load axes. The rollers are supported by bearings in special bearing blocks and can be prevented from rotating with a shaft lock.

The primary tensile force (F_1 in Fig. 2) is produced by the MTS 44.5-kN (10,000-lb) actuator. The sheet metal strip is clamped in a grip, which is attached to a 44.5-kN (10-kip) load cell mounted on the MTS actuator. The MTS system has a maximum stroke of 152 mm (6 in.), which is the limiting factor in the length of strip that may be tested.

Finally, a hydraulic system provides the controlled back-tension (F_2 in Fig. 2). A hydraulic system allows for a wide variation in back tensile forces, while maintaining a steady force at any one setting. The system consists of a hydraulic cylinder with a back-pressure regulating valve. The valve bleeds fluid out of the forward cavity of the cylinder to maintain a steady pressure, as the sheet metal strip is displaced. The system was originally configured with a mechanical valve with multiple control ranges.[18] The system has been modified recently to provide direct servohydraulic control of the force exerted by the back-tension cylinder.[23] The system may be adjusted to provide for a minimum force of 0.18 kN (40 lb) up to a maximum back-tension force in excess of the test strip yield strength. The back tension is directly measured with an in-line 44.5-kN (10,000-lb) load cell.

Friction tests were performed in stroke control (*i.e.*, a constant displacement rate) at rates between 250 mm/min (10 in./min) and 5000 mm/min (200 in./min) by using the standard control system. The back-tension force was set at the desired load with the back-pressure control valve. Thus, the sheet metal strip displacement rate and back-tension force were controlled. The pulling force required to overcome the back-tension, friction, and bending forces of the sheet metal strip was directly

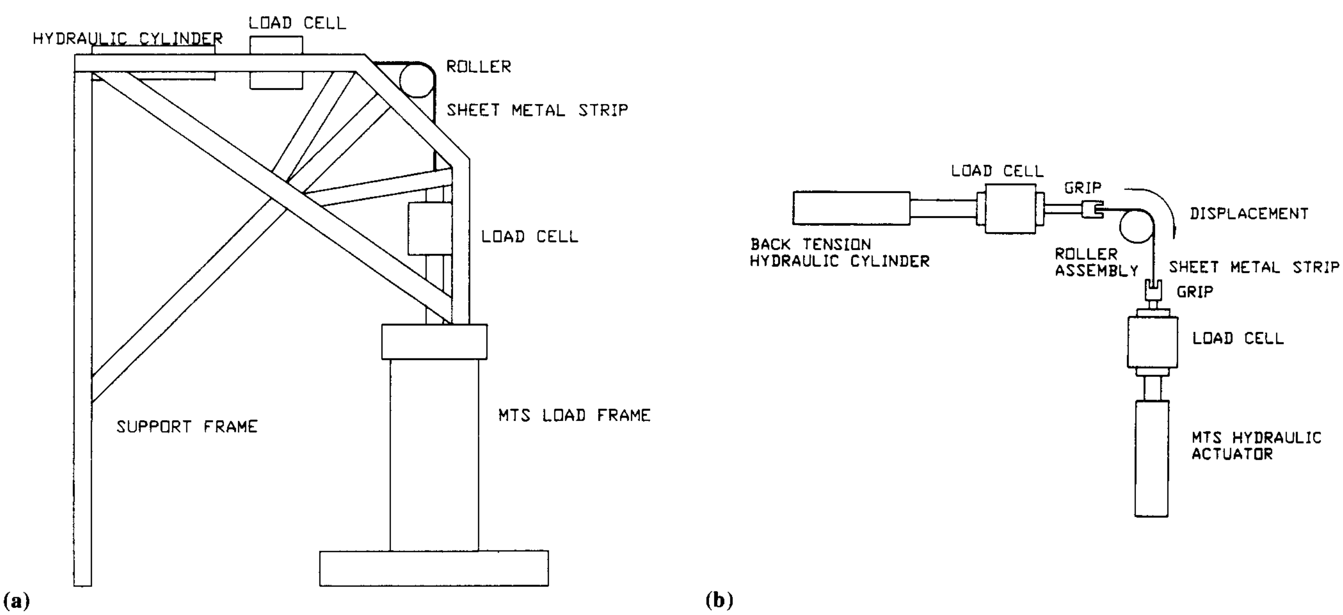

Fig. 4 **(a)** Schematic of bending-under-tension test system as attached to MTS load frame. **(b)** Detail of friction tester components used to implement the bending-under-tension test.

measured with a load cell mounted on the end of the control actuator (Fig. 4b). Pulling force, back-tension force, displacement, and time were continuously recorded with a computer-based data acquisition system.

At each back-tension force, two samples were tested, one with a free roller and one with a fixed roller. Test samples were sheared to the final dimensions of 51 by 534 mm (2 by 21 in.), cleaned with mineral spirits, and lubricated by wiping with a mineral seal oil. The specific test procedure for both fixed and free roller tests is as follows:

- Specify geometry (roller diameter, sheet width, material) and parameters (displacement rate and back-tension force)
- Fully extend the MTS actuator
- Insert a cleaned and lubricated sample and contract the back-tension cylinder to impose the preset back force
- Initiate the test

The applicability of the bending-under-tension test to commercially produced coated sheet steels was evaluated with the six materials summarized in Table 3.[16,19,24,25] The coated steels, which are characteristic of automotive steels, include three pure zinc electrogalvanized steels (EG1, EG2, and EG3), one pure zinc hot dip coated material (HD), and two steels with zinc-iron alloy coatings (N, a galvannealed product, and EA, an electrogalvanized sheet). The three EG materials were produced on different coating lines.

4. Data Analysis

Friction coefficients from the bending-under-tension test are obtained by analyzing several data sets obtained with different back-tension force levels. Each data set involves four load measurements: F_1, F_2, F_1^*, and F_2^*. Two load measurements are obtained from a free roller test and two load measurements are obtained from the corresponding fixed roller test. During the free roller test, no sliding takes place between the sheet steel strip and the roller, and only bending of the sheet occurs. In the fixed roller case, the sheet steel strip slides over the roller causing both bending and sliding friction. Both the free roller and fixed roller load pairs are obtained by averaging measured forces over the time associated with the imposed displacement in the test.

Representative load versus time data for bending-under-tension tests performed at a displacement rate of 2540 mm/min

Table 3 Commercial Coated Sheet Steels[26]

Designation	Description	Gage, mm	R_a, µin.	Tensile yield strength, MPa	Ultimate tensile strength, MPa	Total ductility, %
EG1	Electrogalvanized with 70 g/m^2 zinc per side	0.71	34	207	320	41.3
EG2	Electrogalvanized with 70 g/m^2 zinc per side	0.71	33	214	335	42.2
EG3	Electrogalvanized with 60 g/m^2 zinc per side	0.76	49	165	296	46.4
HD	Hot dip; 60 g/m^2 zinc per side	0.81	54	207	344	41.0
N	Galvannealed; 60 g/m^2 and 40 g/m^2 on each side	0.76	63	158	300	42.4
EA	Zinc-iron electrogalvanized; 40 g/m^2 on each side	1.24	NA	152	284	44.2

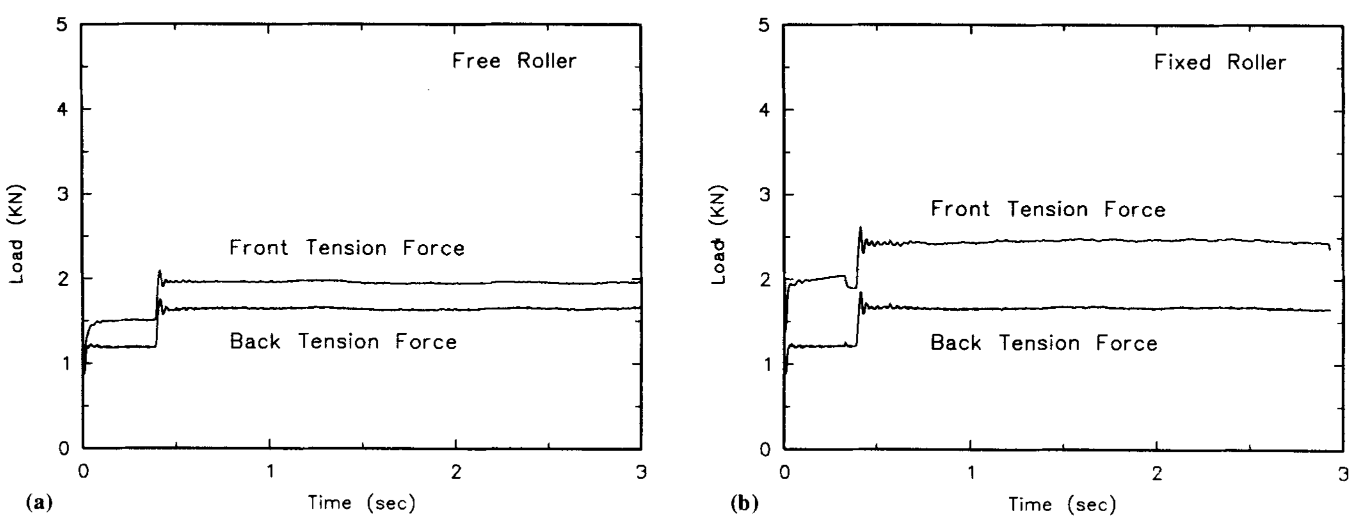

Fig. 5 Load versus time data as obtained with bending under tension test system. **(a)** Free roller test. **(b)** Fixed roller test.

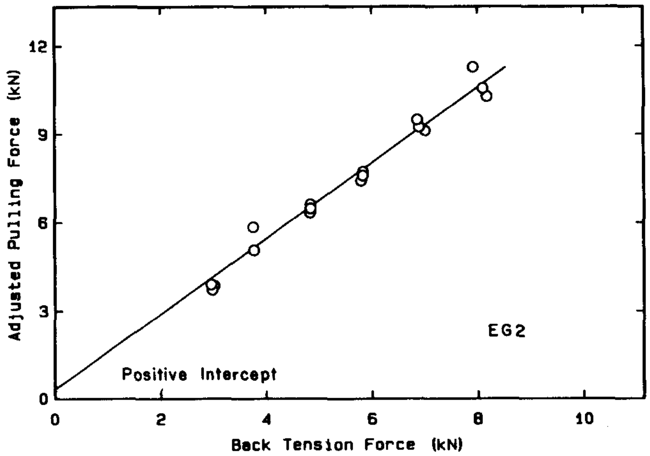

Fig. 6 Effect of back-tension force F_2 on the adjusted pulling force F_{ADJ} for steel EG2. The line represents a least-squares linear regression best-fit line through the data. Increasing back tension increases the average contact pressure between the test strip and the roller.

Table 4 Comparison of Friction Coefficients Measured with the Drawbead Simulator and Bending-Under-Tension Tests

Material	Friction coefficient, μ: Bending-under-tension test	Drawbead simulator test
EG1	0.19	0.18
EG2	0.19	0.17
N	0.15	0.14
EG3	0.12	0.15

(100 in./min) on a cold rolled AKDQ steel are shown in Fig. 5 for free and fixed rollers. The back-tension force was controlled with the servohydraulic control system. The resulting loads are essentially constant with time, and the difference between the front and back-tension forces is significantly greater for the fixed roller test. The magnitude of the initial spike on loading can be controlled by incrementally increasing the displacement rate as opposed to directly imposing the maximum rate, as was used for the data in Fig. 5.

Statistically significant calculations of μ can be obtained by performing multiple tests at the same back-tension force. However, due to inherent scatter in friction test measurements, significant improvements to the data can be obtained by analyzing sets of force data obtained with a range of back-tension forces and correspondingly a range of contact pressures. Although variations in back-tension force were obtained directly from the hydraulic controls in this study, it is interesting to note that Fox *et al.*[21] varied back tension by varying the angle of wrap, θ. The data sets were analyzed to determine data pairs of the back-tension force, F_2, and the adjusted pulling force F_{ADJ} (where $F_{ADJ} = F_1 - F_b$). Because the friction coefficient is a function of the natural log of the ratio of F_{ADJ} to F_2, the data are plotted as F_{ADJ} versus F_2, as shown in Fig. 6, and μ is determined from the slope. Linear regression analysis of the data

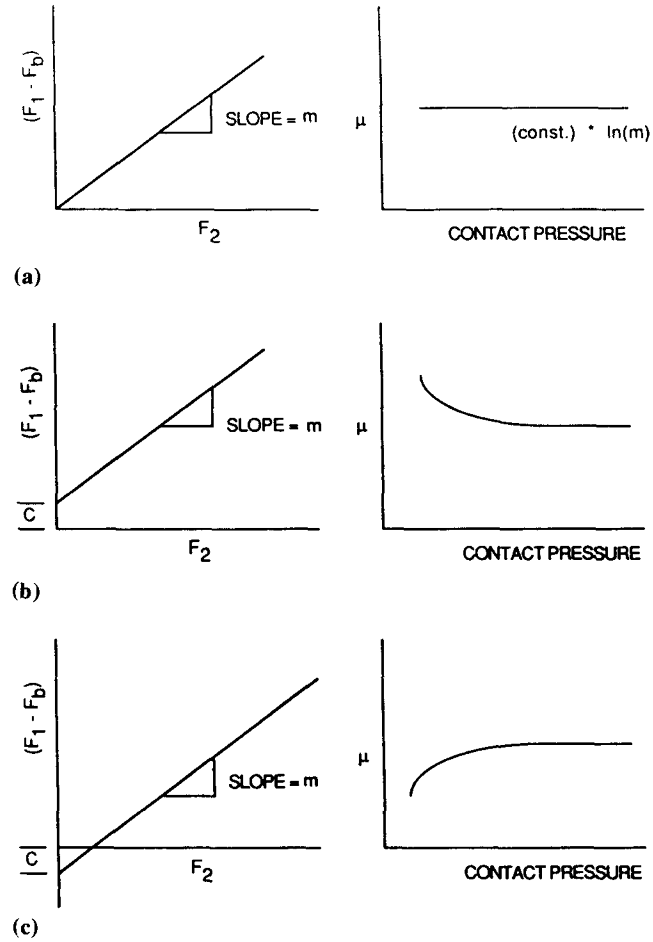

Fig. 7 Schematic illustrating the different behavior of μ as a function of contact pressure for Coulombic friction in a bending-under-tension friction test. **(a)** Zero *y*-intercept. **(b)** Positive *y*-intercept. **(c)** Negative *y*-intercept.[19]

in Fig. 6 shows that the results are accurately described by the indicated linear function.

As indicated above, the derivations of the friction coefficient from bending-under-tension data are based on two primary assumptions: Coulombic friction applies and the force distribution over the arc of contact obeys the pulley equation. If the coefficient of friction is a constant, then for the same strip thickness, t, and roll radius, r, the ratios of F_{ADJ} to F_2 would be constant. If F_{ADJ} were plotted as a function of F_2, then this constant ratio would imply that the plot would yield a straight line. The slope of this line would equal $e^{\mu/C}$, where $C = (2/\pi)((r + 0.5t)/r)$. Because of the linear relationships between F_{ADJ} and F_2, the data obtained and discussed in this article are presented in plots, as shown in Fig. 6.

Based on the energy analysis of Sulonen *et al.*,[15] the forces in a bending-under-tension test are related to the friction force F_μ by:

$$F_1 - F_2 = F_\mu + F_b \qquad [10]$$

If there were no back-tension force (*i.e.*, $F_2 = 0$), F_{ADJ} would equal F_μ, the force due to sliding friction. For this case, the av-

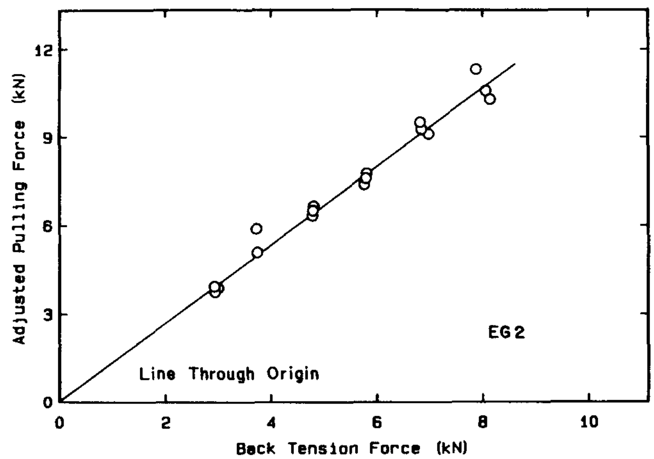

Fig. 8 Data for EG2 in Fig. 6 replotted with the straight line constrained through the origin.

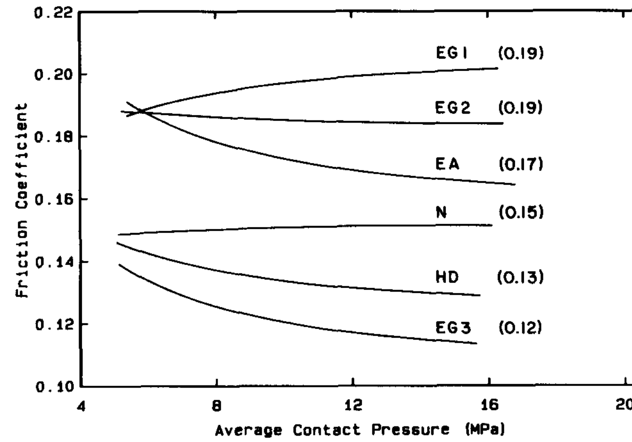

Fig. 9 Effect of average contact pressure on the friction response of a series of commercially produced zinc-based coated sheet steels. The values in parentheses were obtained by constraining the F_2 versus F_{ADJ} data through the origin.

erage contact pressure would be zero and hence F_μ would be zero. Therefore, a plot of F_{ADJ} versus F_2 should be linear through the origin, *i.e.*, a best-fit line through a set of experimental data should have a y-intercept of zero. Figure 6, a plot of data obtained on steel EG2, has a positive y-intercept and shows that the experimental data deviate from theoretical predictions. The data suggest that the friction coefficient varies with contact pressure, an observation that is in violation of the Coulombic friction assumptions used to derive the basic bending-under-tension friction equation.

Experimental data similar to those shown in Fig. 6 may yield three possible y-intercepts: zero, positive, or negative values. The significance of the different intercepts on the apparent pressure dependence of measured friction coefficients is illustrated schematically in Fig. 7, which shows calculated μ versus average contact pressures for each of the three possible extrapolations. The predictions in Fig. 7 were obtained by substituting the linear equation, $F_1 - F_b = F_{ADJ} = mF_2 + b$, into Eq 6; m-slope, b = y-intercept. A zero y-intercept would produce a constant μ; a positive y-intercept would result in a decrease in μ with pressure, and a negative y-intercept would result in an increase in μ with pressure.

To eliminate the effects on a non-zero y-intercept, the plot of F_{ADJ} versus F_2 can be constrained to plot through the origin. To illustrate this procedure, the data in Fig. 6 for EG2 are replotted in Fig. 8 with the extrapolation constrained to go through zero. The significance of the different data plotting methods is considered in the following discussion of friction in the six coated sheet steels.

5. Applicability of the Bending-Under-Tension Test to Coated Sheet Steels

Bending-under-tension test data were obtained for the six coated sheet steels summarized in Table 3 and analyzed following the procedures discussed in conjunction with Fig. 6 and 8. According to Fig. 6, the resulting plots of μ, calculated with Eq 6, versus average contact pressure, as calculated with Eq 9, are presented in Fig. 9. The data indicate that the frictional behavior of the six commercial steels, as monitored with the bending-under-tension test, are significantly different and that all three of the possible behaviors illustrated in Fig. 7 are exhibited by the six steels.

The friction coefficient appears to decrease with pressure for EA, HD, and EG3, remain constant for N and EG2, and increase with pressure for EG1. The observed variations in friction coefficient are in violation of the assumption of Coulombic friction in the derivation of the equations for μ. The friction coefficients calculated from data plots of F_{ADJ} versus F_2 constrained through the origin are shown in parentheses in Fig. 9. The friction coefficients from this second method are similar to the average values obtained from a best-fit through the experimental data.

An error analysis of the friction test based on Eq 6 yields an accuracy of the calculated friction coefficient of ±0.01. Both methods used to reduce the data yield virtually the same results as shown in Fig. 9. Although the actual friction data deviate from linearity as required for Coulombic friction, the deviation is small within the range of average contact pressures investigated. The friction coefficient varies with average contact pressure, but the variation is negligible, as a change in friction coefficient of 0.01 is relatively small.

To further evaluate the potential applicability of the bending-under-tension test, four of the materials presented in Fig. 9 also were evaluated with a drawbead simulator test, and a comparison of the results is presented in Table 4. The data for the bending-under-tension test correspond to the data constrained through the origin. Although there are slight differences in the measured friction coefficients, the frictional behaviors of the materials are ranked in essentially the same order.

Analysis of the fundamental differences in friction behavior and the formability of the six experimental steels has been the subject of several recent investigations.[16-19,24-26] The friction behavior depends on coating strength, surface roughness, and crystallographic texture. For example, compare the behavior of EG1, with an average friction coefficient of 0.19, to EG3, with μ approximately 0.12. Coating EG1 possesses a crystallographic texture that promotes coating deformation during forming, whereas EG3 has a significantly different crystallographic texture and cracks to accommodate imposed strains. The cracks in EG3 were interpreted[18] as providing entrapment sites for lubricant, which improved the overall lubrication at the interface compared to EG1. A complete analysis of the metallurgical variables that control friction is the subject of another publication.[27]

The bending-under-tension test has been shown to provide significant information on the fundamental behavior of coatings during die/metal contact. However, several of the assumptions invoked in the derivations of μ require further attention. The pressure distribution was assumed to be uniform as controlled by the pulley equation. However, due to bending on entrance to the roll and unbending on exiting from the roll, pressure peaks develop, which are not incorporated in the analysis.[28] The necessity of measuring F_b at each average pressure has been questioned by Davies and Stewart,[17] who suggest that, when F_b is independent of back-tension force, then F_b can be obtained directly from the y-intercept of a plot of F_2 versus F_1. A more thorough investigation of the influence of pressure distribution on the friction coefficient along with a direct analysis of the influence of bending and lubrication is required to optimize the bending-under-tension test.

6. Conclusions

The bending-under-tension test provides a controllable test method to directly evaluate the friction behavior of single sides of coated sheet steels. The technique is suitable for both fundamental studies of coated sheet steels and analysis of sheet steels for production operations.

A comparison of the different equations used to calculate μ from the bending-under-tension test shows that the contribution due to bending is significant and cannot be ignored. Furthermore, the solution based on an energy analysis is equivalent to the solution based on a force balance.

Back-tension force versus adjusted pulling force data should be analyzed with a linear extrapolation through the origin to obtain friction coefficients that are pressure independent and consistent with Coulombic friction. Deviations from this extrapolation in coated sheet steels indicate that the friction coefficient in coated sheet steels varies with contact pressure.

Further analysis of the assumptions invoked in the solutions for μ is required to optimize the applicability of the bending-under-tension test.

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