BDX-613-1316
Distribution Category UC-38

FRICITION AND WEAR STUDIES FOR
BONDED THIN-FILM LUBRICANTS

Published August 1976

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PDO 6984735
Final Report

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FRICITION AND WEAR STUDIES FOR BONDED THIN-FILM LUBRICANTS

BDX-613-1316, UNCLASSIFIED Final Report, Published August 1976

Prepared by D. P. Pope, D/822, under PDO 6984735

Ten commercially available, bonded dry-film lubricants were evaluated for use in electromechanical devices which must be highly reliable after extended periods of storage with exposure to extreme temperatures and other extreme atmospheric conditions. A tribometer utilizing the oscillating-bar concept was developed to determine the friction coefficients of lightly loaded samples by several methods. Interaction among friction-influencing variables was investigated, mean and standard deviations were calculated for comparison, and wear characteristics of the lubricants were determined. Test results indicate that Electro-film 77S is the best of the tested lubricants for light-load applications.

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A prime contractor for the United
States Energy Research and
Development Administration
Contract Number E(29-1)-613 USERDA

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SUMMARY

Bonded dry-film lubricants were evaluated to select a suitable lubricant for use in electromechanical devices which must be highly reliable after extended periods of storage and exposure to extreme atmospheric conditions. Mechanism tolerances and clearances require that the bonded lubricants be applied in thin films from 0.0002 to 0.0005 inch (5.08 to 12.7 μm) in thickness and that they exhibit a low friction coefficient both at the light loads of ambient environments and the loads of high-g environments.

A tribometer, needed to obtain friction data by several techniques from dry-film-lubricant samples at light and moderate loads, was developed, since a suitable instrument was not commercially available. The tribometer utilized a slightly modified oscillating-bar concept to test different shapes of samples in several test modes. The tester successfully determined the friction coefficients of lightly loaded samples by several methods. A correlation of the data taken increased its reliability.

Light-load friction and wear characteristics were determined for ten commercially available, bonded dry-film lubricants. Of the six lubricants originally evaluated, Electrofilm 4396 was judged to be the best of the group. A later preliminary study of four other lubricants revealed that Electrofilm 77S may be better than 4396 for use in the small precision electromechanical devices manufactured by Bendix.

After tests on the original six lubricants found 4396 to be superior, a factorial design experiment used it to find the interaction between such friction-influencing variables as substrate finish and hardness, mating-part finish and hardness, and film polishing. Despite the complexity of the resulting data, the range of friction coefficients for many friction systems was determined. Additional experiments found that friction coefficients decreased with increasing load, were not affected by film polishing or film compaction, and were relatively unaffected by sliding velocity.

Light-load-coefficient and wear-life data were compiled for Electrofilm 77S, 4396, and 4396S bonded dry-film lubricants. Static and dynamic friction performance were studied using the tribometer specially designed to test bonded films, and mean and standard deviations were calculated for comparison. The data indicated that Electrofilm 77S was the best lubricant for light-load applications.

DISCUSSION

SCOPE AND PURPOSE

This project was initiated to study lubricating systems for use in electromechanical devices which must be highly reliable after extended periods of storage and exposure to extreme atmospheric conditions. Bonded dry films are one of the few types of lubricants that can be used under these circumstances. Extensive evaluations of many bonded dry-film-lubricant formulas therefore were conducted. Emphasis was directed toward gathering light-load friction and wear data.

Because an acceptable friction tester was not commercially available, a tribometer was developed and used to evaluate the bonded dry-film lubricants selected for study.

ACTIVITY

Testers have been devised to measure frictional forces and wear of machine components. The inclined plane has long been used in elementary classroom demonstrations and as a research tool to evaluate the static frictional resistance of materials. Wear testers to evaluate surface treatments have been designed for many kinds of motion. These testers have used rotating journals, reciprocating or rotating wear blocks pressed against flat surfaces, and the rotation of spheres against spheres. Many special testers have been made for such particular machine elements as ball bearings, roller bearings, and gears.

Lubrication problems in space vehicles, operating at temperature extremes and in the vacuum of space, have led to the development of "solid" lubricants. The principal objective of the development has been to improve the wear-life of frictional surfaces. A reasonable coefficient of friction has been accepted without concerted attempt to minimize friction forces. Currently accepted theory suggests that the frictional force is independent of the "real" contact area, and depends on the shear-strength and hardness of the materials. The true nature of the friction force is probably associated with elastic deformation and contact-stress distribution in the vicinity of the sliding.

Frictional coefficients are commonly considered to be independent of both sliding velocity and gross contact pressure, although evidence exists that friction tends to be reduced with increased contact stress.

Development of Friction Tester

The tribometer developed for this activity uses J. Den Hartog's oscillating-bar concept, but simple modification yields several different methods of friction-coefficient determination. Basically, the tribometer (Figure 1) consists of two rollers mounted on shafts driven by a variable-speed motor through a spiral helical-gear train. The rollers rotate either in the same direction or in opposite directions by interchanging right- and left-hand driving gears (Figure 2). A bar coated with the test lubricant is placed on the rollers. The tester senses the friction at the interface of the coated sample and the rollers when measuring the friction between the rollers and the bar. The friction is measured between the drive shaft and the journal when the device is operated as a bearing tester. The plate on which the shafts, bearings, and gears rest may be tilted.

The five methods of friction measurement which were used more frequently during the evaluation of the tester are discussed in the following five sections of this report; six additional modes of operation are listed in Appendix A.

Oscillating Mode With Sliding Friction

Figure 3 is a schematic of the apparatus showing the rollers rotating at the same speed, but in opposite directions. If the rollers rotate at a sufficiently high speed, sliding contact exists at each roller. A displacement of the bar from its centered position changes the amount of weight carried by each roller; the roller closer to the bar's center of gravity (cg) supports a greater portion of the weight than the remote roller. With the frictional forces thus unbalanced, a net frictional force acts to return the bar to the centered position. If no energy were dissipated within the system, the bar would oscillate indefinitely in a harmonic manner; however, the surrounding atmosphere damps the oscillation, and the motion decays until the bar stops. The dynamic (sliding) coefficient of friction may be calculated from the following equation, the derivation of which is given in Appendix A.

$$\mu_D = \frac{H}{\frac{gT^2}{4\pi^2} + h}, \quad (1)$$

where

μ_D is the dynamic coefficient of friction,

T is the time (in seconds) of an oscillation, and

g is gravitational acceleration (386 inches/sec², or 9.8 m/s²).

Figure 1. Tribometer

Figure 2. Interchangeable Tribometer Right- and Left-Hand Driving Gears

Figure 3. Schematic of Oscillating Mode of Tribometer Operation

The most reliable performance is obtained if the test lubricant has a friction coefficient in the range from 0.05 to 0.4. A stopwatch, calibrated to hundredths of a second, was used to determine the time period of the bar's oscillation. Friction coefficients calculated from such data are estimated to be repeatable within 8 percent; such instrumentation as photocells and electronic timers might improve the repeatability to 2 to 4 percent.

As shown in Appendix A, a negligible error is associated with atmospheric damping of the bar if a minimum of four cycles is completed before the oscillation stops.

To apply Equation 1, the bar must oscillate in a harmonic manner; to assure this, the surface speed of the roller at the point of rubbing must always be greater than the linear velocity of the bar. If the rollers are driven too slowly, "stick-slip" motion will occur (see Appendix A) and Equation 1 will not apply.

The variable relative velocity between the bar and the rollers does not appear to affect the friction coefficient measurement if the roller surface speed is from approximately 600 to 1000 inches per minute (254.0 to 423.3 mm/s). A typical maximum bar translation velocity is 480 inches per minute (203.2 mm/s); that is, $\mu = 0.20$, $h = 2$ inches (50.8 mm), and $x_0 = 1.5$ inches (38.1 mm).

Stationary Mode With Sliding Friction

If both rollers are rotated in the same direction, the friction forces will drive the bar toward one end. In this case, the bar is restrained by a force-measuring device (Figure 4), and the sliding coefficient of friction is calculated from Equation 2, the derivation of which is given in Appendix A.

$$\mu_D = \frac{P}{W}, \quad (2)$$

where

P is the measured force, and

W is the weight of the bar.

Experience with the tribometer indicates satisfactory agreement of values found by oscillation and by force measurement, although the force measurement method yields values up to 20 percent higher. This difference has not yet been explained.

Data can be taken with a dial-indicator type of force gage or an electrical force transducer which will increase the reliability of the force measurements.

Figure 4. Sliding-Friction Determination Using Force Measurement

The operating range of this mode depends upon the sensitivity limits of the force-measuring instrument. A mechanical force gage satisfactorily determined friction coefficients in the range from 0.050 to 0.8. When performing dynamic tests at high roller speeds, care must be taken to avoid obtaining incorrect data caused by lubricant-film breakthrough or by excessive temperature increases at the rubbing interface.

Stationary Mode With Static Friction

With the bar in static contact with nonrotating rollers, a force gage is brought into contact with one end of the bar and the force is increased until the bar slips. Both rollers must be locked so

that neither can turn and take up the backlash in the drive mechanism. The static friction coefficient is calculated from the following equation, the derivation of which is shown in Appendix A.

$$\mu_S = \frac{P}{W}. \quad (3)$$

Several readings are taken, and the average is calculated. The variation in individual observations may be substantial if the surface finish is rough or if wear has accumulated.

Tilt Method With Sliding Friction

If the entire apparatus is tilted from its level position (Figure 5), the principle of the elementary inclined plane is used to evaluate the friction coefficient. Both rollers are rotated in the same direction, with the friction forces acting to hold the bar in equilibrium in opposition to gravity which otherwise would pull the bar down the incline. The coefficient of sliding friction is the tangent of the angle of inclination, as expressed in the following equation.

$$\mu_D = \tan \alpha. \quad (4)$$

The derivation of Equation 4 is given in Appendix A.

Tilt Method With Static Friction

In this mode for determining the static friction coefficient, the rollers are locked in place and the entire mechanism platform is tilted until the bar slips on the rollers. The following equation states the corresponding dynamic case.

$$\mu_S = \tan \alpha. \quad (5)$$

Repeatability of the static coefficient thus measured is limited by surface variations in both the bar and the rollers; however, such variations reflect the real case when components of these geometric shapes are required to move in relation to each other.

The accuracy of the tilt mode depends upon the precision of the angle-measuring device. The practical measurable range of friction coefficients with this mode is approximately 0.020 to 0.8. Caution also must be taken during dynamic tests using this method to avoid overheating and film breakdown.

Figures 6, 7, and 8 detail the rollers and test bar used as test specimens for this study.

Figure 5. Tribometer Tilted for Inclined-Plane Friction Measurement

Cost and Operating Characteristics

The tribometer's cost was approximately \$2500, excluding the expense of measuring instruments and motor-speed controls. All hardware consists of standard commercial items; only the mounting blocks, plates, and shafts required special fabrication. Experience with the tribometer suggests that the following design changes would enhance its performance and operating characteristics.

- The present hardened gears should be replaced with unhardened gears; smoother running would result from the wear-in of mating gears.
- The bronze sleeve-bearing pillow blocks should be replaced with self-aligning, sealed, precision ball-bearing pillow blocks. Smoother running and less maintenance would result from reduced vibration, wear, and misalignment.

Figure 6. Flat Test Bar Assembly and Flat Rollers

Figure 7. Round Test Bar Assembly and V-Shaped Rollers

Figure 8. Round Test Bar Disassembled

- Weight and size of the tribometer should be reduced to increase portability.
- Total instrumentation of data-gathering functions would eliminate operator influence.

This tribometer was designed specifically for light-load friction-coefficient determination (contact loads from 5 grams to several hundred grams); however, operation at contact loads of approximately 5 pounds (22.2 N) per roller is satisfactory.

In addition to its use in evaluating the frictional characteristics of bars or rods in contact with rollers, the device is useful for studies of the frictional characteristics of journal and antifriction bearings. The rollers are supported by test bearings mounted between the roller shafts and the rollers. Relative motion exists within the test bearings while the bar maintains static rolling contact with the bearings at all times.

Testing of Bonded Dry-Film Lubricants, Part 1

Test Lubricants

After reviewing data in commercial publications, private research papers, and government-sponsored research reports, the following six lubricants were initially selected for evaluation.

- Electrofilm 4396--an organic, thermosetting, resin-bonded solid film lubricant consisting of approximately 90 percent molybdenum disulfide (MoS_2) and 10 percent graphite, by weight, and cured at 350 to 375°F (176 to 191°C) in 1 hour. This is the most frequently used lubricant at Bendix when the high cure-temperature is permissible.
- Electrofilm 5396--a thermosetting, resin-bonded solid film lubricant of the type described in Mil-L-8937. It cures at 300°F (148°C) in 1 hour and is from 0.0002 to 0.0005 inch (5.08 to 12.7 μm) in thickness.
- Electrofilm 4306--a solid film with lubricant solids of 100-percent MoS_2 in a resin binder that cures at 375°F (191°C) in 1 hour. The typical thickness is from 0.0002 to 0.005 inch (5.08 to 127.0 μm).
- Electrofilm 5306--a solid film lubricant made from a combination of lubricating pigments, a low-temperature thermosetting resin, and MoS_2 . Its typical thickness is from 0.0002 to 0.0005 inch.
- Electrofilm 99A--a resin-bonded solid lubricant of MoS_2 and graphite that cures at 250°F (121°C). The typical thickness is from 0.0002 to 0.0005 inch.
- Dow Corning's I-3943--a new, bonded, solid-film lubricant of MoS_2 and antimony oxide (Sb_2O_3) in a silicone binder that cures at 480°F (248°C) in 1/2 hour, or 24 hours in air.

The tribometer rollers, made of 303 stainless steel with a 4AA to 16AA surface finish, were cleaned with trichloroethylene prior to tests of each lubricant. Dynamic tests were conducted at a roller speed of 360 rpm.

After the first six lubricants were tested, four other lubricants were chosen for evaluation:

- Electrofilm 77S--a new, bonded, solid lubricant using new high-adhesion, thermosetting resins that cure at 375°F (191°C) in 1 hour, new micron-size solid lubricants, new milling techniques, and new plasticizers and solvents.
- MLF-5--a formula consisting of 77 percent MoS_2 , 5 percent graphite, and 18 percent sodium silicate binder.
- MLR-2--a formula of 37 percent MoS_2 , 30 percent Sb_2O_3 , and 30 percent polyimide binder.
- MLF-6--a formula with a lubricant-to-binder ratio of 10 parts MoS_2 , 1 part graphite, 3 parts bismuth, and 7 parts sodium silicate binder.

Testing of Original Six Lubricants

Data was taken for each lubricant using both Type 1 and Type 2 test bars (Figure 9). The Type 1 test bar, made of unhardened A6 tool steel, simulates the contact loads found in miniature mechanisms. The Type 2 test bar, used early in the study, is heavier than the Type 1 bar. The Type 2 bar is made of 303 stainless steel, vapor-blasted with 500-grit Al_2O_3 prior to lubricant application; it was used with all lubricants except Electrofilm 99A, which was applied to a 6061-T6 aluminum test bar.

Five methods of friction measurement were used in this study:

- Oscillating mode with sliding friction;
- Stationary mode with sliding friction (dynamic force gage method);
- Tilt mode with sliding friction (tilt method);
- Stationary mode, static friction (static force gage method); and
- Tilt mode, static friction.

Life tests were performed on Type 1 test bars in the initial evaluation of the six lubricants. The friction coefficients were measured using the dynamic force gage over a 2-hour period. Readings were taken every 15 minutes (Figures 10 and 11). Lubricants 4396 and 99A exhibited longer wear life than the other lubricants tested.

To rank the wear-life performance of all six lubricants, the friction coefficients measured at 0.0, 15, 30, and 45 minutes of the wear tests were compared: the lower the coefficient, the higher the ranking. A time ranking then was made; because the lubricant must perform well initially, the highest ranking was given to the lubricant with the lowest time.

The friction-coefficient ranking and time ranking for each reading were multiplied and the results were added together. The higher numbers obtained indicated better wear-life performance (Table 1).

Statistical Analysis of Friction-Coefficient Tests of Original Six Lubricants

This analysis helped determine which of the six lubricants exhibited the lowest friction coefficient and the longest wear life. The data consisted of both dynamic and static friction-coefficient readings taken by the force gage and tilt methods, and of dynamic friction readings taken by the oscillation method. Type 1 and Type 2 test bars for each of the six lubricants were used. Data

Figure 9. Type 1 Test Bar (Next to Scale) and Type 2 Test Bar

on unpolished and lightly polished samples were combined for each test method because of the insignificant difference between the two. To provide identification, each lubricant was assigned a number.

The first step in the analysis determined the average and standard deviations for each lubricant at each of the ten condition combinations of friction, type, method, and load (Table 2).

The second step, a two-way analysis of variance for each method, determined the differences among lubricants; the six lubricants were one factor, and the standard and nonstandard loads were the second factor. The analysis of variance showed a significant difference between loads and a significant interaction between lubricants and loads.

Because of the statistical significance of this interaction, the standard and nonstandard loads were analyzed separately as the third step. A one-way analysis of variance for each of the ten combinations confirmed a significant difference among lubricants in each case. This analysis was not precise because of a significant difference among the lubricant variables for all but the oscillation method and the tilt method with nonstandard bar and dynamic friction; however, the inaccuracy was small because the calculated "f" value far exceeded the significant test criterion.

The fourth step, a separation-of-means test to determine which lubricants are different, based on the calculated averages, was performed separately for each combination because of the significant

Text continued on page 25.

Figure 10. Lubricant Wear-Test Plot, Test 1

Figure 11. Lubricant Wear-Test Plot, Test 2

Table 1. Lubricant Wear-Life Test Results

Lubricant	Test Number	Friction Coefficient Value X Time Value (Minutes)				Total	Rank
		0.0	15	30	45		
99A	1	3 x 1.0*	5 x 0.8	5 x 0.6	5 x 0.4	12	1
	2	5 x 1.0	5 x 0.8	5 x 0.6	5 x 0.4	14	1
5396	1	3 x 1.0	5 x 0.8	5 x 0.6	5 x 0.4	12	1
	2	5 x 1.0	5 x 0.8	3 x 0.6	3 x 0.4	12	2
4396	1	5 x 1.0	5 x 0.8	3 x 0.6	3 x 0.4	12	1
	2	5 x 1.0	5 x 0.8	5 x 0.6	5 x 0.4	14	1
I-3943	1	3 x 1.0	5 x 0.8	3 x 0.6	3 x 0.4	10	2
	2	5 x 1.0	5 x 0.8	5 x 0.6	5 x 0.4	14	1
4306	1	5 x 1.0	3 x 0.8	1 x 0.6	1 x 0.4	8.4	3
	2	5 x 1.0	3 x 0.8	1 x 0.6	1 x 0.4	8.4	3
5306	1	1 x 1.0	3 x 0.8	1 x 0.6	1 x 0.4	4.4	4
	2	5 x 1.0	5 x 0.8	1 x 0.6	1 x 0.4	10	4
*The higher the coefficient value, the lower the coefficient; the higher the time value, the better the lubricant performed initially.							

Table 2. Average and Standard Deviations Plotted by Statistical Analysis

Friction	Test Method	Load	Item	Lubricant					
				4396	5396	4306	5306	99A	I-3943
Dynamic	Oscillation	Standard	Standard Deviation	23.86	24.04	24.65	36.18	33.47	19.83
			Average	189.95	212.08	216.17	214.67	239.58	243.89
			Sample Size	19	12	12	12	12	9
		Nonstandard	Standard Deviation	19.33	11.47	11.06	16.62	12.65	11.78
			Average	192.26	187.54	204.08	203.54	209.19	264.07
			Sample Size	27	24	12	24	16	12
	Force Gage	Standard	Standard Deviation	12.64	12.36	41.93	21.47	29.16	5.72
			Average	186.05	199.17	221.00	216.00	206.83	258.67
			Sample Size	19	12	12	12	12	6
		Nonstandard	Standard Deviation	4.95	5.16	9.15	18.98	20.73	15.81
			Average	179.70	179.67	233.83	191.67	213.44	274.92
			Sample Size	27	24	12	24	16	12
	Tilt	Standard	Standard Deviation	32.42	11.01	27.41	28.62	21.66	10.75
			Average	190.44	210.75	242.75	226.50	231.67	291.00
			Sample Size	18	12	12	12	12	6
		Nonstandard	Standard Deviation	10.58	15.51	13.71	12.78	10.20	13.29
			Average	184.48	182.50	247.33	222.71	222.00	297.58
			Sample Size	27	24	9	24	15	12
Static	Force Gage	Standard	Standard Deviation	36.53	8.89	46.03	18.95	21.94	20.37
			Average	148.53	133.17	148.42	143.33	202.92	237.83
			Sample Size	19	12	12	12	12	6
		Nonstandard	Standard Deviation	20.56	21.77	42.40	13.74	32.18	34.87
			Average	118.67	109.79	146.17	121.22	200.79	291.75
			Sample Size	27	24	12	23	14	12
	Tilt	Standard	Standard Deviation	27.83	16.21	36.06	24.03	9.76	17.35
			Average	170.78	178.17	178.42	171.42	202.17	232.83
			Sample Size	18	12	12	12	12	6
		Nonstandard	Standard Deviation	7.60	8.10	14.36	25.38	27.60	14.89
			Average	147.52	117.91	152.08	185.38	193.50	292.92
			Sample Size	27	23	12	24	14	12
Resulting Static Friction Coefficients				0.1596	0.1555	0.1634	0.157	0.2025	0.235
Resulting Dynamic Friction Coefficients				0.1888	0.2072	0.2266	0.2188	0.2260	0.2640

differences in load and interaction. Table 3 shows the results: any two means not underscored by the same line are significantly different. Lubricants 4396 and 5396, yielding the lower friction coefficients, are the better lubricants of the six tested.

Further Testing of 4396 and 5396

The friction-test analysis showed 4396 and 5396 to be superior to the other four lubricants; their friction coefficients were statistically the same, while 5396 exhibited slightly better repeatability than 4396. Because the results were so close for lubricants 4396 and 5396, further data using a larger sample size was gathered so that a more confident analysis could be conducted. The results of this evaluation showed that the friction coefficients were statistically equal, but they also indicated that lubricant 4396 was slightly more repeatable. Lubricant 4396 therefore was selected as the best of the six lubricants.

Data was taken using only the Type 1 test bar. Test specimens made from hardened A6 tool steel and 6061-T6 aluminum were vapor-blasted with 500-grit Al_2O_3 prior to being spray-coated with each lubricant. Friction coefficients were measured using all five tester modes of operation (Table 4). The 303 stainless steel rollers had been vapor-blasted with 220-grit Al_2O_3 to a 28-microinch (711 nm) AA surface finish. The friction tests were conducted on the lubricant-coated samples both as received and after being rubbed firmly with a paper towel. In the five frictional tests, one average favored lubricant 5396, while two standard deviations favored lubricant 4396. All other averages and standard deviations for the different test methods were statistically equal.

Factorial Design Experiment and Statistical Analysis of 4396

Because lubricant 4396 was chosen as the best of the six lubricants, it was used in tests of different combinations of the following friction-influencing variables: substrate (material under film) finish, substrate hardness, film burnishing, and mating-part finish and hardness. Sets of 303 and 440 C stainless steel hardened to R_C 60 and 6061-T6 aluminum tester rollers were used. One set of the test rollers had a 4-microinch (101 nm) AA finish, and the other set had a 30-microinch (762 nm) AA finish. Eight test bars each of hardened steel, unhardened steel, hardened aluminum, and unhardened aluminum were vapor-blasted, half with 500-grit abrasive and the other half with 220-grit abrasive, before the lubricant was applied.

In general, five variables were studied; if only four or three variables were studied, however, either the first variable or the first two variables were held constant. The first two variables represented two components of the test setup.

Table 3. Separation-Of-Means Analysis Results

Friction	Test Method	Load	Lubricant Number** and Test Results*						
Dynamic	Oscillation	Standard	1 *** 189.95	2 **** <u>212.08</u>	4 †† 214.67	3 † <u>216.17</u>	5 ††† <u>239.58</u>	6 †††† <u>243.89</u>	
		Nonstandard	2 <u>187.54</u>	1 <u>192.26</u>	4 <u>203.54</u>	3 204.08	5 <u>209.19</u>	6 264.07	
	Force Gage	Standard	1 186.05	2 <u>199.17</u>	5 <u>206.83</u>	4 <u>216.00</u>	3 <u>221.00</u>	6 258.67	
		Nonstandard	2 <u>179.67</u>	1 <u>179.70</u>	4 191.67	5 213.44	3 233.83	6 274.92	
	Tilt	Standard	1 190.44	2 <u>210.75</u>	4 226.50	5 231.67	3 <u>242.75</u>	6 291.00	
		Nonstandard	2 <u>182.50</u>	1 <u>184.48</u>	5 <u>222.00</u>	4 <u>222.71</u>	3 247.33	6 297.58	
	Static	Force Gage	Standard	2 <u>133.17</u>	4 <u>143.33</u>	3 <u>148.42</u>	1 <u>148.53</u>	5 202.92	6 237.81
			Nonstandard	2 <u>109.79</u>	1 <u>118.67</u>	4 <u>121.22</u>	3 146.17	5 200.79	6 291.75
Tilt		Standard	1 <u>170.78</u>	4 <u>171.42</u>	2 <u>178.17</u>	3 <u>178.42</u>	5 202.17	6 232.81	
		Nonstandard	2 117.91	1 <u>147.52</u>	3 <u>152.09</u>	4 <u>185.38</u>	5 <u>193.50</u>	6 292.90	

*Tabular value is the average friction coefficient per lubricant in 10⁻³ unit

**Underlined lubricants are equivalent

***1 - Electrofilm 4396

****2 - Electrofilm 5396

†3 - Electrofilm 4306

††4 - Electrofilm 5306

†††5 - Electrofilm 99A

††††6 - Dow Corning's I-3943

Table 4. Friction Coefficients of Electrofilm 4396 and 5396

Coefficient	Test Method*	4396	5396
Dynamic	Oscillation	— 7.56	9.39
		x 161.50	155.69**
		n 34	36
	Force Gage	— 10.29 **	17.22
		x 170.60	169.39
		n 35	33
	Tilt	— 9.22**	30.49
		x 175.94	174.15
		n 32	34
Static	Force Gage	— 39.33	30.49
		x 198.16	197.80
		n 37	39
	Tilt	— 17.76	25.09
		x 190.94	187.47
		n 32	34

*Averages and Standard Deviations in 10⁻³ units

**No significant difference in the averages or standard deviations.

- Roller Material

Level 1--303 stainless steel, Condition B

Level 2--440C stainless steel, hardened R_C 55-64

- Roller Finish

Level 1--Surface variation of 4 microinches (101 nm) AA

Level 2--Surface variation of 30 microinches (762 nm) AA

The third and fourth variables represented two components of possible types of product.

- Sample Material

Level 1--A6 tool steel, unhardened, or 6061-0 aluminum alloy

Level 2--A6 tool steel, air hardened, or 6061-T6 aluminum alloy

- Sample Finish

Level 1--Surface-treated with 500-grit Al_2O_3

Level 2--Surface-treated with 220-grit Al_2O_3

In the three-variable experiments, all four possible levels of sample material were studied, whereas all other experiments considered only two levels of sample material. The fifth variable represented polishing the dry lubricant after application to each possible product type.

- Polish Condition

Level 1--Polish with wood wheel or rag wheel

Level 2--Friction measured, as received

The coefficient of static friction was measured using the force gage and tilt methods, and the coefficient of dynamic friction was measured using the force gage, tilt, and oscillation methods. Seven experiments consisted of five variables each, five consisted of four variables each, and three consisted of three variables each. Five repeat measurements were made for each variable combination in each experiment.

In general, the different methods of measuring the friction coefficient (force gage, tilt, and oscillation) yielded different overall average values of the friction coefficient. The results of the experiments also showed that different variables affect the friction coefficient differently from experiment to experiment. An exception to this is roller finish, which is a significant factor in each dynamic friction experiment, with the 30 AA finish yielding a significantly lower friction coefficient than the 4 AA finish. However, roller finish was not a significant factor in any of the static friction experiments.

The results of the factorial design experiment (Table 5) showed that the interaction of the variables is very complex; an average coefficient cannot be accurately predicted for any friction system. The experiment did reveal, however, that a surface finish of 30 microinches AA results in lower dynamic friction coefficients than a 4-microinch AA finish for both 303 stainless steel, Condition B, and 440C stainless steel, R_c 55 to 64.

Table 5. Range of Friction Coefficients From All Factorial Design Experiments Using Electrofilm 4396

Test Description	Static Friction			Dynamic Friction		
	High	Medium	Low	High	Medium	Low
Force Gage--All Tests Combined	0.222	-0.178	-0.141	0.256	-0.201	0.148
Oscillation--All Tests Combined				0.258	-0.176	0.125
Tilt--All Tests Combined	0.198	0.170	0.144	0.261	-0.208	0.166
Overall Methods and Tests	0.210	0.174	0.125	0.258	-0.195	0.146

Additional Testing of Friction-Influencing Variables

Lubricant 4396, judged best in testing and statistical analyses, was used in a study of variables which might or might not reduce friction. Polishing, sliding velocity, load, compaction, and cure-temperature were evaluated.

Experiment 1 found that the friction coefficient is not significantly affected by polishing. Using aluminum and 303 stainless steel test bars, four degrees of polishing were evaluated: unpolished; lightly polished with a paper towel; medium polished with light pressure against a rotating cotton wheel; and highly polished with a slick, rotating wooden wheel.

Six readings were taken for each polish condition, material, and friction-coefficient measuring method. Dynamic friction coefficients were measured by oscillation, and static friction coefficients were measured by the force gage. The average and standard deviation for each set of six readings were calculated (Table 6) along with the results of the significant tests to determine whether differences among polish conditions existed.

There was no significant difference among polish conditions for the static friction coefficient; however, there was a significant difference for the dynamic friction coefficient. A separation-of-means test (Table 7) showed which polish conditions were significantly different: any averages underscored by the same line are not significantly different. For the aluminum samples, for example, medium polish yielded a significantly higher dynamic friction coefficient than no polish at all. For the 303 stainless steel samples, however, medium polish yielded a significantly higher dynamic friction coefficient than high polish. Based on the data, no other differences existed.

Table 6. Average and Standard Deviations for Different Polish Conditions

Material	Coefficient	Deviation	Unpolished	Light Polish	Medium Polish	High Polish	Significantly Different*
<u>Aluminum</u>	Dynamic	Average	0.1695	0.1910	0.2028	0.1882	Yes
		Standard	0.0085	0.0219	0.0225	0.008	Yes
	Static	Average	0.1240	0.1507	0.1473	0.1343	No
		Standard	0.0207	0.0210	0.0244	0.0065	No
<u>303 Stainless Steel</u>	Dynamic	Average	0.1845	0.1840	0.1935	0.1797	Yes
		Standard	0.0091	0.0066	0.0040	0.0037	No
	Static	Average	0.1007	0.1103	0.1192	0.1175	No
		Standard	0.0072	0.0220	0.0111	0.0142	No
*Significant with 95 percent confidence							

Table 7. Separation-Of-Means Analysis for All Polish Conditions

Aluminum Dynamic	Unpolished* <u>0.1695</u>	High 0.1882	Light <u>0.1910</u>	Medium 0.2028
303 Stainless Steel Dynamic	High <u>0.1797</u>	Light 0.1840	Unpolished <u>0.1845</u>	Medium 0.1935
*Polish conditions underscored by the same line are not significantly different.				

Although polishing did not reduce friction, it remains an important preoperative process. If properly done, it will remove surface contamination which otherwise would be bonded within the lubricating film. Such contamination as dirt or lint, if present on the surface, greatly increases the apparent static starting-friction coefficient.

Experiment 2 determined that sliding velocity had little effect on the friction coefficient (Figure 12) in the range of sliding velocity investigated. Test bars of A6 hardened tool steel were vapor-blasted with 500-grit Al_2O_3 and spray-coated with 4396. Friction testing employed 440C stainless steel rollers having R_C 55 to 60 finishes of 4 microinches (101 nm) and 30 microinches (762 nm) AA. Ambient conditions consisted of a temperature of 70°F (21°C) and a relative humidity of 20 percent. For lubricant 77S, the substrate material was 302 stainless steel; for lubricant 99A, it was brass. The oscillation method was used to determine the dynamic friction coefficient.

Experiment 3 showed that the friction coefficient decreased as the load increased (Table 8). Test bars of stainless steel and aluminum, coated with 4396, were tested in air at room temperature by oscillation with different loads.

Experiment 4 found that compacting the lubricant had no effect on its friction coefficient. Aluminum test bars with a 10- to 16-microinch (254 to 406 nm) AA surface finish were vapor-blasted with 500-grit Al_2O_3 and spray-coated with 4396. Using an autoclave engineer's research scale isostatic press, the lubricant was compacted at 30,000 psi (206 MPa) for 15 minutes (Figure 13). From the information shown in Tables 9 and 10, the pressed lubricant, as received, showed significantly lower static friction, but the difference vanished with only a slight disturbance of its surface.

Figure 12. Effect of Decrease in Sliding Velocity on Friction Coefficients

Experiment 5 found that cure-temperature had little effect on friction coefficients. In the first of two tests, dry 4396 lubricant was cured at 390 to 410°F (198 to 210°C), then sprayed onto stainless steel test bars and tested at a load of 192 to 760 grams. In the second test, dry lubricant cured at 325°F (163°C) was sprayed onto aluminum test bars and tested at a load of 67 to 635 grams. The following were the results of the tests.

- Using oscillation, the dynamic coefficients were 0.169 to 0.202 for Test 1, and 0.183 to 0.214 for Test 2.
- Using the gram gage method, the dynamic coefficients were 0.187 to 0.204 for Test 1, and 0.187 to 0.213 for Test 2. The static coefficients were 0.167 to 0.260 for Test 1, and 0.172 to 0.286 for Test 2. These gram gage values were taken at the lowest reading before the gage reading began to rise because of wearing of the lubricant film.

Preliminary Testing of Four Additional Lubricants

Following the study of the first six lubricants, preliminary tests were conducted on the additional four lubricants previously

Table 8. Decrease in Friction
Coefficients Caused by
Increase in Load

Load per Roller (g)	Bar Material	Roller Material	Friction Coefficient (Oscillation)
18	S/S	S/S	0.190
96	S/S	S/S	0.169
380	S/S	S/S	0.175
2270	S/S	S/S	0.171
38	S/S	S/S	0.214
96	AL	S/S	0.200
317	AL	S/S	0.183
1236	AL	S/S	0.173
2208	AL	S/S	0.171

mentioned. Although they were not as extensive, the tests indicated that Electrofilm 77S should be examined further (Table 11).

Testing of Bonded Dry-Film Lubricants, Part 2

Lubricants Tested

Electrofilm 77S is a coating which contains molybdenum disulfide (MoS_2) and graphite in a thermosetting resin that cures in 1 hour at 375°F (190.5°C). The manufacturer advertises that particular attention has been paid to new, high-adhesion thermosetting resins, micron-size solid lubricants, milling techniques, and new plasticizers and solvents.

Electrofilm 4396 is an organic, thermosetting, resin-bonded solid film lubricant containing approximately 90 percent molybdenum disulfide (MoS_2) and 10 percent graphite, by weight, cured at 350 to 375°F (176 to 191°C) for 1 hour. This lubricant is more frequently used at Bendix when the high cure-temperature is permissible.

Electrofilm 4396-S is similar to 4396, but the clay suspension agents are omitted.

Figure 13. Isostatic Compression of Film Lubricant

Coefficient-Of-Friction Testing

Samples for six test-bar assemblies (Figure 6) were spray-coated with each lubricant from 0.0002 to 0.0005 inch (5.08 to 12.7 μm) thick and cured to manufacturers' specifications. Before application of the lubricant, the samples, made from 302 stainless steel, Condition B, were vapor-blasted with 500-grit Al_2O_3 .

Two sets of 303 stainless steel, Condition B, rollers were used in the tests. One set had a smooth (8-16 rms) surface finish and the other set had a rough (16-32 rms) surface finish. The friction tests were performed according to the sequence shown in Table 12. Three of the six test-bar assemblies for each lubricant were subjected to a 1-3-5 test sequence and the remaining three test bars to a 2-4-6 sequence so that half of the samples were tested on smooth rollers and half were tested on rough rollers.

Lubricant 77S consistently performed better than either 4396 or 4396-S. The following paragraphs describe the tests that were used.

Static Friction, Force Transducer Method. The rollers remain stationary while the force transducer is advanced against the end of the test-bar assembly until the bar slips on the rollers. A force signature is recorded on a Visicorder, and the static friction coefficient is calculated from Equation 3, where P is the breakaway force and W is the weight of the bar (Tables 13 through 15). A typical recorded force signature is shown in Figure 14.

Dynamic Friction, Force Transducer Method. The test-bar assembly is placed on the rollers with one end touching the force transducer. The driven rollers force the bar against the transducer. The dynamic friction coefficient is calculated from Equation 2,

Text continued on page 38.

Table 9. Friction Data for Electrofilm 4396 Isostatically Pressed at 30,000 psi
(206 MPa)

Test	Film Description	Load (g)	Roller Material and Finish (microinches)*	Environment	Oscillation	Force	Gage
1	As Received	28.7	303 S/S, 16 AA	Room Temperature Air	---	0.196	0.096
2	After light wear	28.7	303 S/S, 16 AA	Room Temperature Air	---	0.220	0.169
3	Above sample cleaned	28.7	303 S/S, 16 AA	Room Temperature Air	---	0.225	0.186
4	Test No. 3 repeated Dynamic method only	28.7	303 S/S, 16 AA	Room Temperature Air	---	0.209	---
5	Rotating wood wheel Polished	36	440C 30 AA Rc 60	T = 73° R. H. = 31%	---	0.175	0.164
6	Rotating wood wheel Polished	36	440C 4 AA Rc 60	T = 73° R. H. = 31%	---	0.252	0.174
*1 microinch = 25.4 nm							

Table 10. Friction Data for Normal and Compressed Lubricant at Room Temperature

Film Description	Load (g)	Roller Material and Finish (microinches)*	Oscillation	Force Gage	Gram Gage
Unpressed Film As Received	28.9	303, 30 AA	0.160	0.184	0.217
Pressed Film As Received	28.7	303, 16 AA	---	0.196	0.096
Unpressed Film After Light Wear	28.7	303, 30 AA	0.162	0.180	0.181
Pressed Film After Light Wear	28.7	303, 16 AA	---	0.220	0.169
*1 microinch = 25.4 nm					

Table 11. Friction Coefficients for Four Additional Lubricants on 302 Stainless Steel Substrates

Lubricant	Film Condition	Rollers Material and Finish (Microinch)*	Friction Coefficients				
			Dynamic			Static	
			Force Gage	Tilt	Oscillation	Force Gage	Tilt
MLF-6	As Received	303 s/s 16 AA finish	0.282	0.232	0.160	0.181	0.184
MLF-6	Light Polish	303 s/s 16 AA finish	0.233	0.228	0.165	0.178	0.193
MLF-5	As Received	303 s/s 16 AA finish	0.208	0.220	0.161	0.191	0.216
MLF-5	Light Polish	303 s/s 16 AA finish	0.185	0.202	0.197	0.202	0.212
MLR-2	As Received	303 s/s 16 AA finish	0.158	0.183	0.207	0.197	0.239
MLR-2	Light Polish	303 s/s 16 AA finish	0.205	0.233	0.235	0.244	0.267
MLR-2	As Received	303 s/s 4 AA finish	0.191			0.218	
MLF-6	As Received	303 s/s 4 AA finish				0.221	
MLF-5	As Received	303 s/s 4 AA finish				0.181	
77-S n = 10	As Received	440C s/s Rc 60, 4 AA finish			0.133	0.160	
77-S n = 10	As Received	440C s/s Rc 60, 30 AA finish			0.128	0.161	
77-S	As Received	440C s/s Rc 60, 30 AA finish			0.132	0.173	
77-S	As Received	440C s/s Rc 60, 4 AA finish			0.146	0.179	
77-S	Light Polish	440C s/s Rc 60, 4 AA finish	0.143		0.171	0.188	
77-S	Light Polish	440C s/s Rc 60, 30 AA finish	0.164		0.149	0.156	
77-S	Light Polish	440C s/s Rc 60, 4 AA finish	0.181		0.134	0.191	
77-S	Light Polish	440C s/s Rc 60, 30 AA finish	0.163		0.148	0.161	

*1 microinch = 25.4 nm

Table 12. Sequence of Friction Tests

Test Number	Test Type	Test Method	Roller Surface
1	Static	Force Transducer	Rough
2	Static	Force Transducer	Smooth
3	Dynamic	Force Transducer	Rough
4	Dynamic	Force Transducer	Smooth
5	Dynamic	Oscillation	Rough
6	Dynamic	Oscillation	Smooth

where P is the force against the transducer and W is the weight of the bar (Tables 16 through 19). A typical Visicorder force signature is shown in Figure 15.

Dynamic Friction, Oscillation Method. With the rollers rotating in opposite directions, the bar is displaced from the central position and released. If the roller speed is sufficient to maintain slippage between the rollers and the bar, the bar will oscillate in a harmonic motion until damping decays its displacement from the center (Figure 3). The dynamic friction coefficient is calculated from Equation 1, where T is the time period of oscillation and g is the acceleration of gravity (386 in./sec² or 9.2 m/s²). The time period of oscillation was determined by using a photocell and Visicorder arrangement (Tables 20 through 22).

Wear Testing

After friction testing was completed, two test-bar assemblies for each lubricant were subjected to a 10-minute wear test. The friction was monitored continually during the test using the force-transducer method. After the first 10 minutes, the bar was moved slightly and the test was repeated. Range-of-wear performance graphs then were plotted for the lubricants for each test bar (Figures 16 and 17).

ACCOMPLISHMENTS

A tribometer was developed that yielded light-load friction-coefficient data and wear characteristics for dry-film lubricants. With the tribometer, the friction coefficients were determined by several methods; a correlation of the resulting data increased its reliability. The performance characteristics of the tester have been determined chiefly as it operates in the mode of flat

Table 13. Static-Friction Data

Lubricant	Roller Finish	Sample	Coefficient of Friction				
77-S	Smooth	1	0.109	0.128	0.123	0.120	0.140
		2	0.111	0.142	0.142	0.109	0.109
		3	0.111	0.136	0.125	0.136	0.123
4396	Smooth	7	0.134	0.128	0.128	0.117	0.125
		8	0.134	0.178	0.201	0.142	0.117
		9	0.128	0.123	0.150	0.145	0.142
4396-S	Smooth	13	0.106	0.111	0.139	0.134	0.117
		14	0.142	0.150	0.136	0.139	0.148
		15	0.145	0.136	0.145	0.136	0.139
77-S	Rough	4	0.125	0.120	0.128	0.097	0.134
		5	0.142	0.114	0.145	0.123	0.103
		6	0.123	0.117	0.106	0.103	0.123
4396	Rough	10	0.198	0.128	0.162	0.145	0.106
		11	0.201	0.240	0.217	0.120	0.189
		12	0.150	0.153	0.153	0.153	0.150
4396-S	Rough	16	0.139	0.114	0.142	0.134	0.111
		17	0.167	0.134	0.156	0.156	0.148
		18	0.167	0.167	0.148	0.164	0.195

Table 14. Means and Standard Deviations Calculated From Table 13

Lubricant	Roller Finish								
	Smooth			Rough			Smooth and Rough		
	$\bar{X}$		σ	$\bar{X}$		σ	$\bar{X}$		σ
77-S	0.124	(B)	0.013	0.120	(A)	0.014	0.122	(G)	0.013
4396	0.139	(D)	0.023	0.165	(F)	0.037	0.152	(I)	0.032
4396-S	0.135	(C)	0.013	0.149	(E)	0.022	0.142	(H)	0.019
All Lubes	0.133	(J)	0.018	0.145	(K)	0.032			

Table 15. Separation of Means for Static-Friction Data (Means underlined are statistically alike.)

A	B	C	D	E	F
<u>0.120</u>	<u>0.124</u>	<u>0.135</u>	<u>0.139</u>	<u>0.149</u>	<u>0.165</u>
<hr/>					
G	H	I			
<u>0.122</u>	<u>0.142</u>	<u>0.152</u>			
J	K				
<u>0.133</u>	<u>0.145</u>

Figure 14. Typical Static-Friction Visicorder Signature

bar against flat rollers. Journal and antifriction bearing experience is limited, and the design using the round rod and V-roller configuration also has not been fully evaluated.

Light-load (approximately 18 grams) friction-coefficient data for ten commercial bonded solid lubricants was determined. Extensive evaluation and comparison of six lubricants showed Electrofilm 4396 and 5396 to be statistically equal to each other and superior to the other four. A factorial design experiment using 4396 indicated a mating-part finish of 30 AA yielded lower friction coefficients than a 4 AA finish. The results indicated no clear effect of the other variables studied.

Table 16. Force-Transducer-Method Dynamic-Friction Data

Lubricant	Roller Finish	Sample	Friction Coefficient After Lapsed Running Time (Seconds)			
			1	3	10	20
77-S	Smooth	1	0.228	0.253	0.265	0.279
		2	0.214	0.226	0.240	0.253
		3	0.206	0.214	0.228	0.242
4396	Smooth	7	0.220	0.231	0.240	0.248
		8	0.240	0.234	0.245	0.251
		9	0.242	0.251	0.259	0.267
4396-S	Smooth	13	0.259	0.265	0.270	0.276
		14	0.256	0.259	0.270	0.281
		15	0.251	0.265	0.270	0.276
77-S	Rough	4	0.212	0.201	0.201	0.203
		5	0.195	0.195	0.192	0.201
		6	0.203	0.198	0.195	0.203
4396	Rough	10	0.220	0.209	0.189	0.184
		11	0.209	0.201	0.203	0.206
		12	0.212	0.201	0.178	0.184
4396-S	Rough	16	0.181	0.187	0.178	0.195
		17	0.195	0.192	0.198	0.201
		18	0.214	0.203	0.198	0.201

Table 17. Comparison by Lubricant, Time, and Roller Finish of Data Presented in Table 16

	1 Second		3 Seconds		10 Seconds		20 Seconds		1 Through 20	
	$\bar{X}$	σ	$\bar{X}$	σ	$\bar{X}$	σ	$\bar{X}$	σ	$\bar{X}$	σ
Smooth										
77-S	0.216	0.011	0.231	0.020	0.244	0.019	0.258	0.019	0.237	0.022
4396	0.234	0.012	0.239	0.011	0.248	0.010	0.255	0.010	0.244	0.013
4396-S	0.255	0.004	0.263	0.0035	0.270	0.000	0.278	0.003	0.267	0.009
All	0.235	0.010	0.244	0.013	0.254	0.012	0.264	0.0126	0.249	0.020
Rough										
77-S	0.203	0.0085	0.198	0.003	0.196	0.0046	0.202	0.001	0.200	0.009
4396	0.214	0.0057	0.204	0.0046	0.190	0.0125	0.191	0.0127	0.200	0.013
4396-S	0.197	0.0166	0.194	0.008	0.191	0.0115	0.199	0.0035	0.195	0.010
All	0.205	0.011	0.199	0.0057	0.192	0.010	0.198	0.0076	0.198	0.010

Table 18. Separation of Means for Smooth Roller Data Presented in Table 17

Identification	Time (Seconds)			
	1	3	10	20
77-S	0.216	0.231	0.244	0.258
4396	0.234	0.239	0.248	0.255
4396-S	0.255	0.263	0.270	0.277
All Lubes	0.235	0.244	0.254	0.264
At 1 Second	77-S 0.216	4396 0.234	4396-S 0.255	
At 3 Seconds	77-S 0.231	4396 0.239	4396-S 0.263	
At 10 Seconds	77-S 0.244	4396 0.248	4396-S 0.270	
At 20 Seconds	4396 0.255	77-S 0.258	4396-S 0.277	
Data for 1, 3, 10, 22, and 20 Seconds Com- bined	77-S 0.237	4396 0.244	4396-S 0.267	
All data combined 0.249.				

Four other bonded lubricants were tested after the original six. Preliminary friction coefficient data suggest that Electrofilm 77S may exhibit lower friction than the other nine lubricants studied: the overall coefficient of dynamic friction for 4396 is 0.189 and its static-friction coefficient is 0.178, while 77S exhibits a dynamic value of 0.149 and a static value of 0.171.

Additional experiments showed that load, film density, sliding velocity, and polishing had little effect on the friction coefficient of 4396 in the ranges applicable to miniature mechanisms.

Based upon the statistical analysis of the friction coefficients and wear measurements gathered during this study, the following conclusions were reached regarding the performance of lubricants 77S, 4396-S, and 4396.

The average dynamic-friction coefficient, using the oscillation method, was approximately 20 percent lower during the rough-roller tests. Lubricant 77S against the rough rollers exhibited approximately a 20-percent lower dynamic-friction coefficient than against the smooth rollers or than lubricants 4396 and 4396-S exhibited against either roller.

Text continued on page 46.

Table 19. Separation of Means for Rough Roller Data Presented in Table 17

Identification	Time (Seconds)			
	1	3	10	20
77-S	0.196	0.198	0.202	0.203
4396	0.190	0.191	0.203	0.217
4396-S	0.191	0.194	0.197	0.199
All Lubes	0.192	0.198	0.199	0.205
At 1 Second	4396-S 0.197	77-S 0.203	4396 0.217	
At 3 Seconds	4396-S 0.194	77-S 0.198	4396 0.203	
At 10 Seconds	4396 0.190	4396-S 0.191	77-S 0.196	
At 20 Seconds	4396 0.191	4396-S 0.199	77-S 0.202	
Data for 1, 3, 10, and 20 Combined	4396-S 0.195	77-S 0.200	4396 0.200	
All Data Combined	Rough Roller 0.198		Smooth Roller 0.249

Figure 15. Typical Dynamic-Friction Visicorder Signature Using a Force Transducer Against a Smooth Roller (Left) and a Rough Roller (Right)

Table 20. Oscillation-Method Dynamic-Friction Data

Lubricant	Roller Finish	Sample	Coefficient of Friction
77-S	Smooth	1	0.163
		2	0.169
		3	0.152
4396	Smooth	7	0.173
		8	0.188
		9	0.186
4396-S	Smooth	13	0.202
		14	0.206
		15	0.201
77-S	Rough	4	0.122
		5	0.138
		6	0.124
4396	Rough	10	0.191
		11	0.162
		12	0.202
4396-S	Rough	16	0.165
		17	0.143
		18	0.165

Table 21. Means and Standard Deviations Calculated From Table 20

Lubricant	Roller Finish								
	Smooth			Rough			Smooth and Rough		
	$\bar{X}$		σ	$\bar{X}$		σ	$\bar{X}$		σ
77-S	0.161	(B)	0.009	0.128	(A)	0.009	0.145	(G)	0.020
4396	0.182	(D)	0.008	0.169	(F)	0.019	0.175	(I)	0.015
4396-S	0.203	(C)	0.003	0.157	(E)	0.012	0.180	(H)	0.026
All Lubes	0.182	(J)	0.019	0.152	(K)	0.022			

Table 22. Separation of Means for Oscillation-Method Dynamic-Friction Data
(Underlined means are statistically alike.)

A	E	B	F	D	C
<u>0.128</u>	<u>0.157</u>	0.161	0.169	<u>0.182</u>	0.203
G	I	H			
<u>0.145</u>	<u>0.175</u>	0.180			
K	J				
<u>0.152</u>	<u>0.182</u>

Figure 16. Range of Wear for Tests 1 and 2

Figure 17. Range of Wear for All Three Tests

The use of rough rollers resulted in approximately a 20-percent lower dynamic-friction coefficient, using the force-transducer method, when data for all lubricants and all wear times were combined. The friction coefficients increased significantly as the time of wear increased for the smooth rollers, but they were similar for the rough rollers.

Results of the wear tests indicate that all lubricants degrade during operation with a faster increase in the friction coefficient occurring during early operation. This would not be true were the film broken to allow metal-to-metal contact of the mating parts.

The overall data seems to indicate that lubricant 77S is slightly superior to 4396 and 4396-S in light-load, moderate-wear applications. However, because lubricant 77S is thinner and slower-drying than the other two lubricants, it is prone to settling in the spray reservoir, running during application, and presenting handling problems until the solvent evaporates. Additional experience and equipment will be needed at this facility before 77S can be controlled satisfactorily for production use.

Appendix A

FRICTION REDUCTION EQUATIONS

OSCILLATING MODE, SLIDING FRICTION

From the free body force diagram for the bar displaced +X inches,

$$F_x = 0 = \mu F_l - \mu F_r - \frac{W}{g} \ddot{X} \quad (1)$$

$$F_y = 0 = F_l + F_r - W \quad (2)$$

$$\text{MOM}_{c.g.} = 0 = F_l (H + X) - F_r (H - X) - \mu (F_l - F_r) h \quad (3)$$

Substituting for F_l from (2) in (1) and solving for F_r ,

$$F_r = \frac{W}{2} \frac{(H + X - \mu h)}{(H - \mu h)} \quad (4)$$

Similarly,

$$F_l = \frac{W}{2} \frac{(H - X - \mu h)}{(H - \mu h)} \quad (5)$$

Substitution of (4) and (5) into (1) leads to:

$$\ddot{X} + \frac{\mu g}{H - \mu h} X = 0 \quad (6)$$

Equation (6) describes all possible motions of the bar. Correct solution uses the proper values of the friction coefficients and initial conditions of motion. No damping term appears and its effect has been assumed to be negligible.

If μ is assumed to be constant, the complete solution of (6) is

$$X = C_1 \sin wt + C_2 \cos wt \quad (7)$$

$$\text{where } w = \sqrt{\frac{\mu g}{H - \mu h}} \quad (8)$$

If the time of one complete oscillation is T seconds,

$$wT = 2\pi \quad (9)$$

Combining (8) and (9) yields

$$\mu = \frac{H}{\left[\frac{gT^2}{4\pi^2} + h \right]} \quad (10)$$

The error associated with neglect of damping may be evaluated as follows. Equation (10) is based on the assumption that the observed (damped) natural frequency is equal to the undamped natural frequency. The damped value is somewhat lower than the undamped value, depending on the degree of damping present. If T_d equals the damped period, and T_n equals the undamped period, then

$$T_d = \frac{2\pi}{w_d} \quad (11)$$

$$T_n = \frac{2\pi}{w_n} \quad (12)$$

Dividing (11) by (12) yields

$$\frac{T_d}{T_n} = \frac{w_n}{w_d} \quad (13)$$

From elementary vibration theory*

$$\frac{w_d}{w_n} = \sqrt{1 - \left(\frac{C}{C_c} \right)^2} \quad (14)$$

*Wylie, "Advanced Engineering Mathematics", p. 79.

where $\frac{C}{C_c}$ is the fraction of critical damping present. It is apparent that w_d is less than w_n and T_d is greater than T_n . The friction coefficient calculated from equation (10) will be somewhat lower than the true value; the extent of the error depends on the damping actually present.

For a system with linear damping,

$$\frac{C}{C_c} = \frac{\delta}{\delta^2 + 4\pi^2} \quad (15)$$

where δ is the logarithmic decrement, defined as the ratio of the amplitudes of any two successive excursions; that is,

$$\delta = \ln \frac{X_1}{X_2} = \ln \frac{X_2}{X_3} = \ln \frac{X_{n-1}}{X_n} \quad (16)$$

Let

$$\frac{X_1}{X_2} = \frac{X_2}{X_3} = \frac{X_{n-1}}{X_n} = K$$

$$\text{Then } \frac{X_1}{X_n} = K^n$$

$$\text{or } K = \left(\frac{X_1}{X_n} \right)^{\frac{1}{n}}$$

$$\text{and } \delta = \frac{1}{n} \ln \frac{X_1}{X_n} \quad (17)$$

Substituting (17) in (15) and in turn into (14),

$$\frac{w_d}{w_n} = 2\pi \sqrt{\frac{1}{\left(\frac{1}{n} \ln \frac{X_1}{X_n} \right)^2 + 4\pi^2}} \quad (18)$$

Combining (18) and (13) gives

$$\frac{T_d}{T_n} = \frac{1}{2\pi} \sqrt{4\pi^2 + \left(\frac{1}{n} \ln \frac{X_1}{X_n}\right)^2} \quad (19)$$

In equation (10), if h is much less than $\frac{gt^2}{4\pi}$ (which will be true for most test configurations) then

$$\mu \approx \frac{4\pi^2 H}{gT^2} \quad (20)$$

The correct value of T in equation (20) should be T_n , and

$$\frac{\mu \text{ correct}}{\mu \text{ approximate}} \approx \left(\frac{T_d}{T_n}\right)^2 \quad (21)$$

A correction factor, from (19), is

$$\frac{\mu \text{ correct}}{\mu \text{ approximate}} = 1 + \left(\frac{\frac{X_1}{X_n}}{2\pi n}\right)^2 \quad (22)$$

If the last observed oscillation has an amplitude of 0.05 inch, and the first an amplitude of 1 inch, and four oscillations occur, then

$$\frac{\mu \text{ correct}}{\mu \text{ approximate}} \approx 1 + \left(\frac{\ln 20}{2\pi \times 4}\right)^2 = 1.014$$

An error of less than 2 percent exists.

STATIONARY MODE WITH SLIDING FRICTION

Both rollers are driven in the same direction, and a force gage is used to restrain the motion and to measure the sum of the two friction forces.

For static force equilibrium,

$$F_x = 0 = \mu (F_l + F_r) - P \quad (1)$$

$$F_y = 0 = F_l + F_r - W \quad (2)$$

Combination of these two equations gives

$$\mu = \frac{P}{W} \quad (3)$$

The point of the application of P (the value of Z) is not important. Equation (3) is also valid in finding the static coefficient: the rollers are locked, and P is increased until slip occurs.

TILT METHOD

Consider the bar to be "tilted" to angle ϕ by the friction forces in contact with the rollers. Force equilibrium requires

$$F_x = 0 = \mu F_l + \mu F_r - W \sin \phi \quad (1)$$

$$F_y = 0 = F_l + F_r - W \cos \phi \quad (2)$$

$$\text{Moment} = 0 = HW \cos \phi - 2HF_r - Wh \sin \phi \quad (3)$$

about left end

From (3), $F_r = \frac{W}{2} \cos\phi - \frac{Wh \sin\phi}{2H}$ (4)

Substitute (4) in (2), $F_1 = \frac{W}{2} \cos\phi + \frac{Wh \sin\phi}{2H}$ (5)

Solving for F_1 from (1) and (2)

$$F_1 = W \cos\phi - F_r$$

$$F_1 = \frac{W \sin\phi}{\mu} - F_r$$

Solving for μ from (7)

$$\mu = \frac{W \sin\phi}{F_1 + F_r}$$

Substitution of (4) and (5) into (8) leads to

$$\mu = \tan\phi$$

OSCILLATING BAR WITH "STICK-SLIP" MOTION

If the roller speed is below a critical value when operating in the oscillating mode (rollers turning at the same speed but opposite directions), the velocity of the bar will become equal to the peripheral velocity of a roller, and the bar will "lock on" one roller if the static friction coefficient is higher than the sliding value. The resulting motion is oscillatory, but not of true sinusoidal nature. Under the assumption of a distinct, two-value friction coefficient (Figure A-1), the motion may be shown to consist of a combination of a constant velocity translation and a sinusoidal deceleration and acceleration toward the ends of bar travel. The oscillation may be visualized by plotting the velocity of the bar as a function of position

Figure A-2. The portion of the plot which is of straight lines (constant velocity) depends on the roller speed, with a decreasing constant velocity duration with increase in roller rpm. The critical roller rpm corresponds to the point at which a constant velocity vanishes.

Figure A-1. Friction Versus Velocity

Figure A-2. Velocity Versus Position

The condition for existence of "stick-slip" operation is

$$\frac{\rho \Omega}{c_o w} < 1$$

where $w = \sqrt{\frac{\mu_d g}{H - \mu_d h}}$

If "stick-slip" exists, the oscillation will continue without change in amplitude of motion with time, and the ratio of static and dynamic friction coefficients may be calculated from

$$\frac{\mu_s}{\mu_d} = \frac{\frac{H}{X_{\max}} + \sqrt{1 - \left(\frac{\rho \Omega}{X_{\max} \cdot w}\right)^2}}{\frac{H}{X_{\max}} - \sqrt{1 - \left(\frac{\rho \Omega}{X_{\max} \cdot w}\right)^2}}$$

Experience with the apparatus has demonstrated that "stick-slip" operation may exist at low roller speeds, but that this method has not proven to be reliable and repeatable. The actual value of the friction coefficient at very low sliding velocities is erratic, and while a continuing oscillation exists, the excursion may change substantially from cycle to cycle.

The derivation is as follows.

Force equilibrium requires that

$$F_x = 0 = \mu_l F_l - \mu_r F_r - \frac{W}{g} \ddot{X} \quad (1)$$

$$F_y = 0 = F_l + F_r - W \quad (2)$$

$$\text{Moment}_{c.g.} = 0 = F_l (H + X) - F_r (H - X) + (\mu_r F_r - \mu_l F_l) h \quad (3)$$

Complete solution of the above equations must recognize the proper values of the friction coefficients: if slip occurs, the friction coefficient will be constant, whereas if the bar is locked on a roller, static friction exists and is dependent on its position up to the limiting break-away value.

Eliminating F_1 from (1) and (2) and solving for F_r :

$$F_r = W \left[\frac{H + X - \mu_1 h}{2H - h(\mu_r + \mu_1)} \right] \quad (4)$$

Similarly,

$$F_1 = W \left[\frac{H - X - \mu_r h}{2H - h(\mu_r + \mu_1)} \right] \quad (5)$$

Substitution of (4) and (5) in (1) leads to

$$\ddot{X} + \left[\frac{(\mu_1 + \mu_r) g}{2H - h(\mu_1 + \mu_r)} \right] X = \frac{(\mu_1 - \mu_r) Hg}{2H - h(\mu_1 + \mu_r)} \quad (6)$$

Consider the bar to be moving in the +X direction, sliding on the right roller, and locked on the left roller. Set time t to equal 0 at the instant that the left wheel first slips. At $t = 0$

$$\dot{X} = \rho \Omega \quad (7)$$

for equation (6), $\ddot{X} = 0$ for $t \leq 0$, and

$$X|_{t=0} = \frac{(\mu_{1s} - \mu_{rd})}{(\mu_{1s} + \mu_{rd})} H \quad (8)$$

Sequential to $t = 0$, and until the bar has reversed direction of travel, moved toward the left, and locked on the right roller, sliding exists on both rollers with constant dynamic friction in play. For the second part of the motion $\mu_1 = \mu_r = \mu_d$. The complete solution of (6) is

$$X = C_1 \sin wt + C_2 \cos wt + \frac{(\mu_{1d} - \mu_{rd})}{(\mu_{1d} + \mu_{rd})} H \quad (9)$$

in which C_1 and C_2 depend upon (7) and (8) as initial conditions. From (7) and (9),

$$\dot{X}|_{t=0} = C_1 w \cos(o) - C_2 w \sin(o) = \rho \Omega$$

$$C_1 = \frac{\rho \Omega}{w} \quad (10)$$

from (8) and (9), taking $\mu_{ld} = \mu_{rd}$,

$$X_o = \frac{(\mu_{ls} - \mu_{rd})}{(\mu_{ls} + \mu_{rd})} H = C_1 \sin(o) + C_2 \cos(o) + 0$$

$$\text{and } C_2 = \left(\frac{(\mu_s - \mu_d)}{(\mu_s + \mu_d)} \right) H \quad (11)$$

Thus for $0 \leq t \leq t_1$, where t_1 is the time at which motion reverses direction,

$$X = \frac{\rho \Omega}{w} \sin wt + \frac{(\mu_s - \mu_d)}{(\mu_s + \mu_d)} H \cos wt \quad (12)$$

At $t = t_1$, the bar is at the extreme right position, and $\dot{X} = 0$ from (12)

$$\dot{X}|_{t=t_1} = 0 = \rho \Omega \cos wt_1 - \frac{(\mu_s - \mu_d)}{(\mu_s + \mu_d)} Hw \sin wt_1 \quad (13)$$

from which

$$\frac{\sin wt_1}{\cos wt_1} = \tan wt_1 = \frac{\rho \Omega (\mu_s + \mu_d)}{Hw (\mu_s - \mu_d)} \quad (14)$$

$$\text{from (14), } \sin wt_1 = \frac{\rho \Omega (\mu_s + \mu_d)}{\sqrt{\left[\rho \Omega (\mu_s + \mu_d) \right]^2 + \left[Hw (\mu_s - \mu_d) \right]^2}} \quad (15)$$

$$\text{and } \cos wt_1 = \frac{Hw (\mu_s - \mu_d)}{\sqrt{\left[\rho \Omega (\mu_s + \mu_d) \right]^2 + \left[Hw (\mu_s - \mu_d) \right]^2}} \quad (16)$$

Substituting (16) and (15) in (12)

$$X \text{ max} = \left[\begin{aligned} &+ \frac{(\rho \Omega)}{(w)} \frac{\rho \Omega (\mu_s + \mu_d)}{\sqrt{\left[\rho \Omega (\mu_s + \mu_d) \right]^2 + \left[Hw (\mu_s - \mu_d) \right]^2}} \\ &w + H \frac{(\mu_s - \mu_d)}{(\mu_s + \mu_d)} \frac{Hw (\mu_s - \mu_d)}{\sqrt{\left[\rho \Omega (\mu_s + \mu_d) \right]^2 + \left[Hw (\mu_s - \mu_d) \right]^2}} \end{aligned} \right] \quad (17)$$

solving the last equation for μ_s :

$$\mu_s = \mu_d \frac{\frac{H}{X \text{ max}} + \sqrt{1 - \left(\frac{(\rho \Omega)}{(X \text{ max } w)} \right)^2}}{\frac{H}{X \text{ max}} - \sqrt{1 - \frac{(\rho \Omega)^2}{(X \text{ max } w)^2}}} \quad (18)$$

$$\text{where } w = \sqrt{\frac{\mu_d g}{H - \mu_d h}} \quad (19)$$

The condition for existence of stick-slip motion is simply that stick will occur only if the velocity of the bar during the harmonic portion of motion becomes equal to the peripheral velocity of a roller. If stick were to occur,

$$\rho \Omega = \dot{X}$$

The equation of motion of a completely harmonic oscillation, with time started at $X = 0$, and of amplitude C_0 is:

$$X = C_0 \sin wt$$

The maximum velocity occurs at $X = 0$, and is

$$\dot{X}_{\text{max}} = C_0 w$$

Hence stick-slip will occur if

$$C_o W > \rho \Omega$$

or

$$\frac{\rho \Omega}{C_o W} < 1 \quad (20)$$

STATIONARY MODE: STATIC TO DYNAMIC RATIO

The ratio of the limiting value of the static friction coefficient to the sliding dynamic value may be determined by operating one roller at a sufficiently high speed to produce sliding on the bar, and rotating the other roller at a slow speed in a direction opposite to the high speed roller, so that static contact exists on the slowly rotated roller. The bar is placed in motion with the center of gravity displaced toward the slow speed roller. The unbalance of friction forces will move the bar toward the high speed roller until the limiting static and the dynamic friction forces are equal. The ratio of friction coefficients is given by:

$$\frac{\mu_s}{\mu_d} = \frac{H + X_{\max}}{H - X_{\max}}$$

This relationship is identical to the stick-slip mode relation for $W = 0$: that is, stick-slip without harmonic reversal of bar travel.

Experience with the apparatus concludes that this method of finding the ratio of friction coefficients is unsatisfactory because of the variable nature of static friction from point to point on the surface of the bar and rollers. The values found are not repeatable within desired limits.

The derivation is as follows.

Let the right roller run at a sufficiently high speed to assure slip against the bar, and rotate the left roller slowly in static contact with the bar. The bar is placed in motion with the center of gravity displaced toward the left (slow speed). For the roller, the bar moves to the right at a constant velocity

until the limiting static friction forces exist. The three equilibrium equations of Appendix IV apply, and the acceleration $X = 0$. Hence, from equation (6) of Appendix IV

$$(\mu_s + \mu_d) X_{\max} = (\mu_s - \mu_d) H$$

Expanding and collecting terms

$$\mu_s (H - X_{\max}) = \mu_o (H + X_{\max})$$

$$\text{or } \frac{\mu_s}{\mu_d} = \frac{H + X_{\max}}{H - X_{\max}}$$

Substituting (4) and (5) in (1) and solving for μ equals

$$\tan\phi = \frac{P}{W \cos\phi} \quad (6)$$

To measure dynamic friction, rotate both rollers in the same direction so as to hold the bar on the incline, without force P . To measure static friction, tilt the apparatus (with rollers locked) until slip occurs. In either case, $P = \mu$, and

$$\mu = \tan\phi$$

If $\phi = 0$ (No "TILT"), and P is applied,

$$\mu = \left| \frac{P}{W} \right|$$

ROTATING BEARING DYNAMIC FRICTION BY OSCILLATING MODE

The bearing friction coefficient is defined as the ratio of the tangential force at the shaft surface resisting rotation divided by the radial load applied to the bearing. If this definition is applied to both journal and rolling-element bearings, a direct comparison of bearings of various designs is possible.

If the roller weight is small in comparison with the weight of the bar, the dynamic bearing friction coefficient may be calculated from:

$$f = \frac{H\rho}{r} \left[\frac{1 + \frac{2gJ}{W\rho^2}}{\frac{gT^2}{4\pi^2} + h} \right]$$

If $J \rightarrow 0$, this is identical to the sliding coefficient of the bar against rollers except multiplied by the ratio of the roller radius to the shaft radius, ρ/r .

The derivation is as follows.

The figures below are free-body force diagrams of the bar and the two rollers as they exist in dynamic oscillatory motion.

The bar remains in rolling contact with the rollers at all times, and sliding occurs at bearing radius r at all times.

Dynamic force equilibrium for the bar requires

$$F_x = 0 = \mu_l F_l - \mu_r F_r - \frac{W}{g} \ddot{X} \quad (1)$$

$$\Sigma F_y = 0 = F_l + F_r - W \quad (2)$$

$$\text{Moment} = 0 = -\mu_l F_l h + \mu_r F_r h + F_l (H + X) - F_r (H - X) \quad (3)$$

For the left roller:

$$\text{Moment} = 0 = -\mu_l F_l \rho - J \ddot{\Omega} + f_l F_l r \quad (4)$$

about 0

For the right roller:

$$\text{Moments} = 0 = \mu_r F_r \rho - J \ddot{\Omega} - f_r F_r r \quad (5)$$

about 0

Since the bar does not slip on the rollers,

$$\rho \Omega = \dot{X}$$

$$\text{and } \rho \ddot{\Omega} = \ddot{X} \quad (6)$$

The above equations may be solved to find the bearing coefficient of friction, f .

From equations (4) and (6):

$$(\mu_l F_l) = -\frac{J \ddot{X}}{\rho^2} + \frac{f_l r}{\rho} F_l \quad (7)$$

From equations (5) and (6):

$$(\mu_r F_r) = +\frac{J \ddot{X}}{\rho^2} + \frac{f_r r}{\rho} F_r \quad (8)$$

Substituting for F_r from equation (2) into (8):

$$(\mu_r F_r) = + \frac{J\ddot{X}}{\rho} + \frac{f_r r}{\rho} (W - F_1) \quad (9)$$

Substitute equation (7), (9) and (2) into (3)

$$\left[\frac{J\ddot{X}}{\rho} - \frac{f_r r}{\rho} F_1 \right] h + \left[\frac{J\ddot{X}}{\rho} + \frac{f_r r}{\rho} (W - F_1) \right] h + F_1 (H + X) - \quad (10)$$

$$(W - F_1) (H - X) = 0$$

Solving equation (10) for F_1 :

$$F_1 = \frac{-\frac{2Jh}{\rho} \ddot{X} - \frac{f_r r h}{\rho} W + WH - WX}{2H - \frac{(f_1 + f_r) r h}{\rho}} \quad (11)$$

Substituting equation (11) into (7) gives:

$$(\mu_1 F_1) = - \frac{J\ddot{X}}{\rho} + \frac{f_r r}{\rho} \left[\frac{-\frac{2Jh}{\rho} \ddot{X} - \frac{f_r r h}{\rho} W + WH - WX}{2H - \frac{(f_1 + f_r) r h}{\rho}} \right] \quad (12)$$

Substitution of equation (11) into (9) leads to :

$$(\mu_f F_r) = + \frac{J\ddot{X}}{\rho} + \frac{f_r r}{\rho} \left[\frac{\frac{2Jh}{\rho} \ddot{X} - \frac{f_r r h}{\rho} W + WH + WX}{2H - \frac{(f_1 + f_r) r h}{\rho}} \right] \quad (13)$$

Combination of equations (12), (13), and (1) yields:

$$\left[\begin{array}{l} + \frac{-J\ddot{X}}{\rho^2} + \frac{f_l r}{\rho} \left[\frac{-\frac{2Jh}{\rho^2} \ddot{X} - \frac{f_r r h}{\rho} W + WH - WX}{2H - \frac{(f_l + f_r) r h}{\rho}} \right] \\ - \frac{J\ddot{X}}{\rho^2} + \frac{f_r r}{\rho} \left[\frac{+\frac{2Jh}{\rho^2} \ddot{X} - \frac{f_l r h}{\rho} W + WH - WX}{2H - \frac{(f_l + f_r) r h}{\rho}} \right] \\ - \frac{W}{g} \ddot{X} \end{array} \right] = 0 \quad (14)$$

Simplified, the last equation becomes

$$\ddot{X} + \frac{(f_l + f_r) r \rho g W}{4gJH + 2W\rho^2 H - W\rho r h (f_l + f_r)} X = \frac{(f_l - f_r) r \rho g WH}{4gJH + 2W\rho^2 H - W\rho r h (f_l + f_r)} \quad (15)$$

It follows that the circular frequency W of the motion is

$$W = \sqrt{\frac{(f_l + f_r) r \rho g W}{4gJH + 2W\rho^2 H - W\rho r h (f_l + f_r)}} \quad (16)$$

If the period of one cycle is T seconds

$$WT = 2\pi \quad (17)$$

$$\text{or } W = \frac{2\pi}{T}$$

If all bearings are alike and operate dynamically,

$$f_l = f_r = f_d$$

Equation (16) reduces to

$$w = \sqrt{\frac{fr\rho \cdot gW}{2gJH + W \cdot \rho^2 H - W\rho r \cdot hf}} \quad (18)$$

Setting equation (17) to equal (18), and solving for f,

$$f = \frac{2H\rho}{r} \left[\frac{1 + \frac{2gJ}{W\rho^2}}{\frac{gT^2}{2\pi^2} + 2h} \right] \quad (19)$$

ROTATING BEARING DYNAMIC FRICTION: STATIONARY MODE

If both roller shafts are rotated in the same direction, and the bar is held in a central position by force P as shown above, the rotating friction coefficient of the bearings may be calculated from

$$f = \sqrt{\frac{1}{\frac{r^2}{\rho^2} \left[\left(1 + \frac{2WH}{WH - PZ} \right)^2 \frac{W^2}{P^2} + 1 \right] - 1}}$$

A bar roller combination of high static friction coefficient must be used in this mode to avoid slip at the bar roller interface.

ROTATING BEARING STATIC FRICTION: STATIONARY MODE

Both shafts are locked to prevent rotation, and a force gage is used to apply force to one end of the bar with increasing force until motion is started. The static friction coefficient is calculated by the same relationship as for the corresponding dynamic case.

"TILT METHOD" OF BEARING FRICTION MEASUREMENT

Consider the left roller as it exists in equilibrium contact with the "Tilted" bar.

The rotating friction coefficient of the bearings may be calculated from

$$f = \sqrt{\frac{1}{\frac{r^2 \left(w + \frac{W}{2} + \frac{Wh}{2H} \tan^2 \phi \right)}{\rho^2 \tan^2 \phi \left(\frac{W}{2} \cos \phi + \frac{Wh}{2H} \sin \phi \right)} - 1}}$$

BEARING FRICTION: STATIONARY MODE, STATIC TO DYNAMIC RATIO

The ratio of the "starting" to the "running" friction coefficients may be found by rotating one shaft at a substantially higher speed than the other shaft. The directions of rotation are opposite. The low-speed shaft must

turn slowly enough to avoid relative motion between the shaft and its roller so that this bearing does not turn. The bar is placed in motion with the center of gravity of the bar displaced toward the slow speed shaft. The unbalance of friction forces will cause the bar to move at constant velocity toward the high speed shaft until the limiting static and the dynamic forces are equal. If the roller weight is small in comparison with the bar, the ratio of the "starting" to the "running" friction coefficients may be calculated from

$$\frac{f_s}{f_d} = \frac{H + X_{\max}}{H - X_{\max}}$$

where $X_{\max}$ is the maximum displaced position of the center of gravity, observed at the instant of initial rotation of the static bearings.

SUMMARY OF FRICTION FORMULAS

	Bar Against Rollers	Bearing Inside Rollers
A. Dynamic Oscillation	$\mu_d = \frac{H}{\frac{gT^2}{4\pi^2} + h}$	$f_d = \frac{2\rho H}{r} \left[\frac{1 + \frac{2gJ}{W\rho^2}}{\frac{gT^2}{2\pi^2} + 2h} \right]$
B. Force Measurement	$\mu = \frac{P}{W}$	$f = \sqrt{\frac{1}{\frac{r^2}{\rho^2} \left[\left(1 + \frac{2wH}{WH - PZ} \right)^2 \frac{W^2}{P^2} + 1 \right]} - 1}$
C. Tilted Position	$\mu = \tan\phi$	$f = \sqrt{\frac{1}{\frac{r^2}{\rho^2} \left(w + \frac{W}{2} + \frac{Wh}{2} \tan^2\phi \right) \tan^2\phi \left(\frac{W}{2} \cos\phi + \frac{Wh}{2H} \sin\phi \right)} - 1}$

NOTATIONS AND SUBSCRIPTS USED

Notations

- C = A constant
c = Damping constant

F = Radial contact force, bar against roller (lb)
 f = Bearing friction coefficient
 g = Gravitational acceleration (386 in/sec²)
 h = Height of bar center of gravity above roller surface (in)
 H = Half of center distance between rollers (in)
 J = Mass moment of inertia of one roller (lb/in/sec²)
 K = A constant
 n = Number of oscillations
 P = Friction force (lb)
 r = Bearing bore radius (in)
 T = Time period of one oscillation (sec)
 t = Time (sec)
 W = Weight of bar (lb)
 w = Weight of one roller (lb)
 X = Coordinate location of bar center of gravity in direction of travel
 y = Coordinate normal to x axis
 μ = Coefficient of friction, bar against roller
 ω = Circular frequency of oscillation
 Ω = Angular velocity of roller (radians/sec)
 ρ = Roller radius (in)

Subscripts

s = Static condition
 d = Dynamic condition
 l = Left roller
 r = Right roller
 n = Undamped valve
 d = Damped valve

Appendix B

PROPOSED THEORIES ABOUT FACTORS INFLUENCING THE FRICTION COEFFICIENT

These theories have been advanced to explain the nature of friction.

The following consists of excerpts taken from a publication by Ernest Rabinowicz entitled Friction and Wear of Solids.

"If two solids are pressed in contact by a load L , they will touch over a few small regions, called junctions, and there will be sizeable separation over the rest of the apparent area of contact A . The total area of the junctions is real area of contact A_r and, on the assumption the plastic deformation has occurred at the junctions, it may readily be shown that the value of A_r is given by the equation

$$L = A_r \cdot p \quad (1)$$

where p is the hardness of the softer of the two surfaces.

"The problem we shall now discuss is this. What happens when we apply a tangential force F to the surfaces, so that one surface slides over the others? Looking at one junction in detail, it is clear that when a shear force is applied slip will generally occur along the interface, since this is the surface of smallest cross-section. If the shear strength of the interface is τ_s we then have

$$F = A_r \cdot \tau_s \quad (2)$$

For the friction coefficient f , we have

$$f = \frac{F}{L} = \frac{A_r \cdot \tau_s}{A_r \cdot p} = \frac{\tau_s}{p} \quad (3)$$

"This equation states that the friction coefficient depends only on the shear strength of the interface between the contacting surfaces, and the hardness of the softer surface. In typical sliding situations, both the shear strength and the hardness are material constants, and then the friction coefficient too is virtually a constant, being nearly independent of engineering parameters such as load, apparent area of contact, sliding velocity, and surface roughness.

"The above discussion assumes that if the interface consists of a film lubricant that the interface does not wear away allowing some base material contact.

"Our derivation of Equation 3 and our discussion of it has been quite general, applying both to clean surfaces and to surfaces covered by thin films of lubricant. Now, however, some way must be found of specifying the magnitude of τ_s and this involves making some assumptions about what happens at the interface.

FRICITION OF CLEAN SURFACES

"According to the widely accepted adhesion theory (Bowden and Tabos, 1950) we assume that when clean surfaces are pressed together, strong bonds are formed between them, so that the interface becomes as strong as the weaker material. In that case τ_s , the shear strength of the interface may be taken to be equal to S_m , the plastic shear strength of the weaker material. Accordingly, Equation 3 becomes

$$f_m = \frac{S_m}{P} \quad (4)$$

where f_m represents the friction coefficient of the clean materials.

"It may be shown (for example in McClintock and Argon, 1966) that S_m is equal to $\delta y/2$ where δy is the yield strength of the material in uniaxial tension, while p is equal to $3 \delta y$ approximately. Hence, the ratio S_m/P should have a constant value of 0.167 for all material--soft and hard, metals, oxides, plastics, and so forth. To some extent this is born out in practice. The friction coefficients of a wide variety of materials is indeed relatively constant. However, the value of the friction coefficient is found to be about 0.3 or 0.4, about a factor of two greater than predicted by theory. This discrepancy between theory and experiment may be explained by the supposition that, as sliding starts, the real area of contact A_r grows and its value rises by a factor of 2 above the value given in equation 1; accordingly the friction coefficient is higher than it should be by about a factor of 2.

FRICITION OF SURFACES COVERED BY A SOLID FILM LUBRICANT

"In this case, there is at the interface a soft film, which may be a metal, an inorganic material (for example, graphite or molybdenum disulfide) or an organic material. The material at the interface has a low value of shear strength, which we shall call S film, and lowers the friction correspondingly.

$$f_{\text{film}} = \frac{S_{\text{film}}}{P} \quad (5)$$

The lubricant film at the interface is under high hydrostatic stress, and hence its shear strength is not the same as the strength measured under normal circumstances."

The following consists of excerpts taken from a publication by Arthur J. Haltner, of General Electric Company, Space Science Laboratory, Philadelphia, entitled The Properties of Solid Lubricants.

"In considering the properties of a solid to be used as a lubricant, the total environment must be included. The bulk mechanical and chemical properties of the metal substrate are important as well as the properties of any surface films which arise from contamination or chemical reaction. Often the nature of the gaseous environment can be crucial in determining the sliding behavior. In normal solid film lubrication, a number of complex interfacial areas are involved compared to the idealized case in which a crystal of solid lubricant slides against another similar solid in a perfect vacuum.

"The literature on the properties of solid lubricant materials is increasing rapidly, testifying to active research interest in this field. Much of the data is confusing and even apparently contradictory. However, it appears that the subject of solid lubricant properties can best be developed from a consideration of structural chemistry and the various forces between atoms and molecules that lead them to adhere to certain stable arrangements in space. The properties of interest may quite generally be divided into two categories. Bulk properties (density, shear strength, modules, thermal and electrical conductivity) are closely allied to crystal structure and the nature of the forces acting in the solid. Chemical properties can be related to composition and phenomena at the solid interface.

"According to the shear strength theory, the frictional force F should be given by

$$F = \tau_s A_r \quad (1)$$

where A_r is the real area of contact and τ_s is the shear strength. Since the coefficient of friction f is given by

$$f = \frac{F}{W} \quad (2)$$

$$f = \tau_s \frac{A_r}{W} \quad (3)$$

$$f = \tau_s P_m^{-1} \quad (4)$$

where P_m is the pressure and will be associated here with a yield pressure.

"It is obvious that if τ_s does not increase with pressure or at least varies with a power of pressure lower than the first power, the friction coefficient of such a film will decrease as the load increases. Plotting high pressure shear curves for MoS_2 and WS_2 shows that the shearing stress versus pressure curves for MoS_2 levels off at high pressures. This means that the shearing stress becomes independent of pressure at sufficiently high pressures.

Many solid lubricant materials are available possessing a wide range of properties. Available materials to the engineer and designer will probably increase at a rapid rate in the immediate future since work in this field is accelerating. It appears certain that the impact of solid lubricants on technology will be limited by our ingenuity in taking advantage of their unique properties."

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