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NASA Technical Memorandum 81601

Evaluation of Boundary Lubricants Using Steady-State Wear and Friction

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Prepared for the
Thirty-sixth Annual Meeting of the American Society of Lubricant Engineers
Pittsburgh, Pennsylvania, May 11-14, 1981

NASA

EVALUATION OF BOUNDARY LUBRICANTS USING STEADY-STATE WEAR AND FRICTION

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E-716

ABSTRACT

A friction and wear study was made at 20° C to establish operating limits and procedures for obtaining improved reproducibility and reliability in boundary lubrication testing. Ester base and C-ether base fluids were used to lubricate a pure iron rider in sliding contact with a rotating M-50 steel disk in a pin-on-disk apparatus. Results of a parametric study with varying loads and speeds showed that satisfactory test conditions for studying the friction and wear characteristics in the boundary lubrication regime with this test device were found to be 1-kilogram load; 7 to 9 meters-per-minute (50 rpm) surface speed; dry air test atmosphere (<100 ppm H_2O); and use of a time stepwise procedure for measuring wear. Highly reproducible steady-state wear rates resulted from the two fluid studies which had a linearity of about 99 percent after initially higher wear rates and friction coefficients during run-in periods of 20 to 40 minutes.

INTRODUCTION

A number of boundary lubrication studies have been made of candidate hydraulic fluids and lubricants for use in advanced aircraft and re-entry vehicles (1)-(3). Friction and wear measurements were made with various fluids using a pin-on-disk sliding friction apparatus as a screening

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device. Generally, the experimental conditions for these earlier tests were standardized as follows: Both test components (i.e., rider and disk) were hardened CVM M-50 steel (Rockwell C62 to 64), a normal load of 1 kilogram was applied with a deadweight, and a 17-meters-per-minute (100 rpm) disk surface speed was maintained for a 25-minute test duration. Test run variables included comparative runs in dry air (<100 ppm H_2O) and moist air (relative humidity of 50 percent at 25° C) and runs at various disk temperatures between room temperature and 300° C (572° F).

Results of these earlier lubrication studies indicated very low wear rates with the operating mode applied. C-ether base and formulated fluids at a temperature of 25° C (77° F) in a dry air atmosphere gave wear rates in the range of 3.0×10^{-15} to 4.5×10^{-15} cubic meter per meter of sliding distance. Contributing to these low wear rates were (1) the fact that both rubbing components are made of the same wear-resistant tool steel, and (2) the probability that operating conditions resulted in "mixed" lubrication (especially at the lower temperatures), as described by Dowson in Ref. (4), where lower wear would be expected. Variations in test parameters from test to test and their effect on wear rate are accentuated in low-wear situations and thereby lessen the chances of obtaining reproducible results. Furthermore, the unpredictability of higher run-in wear during the first 10 to 20 minutes of these test runs also adversely affected reproducibility. With all prior testing having been done over a single, fixed test duration, the so-called rider wear rates reported were simply wear volume measurements. Equilibrium or "steady-state" wear rates and friction were not determined. The idea of a run-in, high initial wear period, followed by a fairly constant wear rate period is common and is cited frequently in text books as well as papers on wear (5)-(8). However, quantification of these wear rates has been needed, especially in the practice of using steady-state values for repetitive evaluation of lubricating fluids.

Another significant result of these previous lubrication studies was that in general, lower wear rates were observed when fluids were tested in moist air as compared with a dry air atmosphere. In selecting the test atmosphere to use for future studies of this type it would seem appropriate to consider dry air, which generally is the more severe condition (1).

The objectives of this investigation were (1) to establish operating test parameters and procedures for obtaining more reproducible wear rates and friction coefficients for boundary lubrication studies using liquid lubricants and (2) to determine wear rate and friction coefficient for "steady-state" operation. Pure annealed iron (99.99 percent iron) riders were used instead of M-50 tool steel to achieve larger and more easily measured wear scars, the iron being much softer than M-50 tool steel disks. Surface sliding velocities and loads were selected which avoided elastohydrodynamic and "mixed" lubrication regions.

Other experimental conditions included: 1/2- and 1-kilogram loads (initial Hertz stresses of 0.8×10^9 and 1×10^9 N/m^2, respectively); sliding velocities of 3.6- to 18.2- meters per minute; time sequences for each test run of 1 to 250 minutes; and use of an ester base fluid (Specification MIL-L-27502 candidate base) and a C-ether base fluid as experimental lubricants.

APPARATUS

The sliding friction and wear apparatus used in this study is shown in Fig. 1. The test specimens were contained inside a stainless-steel chamber, and the chamber atmosphere was controlled with respect to moisture content. A stationary 0.476- centimeter-radius, hemispherically tipped rider was placed in sliding contact with a rotating 6.35- centimeter-diameter disk. Rider holders of three different lengths were used to permit three parallel (or concentric) wear tracks to be run on a single disk, thereby eliminating the need to refinish disk surfaces after each run. Sliding velocities ranged from about 3.6- to 18.2- meters per minute as disk rotational speeds were maintained at 25, 50, 75, and 100 rpm when using each of the three rider holders.

A normal load of 1/2- or 1-kilogram was applied with a deadweight. Disks were made of CVM M-50 tool steel and heat treated to a hardness of Rockwell C62 to 64. Riders were made of pure iron (99.99 percent iron) and were annealed to a DPH (diamond pyramid hardness number) hardness of 70 to 92 kilograms per square millimeter as measured on an Eberbach microhardness tester at a 150-gram load.

The disk was partially submerged in a polyimide cup containing the test lubricant, and bulk lubricant temperature was measured with a thermocouple. Frictional force was measured with a strain gage and was recorded on an X-Y chart recorder.

The test chamber atmosphere used in this study was dry air ($<$100 ppm H_2O) obtained by drying and filtering service air. The moisture concentration was monitored by a moisture analyzer with an accuracy of ±10 parts per million.

PROCEDURE

Disks were ground and lapped to a surface finish of 10×10^{-8} to 20×10^{-8} meter (4 to 8 μin.), rms. Rider tips were machined and polished to a surface finish of 5×10^{-8} to 10×10^{-8} meter (2 to 4 μin.), rms. Specimens were scrubbed with a paste of levigated alumina and water, rinsed with tap and distilled water, then placed in a desiccator.

Test lubricants were degassed at approximately 150° C (302° F) at 2.7×10^2 newtons-per-square meter pressure for 1 hour. Measurements using the Karl Fisher technique indicate that this degassing procedure reduces dissolved water content in the test fluids to less than 20 ppm.

The specimens were assembled, and approximately 3×10^{-5} cubic meter (30 ml) of lubricant were placed in the lubricant cup. The test chamber (3.7×10^{-3} m^3 volume) was purged with the dry air test atmosphere for 10 minutes at a flow rate in excess of 5×10^{-2} cubic meter per hour. The rider was then loaded against the disk with the deadweight, and the test run period was started at that time. Dry air atmosphere flow rate was reduced to 3.5×10^{-2} cubic meter per hour, and a 6.9×10^3-newtons-per-square meter (1-psig) pressure was maintained in the test chamber.

Rider wear scar diameters were measured and recorded after sliding experiment intervals that varied from 1 to 250 minutes. Most wear measurements were made after 1, 3, 10, 20, 40, 70, and 130 minutes. Disk wear with pure iron riders was so small that it was immeasurable. Operating times were selected for each disk speed so that total sliding distance for all runs was in the range of about 900 to 1200 meters. Frictional force was continuously recorded during the experiments.

EXPERIMENTAL LUBRICANTS

The experimental fluids used in this study were an ester-base fluid and a C-ether base fluid. Some typical properties of the test fluids appear in Table 1. The ester fluid is a special synthesized unformulated mixture of the following: Hindered polyol esters of straight chain fatty acids (C_5 to C_{10}), a polyester, and dipentaerythritol esters. This ester blend has been used as a base stock for a specification MIL-L-27502 candidate lubricant (-40° to 465° F) and is described in Refs. (9)-(10). The C-ether base fluid used in this study was originally described in Ref. (11) and more recently in Ref. (12). This fluid is a blend of three-ring and four-ring components, which are structurally similar to the polyphenyl ethers, and contains an antifoam additive.

RESULTS AND DISCUSSION

Lubrication Regime

A series of wear tests at 20° C was run with the ester base fluid at four disk speeds from 25 to 100 rpm and at loads of 1/2 and 1 kilogram. The 25 rpm rate is about the lowest controllable speed for this test rig, and the 100 rpm rate is the speed used in previous studies (1)-(3) with this apparatus. Although generally a 1 kilogram load had been employed previously, the 1/2-kilogram load was used to determine the effects of a lower load and lower speeds. Since practically all bearing failures (regardless of type) involve boundary lubrication to some degree (13), it is important that the test conditions be chosen for that regime.

Results of those wear tests, plotted in Fig. 2, show consistent linear relations with sliding distances in the range of 150 to 1,200 meters at constant loads and disk speeds. Wear rates were calculated as the slopes of these lines using the linear regression fit formula (least squares estimates) as described in Refs. (14)-(15). High wear rate values were observed during these wear tests before the inception of the constant rates shown in Fig. 2. A discussion of this initial, run-in wear phenomenon will be given in a later section of this report.

A presentation of these computed wear data is given in Table 2 along with the correlation coefficients R for each set of operating conditions. The correlation coefficient is the approximate measure of the degree of fit for a linear relation of the two variables (14). Thus, the consistently high absolute values of R for the reported variables indicate a high degree of association between them.

The square of the correlation coefficient R^2 is an even more significant parameter for determining these variable relations. It measures the proportion of the total variation between the two parameters accounted for by the regression equation (16). Thus, the data given in Fig. 2 and Table 2 indicate that between 98.9 and 99.8 percent of the relation between rider wear and sliding distance can be shown as a straight line function.

Good reproducibility of these wear rate results is shown in two sets of five test runs where the operating conditions were the same (1-kg load, 20° C disk temperature), while the disk speeds were held at 50 and 100 rpm, respectively. Standard deviations from their mean values indicate a variance of less than 9 percent.

A presentation of these wear rates as a function of disk speed (rubbing surface velocity) at the two loads is given in Fig. 3. We conclude from these data that wear tests were being performed in the boundary lubrication regime in the lower disk speed range of 25 to 50 rpm and at the higher load value of 1 kilogram since the wear rates at these speeds are essentially at a constant value (~23.5×10^{-14} m^3/m). This conclusion is based on an earlier determination (4) that for practical purposes wear rate in the region of boundary lubrication is independent of the rubbing speed, which is the case here. For all other operating conditions shown in Fig. 3, wear rates decreased as surface velocities were increased. These would be the cases where film thicknesses would increase with speed increases as in the mixed lubrication regime (17).

The effect of load on wear rate is further evidence that operating conditions are within the boundary lubricating regime where higher loads resulted in increased wear rates. Although the data were limited to only the two loads under study, the results plotted in Fig. 3 do show that the greatest effect of load on wear rate occurred at a disk speed of 50 rpm. Previous boundary lubrication studies (4), (8), (18) had shown an increase in wear rate with increasing loads, although there does not seem to be any simple, general functional relation.

Additional evidence which verifies that the lower sliding speed range should be used for boundary lubrication studies with this system involves use of the dimensionless parameter Λ. This parameter was defined by Tallian, et al. (19), and has been used for predicting the degree of surface interactions. It is the ratio of the central or minimum film thickness to the composite surface roughness. Mathematically it is shown as

$$\Lambda = \frac{h_o}{\left(\sigma_1^2 + \sigma_2^2\right)^{1/2}}$$

where h_o is the central film thickness as calculated from the Dowson and Higginson equation of Ref. (20), and σ_1 and σ_2 are the roughness values of the mating surfaces. In this study, the two surfaces are the pure iron rider and the M-50 steel disk.

Calculated values of h_o and Λ for these ester base fluid wear tests are given in Table 3. The range in Λ values for each disk speed shown is due to the variation in disk surface finishes of 10×10^{-8} to 20×10^{-8} meters (4 to 8 μin.). Based on these data, lubrication is undoubtedly in the mixed regime at 75 and 100 rpm disk speeds where higher Λ numbers are in the range of 0.8 to 0.9; whereas, lubrication should be primarily boundary at 25 and 50 rpm speeds for Λ values between about 0.2 and 0.5. Czichos (21) reports that the change from a full EHD film to continuous asperity contact (as in boundary lubrication) occurs as Λ decreases from 2.5 down to 0.7. These experimental findings by Czichos are consistent with the theoretical predictions of Johnson, et. al (22). However, more recent work (23) has shown that discretion should be exercised in using Λ values to predict asperity interactions. This is due to the fact that as the central film thickness decreases to very low values, the micro-EHD effects become dominant; i.e., the local topographical geometry (asperities and depressions) determines the actual separation and stresses to a greater degree than the global geometry from which film thickness is usually calculated. This is especially true under sliding conditions.

Results of this parametric study are much as expected, in that lower speeds and higher loads tend to drive most systems towards boundary lubrication (13). All studies with this apparatus using the C-ether base fluid were at a load of 1 kilogram and at a disk speed of 50 rpm to insure that boundary lubrication conditions are realized.

Steady-State Wear Rate

A series of wear tests at 20° C was run with the C-ether base fluid at 1-kilogram load and 50 rpm disk speed. Time intervals for measuring rider wear scars were at 10, 50, 90 and 130 minutes. Plots of rider wear volume versus sliding distance are shown in Fig. 4 for each of the three different length rider holders, which resulted in sliding speeds of 7.1, 8.1, and 9.1 meters per minute. These wear-sliding distance curves are characteristic of the type described by Hirst and Lancaster (8) for relatively soft metals in contact with a hard metal where the essential feature is that the wear rate is initially high but later decreases to a lower constant value. The slopes of the linear relations of Fig. 4 in the range of 50 to 130 minutes are herein defined as the "steady-state wear rates" for these studies. These calculated wear rates were determined to be very constant and reproducible values of 8.3x10^{-15}, 8.6x10^{-15}, and 8.5x10^{-15} cubic meter per meter of sliding distance. These give a mean value of 8.45x10^{-15} and a standard deviation of ±0.021x10^{-15} cubic meter per meter, a variance of only ±2.5 percent. This is considered high reproducibility for sliding wear tests under these conditions. Using R^2 values, these variables have an estimated 98 percent straight line relation. As shown in Fig. 4, if wear rates were calculated from one 25-minute wear measurement (as in Ref. (1)), their values of 116x10^{-15}, 106x10^{-15}, and 83x10^{-15} cubic meter per meter (mean, 101.7x10^{-15}, standard deviation of ±16.9x10^{-15}, and variance of ±17 percent) would be over an order of magnitude higher than the steady-state rates, and also less reproducible.

Steady-State Coefficient of Friction

Figure 5 shows plots of the friction coefficients measured during the stepwise wear tests for C-ether base fluids described in the previous section. All three of these test runs showed the same consistent and repeatable friction pattern. Friction coefficients were initially high at 0.130 to 0.135 after 1 minute and then gradually decreased throughout the 10- to 35-minute run-in period to a constant or steady-state level of about 0.09 for the remainder of the 130-minute run. These constant friction values shall be defined as "steady-state friction coefficients" and are analogous to corresponding steady-state wear rates in that they occur during the same experimental time period.

These coefficient of friction results are further evidence that the system is operating within the boundary lubrication regime. It has been reported (4) that by using the Stribeck diagram (a plot of friction coefficient as a function of $\mu N/p$ where μ is the oil viscosity, N the speed, and p the normal load; see Refs. (24) and (25)) for friction coefficient data, it can be predicted that coefficient of friction should be at a steady value of about 0.1 for boundary lubrication where lower speeds and higher contact stress levels are maintained. This is the case at the chosen operating conditions with the system under study.

Run-in Period

As shown from rider wear results of Fig. 4, a run-in period of 20 to 30 minutes was needed before constant or steady-state wear rates were established. Rider wear volume measurements at three run times (50, 90 and 130 min) were sufficient to establish a reliable linear wear-sliding distance relation.

Likewise, friction coefficient results of Fig. 5 from the same test runs showed the ability to predict run-in periods independently of wear results. The determination of run-in periods from the two methods were in good agreement with each other.

Another reliable method for determining run-in periods and for enhancing the predictability of the wear phenomenon involves the plotting of unit contact stresses over the time period studied. An example of this is shown in Fig. 6 where unit contact stress data (normal load per wear-scar area) are plotted as a function of sliding distance for several ester base fluid test runs at 1/2 and 1 kilogram loads and a disk speed of 60 rpm. Contact stresses begin to taper off to levels of 3.0×10^6 to 2.5×10^6 N/m^2 (435 to 362 psi) at sliding distances of greater than 300 meters. This corresponds to an elapsed run-in time of about 35 minutes after initially high stress values were experienced ($>30 \times 10^6$ N/m^2 after 1 minute and 8×10^6 N/m^2 after 10 minutes).

Run-in Wear

Run-in wear is the initial high-rate wear that is experienced before the onset of the usually lower and linear "steady-state wear rate" phenomenon. Run-in wear behavior of some liquid lubricants in the boundary lubrication regime could be important if it is a significant fraction of the total wear that occurs during "lifetime" or prolonged operating times. Table 4 shows typical run-in wear and steady-state wear values for both test fluids. The sum of these two wear values is the total wear for the test run. This tabulation indicates that run-in wear during the 130-minute test period was significant for the C-ether base fluid (76 percent of the total wear) but was less significant for the ester base fluid (only 16 percent of total wear). Furthermore, it is shown that for more prolonged runs of 300 and 600 minutes, where the steady-state wear is extrapolated to more realistic running times, run-in wear becomes much less significant (37 percent for C-ethers and only 3.4 percent for esters).

The most reproducible comparisons of wear behavior appear to be of steady-state wear rate determinations where run-in wear is not the dominant factor. As a further refinement in defining lubricating ability of a fluid in this system, consideration could be given to including run-in wear, run--in wear rate, or total wear.

SUMMARY OF RESULTS

A friction and wear study was made at 20° C to establish some test operating limits and procedures to insure better reproducibility in the boundary lubrication regime. Ester base and C-ether base fluids were used in a pin-on-disk sliding friction apparatus (pure iron on CVM M-50 steel) in a dry air (<100 ppm H_2O) test atmosphere. Other conditions were 1/2- and 1-kilogram loads (initial Hertz stresses of $0.8x10^9$ and $1x10^9$ N/m^2), sliding velocities of 3.6 to 18.2 meters per minute, and test times from 1 to 250 minutes. Wear rate results were calculated from wear measurements by the linear regression formula and compared with those as obtained in previous studies where a single, fixed test duration (25 min) was used. The major results were the following:

1. Satisfactory test conditions for studying boundary lubrication friction and wear characteristics of C-ether base and ester base fluids in the apparatus used in this study were found to be pure iron rider on M-50 steel disk; dry test atmosphere (<100 ppm H_2O); 1-kilogram normal load; 50 rpm disk speed; and use of a time stepwise procedure for measuring wear with as few as three time intervals to establish reliable linear wear rates up to about 130 minutes (~1,000 meters of sliding distance). Higher speed or lower load produced mixed lubrication.

2. Steady-state wear rates appear to be the best single parameter for determining comparative lubrication behavior among fluids. Highly reproducible steady-state wear rates were determined for C-ether base ($\sim 8.5 \times 10^{-15}$ m^3/m) and ester base ($\sim 2.3 \times 10^{-13}$ m^3/m) fluids under boundary lubrication conditions. These wear-sliding distance relations were found to have 98 to 99.8 percent linearity following an initial 20 to 40 minute run-in period during which higher wear rates, friction coefficients, and unit contact stess levels prevailed. Run-in wear rates were more than an order of magnitude higher and less reproducible than the "steady-state" values.

3. Reproducible steady-state or equilibrium coefficients of friction were measured during the same experimental time periods as corresponding steady-state wear rates. Constant values of about 0.09 were recorded after the run-in period.

REFERENCES

1. Jones, William R., Jr., "Boundary Lubrication of Formulated C-Ethers in Air to 300° C," Lubr. Eng., 32, 10, pp 530-538 (1976).

2. Jones, William R., Jr. and Hady, William F., "Effect of Humidity and a Wettability Additive on Polyphenyl Ether Boundary Lubrication of Steel in Air and Nitrogen to 350° C," NASA TN D-6055 (1970).

3. Jones, William R., Jr. and Hady, William F., "Boundary Lubrication and Thermal Stability Studies with Five Liquid Lubricants in Nitrogen to 400° C," NASA TN D-6251 (1971).

4. Ling, F. F., Klaus, E. E., and Fein, R. S., Boundary Lubrication, an Appraisal of World Literature, American Society of Mechanical Engineers, New York (1969), pp 229-240.

5. Sarkar, A. D., Wear of Metals, Pergamon Press, Inc., New York (1976), pp 46-48.

6. Burwell, J. T., Jr., "Survey of Possible Wear Mechanisms," Wear, 1 pp 119-141 (1957-58).

7. Queener, C. A., Smith, T. C., and Mitchell, W. L., "Transient Wear of Machine Parts," Wear, 8, pp 391-400 (1965).

8. Hirst, W. and Lancaster, J. K., "Surface Film Formation and Metallic Wear," J. Appl. Phys., 27, 9, pp 1057-1065 (1956).

9. Clark, F. S., Miller, D. R., and Reid, S. L., "Development of a Gas Turbine Engine Oil for Bulk Oil Temperatures of -40° to 465° F, Part I," Report MRC-SL-477, Monsanto Research Corp., 1975. (AFML-TR-74-247, AD-A027068.)

10. Clark, F. S., Herber, J. F., and Reid, S. L., "Development of a Gas Turbine Engine Oil for Bulk Oil Temperatures of -40° to 465° F, Part II," Report MRC-SL-538-Pt. 2, Monsanto Research Corp., 1975. (AFML-TR-74-247-Pt. 2, AD-A027068.)

11. McHugh, Kenneth L. and Stark, Louis R., "Properties of a New Class of Polyaromatics for Use as High-Temperature Lubricants and Functional Fluids," ASLE Trans., 9, 1, pp 13-23 (1966).

12. Clark, Frank S., Green, Richard L., and Miller, David R., "Synthesis and Evaluation of C-Ether Formulations for Use as High Temperature Lubricants and Hydraulic Fluids," Report MRC-SL-460, Monsanto Research Corp., NASA CR-134643 (1974).

13. Bisson, Edmond E. and Anderson, William J., "Advanced Bearing Technology," NASA SP-38 (1964).

14. Bowker, Albert H. and Lieberman, Gerald J., Engineering Statistics, Prentice Hall, Inc., Englewood Cliffs (1959), p 273.

15. Freund, John E., Mathematical Statistics, Prentice Hall, Inc., Englewood Cliffs (1962).

16. Atherton, Murle J., "Some Common Errors Associated with Regression Analysis," Technical Paper 679-OP, The International Nickel Company, Inc. (1970).

17. Wedeven, Lavern D., "What is EHD?," Lubr. Eng., 31, 6, pp 291-296 (1975).

18. Burwell, J. T. and Strang, C. D., "On the Empirical Law of Adhesive Wear," J. Appl. Phys., 23, 1, pp 18-28 (1952).

19. Tallian, T. E., McCool, J. I., and Sibley, L. B., "Partial Elastohydrodynamic Lubrication in Rolling Contact," Proc. Inst. Mech. Engrs., 180, Part 3B, pp 169-184 (1965-66).

20. Dowson, D. and Higginson, G. R., Elasto-Hydrodynamic Lubrication, The Fundamentals of Roller and Gear Lubrication, Pergamon Press Ltd., Oxford (1966).

21. Czichos, H., "Influence of Asperity Contact Conditions on the Failure of Sliding Elastohydrodynamic Contacts," Wear, 41, 1, pp 1-14 (1977).

22. Johnson, K. L., Greenwood, J. A., and Poon, S. Y., "A Simple Theory of Asperity Contact in Elastohydrodynamic Lubrication," Wear, **19**, 1, pp 91-108 (1972).

23. Cusano, C. and Wedeven, L. D., "Elastohydrodynamic Film Thickness Measurements of Artificially Produced Nonsmooth Surfaces," NASA TM-79214 (1979).

24. Stribeck, R., "Characteristics of Plain and Roller Bearings (in German)," Zeit. V.D., **46**, pp 1341-48, 1432-38, 1463-70 (1902).

25. Wilson, R. E. and Barnard, D. P., "The Mechanism of Lubrication. II - Methods of Measuring the Property of Oiliness," J. SAE, **11**, pp 143T-157T (1922).

TABLE 1. - TYPICAL PROPERTIES OF THE EXPERIMENTAL FLUIDS

Properties*	Ester base fluid†	C-ether base fluid
Kinematic viscosity, m^2/sec (cS):		
At 38° C (100° F)	3.96×10^{-5} (39.6)	2.5×10^{-5} (25)
At 99° C (210° F)	7.02×10^{-6} (7.02)	4.1×10^{-6} (4.1)
At 150° C (302° F)	2.80×10^{-6} (2.80)	1.9×10^{-6} (1.9)
At 260° C (500° F)	1.06×10^{-6} (1.06)	7.6×10^{-7} (0.76)
At 300° C (572° F)	‡8.6×10^{-7} (0.86)	6.9×10^{-7} (0.69)
Pour point, °C (°F)	-51 (-60)	-29 (-20)
Flash point, °C (°F)	274 (525)	239 (445)
Fire point, °C (°F)	----------------	285 (540)
Density at 38° C (100° F), kg/m^3 (g/ml)	§0.994	1.19×10^3 (1.19)
Thermal decomposition (isoteniscope), °C (°F)	298 (536)	390 (734)
Vapor pressure, N/m^2:		
At 220° C (428° F)	1.33×10^2	----------------
At 371° C (600° F)	----------------	1.86×10^4
Surface tension at 23° C (73° F), N/cm (dynes/cm)	----------------	‖4.48×10^{-4} (44.8)

*Manufacturer's data.
†Base stock for formulating a MIL-L-27502 candidate lubricant.
‡Extrapolated.
§Specific gravity, 15.6° C/15.6° C (60° F/60° F).
‖Measured in authors' laboratory.

TABLE 2. - RIDER WEAR RATE AND CORRELATION COEFFICIENT AS CALCULATED BY LINEAR REGRESSION ANALYSIS FOR ESTER BASE FLUID LUBRICATION TEST

[Disk temperature, 20° C; test atmosphere, dry air (<100 ppm H_2O); M-50 steel disk and pure iron rider.]

Load, kg	Disk speed, rpm	Wear rate (slope), m^3/m	Correlation coefficient,* R	Square of correlation coefficient, R^2
0.5	25	18.15×10^{-14}	0.9944	0.9888
	50	9.27	.9991	.9982
	75	5.97	.9964	.9928
	100	4.53	.9980	.9960
1.0	25	23.73×10^{-14}	0.9984	0.9968
	50	23.51	.9986	.9972
	75	14.21	.9943	.9886
	100	9.64	.9951	.9902

*Ref. (14).

TABLE 3. - CALCULATED FILM THICKNESSES AND FILM PARAMETERS FOR ESTER BASE FLUID

Disk speed, rpm	Film thickness* = h_o, m (μin.)	Dimensionless film parameter† = Λ, $\Lambda = h_o/\left(\sigma_1^2 + \sigma_2^2\right)^{1/2}$
100	9.25×10^{-8} (3.7)	0.90 to 0.46
75	7.75×10^{-8} (3.1)	0.76 to 0.38
50	5.75×10^{-8} (2.3)	0.55 to 0.29
25	3.87×10^{-8} (1.55)	0.38 to 0.19

*Central film thickness calculated from Dowson and Higginson equation (Ref. 20) for 20° C disk temperature, 1-kilogram load, and fluid viscosity of 89 cP at 20° C.

†Range of Λ values for $\sigma_1 = 2.5 \times 10^{-8}$ m (rider roughness) and $\sigma_2 = 10 \times 10^{-8}$ to 20×10^{-8} m (disk roughness range).

TABLE 4. - TYPICAL RUN-IN AND STEADY-STATE WEAR VALUES FOR

C-ETHER AND ESTER BASE FLUIDS

[Disk temperature, 20° C; load, 1 kilogram; disk speed, 50 rpm.]

Test fluid	Run-in period		Steady-state period			
	Duration, min.	Wear, m^3	Wear rate after run-in, m^3/m	Wear volume increase, m^3, after -		
				110 min (during test)	300 min (*)	600 min (*)
C-etner base	20	2.1×10^{-11}	8.3×10^{-15}	0.65×10^{-11}	1.8×10^{-11}	3.6×10^{-11}
Ester base	20	4.0×10^{-11}	2.35×10^{-13}	2.09×10^{-10}	5.71×10^{-10}	11.42×10^{-10}

*Extrapolated values.

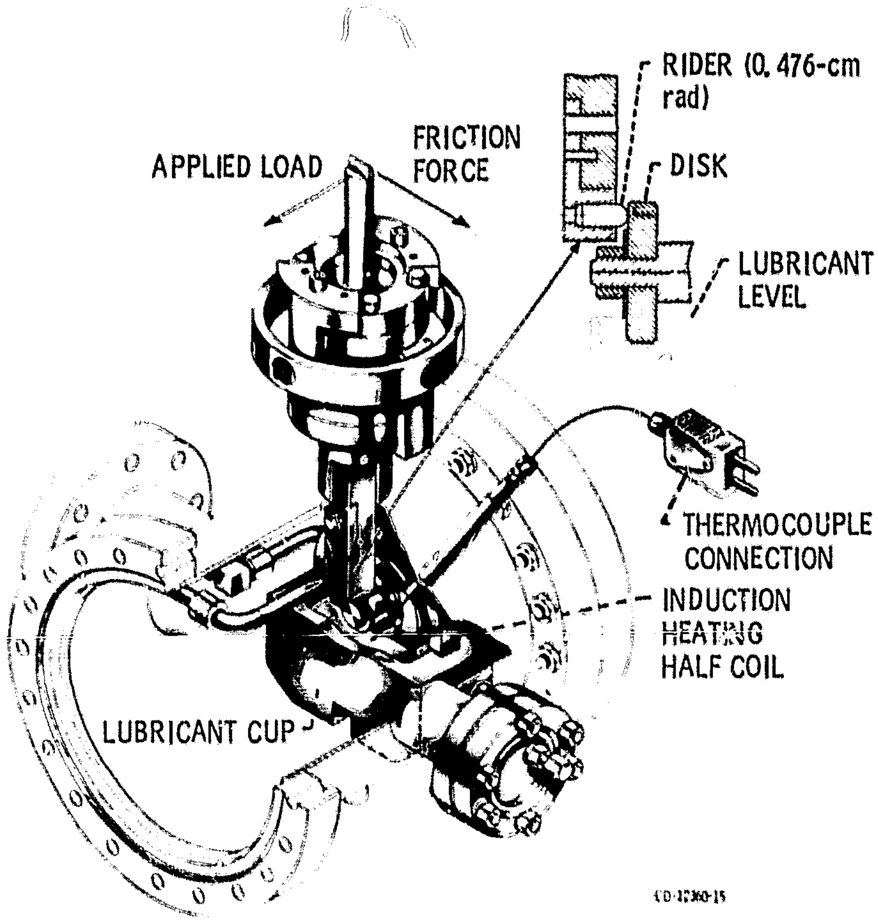
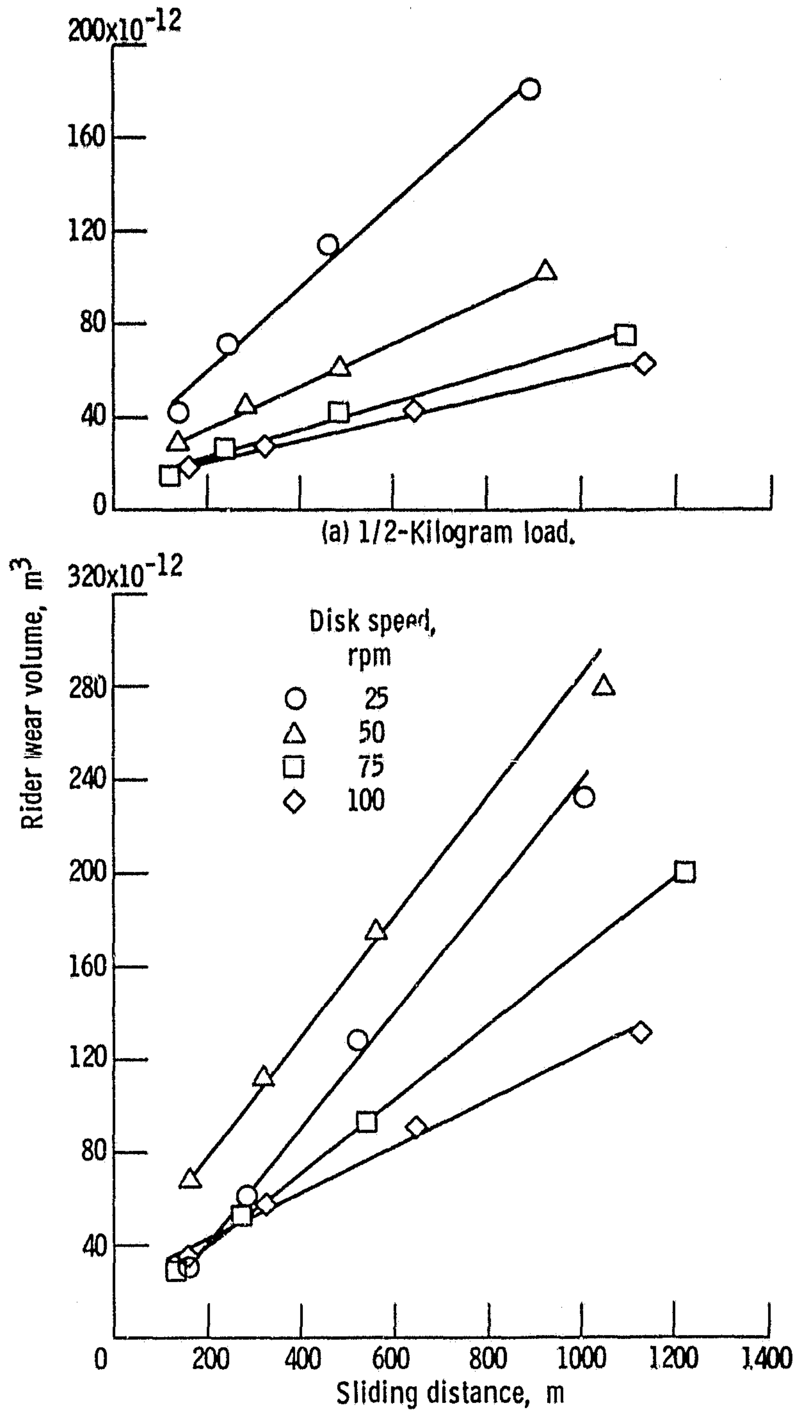


Fig. 1. - Friction and wear apparatus.



(b) 1-Kilogram load.

Fig. 2 - Rider wear as function of sliding distance for ester base lubricant. Atmosphere, dry air; disk temperature, 20^{0} C; pure iron rider on M-50 disk.

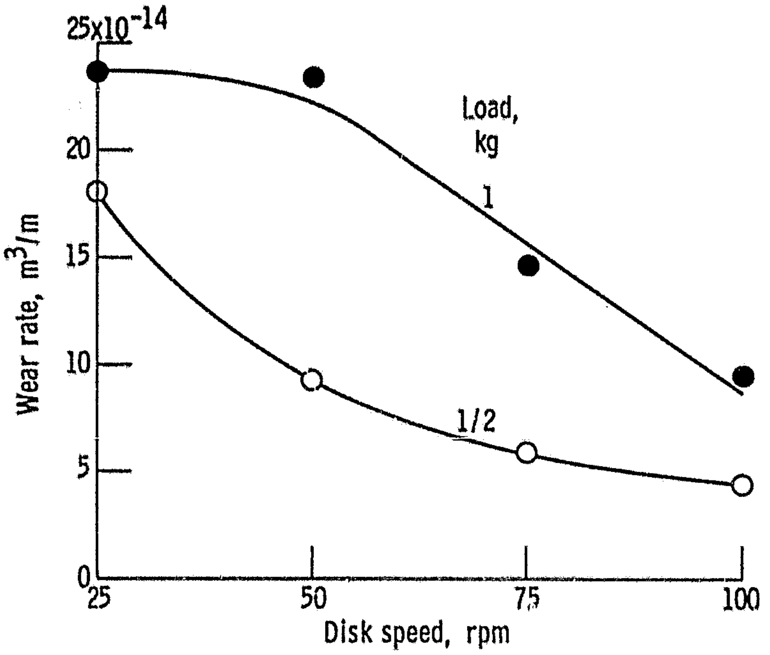


Fig. 3 - Effect of disk speed on rider wear rate at two loads with ester base lubricant. Atmosphere, dry air; disk temperature, 20° C; pure iron rider on M-50 disk.

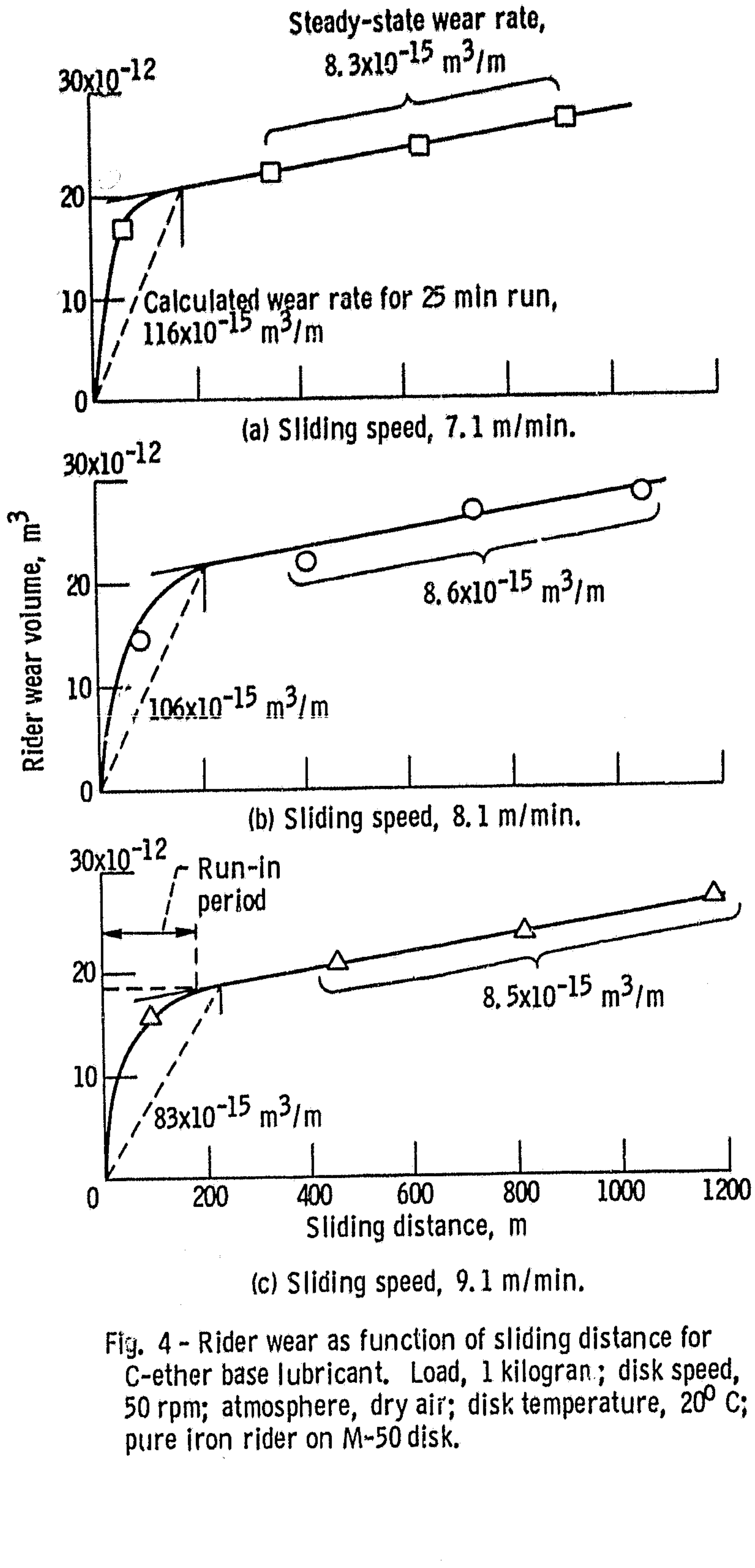


Fig. 4 - Rider wear as function of sliding distance for C-ether base lubricant. Load, 1 kilogram; disk speed, 50 rpm; atmosphere, dry air; disk temperature, 20^0 C; pure iron rider on M-50 disk.

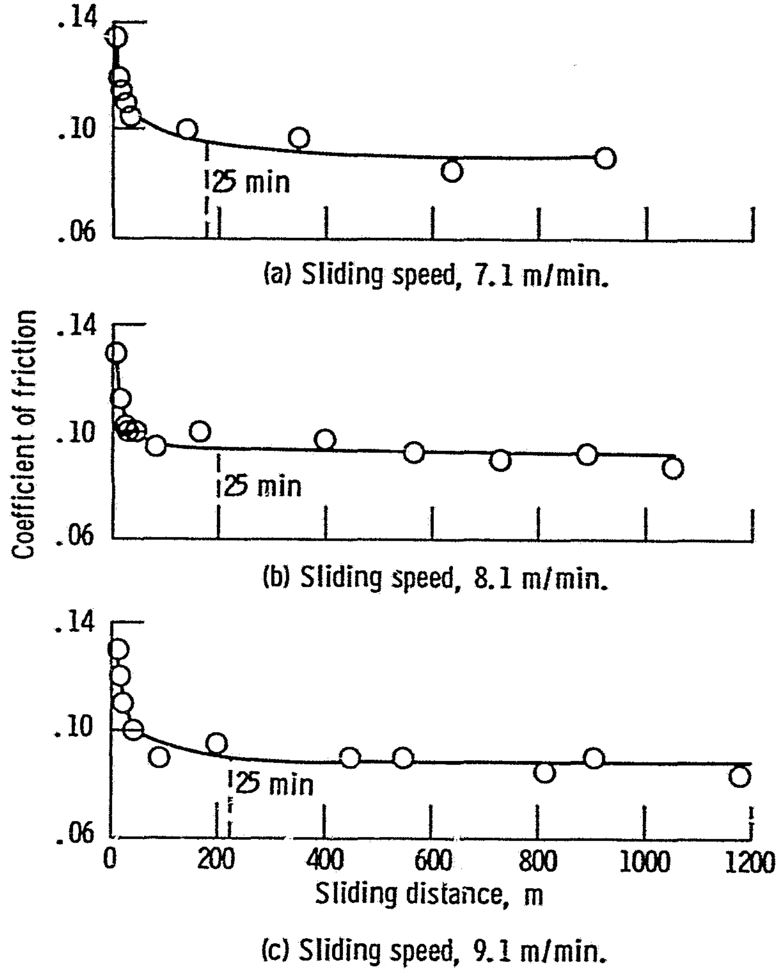


Fig. 5 - Coefficient of friction as function of sliding distance for C-ether base lubricant. Load, 1 kilogram; disk speed, 50 rpm; atmosphere, dry air; disk temperature, 20^0 C; pure iron rider on M-50 disk.

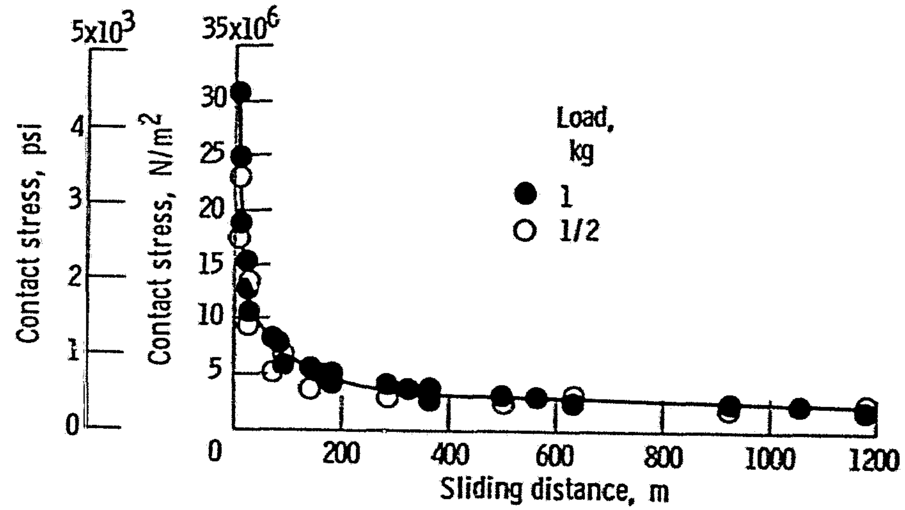


Fig. 6 - Contact stress as function of sliding distance for ester base lubricant. Disk speed, 50 rpm; atmosphere, dry air; disk temperature, 20^0 C; pure iron rider on M-50 disk.