Simulative Friction and Wear Study of Retrofitted Swash Plate and Rolling Piston Compressors

T. Sheiretov, W. Van Glabbeek, and C. Cusano

ACRC TR-29 December 1992

For additional information:

Air Conditioning and Refrigeration Center
University of Illinois
Mechanical & Industrial Engineering Dept.
1206 West Green Street
Urbana, IL 61801

(217) 333-3115

Prepared as part of ACRC Project 4
Compressor--Lubrication, Friction, and Wear
C. Cusano, Principal Investigator

The Air Conditioning and Refrigeration Center was founded in 1988 with a grant from the estate of Richard W. Kritzer, the founder of Peerless of America Inc. A State of Illinois Technology Challenge Grant helped build the laboratory facilities. The ACRC receives continuing support from the Richard W. Kritzer Endowment and the National Science Foundation. The following organizations have also become sponsors of the Center.

Acustar Division of Chrysler
Allied-Signal, Inc.
Amana Refrigeration, Inc.
Bergstrom Manufacturing Co.
Caterpillar, Inc.
E. I. du Pont de Nemours & Co.
Electric Power Research Institute
Ford Motor Company
General Electric Company
Harrison Division of GM
ICI Americas, Inc.
Johnson Controls, Inc.
Modine Manufacturing Co.
Peerless of America, Inc.
Environmental Protection Agency
U. S. Army CERL
Whirlpool Corporation

For additional information:

Air Conditioning & Refrigeration Center
Mechanical & Industrial Engineering Dept.
University of Illinois
1206 West Green Street
Urbana IL 61801

217 333 3115

Introduction

The production of R12 will cease in the near future. Many air conditioning and refrigeration systems, however, will continue to operate with R12 for many years to come. If the refrigerant in these systems must be changed for reasons such as leakage or maintenance, then the old R12/oil mixture will probably be replaced by a R134a/oil mixture. Upon removing the R12/oil mixture, some residual amounts of both the R12 and the oil may remain in the system and may influence the tribological properties of the substituted R134a/oil mixture. Therefore, the goal of this study was to evaluate the tribological properties of small residual amounts of R12/oil mixtures in R134a/oil mixtures.

The tribological properties of the oil/refrigerant mixtures of interest were evaluated by testing specimens in a high pressure tribometer. A complete description of the high pressure tribometer capabilities, the experimental method used, and the test procedures are given in [1] and [2]. For the contacts studied, the operating conditions, materials and lubricants are also the same as those used in [1] and [2]. The specimens' geometry and material are representative of the geometry and the materials of the critical tribo-contacts found in real compressors. The operating and environmental conditions such as load, speed, pressure, and temperature were also approximately simulated.

Two contact geometries were simulated in this study. One was a vane-piston contact in a rolling piston compressor used in home refrigerators, and the other a shoe-plate contact in a swash plate compressor used in automotive air conditioners. The tribological behavior of pure R134a/oil mixtures were compared to mixtures containing a small amount of residual R12-based mixture. For each contact geometry, tests with both base and formulated versions of the lubricant were conducted. Since, in practice, the tribological contacts of interest have been operated in a R12 environment for a certain period of time, it is expected that some changes of the contact surface properties have occurred. In order to simulate this condition, consecutive tests were conducted on the same materials contact pairs, the first with a R12/oil mixture and the second with a R134a/oil mixture. Note that, in this case, the second test was conducted on an already worn surface. Tests on a virgin surface were run as well. Higher accuracy of the wear measurements could be achieved when using virgin surfaces.

Description of the Tests Conducted

Counterformal contact
(vane-piston contact in a rolling piston compressor)

The critical contact in rolling piston compressors is between the vane and the piston. This is a Hertzian contact between two cylindrical surfaces. It is simulated by a contact between a cylindrical pin and a flat disc. The pin has an equivalent diameter such that the contact stresses are the same as in the real vane-piston contact. For more information on the pin and disc contact geometry, and pin diameter derivation see ACRC TR-19, May 1992 [2].

All the test conditions and specimen material properties were kept the same as in our previous studies [1]. The pin, corresponding to the vane in the real compressor, is made from hardened tool steel, while the plate is a hardened gray cast iron. The relevant geometrical and material data for the model are shown in Table 1. Test operating conditions are given in Table 2.

For the counterformal contact, the oil/refrigerant mixtures under study were:

1) R12 + alkylbenzene oil
2) R134a + base polyol ester oil
3) R134a + formulated polyol ester oil
4) R134a + base polyol ester oil
   + residual amounts of R12 and alkylbenzene oil.
5) R134a + formulated polyol ester oil
   + residual amounts of R12 and alkylbenzene oil.

Data on the oils used are given in Table 3. The polyol ester oil is designated as Ester base for the base version, and Ester form for the formulated version. The alkylbenzene oil used was a base oil, designated as Alkylbenz in Table 3.

The R12/oil residue was supplied to the R134a/oil mixture by the following procedure:

1) A known amount of polyol ester oil was supplied to the cup of the tribometer.
2) 20% by weight of alkylbenzene oil was added to the cup.
3) The chamber was charged with R134a refrigerant. The amount of the gas supplied was determined by measuring the weight of the pressure vessel before and after charging.
4) One to five percent by weight of R12 refrigerant was supplied to the chamber from another pressure vessel.

The amount of residual R12 used reflects the fact that almost all of it will escape when the system is refilled. There were some difficulties in supplying such a small quantity of R12 to the chamber. The amount of R134a refrigerant, necessary to maintain a pressure of 225 psi at 177°F in the chamber, was about 200g. Hence, 1% by weight of R12 in the mixture was only 2g . The weight of the pressure vessel itself from which the R12 was supplied was approximately 3000g, i.e. much heavier than the 2g of R12 needed in the chamber. The charging procedure is a dynamic process which makes the monitoring of the weight change difficult. We were able to monitor the weight loss of the vessel with an accuracy of 0.1%, or about 3g. The process of charging continued to the point when approximately 5-6g change in the pressure vessel weight was observed. This corresponded to a range of one to five percent of R12 in R134a.

The amount of alkylbenzene oil used was 20%, since it is expected that a significant quantity of the previously used oil may remain in the system. This amount of oil has been suggested as an extreme but still possible amount of old lubricant that can remain in the system.

Tests were conducted on both virgin materials contact pairs, as well as on contact pairs which had already been tested using R12/oil mixtures. The latter tests were conducted to more accurately simulate actual compressor operation including the possibility of the existence of surface protective films due to a chemical reaction between R12 and the metal surface. These films may continue to protect the surface even after the R12 has been replaced by another refrigerant. For this test series, the first test was conducted with a R12-

alkylbenzene oil mixture, which simulated the operation of the compressor before the refill. Next, another test with a R134a/oil mixture, either with or without residual amount of R12/oil mixture, was conducted on the same set of specimens.

Tests conducted on virgin surfaces with R134a/oil mixtures give results that are easier to compare because the initial surface conditions are the same for each test. On the other hand, for tests conducted to simulate retrofitting conditions, i.e. R134a/oil mixture tests following R12/oil mixture tests, the surfaces are already worn when the R134a/oil mixture is tested, and therefore surface topography, as well as contact pressure distribution might be significantly different for the two pairs of tests. Such differences however can also be expected when a compressor is retrofitted with a new refrigerant/oil mixture after being in operation for a long period of time.

All the tests followed the same procedure which is covered more extensively in [2]. Each test was one hour long, and each test was repeated at least twice with some repeated three times in order to get better reliability of the data. The operating conditions for all the tests were the same.

Area contact (shoe-plate contact in a swash plate compressor)

The critical contact in the swash plate compressor is that between the shoe and the plate. The equivalent geometry used in each test is the same. The area contact is made up of a bronze shoe, loaded by a steel ball to permit the same degree of freedom found in the actual compressor, sliding against a flat ductile cast iron plate. Table 1 shows the geometry, material and surface conditions used in modeling the area contact. The operating conditions shown in Table 2 do not completely simulate conditions in the actual compressor. When the area contact was run at pressures equivalent to those in the real compressor, even relatively low speeds caused hydrodynamic liftoff. In order to avoid the liftoff and generate measurable wear, it was necessary to increase the pressure and decrease the speed relative to compressor's operation.

For the area contact, the oil/refrigerant mixtures under study were:

1) R12 + mineral oil
2) R134a + base PAG oil
3) R134a + formulated PAG oil
4) R134a + base PAG oil
   + residual amounts of R12 and mineral oil.
5) R134a + formulated PAG oil
   + residual amounts of R12 and mineral oil.

PAG (Polyalkaline glycol) oil properties are given in Table 3 under the designation PAG base, or PAG form for the base and the formulated versions of the oil, respectively. The oil used as a component of the residual mixture was mineral oil, designated as Mineral in Table 3. The mixtures were prepared the same way as for the rolling piston tests.

Most of the tests were run on virgin surfaces. Some tests were run on previously tested specimens lubricated by a R12-mineral oil mixture. However these specimens tend to mutually polish their mating surfaces to a "mirror finish", and no measurable wear could be obtained in the subsequent tests on the same specimens. Therefore, data from these tests are not presented. Most of the tests were one hour long. Some longer tests were also conducted in order to verify the expected chemical degradation of the PAG oils in the presence of R12. Such degradation could produce HCl (hydrochloric acid), which would have detrimental effects on the operation of the system. To check for HCl production, a 14 hour test was run, and samples were taken every two hours. Part of the oil sample was shaken with water and the aqueous solution was then studied with pH paper and AgNO3 for the presence of HCl. However, no increase in the acidity was found. Samples were also sent to Caterpillar Oil Testing Laboratory to be tested for their acid numbers, applying a standard ASTM D664 procedure. The results from this test indicated that no acidity change occurred in the PAG base oil. Throughout the test, the acid number remained constant and equal to 0.2. However, it is possible that the duration of these tests was too short to produce oil degradation.

Data Acquisition

Counterformal contact (rolling piston compressor)

In addition to the friction coefficient and wear measurements, which are of prime interest, the roughness and hardness of the lower specimen (cast iron plate) were also measured for each test. The last two measurements were conducted in order to check the possibility that some other factors other than type of oil and refrigerant may affect the test results. The data acquisition is described below:

1) The friction coefficient, specimen and chamber temperatures, forces and moments acting on specimens were monitored and recorded continuously throughout the test using a computer data acquisition system. The friction coefficient reported is the average value of the friction coefficient data points throughout the test. For this particular contact, the friction coefficient remains almost constant with a slight tendency to decrease as the test proceeds.

2) The amount of wear was evaluated by measuring the width of the wear scar on the upper specimen (M2 tool steel pin) by means of an optical microscope. The accuracy of these measurements is determined by the value of a single division on the microscope measuring scale at the highest magnification. This value is 0.0078mm. The average scar width is about 0.2 mm. Therefore, the width of the wear scar can be measured with an error that does not exceed 3.9%. A typical wear scar appearance is shown in Figure 1. The width of the wear scar may vary slightly along the length of the specimen by 1-5 divisions (0.0078-0.039 mm) due to speed variations along the pin. The value reported in this case is the mean of five measurements along the length of the specimen. In some cases, the error in these measurements may be increased by a few percent due to the irregularity of the wear scar boundaries.

The volume worn can be calculated on the basis of the wear scar width using simple geometry. Since the volume worn is a strongly nonlinear function of the wear scar (cubic), the measurement error may accumulate and reach unacceptable values. Therefore, the reported values for the volume worn are based on the average wear scar widths. This averaging technique works for tests conducted on virgin surfaces only. For the tests run on a previously worn surface, the only way to obtain the volume of the material worn during the second test is to subtract the volume worn during the first test from the total volume worn. Since here we have to work with volumes rather than with wear scar widths, it is expected that the results for the tests conducted on a previously worn surface will be less accurate.

Wear on the mating cast iron plates was observed as well. However, the depth of the wear scar obtained is small and is difficult to measure. Therefore it is not used for quantitative assessment of the amount of wear.

3) The surface roughness was obtained using a DEKTAK stylus type profilometer.

4) The surface hardness was obtained using the Vickers micro-hardness testing machine.

Area contact (swash plate compressor)

Friction coefficient, wear, plate hardness and surface roughness of both specimens were obtained. In addition to these measurements, the surface of the bronze shoe specimens was studied for the presence of surface films using XPS (x-ray photoelectron spectroscopy) and SEM (scanning electron microscopy). Since some of our previous studies [1] indicated the formation of protective chlorine-based and fluorine-based films on the metal surfaces when the contact is operated in an R12 environment, the aim of the current analysis was to reveal how these films behave when R12 is replaced by R134a. Three different specimens were analyzed using XPS. The first specimen was tested in a R12/mineral oil mixture. The second and the third specimens were first tested in a R12/mineral oil mixture, and were then tested again in a R134a/PAG oil mixture with and without R12/mineral oil residue.

The acidity of the oils during the tests involving a mixture of R12 with a PAG oil was also measured. This was done by shaking the oil samples with water and testing the aqueous solution obtained for the presence of HCl. Standard ASTM D664 test for oil acid number was applied as well.

All other measurements were made according to the following procedures:

1) The value for the friction coefficient was obtained in the same way as for the counterformal contact. The records of friction coefficient changes during the test were used not only to compute the mean coefficient of friction value, but also to monitor the type of lubrication regime. In addition to this, the load on the specimen was monitored by means of the data acquisition system. The load data

were used to evaluate the possible effect of dynamic loads on the regime of lubrication.

2) The amount of wear was determined by measuring the weight of the bronze shoe specimens before and after the tests. The reported values for the wear are in mg. The accuracy of the measurements was 0.01 mg. No wear was detected on the mating cast iron plate specimens. On the contrary, some transfer of material from the shoe to the plate was observed.

3) The surface profiles of the bronze shoe specimens were measured using the DEKTAK profilometer.

4) The surface roughness and the hardness of the plates were measured in the same way as they were measured in the counterformal contact tests.

Results and Discussion

Counterformal contact

The rolling piston tests give comparatively small scatter of data; therefore, good repeatability of the results was obtained. Table 4 gives the results for the rolling piston test conducted on virgin surfaces. The highlighted numbers are the mean values of the data in the column above them. The only exception is the value for the volume worn which is obtained on the basis of the average scar width. The tests designated by an "*" are those from our previous report [2]. It is evident that data from our recent tests confirm the data obtained before.

Table 5 gives the results for the tests conducted on an already worn surface. The test denoted as "first test" is the one run in a R12/oil mixture environment. The "second test" is the one in which a R134a/oil mixture was used with or without residual amount of R12. Both the first and the second test were one hour long. Since two consecutive tests were run over the same wear scar, data for two wear scar widths and volumes worn are given in Table 5. Friction coefficients are given for the second test only. The surface hardness of the plate is also given. The wear during the second test is obtained by subtracting the volume worn during the first test from the total volume worn. These values can therefore be compared to the amount of volume worn on a virgin surface since both tests lasted one hour. The highlighted values in this table represent the mean values for the data in the column above them.

Since the first tests given in Table 5 were run on a virgin surface, the value for the volume worn can be compared to the values of the volume worn in Table 4. From this comparison it is evident that the R12-alkylbenzene oil-refrigerant mixture is much better from a tribological point of view than any of the R134a/oil combinations. This confirms the results from our studies reported before. Data from Table 4 and Table 5 are the basis for the plots on Figure 2 and Figure 3.

The results from the tests generally confirm our expectations for the tribological behavior of different oil-refrigerant mixtures. The presence of small amounts of R12 seems to slightly improve the tribological characteristics of the mixture no matter what type of surface or oil was studied. Also, the formulated oils seem to perform better, which is also consistent with our previous observations. The differences in tribological properties between the pure R134a/oil mixture and the one containing residual amounts of R12/oil is very small. Therefore, the effect of the residual amounts of R12/oil mixture on the overall performance of the R134a-based mixture is negligible from the stand point of friction and wear.

The possibility that some other factors such as plate hardness may affect the friction and wear results was also considered. Figure 4 shows a regression analysis of the correlation between the wear and surface hardness. It may be concluded that this factor, in the range of 440HV to 650HV, does not significantly affect the results.

Area contact

During these tests, some unexpected scatter of friction and wear data was observed. Tests that were conducted under the same conditions sometimes gave different results. The scatter of data can be due to two different regimes of lubrication, i.e. boundary and mixed. The basic characteristic for the first type is the comparatively high friction coefficient and substantial wear. The second is characterized by a much lower coefficient of friction and wear . Typical records of the high friction regime are given in Figure 5. The low friction regime is illustrated in Figure 6. It is also possible that a switch from one regime of lubrication to the other occurs during the test, as in the test record shown in Figure 7.

Various reasons for this kind of behavior were studied. The surface hardness of all plates were measured. However, for the tests conducted, as with the rolling piston tests, no relationship was found between this parameter and friction and wear. The regression analysis for the influence of the surface hardness on the friction and wear results is shown in Figure 8. Some tests were also run on the same plate, and sometimes these tests showed different regimes of friction and wear. This excludes the possibility that this factor is the major reason for this kind of behavior.

Next, the axial load records were studied. The goal was to check whether some geometrical irregularities of the specimens and the testing machine itself could produce enough fluctuations in the load to cause changes in the lubrication regime. A typical load record is shown in Figure 9. Typically, the oscillations are less than 12% in magnitude compared to the mean force value. The mean force value and the oscillations stay fairly constant throughout the test. The maximum and minimum values of the load always occur at certain position of the spindle which suggests that they are caused by the spindle rotation and some flatness deviations of the specimens. A comparison of the load dynamics and coefficient of friction records shows that the fluctuations of the force are not related to the regime of lubrication.

The possibility that the hydrodynamic liftoff of the lower specimen (the bronze shoe), may be the major factor for the regime of lubrication was also checked. Specimens with both convex, concave, and flat geometries were tested. The surface profile of these specimens were obtained using the DEKTAK profilometer before and after the test. Typical examples of these profiles are shown in Figures 10 and 11. In both figures, the first surface profile is the one taken before the test, while the other two show the same specimen profile after the test. One of the profile measurements after the test was taken in a direction of sliding, while the other was taken perpendicular to this direction. According to hydrodynamics theory, the convex specimens can more easily develop fluid films between the surfaces, resulting in better lubrication condition. However, for the loads and speeds used in these tests, hydrodynamic lubrication does not exist and the initial surface geometry does not play a major role in the lubrication process.

Since none of the factors listed above seem to be the major reason for the lubrication regime, it is concluded that probably a complex combination of all these factors contribute to the changing from one regime to the other during the tests. A possible reason for the presence of the both regimes of lubrication is that the test conditions lie in some transitional zone in which a small change in any of the test parameters, may lead to a change in the regime of lubrication.

Based on the above conclusion, the data obtained from the tests were processed separately for the two different regimes of lubrication. The criterion used for this classification was the mean value of the friction coefficient and the appearance of the coefficient of friction record. In Table 6, data for the tests run with the base version of the PAG oil are presented. Most of the tests ran under boundary lubrication conditions. The reported wear is in milligrams, and is measured as a weight loss on the bronze shoe. Plate hardness is reported as well.

Table 7 is analogous to Table 6 with the only difference that the data are from the tests conducted with formulated version of the PAG oil. The plots on Figures 12 and 13 are based on the data from Tables 6 and 7.

The comparison of the wear values for the tests run with only R134a/PAG oil mixtures with those conducted in the presence of R12/mineral oil residue are, in general, consistent with each other and with the results from the counterformal contact tests. The presence of R12/oil residues slightly improves the tribological properties of the refrigerant/oil mixture, but this improvement may be considered negligible from a practical point of view.

As previously stated, an XPS analysis was done on three separate specimens, and the results show that some films are present on their surfaces. The thickness of these films are of the order 30 angstroms. The chemical composition of these films are given in Table 8. Since the accuracy of the percent concentration is 1%, the results indicate that the amount of chlorine and fluorine on all these specimens is approximately the same. Because chlorine is not present in R134a but is still present on specimens 2 and 3 after the tests, it can be concluded that possible chlorine surface films are formed during the R12/mineral oil test and are not worn off during the one hour R134a/PAG oil test with or without the presence of the R12/mineral oil residue. Since fluorine exists in R134a, it is not possible to say with certainty that the fluorine on the surface of specimens 2 and 3 also originated from the R12 test. The similarity in the composition of the surface films for the tests run with and without R12/mineral oil residue is one possible explanation why the friction and wear results shown in figures 12 and 13 are similar for both tests run with and without R12/mineral oil residue.

Conclusions:

For the tests conducted, the presence of R12 and a residual amount of alkylbenzene or mineral oil, do not have a negative effect on the lubricative properties of the R134a/oil mixture.

For both the counterformal and area contacts under study, the presence of small residual amounts of R12/oil mixture tends to improve the tribological conditions of the contacts. These improvements, however, may be considered negligible. Based on a 14-hour long test, the acid number of the PAG's did not show any measurable increase. Test duration, however might be too short to produce any oil degradation.

For the tests conducted in this research, it may be concluded that, from a tribological point of view, the presence of R12/oil residue does not significantly affect the tribological properties of R134a/oil mixture.

References:

1. Davis, B., Sheiretov, T. and Cusano, C., *Tribological Evaluation of Contacts Lubricated by Oil-Refrigerant Mixtures*, ACRC-TR-16, March, 1992.

2. Davis, B. and Cusano, C., *The Tribological Evaluation of Compressor Contacts Lubricated by Oil-Refrigerant Mixtures*, ACRC-TR-19, May, 1992.

Table 1 - Contact Geometries and Materials

Description	Couterformal contact	Area contact
Geometry: - Upper specimen - Lower specimen	 76.2 mm Ø, Flat disk 6.35 mm Ø, Pin Length=9.25 mm	 76.2 mm Ø, Flat disk 5.08 mm Ø, Flat shoe
Materials: - Upper specimen - Lower specimen	 Gray cast iron Tool steel	 Ductile cast iron Bronze
Average Hardness: - Upper specimen - Lower specimen	 550 HV (53 RC) 840 HV (65 RC)	 426 HV (38 RC) ----
Surface topography: - Upper specimen - Lower specimen	 Ground Ground	 Ground Lapped
Average Surface Finish (Ra): - Upper specimen - Lower specimen	 0.13 µm 0.13 µm	 0.13 µm 0.21 µm

Table 2 - Operating Conditions

Operating Conditions	Counterformal Contact	Area Contact
Contact Pressure (MPa)	1034	124
Type of Motion	Oscillatory	Unidirectional
Speed (m/sec)	± 0.51 max	0.20
Angular Amplitude	± 50°	---
Angular Frequency	5 Hz	---
Env. Pressure (MPa)	1.55	0.172
Env. Temperature (°C)	80.6	73.9

Table 3 - Lubricant Data

Oil	Oil Type	Family *	Additives	Viscosity	
				@ 40°C	@ 100°C
Mineral	Mineral Oil	---	No	102	11.12
Alkylbenz	Alkylbenzene	---	No	57	5.8
PAG base	Polyalkylene glycol	Mono	No	135	25
PAG form	Polyalkylene glycol	Mono	Yes	135	25
Ester base	Polyolester	PE	No	23.94	4.88
Ester form	Polyolester	PE	Yes	23.9	4.87
* PE - Pentaerythritol ester; Mono - Monoether					

Table 4 - Friction and Wear Data for Counterformal Contact, Duration of Each Test=1 hour Tests Run on Virgin Surfaces

Test #	Refrigerant	Oil	Plate Hardness HV	Frict. Coef.	Wear Scar mm	Vol. Worn mm^-3
76RL	R134a	Ester base	514	0.105	0.203	2.092
79RL	R134a	Ester base	644	0.086	0.179	1.434
80RL	R134a	Ester base	571	0.106	0.273	5.089
82RL	R134a	Ester base	606	0.095	0.211	2.349
84RL	R134a	Ester base	606	0.090	0.226	2.887
86RL	R134a	Ester base	464	0.090	0.296	6.488
32RL*	R134a	Ester base	-	0.105	0.211	2.349
33RL*	R134a	Ester base	-	0.104	0.242	3.545
				0.098	0.230	3.043
77RL	R134a + R12	Ester base+Alkylbenz	514	0.082	0.203	2.092
78RL	R134a + R12	Ester base+Alkylbenz	626	0.085	0.156	0.949
81RL	R134a + R12	Ester base+Alkylbenz	571	0.106	0.328	8.829
83RL	R134a + R12	Ester base+Alkylbenz	565	0.096	0.211	2.349
85RL	R134a + R12	Ester base+Alkylbenz	606	0.090	0.226	2.887
87RL	R134a + R12	Ester base+Alkylbenz	463	0.090	0.242	3.545
				0.092	0.228	2.964
36RL*	R134a	Ester form	-	0.108	0.218	2.591
37RL*	R134a	Ester form	-	0.106	0.242	3.545
88RL	R134a	Ester form	532	0.113	0.156	0.949
90RL	R134a	Ester form	516	0.115	0.156	0.949
				0.111	0.193	1.798
89RL	R134a + R12	Ester form+Alkylbenz	532	0.095	0.172	1.272
91RL	R134a + R12	Ester form+Alkylbenz	516	0.094	0.195	1.854
				0.094	0.184	1.558

* Tests included in our previous report [2]

Table 5a - Friction and Wear Data for Counterformal Contact, Duration of Each Test = 1 hour
First test run on virgin surfaces in R12/Alkylbenzene oil environment.
Second test run on worn surfaces from the first test in R134a/Ester base oil.

Second Test #	Refrigerant	Plate HV	Frict. Coef.	WEAR				
				First Test Wear Scar mm	First Test Volume Worn mm^-3	Second Test Wear Scar mm	Total Volume Worn mm^-3	Second Test Volume Worn mm^-3
73RL	R134a	606	0.085	0.140	0.686	0.250	3.908	3.222
75RL	R134a	606	0.100	0.120	0.432	0.226	2.887	2.455
			0.092		0.559			2.839
49RL*	R134a+R12	582	0.100	0.125	0.488	0.242	3.545	3.057
51RL*	R134a+R12	566	0.085	0.133	0.588	0.222	2.736	2.148
			0.093		0.538			2.603

Table 5b - Friction and Wear Data for Counterformal Contact, Duration of Each Test = 1 hour
First test run on virgin surfaces in R12/Alkylbenzene oil environment.
Second test run on worn surfaces from the first test in R134a/Ester formulated oil.

Second Test #	Refrigerant	Plate HV	Frict. Coef.	WEAR				
				First Test Wear Scar mm	First Test Volume Worn mm^-3	Second Test Wear Scar mm	Total Volume Worn mm^-3	Second Test Volume Worn mm^-3
69RL	R134a	464	0.100	0.120	0.432	0.250	3.908	3.476
71RL	R134a	464	0.107	0.125	0.488	0.195	1.854	1.366
94RL	R134a	582	0.096	0.125	0.488	0.211	2.349	1.861
			0.101		0.469			2.234
53RL*	R134a+R12	566	0.901	0.125	0.488	0.226	2.887	2.399
55RL*	R134a+R12	516	0.080	0.120	0.432	0.211	2.349	1.917
57RL*	R134a+R12	516	0.078	0.140	0.686	0.234	3.205	2.520
95RL*	R134a+R12	582	0.086	0.120	0.432	0.195	1.854	1.422
			0.286		0.510			2.065

* Only residual amount of alkylbenzene oil present

Table 6a - Friction and Wear Data for Area Contact: High Friction Regime
Duration of Each Test = 1 hour

Test #	Refrigerant	Oil	Frict. Coef.	Shoe Wear mg	Plate Hardness HV	Contact Pressure psi
97SP*	R134a+R12	PAG base + mineral	0.055	1.30	482	18000
88SP*	R134a+R12	PAG base + mineral	0.037	2.50	466	18000
87SP*	R134a+R12	PAG base + mineral	0.032	2.30	423	18000
49SP*	R134a+R12	PAG base + mineral	0.036	7.50	398	30000
48SP*	R134a+R12	PAG base + mineral	0.037	9.20	466	30000
			0.039	3.22		
89SP	R134a	PAG base	0.047	2.20	466	18000
86SP	R134a	PAG base	0.036	3.90	423	18000
47SP	R134a	PAG base	0.033	10.10	423	30000
			0.039	3.38		

Note: shaded values are weighted averages with respect to contact pressure

Table 6b - Friction and Wear Data for Area Contact: Low Friction Regime
Duration of Each Test = 1 hour

Test #	Refrigerant	Oil	Frict. Coef.	Shoe Wear mg	Plate Hardness HV	Contact Pressure psi
53SP*	R134a+R12	PAG base + mineral	0.013	2.20	419	18000
52SP	R134a	PAG base	0.015	2.30	409	18000

* Only residual amounts of mineral oil present

Table 7a - Friction and Wear Data for Area Contact: High Friction Regime
Duration of Each Test = 1 hour

Test #	Refrigerant	Oil	Frict. Coef.	Shoe Wear mg	Plate Hardness HV	Contact Pressure psi
92SP*	R134a+R12	PAG form + mineral	0.042	1.00	419	18000
56SP*	R134a+R12	PAG form + mineral	0.037	2.3 1.67	471	18000
29SP*	R134a+R12	PAG form + mineral	0.070	3.6 2.61	492	18000
			0.050	1.76		
94SP	R134a	PAG form	0.051	1.50	409	18000
55SP	R134a	PAG form	0.042	3.21 2.37	409	18000
54SP	R134a	PAG form	0.061	3.40 2.52	397	18000
			0.051	2.13		

Table 7b - Friction and Wear Data for Area Contact: Low Friction Regime
Duration of Each Test = 1 hour

Test #	Refrigerant		Frict. Coef.	Shoe Wear mg	Plate Hardness HV	Contact Pressure psi
95SP*	R134a+R12	PAG form + mineral	0.017	0.30	409	18000
91SP*	R134a+R12	PAG form + mineral	0.027	0.40	366	18000
57SP*	R134a+R12	PAG form + mineral	0.010	0.30	419	18000
			0.018	0.33		
93SP	R134a	PAG form	0.022	0.60	419	18000
90SP	R134a	PAG form	0.019	0.30	366	18000
			0.020	0.45		

* Only residual amount of mineral oil present

Table 8 - XPS Analysis Results for Area Contact

Element	% concentration		
	Specimen 1	Specimen 2	Specimen 3
Test #1 :	R12/mineral oil	R12/mineral oil	R12/mineral oil
Test #2 :	no 2nd test	R134a/PAG oil	R134a/PAG oil + residue
Oxygen, O1s	33.81	38.85	31.60
Copper, Cu2p3	13.31	11.74	11.84
Chlorine, Cl2p	0.74	0.64	0.90
Carbon, C1s	52.01	48.51	55.61
Flourine, Fls	0.12	0.27	0.05

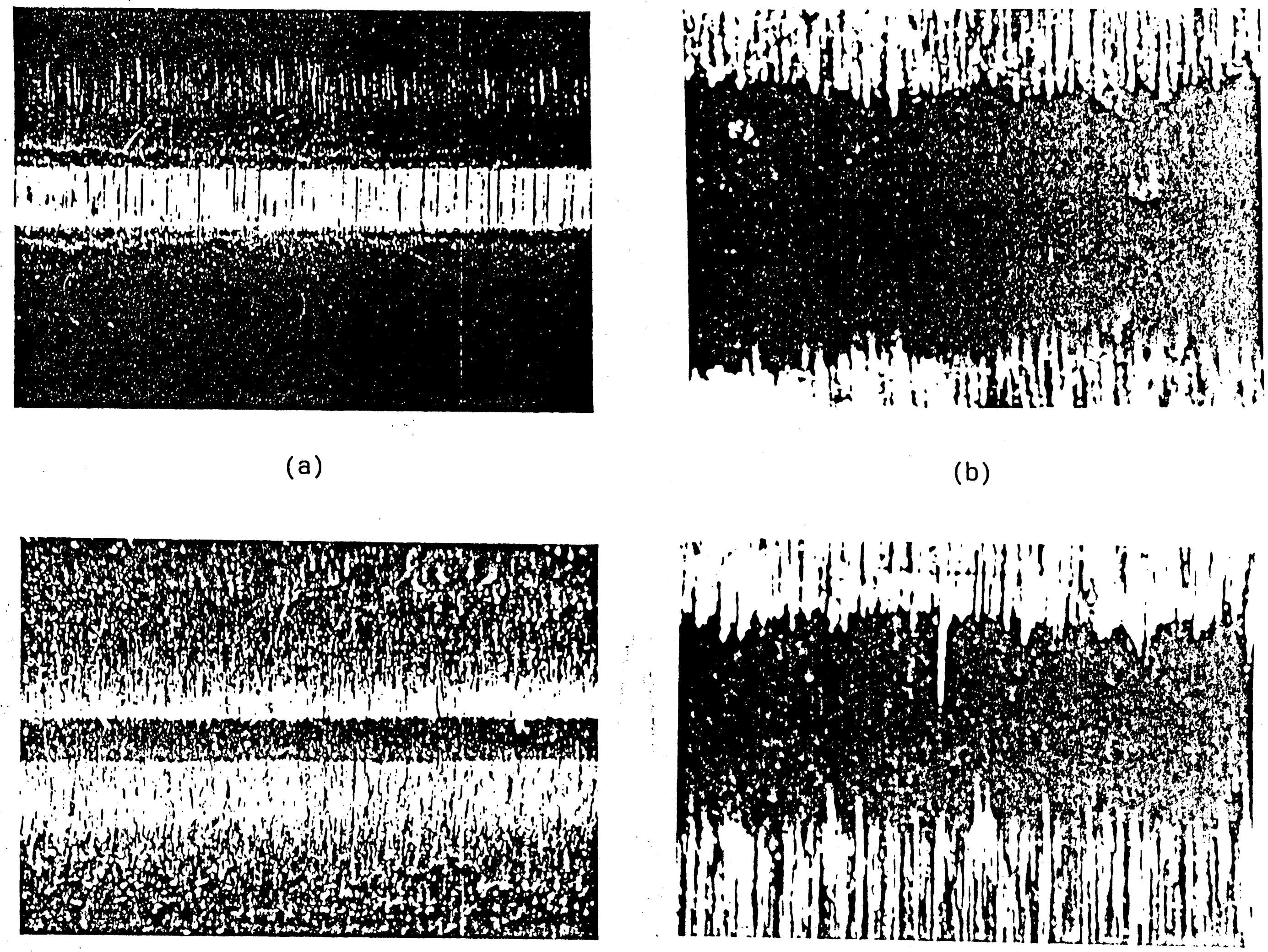


Figure 1 - Typical wear scars on the M2 steel pin.
(a) Test 86RL at magnification 10x
(b) Test 37RL at magnification 63x
(c) Test 89RL at magnification 10x
(d) Test 91RL at magnification 63x

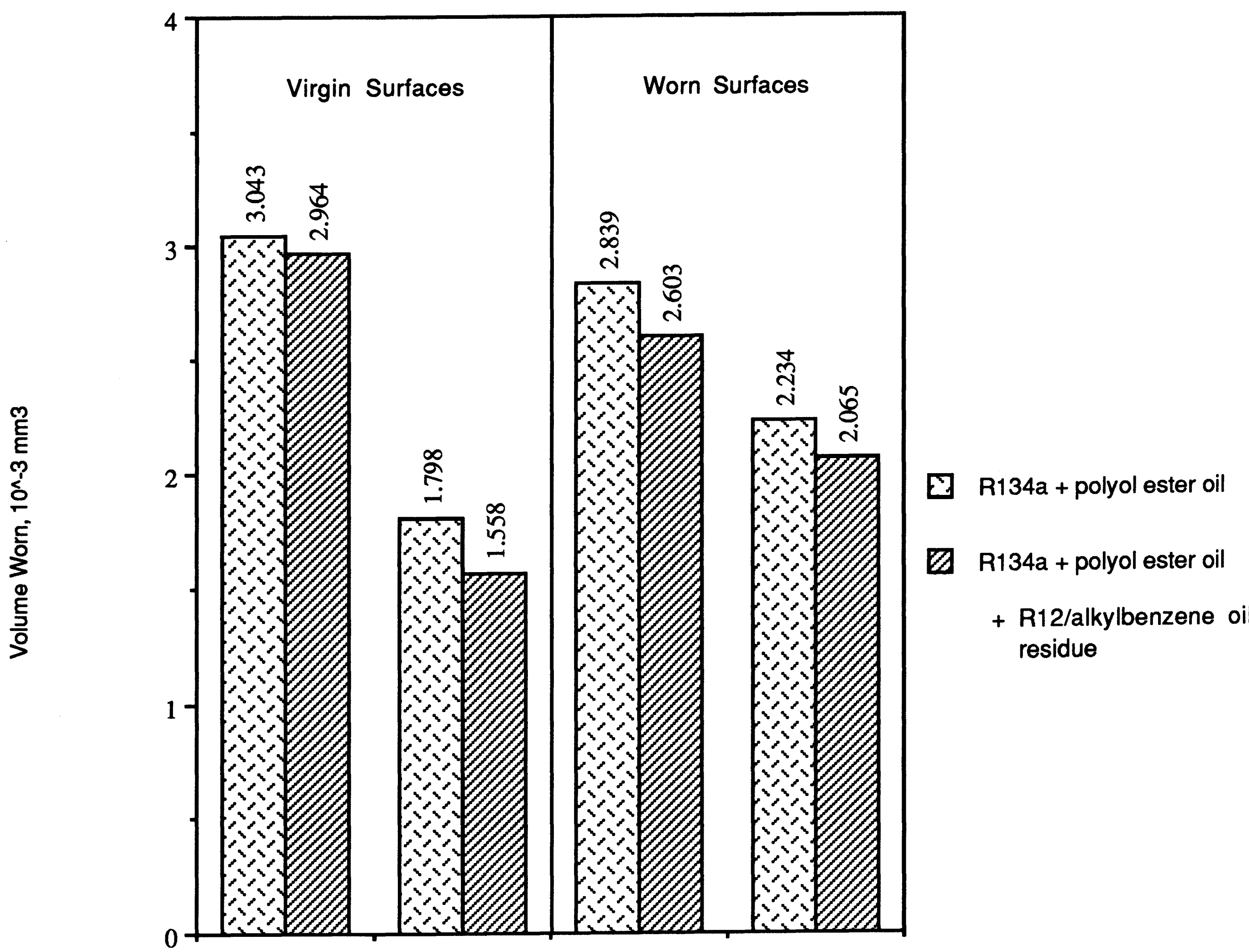


Figure 2 - Wear Results for Counterformal Contact
Duration of Each Test = 1 hour

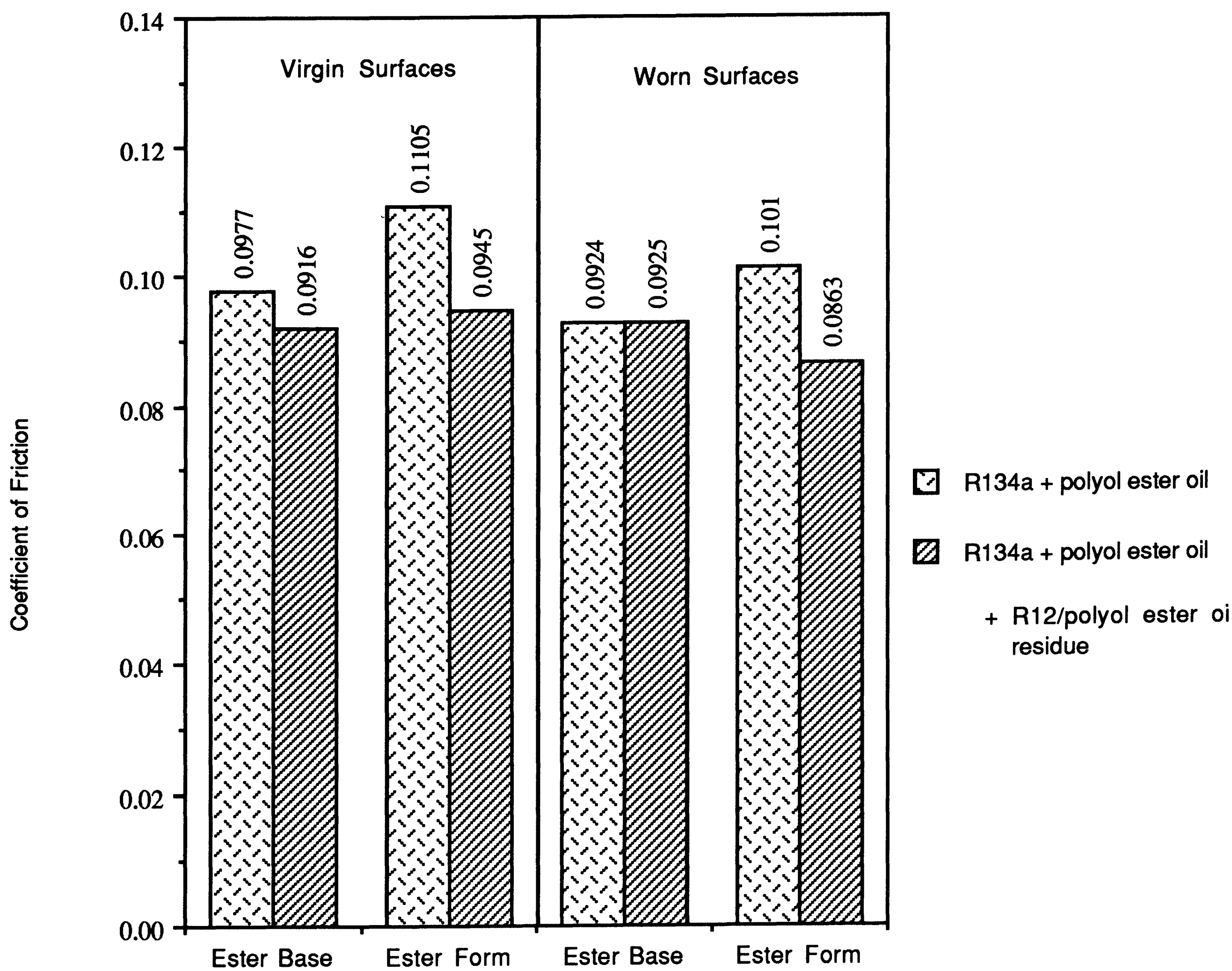


Figure 3 - Coefficient of Friction for Counterformal Contact

Duration of Each Test = 1 hour

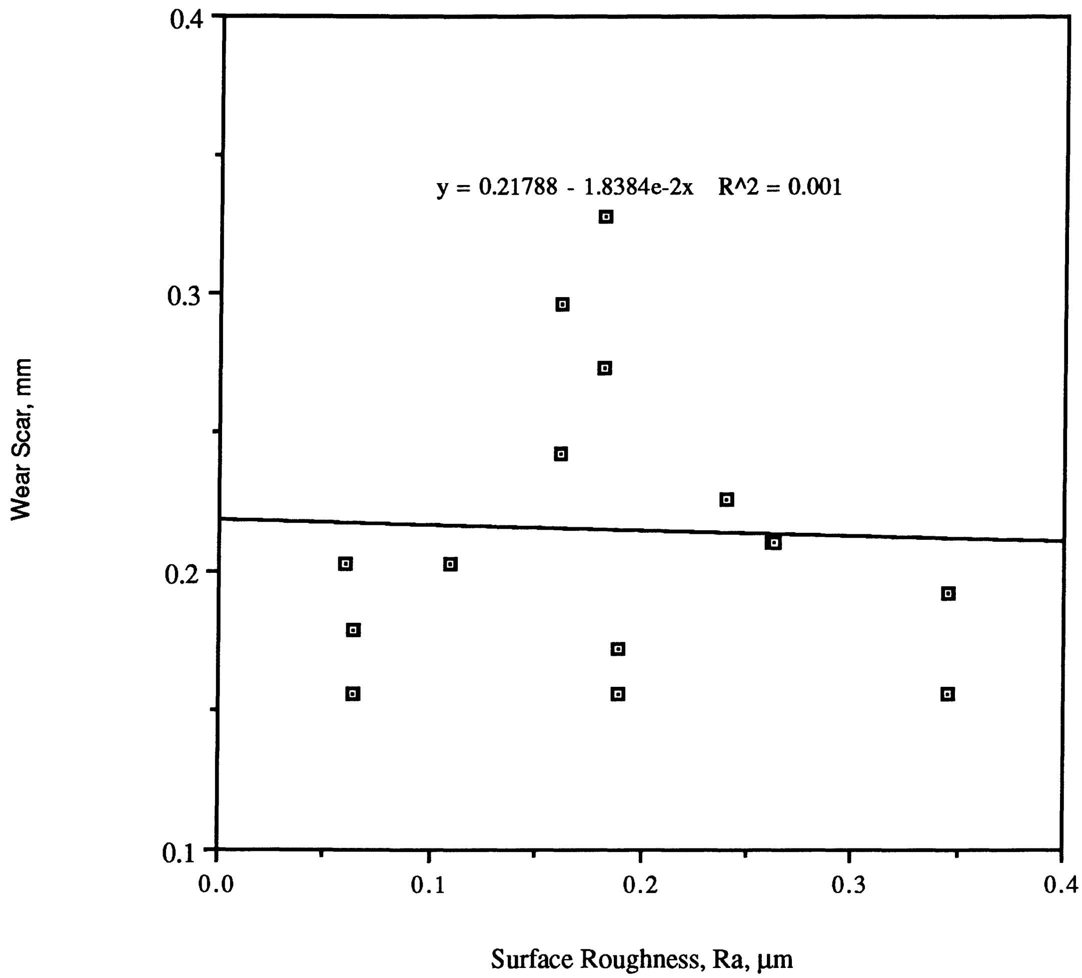
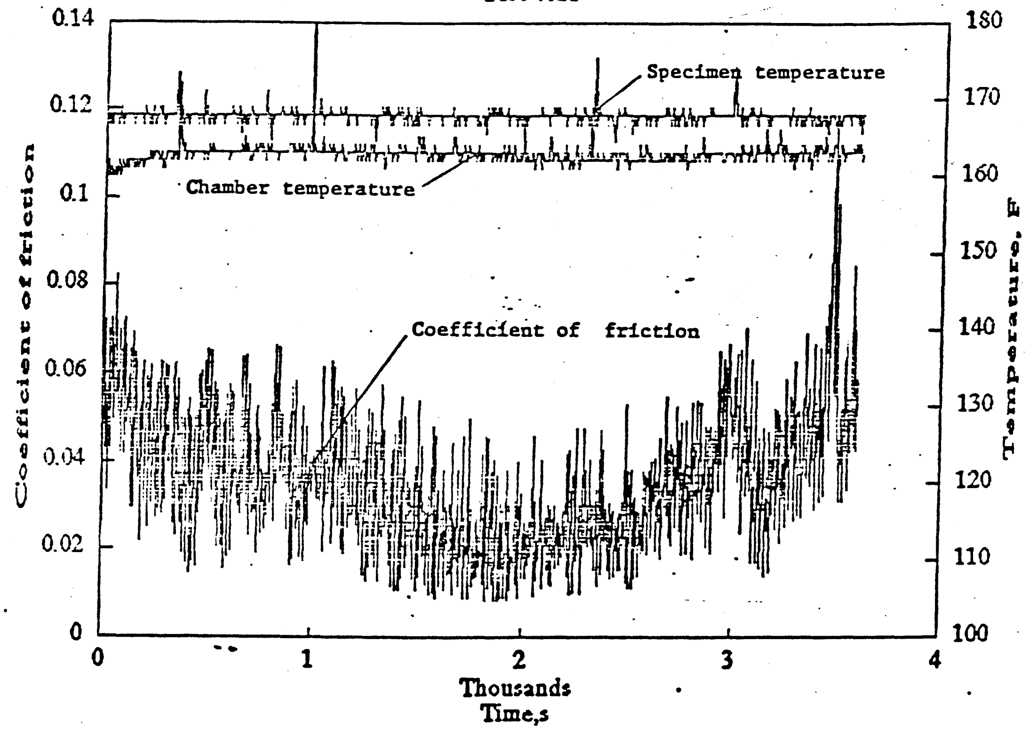


Figure 4 - Correlation Between Wear on Pin and Plate Roughness for the Counterformal Contact

Test 48SP



Test 54SP

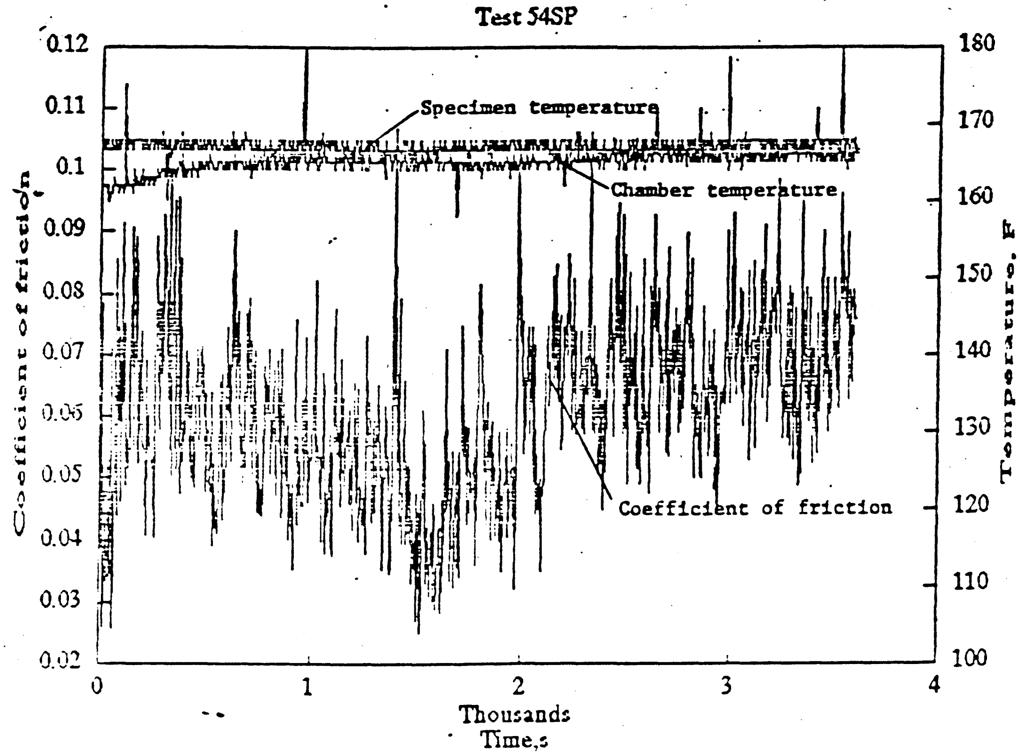
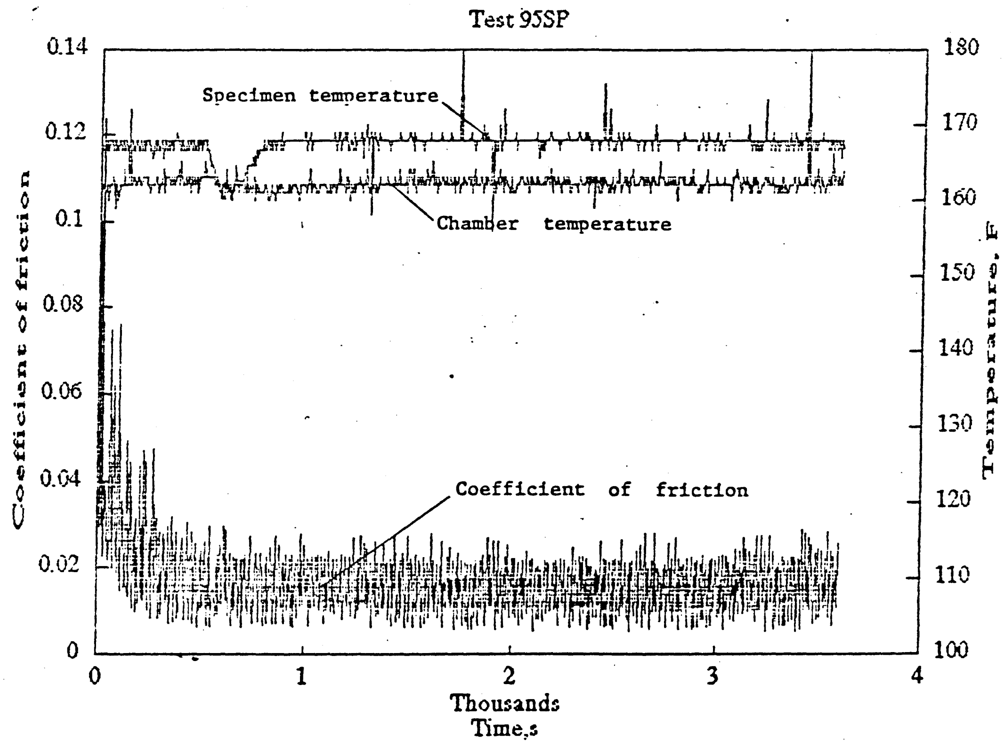


Figure 5 - Typical examples of the high friction regime for the area contact.



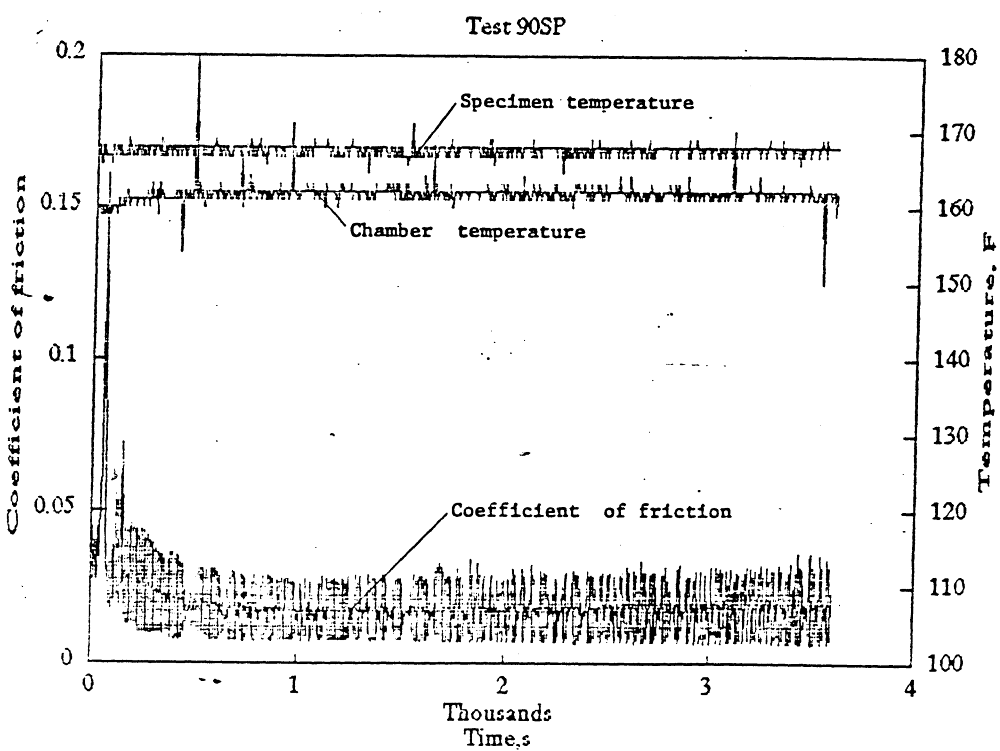


Figure 6 - Typical examples of the low friction regime for the area contact.

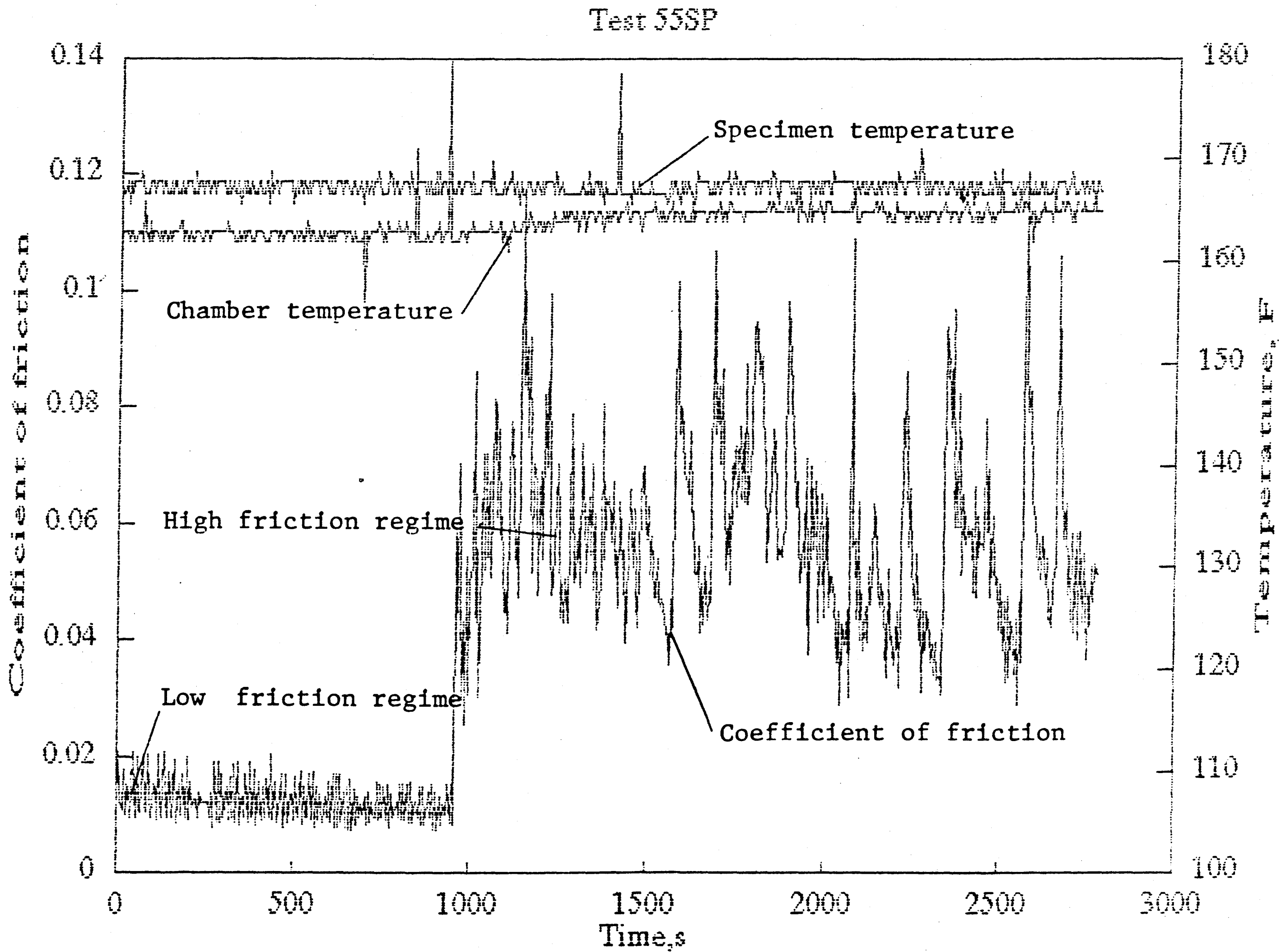


Figure 7 - Typical example of a switch from a low friction to a high friction regime for the area contact.

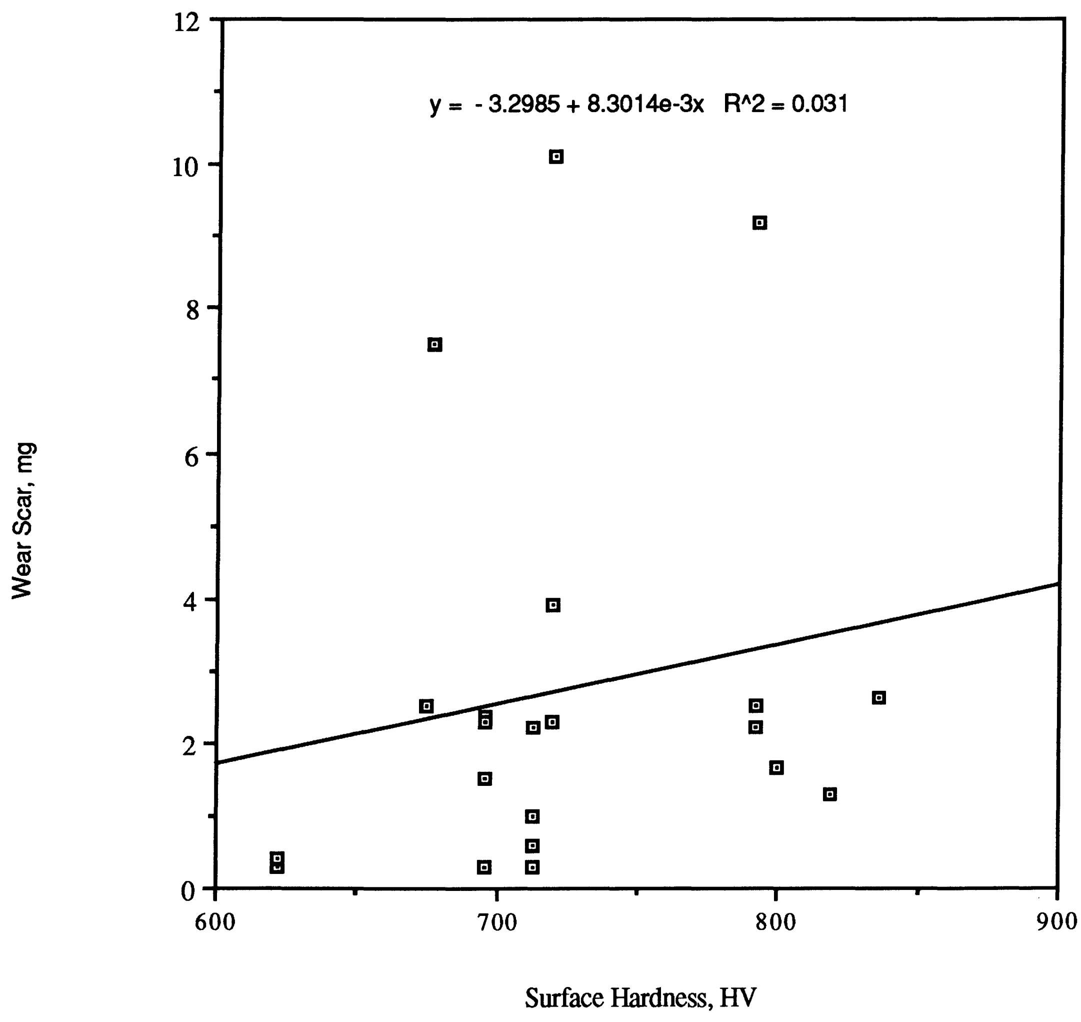


Figure 8 - Correlation Between Wear and Plate Hardness for Area Contact

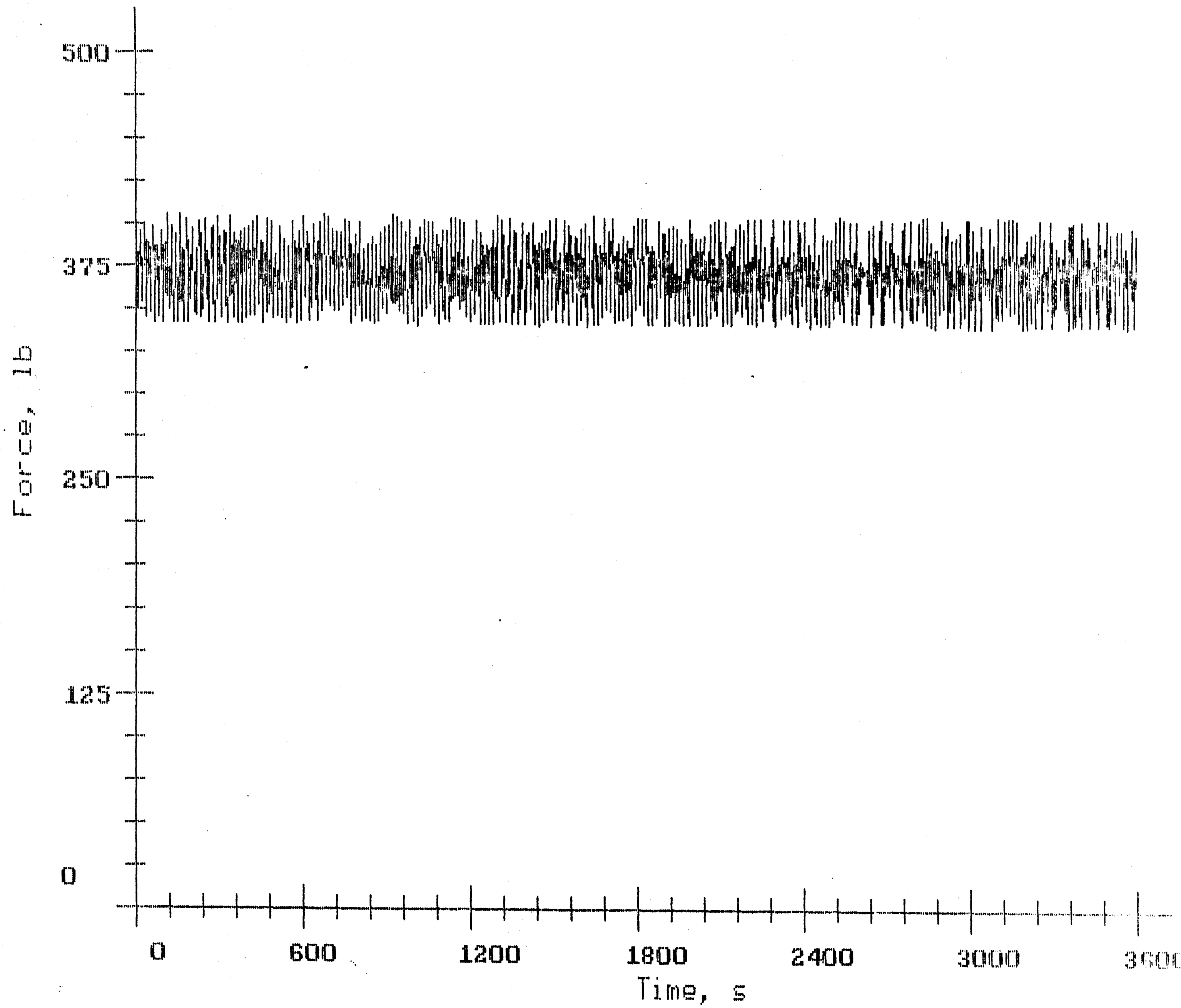
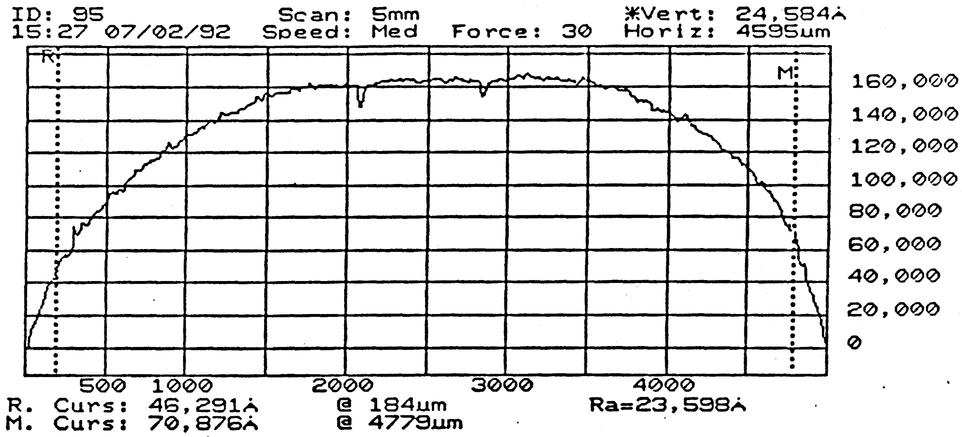
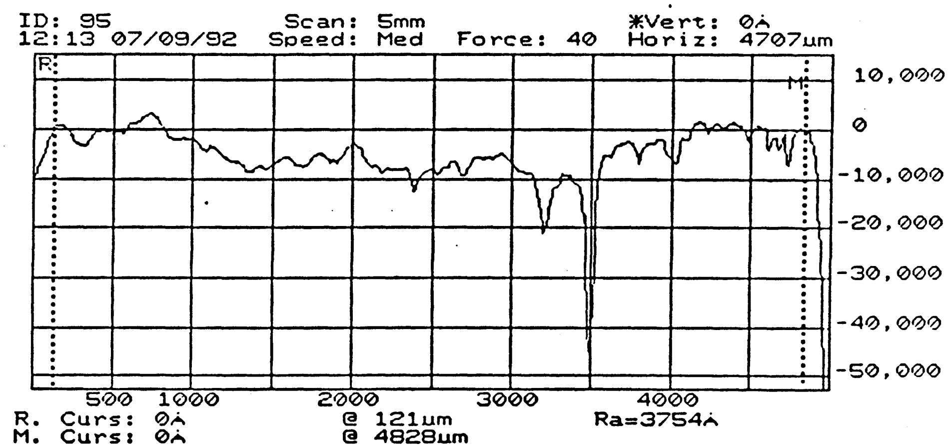


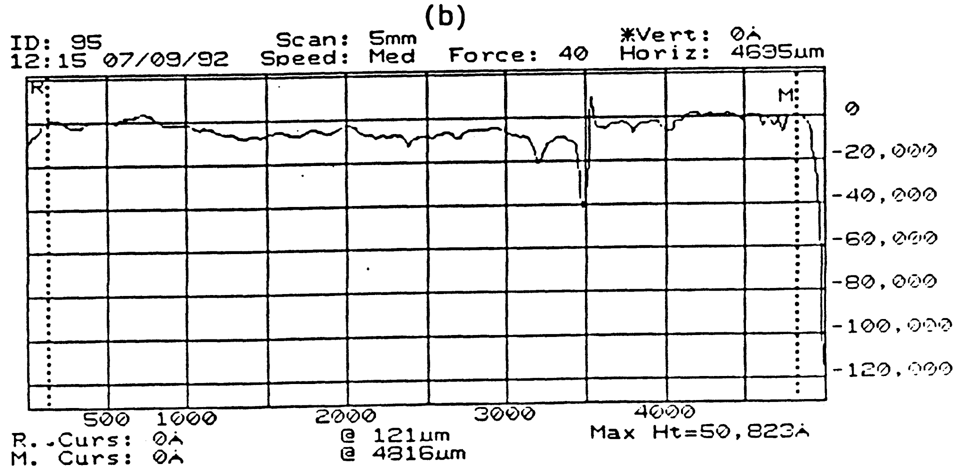
Figure 9 - Axial force record for test 97sp.



(a)



(b)



(c)

Sloan DEKTAK 3030 Rev. 1.3/2.3

Figure 10 - Surface profiles of the bronze shoe specimen.
(a) Surface profile before the test - convex shape.
(b) Surface profile after the test, measured along the direction of motion.
(c) Surface profile after the test, measured perpendicular to the direction of motion.

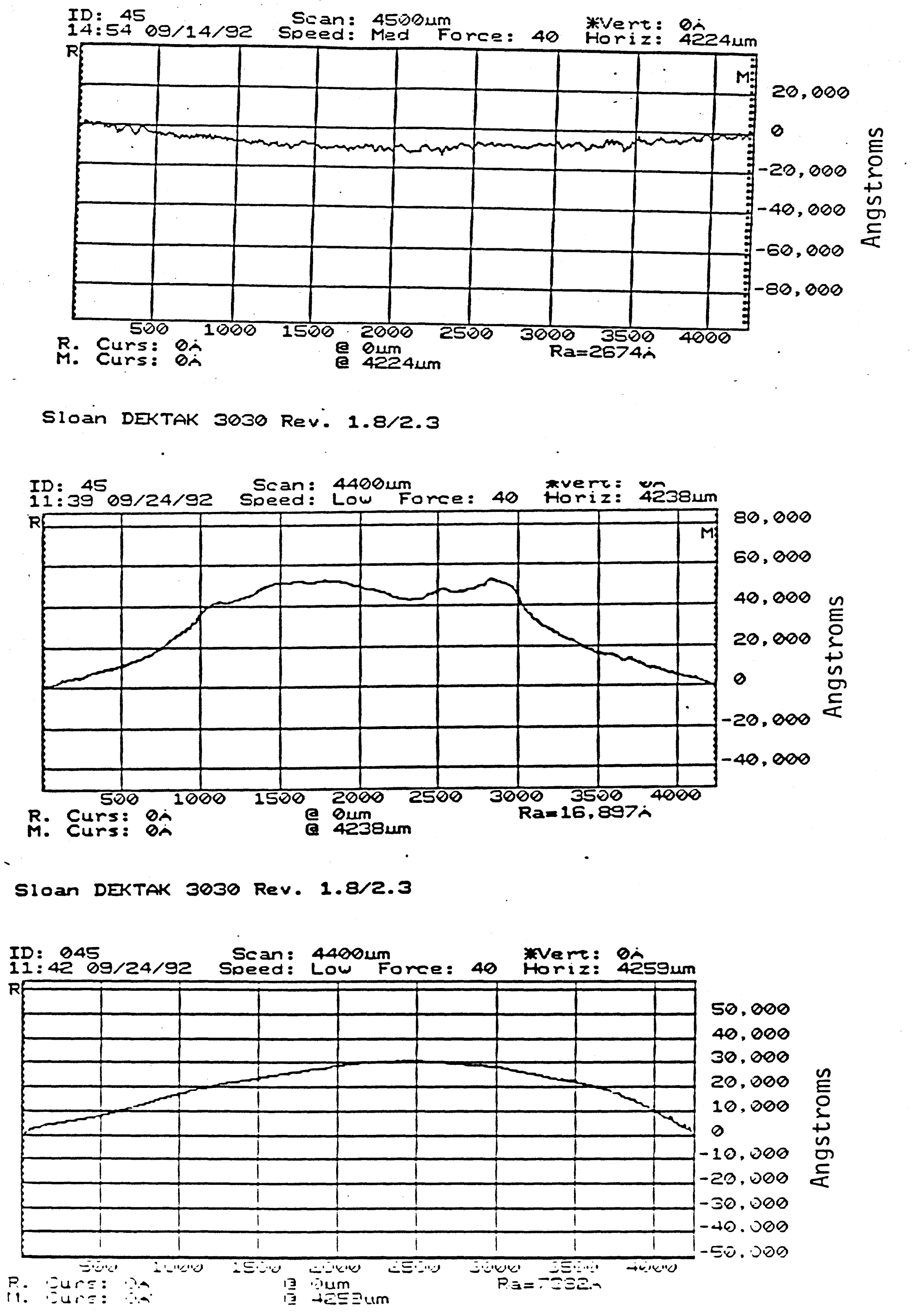


Figure 11 - Surface profiles of the bronze shoe specimen.
(a) Surface profile before the test - concave shape.
(b) Surface profile after the test, measured along the direction of motion.
(c) Surface profile after the test, measured perpendicular to the direction of motion.

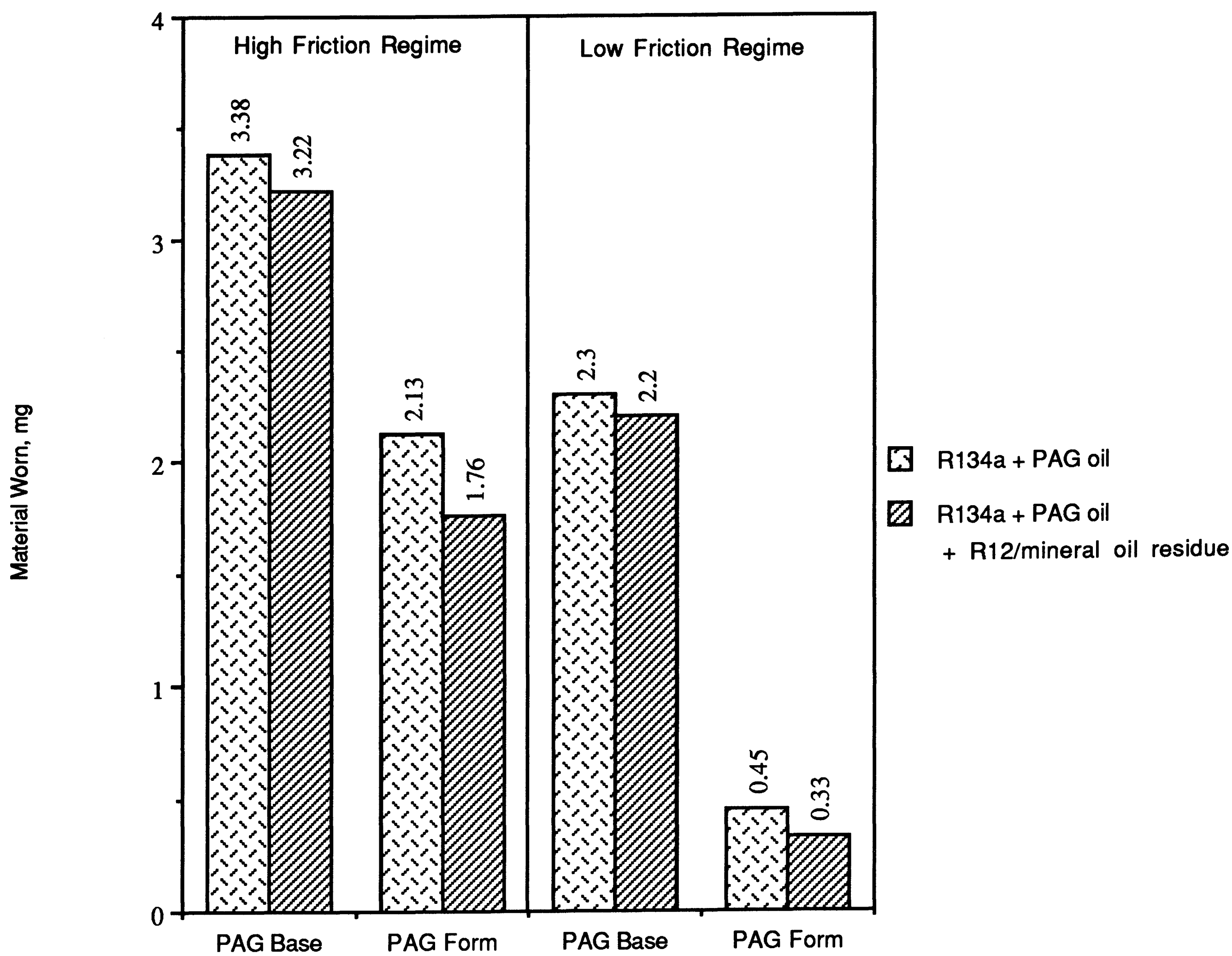


Figure 12 - Wear Results for Area Contact
Duration of Each Test = 1 hour

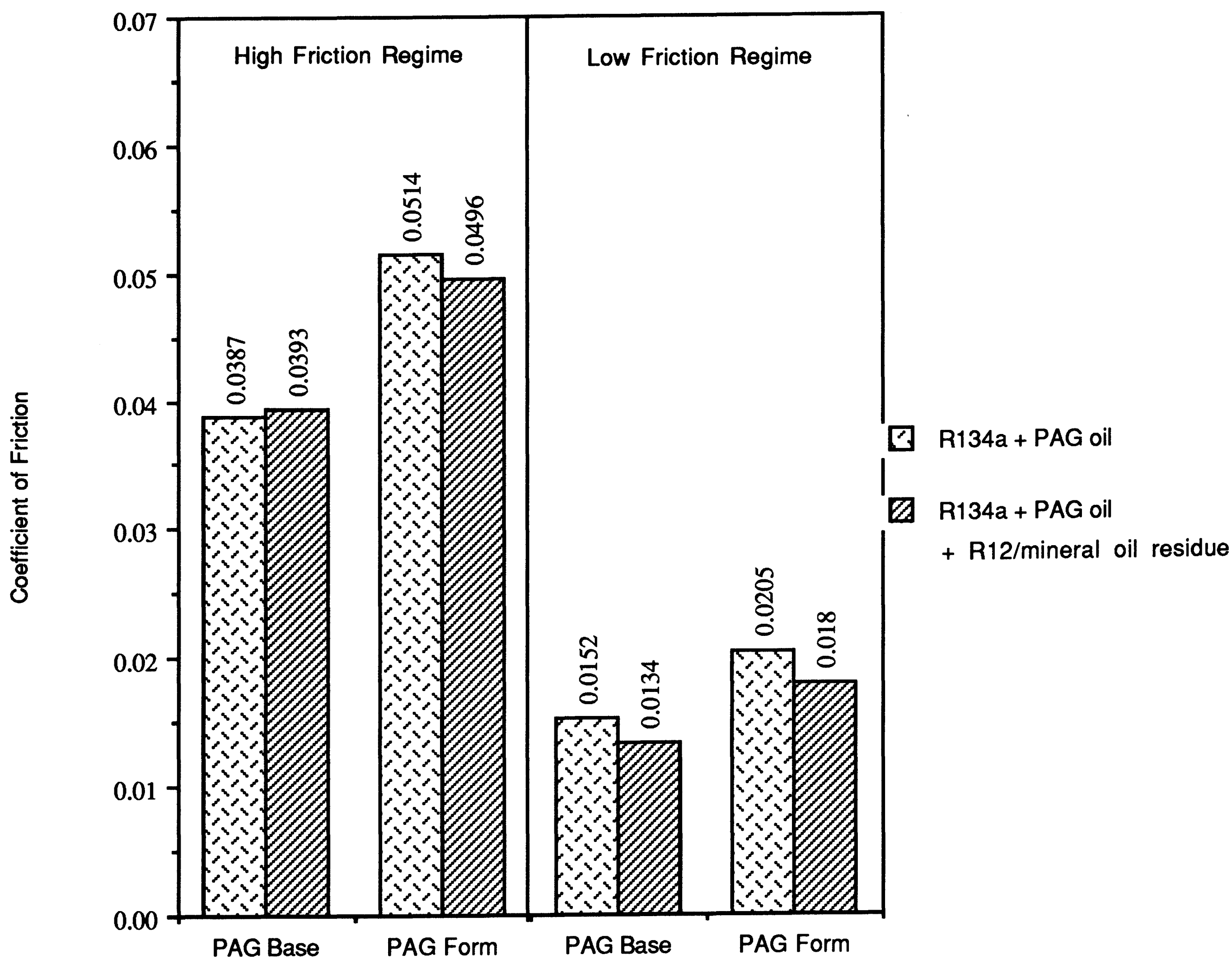


Figure 13 - Coefficient of Friction for Area Contact
Duration of Each Test = 1 hour