DOE/CE/23810-45

ACCELERATED SCREENING METHODS FOR PREDICTING LUBRICANT PERFORMANCE IN REFRIGERANT COMPRESSORS

FINAL REPORT

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November 1994

Prepared for
The Air-Conditioning and Refrigeration Technology Institute

Under
ARTI MCLR Project Number 655-51600

This project is supported, in whole or in part, by U.S. Department of Energy grant number DE-FG02-91CE23810; Materials Compatibility and Lubricants Research (MCLR) on CFC-Refrigerant Substitutes. Federal funding supporting this project constitutes 93.67% of allowable costs. Funding from non-government sources supporting this project consists of direct cost sharing of 6.33% of allowable costs; and in-kind contributions from the air-conditioning and refrigeration industry.

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EXECUTIVE SUMMARY

The friction and wear characteristics of various lubricant/refrigerant mixtures for refrigerant compressors are experimentally investigated by means of a high pressure tribometer (HPT). The HPT is a specimen tester used to approximately simulate specific critical contacts in compressors. Among the variables which are approximately simulated include: environmental pressure and temperature surrounding the contact, lubricant/refrigerant mixture, materials and hardnesses of the contact pair, contact geometry, external load (pressure) and relative motion in the contact.

This study is composed of two parts. In Part I, a comparison is made between wear data obtained from a Falex^{TM} tester and those obtained from the HPT. In Part II, data obtained from the HPT, Falex^{TM} and Four Ball specimen testers are compared to each other as well as to data obtained from accelerated component (compressor) testing. For all component and specimen tests, the ranking is based on three lubricants. A brief description of the approach taken and the results obtained is given below.

In Part I, three companies provided Falex^{TM} data for four different material pairs, three sets of three ester lubricants and two refrigerants (R134a and a blend). All these data were obtained by bubbling the refrigerants through the lubricant. Most of the data provided by these companies were qualitative (best, intermediate, worst). Lubricant rankings obtained from the Falex^{TM} tests were compared to the rankings of the same lubricants tested in the HPT. The HPT tests were conducted in a controlled environment and at lower loads than the Falex^{TM} tests. Materials, contact geometry, lubricants and refrigerants were the same for the two specimen testers. A summary of the data obtained is given in **Table 3.21** (Page 50). From these data, the following conclusions are drawn:

1. Lubricant ranking correlation between the HPT and Falex^{TM} tester is obtained only when relatively large wear differences existed between the lubricants.
2. For a given refrigerant and based on statistical significance, lubricant ranking obtained by means of the HPT was essentially the same under various loads and speeds.
3. A lubricant/refrigerant mixture, which produces relative low wear, will not necessarily produce relative low friction.
4. The ranking of the lubricants can be a function of the materials pair in contact.
5. For the operating condition examined, R134a or blend/ester mixtures generally gave higher wear than the baseline R12 or R22/mineral oil mixtures.

In Part II, a comparison is made between the wear data obtained from the HPT and the Four Ball specimen testers, and those obtained from accelerated component (compressor) tests.

Based on these data, the rankings of the lubricants obtained from the various testers are compared to each other and to the rankings of the same lubricants obtained from the component tests. The accelerated component tests were conducted by four companies. Three companies provided wear data for the wrist pin/bearing contact in a reciprocating compressor. One of these companies also provided wear data for the vane/piston contact in a rotary compressor. A fourth company provided wear data for the piston ring/cylinder contact, also of a reciprocating compressor. Each compressor was tested with three different lubricants. For the reciprocating compressors, all lubricants were esters and the refrigerant was R134a. For the rotary compressor, two alkylbenzene lubricants and a mineral oil were tested with R22. All the data supplied by these companies were qualitative (best, intermediate, worst) and, therefore, the relative wear difference obtained for the various lubricant/refrigerant mixtures is not known. All Four Ball data were obtained with the lubricant only. A summary of the data obtained in Part II is given in **Table 3.56** (Page 93). From these data, the following conclusions are drawn:

1. None of the specimen testers produced data which exactly correlated with the component data.
2. For given conditions and materials pair studied, the presence of R134a with any lubricant consistently increased wear on the specimens as compared to the same lubricant acting alone.
3. As in Part I, a lubricant/refrigerant mixture, which produces relative low wear, will not necessarily produce relative low friction.
4. The HPT data obtained also suggest that lubricant ranking is affected by environmental conditions (pressure and temperature).

The data presented in **Table 3.56** and some of the Falex™ data given in Part I are summarized in **Table 3.57** (Page 96). It should be noted that the Falex™ data given for esters 4 through 6 were obtained using a 356 aluminum which is different from the 380 aluminum used for both the components and HPT tests. For each of the other three lubricants sets for which the Falex™ data are available, the HPT, Falex™ and component data are based on the same materials pair.

Agreement between the data obtained from each of the specimen testers and the component data is approximately 65 percent. Based on these data, the HPT does not seem to be a significant improvement over more common specimen testers for screening lubricants for refrigerant compressors. Obviously, the "correctness" of the data obtained by all specimen testers is based on the assumption that the lubricant ranking obtained from the component tests is accurate. This assumption is discussed in more detail in Section 3.3.2 (Page 95).

The goal of this research was to recommend a specific bench tester, which could be used to predict lubricant performance in refrigerant compressors. The data obtained do not seem to give a clear vision about the development of a new bench tester to accomplish this goal. The HPT tests conducted in air have always given different lubricant performance and generally, different rankings than those conducted in pressurized refrigerant environments. As such, the use of the HPT is likely to be an improvement over presently used lubricant screening testers. Before the HPT can be recommended, however, simulation through specimen testing needs to be based upon more accurate operating and environmental conditions under which simulated components operate. In addition, statistically significant components wear data are required in order to make a more effective comparison to the specimen data.

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ABSTRACT

The tribological characteristics of various lubricant/refrigerant mixtures for refrigerant compressors have been experimentally investigated by means of a unique high pressure tribometer (HPT). In order to identify more effective bench testers for screening lubricants for compressors, a comparison is made between data obtained from the HPT, Four Ball and Falex™ testers, and those obtained from accelerated component (compressor) tests.

Part I of this study is geared toward a comparison between data obtained from a Falex™ specimen tester and those obtained by the HPT. The main purpose of this comparison was to determine if the controlled environment and the lower loads used with the HPT produce different rankings than those obtained from the Falex™ tests. Based on statistically significant data, and for a given materials pair and environmental pressure and temperature, the results obtained from the HPT gave a consistent ranking at different loads and speeds and this ranking did not always correlate with that obtained from the Falex™ tester.

In Part II, the HPT is used to approximately simulate specific critical contacts in compressors to determine the degree to which it could predict lubricant performance. A comparison is made between data obtained from component tests and those obtained from the HPT. For comparison purposes, each lubricant is also tested and ranked based on results obtained in an air environment with both the HPT and a Four Ball machine.

Based on wear data, the rankings of the lubricants obtained from the various testers are compared to the lubricant rankings obtained from the component tests. Assuming that the component data are correct, this comparison shows that approximately a 65 percent accuracy is obtained by means of the HPT operating in a pressurized refrigerant environment and that this accuracy is essentially the same as the accuracy obtained with the HPT operating in an air environment, the Falex™ tester and the Four Ball tester.

1. INTRODUCTION

1.1 Overview

The decrease in the production and use of ozone-depleting refrigerants has forced the air-conditioning and refrigeration industry to examine a number of possible replacements. The prime replacement for R-12 is R-134a or (1,1,1,2) tetrafluoroethane and various blends are being considered as replacements for R-22. These new refrigerants pose new problems for manufacturers, both for their thermodynamic properties as well as their tribological properties with new lubricants. Although R-134a possesses thermodynamic properties similar to those of R-12, it lacks miscibility with the lubricants typically employed with CFC refrigerants and also lacks the inherent antiwear properties of the chlorinated refrigerants. The lack of miscibility can cause both lubrication and overall system performance problems and, therefore, synthetic lubricants which are miscible or partially miscible with R-134a, are being considered for use in refrigerant compressors. Among these lubricants are polyalkylene glycols (PAGs) and polyolesters, both currently used with R-134a. However, these lubricants, in combination with the alternative refrigerants, may not provide as good lubrication compared with CFCs and mineral oil [1].

The lubricant in refrigeration systems plays an important role in the overall system efficiency and reliability. This is due to the direct interaction of the lubricant and refrigerant within the compressor and heat exchangers, as well as other parts of the system. Ever increasing technological advances in compressor and heat exchanger design and optimization have necessitated a deeper understanding of lubricant performance in the systems. In previous work [2,3], the criteria for selecting synthetic lubricants that resulted in optimal performance of refrigeration systems were reviewed. Performance properties of some synthetic lubricants for R134a have been investigated and these were found to be acceptable alternatives to mineral oils [2]. In general, manufacturers of compressors are forced to re-evaluate the tribological behavior of critical contacts in compressors, which are lubricated by these new lubricant/refrigerant mixtures, under prescribed operating and environmental conditions.

1.2 Scope of Research

Due to the rapid changes in the air-conditioning and refrigeration industry, there is an increasing interest in a more effective method of screening lubricants for compressors. The Falex™ test has been widely used in evaluating lubricity with R-12/mineral oil mixtures because it was found that a reasonable correlation exists between such tests and actual compressor tests [4]. In the case of lubricants used with R-134a, however, the correlation between the results obtained from the Falex™ tests and actual component wear tests has been found to be poor [5]. For this reason, the High Pressure Tribometer (HPT), and other specialized test machines have been designed and implemented to see if an improved correlation to actual component behavior could be obtained.

The scope of this work is to do an overall study on the methodology of specimen testing which will aid in the selection of a bench tester for screening lubricants used in compressors of air-conditioning and refrigeration systems. The approach taken was to tribologically simulate critical contacts found in compressors. Part I of this study is geared toward a comparison between data obtained by a Falex™ specimen tester and those obtained by a high pressure tribometer (HPT). In Part II, the HPT is used to obtain friction and wear data under conditions which simulate, as closely as possible, the conditions that exist in critical tribo-contacts of compressors. For comparison purposes, HPT and Four Ball data in an air environment are also obtained. Details on the approach taken are given below.

<u>Part I</u>

Obtain Falex™ data from various manufacturers of compressors, including lubricant/refrigerant mixtures and materials contact pairs used to obtain these data, and compare these data with those obtained from the HPT.

<u>Part II</u>

1. Interact with compressor manufacturers to determine which components are more likely to have tribological problems in practice.
2. Collect from the manufacturers as much tribological data as possible about these critical components. Determine from the data collected approximate operating and environmental conditions under which the components are operating, the materials in contact, contact geometry and surface topography.
3. Select representative contact geometries to be tested in the HPT.
4. Based on input from manufacturers of compressors, select lubricants to be screened and refrigerants to be used.
5. Test the contact geometries with selected lubricant/refrigerant mixtures using the high pressure tribometer (HPT), simulating as closely as possible operating and environmental conditions experienced by the corresponding components in the compressors.
6. Obtain data, for the lubricant alone, by means of both the HPT and a Four Ball tester.
7. Compare the lubricant ranking obtained in steps 5 and 6 above with the ranking obtained by the manufacturers from accelerated component tests.

Recommendations concerning a specific bench test facility which can be used for accelerated lubricant screening will be based on the correlation obtained in step 7 above, as well as the correlation obtained between the Falex™ data (Part I) and the component data.

1.3 Summary of Overall Research Program

The main emphasis of the testing program is to determine if the HPT can accurately predict lubricant performance in compressors by conducting tests under conditions, which approximately simulate component operation. The main advantage of the HPT over more standard bench testers is its ability to provide a controlled pressure and temperature environment during a test. Complexity and high costs are its main disadvantages. In order to determine if a simpler and/or less costly bench tester can predict lubricant performance to the same degree as the HPT, a comparison was made between the data obtained from the Falex™ pin and vee-block tester and the HPT to those obtained from accelerated wear in operating compressors. Four Ball and HPT tests in an air environment were also conducted with each lubricant. In this research, specific material pairings, contact geometries and lubricants of interest to the participating companies were used.

With the assistance of manufacturers of compressors, critical contacts were identified, as were the approximate operating conditions. Included among these conditions were environmental (refrigerant) pressure and temperature, contact temperature, load, velocity, materials, geometry, surface finish and lubricant type. A description of the high pressure tribometer is presented in full detail in Appendix A, while the procedures used in the testing program are given in Appendix B. Based upon the data obtained from the specimen testing program and those obtained from the accelerated components tests, recommendations are made about which bench tester, if any, is effective in screening lubricants for compressors of air-conditioning and refrigeration systems.

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2. BACKGROUND AND TECHNICAL APPROACH

2.1 General Requirements of Simulative Testing

The successful operation of compressors used in air-conditioning and refrigeration systems is mainly governed by the tribological behavior at the critical contacts within the compressor (i.e., for example, wrist pin contact in reciprocating compressors). This behavior can be examined by testing compressors under conditions that they experience in service. Such a testing program attempts to ensure that all contacts closely reproduce expected operating and environmental conditions. The main drawback to long term component testing is the time and cost requirements, especially when it is desirable to examine a large number of variables; for example, a number of lubricants or materials contact pair.

A less costly approach for screening lubricants is to conduct accelerated compressor tests where measurable component wear takes place in months rather than years. In addition to costs, the disadvantage of this approach is that both the operating and environmental conditions experienced by critical components can be significantly different than those experienced by the same components operating under normal operating conditions. Therefore, lubricant rankings obtained under accelerated conditions may be questionable.

Effective specimen testing could prove superior to component testing since both cost and time could be significantly reduced in the screening of multiple lubricants and materials. Additionally, specimen testing would likely provide an increased understanding of contact behavior since the numerous variables, which the contact experiences in practice, can be more easily controlled. Specimen testing, however, can lead to erroneous conclusions if the conditions under which the tests are conducted do not simulate, as closely as possible, the failure mode and conditions experienced by the component.

For any accelerated lubricant screening program to be effective, simulative specimen testing should be conducted. Some of the requirements of such a program include:

1. Similarity of operating variables between component and specimen.
These variables include: (a) contact kinematics, (b) load (pressure),
(c) lubricant/refrigerant mixture and (d) environmental temperature and pressure.
2. Similarity of structure between component and specimen. The structure includes: (a) material properties of contacting surfaces and (b) geometrical and contact conditions of the surfaces with appropriate scaling factors.
3. Similarity of the mode of failure, i.e., same failure mechanism occurs in both component and specimen.

One of the problems in conducting simulative specimen testing is that the component and the specimens are generally not attached to parts, which have the same geometry and material composition. This may prove critical due to the difference of the rate of heat dissipation between the specimens and component contacts. As a result, the heat generation rates due to friction might be similar, however, the temperatures experienced by the specimens and component contacts may differ significantly due to the different heat dissipation characteristics of the materials surrounding the two contacts. In simulative specimen testing, it becomes essential that the contact temperature in the component and in the test specimen be approximately the same.

Another difficulty encountered in simulative specimen testing is the necessity of obtaining measurable wear in a relatively short period of time. This problem is similar to the problem encountered when conducting accelerated compressor testing. If the load and speed actually experienced by the component are used in the specimen testing program, it is likely that no measurable wear will occur during the time frame prescribed for a test. If a different load and/or speed is used in order to obtain measurable wear, it is usually necessary to examine the effects of these parameters on wear rates to ensure that the wear mechanisms are the same in both the component and specimens.

The purpose of this discussion is to emphasize the importance of simulative testing in determining the tribological characteristics of tribo-contacts. Simulating critical contacts in compressors is further complicated by the need for equivalent environmental pressures and temperatures in both specimen tests and component tests. This is true because the pressure and temperature directly affects the amount of refrigerant dissolved into the lubricant and, therefore, the lubricating characteristics of the lubricant/refrigerant mixture. The solution to an effective lubricant screening specimen test facility for compressors, therefore, might not be a simple one.

2.2 Historical Background

As previously stated, the initial tribological evaluation of critical components in compressors of air-conditioning and refrigeration systems has mainly been based on accelerated compressor tests. Because of the time and cost requirements, standard specimen test equipment such as Falex™ or Four Ball testers have also been used as initial lubricant screening tools. The specimens tests are usually conducted by bubbling the refrigerant through the lubricant to achieve a lubricant/refrigerant mixture. Huttenlocher [1] used this approach to determine the tribological characteristics of R-12/mineral oil mixtures using a Falex™ machine. Kitaich [2] also carried out Falex™ tests in a variety of synthetic lubricants/R-134a mixtures to evaluate the lubricity of these mixtures. Espinoux, et al. [3] used Falex™, Four Ball and Plint-Cameron™ testers to evaluate and rank several lubricant candidates, which could be used with R-134a. The results show that the lubricant rankings obtained from each tester did not generally correlate with one other. Although the Falex™ tester has been utilized as a screening and development tool for many years, the validity of these tests in modeling actual compressor conditions has yet to be established with any degree of confidence. If the lubricant and refrigerant are miscible, increasing the pressure and decreasing the temperature will increase the amount of refrigerant, which will saturate into the lubricant medium. The limitations of standard testing equipment is their inability to accurately model the typical environmental conditions found in actual compressors. This generally results in lubricant/refrigerant mixtures being simulated at atmospheric pressure, where the possibility of other gases saturating into the mixture is very likely. Another major drawback to such testing methodologies is the inability of the apparatus to specifically model the different critical contact geometries as well as the kinematic conditions found within real compressors.

Sundaresan [4,5] has completed many accelerated tests in which various PAGs and esters were used with R-134a in a reciprocating piston compressor. Accelerated compressor tests have also been carried out by Davis [6] and Reyes-Gavilan [7]. Davis used reciprocating and hermetic compressors to determine the tribological characteristics of various R134a/ester mixtures, while Reyes-Gavilan [7] evaluated the lubricating ability and materials compatibility of mineral oils in reciprocating compressors charged with R-134a. In [7], alkylbenzene, PAG and ester lubricants were compared in the same types of systems. The results from accelerated compressor tests are useful in lubricant screening for specific compressors; however, the behavior of components in other types of compressors cannot be accurately inferred. Additionally, this type of testing is costly and its effectiveness in predicting proper lubricant ranking with new lubricant/refrigerant mixtures is not known.

Recent trend in the tribological evaluation of tribo-contacts in compressors has been towards the use of pressurized friction and wear machines. Such machines have been used by Sanvordenker and Gram [8] and Sanvordenker [9] who utilized the modified Falex™ machine to test R-12/mineral oil mixtures under pressure. Although this work did not model the wide range of conditions found in actual compressors, it was capable of reasonably modeling the lubricant/refrigerant mixture at low pressures. More recent innovations and advances in pressurized testing equipment for tribological evaluation have led to the development of high pressure friction and wear machines. Such a machine has been used by Komatsuzaki, et al. [10], Komatsuzaki and Homma [11], and Komatsuzaki, et al. [12], for the tribological evaluation of R-12/mineral oil, R-12/alkylbenzene, and R-134a/PAG combinations, respectively. The apparatus used in their investigation was a modified Four Ball tester equipped with a pressure chamber surrounding the contact. The pressure capability of the machine, however, was not adequate for most of the pressures found in compressors.

Since the pressure directly affects the composition of the lubricant/refrigerant mixture (i.e., the amount of refrigerant saturated into the lubricant), any testing apparatus should adequately simulate the pressure existing in real compressors. In addition, the apparatus should have the capability to simulate a wide variety of contact geometries for a wide range of operating conditions. The temperature ranges at and around the contact should be controllable as well. To satisfy most of these criteria, a high pressure tribometer (HPT) was developed as part of the Air Conditioning and Refrigeration Center (ACRC) at the University of Illinois at Urbana-Champaign. This apparatus is capable of approximately simulating environments found in most all compressors. Data obtained by means of the HPT will be used in the evaluation of existing, and possibly new, accelerated lubricant screening methods for refrigerant compressors.

2.3 Technical Approach

The decision making process used to recommend a bench facility, if any, for the accelerated screening of lubricants is based on a specimen testing program which simulates, as closely as possible, conditions which exist at the tribo-contact of the component. It is believed that this approach, even though more complex and costly than most standard specimen testing apparatus, has the best chance of predicting component behavior in practice. The proposed approach is based on the fact that the tribological behavior of a tribo-contact, under boundary lubrication conditions, is a complex function of the following: material properties and microstructure, lubricants and lubricant additives, refrigerants, contact geometry, interfacial temperature, contact kinematics, load, initial surface roughness and environmental pressure and temperature. In many testing machines, lubricants cannot be exposed to refrigerant environments at controlled pressures and temperatures, usually only one contact geometry can be tested and the kinematic conditions are restricted to only rotational or oscillatory motion. Additionally, the contacts generally experience very high friction, which generates very high interfacial temperatures not found in the operation of the components being simulated. Therefore, the tribological data obtained from these testing machines may not predict lubricant performance in a compressor based on some of the following limitations. First, the amount of refrigerant in a lubricant, which is a function of pressure and temperature, can significantly affect the lubricating characteristics of the lubricant. Second, the state of stress, generation of lubricating films and local surface deformation which, in turn, determines lubrication regimes (i.e., hydrodynamic, elastohydrodynamic or boundary), are all functions of the contact geometry. Third, the generation of lubricating films and surface damage, under boundary lubrication conditions, depends on the kinematic conditions of the contacts; i.e., unidirectional and cyclic motion might give different results. Finally, interfacial temperature should be simulated as closely as possible since it plays a critical role in the tribological behavior of lubricants, especially those, which are formulated.

A select number of compressor manufacturers have provided the current research program with qualitative lubricant rankings based upon limited Falex^{TM} tests and "in-house" accelerated component (compressor) tests. Additionally, estimated operating and environmental conditions found in the component tests were provided by these manufactures and were approximately simulated with the HPT.

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3. SIGNIFICANT RESULTS

3.1 PART I: HIGH PRESSURE TRIBOMETER vs. FALEX™ TESTER

3.1.1 Overview

As previously stated the primary goal of Part I of this project is to collect Falex™ data from a few manufacturers and compare these data with those obtained from the HPT. With the exception of load and environmental pressure and temperature, specimen testing is conducted under approximately the same conditions in both Falex™ and HPT testers. For the Falex™ data provided, the refrigerant was bubbled through the lubricant, the temperature was not controlled and the tests were conducted at relatively high contact loads. A schematic configuration of the Falex™ tester is shown in Appendix A (**Figure A.6**). The loads used to conduct tests with the HPT are smaller than those used on the Falex™ and the environments chosen for the HPT are as closely as possible to those found in specific contacts in compressors.

3.1.2 Material Pairings and Contact Geometry

The various contact geometries and material pairings used in the HPT tests are shown in **Table 3.1**. These contact geometries and materials are the same as those used to obtain the Falex™ data by the manufacturers. Two companies supplied information for only one material pairing and one contact geometry (Cases 1 and 2), while another company offered data for two different contact material pairings (Cases 3 and 4).

The contact geometries for the specimens used in the HPT are shown in **Figure 3.1** for the unidirectional motion and **Figure 3.2** for the oscillatory motion. The lower specimen is secured in place by a specimen holder for the unidirectional tests; however, both the upper and lower specimens are mounted in specimen holders for the oscillatory tests. Specimen holders are shown in Appendix B.

Table 3.1- Specimen Data for HPT (Material Pairings and Contact Geometry)

Description	Case 1	Case 2	Case 3	Case 4
Geometry: upper specimen	3 in. dia. Disk	3 in. dia. Disk	1 in. x 1 in. Pad	3 in. dia. Disk
Geometry: lower specimen	0.25 in. dia. by 0.375 in. long Pin	0.25 in. dia. by 0.375 in. long Pin	2.25 in. rad. by 0.375 in. long Pin	2.25 in. rad. by 0.375 in. long Pin
Materials: upper specimen	Gray Cast Iron	356 Die Cast Aluminum	380 Die Cast Aluminum	Gray Cast Iron
Materials: lower specimen	333 Die Cast Aluminum	Hardened Drill Rod	Carburized 1018 Steel	Carburized 1018 Steel
Ave. Hardness: upper specimen	187 Hv	67 Hv	83 Hv	187 Hv
Ave. Hardness: lower specimen	113 Hv	886 Hv	817 Hv	817 Hv
Surface Topo: upper specimen	Ground	Ground	Ground	Ground
Surface Topo: lower specimen	Ground	Ground	Ground	Ground
Ave. Surface Finish: upper specimen	0.047 μm Ra	0.406 μm Ra	0.188 μm Ra	0.042 μm Ra
Ave. Surface Finish: lower specimen	0.603 μm Ra	0.286 μm Ra	0.170 μm Ra	0.101 μm Ra

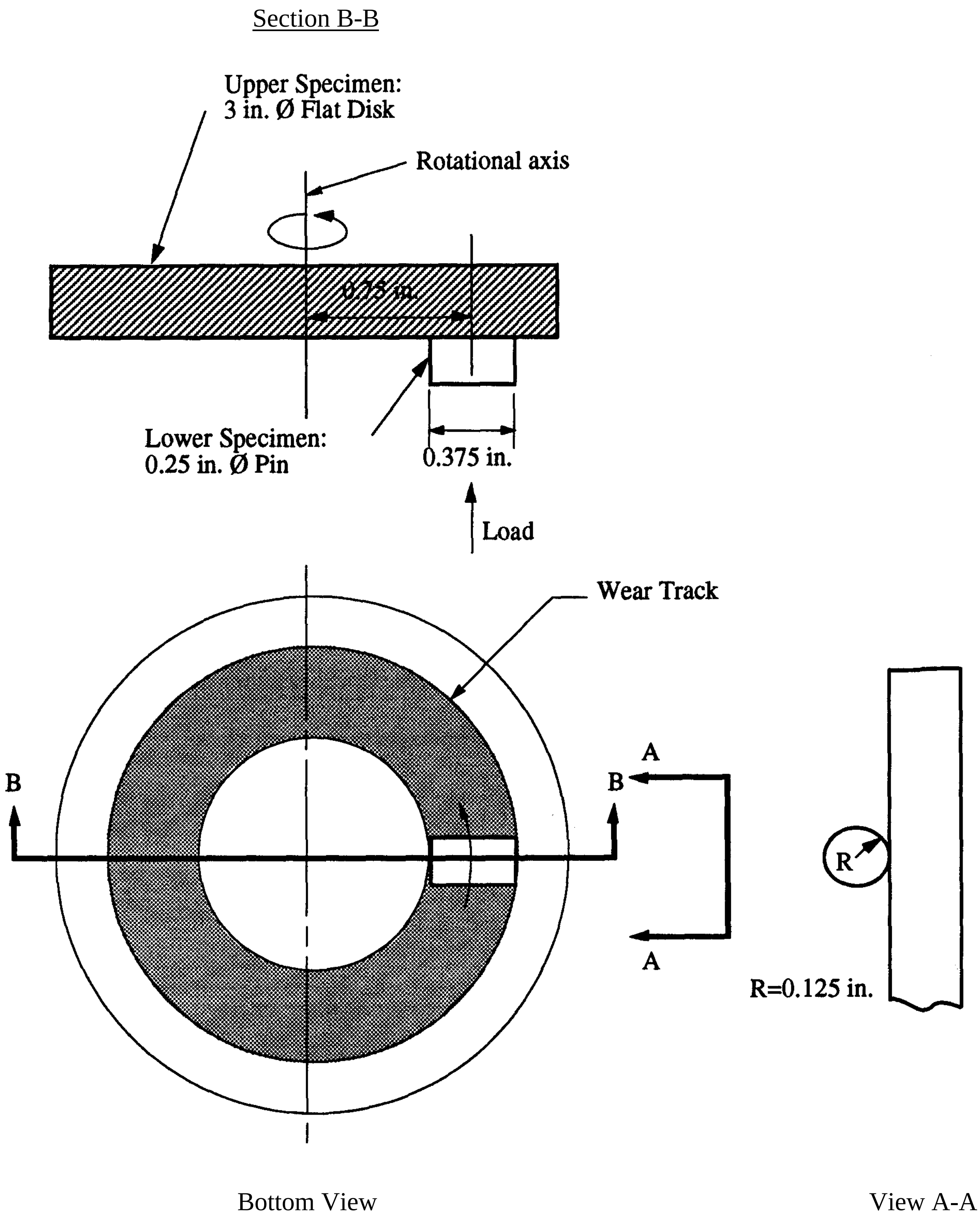


Figure 3.1- Contact Geometry of Specimens for Unidirectional Motion

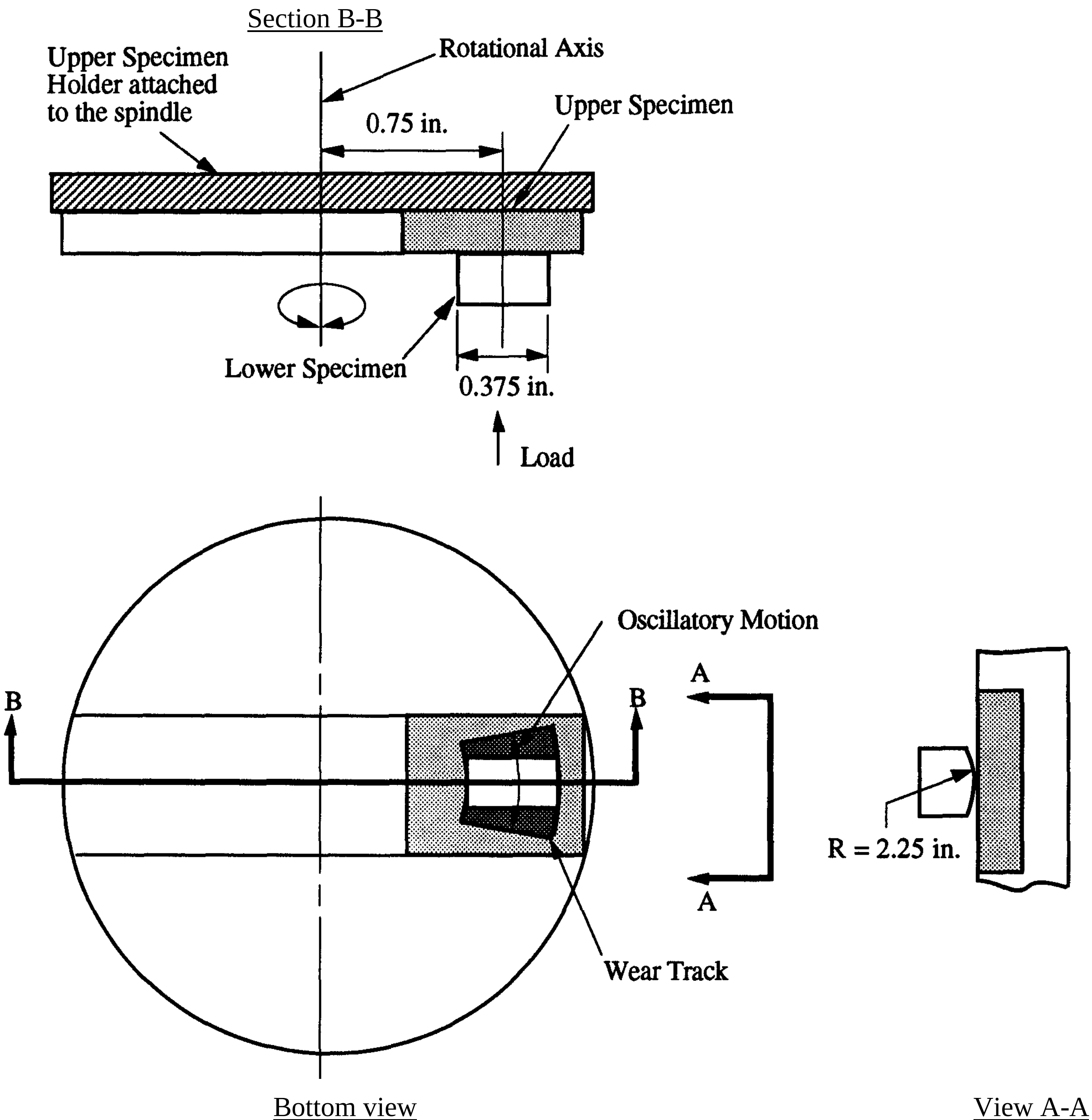


Figure 3.2 - Contact Geometry of Specimens for Oscillatory Motion

3.1.3 Environmental and Operating Conditions

As previously stated, tests on the HPT are conducted under environmental conditions which approximately simulate those existing in critical contacts of compressors. Some estimated environmental and operating conditions found in critical contacts (supplied by the companies) and the environmental and operating conditions used in the HPT are tabulated in **Table 3.2** and **Table 3.3**, respectively.

The environmental conditions of Case 1 simulate a critical contact in a scroll compressor, whereas, conditions for Cases 2 through 4 simulate the wrist pin contact in reciprocating compressors. From **Table 3.2**, it is noted that the wrist pin experiences temperatures above 266°F, however, this is the maximum temperature, which can be accommodated by the HPT. It should be emphasized that the minimum loads used for the HPT are based on obtaining measurable wear. The lubrication conditions which results from these loads are much more severe than those expected in the component, but not as severe as those used in the Falex™ tests. Different loads and speeds were chosen to see their effects on lubricant ranking.

For Case 3, due to the unavailability of 380 die cast aluminum in a size required to conduct unidirectional tests, oscillatory motion was used to conduct tests with this material. This motion, at a frequency of 4.61 Hz, amplitude of ±15° and mean radius of 0.975 in., gives an average speed of 3.8 in./s, the unidirectional speed used with the Falex™ tests. For Case 4, a speed of 15.9 in./s was used in order to obtain measurable wear in the test duration time of one hour.

The loads used to conduct the Falex™ tests are given in **Table 3.4**. These loads were not used with the HPT since the HPT does not have the capability to generate such loads and, more importantly, such loads might not be realistic since they produce stresses which are much higher than those experienced by the components.

Table 3.2 - Component Environmental and Operating Conditions Provided by the Companies

Operating Conditions	Case 1	Case 2	Cases 3 and 4
Compressor Type	Scro.C.	Rec.C.	Rec.C.
Components	TB:OC	WP:B	WP:B
Pressure	300 psi	4,000* psi	2,500* psi
Env. Pressure (psig)	85	30	7
Env. Temperature (°F)	70	275	325

Scro.C. = Scroll Compressor; Rec.C. = Reciprocating Compressor
TB:OC = Thrust Bearing:Oldham Coupling; WP:B = Wrist Pin:Bearing
* Load per Unit Projected Area

Table 3.3 - Environmental and Operating Conditions for HPT

Operating Conditions	Case 1	Case 2	Case 3	Case 4
Contact Load (lbf)	185 370	50 100	50	300 400
Max. Initial Contact Stresses (psi)	91,200 129,000	53,800 76,100	12,700	34,520 39,870
Type of Motion	Unidirectional	Unidirectional	Oscillatory	Unidirectional
Speeds (in./s)	3.8 23	3.8 23	± 3.8 (Average)	15.9
Angular Frequency	—	—	4.61 Hz	—
Angular Amplitude	—	—	± 15 °	—
Env. Pressure (psig)	85	30	7	7
Env. Temperature (°F)	70	266	266	266
Test Duration	1 hr	1 hr	1 hr	1 hr

Table 3.4 - Load and Speed used in the Falex™ Tester Provided by the Companies

Description	Case 1	Case 2	Case 3	Case 4
Applied Initial Load on the Lever Arm (lbf)	350	141	550	250
Contact (Normal) Load at Interface (lbf)	2,120	854	3,330	1,515
Max. Initial Contact Stress (psi)	273,900	198,600	145,160	97,800
Speed (in./s)	3.8	3.8	3.8	

3.1.4 Lubricant/Refrigerant Mixtures

The viscosity data for the lubricants are given in **Table 3.5.** Those lubricants, which are known to have the same base, are designated by "#". The various lubricant/refrigerant combinations studied are given in **Table 3.6.** The blend shown in the table consists of 30% R-32, 10% R-125, and 60% R-134a. The lubricants that have been evaluated are mineral oils and various esters. Mineral oils were used with R-12 and R-22 for obtaining baseline friction and wear data, while several ester lubricants were used with R-134a and blend refrigerants. Lubricants used to obtain the baseline data are those commonly used with R-12 and R-22. Again, both the Falex™ and HPT data were obtained by using the same lubricants.

Table 3.5 - Lubricant Viscosity

Lubricant	Type	Viscosity (cS)	
		@ 40 °C	@ 100 °C
Mineral 1	P	59.7	6.8
Mineral 2	N	29.8	4.4
Ester 1	PE	33.5	6.2
#Ester 2	PE	23.9	4.9
Ester 3	PE	74.0	8.9
#Ester 4	PE	23.9	4.9
#Ester 5 *	PE	23.9	4.9
Ester 6 *	PE	32.0	5.6
#Ester 7 *	PE	23.9	4.9
Ester 8	PE	32.0	5.4
Ester 9	PE	32.0	5.6
* Formulated; P: Paraffinic; N: Naphthenic; PE: Polyolester; # Same Base Lubricant			

Table 3.6 - Lubricant/Refrigerant Combinations

Company	Refrigerant	Lubricant
Case 1	R-22 (baseline) R-134a R-134a R-134a Blend Blend Blend	Mineral 1 (baseline) Ester 1 Ester 2 Ester 3 Ester 1 Ester 2 Ester 3
Case 2	R-12 (baseline) R-134a R-134a R-134a	Mineral 2 (baseline) Ester 4 Ester 5 Ester 6
Cases 3 and 4	R-12 (baseline) R-134a R-134a R-134a	Mineral 2 (baseline) Ester 7 Ester 8 Ester 9

3.1.5. Results and Discussion

The contact geometry and material pairings given in **Table 3.1**, with the lubricant/refrigerant mixtures given in **Table 3.5** were tested in the HPT with the environmental and operating conditions given in **Table 3.3**. Comparative tests with R-134a and the blend refrigerants with various esters were completed for the same contacts and operating conditions as the baseline tests (R12 or R22/mineral oil mixtures). In general, these tests show worse wear results than the baseline tests.

For each operating condition, the HPT was used to conduct two tests for each lubricant/refrigerant mixture. The friction coefficients, 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 was the average value of the friction coefficient data points throughout the test. These data points are collected at a frequency of 300 samples per second. The amount of wear was evaluated using one of the methods described in Appendix C. The wear data obtained were compared with the Falex™ wear data supplied by the companies. The statistical significance of all HPT data obtained is discussed at the end of each Case.

3.1.5.1 Case 1: HPT Friction and Wear Results - 333 Aluminum Pin on Gray Cast Iron Disk (Scroll Compressor)

The reported wear data were obtained by averaging the wear scar widths of the lower specimen (333 die cast aluminum pin) as discussed in Appendix C. A typical wear scar on the aluminum pin is shown in **Figure 3.3**. The widths of the wear scars are uniform along the length of the specimen. The surface of the mating piece, the gray cast iron plate, showed only minor polishing wear. This wear probably resulted in slight but negligible changes in the surface finish; therefore, no measurements were taken. Surface roughness of the plate specimens is considered an important parameter since it affects the wear behavior of the contacts. Therefore, in order to minimize the effects of surface roughness, care was taken in maintaining a consistent surface finish from specimen to specimen. The range of the surface roughness readings was from 0.0321 µm to 0.0592 µm Ra with the average of 0.0466 µm.

Figure 3.4 illustrates the friction and wear data collected by means of the HPT for two different speeds and loading conditions. For all friction and wear data, the speed and loading conditions are specified. The average volume loss on the pin is reported. The raw data, given in the Appendix D, show an average scatter of wear data between repeated tests of 5.87%. This suggests that the repeatability of experimental data is quite good. The repeatability of the coefficient of friction for all tests is within 7%.

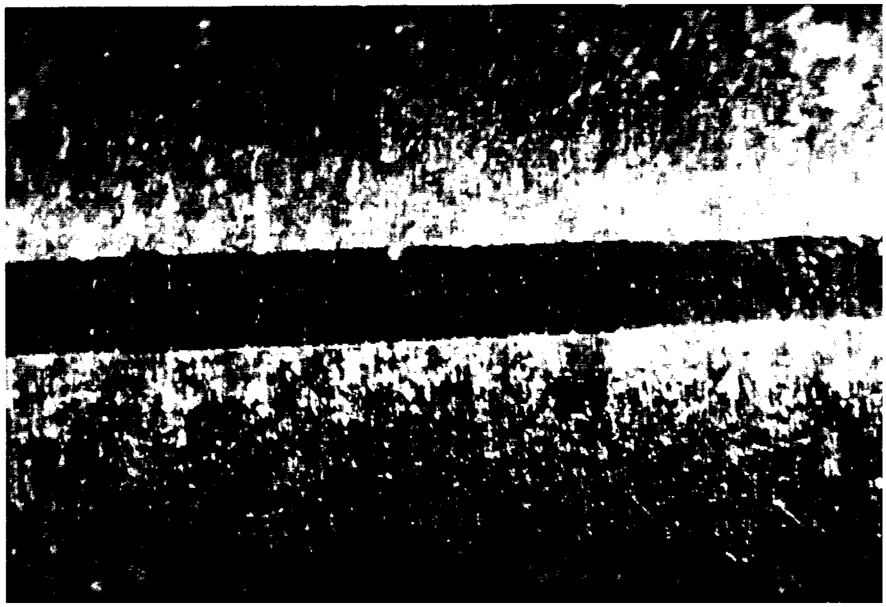


Figure 3.3 - Typical Wear Scar on the Pin

A comparison between the data obtained from the Falex™ tester and those obtained from the HPT is summarized in **Table 3.7.** The results obtained from the HPT were for three different combinations of load and speed. These test conditions are also shown in the table and the ranking of the lubricants determined by use of the HPT are compared for the different operating conditions. Note that for a given operating and environmental condition, materials and refrigerant, a set of three lubricants were tested and ranked by their wear behavior. Lubricants, which have an "ns" in parenthesis within this set, indicate that their relative wear difference is not significant enough to differentiate their relative wear behavior.

Table 3.7-- Ranking of Lubricants by Wear (Case 1)

Lubricant	Ref	Falex™ 3.8 in./s at 2,120 lbf	HPT 3.8 in./s at 185 lbf (mm^3)	HPT 23 in./s at 185 lbf (mm^3)	HPT 23 in./s at 370 lbf (mm^3)
Ester 1	R-134a	na	0.016 (ns)	0.019 (ns)	0.043 (B)
Ester 2	R-134a	na	0.013 (ns)	0.018 (ns)	0.049 (I)
Ester 3	R-134a	na	0.014 (ns)	0.020 (W)	0.061 (W)
Ester 1	Blend	(W)	0.020 (ns)	0.023 (W)	0.054 (ns)
Ester 2	Blend	(B)	0.019 (ns)	0.020 (I)	0.052 (ns)
Ester 3	Blend	(I)	0.011 (B)	0.013 (B)	0.046 (ns)

ns : Statistically Not Significant; na: Not Available; B: Best; I: Intermediate; W: Worst

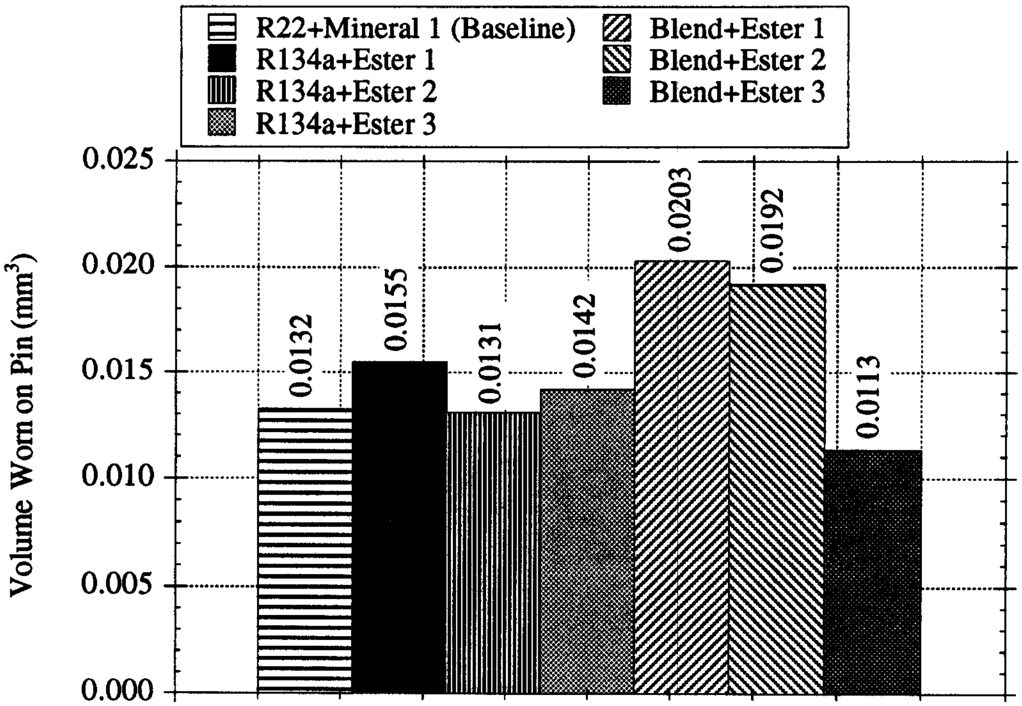


Figure 3.4a - Wear Results (3.8 in./s at 185 lbf)

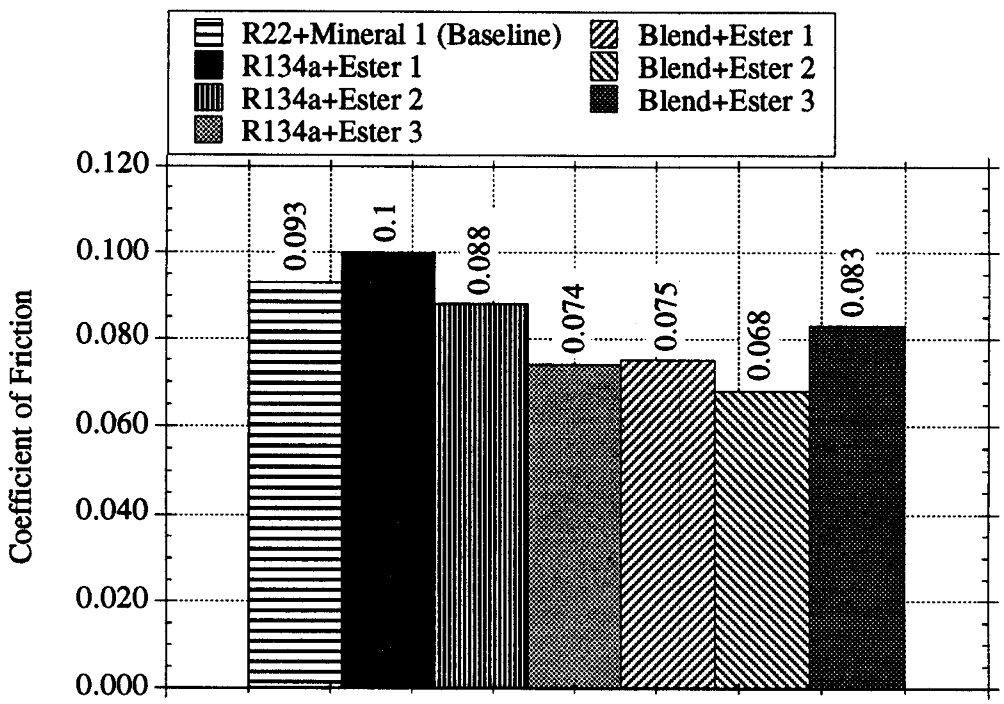


Figure 3.4b - Coefficient of Friction (3.8 in./s at 185 lbf)
Duration of Each Test =1 Hour

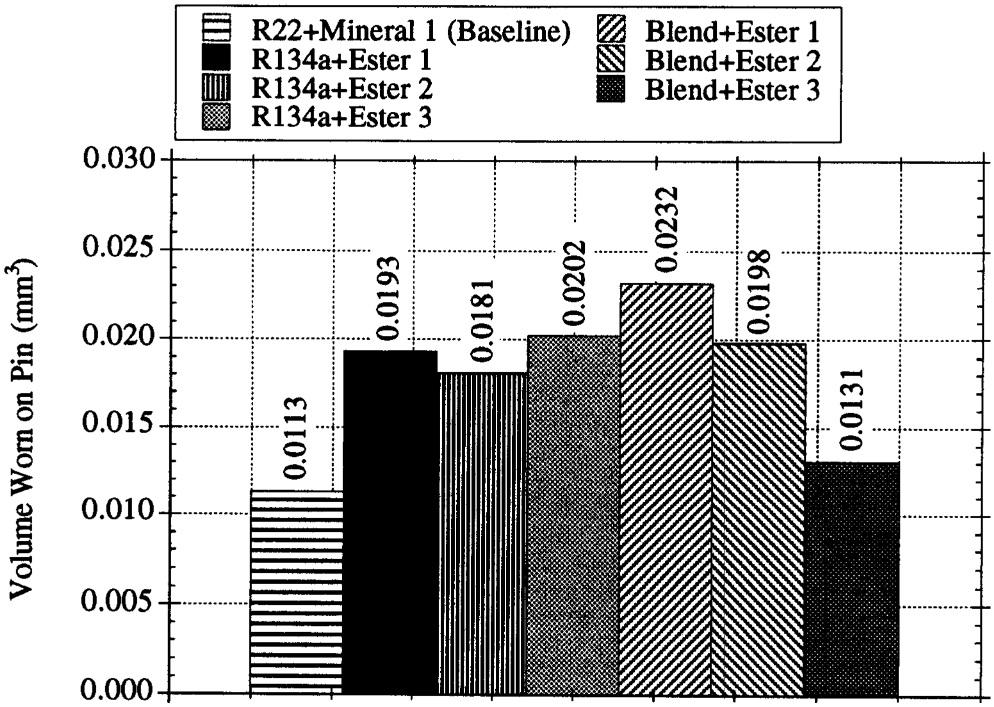


Figure 3.4c - Wear Results (23 in./s at 185 lbf)

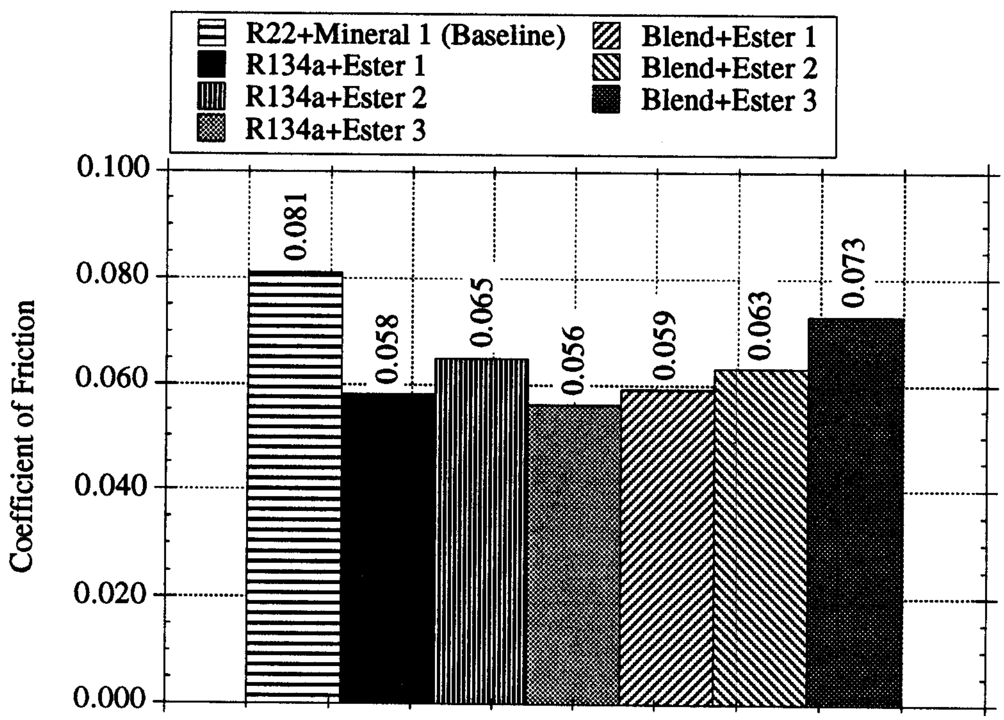


Figure 3.4d - Coefficient of Friction (23 in./s at 185 lbf)
Duration of Each Test =1 Hour

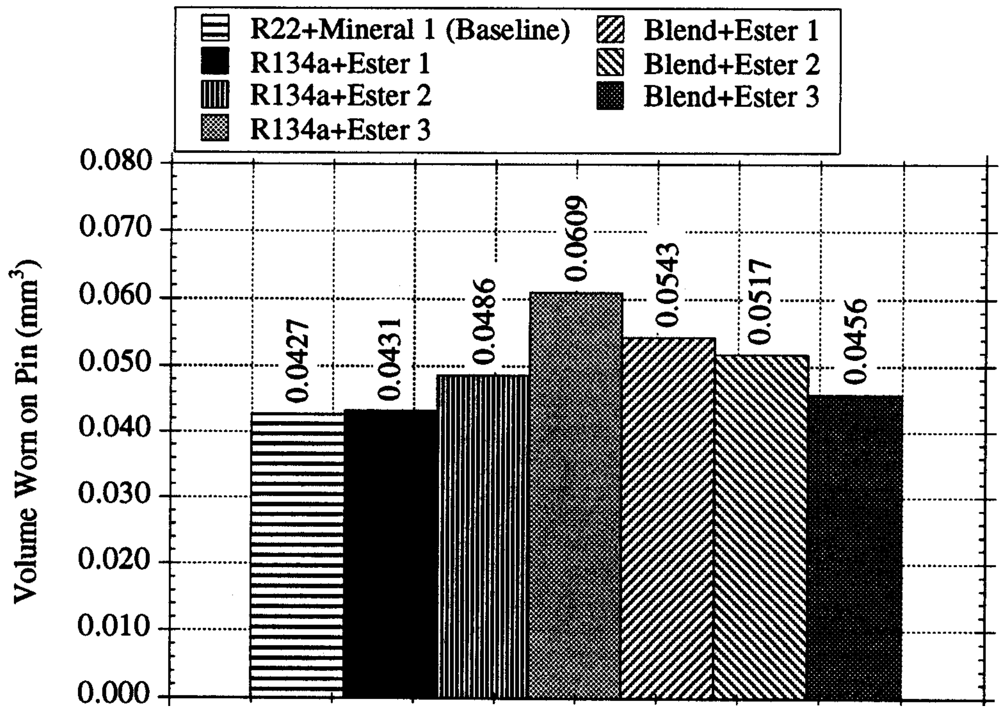


Figure 3.4e - Wear Results (23 in./s at 370 lbf)

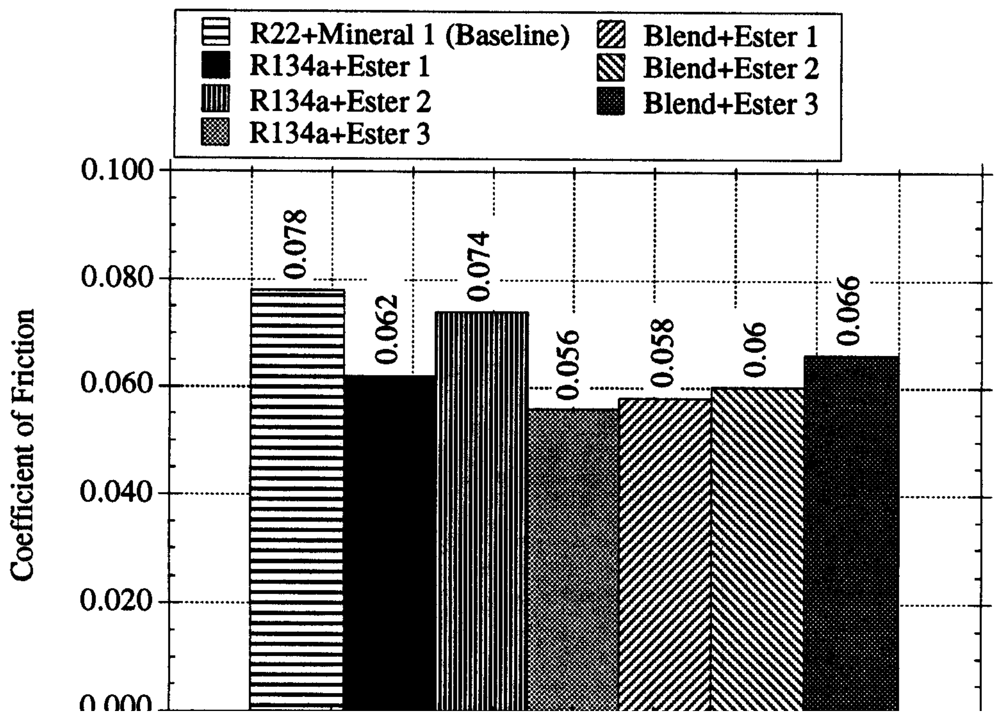


Figure 3.4f - Coefficient of Friction (23 in./s at 370 lbf)
Duration of Each Test =1 Hour

For the lubricants/blend mixtures, it is seen that the HPT produces the same lubricant ranking under the different operating conditions and the lubricant ranking obtained from the HPT does not correlate with that obtained from the Falex™ tester. The Falex™ ranks Ester 2, Ester 3 and Ester 1 as best, intermediate and worst, respectively, while the HPT consistently ranks Ester 3, Ester 2 and Ester 1 as best, intermediate and worst, respectively at the various testing conditions.

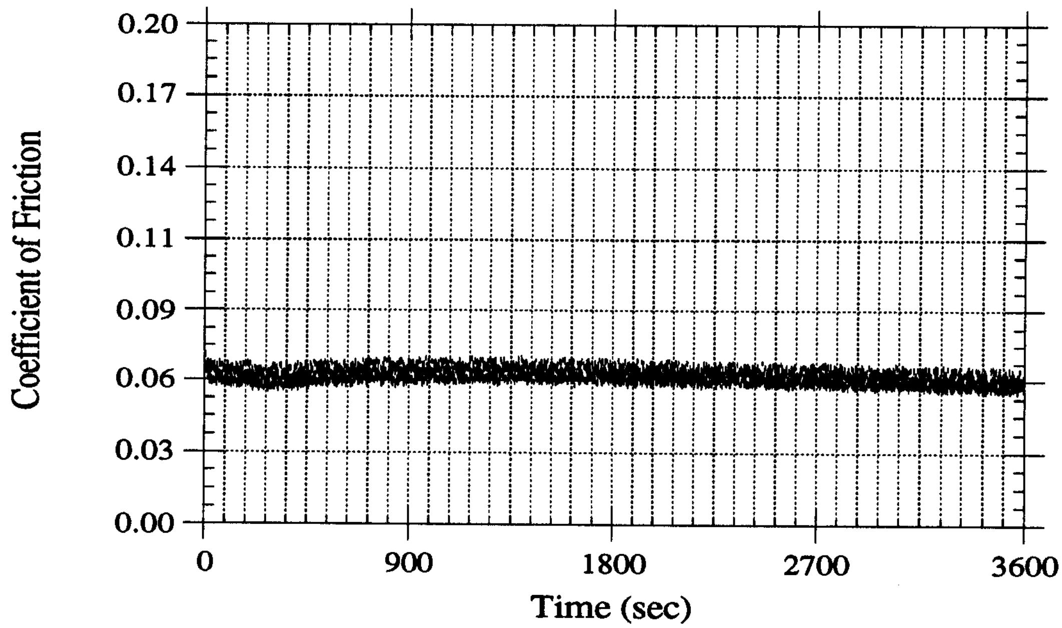
For the lubricants/R-134a mixtures, the lubricant ranking from the Falex™ tester is not available. Data from the HPT show that there exists a slight discrepancy in the ranking for the lubricants/R-134a mixtures (**Figs. 3.4a**, **3.4c** and **3.4e**). It was found that, at the conditions of 23 in./s at 370 lbf, the HPT ranked Ester 1, Ester 2 and Ester 3 as best, intermediate and worst, respectively (**Table 3.11**). Note, however, that for statistically significant data, the HPT rankings are consistent.

The friction and wear results further showed that wear generally increases with increasing load and speed, however, the friction coefficients remain almost constant. Typical graphical representations of the friction coefficient vs. time are given in **Figure 3.5**. The friction coefficient data plotted in these figures are collections of instantaneous values, which are used to compute the average coefficient of friction. The fluctuations of the friction coefficient vary slightly for each of the tests, although the mean value of the friction coefficient remained approximately constant for all tests conducted. It should be emphasized that the friction plotted is based on boundary lubrication conditions. The various lubricants may not have the same friction ranking if the lubrication process is predominantly hydrodynamic where lubricant viscosity plays a major role.

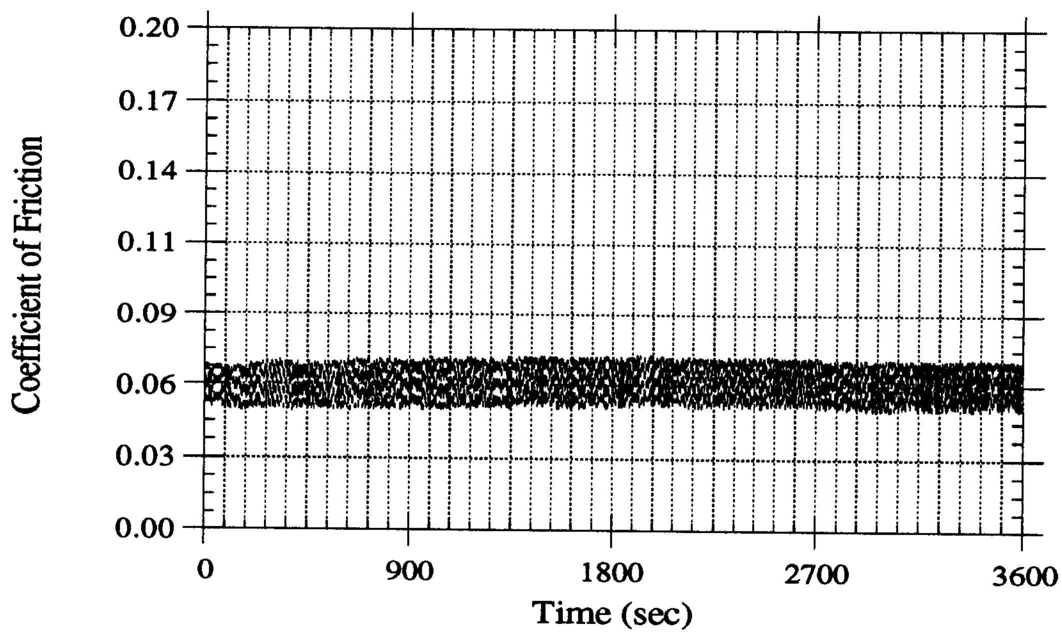
It is interesting to note that Ester 3 with the blend refrigerant shows the best wear resistance, however, its friction coefficient is among the worst. This indicates that there is no general relationship between friction and wear, even though frictional changes are often good indicators of changes in the wear mode or wear transitions. Both friction and wear are governed by complex interfacial phenomena. A tribo-contact, which has high friction, will generate more heat and, therefore, lower efficiencies and higher operating temperatures can be expected relative to a contact, which has low friction. High steady friction may not necessarily results in high wear. It is thought, however, that a high friction with large variations in magnitude is a good indicator of relatively high wear. The relationship between friction and wear is influenced by [1]: (a) relative thermal conductivities of the sliding materials, (b) relative fracture toughness of the materials, (c) the extent of micro-cutting and plowing in the material, (d) the presence of debris and/or transfer layers and films to either protect the surfaces or lubricate the surfaces, (f) the geometry of the contact as it affects heat transfer out of the interface, (g) the presence of a cooling lubricant and (h) the diversion of energy into the formation of tribochemical reaction products.

For given operating conditions and refrigerants in **Figure 3.4**, only the lubricant is changed. Therefore, from the list above, item (g) is the probable cause of the observed friction and wear behavior. Again, this phenomenon is not unusual in boundary lubrication because complex mechanical and chemical interactions take place at the interface. In addition to the operating condition, this interaction is affected by the materials in contact, lubricant and its additives and the environment.

Because of time limitation, all HPT data presented in Part I (Section 3.1) of this study are based on the average of two tests per lubricant. The statistical significance of the data obtained is evaluated by Small Sampling Theory (SST) [2]. The SST enables one to find the statistical significance between two sets of data points. Confidence intervals in excess of 80%, 90% and 95% are considered acceptable, very good and excellent, respectively. Anything less than 80% is generally considered poor confidence and is regarded as statistically insignificant.



(a)



(b)

Figure 3.5 - Coefficient of Friction vs. Time (Blend/Ester 1)
(a) Test 47FT (23 in./s at 185 lbf)
(b) Test 65FT (23 in./s at 370 lbf)

Statistical wear data for the lubricants in the presence of the blend refrigerant and in the presence of R-134a are shown in **Table 3.8** and **Table 3.10**, respectively. Based on SST analysis, the confidence intervals for the lubricants/blend and the lubricants/R-134a mixtures are tabulated in **Table 3.9** and **3.11**. In general, more tests are required to increase the degrees of freedom necessary for a statistical analysis to be more significant. In this study, the statistical significance of the data obtained for each company will be tabulated at the end of its corresponding section.

Table 3.8 - Statistical Wear Data for Case 1
Comparison of Wear Data Between Lubricants in the Presence of Blend Refrigerant

Ester 1			Ester 2			Ester 3		
X_1	σ_1	N_1	X_2	σ_2	N_2	X_3	σ_3	N_3
Test condition: 3.8 in./s at 185 lbf								
0.020	0.00368	2	0.019	0.00361	2	0.011	0.00262	2
Test condition: 23 in./s at 185 lbf								
0.023	0.00042	2	0.020	0.00085	2	0.013	0.00057	2
Test condition: 23 in./s at 370 lbf								
0.054	0.00240	2	0.052	0.00552	2	0.046	0.00926	2
X_i = Mean; σ_i = Standard Deviation; N_i = Number of Tests								

Table 3.9 - Confidence Intervals for Case 1 Using Small Sample Theory
Comparison of Wear Data Between Lubricants in the Presence of Blend Refrigerant

Test condition: 3.8 in./s at 185 lbf									
X_1-X_2	0.001	σ_{12}	0.00516	t_{12}	0.194	n_{12}	2	$\%_{12}$	57
X_2-X_3	0.008	σ_{23}	0.00446	t_{23}	1.794	n_{23}	2	$\%_{23}$	89
X_1-X_3	0.009	σ_{13}	0.00452	t_{13}	1.991	n_{13}	2	$\%_{13}$	90
Test condition: 23 in./s at 185 lbf									
X_1-X_2	0.003	σ_{12}	0.00095	t_{12}	3.158	n_{12}	2	$\%_{12}$	95
X_2-X_3	0.007	σ_{23}	0.00102	t_{23}	6.863	n_{23}	2	$\%_{23}$	99
X_1-X_3	0.010	σ_{13}	0.00071	t_{13}	14.09	n_{13}	2	$\%_{13}$	99
Test condition: 23 in./s at 370 lbf									
X_1-X_2	0.002	σ_{12}	0.00602	t_{12}	0.332	n_{12}	2	$\%_{12}$	61
X_2-X_3	0.006	σ_{23}	0.01078	t_{23}	0.589	n_{23}	2	$\%_{23}$	69
X_1-X_3	0.008	σ_{13}	0.00957	t_{13}	0.836	n_{13}	2	$\%_{13}$	75
$\sigma_{ij} = \sqrt{\dfrac{N_i\sigma_i^2 + N_j\sigma_j^2}{N_i + N_j - 2}}$; $t = \dfrac{X_i - X_j}{\sigma_{ij}\sqrt{1/N_i + 1/N_j}}$						$n = N_i + N_j - 2$; % = Percentage Value for the t Distribution (Confidence Level)			

Table 3.10 - Statistical Wear Data for Case 1
Comparison of Wear Data Between Lubricants in the Presence of R-134a

Ester 1			Ester 2			Ester 3		
X_1	σ_1	N_1	X_2	σ_2	N_2	X_3	σ_3	N_3
Test condition: 3.8 in./s at 185 lbf								
0.016	0.00141	2	0.013	0.00325	2	0.014	0.00148	2
Test condition: 23 in./s at 185 lbf								
0.019	0.00042	2	0.018	0.00141	2	0.020	0.00057	2
Test condition: 23 in./s at 370 lbf								
0.043	0.00332	2	0.049	0.00332	2	0.061	0.00064	2
X_i = Mean; σ_i = Standard Deviation; N_i = Number of Tests								

Table 3.11- Confidence Intervals for Case 1 Using Small Sample Theory
Comparison of Wear Data Between Lubricants in the Presence of R-134a

Test condition: 3.8 in./s at 185 lbf									
X_1-X_2	0.003	σ_{12}	0.00354	t_{12}	0.847	n_{12}	2	$\%_{12}$	75
X_2-X_3	0.001	σ_{23}	0.00357	t_{23}	0.280	n_{23}	2	$\%_{23}$	60
X_1-X_3	0.002	σ_{13}	0.00204	t_{13}	0.980	n_{13}	2	$\%_{13}$	78
Test condition: 23 in./s at 185 lbf									
X_1-X_2	0.001	σ_{12}	0.00147	t_{12}	0.680	n_{12}	2	$\%_{12}$	72
X_2-X_3	0.002	σ_{23}	0.00152	t_{23}	1.316	n_{23}	2	$\%_{23}$	84
X_1-X_3	0.001	σ_{13}	0.00071	t_{13}	1.408	n_{13}	2	$\%_{13}$	85
Test condition: 23 in./s at 370 lbf									
X_1-X_2	0.006	σ_{12}	0.00470	t_{12}	1.277	n_{12}	2	$\%_{12}$	83
X_2-X_3	0.012	σ_{23}	0.00338	t_{23}	3.550	n_{23}	2	$\%_{23}$	96
X_1-X_3	0.018	σ_{13}	0.00338	t_{13}	5.325	n_{13}	2	$\%_{13}$	98
$\sigma_{ij} = \sqrt{\frac{N_i\sigma_i^2 + N_j\sigma_j^2}{N_i + N_j - 2}}; t = \frac{X_i - X_j}{\sigma_{ij}\sqrt{1/N_i + 1/N_j}}$					$n = N_i + N_j - 2$; % = Percentage Value for the t Distribution (Confidence Level)				

3.1.5.2 Case 2: HPT Friction and Wear Results - Drill Rod Pin on 356 Aluminum Disk (Reciprocating Compressor)

The reported wear data were obtained by measuring the weight of the aluminum plates before and after the test. Typical wear scars on the 356 Die Cast aluminum specimens are shown in **Figure 3.6**. Because the mating piece, the drill rod pin, is much harder than the aluminum, only small polishing marks are observed on the pin, therefore, wear was not measured on the pin.

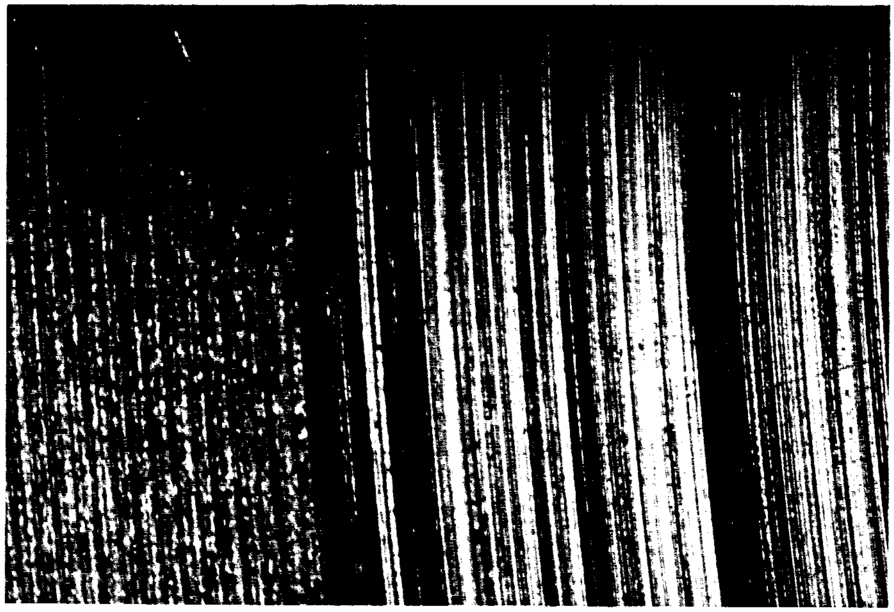
Figure 3.7 illustrates the friction and wear results obtained by means of the HPT. The raw friction and wear data are given in the Appendix D. The data show that an average scatter between wear results of repeated tests is about 11%. The average scatter for the coefficient of friction between repeated tests is 10.6% **Figures 3.7a** and **3.7c** show that the HPT data do not seem to give a consistent ranking at the different testing conditions. However, based on the statistical wear data (**Table 3.13**), the confidence intervals (**Table 3.14**) which were obtained at the conditions of 23 in./s at 50 lbf are 92% between Ester 4 and Ester 5, 52% between Ester 5 and Ester 6, and 87% between Ester 4 and Ester 6. This indicates that Ester 4 is clearly the worst and the relative ranking between Ester 5 and Ester 6 is inconclusive. Therefore, it would not be appropriate to rank these lubricants based on the data obtained. Again, more tests might yield statistically significant results between these lubricants. It is also possible that these lubricants have approximately the same lubricity characteristics.

From **Table 3.12**, the Falex™ ranks Ester 5, Ester 6 and Ester 4 as best, intermediate and worst, respectively. The data supplied by the company were qualitative (best, intermediate, worst) and, therefore, the relative wear difference between the best and worst is not known.

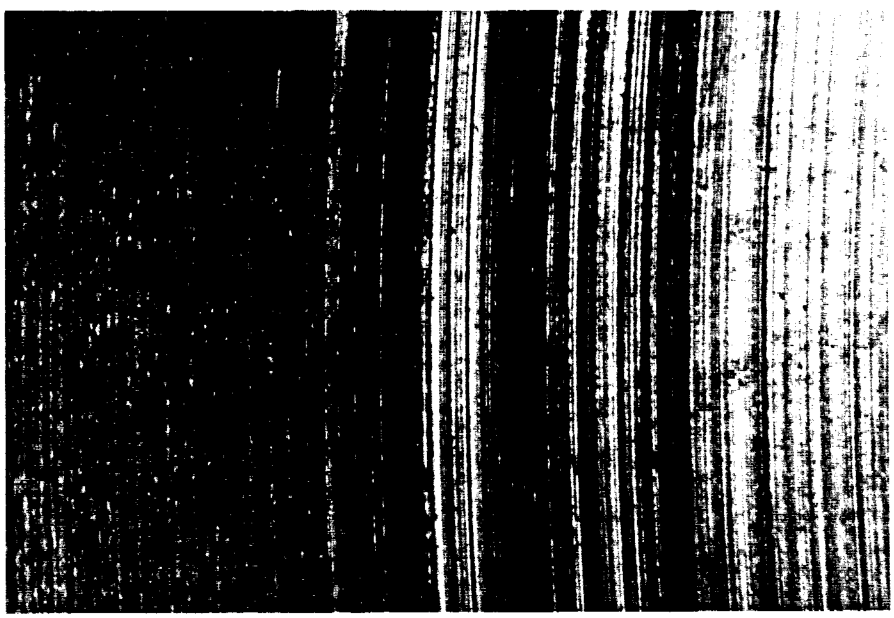
A graphic representation of the friction coefficient plots and a typical load record are shown in **Figure 3.8** and **3.9**, respectively. The mean force value and the oscillations stayed fairly constant throughout the test.

Table 3.12 - Ranking of Lubricants by Wear (Case 2)

Lubricant	Ref	Falex™ 3.8 in./s at 854 lbf	HPT 3.8 in./s at 50 lbf (gram)	HPT 23 in./s at 50 lbf (gram)	HPT 3.8 in./s at 100 lbf (gram)
Ester 4	R-134a	(W)	0.085 (ns)	0.402 (W)	0.100 (ns)
Ester 5	R-134a	(B)	0.079 (ns)	0.374 (ns)	0.101 (ns)
Ester 6	R-134a	(I)	0.094 (ns)	0.373 (ns)	0.103 (ns)
ns : Statistically Not Significant; B: Best; I: Intermediate; W: Worst					

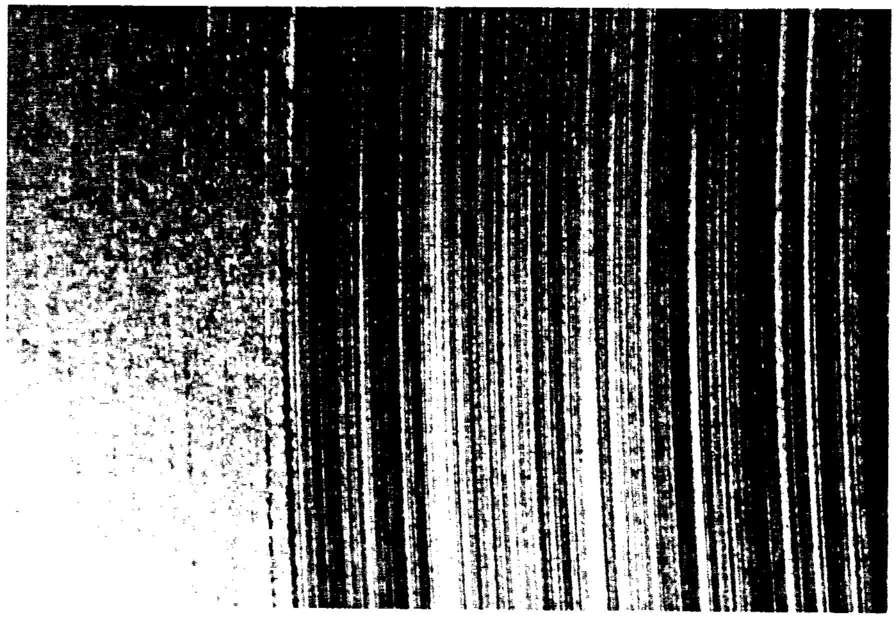


(a)

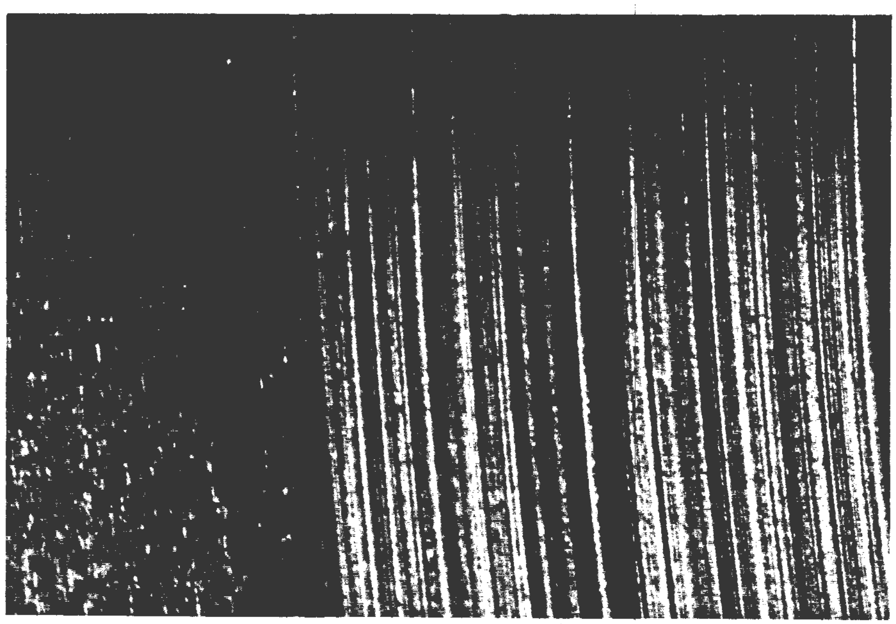


(b)

Figure 3.6a - Typical Wear Scars on the 356 Die Cast Al Plate
(a) Tested with R-12/Mineral 2 (10 X Mag.)
(b) Tested with R-134a/Ester 4 (10 X Mag.)



(c)



(d)

Figure 3.6b - Typical Wear Scars on the 356 Die Cast Al Plate
(c) Tested with R-134a/Ester 5 (10 X Mag.)
(d) Tested with R-134a/Ester 6 (10 X Mag.)

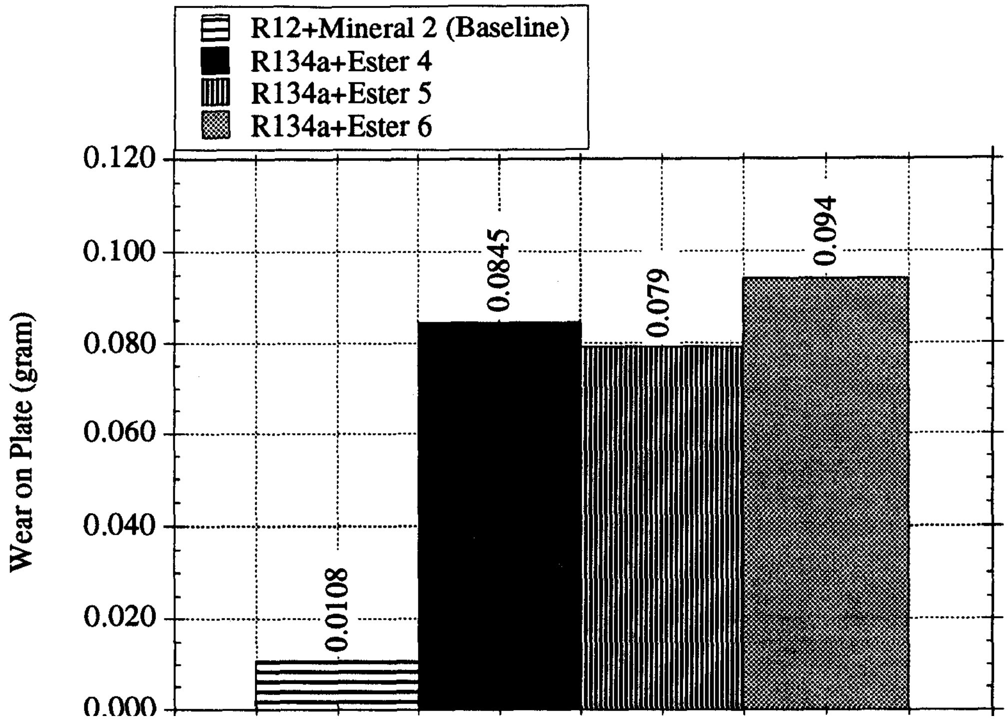


Figure 3.7a - Wear results (3.8 in./s at 50 lbf)

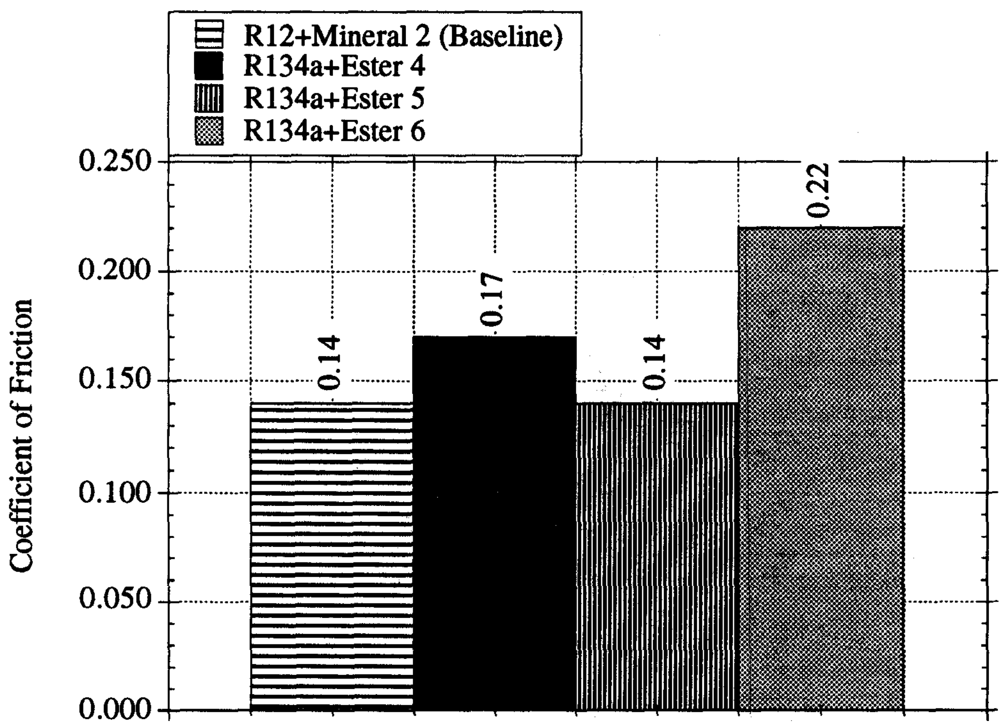


Figure 3.7b - Coefficient of Friction (3.8 in./s at 50 lbf)
Duration of Each Tests =1 Hour

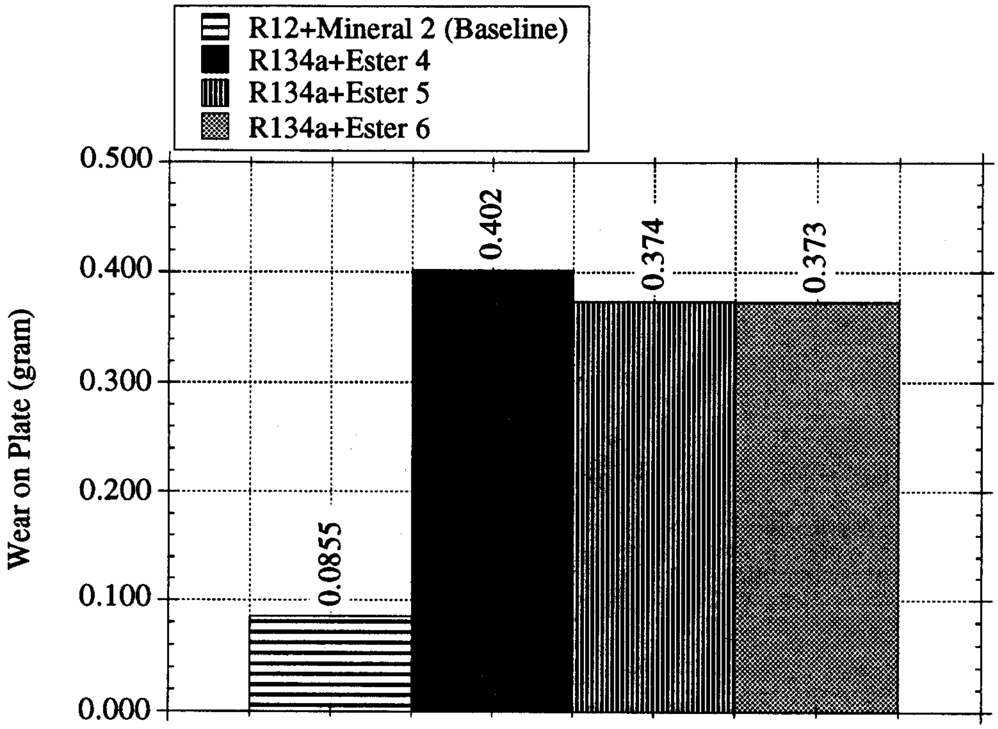


Figure 3.7c - Wear Results (23 in./s at 50 lbf)

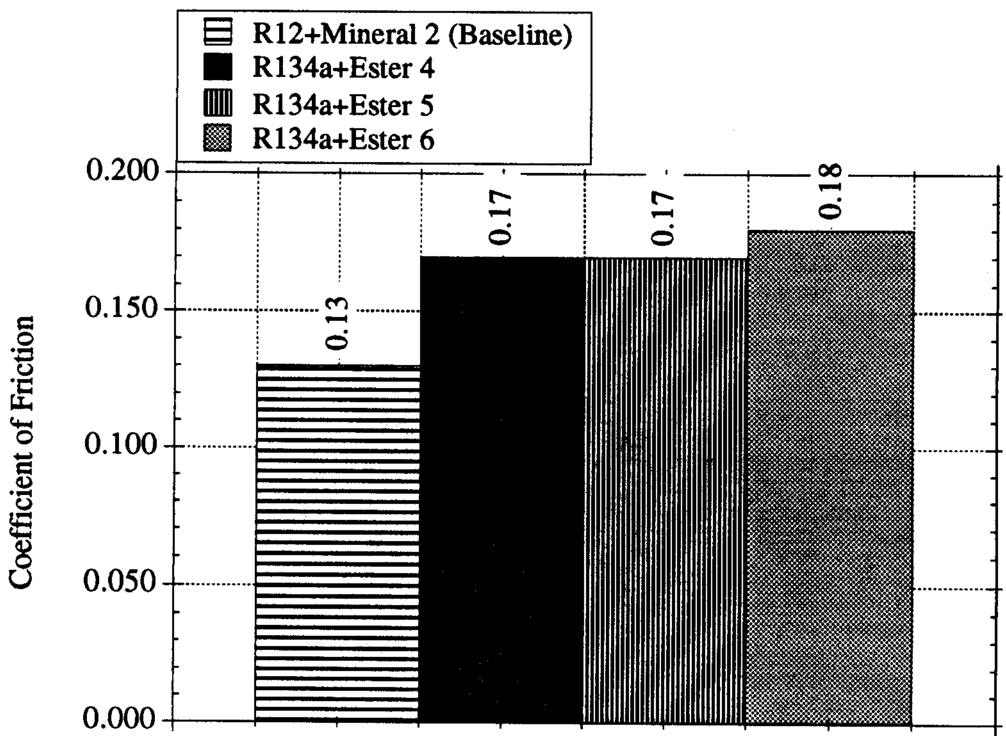


Figure 3.7d - Coefficient of Friction (23 in./s at 50 lbf)
Duration of Each Test =1 Hour

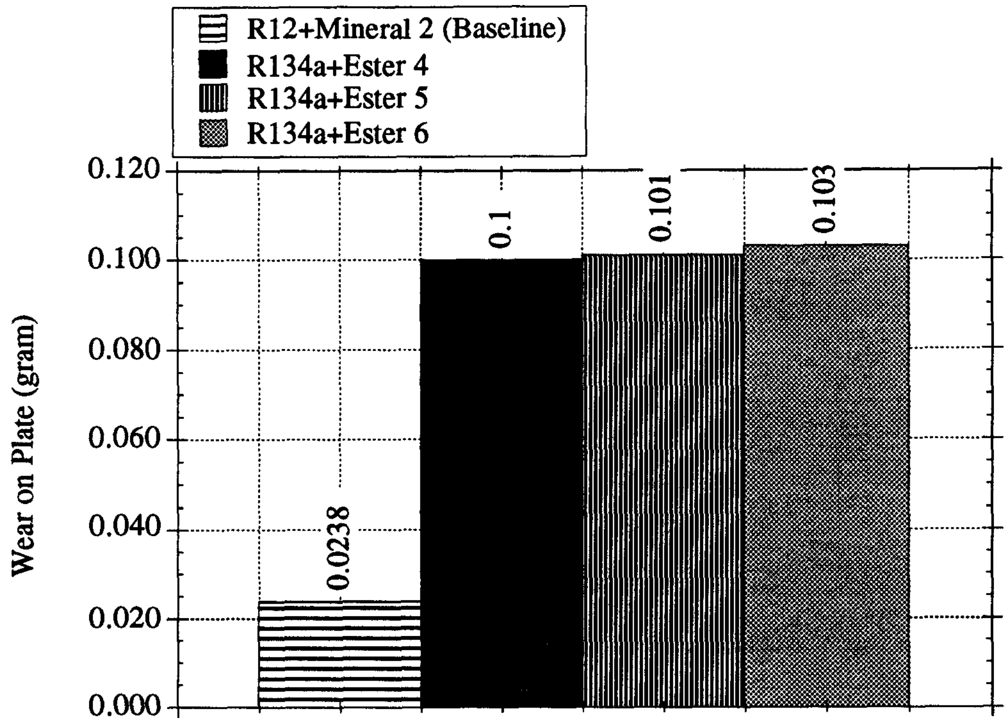


Figure 3.7e - Wear results (3.8 in./s at 100 lbf)

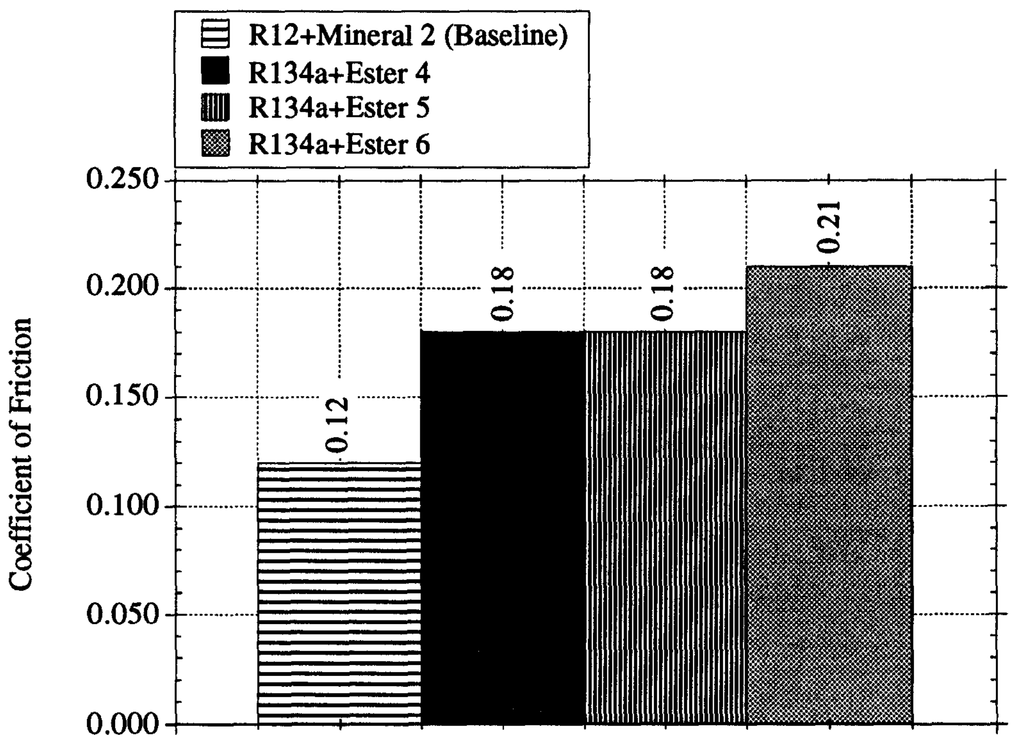


Figure 3.7f - Coefficient of Friction (3.8 in./s at 100 lbf)
Duration of Each Tests =1Hour

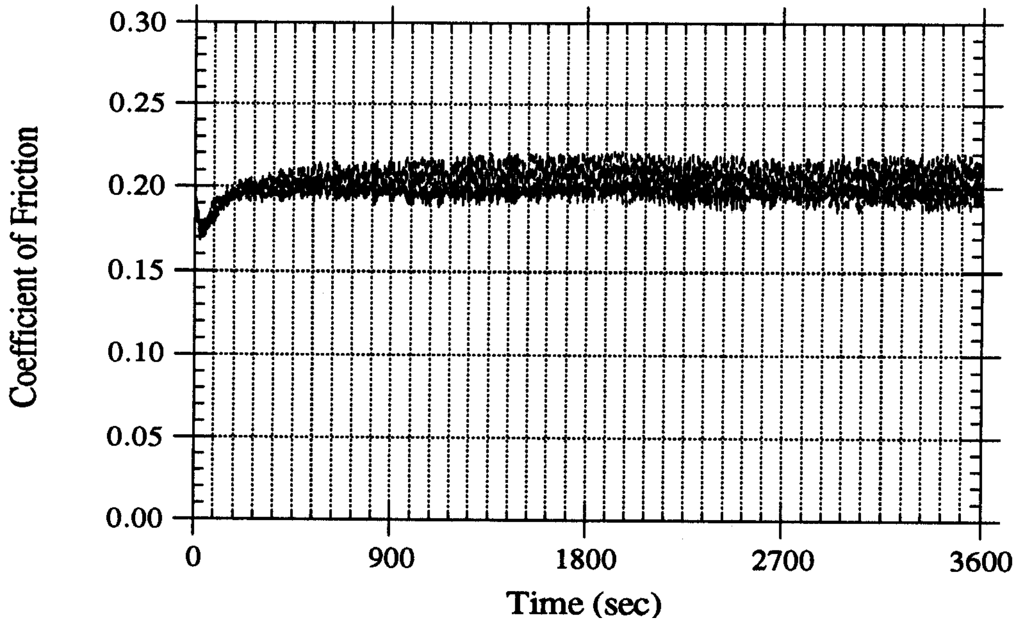


Figure 3.8 - Coefficient of Friction vs. Time
Test 55FC (3.8 in./s at 50 lbf)

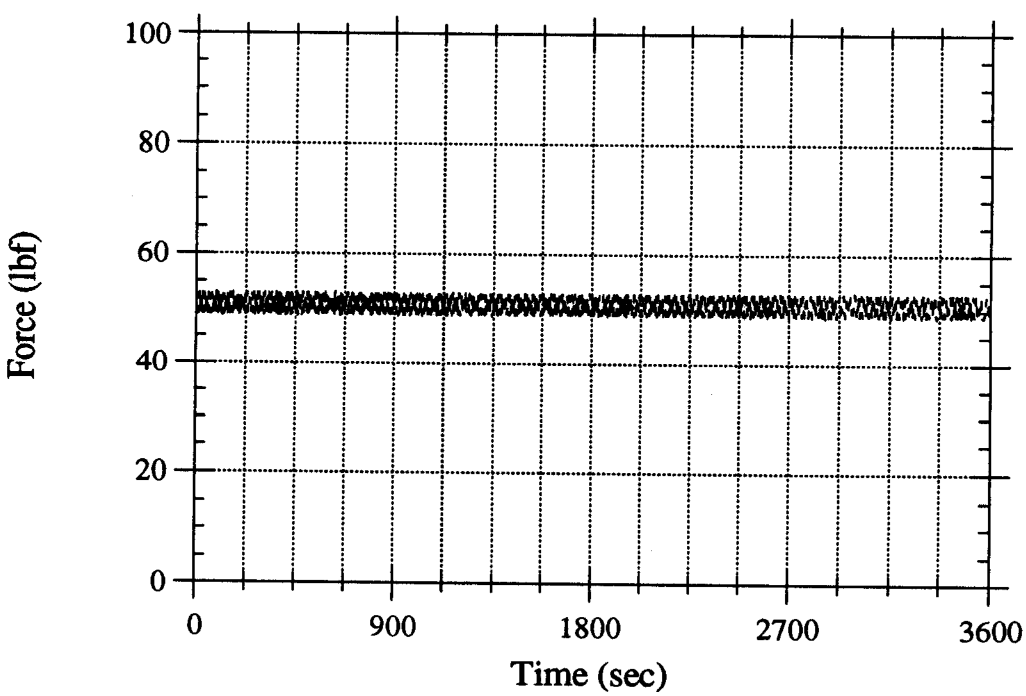


Figure 3.9 - Typical Axial Force Record

Table 3.13 - Statistical Wear Data for Case 2
Comparison of Wear Data Between Lubricants in the Presence of R-134a

Ester 4			Ester 5			Ester 6		
X_1	σ_1	N_1	X_2	σ_2	N_2	X_3	σ_3	N_3
Test condition: 3.8 in./s at 50 lbf								
0.085	0.00679	2	0.079	0.00382	2	0.094	0.01075	2
Test condition: 23 in./s at 50 lbf								
0.402	0.00064	2	0.374	0.01181	2	0.373	0.01761	2
Test condition: 3.8 in./s at 100 lbf								
0.100	0.00304	2	0.101	0.00742	2	0.103	0.00643	2
X_i = Mean; σ_i = Standard Deviation; N_i = Number of Tests								

Table 3.14 - Confidence Intervals for Case 2 Using Small Sample Theory
Comparison of Wear Data Between Lubricants in the Presence of R-134a

Test condition: 3.8 in./s at 50 lbf									
X_1-X_2	0.006	σ_{12}	0.00779	t_{12}	0.770	n_{12}	2	$\%_{12}$	74
X_2-X_3	0.015	σ_{23}	0.01141	t_{23}	1.315	n_{23}	2	$\%_{23}$	84
X_1-X_3	0.009	σ_{13}	0.01271	t_{13}	0.708	n_{13}	2	$\%_{13}$	72
Test condition: 23 in./s at 50 lbf									
X_1-X_2	0.028	σ_{12}	0.01183	t_{12}	2.367	n_{12}	2	$\%_{12}$	92
X_2-X_3	0.001	σ_{23}	0.02120	t_{23}	0.047	n_{23}	2	$\%_{23}$	52
X_1-X_3	0.029	σ_{13}	0.01762	t_{13}	1.646	n_{13}	2	$\%_{13}$	87
Test condition: 3.8 in./s at 100 lbf									
X_1-X_2	0.001	σ_{12}	0.00802	t_{12}	0.125	n_{12}	2	$\%_{12}$	54
X_2-X_3	0.002	σ_{23}	0.00982	t_{23}	0.204	n_{23}	2	$\%_{23}$	57
X_1-X_3	0.003	σ_{13}	0.00711	t_{13}	0.422	n_{13}	2	$\%_{13}$	64
$\sigma_{ij} = \sqrt{\frac{N_i\sigma_i^2 + N_j\sigma_j^2}{N_i + N_j - 2}}$; $t = \frac{X_i - X_j}{\sigma_{ij}\sqrt{1/N_i + 1/N_j}}$						$n = N_i + N_j - 2$; % = Percentage Value for the t Distribution (Confidence Level)			

3.1.5.3 Case 3: HPT Friction and Wear Results - Carburized 1018 Steel Pin on 380 Die Cast Aluminum Pad (Reciprocating Compressor)

As previously stated, oscillatory motion was used in this part of the testing program due to the unavailability of 380 die cast aluminum in a size required for unidirectional tests. The different kinematic conditions of the contacts might affect the generation of lubricating films and surface damages under boundary lubrication. The average speed used for the oscillatory HPT tests is the same as that used to conduct the Falex™ tests. The wear depth is determined by tracing across the wear scar at three different locations with a Dektak stylus surface profiler, and taking the average of these measurements. **Figure 3.10** and **3.11** show a typical wear scar on the 380 die cast aluminum pad specimen and its surface profile, respectively.

Wear and friction data obtained from the HPT are given in **Figures 3.12**. From raw wear data shown in the Appendix D, the average scatter between repeated tests is about 16%. The maximum scatter for the coefficient of friction between repeated tests is 16% as well.

Quantitative data obtained from both the Falex™ tester and the HPT are summarized in **Table 3.15**. The HPT ranks Ester 8, Ester 7 and Ester 9 as best, intermediate and worst, respectively. Based on the statistical wear data given in **Table 3.16**, the statistical analysis (**Table 3.17**) shows 98% confidence interval for the relative ranking between Ester 8 and Ester 9, 99% between Ester 7 and Ester 9, and only 76% between Ester 7 and Ester 8. This indicates that Ester 9 is the worst and the relative ranking between Ester 7 and Ester 8 is inconclusive. However, the Falex™ ranks Ester 8, Ester 9 and Ester 7 as best, intermediate and worst, respectively. Once again the ranking of the lubricants based on the Falex™ data and the data from the HPT do not match. It should be noted that the difference in the rankings might be caused by the different kinematic conditions.

Table 3.15 - Ranking of Lubricants by Wear (Case 3)

Lubricant	Ref	Falex™ 3.8 in./s at 3,330 lbf (gram)	HPT ± 3.8 in./s at 50 lbf (µm)
Ester 7	R-134a	0.0106 (W)	29.9 (ns)
Ester 8	R-134a	Negligible (B)	26.7 (ns)
Ester 9	R-134a	0.0031 (I)	50.9 (W)
ns : Statistically Not Significant; B: Best; I: Intermediate; W: Worst			

Even though the wear results are appreciably different among the lubricants, the coefficient of friction is approximately the same for all esters/R134a mixtures tested. The mineral oil/R12 mixture shows very good wear characteristics compared to the esters with R134a, but its friction characteristics tend to be worse than those of the latter mixtures. This indicates that the wear results do not generally correlate well with the friction results. This phenomenon has been observed throughout this study. A typical example of the friction coefficient vs. time obtained from the steel/aluminum contact is shown in **Figure 3.13**. Compared to the friction coefficient vs. time data obtained from the other tests, large fluctuations in the friction coefficients are observed. This is due to the type of motion used for this test (oscillatory).

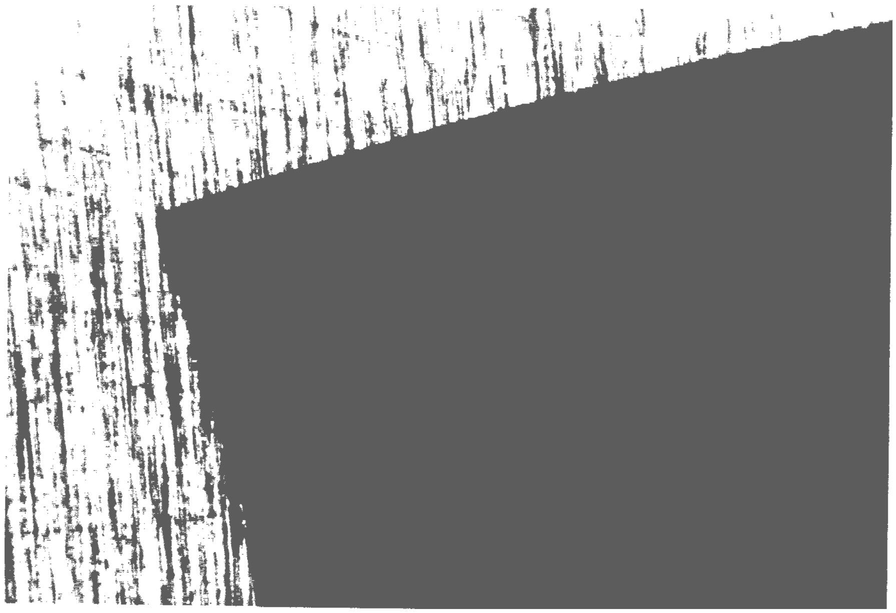


Figure 3.10 - Typical Wear Scar on the 380 Die Cast Al Specimen Caused by Oscillatory Motion

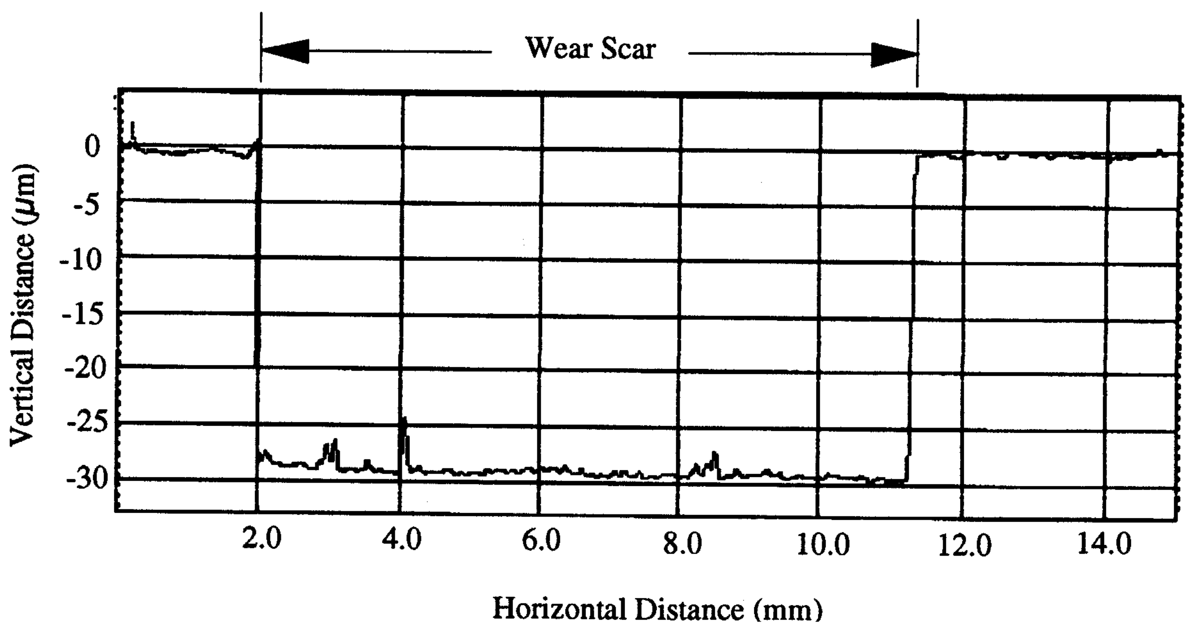


Figure 3.11 - Surface Profile of the 380 Die Cast Al specimen

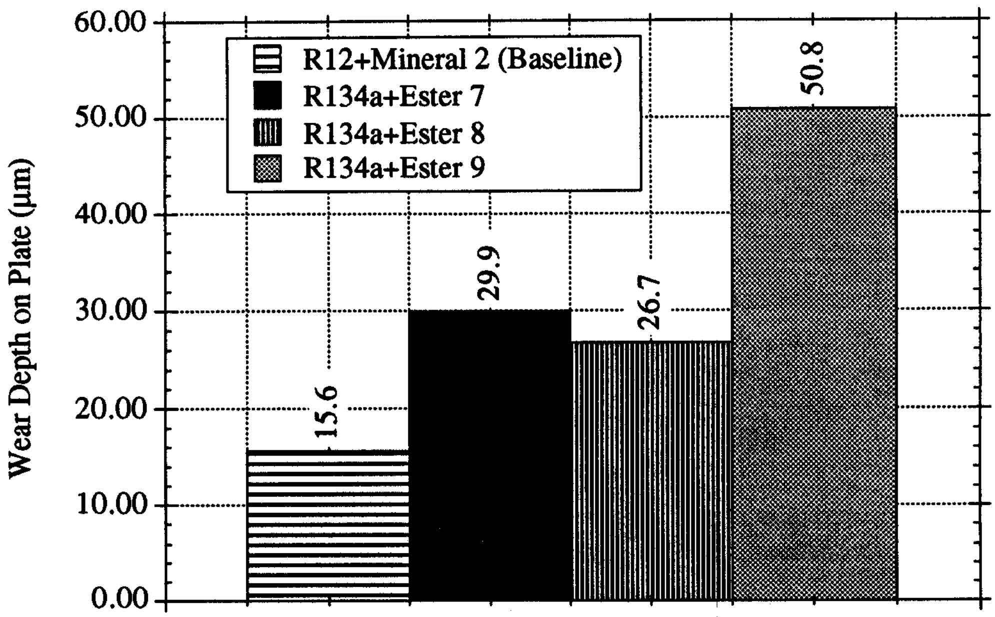


Figure 3.12a - Wear results (± 3.8 in./s at 50 lbf)

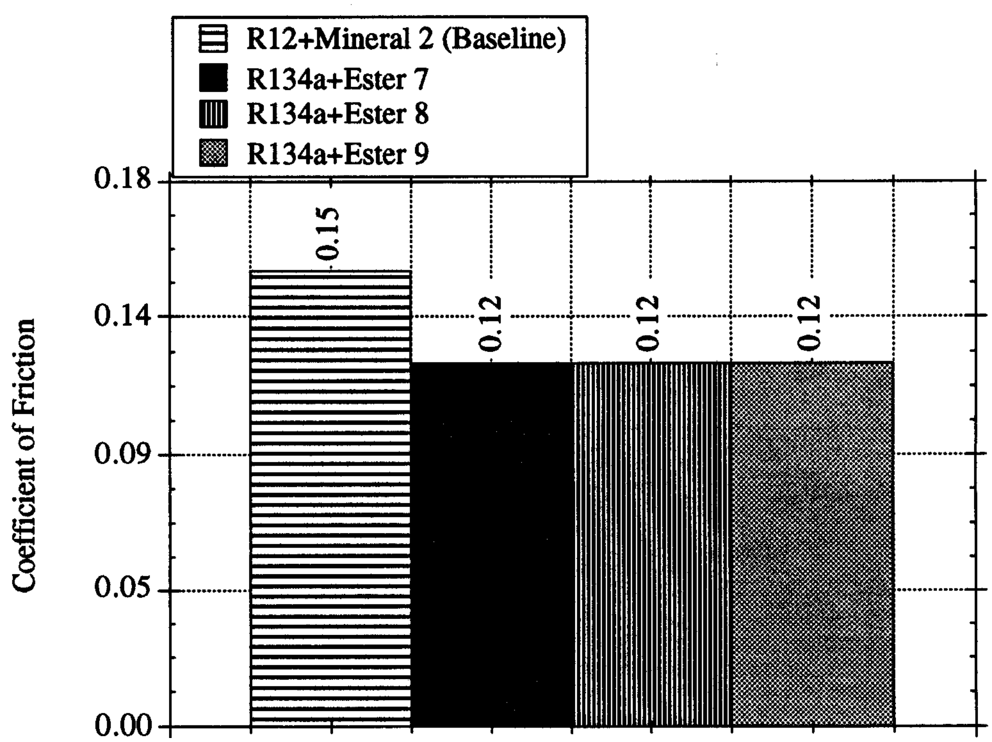


Figure 3.12b - Coefficient of Friction (± 3.8 in./s at 50 lbf)
Duration of Each Test = 1 Hour

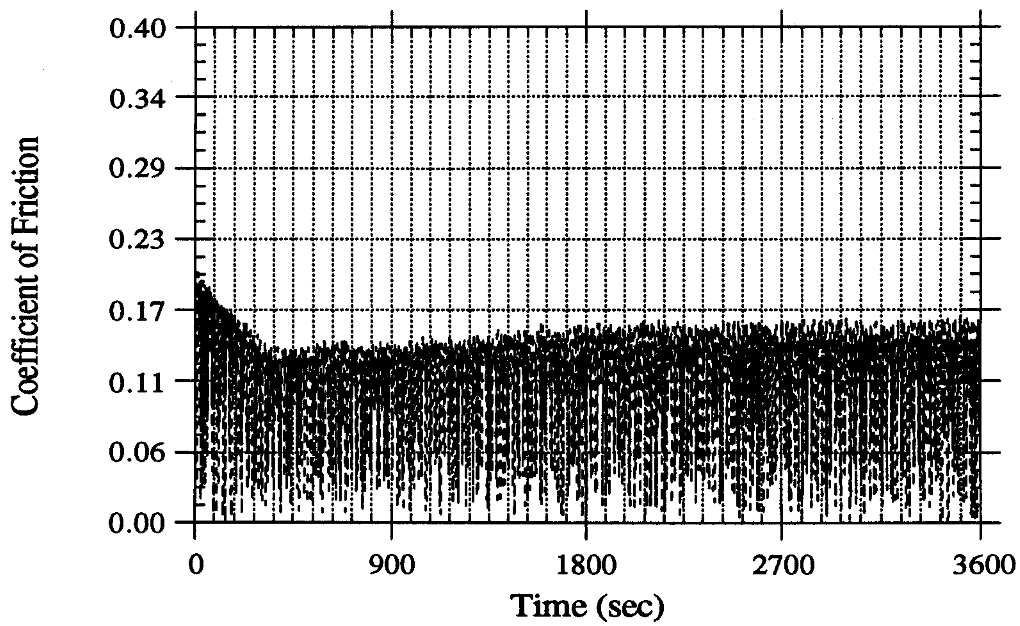


Figure 3.13 - Coefficient of Friction vs. Time
Test 50FC (± 3.8 in./s at 50 lbf)

Table 3.16 - Statistical Wear Data for Case 3

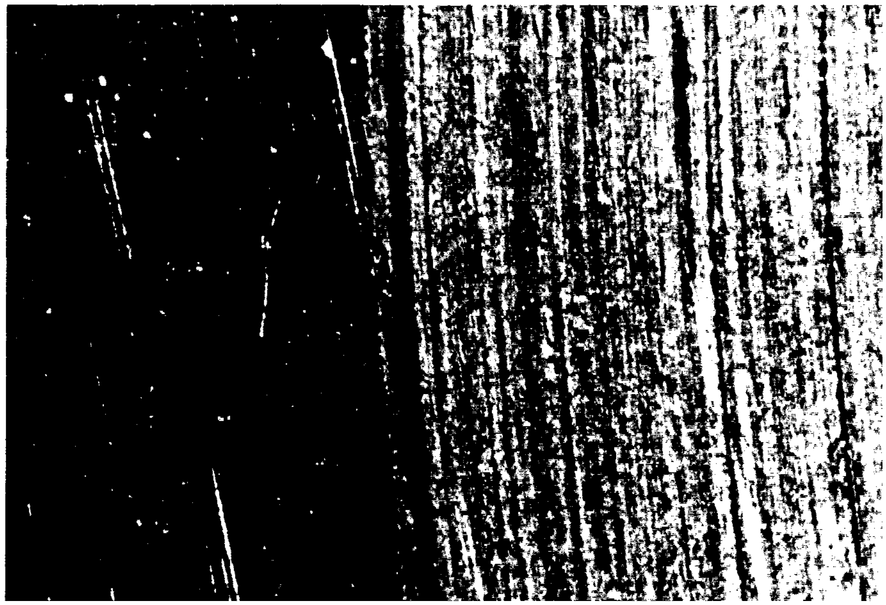
Ester 7			Ester 8			Ester 9		
X_1	σ_1	N_1	X_2	σ_2	N_2	X_3	σ_3	N_3
29.9	0.14142	2	26.7	3.5355	2	50.9	2.3335	2
X_i = Mean; σ_i = Standard Deviation; N_i = Number of Tests								

Table 3.17 - Confidence Intervals for Case 3 Using Small Sample Theory

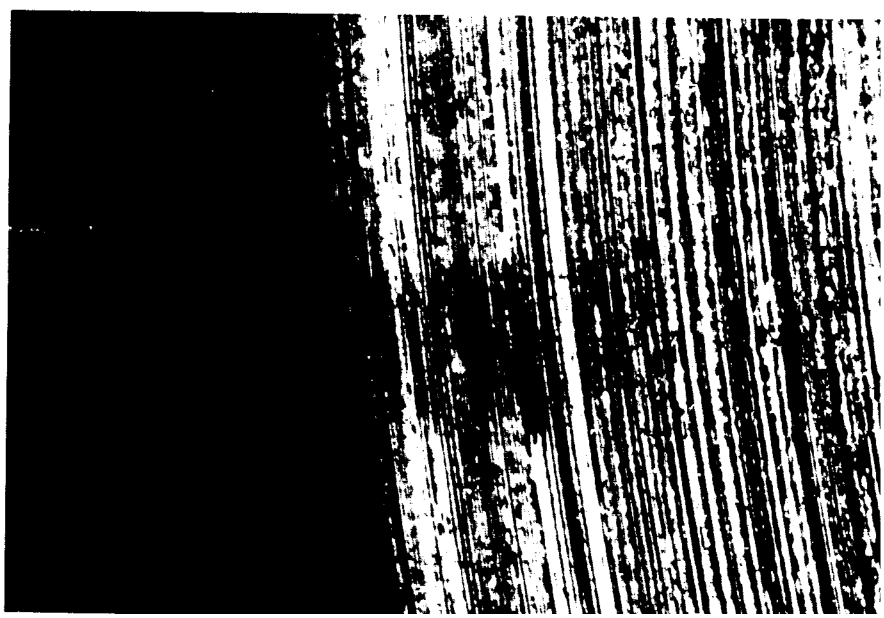
X_1-X_2	3.2	σ_{12}	3.5383	t_{12}	0.904	n_{12}	2	$\%_{12}$	76
X_2-X_3	24.2	σ_{23}	4.2362	t_{23}	5.713	n_{23}	2	$\%_{23}$	98
X_1-X_3	21.0	σ_{13}	2.3378	t_{13}	8.983	n_{13}	2	$\%_{13}$	99
$\sigma_{ij} = \sqrt{\dfrac{N_i\sigma_i^2 + N_j\sigma_j^2}{N_i + N_j - 2}}$; $t = \dfrac{X_i - X_j}{\sigma_{ij}\sqrt{1/N_i + 1/N_j}}$					$n = N_i + N_j - 2$; % = Percentage Value for the t Distribution (Confidence Level)				

3.1.5.4 Case 4: HPT Friction and Wear Results - Carburized 1018 Steel Pin on Gray Cast Iron Disk (Reciprocating Compressor)

The wear data for Case 4 are obtained by measuring the wear depth on the gray cast iron plates. The wear depth is determined by the same method described in the previous section. The mating piece, the 1018 carburized steel pin, is much harder than the gray cast iron. Only small polishing marks are observed on the pin. Therefore, no measurements were taken on the pin. Typical wear scar appearances and the corresponding surface profiles are shown in **Figure 3.14** and **Figure 3.15**, respectively.

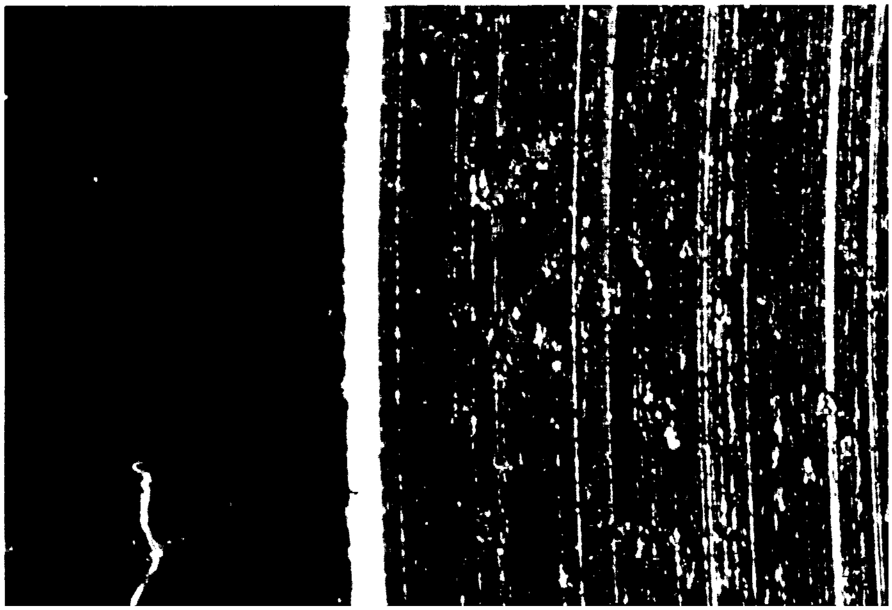


(a)

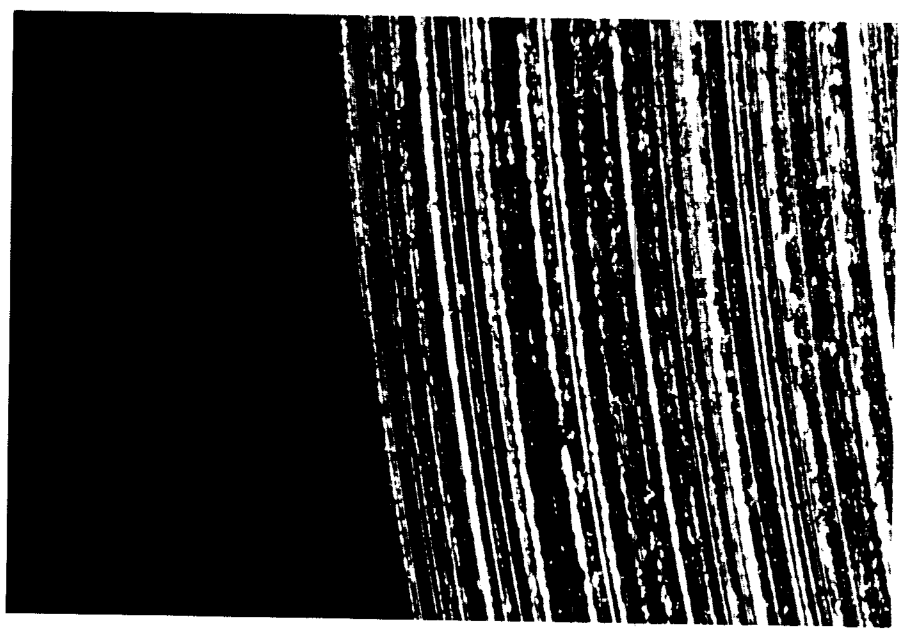


(b)

Figure 3.14a - Typical Wear Scars on the Gray Cast Iron Plate
(a) Tested with R-12/Mineral 2 (10 X Mag.)
(b) Tested with R-134a/Ester 7 (10 X Mag.)

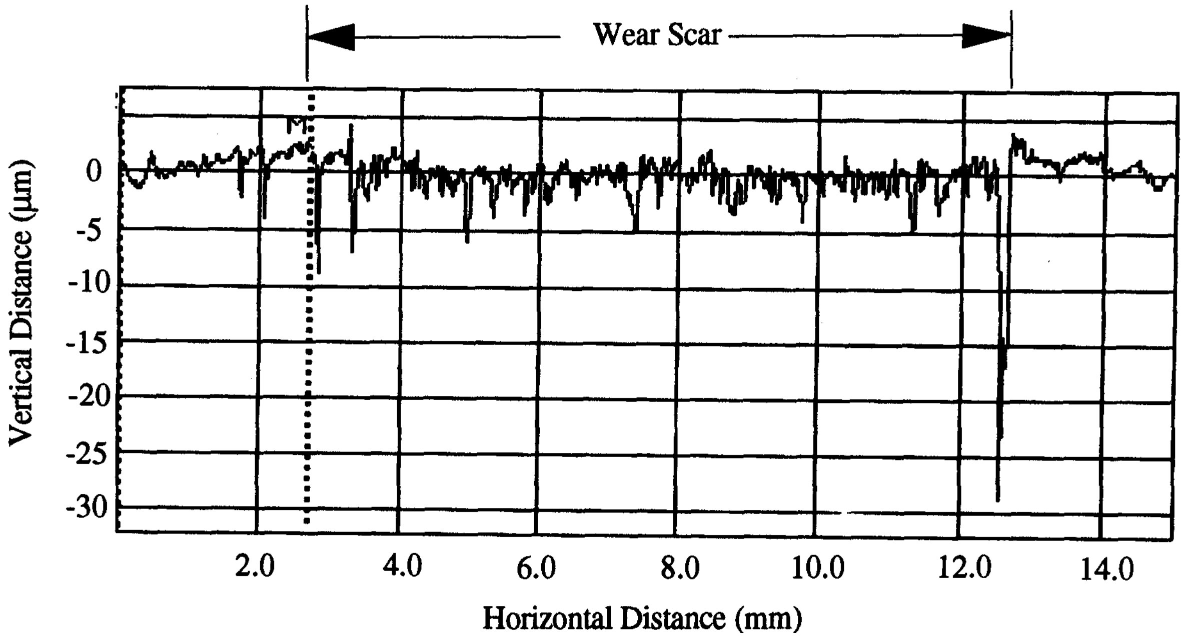


(c)

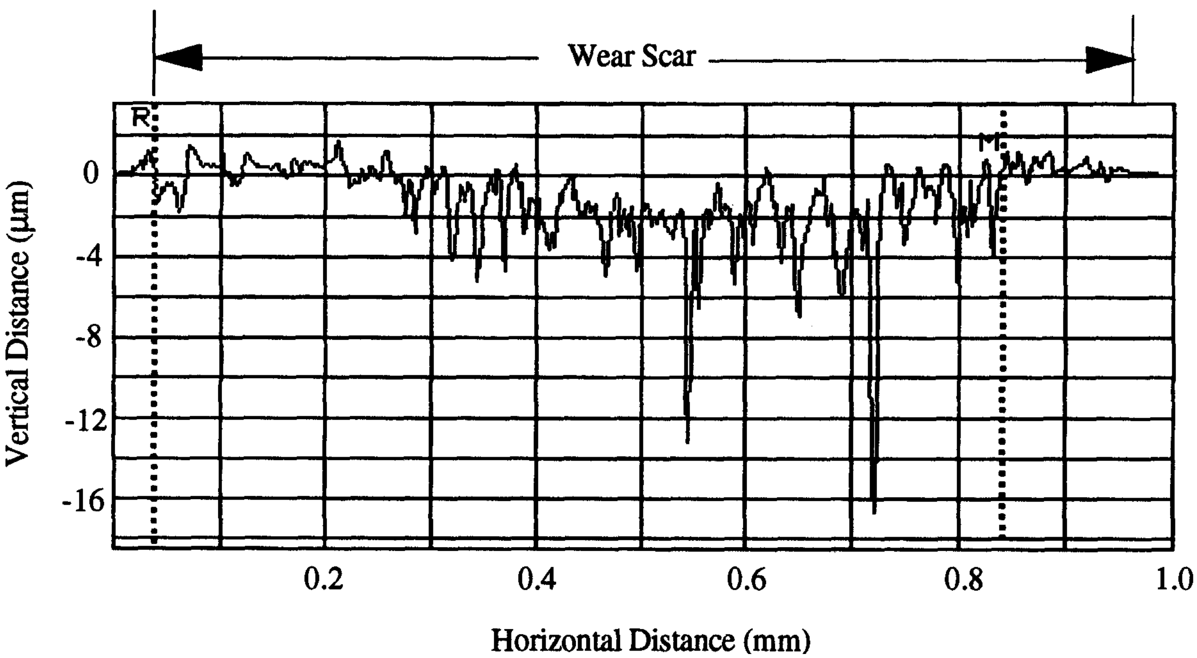


(d)

Figure 3.14b - Typical Wear Scars on the Gray Cast Iron Plate
(c) Tested with R-134a/Ester 8 (10 X Mag.)
(d) Tested with R-134a/Ester 9 (10 X Mag.)

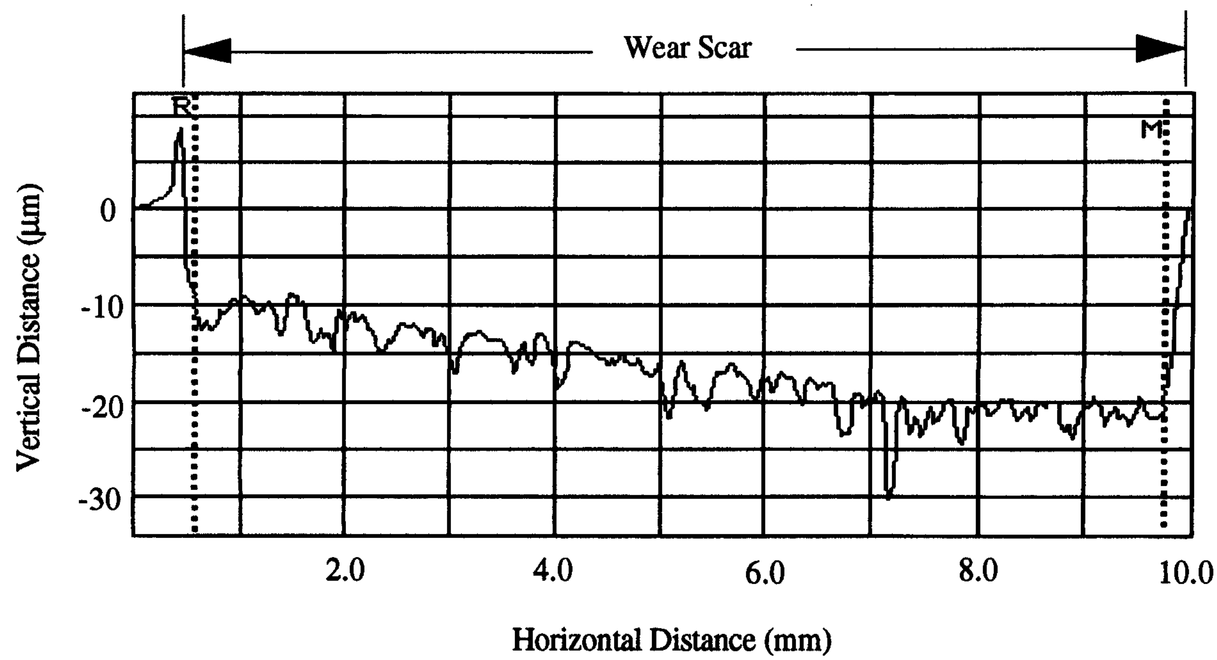


(a)

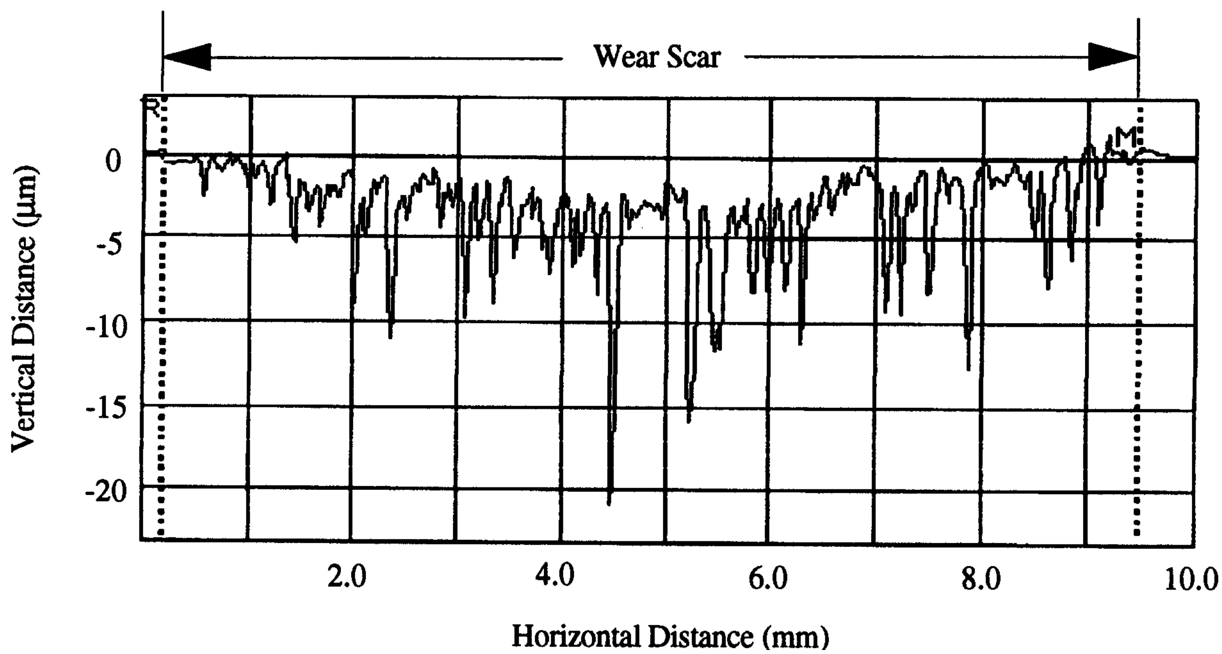


(b)

Figure 3.15a - Wear Scars on the Gray Cast Iron Plates (300 lbf at 15.9 in./s)
(a) Tested with R-12/Mineral 2
(b) Tested with R-134a/Ester 7



(c)



(d)

Figure 3.15b - Wear Scars on the Gray Cast Iron Plates (300 lbf at 15.9 in./s)
(c) Tested with R-134a/Ester 8
(d) Tested with R-134a/Ester 9

Friction and wear results for Case 4 are shown in **Figure 3.16**. From the raw data given in the Appendix D, the average scatter between repeated wear tests is 16%. The repeatability of the coefficient of friction for all tests is within 5%. From **Figures 3.16a** and **3.16c**, it is seen that a consistent ranking exists for the lubricants under the two different loading conditions. For both 300 lbf and 400 lbf, with the same speed, the HPT ranks Ester 7, Ester 9 and Ester 8 as best, intermediate and worst, respectively. The wear differences are appreciable between the lubricants tested. The rankings of lubricants/R-134a mixtures for Case 4 were found to be statistically significant. For both of the testing conditions, based on statistical wear data (**Table 3.19**), the confidence intervals (**Table 3.20**) for the relative ranking between lubricants are in excess of 98%. This means that there is a greater than 98% chance that the relative rankings between the three lubricant/refrigerant mixtures are correct. Quantitative data obtained from both the Falex™ tester and the HPT are summarized in **Table 3.18.** The results show that a consistent lubricant ranking is obtained from both the Falex™ and HPT testers.

Table 3.18 - Ranking of Lubricants by Wear (Case 4)

Lubricant	Ref	Falex™ 3.8 in./s at 1,515 lbf (gram)	HPT 15.9 in./s at 300 lbf (μm)	HPT 15.9 in./s at 400 lbf (μm)
Ester 7	R-134a	0.0013 (B)	1.91 (B)	4.63 (B)
Ester 8	R-134a	0.0167 (W)	18.1 (W)	56.11 (W)
Ester 9	R-134a	0.0039 (I)	3.77 (I)	34.5 (I)
ns : Statistically Not Significant; B: Best; I: Intermediate; W: Worst				

It is interesting to note that when tests are conducted with an aluminum/steel contact (Case 3), ester 7 showed the worst wear characteristics, while with a gray cast iron/steel contact (Case 4), ester 7 showed the best wear characteristics. Therefore, the type of the material pair used for the test is one of the factors that affect the ranking of the lubricants. The results further showed that wear increases significantly with increasing load. Even though the differences of wear results are appreciable among the R134a/ester mixtures, the coefficient of friction is about the same among these mixtures for the two loading conditions. Typical records of the friction coefficient as a function of time are given in **Figure 3.17**.

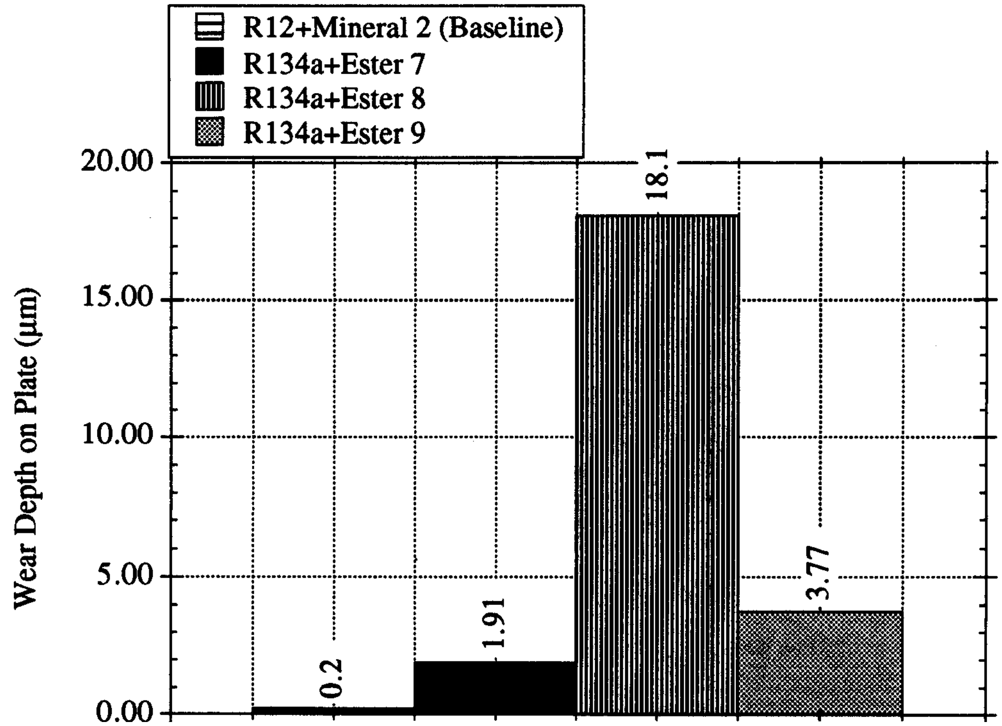


Figure 3.16a - Wear Results (15.9 in./s at 300 lbf)

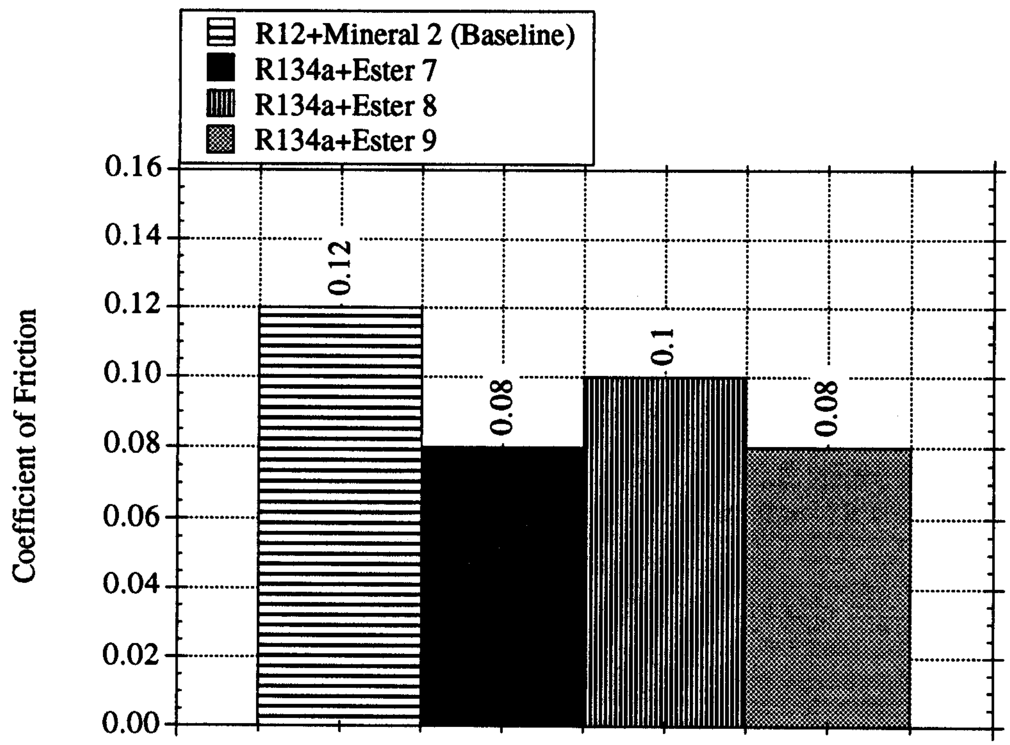


Figure 3.16b - Coefficient of Friction (15.9 in./s at 300 lbf)
Duration of Each Test =1 Hour

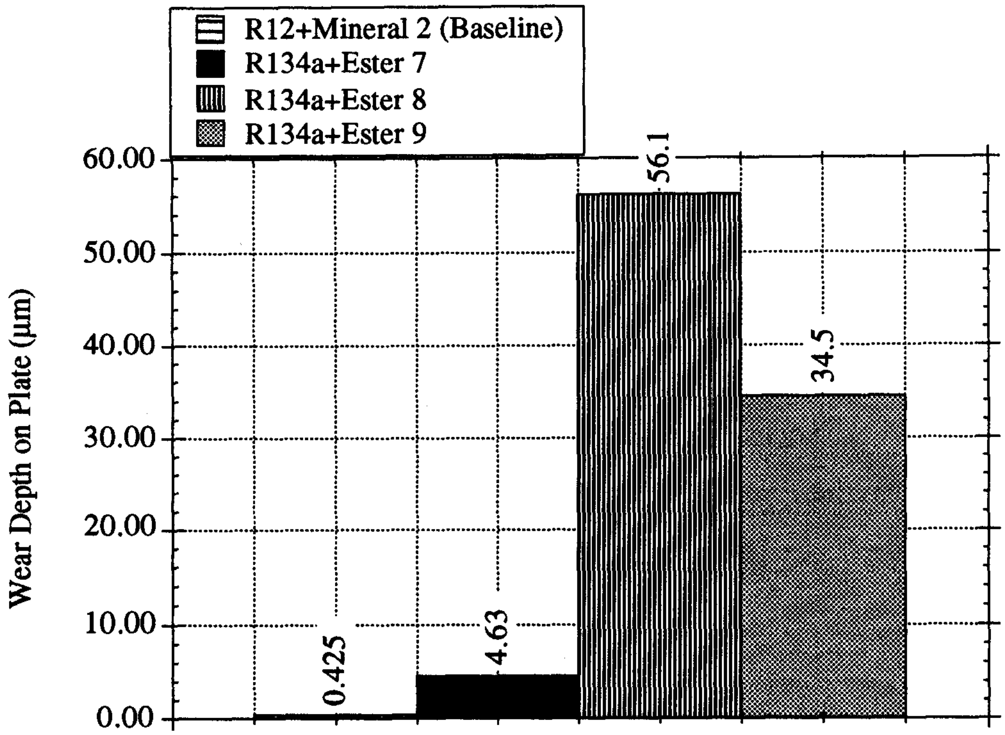


Figure 3.16c - Wear Results (15.9 in./s at 400 lbf)

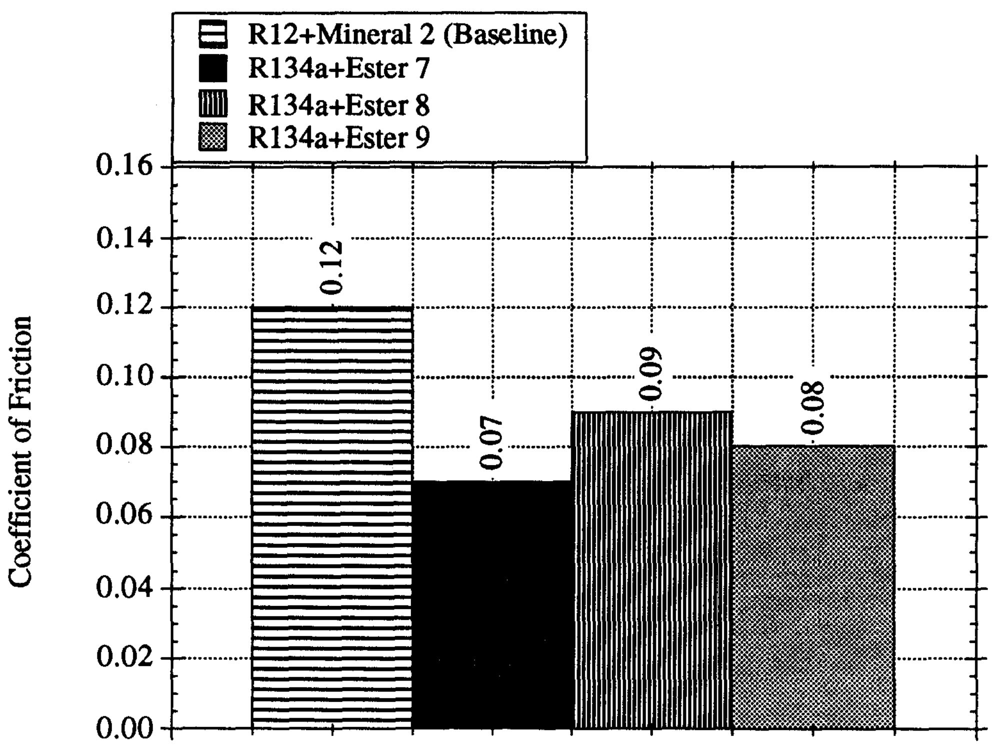
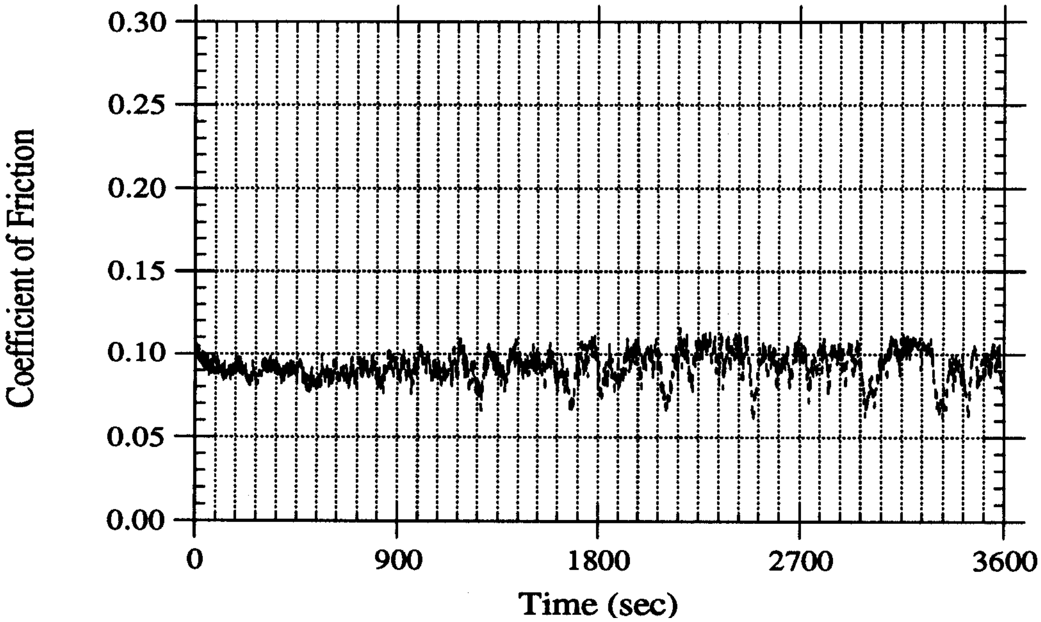
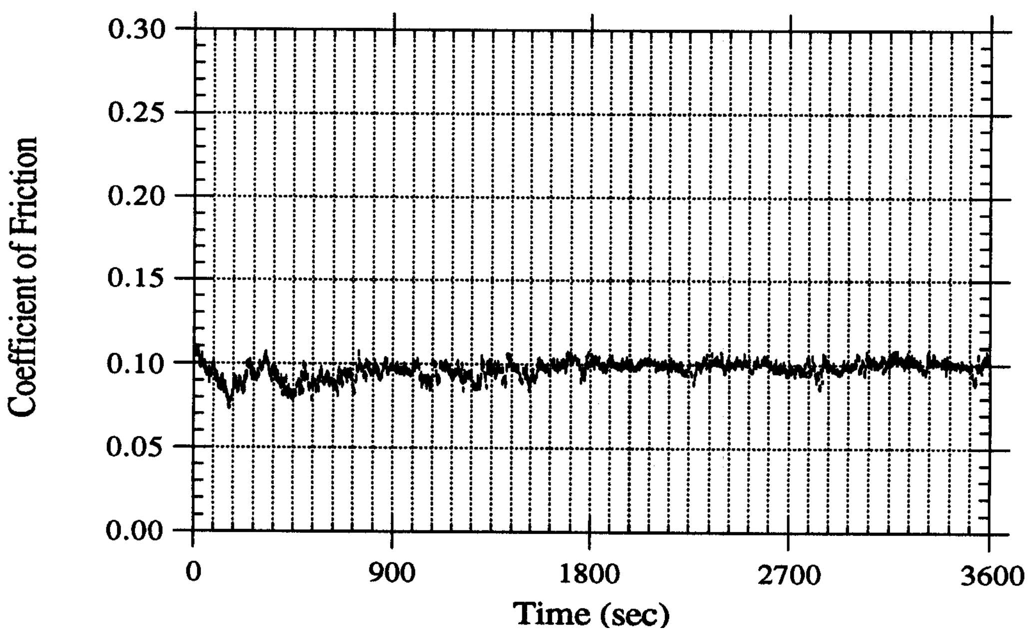


Figure 3.16d - Coefficient of Friction (15.9 in./s at 400 lbf)
Duration of Each Test =1 Hour



(a)



(b)

Figure 3.17 - Coefficient of Friction vs. Time
Test 24TE (15.9 in./s at 400 lbf)
Test 28TE (15.9 in./s at 300 lbf)

Table 3.19 - Statistical Wear Data for Case 4
Comparison of Wear Data Between Lubricants in the Presence of R-134a

Ester 7			Ester 8			Ester 9		
X_1	σ_1	N_1	X_2	σ_2	N_2	X_3	σ_3	N_3
Test condition: 15.9 in./s at 300 lbf								
1.91	0.16263	2	18.1	1.0607	2	3.77	0.25456	2
Test condition: 15.9 in./s at 400 lbf								
4.63	0.89095	2	56.11	3.4083	2	34.5	1.5556	2
X_i = Mean; σ_i = Standard Deviation; N_i = Number of Tests								

Table 3.20 - Confidence Intervals for Case 4 Using Small Sample Theory
Comparison of Wear Data Between Lubricants in the Presence of R-134a

Test condition: 15.9 in./s at 300 lbf									
X_1-X_2	16.19	σ_{12}	1.0731	t_{12}	15.09	n_{12}	2	$\%_{12}$	99
X_2-X_3	14.33	σ_{23}	1.0908	t_{23}	13.14	n_{23}	2	$\%_{23}$	99
X_1-X_3	1.860	σ_{13}	0.3021	t_{13}	6.157	n_{13}	2	$\%_{13}$	98
Test condition: 15.9 in./s at 400 lbf									
X_1-X_2	51.48	σ_{12}	3.5228	t_{12}	14.61	n_{12}	2	$\%_{12}$	99
X_2-X_3	21.61	σ_{23}	3.7465	t_{23}	5.768	n_{23}	2	$\%_{23}$	98
X_1-X_3	29.87	σ_{13}	1.7927	t_{13}	16.66	n_{13}	2	$\%_{13}$	99
$\sigma_{ij} = \sqrt{\dfrac{N_i\sigma_i^2 + N_j\sigma_j^2}{N_i + N_j - 2}}$; $t = \dfrac{X_i - X_j}{\sigma_{ij}\sqrt{1/N_i + 1/N_j}}$					$n = N_i + N_j - 2$; % = Percentage Value for the t Distribution (Confidence Level)				

All the results obtained from Part I are summarized in **Table 3.21**. The statistical significance of the HPT data are also indicated in the table. The statistically significant data are shown as "B" for best, "I" for intermediate or "W" for worst and the statistically insignificant data are shown as "ns" which indicates "not significant" and "na" indicates "not available".

Table 3.21- A Comparison of Lubricant Rankings Based on Wear Data Obtained From the Falex™ Tester and the HPT

Lubricant	Ref	Falex™ (Case 1) 3.8 in./s at 2,120 lbf	HPT (Case 1) 3.8 in./s at 185lbf (mm^3)	HPT (Case 1) 23 in./s at 185 lbf (mm^3)	HPT (Case 1) 23 in./s at 370 lbf (mm^3)
Ester 1	R-134a	na	0.016 (ns)	0.019 (ns)	0.043 (B)
Ester 2	R-134a	na	0.013 (ns)	0.018 (ns)	0.049 (I)
Ester 3	R-134a	na	0.014 (ns)	0.020 (W)	0.061 (W)
Ester 1	Blend	(W)	0.020 (ns)	0.023 (W)	0.054 (ns)
Ester 2	Blend	(B)	0.019 (ns)	0.020 (I)	0.052 (ns)
Ester 3	Blend	(I)	0.011 (B)	0.013 (B)	0.046 (ns)
Lubricant	Ref	Falex™ (Case 2) 3.8 in./s at 854 lbf	HPT (Case 2) 3.8 in./s at 50 lbf (gram)	HPT (Case 2) 23 in./s at 50 lbf (gram)	HPT (Case 2) 3.8 in./s at 100 lbf (gram)
Ester 4	R-134a	(W)	0.085 (ns)	0.402 (W)	0.100 (ns)
Ester 5	R-134a	(B)	0.079 (ns)	0.374 (ns)	0.101 (ns)
Ester 6	R-134a	(I)	0.094 (ns)	0.373 (ns)	0.103 (ns)
Lubricant	Ref	Falex™ (Case 3) 3.8 in./s at 3,330 lbf (gram)	HPT (Case 3) ± 3.8 in./s at 50 lbf (μm)	Falex™ (Case 4) 3.8 in./s at 1,515 lbf (gram)	HPT (Case 4) 15.9 in./s at 300 lbf 15.9 in./s at 400 lbf (μm)
Ester 7	R-134a	0.0106 (W)	29.9 (ns)	0.0013 (B)	1.91 (B) 4.63 (B)
Ester 8	R-134a	Negligible (B)	26.7 (ns)	0.0167 (W)	18.1 (W) 56.11 (W)
Ester 9	R-134a	0.0031 (I)	50.9 (W)	0.0039 (I)	3.77 (I) 34.5 (I)
ns : Statistically Not Significant; na: Not Available; B: Best; I: Intermediate; W: Worst					

3.2. PART II: HIGH PRESSURE TRIBOMETER AND FOUR BALL TESTER vs. COMPONENT TESTING

3.2.1 Overview

In Part II of this project, accelerated compressor wear data for specific components were supplied by five companies. These data were compared with data obtained from the HPT and the Four Ball tester. The HPT operating and environmental conditions used for Case 2 of Part I of this report are the same as those used in Case 5 of this section (Part II). Also, the HPT material pairs and operating and environmental conditions used in Case 3 of Part I of this report are the same as those used in Case 6 of this section.

3.2.2 Material Pairings and Contact Geometry

Operating conditions were chosen to approximately simulate those found at critical contacts in compressors. The HPT contact is between either a large or small radius pin and a flat disk, or a large radius pin on a pad. The large radius pin on a pad or disk was used to approximately simulate the wrist pin/bearing contact or the piston ring/cylinder contact of a reciprocating compressor. It should be noted that a much larger pin radius was needed to completely simulate the contact geometry of these components. However, the 48 in. radius used is the maximum which could be machined with the available equipment. The small radius (0.22 in.) pin on a disk was used to simulate the vane/piston contact of a rotary compressor. A representation of the two contact geometries is given in **Figure 3.18** and **Figure 3.19**, respectively.

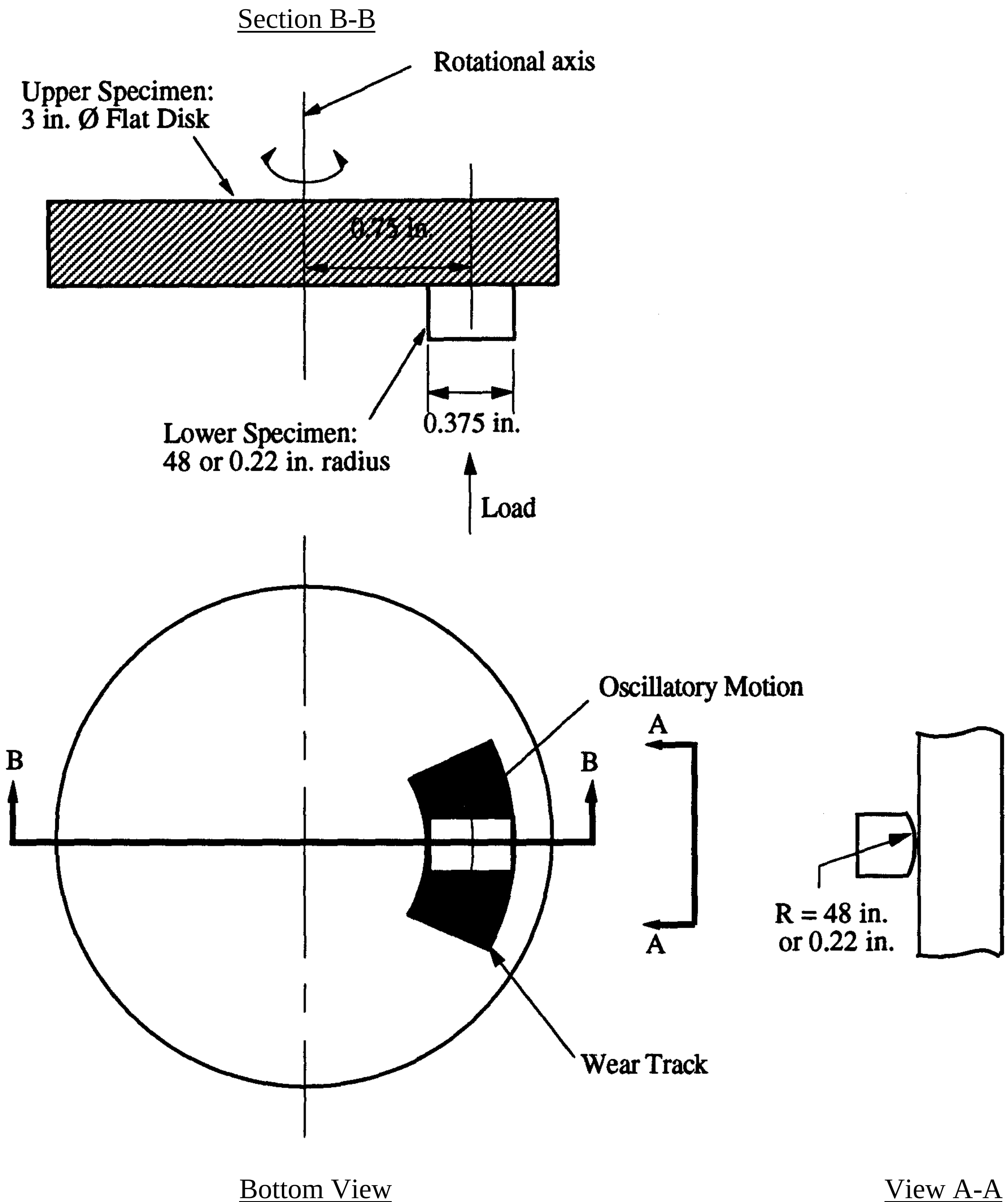


Figure 3.18- Pin on Disk Configuration

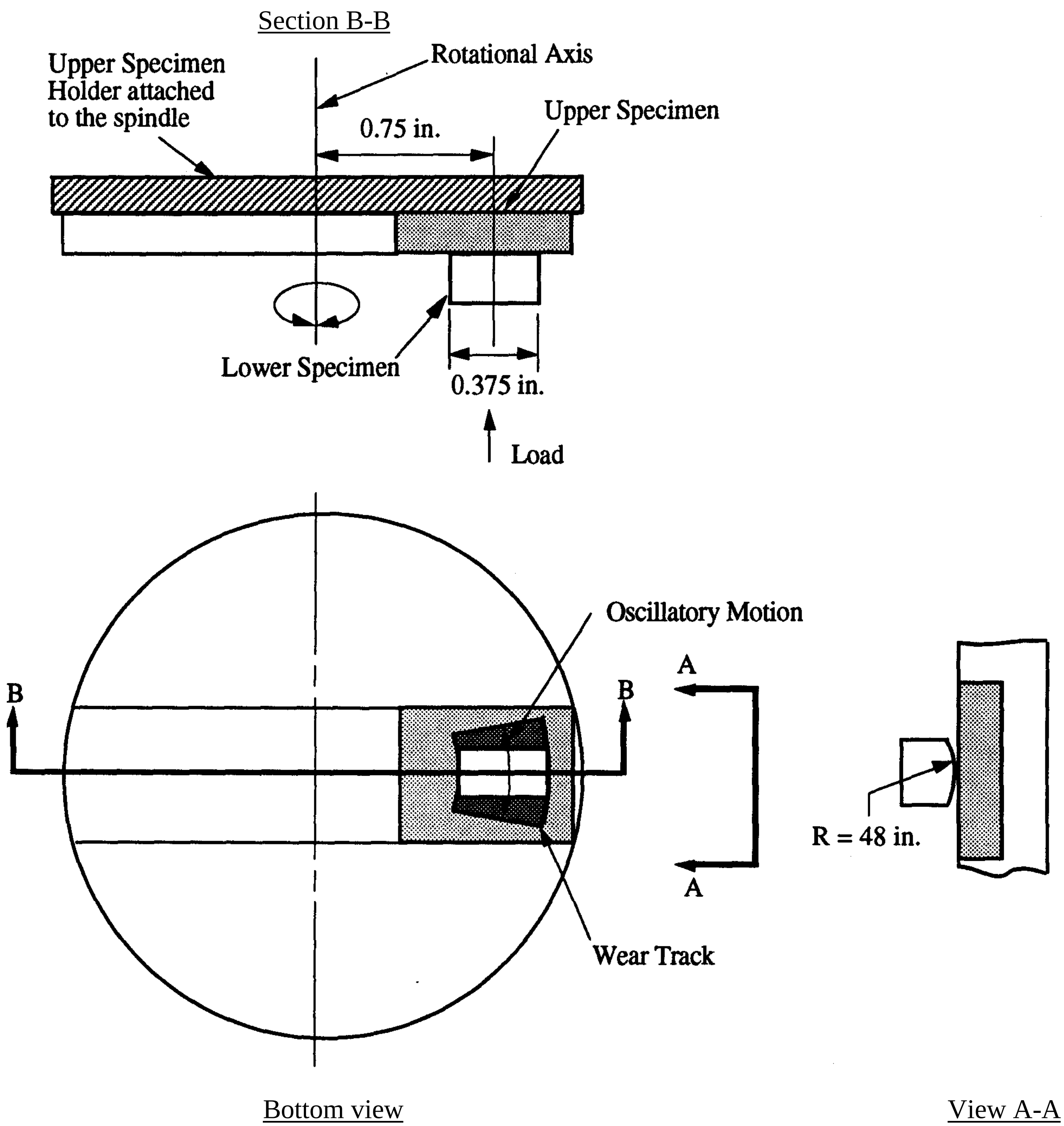


Figure 3.19- Pin on Pad Configuration

The relevant geometrical and material data used for each of the companies are displayed in **Table 3.22** and **3.23**.

Table 3.22 - Contact Geometries and Materials for HPT Tests

Description	Case 5 and 8	Case 6	Case 7	Case 9
Geometry: upper spec.	1 in. x 1 in. Pad	1 in. x 1 in. Pad	3 in. dia. Disk	3 in. dia. Disk
Geometry: lower spec.	48 in. rad. by 0.375 in. long Pin	48 in. rad. by 0.375 in. long Pin	48 in. rad. by 0.375 in. long Pin	0.22 in. rad. by 0.375 in. long Pin
Materials: upper spec.	380 Die Cast Aluminum	380 Die Cast Aluminum	Ductile Cast Iron	Sintered Ferrous Material
Materials: lower spec.	Carburized 1018 Steel	Carburized 1018 Steel	Ductile Cast Iron	Sintered Ferrous Material

Table 3.23 - Material Properties of the Specimens for HPT Tests

Material	Chemical Composition (%)
380 Die Cast Aluminum (Cases 5 and 8)	3.5 Cu, 0.10 Mg, 0.50 Mn, 8.5 Si, 2.0 Fe, 0.50 Ni, 3.0 Zn, 0.35 Sn
380 Die Cast Aluminum (Case 6)	4.5 Cu, 0.10 Mg, 0.50 Mn, 5.5 Si, 1.0 Fe, 1.0 Zn, 0.25 Ti
1018 Steel	0.15-0.20 C, 0.6-0.9 Mn, 0.04 P, 0.05 S
Ductile Cast Iron	3.0-4.0 C, 0.1-1.0 Mn, 1.8-2.8 Si, 0.01-0.1 P, 0.03 S
Sintered Ferrous Material (plate)	94.7-97.5 Fe, 0.35-0.65 C, 1.8-2.2 Ni, 0.4-0.8 Mo, 0.2-0.4 Mn, 0.15 Cu (max)
Sintered Ferrous Material (pin)	90.6-92.2 Fe, 5.5-6.0 Cr, 0.5-0.8 C, 0.7-1.0 Ni

3.2.2.1 Upper Specimens - Surface Characteristics

The surface roughness of each specimen was measured before each test using the Dektak surface profilometer as explained in Appendix C.2. The average roughness and range for the aluminum pads, ductile cast iron plates and sintered ferrous material (SFM) plates are given in **Table 3.24**.

Table 3.24 - Upper Specimens: Surface Roughness

Description	Cases 5 and 8	Case 6	Case 7	Case 9
Materials: upper spec.	380 Die Cast Aluminum	380 Die Cast Aluminum	Ductile Cast Iron	Sintered Ferrous Material
Range:	0.1424 μm - 0.4281 μm Ra	0.9331 μm - 4.6384 μm Ra	0.0641 μm - 0.0841 μm Ra	0.0359 μm - 0.0692 μm Ra
Average:	0.247 μm Ra	2.524 μm Ra	0.073 μm Ra	0.054 μm Ra

The surface hardness of the specimens was measured after each test using a Brinell Hardness Tester for the 380 aluminum pads (Case 6) and a Vickers Hardness Tester for the 380 aluminum pads (cases 5 and 8), cast iron plates and SFM, as explained in Appendix C.4. The designations for the Brinell and Vickers Hardness are HB and Hv, respectively. The average hardness and range for the specimens are given in **Table 3.25**.

Table 3.25 - Upper Specimens: Surface Hardness

Description	Cases 5 and 8	Case 6	Case 7	Case 9
Materials: upper spec.	380 Die Cast Aluminum	380 Die Cast Aluminum	Ductile Cast Iron	Sintered Ferrous Material
Range:	117 - 169 Hv	81 - 99 HB	251 - 334 Hv	198 - 252 Hv
Average:	141 Hv	89 HB	295 Hv	218 Hv

3.2.2.2 Lower Specimens - Surface Characteristics

The surface roughness of each specimen was measured before each test using the Dektak surface profilometer as explained in Appendix C.3. The average roughness and range for the ductile cast iron, carburized 1018 steel and sintered ferrous material (SFM) pins are given in **Table 3.26**.

Table 3.26 - Lower Specimens: Surface Roughness

Description	Cases 5, 6, and 8	Case 7	Case 9
Materials: lower spec.	Carburized 1018 Steel	Ductile Cast Iron	Sintered Ferrous Material
Range:	0.1424 - 0.3051 μm Ra	0.1141 - 0.2341 μm Ra	3.010 - 4.189 μm Ra
Average:	0.217 μm Ra	0.160 μm Ra	3.601 μm Ra

The surface hardness of the specimens was measured using a Vickers Hardness Tester as explained in Appendix C.4. The average hardness and range for the specimens are given in **Table 3.27**.

Table 3.27 - Lower Specimens: Surface Hardness

Description	Cases 5, 6, and 8	Case 7	Case 9
Materials: lower spec.	Carburized 1018 Steel	Ductile Cast Iron	Sintered Ferrous Material
Range:	750 - 880 Hv	250 - 330 Hv	370 - 415 Hv
Average:	810 Hv	295 Hv	392 Hv

3.2.3 Environmental and Operating Conditions

As stated previously, tests on the HPT are conducted under environmental conditions, which approximately simulate those existing in compressors. **Table 3.28** contains the estimated environmental and operating conditions found in the component tests as provided by the participating companies. The environmental and operating conditions used in the HPT are tabulated in **Table 3.29**. Note that some of the environmental pressures and temperatures used with the HPT are somewhat lower than those estimated to exist in the compressor due to the limitations of the HPT.

In Cases 5, 6 and 8, the component pressure is given as load per unit projected area. Under nominal operating condition, hydrodynamic lubrication conditions can be expected for the wrist pin/bearing system. The HPT tests, which approximately simulate this component, were conducted at lower speeds and with less conformal geometry so that measurable wear can be obtained. For Cases 7 and 9 , the HPT loads chosen are higher than the expected loads on the component so that again measurable wear is obtained under the test conditions used. The obvious assumption made is that the same dominant wear mechanisms occur in both the specimen and component tests. The HPT operating conditions used are only severe enough to make a comparative wear study of various lubricants and even though these conditions produce more severe lubrication conditions that could nominally be expected in the components, the wear mechanisms in both component and HPT tests are not expected to be appreciably different.

Table 3.28 - Component Environmental and Operating Conditions Provided by the Companies

Operating Conditions	Case 5	Case 6	Case 7	Case 8	Case 9
Compressor Type	Rec.C.	Rec.C.	Rec.C.	Rec.C.	Rot.C.
Components	WP:B	WP:B	PR:C	WP:B	V:P
Pressure or Load	*4,000 psi	*2,500 psi	na	*6,000 psi	**43-65 lb/in.
Type of Motion	Oscillatory	Oscillatory	Oscillatory	Oscillatory	Oscillatory
Env. Pressure (psig)	30	7	170	60	400
Env. Temperature (°F)	275	325	160	140	200
PR:C = Piston Ring:Cylinder; V:P = Vane:Piston; WP:B = Wrist Pin:Bearing Rec.C. = Reciprocating Compressor; Rot.C. = Rotary Compressor * Load per Unit Projected Area; ** Load per Unit Length					

Table 3.29 - Environmental and Operating Conditions Used in the HPT

Operating Conditions	Case 5	Case 6	Case 7	Case 8	Case 9
Contact Load (lbf)	50	25	350	50	250
Initial Maximum Hertz Contact Stress (psi)	4,000	2,800	10,000	4,000	105,000
Type of Motion	Oscillatory	Oscillatory	Oscillatory	Oscillatory	Oscillatory
Average Speed (in./s)	± 3.8	± 3.8	± 20.6	± 3.8	± 20.6
Angular Frequency	4.61 Hz	4.61 Hz	5.00 Hz	4.61 Hz	5.00 Hz
Angular Amplitude	± 15°	± 15°	± 75°	± 15°	± 75°
Env. Pressure (psig)	30	7	170	60	225
Env. Temperature (°F)	266	266	160	140	200
Test Duration	1 hr	1 hr	1 hr	1 hr	1 hr

3.2.4 Lubricants and Refrigerants

Table 3.30 contains information on the various properties of the lubricants used throughout the testing program, including lubricant type, viscosity and solubility data. Those lubricants which are known to be the same are designated as such. The viscosity was measured with a Brookfield Digital Viscometer as described in Appendix C.5. The weight percent of refrigerant saturated into the lubricant was obtained by the sampling facility described in Appendix C.6. The sampling procedure is based on ASHRAE standards [3]. In this procedure, a sample of the lubricant/refrigerant mixture is taken during a test and then weighed. The refrigerant is carefully removed from the lubricant, and the final weight is measured. The amount of refrigerant saturated into the lubricant can then be determined. The lubricant/refrigerant combinations used for the HPT tests are the same as those used in the compressor tests. These combinations are given in **Table 3.31**. All tests were conducted in both refrigerant and air environments to better understand the tribological behavior of the contact with and without the presence of the refrigerant saturated into the lubricant.

Table 3.30 - Lubricant Viscosity, Solubility and Environmental Test Conditions

Lubricant	Type	Viscosity, μ (cS)		Refrig	Pressure	Temp	Solubility
		@40°C	@100°C		(psi)	(°F)	% Ref in Oil
Mineral 2	N	29.8	4.4	R-12	30	266	2.0
#Ester 4	PE	23.9	4.9	R-134a	30	266	0.3
#Ester 5 *	PE	23.9	4.9	R-134a	30	266	0.3
Ester 6 *	PE	32.0	5.6	R-134a	30	266	0.5
#Ester 7 *	PE	23.9	4.9	R-134a	7	266	0.3
Ester 8	PE	32.0	5.4	R-134a	7	266	0.3
Ester 9	PE	32.0	5.6	R-134a	7	266	0.1
Mineral 2	N	29.8	4.4	R-12	7	266	0.1
Ester 10 *	PE	61.7	8.1	R-134a	170	160	4.7
Ester 11	PE	67.6	8.7	R-134a	170	160	4.1
Ester 12	PE	72.3	9.8	R-134a	170	160	10.4
Ester 13	DAE	7.7	2.3	R-134a	60	140	1.2
Ester 14	DAE	9.4	2.8	R-134a	60	140	1.3
Ester 15	PE	9.5	2.6	R-134a	60	140	1.0
Alkbenz 1 *	AB	28.0	4.1	R-22	225	200	18.0
Alkbenz 2 *	AB	57.0	5.8	R-22	225	200	15.0
Mineral 3 *	P/N	35.6	7.0	R-22	225	200	7.8

AB: Alkylbenzene; DAE: Dibasic-acid ester; N: Naphthenic mineral oil;
P/N: Parrafinic/Naphthenic mineral oil mixture; PE: Polyolester; * Formulated
Same Base Lubricant

Table 3.31 - Lubricant/Refrigerant Combinations

Company	Refrigerant	Lubricant
Case 5	R-12 (baseline)	Mineral 2
	R-134a	Ester 4
	R-134a	Ester 5
	R-134a	Ester 6
Case 6	R-12 (baseline)	Mineral 2
	R-134a	Ester 7
	R-134a	Ester 8
	R-134a	Ester 9
Case 7	R-134a	Ester 10
	R-134a	Ester 11
	R-134a	Ester 12
Case 8	R-134a	Ester 13
	R-134a	Ester 14
	R-134a	Ester 15
Case 9	R-22	Alkbenz 1
	R-22	Alkbenz 2
	R-22	Mineral 3

3.2.5 Four Ball Tests

A Four Ball Wear Test Machine was used throughout this study to determine the relative wear preventative properties of the various lubricants without a refrigerant environment. A schematic configuration of Four Ball tester is shown in Appendix A (**Figure A.5**).

3.2.5.1 Summary of Method

Three 0.5 inch diameter steel balls are clamped together and covered with the lubricant to be evaluated. A fourth 0.5 inch diameter steel ball, referred to as the top ball, is loaded against the lower three balls. Lubricants are evaluated by comparing the average size of the wear scar diameters on the three lower balls.

3.2.5.2 Materials

The chrome alloy steel test balls used with the Four Ball Tester are AISI standard steel No. E-52100, with diameter of 12.7 mm (0.5 in.) Grade 25 EP (Extra Polish) with a Rockwell C hardness ranging from 64 to 66.

3.2.5.3 Test Conditions for The Four Ball Wear Test Machine

The test conditions and allowable limits used with the Four Ball Tester are tabulated in **Table 3.32** [4].

Table 3.32 – Test Conditions for The Four Ball Tester

Temperature	75 ± 2°C (167 ± 4°F)
Speed	1200 ± 60 rpm
Duration	60 ± 1 min
Load	392 ± 2 N (40 ± 0.2 kgf)

3.2.6. Results and Discussion

The HPT tests were conducted using the material pairings, contact geometries, and environmental and operating conditions given in **Tables 3.22** and **3.29**. The tests were conducted in both lubricant/refrigerant (**Table 3.31**) and lubricant/air environments. The lubricant/air tests helped establish the influence of the refrigerant on the behavior and ranking of the lubricants. In general, lubricants in refrigerant environments did not give the same rankings as lubricants in air environments, indicating the effects of the refrigerant on the general lubrication behavior of the lubricant. Additionally, standardized Four Ball tests [4] were conducted "in-house", for all of the lubricants, to make a preliminary evaluation of the anti-wear properties of the lubricants acting alone. Due to time limitations, only two tests per lubricant were conducted with the Four Ball tester. The raw wear data obtained with the Four Ball tester and statistical analysis of these data are given in Appendix F.

For each operating condition, from two to four tests were conducted using the HPT for each lubricant/refrigerant and for each lubricant acting alone in an air environment. The raw data from all of the tests conducted have been tabulated in Appendix E. The friction coefficients, specimen and chamber temperatures, forces and moments acting on specimens were monitored and recorded continuously throughout each test using a computer data acquisition system [5]. The reported friction coefficient is an average value of the sum total of friction data points collected throughout the duration of a test. Upon completion of a test, the amount of wear on the pad and/or disk was evaluated using the wear scar measuring technique as described in Appendix C. The wear data obtained were compared with the qualitative component wear data supplied by the participating companies.

3.2.6.1 Case 5: HPT Friction and Wear Results - Carburized 1018 Steel Pin on 380 Die Cast Aluminum Pad (Reciprocating Compressor)

Figure 3.20 shows a typical surface profile of a wear scar on a 380 die cast aluminum pad. It is clear from this profile that the damage to the pad specimen is very uniform. **Figure 3.21** shows actual wear scar on the aluminum pad caused by a large radius mating pin made from carburized 1018 steel. Since the pins were much harder (810 Hv) than the aluminum pads (141 Hv), the wear was almost exclusively confined to the pads and only slight polishing marks appeared on the pins. Therefore, all of the wear measurements and subsequent lubricant rankings were obtained from the wear on the aluminum pads.

Figure 3.22a is a comparative bar chart which graphically represents the wear damage on the die cast aluminum pads for each lubricant/refrigerant and for each lubricant acting alone in an air environment. Additionally, baseline R-12/mineral lubricant tests were run for comparative purposes. All tests operated in the boundary lubrication regime. The decrease in the effective viscosity of the lubricant, due to the presence of the refrigerant, may or may not necessarily explain the apparent increase in wear for lubricants in refrigerant environments as compared to lubricants acting alone in an air environment. In the boundary lubrication regime, the wear behavior is mainly determined by the materials in contact, lubricant additives and chemical interactions at material/lubricant interface. All these effects can be significantly influenced by environmental conditions. Similarly, the coefficient of friction obtained in an air environment is smaller than that obtained in refrigerant environments (**Figure 3.22b**). For both the friction and wear data, the speed and loading conditions are specified. The raw data, given in Appendix E (**Table E.1**), has an average scatter for the wear between repeated tests of 7.6%. The average scatter of the friction coefficients for all of the tests are within 7.4%. Once again, the HPT maintains very good repeatability.

Figure 3.23 is a graphic representation of the friction coefficient plots for both a lubricant/refrigerant mixture oscillation test and a lubricant/air oscillation test. These figures illustrate the effects of the refrigerant on the apparent coefficient of friction.

Figure 3.24 depicts a typical axial force record for a test. Every test run has an axial force record similar to this and is checked to make sure that the appropriate load was continually applied throughout all tests.

A comparison between the wear data obtained from the HPT, Four Ball Tester and the component is summarized in **Table 3.33**. For both the component tests and the HPT, Ester 5 is clearly the best in terms of wear. The differences between the Ester 4 and Ester 6 only yield a confidence interval of 58% (**Table 3.35**). In general, confidence intervals in excess of 80%, 90% and 95% are considered acceptable, very good and excellent, respectively. Anything less than 80% is generally considered poor confidence and is regarded as statistically insignificant. Based upon a lack of statistical significance, it would be inappropriate to assign a rank to either of these lubricants. Once again, the relative differences in the qualitative rankings provided by the companies are not known.

For lubricant in an air environment, the HPT and the Four Ball Tester both rank Ester 5, Ester 6 and Ester 4 as best , intermediate and worst, respectively. The differences between Ester 6 and the Ester 4 for the lubricant/air tests yield a confidence interval of 68% (**Table 3.37**).

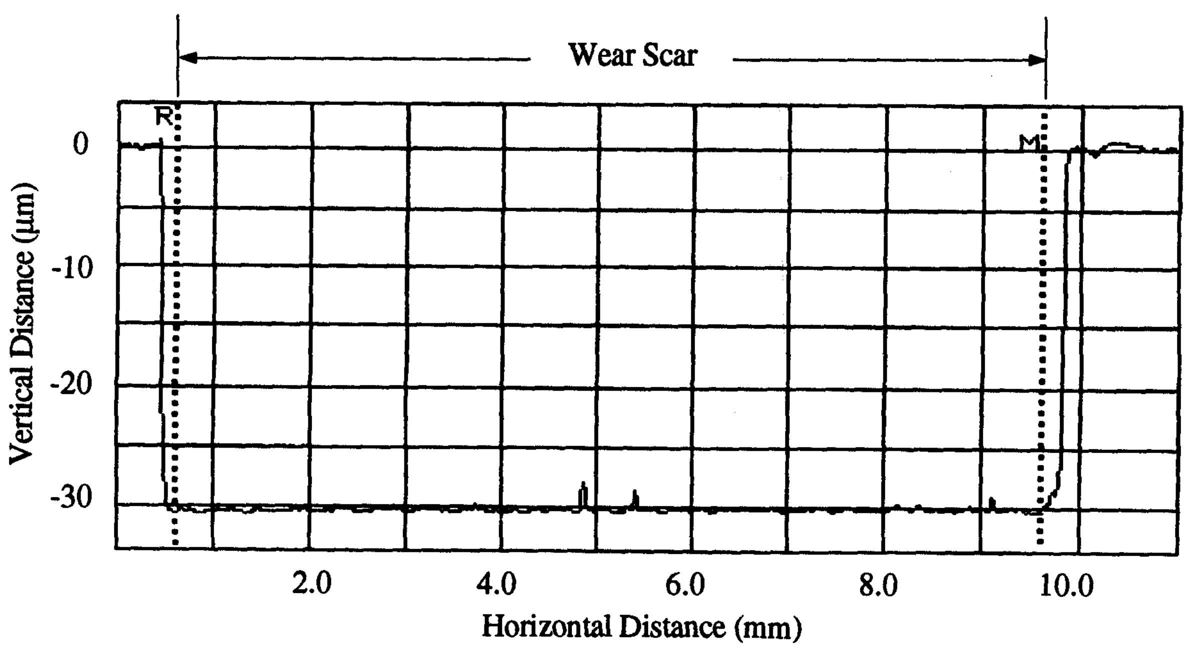
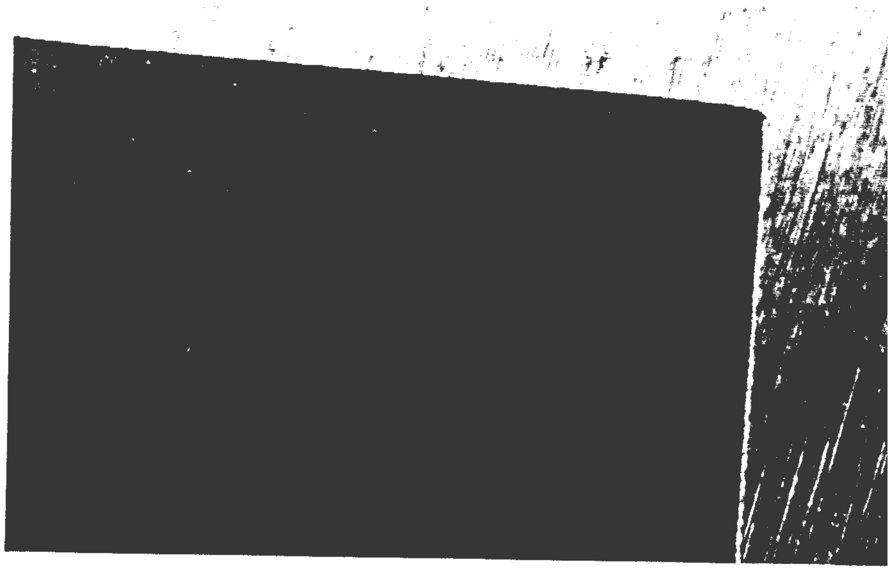
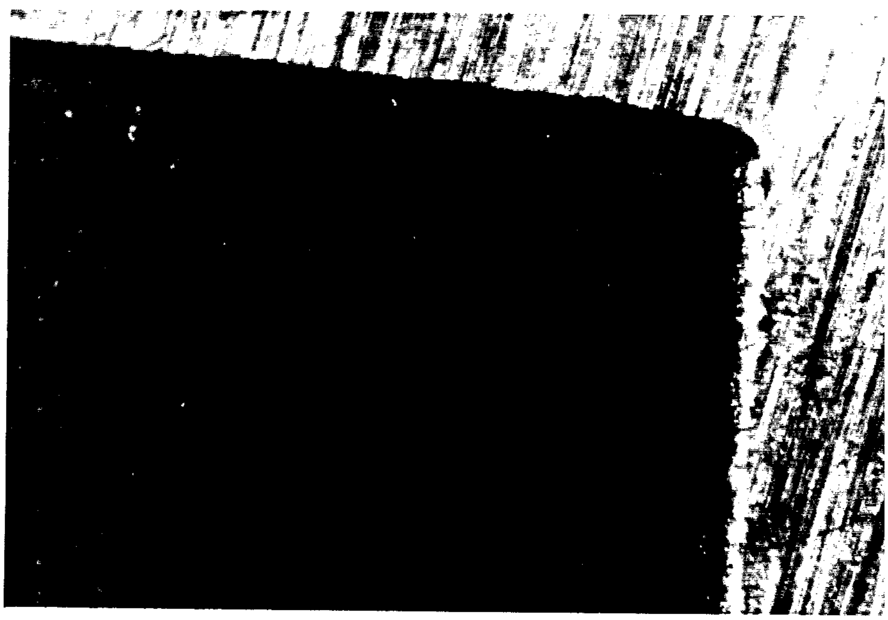


Figure 3.21 - Surface Profile of 380 Die Cast Aluminum Pad Specimen No. 13



(a)



(b)

Figure 3.20 - Typical Wear Scar on a 380 Die Cast Aluminum Pad
(a) Test 13CO-2/SN 13 (10 X Mag.)
(b) Test 13CO-2/SN 13 (40 X Mag.)

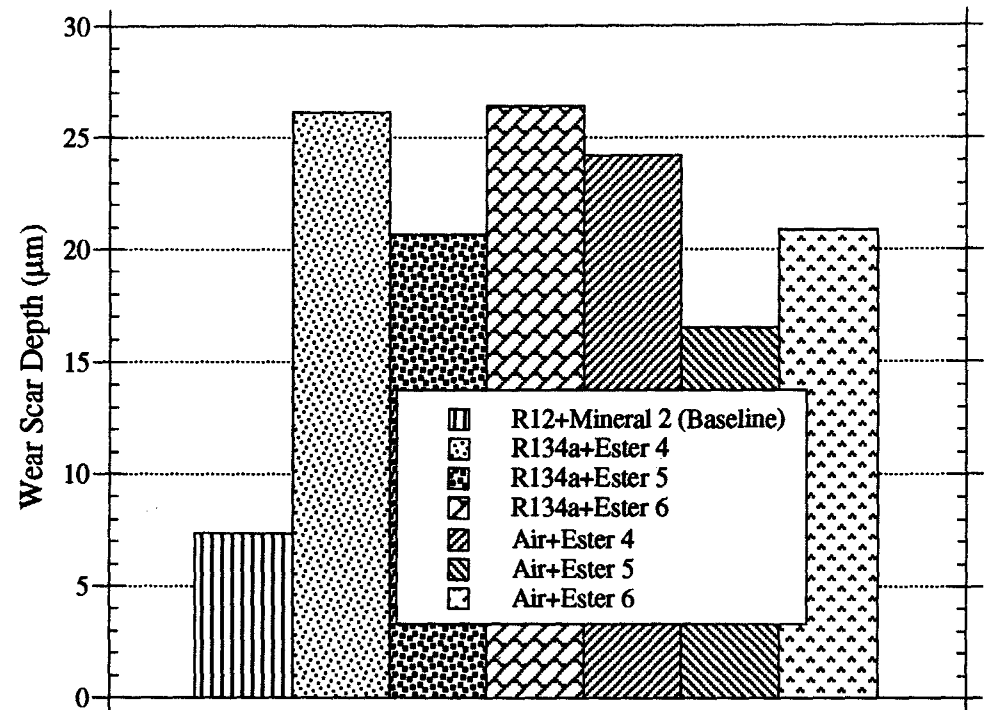


Figure 3.22a - Wear Results (± 3.8 in./s at 50 lbf)

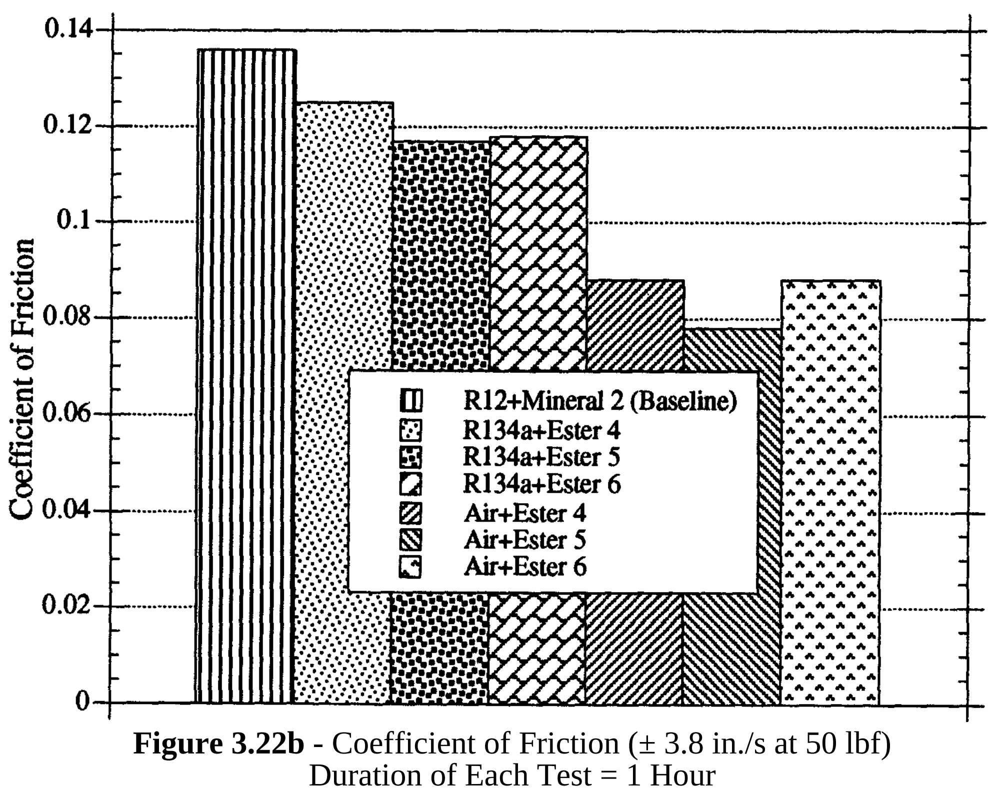
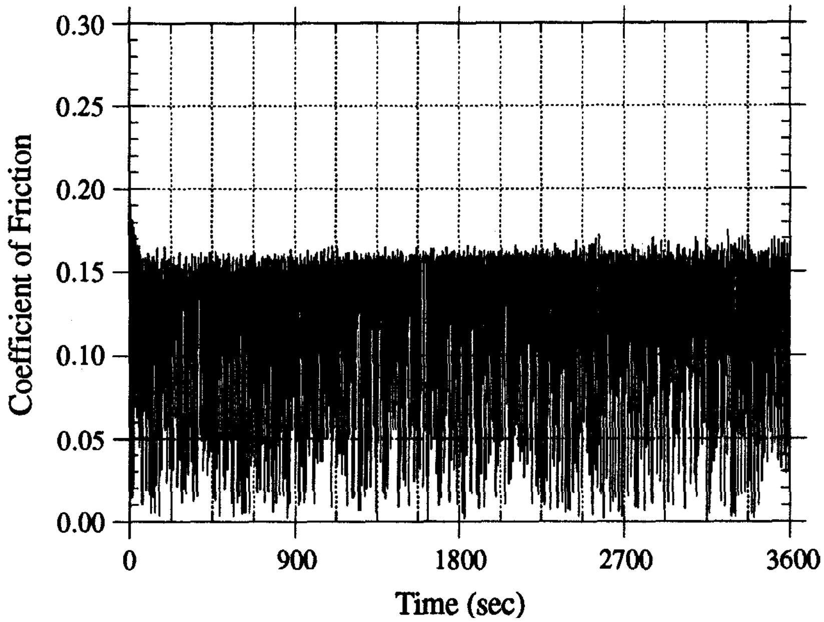
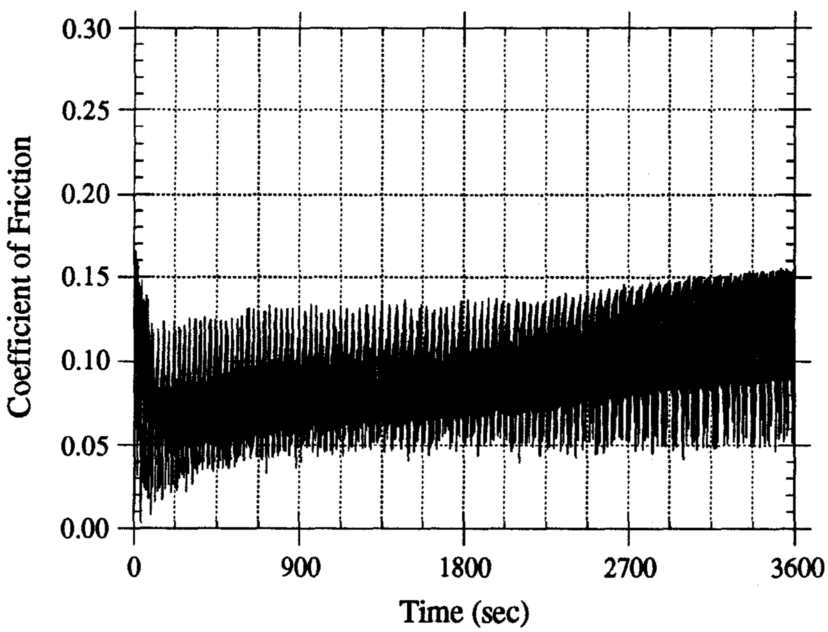


Figure 3.22b - Coefficient of Friction (± 3.8 in./s at 50 lbf)
Duration of Each Test = 1 Hour



(a)



(b)

Figure 3.23 - Coefficient of Friction vs. Time
(a) Test 13CO-2: Lubricant/Refrigerant Environment (± 3.8 in./s at 50 lbf)
(b) Test 21CO-2: Lubricant/Air Environment (± 3.8 in./s at 50 lbf)

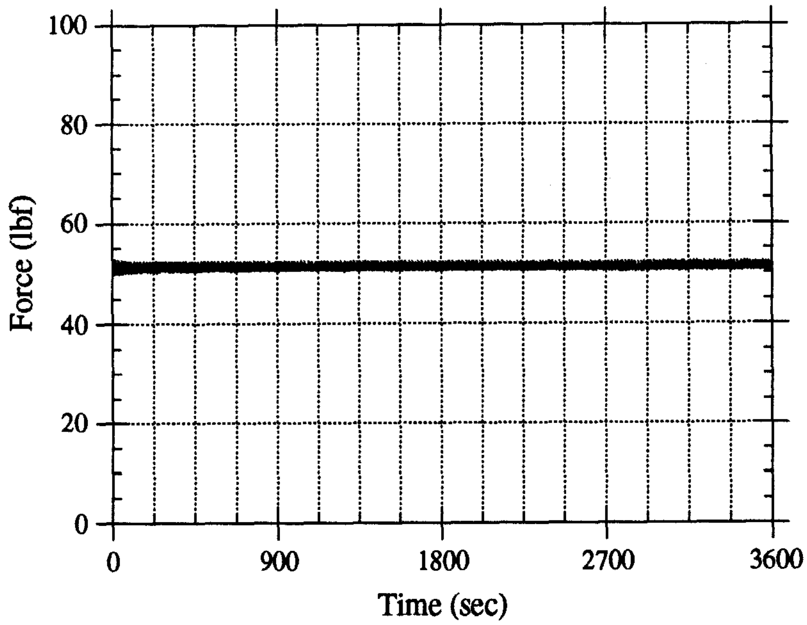


Figure 3.24 - Axial Force Record: Test 13CO-2

Table 3.33 - Ranking of Lubricants by Wear (Case 5)

Lubricant	Environment	Component Qualitative Ranking	HPT ± 3.8 in./s at 50 lbf Wear Scar Depth (μm)	Four Ball Tester Average Wear Scar Diameter (mm)
Ester 4	R-134a	(W)	26.1 (ns)	na
Ester 5	R-134a	(B)	20.7 (B)	na
Ester 6	R-134a	(I)	26.4 (ns)	na
Ester 4	Air	na	24.2 (ns)	0.78 (ns)
Ester 5	Air	na	16.5 (B)	0.58 (B)
Ester 6	Air	na	20.9 (ns)	0.77 (ns)
ns : Statistically Not Significant; na: Not Available; B: Best; I: Intermediate; W: Worst				

Based on a statistical analysis similar to that in Part 1, the confidence intervals of the relative differences between lubricants, as tabulated in **Table 3.35**, are 95% between Ester 5 and Ester 6, 93% between Ester 4 and Ester 5 and only 58% between Ester 4 and Ester 6 for the lubricant/refrigerant tests. Similarly, the lubricant/air tests produced confidence intervals (**Table 3.37**) of 75% between Ester 5 and Ester 6, 69% between Ester 4 and Ester 6 and 86% between Ester 4 and Ester 5. A minimum of four and two tests were run for the lubricant/refrigerant and lubricant/air conditions, respectively.

Table 3.34 - Statistical Wear Data for Case 5
Comparison of HPT Wear Data Between Lubricants in the Presence of R-134a

Ester 4			Ester 5			Ester 6		
X_1	σ_1	N_1	X_2	σ_2	N_2	X_3	σ_3	N_3
26.1	1.4408	4	20.7	3.7864	4	26.4	0.6758	4
X_i = Mean; σ_i = Standard Deviation; N_i = Number of Tests								

Table 3.35 - Confidence Intervals for Case 5 Using Small Sample Theory
Comparison of HPT Wear Data Between Lubricants in the Presence of R-134a

X_1-X_2	5.4	σ_{12}	3.308	t_{12}	2.309	n_{12}	6	$\%_{12}$	93
X_2-X_3	5.7	σ_{23}	3.410	t_{23}	2.567	n_{23}	6	$\%_{23}$	95
X_1-X_3	0.3	σ_{13}	1.299	t_{13}	0.327	n_{13}	6	$\%_{13}$	58
$\sigma_{ij} = \sqrt{\dfrac{N_i\sigma_i^2 + N_j\sigma_j^2}{N_i + N_j - 2}}; t = \dfrac{X_i - X_j}{\sigma_{ij}\sqrt{1/N_i + 1/N_j}}$					$n = N_i + N_j - 2$ % = Percentage Value for the t Distribution (Confidence Level)				

Table 3.36 - Statistical Wear Data for Case 5
Comparison of HPT Wear Data Between Lubricants in Air

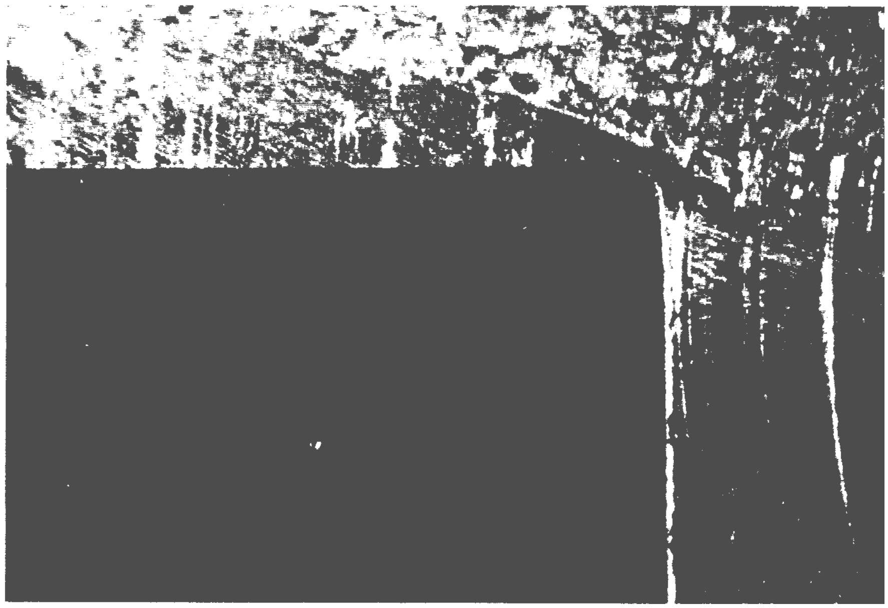
Ester 4			Ester 5			Ester 6		
X_1	σ_1	N_1	X_2	σ_2	N_2	X_3	σ_3	N_3
24.2	3.0406	2	16.5	0.7071	2	20.9	3.1113	2
X_i = Mean; σ_i = Standard Deviation; N_i = Number of Tests								

Table 3.37 - Confidence Intervals for Case 5 Using Small Sample Theory
Comparison of HPT Wear Data Between Lubricants in Air

X_1-X_2	7.7	σ_{12}	3.122	t_{12}	2.466	n_{12}	2	$\%_{12}$	86
X_2-X_3	4.4	σ_{23}	3.191	t_{23}	1.379	n_{23}	2	$\%_{23}$	75
X_1-X_3	3.3	σ_{13}	4.350	t_{13}	0.759	n_{13}	2	$\%_{13}$	69
$\sigma_{ij} = \sqrt{\dfrac{N_i\sigma_i^2 + N_j\sigma_j^2}{N_i + N_j - 2}}; t = \dfrac{X_i - X_j}{\sigma_{ij}\sqrt{1/N_i + 1/N_j}}$					$n = N_i + N_j - 2$ % = Percentage Value for the t Distribution (Confidence Level)				

3.2.6.2 Case 6: HPT Friction and Wear Results - Carburized 1018 Steel Pin on 380 Die Cast Aluminum Pad (Reciprocating Compressor)

Figure 3.25 shows a typical wear scar on a 380 die cast aluminum pad caused by a large radius mating pin made from carburized 1018 steel. Again, wear mainly occurred on the pad. Therefore, all of the wear measurements and subsequent lubricant rankings were obtained from the surface profiles of the aluminum pad. **Figure 3.26** shows a typical surface profile of a wear scar on the aluminum pad. The damage to the aluminum pad specimen is again very uniform.



(a)

(b)

Figure 3.25 - Typical Wear Scar on a 380 Die Cast Aluminum Pad
(a) Test 20TE-II/SN 20 (10 X Mag.)
(b) Test 20TE-II/SN 20 (40 X Mag.)

Figure 3.27a is a comparative bar chart which graphically represents the wear damage on the 380 die cast aluminum pads for each lubricant/refrigerant mixture and for each lubricant acting alone in an air environment. Again, baseline R-12/mineral lubricant tests were run for comparative purposes. As in the Case 5, the figure depicts the damaging effects of the lubricant/refrigerant mixture as opposed to the lubricant acting alone. R-134a does not possess the beneficial lubricative properties of R-12, as is evident in **Figure 3.27a**. Again, the lack of correlation between friction and wear becomes evident when analyzing the two graphs (**Figure 3.27a** and **3.27b**). In terms of wear, the R-12/mineral lubricant is the best, but in terms of friction, it is among the worst. Additionally, for reasons unknown, the friction is also worse for all three lubricant/air cases as compared to the lubricant/refrigerant tests, a trend that only exists for Case 6. For both the friction and wear data, the speed and loading conditions are specified. The raw data, given in Appendix E (**Table E.2**), has an average scatter for the wear between repeated tests of 7.8%. The average scatter of the friction coefficients for all of the tests is within 9.1%. Once again, the HPT maintains very good repeatability.

Figure 3.28 is a graphic representation of the friction coefficient plots for both a lubricant/refrigerant mixture oscillation test and a lubricant/air oscillation test. The general trend is the same for both plots, first a "run-in" followed by an approximate steady state region.

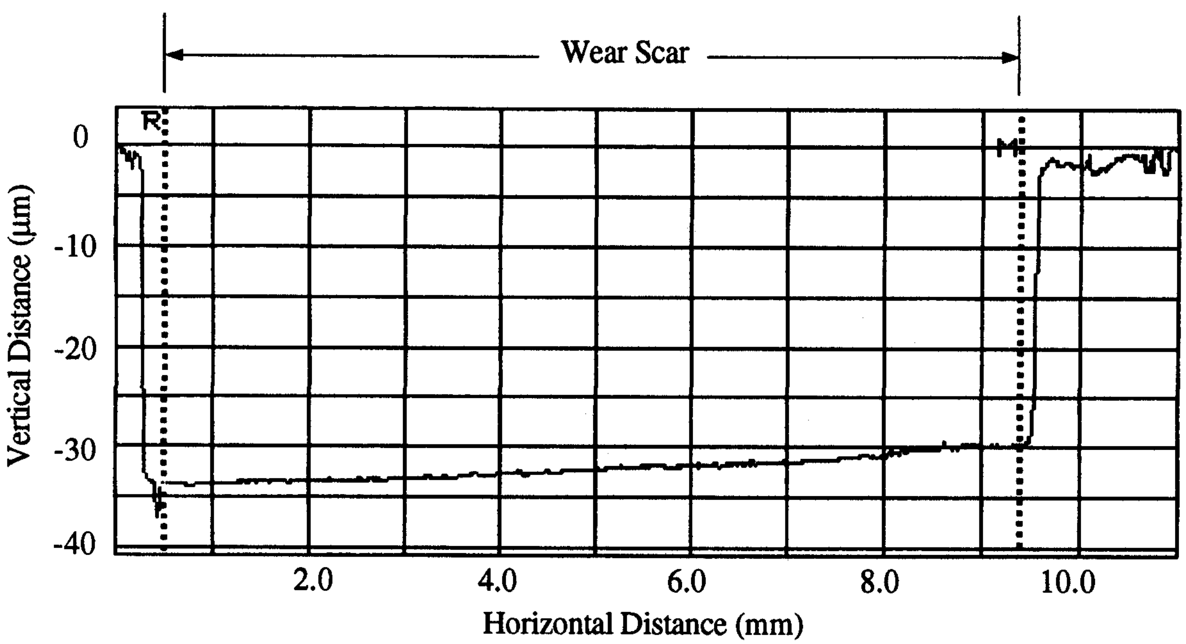


Figure 3.26 - Surface Profile of 380 Die Cast Aluminum Pad Specimen No. 20

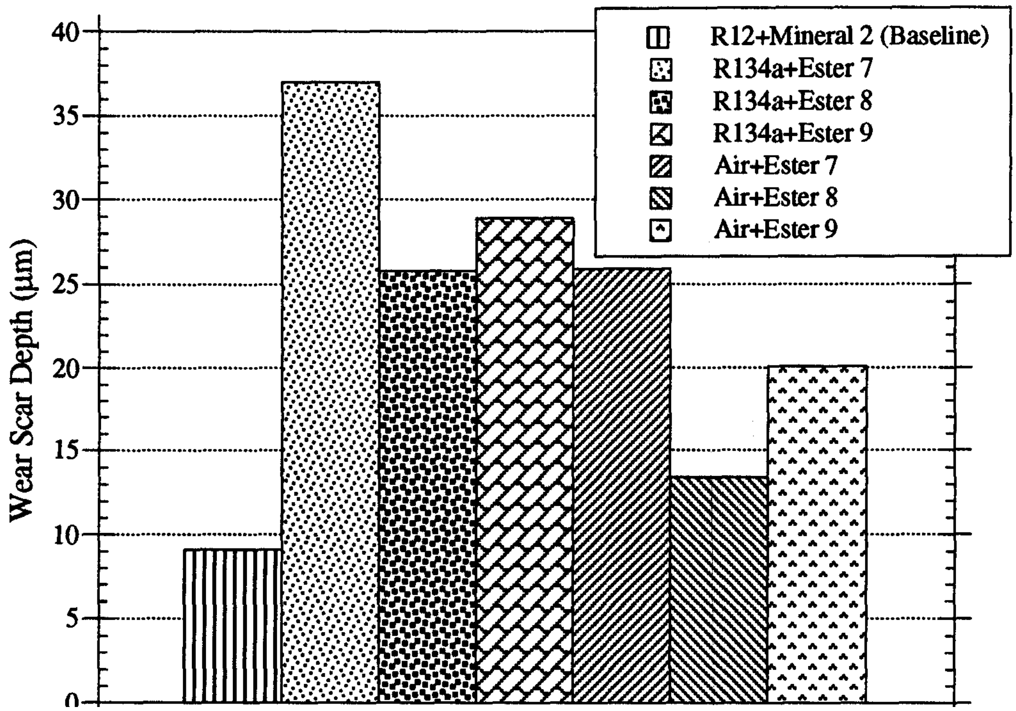


Figure 3.27a - Wear Results (± 3.8 in./s at 25 lbf)

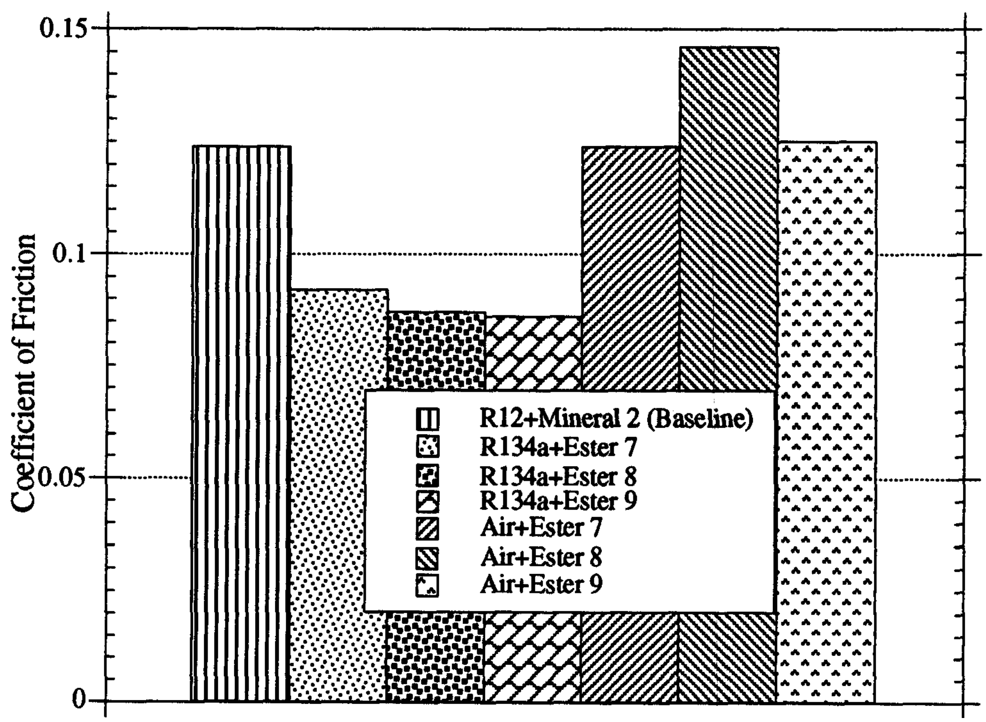
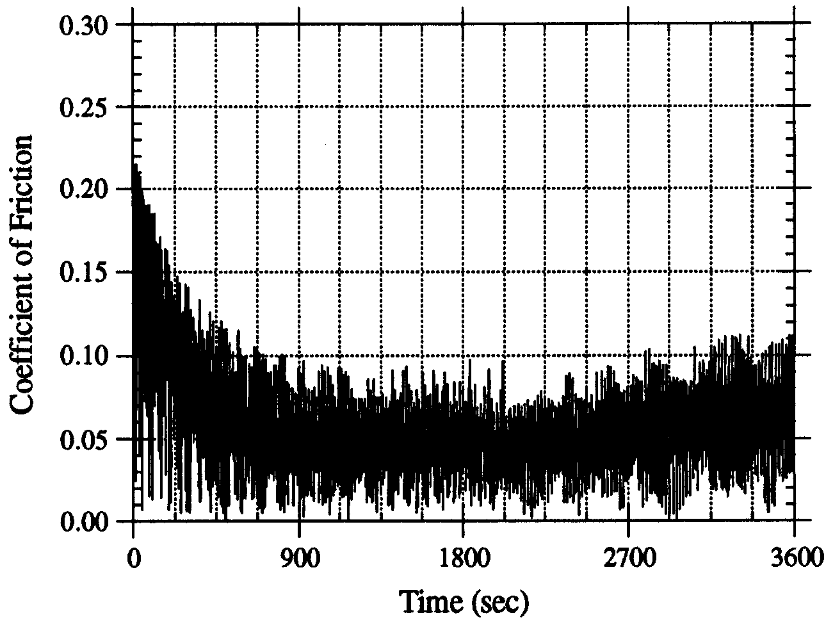
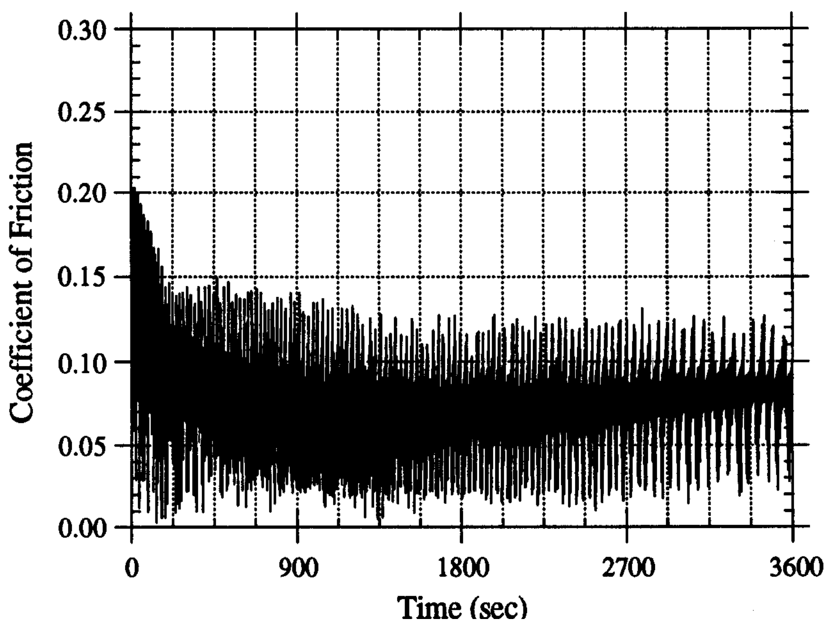


Figure 3.27b - Coefficient of Friction (± 3.8 in./s at 25 lbf)
Duration of Each Test = 1 Hour



(a)



(b)

Figure 3.28 - Coefficient of Friction vs. Time
(a) Test 20TE-II: Lubricant/Refrigerant Environment (± 3.8 in./s at 25 lbf)
(b) Test 31TE-II: Lubricant/Air Environment (± 3.8 in./s at 25 lbf)

Figure 3.29 depicts a typical axial force record for a test. It maintains an approximate constant value throughout the test.

A comparison between the wear data obtained from the HPT, Four Ball Tester and the component is summarized in **Table 3.38**. Both the HPT rankings and the rankings provided by the company show that Ester 8 is the best lubricant, however, the relative ranking between Ester 7 and Ester 9 obtained from the HPT is different from that obtained from the component tests. The quantitative results from the HPT show that Ester 7 is clearly the worst, with the difference between Ester 8 and Ester 9 not being statistically significant (**Table 3.40**).

For the lubricant/air tests, the HPT and the Four Ball Tester do not agree. The HPT ranks Ester 8, Ester 9 and Ester 7 as best, intermediate and worst, respectively. Whereas, the Four Ball Tester ranks Ester 9, Ester 7 and Ester 8 as best, intermediate and worst, respectively. It is quite possible that the materials in contact for these tests may have influenced the ranking. In general, the 380 die cast aluminum specimens tended to be porous. The results for the lubricant/air tests produced confidence intervals of no less than 97% (**Table 3.42**).

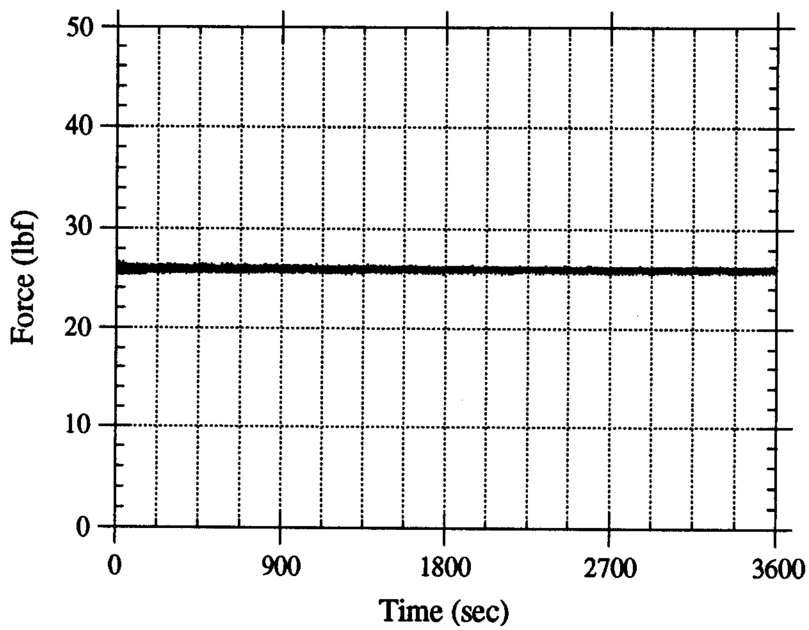


Figure 3.29 - Axial Force Record: Test 20TE-II

Table 3.38 - Ranking of Lubricants by Wear (Case 6)

Lubricant	Environment	Component Qualitative Ranking	HPT ± 3.8 in./s at 25 lbf Wear Scar Depth(μm)	Four Ball Tester Average Wear Scar Diameter (mm)
Ester 7	R-134a	(I)	37.0 (W)	na
Ester 8	R-134a	(B)	25.8 (ns)	na
Ester 9	R-134a	(W)	28.9 (ns)	na
Ester 7	Air	na	25.9 (W)	0.79 (ns)
Ester 8	Air	na	13.4 (B)	1.06 (W)
Ester 9	Air	na	20.1 (I)	0.78 (ns)
ns : Statistically Not Significant; na: Not Available; B: Best; I: Intermediate; W: Worst				

For the lubricant/refrigerant mixtures tested, the confidence intervals of the relative differences between lubricants, as tabulated in **Table 3.40**, are 97% between Ester 8 and Ester 7, 89% between Ester 7 and Ester 9 and 73% between Ester 8 and Ester 9. Similarly, the lubricant/air tests produced confidence intervals (**Table 3.42**) of greater than 99% between Ester 8 and Ester 7, 97% between Ester 7 and Ester 9 and 97% between Ester 8 and Ester 9. A minimum of three tests were run for the lubricant/refrigerant and lubricant/air conditions.

Table 3.39 - Statistical Wear Data for Case 6
Comparison of HPT Wear Data Between Lubricants in the Presence of R-134a

Ester 7			Ester 8			Ester 9		
X_1	σ_1	N_1	X_2	σ_2	N_2	X_3	σ_3	N_3
37.0	4.2123	3	25.8	1.7098	3	28.9	3.5557	3
X_i = Mean; σ_i = Standard Deviation; N_i = Number of Tests								

Table 3.40 - Confidence Intervals for Case 6 Using Small Sample Theory
Comparison of HPT Wear Data Between Lubricants in the Presence of R-134a

X_1-X_2	11.2	σ_{12}	3.937	t_{12}	3.484	n_{12}	4	$\%_{12}$	97
X_2-X_3	3.1	σ_{23}	3.417	t_{23}	1.111	n_{23}	4	$\%_{23}$	73
X_1-X_3	8.1	σ_{13}	4.774	t_{13}	2.078	n_{13}	4	$\%_{13}$	89
$\sigma_{ij} = \sqrt{\dfrac{N_i\sigma_i^2 + N_j\sigma_j^2}{N_i + N_j - 2}}\,;\, t = \dfrac{X_i - X_j}{\sigma_{ij}\sqrt{1/N_i + 1/N_j}}$					$n = N_i + N_j - 2$; % = Percentage Value for the t Distribution (Confidence Level)				

Table 3.41- Statistical Wear Data for Case 6
Comparison of HPT Wear Data Between Lubricants in Air

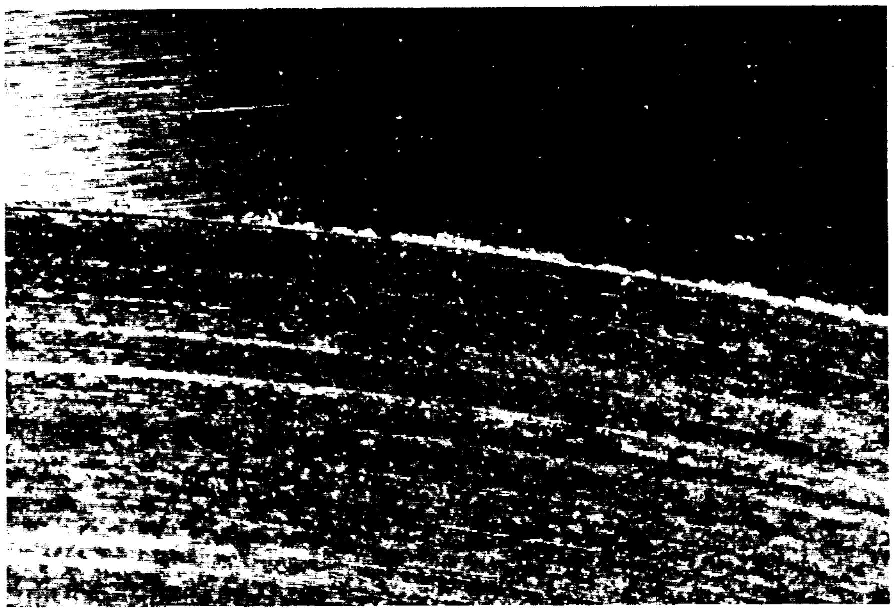
Ester 7			Ester 8			Ester 9		
X_1	σ_1	N_1	X_2	σ_2	N_2	X_3	σ_3	N_3
25.9	2.3245	3	13.4	2.5384	3	20.1	0.9074	3
X_i = Mean; σ_i = Standard Deviation; N_i = Number of Tests								

Table 3.42 - Confidence Intervals for Case 6 Using Small Sample Theory
Comparison of HPT Wear Data Between Lubricants in Air

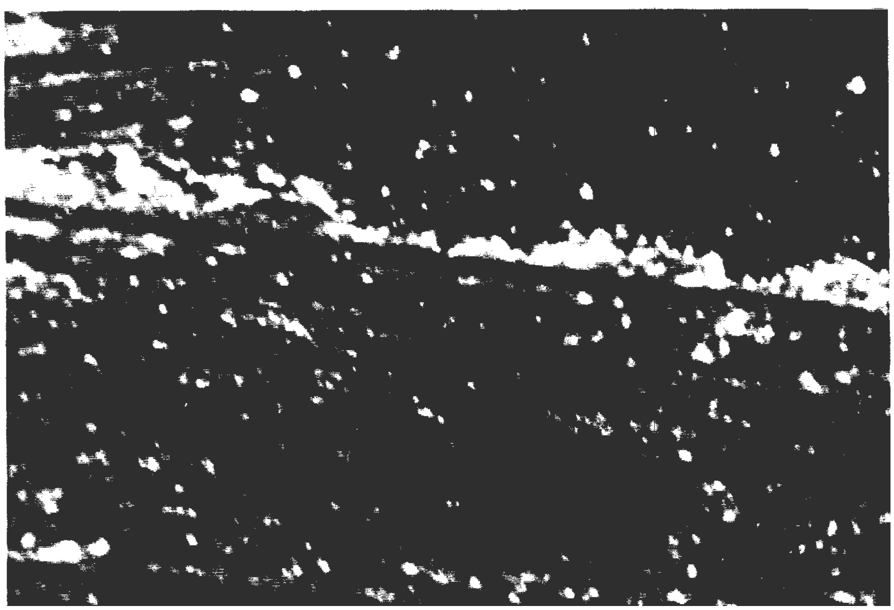
X_1-X_2	12.5	σ_{12}	2.981	t_{12}	5.136	n_{12}	4	$\%_{12}$	>99
X_2-X_3	6.7	σ_{23}	2.335	t_{23}	3.514	n_{23}	4	$\%_{23}$	97
X_1-X_3	5.8	σ_{13}	2.161	t_{13}	3.287	n_{13}	4	$\%_{13}$	97
$\sigma_{ij} = \sqrt{\frac{N_i\sigma_i^2 + N_j\sigma_j^2}{N_i + N_j - 2}}; t = \frac{X_i - X_j}{\sigma_{ij}\sqrt{1/N_i + 1/N_j}}$					$n = N_i + N_j - 2$; % = Percentage Value for the t Distribution (Confidence Level)				

3.2.6.3 Case 7: HPT Friction and Wear Results - Ductile Cast Iron Pin on Ductile Cast Iron Disk (Reciprocating Compressor)

Figure 3.30 shows a typical wear scar on a ductile cast iron plate caused by a large radius mating pin of the same material. Due to the relative hardness, the wear was measured only on the plate. The wear on the pin was small and, therefore, considered inconclusive in the analysis and subsequent ranking. The wear on the plates, however, were very consistent and provided a very precise set of data for both lubricant/refrigerant and lubricant/air tests.



(a)



(b)

Figure 3.30 - Typical Wear Scar on a Ductile Cast Iron Plate
(a) Test 17CAR/SN 9B (10 X Mag.)
(b) Test 17CAR/SN 9B (63 X Mag.)

Figure 3.31 shows a typical surface profile of a wear scar on a cast iron plate. The wear scar depth is the basis by which the lubricant/refrigerant mixtures and lubricant acting alone in an air environment are ranked.

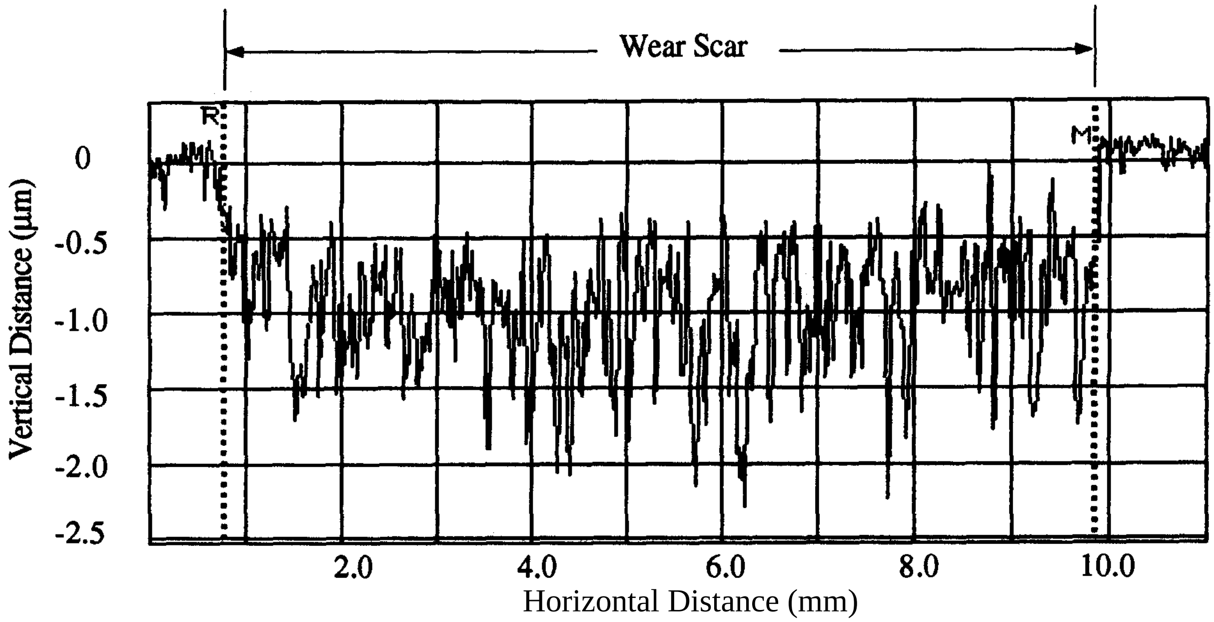


Figure 3.31 - Surface Profile of Ductile Cast Iron Specimen No. 9B

Figure 3.32a is a comparative bar chart, which graphically represents the wear damage on the ductile cast iron plate specimens for each lubricant/refrigerant mixture and lubricant acting alone in an air environment. As in Cases 5 and 6, a refrigerant atmosphere produces more wear than an air atmosphere. Unlike Case 5 and 6, the amount of refrigerant in the lubricant is significant for this case (**Table 3.30**), thus decreasing the effective viscosity of the mixture. This lower viscosity can increase the degree of asperity interaction between the mating surfaces in a mixed lubrication regime or possibly cause a change from hydrodynamic to mixed lubrication. However, when boundary lubrication conditions exist, viscosity generally does not play a major role. Therefore, the viscosity effects might not be dominant for the conditions studied and more complex effects are involved.

Similarly, the coefficient of friction for each case (**Figure 3.32b**) behaves in the same manner. For both the friction and wear data, the average speed and loading conditions are specified. The average scatter of wear (**Table E.3,** Appendix E) between repeated tests is 9.0%, while that for the friction coefficients is within 9.2%.

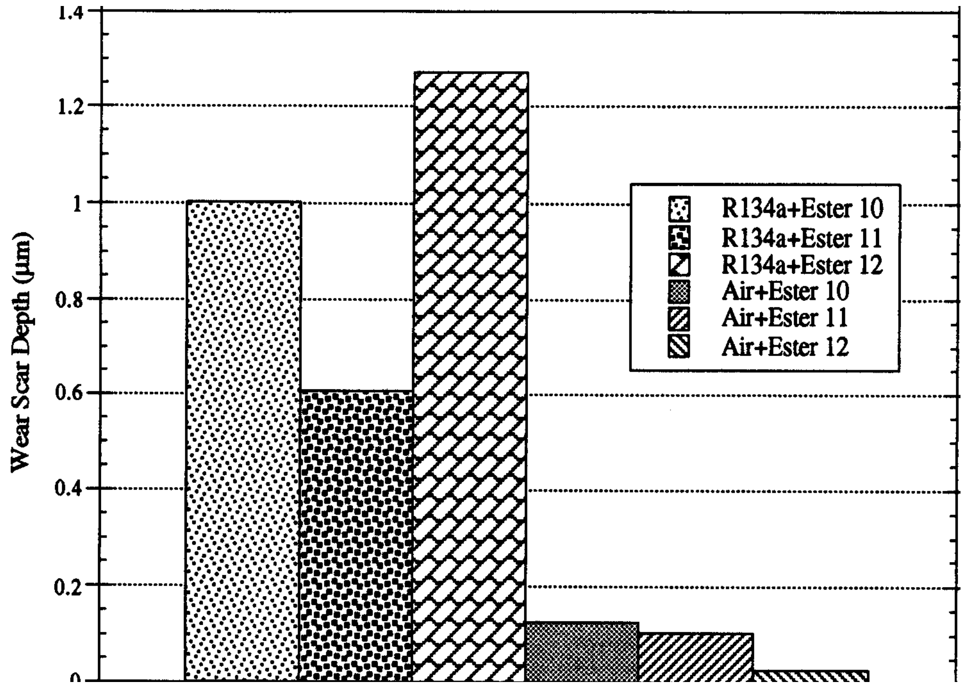


Figure 3.32a - Wear Results (± 20.6 in./s at 350 lbf)

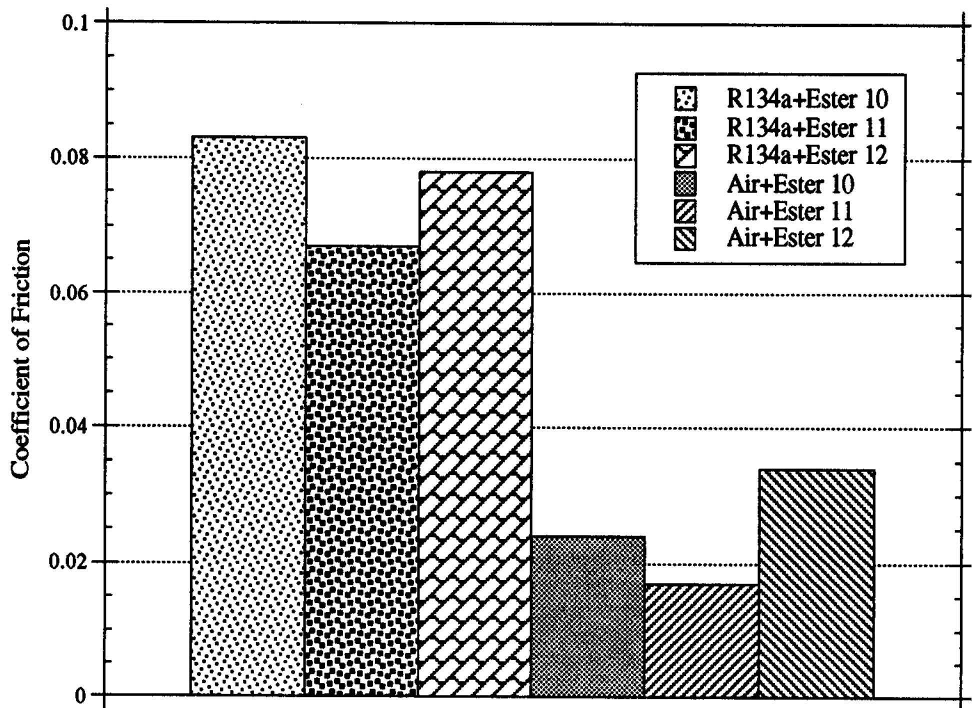
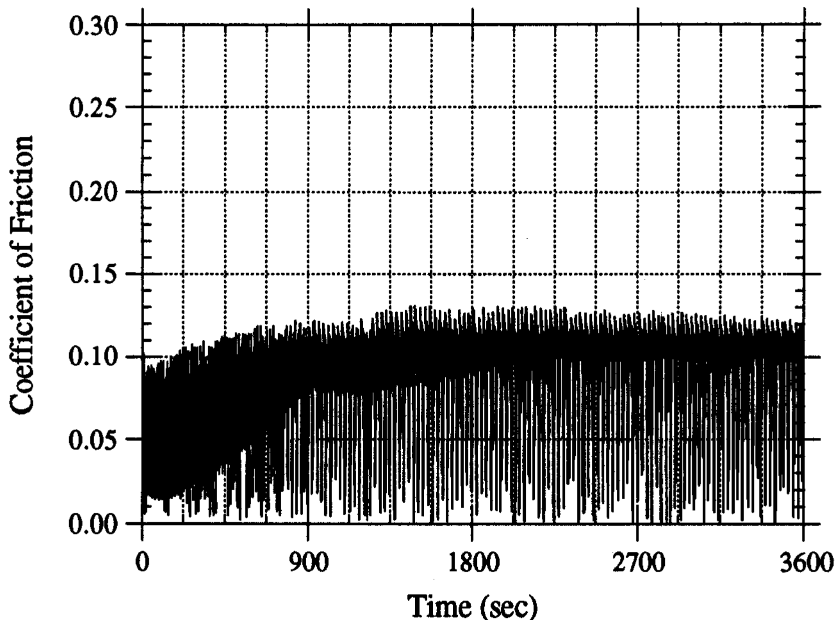


Figure 3.32b - Coefficient of Friction (± 20.6 in./s at 350 lbf)
Duration of Each Test = 1 Hour

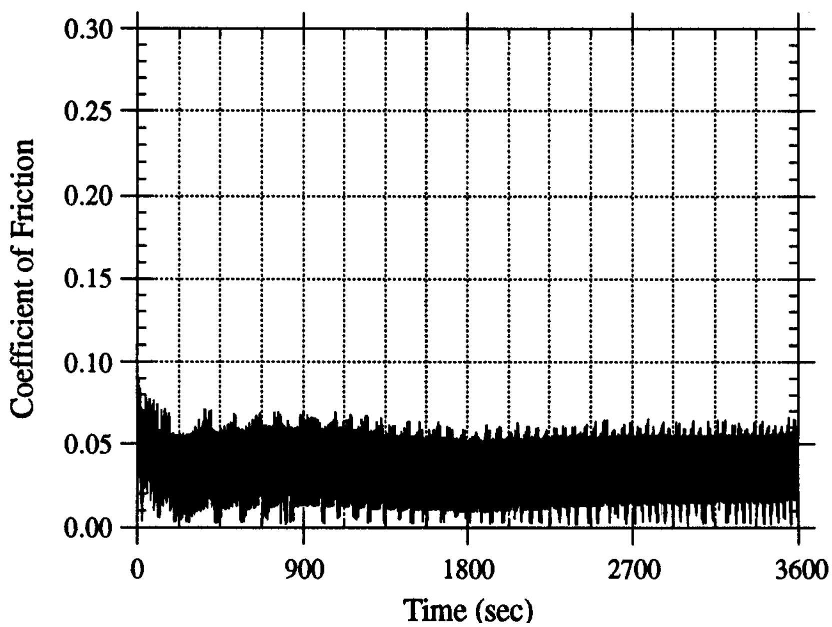
Figure 3.33 is a graphic representation of the friction coefficient plots for both lubricant/refrigerant mixture oscillation test and lubricant/air oscillation test. These figures illustrate the effects of the refrigerant on the apparent coefficient of friction. In general, there is no intuitive relationship between friction and wear. Each property must be evaluated separately. For example, Ester 12 in air shows the best wear resistance; however, its friction coefficient is among the worst when compared to its lubricant/air counterparts. **Figure 3.35** depicts a typical axial force record for a test. Even though the percent of fluctuation is relatively large for this test, the replicated data are within 10%.

A comparison between the wear data obtained from the HPT, Four Ball Tester and the component is summarized in **Table 3.43**. For the lubricant/refrigerant mixtures, both the HPT and component data rank Ester 11 as the best. There exists a discrepancy between the ranking provided by the company and those obtained by the HPT for Ester 10 and Ester 12. It should be noted that the relative difference between Ester 10 and Ester 12 is statistically significant.

For the lubricant/air tests, the HPT and the Four Ball Tester both rank Ester 10 to be the worst, however, the relative ranking between Ester 11 and Ester 12 is inconclusive. It is interesting to note that Ester 12 jumped from best to worst when going from air to a refrigerant atmosphere. One explanation for this phenomenon may lie in the relative solubilities of the three lubricants. This shows the importance of using controlled atmospheres when running friction and wear tests. **Table 3.30** reveals that the solubility of the R-134a in Ester 12 is the greatest (10.4% refrigerant in lubricant) as opposed to Ester 10 and Ester 11, 4.7% and 4.1%, respectively. The greater solubility plus the higher viscosity of Ester 12 (**Table 3.30**) may explain the differences between the lubricant/refrigerant and lubricant/air tests. However, it should again be emphasized that the lubricant viscosity usually does not play a major role in boundary lubrication. Other factor such as interfacial material/environment chemical interactions can play a major role in the lubrication process. Which role is dominant in this case is not known.



(a)



(b)

Figure 3.33 - Coefficient of Friction vs. Time
(a) Test 17CAR: Lubricant/Refrigerant Environment (± 20.6 in./s at 350 lbf)
(b) Test 9CAR: Lubricant/Air Environment (± 20.6 in./s at 350 lbf)

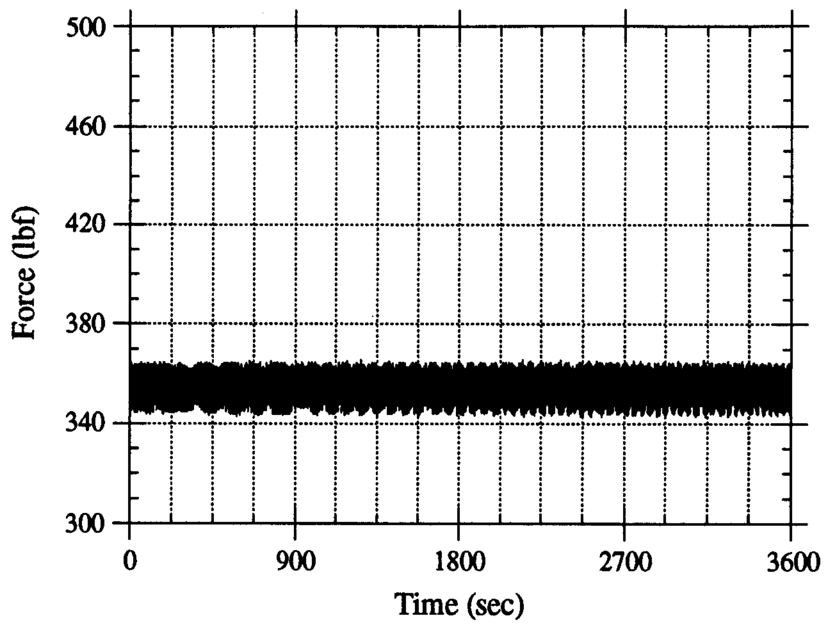


Figure 3.34 - Axial Force Record: Test 17CAR

Table 3.43 - Ranking of Lubricants by Wear (Case 7)

Lubricant	Environment	Component Qualitative Ranking	HPT ± 20.6 in./s at 350 lbf Wear Scar Depth (µm)	Four Ball Tester Average Wear Scar Diameter (mm)
Ester 10	R-134a	(W)	1.003 (I)	na
Ester 11	R-134a	(B)	0.607 (B)	na
Ester 12	R-134a	(I)	1.251 (W)	na
Ester 10	Air	na	0.126 (ns)	1.09 (W)
Ester 11	Air	na	0.106 (ns)	1.00 (ns)
Ester 12	Air	na	0.027 (B)	1.02 (ns)
ns : Statistically Not Significant; na: Not Available; B: Best; I: Intermediate; W: Worst				

Based on a statistical analysis, the confidence interval of the relative differences between lubricants is in excess of 99% for the lubricant/refrigerant data given in **Figure 3.32a** (**Table 3.45**). A minimum of four tests were run for each lubricant/refrigerant mixture and two for each lubricant/air tests. The lubricant/air tests (**Table 3.47**) show a greater than 99% confidence interval for the relative ranking between Ester 11 and Ester 12, 98% between Ester 10 and Ester 12 and only 76% for the relative ranking between Ester 10 and Ester 11. Increasing the number of tests would more than likely increase the statistical significance, but caution must be exercised in situations where two lubricants behave similarly in terms of wear. In such cases, the relative differences may be too small to accurately determine the better lubricant.

Table 3.44 - Statistical Wear Data for Case 7
Comparison of HPT Wear Data Between Lubricants in the Presence of R-134a

Ester 10			Ester 11			Ester 12		
X_1	σ_1	N_1	X_2	σ_2	N_2	X_3	σ_3	N_3
1.003	0.12025	4	0.607	0.0954	4	1.251	0.1779	5
X_i = Mean; σ_i = Standard Deviation; N_i = Number of Tests								

Table 3.45 - Confidence Intervals for Case 7 Using Small Sample Theory
Comparison of HPT Wear Data Between Lubricants in the Presence of R-134a

X_1-X_2	0.125	σ_{12}	0.125	t_{12}	4.480	n_{12}	6	$\%_{12}$	>99
X_2-X_3	0.644	σ_{23}	0.167	t_{23}	5.749	n_{23}	7	$\%_{23}$	>99
X_1-X_3	0.248	σ_{13}	0.176	t_{13}	2.101	n_{13}	7	$\%_{13}$	>99

$$\sigma_{ij} = \sqrt{\frac{N_i\sigma_i^2 + N_j\sigma_j^2}{N_i + N_j - 2}}; t = \frac{X_i - X_j}{\sigma_{ij}\sqrt{1/N_i + 1/N_j}}$$

$n = N_i + N_j - 2$
% = Percentage Value for the
t Distribution (Confidence Level)

Table 3.46 - Statistical Wear Data for Case 7
Comparison of HPT Wear Data Between Lubricants in Air

Ester 10			Ester 11			Ester 12		
X_1	σ_1	N_1	X_2	σ_2	N_2	X_3	σ_3	N_3
0.126	0.0137	2	0.106	0.00042	2	0.027	0.00382	2
X_i = Mean; σ_i = Standard Deviation; N_i = Number of Tests								

Table 3.47 - Confidence Intervals for Case 7 Using Small Sample Theory
Comparison of HPT Wear Data Between Lubricants in Air

X_1-X_2	0.02	σ_{12}	0.014	t_{12}	1.429	n_{12}	2	$\%_{12}$	76
X_2-X_3	0.079	σ_{23}	0.004	t_{23}	19.75	n_{23}	2	$\%_{23}$	>99
X_1-X_3	0.099	σ_{13}	0.014	t_{13}	7.071	n_{13}	2	$\%_{13}$	98

$$\sigma_{ij} = \sqrt{\frac{N_i\sigma_i^2 + N_j\sigma_j^2}{N_i + N_j - 2}}; t = \frac{X_i - X_j}{\sigma_{ij}\sqrt{1/N_i + 1/N_j}}$$

$n = N_i + N_j - 2$
% = Percentage Value for the
t Distribution (Confidence Level)

3.2.6.4 Case 8: Friction and Wear Results - Carburized 1018 Steel Pin on 380 Die Cast Aluminum Pad (Reciprocating Compressor)

Figure 3.35 shows a typical wear scar on a 380 die cast aluminum pad caused by a large radius mating pin made from carburized 1018 steel. Again, the wear scar was taken from the aluminum pad due to the negligible wear on the pin.

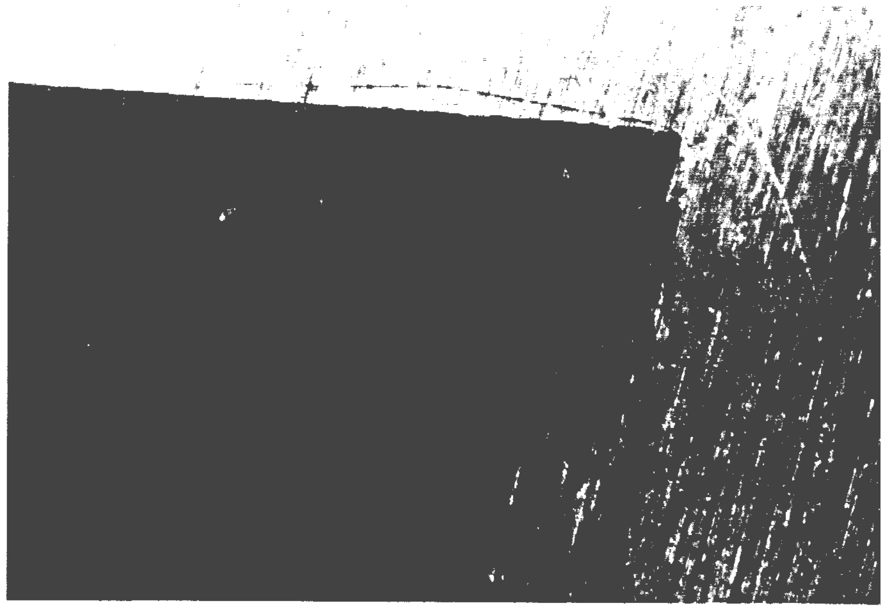
Figure 3.36 shows a typical surface profile of a wear scar on the aluminum pad. This scar is the result of a test run in a lubricant with an air environment.

Figure 3.37a shows once again that wear increased in the presence of R-134a. Two of the lubricant/refrigerant mixtures produced relatively large amounts of wear. Only the wear on the pads lubricated with R134a/Ester 15 was comparable to the lubricant/air tests. Again, even though boundary lubrication may be influenced by the apparent drop in the effective viscosity, due to the absorption of refrigerant by the lubricant, it is more likely that other mechanisms play a major role in the friction and wear characteristics of these mixtures. For both the friction and wear data, the speed and loading conditions are specified. The raw data, given in Appendix E (**Table E.4**), has an average scatter of wear between repeated tests of 6.1%. The average scatter of the friction coefficients for all of the tests is within 5.1%.

Both the HPT and the component rankings (**Table 3.48**) shows Ester 15 as the best, for wear, followed by Ester 14 and then Ester 13.

For the lubricant/air tests, the HPT and the Four Ball Tester are also in perfect agreement, ranking Ester 15, Ester 14 and Ester 13 as best, intermediate and worst, respectively.

A graphic representation of the friction coefficient plots and a typical axial force record for a test are shown in **Figure 3.38** and **Figure 3.39**, respectively.



(a)

(b)

Figure 3.35 - Typical Wear Scar an a 380 Die Cast Aluminum Pad
(a) Test 10WIT/SN 10 (10 X Mag.)
(b) Test 10WIT/SN 10 (40 X Mag.)

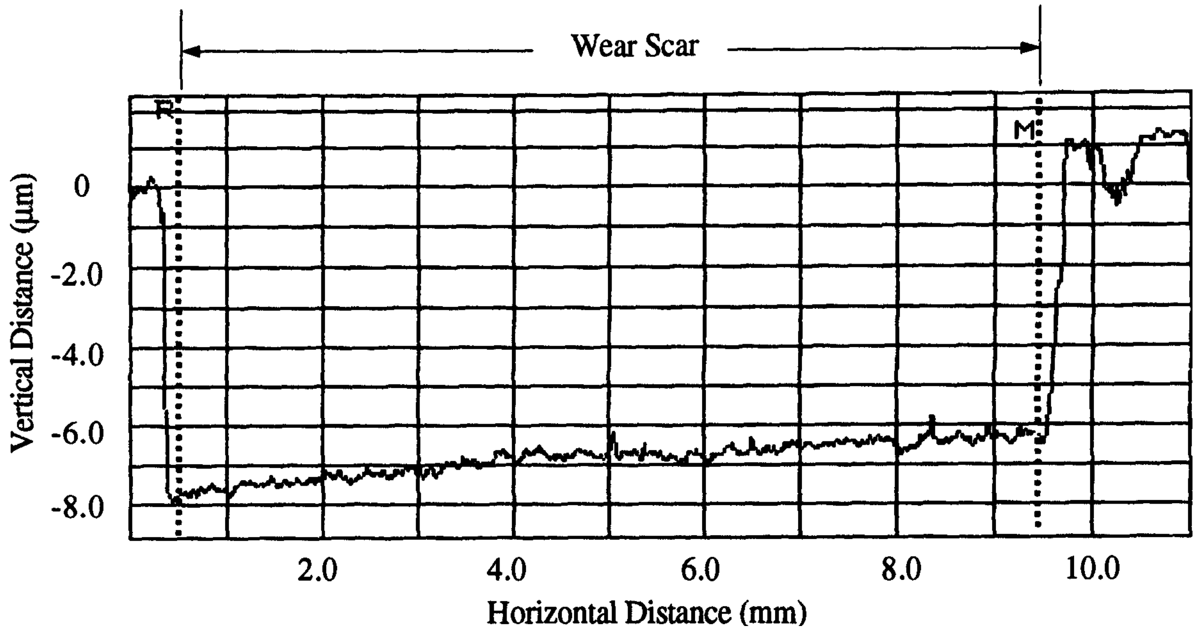


Figure 3.36 - Surface Profile of 380 Die Cast Aluminum Pad Specimen No. 10

Table 3.48 - Ranking of Lubricants by Wear (Case 8)

Lubricant	Environment	Component Qualitative Ranking	HPT ± 3.8 in./s at 25 lbf Wear Scar Depth (µm)	Four Ball Tester Average Wear Scar Diameter (mm)
Ester 13	R-134a	(W)	66.2 (W)	na
Ester 14	R-134a	(I)	35.3 (I)	na
Ester 15	R-134a	(B)	6.4 (B)	na
Ester 13	Air	na	6.5 (W)	0.96 (W)
Ester 14	Air	na	4.8 (ns)	0.88 (I)
Ester 15	Air	na	4.4 (ns)	0.85 (B)
ns : Statistically Not Significant; na: Not Available; B: Best; I: Intermediate; W: Worst				

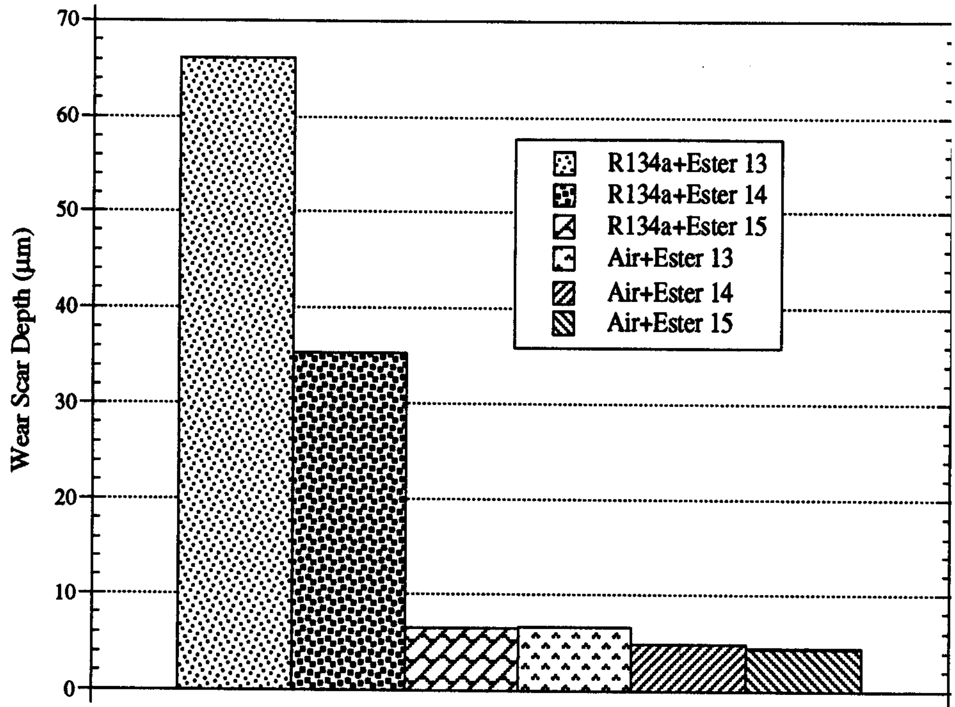


Figure 3.37a - Wear Results (±3.8 in./s at 25 lbf)

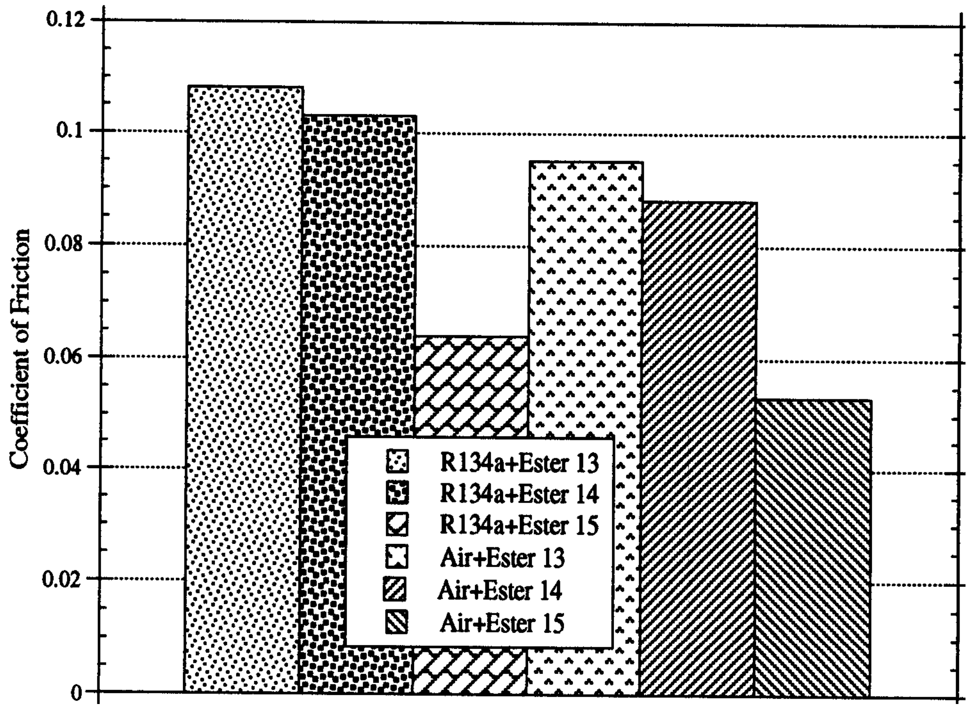
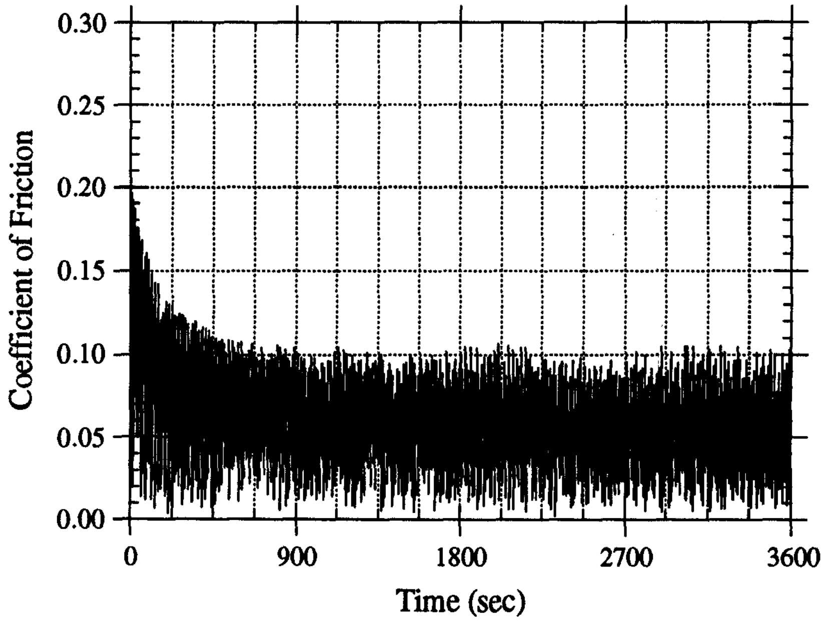
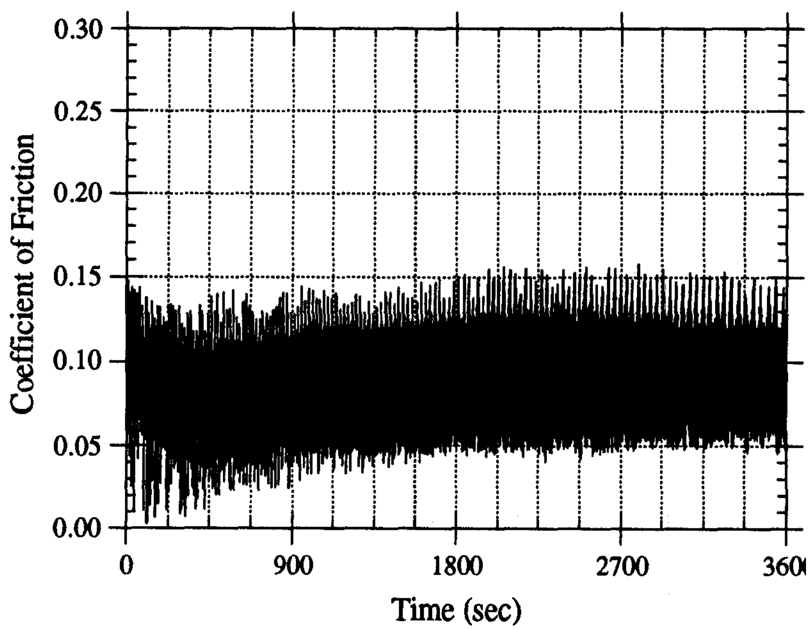


Figure 3.37b - Coefficient of Friction (± 3.8 in./s at 25 lbf)
Duration of Each Test =1 Hour



(a)



(b)

Figure 3.38 - Coefficient of Friction vs. Time
(a) Test 10WIT: Lubricant/Refrigerant Environment (± 3.8 in./s at 25 lbf)
(b) Test 15WIT: Lubricant/Air Environment (± 3.8 in./s at 25 lbf)

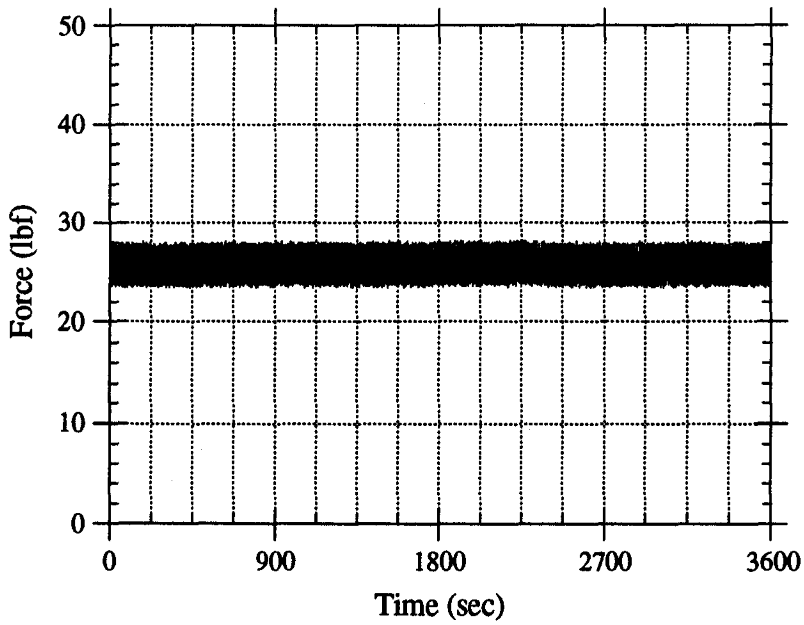


Figure 3.39 - Axial Force Record: Test 10WIT

The statistical analysis shows that the confidence intervals of the relative differences between lubricants, as tabulated in **Table 3.50**, are 98% between Ester 13 and Ester 14, greater than 99% between Ester 14 and Ester 15 and between Ester 13 and Ester 15 for the lubricant/refrigerant tests. Similarly, the lubricant/air tests produced confidence intervals (**Table 3.52**) of 79% between Ester 13 and Ester 14, 69% between Ester 14 and Ester 15 and 82% between Ester 13 and Ester 15. A minimum of two tests were run for the lubricant/refrigerant and lubricant/air tests.

Table 3.49 - Statistical Wear Data for Case 8
Comparison of HPT Wear Data Between Lubricants in the Presence of R-134a

Ester 13			Ester 14			Ester 15		
X_1	σ_1	N_1	X_2	σ_2	N_2	X_3	σ_3	N_3
66.2	4.4548	2	35.3	1.1314	2	6.4	1.2728	2
X_i = Mean; σ_i = Standard Deviation; N_i = Number of Tests								

Table 3.50 - Confidence Intervals for Case 8 Using Small Sample Theory
Comparison of HPT Wear Data Between Lubricants in the Presence of R-134a

X_1-X_2	30.9	σ_{12}	4.596	t_{12}	6.723	n_{12}	2	$\%_{12}$	98
X_2-X_3	28.9	σ_{23}	1.703	t_{23}	16.97	n_{23}	2	$\%_{23}$	>99
X_1-X_3	59.8	σ_{13}	4.633	t_{13}	12.91	n_{13}	2	$\%_{13}$	>99
$\sigma_{ij} = \sqrt{\dfrac{N_i\sigma_i^2 + N_j\sigma_j^2}{N_i + N_j - 2}}; t = \dfrac{X_i - X_j}{\sigma_{ij}\sqrt{1/N_i + 1/N_j}}$					$n = N_i + N_j - 2$; % = Percentage Value for the t Distribution (Confidence Level)				

Table 3.51- Statistical Wear Data for Case 8
Comparison of HPT Wear Data Between Lubricants in Air

Ester 13			Ester 14			Ester 15		
X_1	σ_1	N_1	X_2	σ_2	N_2	X_3	σ_3	N_3
6.5	0.9192	2	4.8	0.2828	2	4.4	0.4243	2
X_i = Mean; σ_i = Standard Deviation; N_i = Number of Tests								

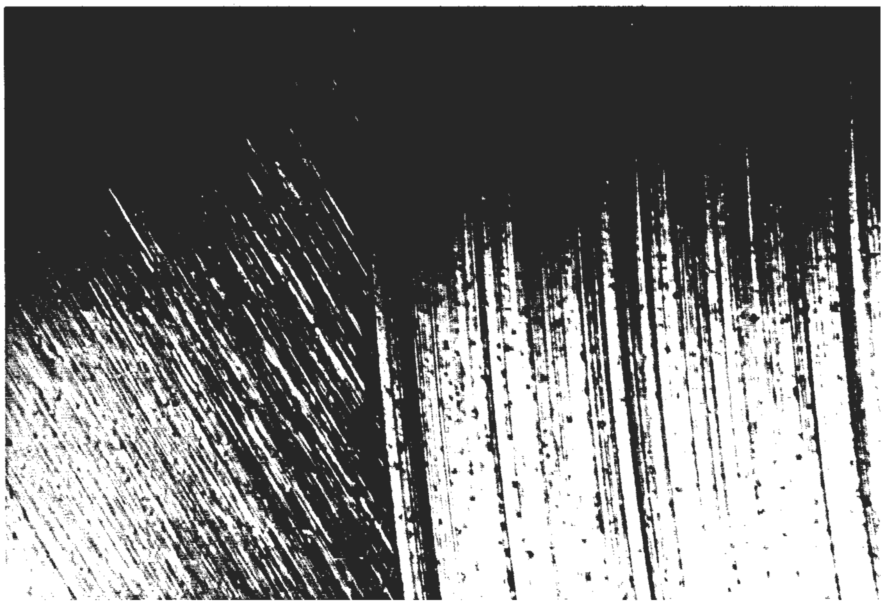
Table 3.52 - Confidence Intervals for Case 8 Using Small Sample Theory
Comparison of HPT Wear Data Between Lubricants in Air

X_1-X_2	1.7	σ_{12}	0.962	t_{12}	1.767	n_{12}	2	$\%_{12}$	79
X_2-X_3	0.4	σ_{23}	0.510	t_{23}	0.784	n_{23}	2	$\%_{23}$	69
X_1-X_3	2.1	σ_{13}	1.012	t_{13}	2.075	n_{13}	2	$\%_{13}$	82
$\sigma_{ij} = \sqrt{\dfrac{N_i\sigma_i^2 + N_j\sigma_j^2}{N_i + N_j - 2}}; t = \dfrac{X_i - X_j}{\sigma_{ij}\sqrt{1/N_i + 1/N_j}}$					$n = N_i + N_j - 2$; % = Percentage Value for the t Distribution (Confidence Level)				

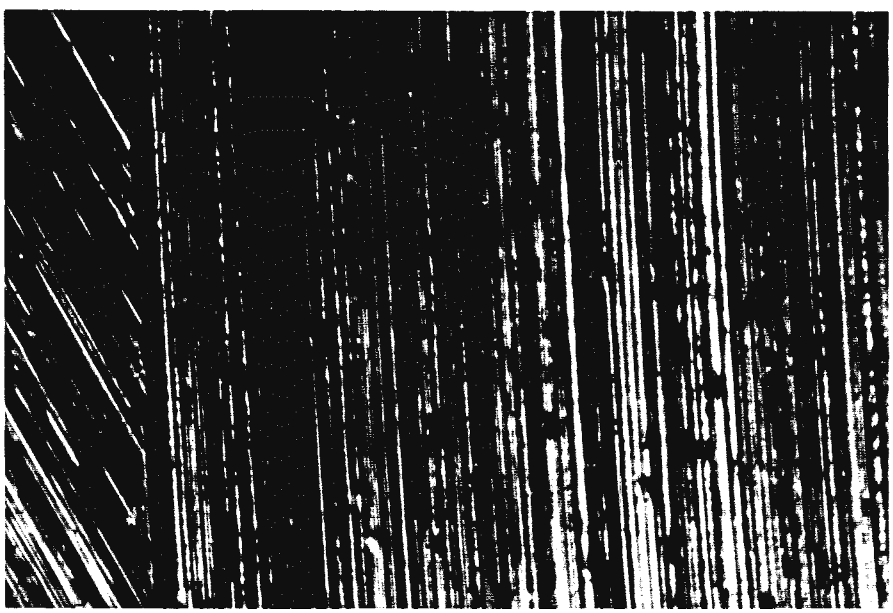
3.2.6.5 Case 9: HPT Friction and Wear Results - Sintered Ferrous Material Pin on Sintered Ferrous Material Disk (Rotary Compressor)

Figure 3.40 shows a typical wear scar on a sintered ferrous material (SFM) plate caused by a small radius mating pin of a sintered material with a slightly different composition (**Table 3.23**). The wear scars were measured on the plates for all of the tests rather than on the pins. The wear on the pin was slight and, therefore, considered inconclusive in the analysis and subsequent ranking. The wear on the plates, however, proved to be very consistent and provided a very precise set of data per lubricant/refrigerant mixture.

Figure 3.41 shows a typical surface profile of a wear scar on a SFM plate. This scar is very pronounced and provides the necessary wear to rank the lubricants.



(a)



(b)

Figure 3.40 - Typical Wear Scar on an SFM Plate
(a) Test 1TE-3/SN 1A (10 X Mag.)
(b) Test 1TE-3/SN 1A (40 X Mag.)

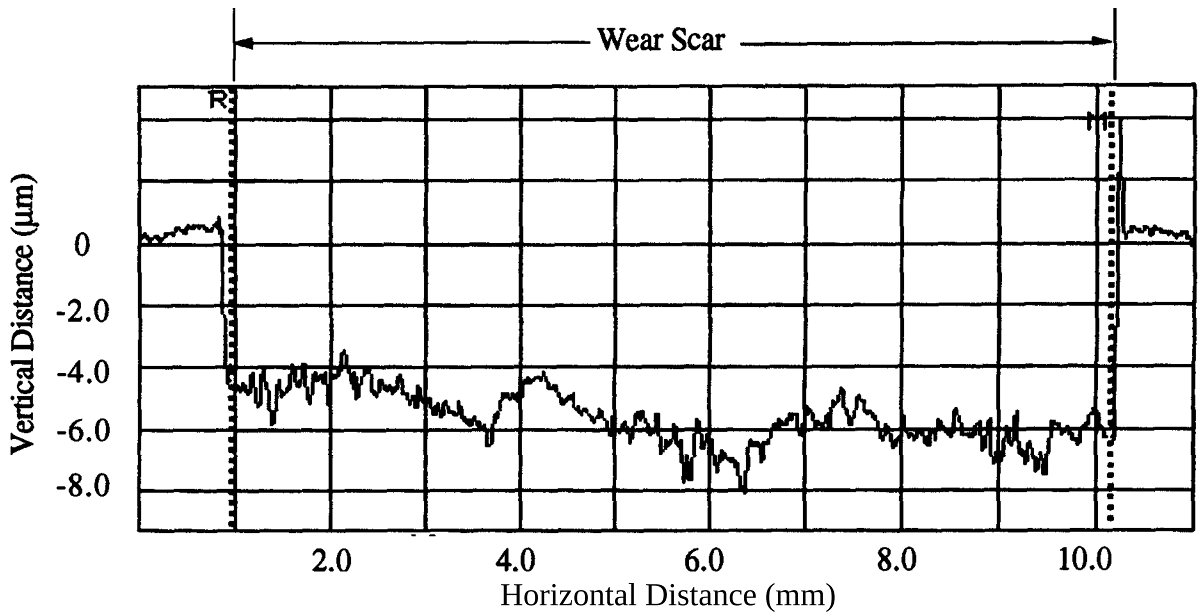


Figure 3.41 - Surface Profile of a SFM Plate Specimen No. 1A

Due to limited numbers of SFM specimens, only lubricant/refrigerant tests and Four Ball tests were conducted. **Figure 3.42** depicts the results for both friction and wear for the three lubricant/refrigerant mixtures. The Alkbenz 1 is far worse than the Alkbenz 2 or the Mineral 3, the latter two being close in terms of wear, with the Alkbenz 2 slightly better than the Mineral 3 (confidence interval of 60% - statistically insignificant, **Table 3.55**). There are no significant differences in terms of friction between the three lubricant/refrigerant combinations.

Table 3.53 summarizes the results and again shows very good agreement between the HPT and the component rankings. The component data indicate that Mineral 3 and Alkbenz 2 lubricant behave virtually the same in terms of wear, with the Alkbenz 1 much worse. The HPT data show that Alkbenz 2 and Mineral 3 are almost identical in terms of wear with Alkbenz 1 far worse.

The Four Ball tests rank the lubricants Mineral 3, Alkbenz 2, and Alkbenz 1 as best, intermediate and worst, respectively.

A graphic representation of the friction coefficient plots and a typical axial force record for a test are shown in **Figure 3.43** and **Figure 3.44**, respectively.

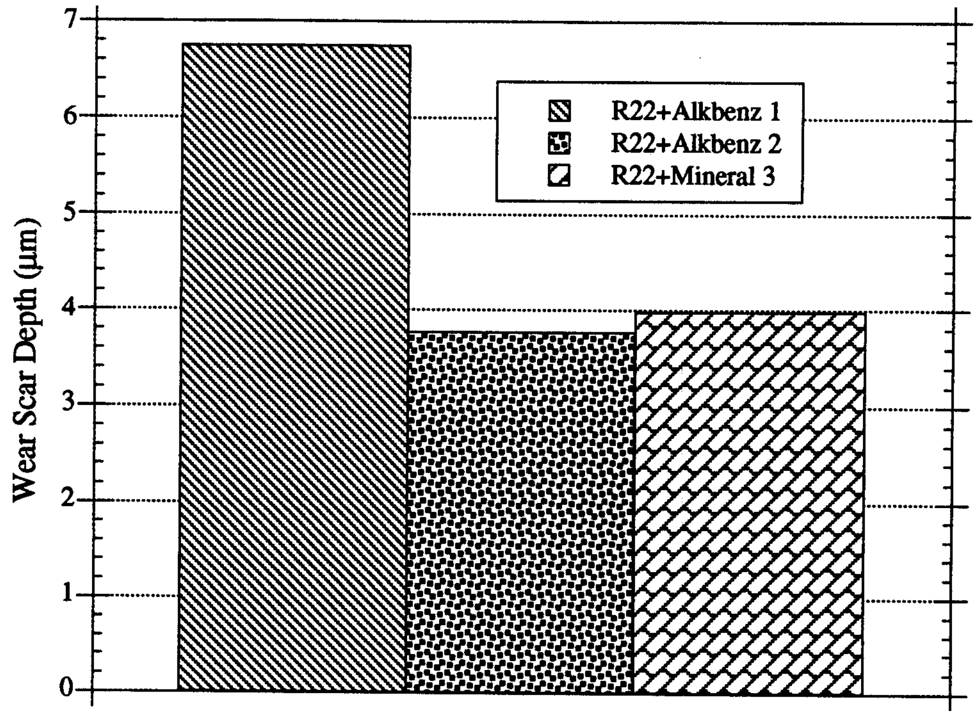


Figure 3.42a - Wear Results (± 20.6 in./s at 250 lbf)

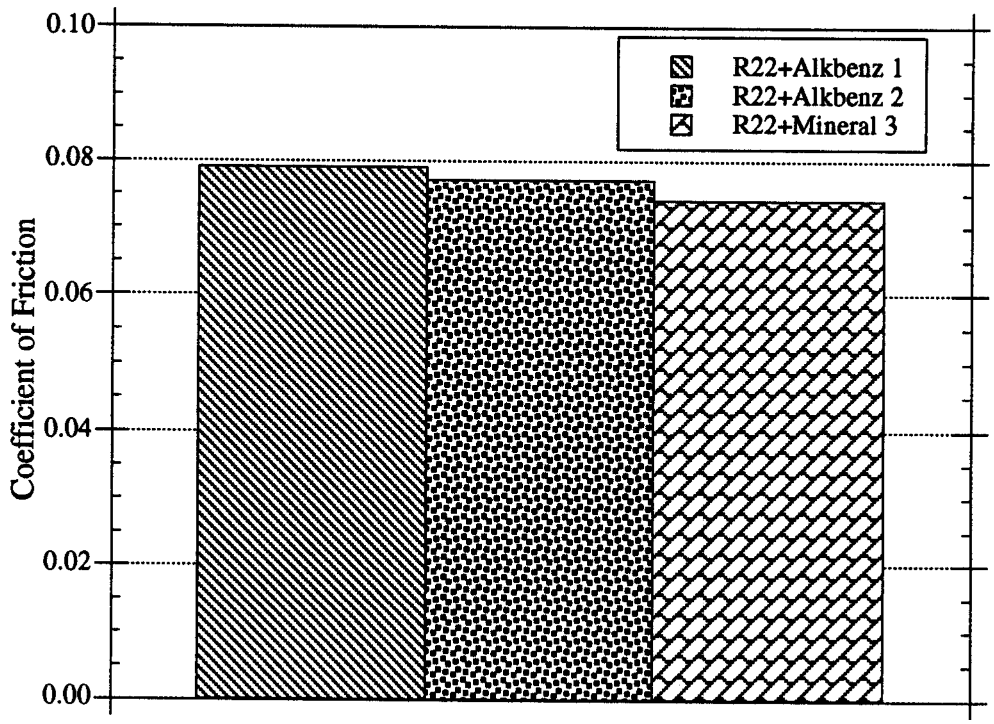


Figure 3.42b - Coefficient of Friction (± 20.6 in./s at 250 lbf)
Duration of Each Test = 1 Hour

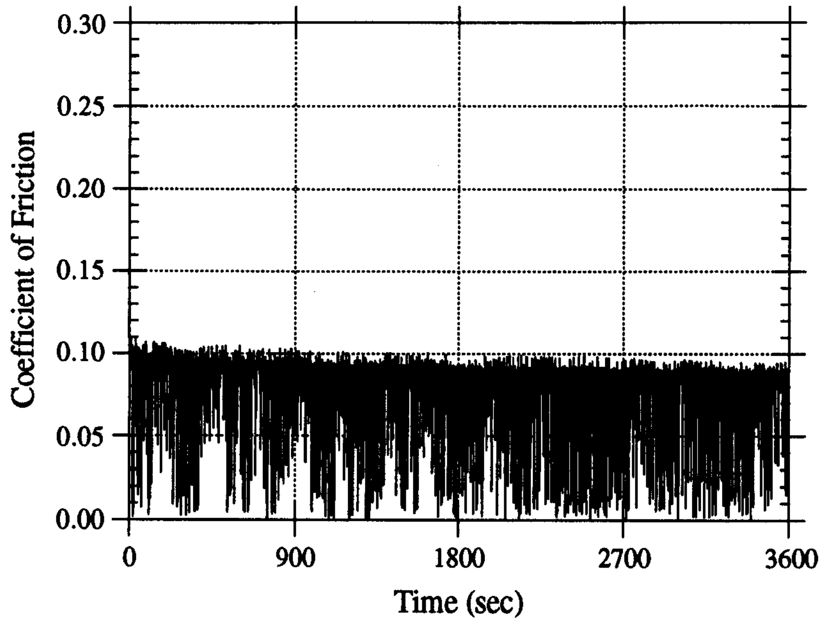


Figure 3.43 - Coefficient of Friction vs. Time
Test 1TE-3: Lubricant/Refrigerant Environment (± 20.6 in./s at 250 lbf)

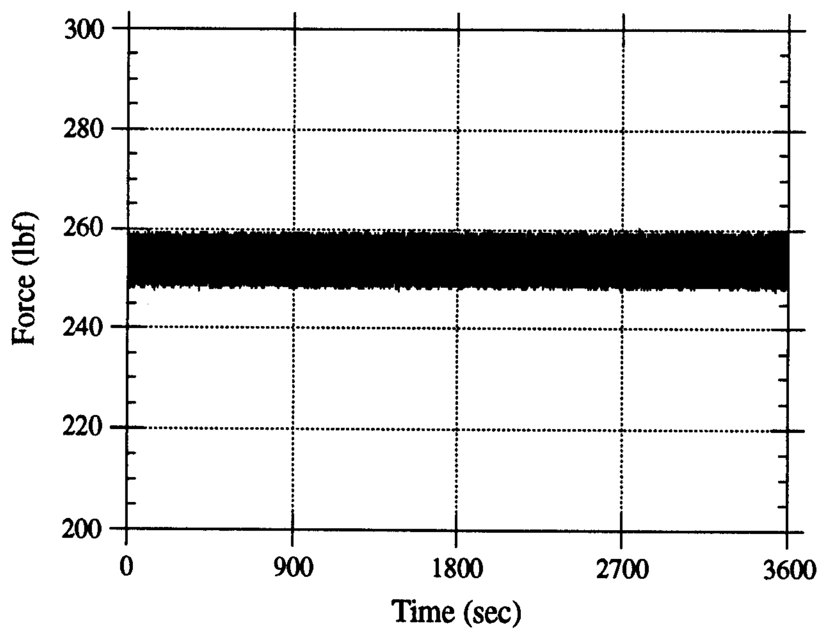


Figure 3.44 - Axial Force Record: Test 1TE-3

Table 3.53 - Ranking of Lubricants by Wear (Case 9)

Lubricant	Environment	Component Qualitative Ranking	HPT ± 20.6 in./s at 250 lbf Wear Scar Depth (μm)	Four Ball Tester Average Wear Scar Diameter (mm)
Alkbenz 1	R-22	(W)	6.74 (W)	na
Alkbenz 2	R-22	(B)(tied)	3.76 (ns)	na
Mineral 3	R-22	(B)(tied)	3.98 (ns)	na
Alkbenz 1	Air	na	na	0.69 (ns)
Alkbenz 2	Air	na	na	0.64 (ns)
Mineral 3	Air	na	na	0.54 (B)
ns : Statistically Not Significant; na: Not Available; B: Best; I: Intermediate; W: Worst				

Based on a statistical analysis, the confidence intervals of the relative differences between lubricants, as tabulated in **Table 3.55**, are 92% between Alkbenz 1 and Alkbenz 2, 60% between Alkbenz 2 and Mineral 3 and 89% between Alkbenz 1 and Mineral 3 for the lubricant/refrigerant tests. As discussed in previous sections, confidence intervals in excess of 80%, 90% and 95% are considered acceptable, very good and excellent, respectively. Anything less than 80% is generally considered poor confidence and is regarded as statistically insignificant. Therefore, the confidence interval of 60% between Alkbenz 2 and Mineral 3 is inconclusive.

Table 3.54 - Statistical Wear Data for Case 9
Comparison of HPT Wear Data Between Lubricants in the Presence of R-134a

Alkbenz 1			Alkbenz 2			Mineral 3		
X_1	σ_1	N_1	X_2	σ_2	N_2	X_3	σ_3	N_3
6.74	2.414	4	3.76	0.2979	4	3.98	0.7902	4
X_i = Mean; σ_i = Standard Deviation; N_i = Number of Tests								

Table 3.55 - Confidence Intervals for Case 9 Using Small Sample Theory
Comparison of HPT Wear Data Between Lubricants in the Presence of R-134a

X_1-X_2	2.98	σ_{12}	1.986	t_{12}	2.122	n_{12}	6	$\%_{12}$	92
X_2-X_3	0.22	σ_{23}	0.690	t_{23}	0.451	n_{23}	6	$\%_{23}$	60
X_1-X_3	2.76	σ_{13}	2.074	t_{13}	1.882	n_{13}	6	$\%_{13}$	89

$$\sigma_{ij} = \sqrt{\frac{N_i\sigma_i^2 + N_j\sigma_j^2}{N_i + N_j - 2}}\ ; t = \frac{X_i - X_j}{\sigma_{ij}\sqrt{1/N_i + 1/N_j}}$$

$n = N_i + N_j - 2$
% = Percentage Value for the t Distribution (Confidence Level)

All the results obtained in Part II are summarized in **Table 3.56**. The statistical significance of the HPT and Four Ball data is also indicated in the table. The statistically significant data are designated as "B" for best, "I" for intermediate or "W" for worst and the statistically insignificant data are designated as "ns" which indicates "not significant" and "na" indicates "not available".

Table 3.56 - A Comparison of Lubricant Rankings Based on Wear Data

Lubricant	Viscosity, cS		HPT (air) Wear Scar Depth (μm)	HPT (R134a) Wear Scar Depth (μm)	Four Ball Wear Scar Dia. (mm)	Component (Provided by Companies)
	@40°C	@100°C				
#Ester 4	23.9	4.9	24.2 (ns)	26.1 (ns)	0.78 (ns)	(W)
#Ester 5*	23.9	4.9	16.5 (B)	20.7 (B)	0.58 (B)	(B)
Ester 6*	32.0	5.6	20.9 (ns)	26.4 (ns)	0.77 (ns)	(I)
#Ester 7*	23.9	4.8	25.9 (W)	37.0 (W)	0.79 (ns)	(I)
Ester 8	32.0	5.4	13.4 (B)	25.8 (ns)	1.06 (W)	(B)
Ester 9	32.0	5.6	20.1 (I)	28.9 (ns)	0.78 (ns)	(W)
Ester 10*	61.7	8.1	0.126 (ns)	1.003 (I)	1.09 (W)	(W)
Ester 11	67.6	8.7	0.106 (ns)	0.607 (B)	1.00 (ns)	(B)
Ester 12	72.3	9.8	0.027 (B)	1.251 (W)	1.02 (ns)	(I)
Ester 13	7.7	2.3	6.5 (W)	66.2 (W)	0.96 (W)	(W)
Ester 14	9.4	2.8	4.8 (ns)	35.3 (I)	0.88 (I)	(I)
Ester 15	9.8	4.2	4.4 (ns)	6.4 (B)	0.85 (B)	(B)
Alkbenz1*	28.0	4.1	na	6.74 (W)	0.69 (ns)	(W)
Alkbenz2*	57.0	5.8	na	3.76 (ns)	0.64 (ns)	(B) (tied)
Mineral 3*	35.6	7.0	na	3.98 (ns)	0.54 (B)	(B) (tied)

ns : Statistically Not Significant; na: Not Available;B: Best; I: Intermediate; W: Worst
Same Base Lubricant; * Formulated

3.3 CONCLUSIONS AND RECOMMENDATIONS

3.3.1 Research Summary

Various materials and contact geometries lubricated with different lubricants, in both refrigerant and air environments, were tribologically evaluated with the HPT. In Part I of the study, the HPT wear data were compared to those obtained by a Falex™ tester. With the exception of load and environmental pressure and temperature, specimen testing was conducted under approximately the same conditions in both Falex™ and HPT testers. Both the environmental pressure and temperature were not controlled during the Falex™ tests. Also, in these tests, refrigerant was bubbled through the lubricant and the contact loads were relatively high. The HPT test loads were less than those used with the Falex™ and the environments in the HPT approximately correspond to those surrounding critical contacts in scroll and reciprocating compressors.

In Part II of the study, the wear data obtained from the HPT and a Four Ball tester are compared to those obtained from accelerated component (compressor) tests. The HPT tests conducted in a refrigerant environment approximately simulated the environmental and operating conditions experienced by the components. Additional data were obtained by means of the HPT and a Four Ball tester in an air environment. These latter data were used to determine the effects of the environment on friction and wear.

The Falex™ and component data were provided by the participating companies. Most of the data supplied by these companies were qualitative (best, intermediate, worst) and, therefore, the relative wear difference is not known. Quantitative wear data are obtained by means of both the HPT and the Four Ball tester, and the lubricants are ranked based on these data. Based on the statistically significant data obtained, the following conclusions can be drawn:

Part I - HPT vs. Falex™ Tester

1. Lubricant ranking correlation between the HPT and Falex™ tester is obtained only when relatively large wear differences existed between the lubricants.
2. For a given refrigerant and based on statistical significance, lubricant ranking obtained by means of the HPT was essentially the same under various loads and speeds.
3. A lubricant/refrigerant mixture which produces relative low wear will not necessarily produce relative low friction.
4. The ranking of the lubricants can be a function of the materials pair in contact.
5. For the operating condition examined, R134a or blend/ester mixtures generally gave higher wear than the baseline R12 or R22/mineral oil mixtures.

Part II - HPT and Four Ball Tester vs. Component Testing

1. None of the specimen testers produced data which exactly correlated with the component data.

2. For given conditions and materials pair, the presence of R134a with any lubricant consistently increased wear on the specimens as compared to the same lubricant acting alone.

3. As in Part I, a lubricant/refrigerant mixture which produces relative low wear will not necessarily produce relative low friction.

4. The HPT data obtained also suggest that lubricant ranking is affected by environmental conditions (pressure and temperature)

3.3.2 Discussion of Results and Recommendations

A comparison of lubricant rankings based on wear data, obtained from various testers, are summarized in **Table 3.57**. As previously stated, the Falex™ and component data were supplied by participating companies. Ester 1, Ester 2, and Ester 3 are not shown in the table because component data are not available for these lubricants. The component data shown were obtained from five different sets of compressor tests. For each of these sets, three different lubricants are qualitatively ranked. The first, second and fourth sets are based on wrist pin contacts in reciprocating compressors, the third on a piston ring/cylinder contact in a reciprocating compressor and finally the fifth on a vane/piston contact in a rotary compressor.

In **Table 3.57**, in addition to the component and HPT data obtained in refrigerant environments, other data shown are: (a) HPT and Four Ball data, for the lubricant alone, obtained in air and (b) Falex™ Failure Load data for the lubricant alone (Esters 13-15) and Falex™ data obtained by bubbling refrigerant through the lubricant (Esters 4-6, 7-9, and Alkbenz 1-2, Mineral 3). Except for lubricants set 4-6, the HPT, Falex™ and component data given are all based on the same materials pair. For Esters 4-6, the component and HPT data are based on a 380 aluminum/steel contact while the Falex™ data are based on a 356 aluminum/steel contact. The company which provided the Falex™ data was unable to get consistent results with the 380 aluminum and the 356 was substituted for the 380.

In trying to analyze the data obtained, the following comments are in order:

1. Based on information obtained from one company, if two lubricants are not significantly different in their overall performance, data for the same component in different compressors nominally operating under the same conditions might show different ranking for the lubricants.

2 . Component data obtained has not been statistically analyzed. For Esters 10-12 and 13-15, only one compressor per lubricant was tested.

3. Component testing is done under accelerated conditions (i.e., high loads, temperature or refrigerant content in the lubricant) which might not be representative of nominal operating conditions. Except for Case 6 (Esters 7-9), the operating and environmental conditions used to obtain the HPT data are based on nominal component operation.

4. Most tribo-contacts in compressors experience transient environmental operating conditions, while specimen testing is conducted under steady state conditions. What effects these transient conditions have, especially on the composition of the lubricant/refrigerant mixture, are not known.

5. The conditions used to conduct HPT tests in refrigerant environments were based on data supplied by participating companies for specific components. How representative these data are to the actual component operation is not known.

6. Due to the limitations of the HPT, some of the environmental conditions found in the components could not be simulated (**Table 3.28** and **3.29**).

7. In order to get measurable wear and due to the speed limitations of the HPT, components speeds and some loads could not be simulated.

Table 3.57 - A Comparison of Lubricant Rankings Based on Wear Data Obtained from Various Testers

Lubricant	Viscosity, cS		HPT(air) Wear Scar Depth (μm)	HPT (R134a) Wear Scar Depth (μm)	Four Ball Wear Scar Dia. (mm)	Falex™ Test (Provided by Companies)	Component (Provided by Companies)
	@40°C	@100°C					
#Ester 4	23.9	4.9	24.2 (ns)	26.1 (ns)	0.78 (ns)	(W)	(W)
#Ester 5*	23.9	4.9	16.5 (B)	20.7 (B)	0.58 (B)	(B)	(B)
Ester 6*	32.0	5.6	20.9 (ns)	26.4 (ns)	0.77 (ns)	(I)	(I)
#Ester 7*	23.9	4.8	25.9 (W)	37.0 (W)	0.79 (ns)	0.0106g (W)	(I)
Ester 8	32.0	5.4	13.4 (B)	25.8 (ns)	1.06 (W)	Negligible(B)	(B)
Ester 9	32.0	5.6	20.1 (I)	28.9 (ns)	0.78 (ns)	0.0031g (I)	(W)
Ester 10*	61.7	8.1	0.126 (ns)	1.003 (I)	1.09 (W)	na	(W)
Ester 11	67.6	8.7	0.106 (ns)	0.607 (B)	1.00 (ns)	na	(B)
Ester 12	72.3	9.8	0.027 (B)	1.251 (W)	1.02 (ns)	na	(I)
Ester 13	7.7	2.3	6.5 (W)	66.2 (W)	0.96 (W)	733 lbf**(W)	(W)
Ester 14	9.4	2.8	4.8 (ns)	35.3 (I)	0.88 (I)	750 lbf**(I)	(I)
Ester 15	9.8	4.2	4.4 (ns)	6.4 (B)	0.85 (B)	975 lbf**(B)	(B)
Alkbenz1*	28.0	4.1	na	6.74 (W)	0.69 (ns)	Tied	(W)
Alkbenz2*	57.0	5.8	na	3.76 (ns)	0.64 (ns)	na	(B) (tied)
Mineral 3*	35.6	7.0	na	3.98 (ns)	0.54 (B)	Tied	(B) (tied)
Agreement relative to component data			67%	67%	67%	64%	100%

Same Base Lubricant; * Formulated; ** Falex™ Failure Load; ns : Statistically Not Significant; na: Not Available; B: Best; I: Intermediate; W: Worst

From the rankings summarized in **Table 3.57**, agreement between the data obtained from each specimen tester and the component data is approximately 65%. A direct correspondence between the HPT data obtained in a refrigerant environment and those obtained from the components does exist for the "best" lubricants. For the first three sets of data (Esters 4-6, 7-9, and 10-12), the "intermediate" and "worst" lubricants do not show the same ranking based on the HPT and component data obtained. The fourth data set (Esters 13-15) shows that, if the performance of the lubricants is significantly different, the same ranking is obtained by all testers and that this ranking corresponds to the component ranking.

For the second set of data, the discrepancy might be due to the material (380 die cast aluminum (Case 6 - **Table 3.23**) used in the HPT tests. As previously stated, these aluminum specimens tended to be porous. The porosity added an extra variable to the data obtained since it could not be controlled from specimen to specimen. When tests were conducted with the slightly different 380 die cast aluminum specimens (Cases 5 and 8 - **Table 3.23**, also Case 3 in Part 1), the ranking of these lubricants (**Table 3.15**) based on the HPT data, is in complete agreement with the ranking from the component tests. Therefore, it is quite possible that the porosity and slightly different material composition for these tests may have influenced the ranking of these lubricants.

In order to see if the different environmental conditions (pressure and temperature) produce different rankings of the lubricants, additional tests were conducted with the HPT. Due to time limitations, only two tests for each lubricants were conducted. Tests were repeated for the first and third sets of the lubricants (**Table 3.57**) under the same operating conditions but different environmental pressure and temperature. As previously stated, the amount of refrigerant in a lubricant, which is a function of pressure and temperature, can significantly affect interfacial lubricative properties. It should again be emphasized that viscosity may not play a major role under boundary lubrication conditions. Chemical interactions at the interface are likely to be more important. The friction and wear results and the amount of refrigerant in lubricants are tabulated in **Table 3.58**.

The results show that the rankings of "intermediate" and "worst" lubricants are reversed at these different environmental conditions. The rankings of the lubricants obtained under these environmental conditions correlate with those obtained from the component tests. Therefore, in order to rank lubricants properly, it is very important to know the actual environmental conditions which are experienced by the tribo-contact of the component, especially if the lubricants to be ranked are not significantly different in their lubricity characteristics.

With the exception of Ester 12, **Table 3.58** also shows that the amount of wear decreases as the amount of refrigerant in the lubricant increases. The common belief that larger amounts of refrigerant in an lubricant will produce more wear, due to lower mixture viscosity, may not be correct under boundary lubrication conditions. Obviously, its effects in mixed or fully hydrodynamic conditions is significant. This leads one to conclude that very complex physical and chemical interactions are taking place at the interface and the HPT is the only tester which has the capability to expose these interactions.

Table 3.58 - Effects of Environmental Conditions on Wear Data

Lubricant	Refrig	Load (lbf)	Pressure (psig)	Temp (°F)	Coeff. of Friction	Wear Scar (μm)	Solubility % Ref. in Oil
Ester 4	R-134a	50	30	266	0.125	26.1	0.3
Ester 5*	R-134a	50	30	266	0.117	20.7	0.3
Ester 6*	R-134a	50	30	266	0.118	26.4	0.5
#Ester 4	R-134a	50	125	70	0.115	6.64	21.0
#Ester 5*	R-134a	50	125	70	0.106	3.98	36.7
#Ester 6*	R-134a	50	125	70	0.113	4.88	48.5
Ester 10*	R-134a	350	170	160	0.082	1.003	4.7
Ester 11	R-134a	350	170	160	0.067	0.607	4.1
Ester 12	R-134a	350	170	160	0.079	1.251	10.4
#Ester 10*	R-134a	350	15	266	0.074	1.244	0.19
#Ester 11	R-134a	350	15	266	0.074	0.913	0.10
#Ester 12	R-134a	350	15	266	0.072	1.178	0.15
* Formulated, # Repeated Tests at Different Pressure/Temperature							

Strictly based on the results obtained, lubricant ranking correlation between the component data and the HPT data obtained in a refrigerant environment is not very good. Obviously, the "correctness" of the data obtained from the HPT is based on the assumption that the lubricant rankings obtained from the component tests are accurate. As previously stated, component testing is done under accelerated conditions. Under these conditions, the component experiences operating and environmental conditions which can be significantly different than those experienced under nominal conditions. If in fact these changes in operating and environmental conditions affect lubricant ranking, then lubricant ranking based on accelerated component (compressor) testing may also be questionable. The uncertainty of the ranking obtained will increase as the differences in the lubricants wear behavior decreases. A possible extension of the work presented in this report is to more thoroughly examine the effects of refrigerant pressure and temperature, and possibly load and speed, on lubricant ranking.

The goal of this research was to recommend a specific bench tester which could be used to predict lubricant performance in refrigerant compressors. The data obtained do not seem to give a clear vision about the development of a new bench tester to accomplish this goal. The HPT tests conducted in air have always given different lubricant performance and generally different lubricant rankings than those conducted in pressurized refrigerant environments. As such, the use of the

HPT is likely to be an improvement over presently used lubricant screening testers. Before the HPT can be recommended, however, simulation through specimen testing needs to be based upon more accurate operating and environmental conditions under which simulated components operate. In addition, statistically significant components wear data are required in order to make a more effective comparison to the specimen data.

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COMPLIANCE WITH AGREEMENT

The University of Illinois at Urbana-Champaign has complied with all terms of the agreement.

PRINCIPAL INVESTIGATOR AND STUDENTS EFFORTS

From 1 October 1992 to 31 August 1994, Cristino Cusano (Principal Investigator) has devoted approximately 1,000 hours to this project. During the same period, Hyung Yoon and Carl Poppe have each devoted approximately 2,000 hours to this project.

APPENDIX A : SPECIMEN TESTERS

A.1 High Pressure Tribometer (HPT)

A.1.1 Overview

The design of the facility for the tribological evaluation of critical contacts in compressors, subjected to pressurized refrigerant environments, centers on the development of a tribometer enclosed in a pressurized chamber. The tribometer was designed and manufactured by Advanced Mechanical Technology Inc. (AMTI) of Newton, Massachusetts. The data acquisition system, peripheral instrumentation and equipment were developed at the University of Illinois at Urbana-Champaign. The completed High Pressure Tribometer (HPT) system, shown in **Figure A.1**, is located in the Tribology Laboratory in the Mechanical Engineering Building (MEB) on the campus of the University of Illinois at Urbana-Champaign. A schematic of the HPT is given in **Figure A.2**.

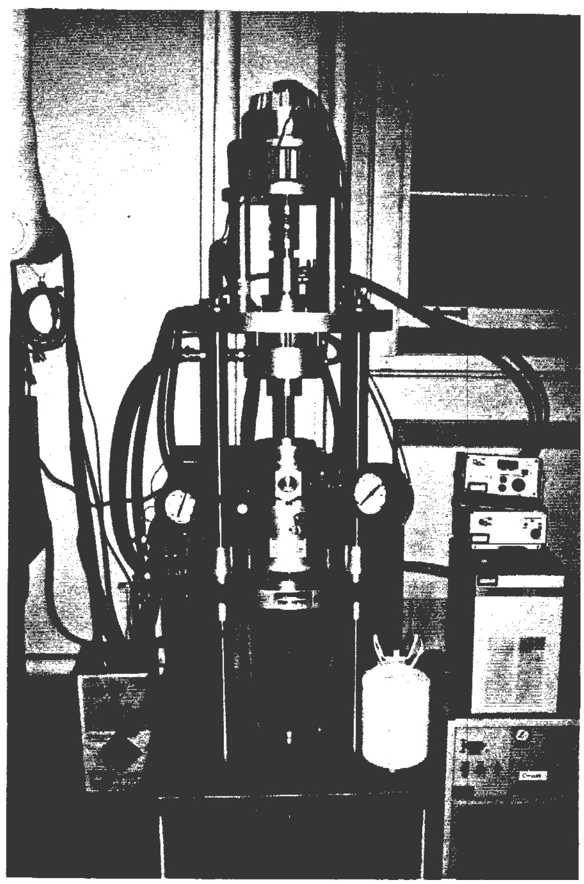


Figure A.1 - The High Pressure Tribometer (HPT)

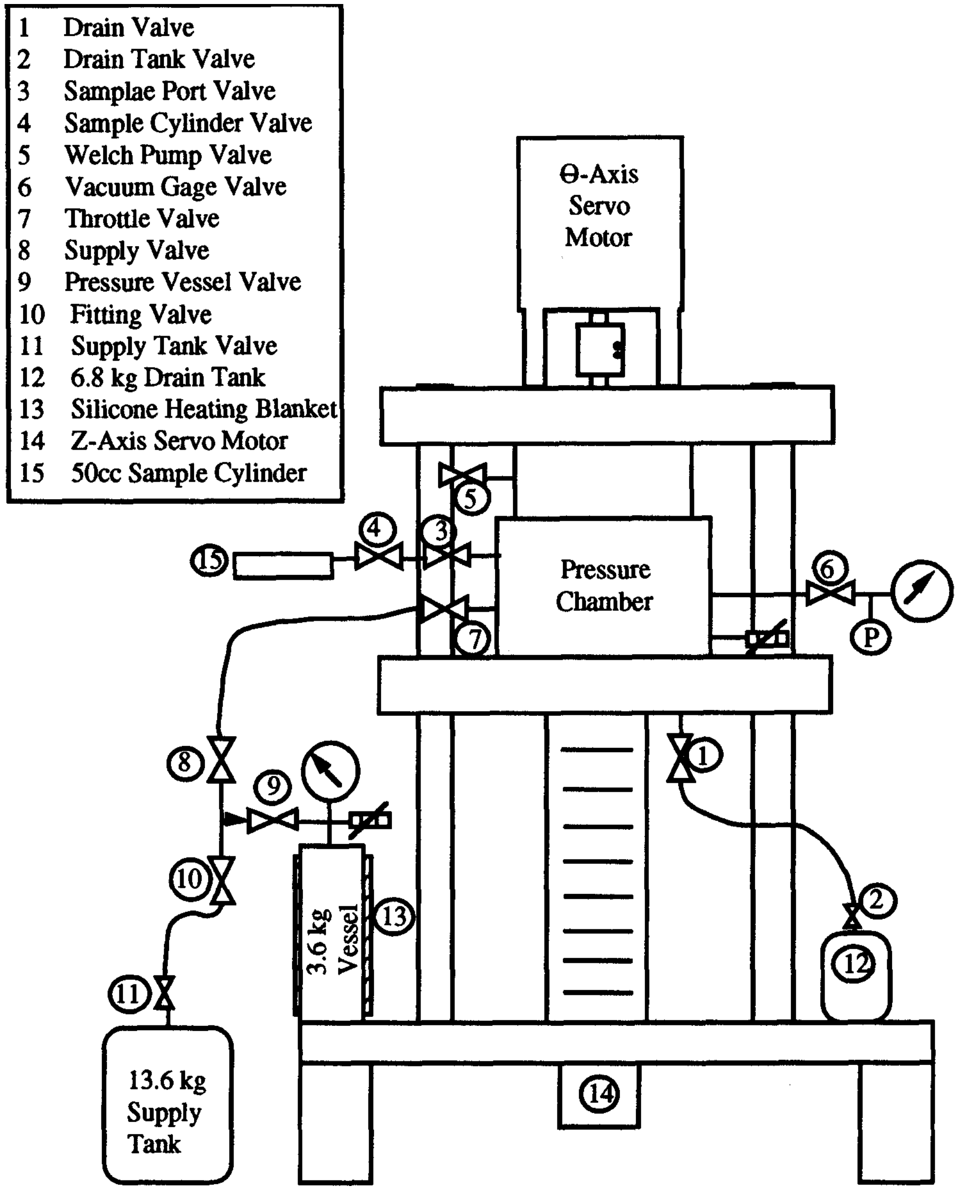


Figure A.2 - Schematic of the High Pressure Tribometer (HPT)

Central to the HPT design is a special pressure/vacuum chamber, which surrounds the tribo-contacts. This chamber is capable of testing any inflammable non-corrosive gas. Multiple thermal control loops are included to permit testing at temperatures typically found in compressors. Two separate servomotors provide motion and loading capabilities. The rotational (θ-axis) motor is capable of unidirectional rotation and oscillatory motion. The load (Z-axis) servo motor provides either static or oscillatory loads for the contact. A complex transducer measures the applied load, frictional forces, and moment during a test. The feedback from this transducer and other sensors, provide the HPT with an excellent control system.

The design of the HPT system consists of five sections: the HPT, a purging facility, a charging facility, a sampling facility and a data acquisition unit. A brief description of the apparatus follows.

A.1.2 Specimen Chamber

To adequately simulate a pressurized refrigerant environment, any test conducted must occur within the boundaries of a pressure chamber. The chamber of the HPT is rated at upwards of 250 psi operating pressure. A schematic of the pressure chamber is shown in **Figure A.3**. The chamber consists of two separate halves. The upper half of the chamber remains stationary while the lower half can be raised or lowered via the Z-axis servo motor. When the lower half engages with the upper half, a seal is formed and the unit may then be purged or pressurized.

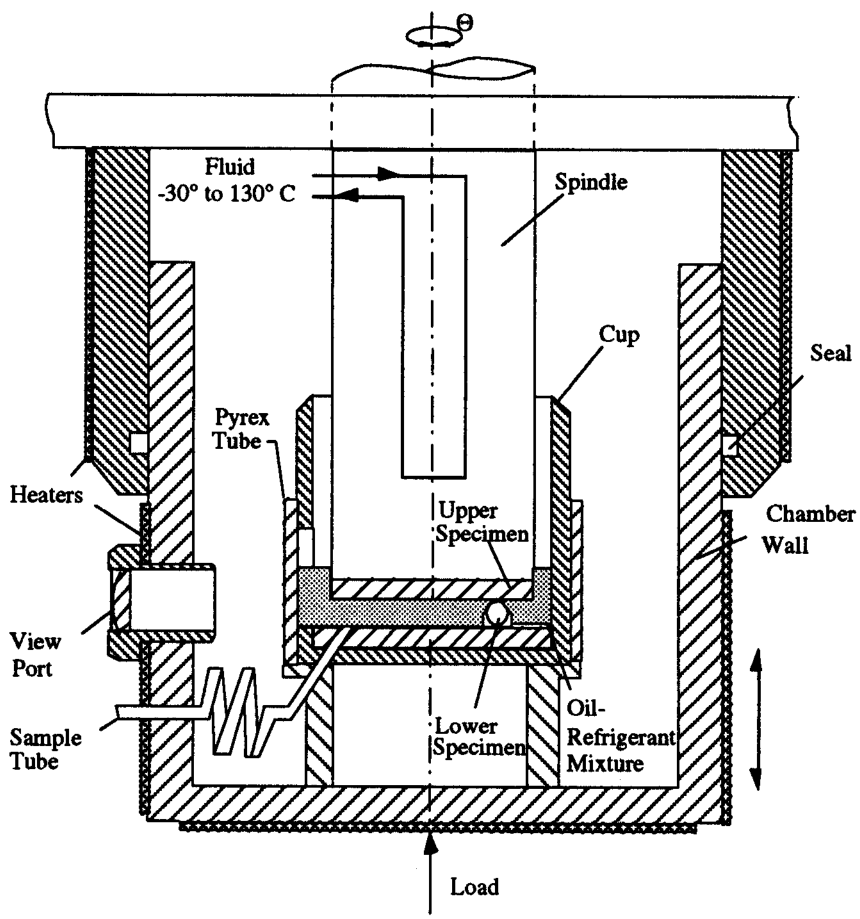


Figure A.3 - The Pressure Chamber

The spindle serves as the mounting face for the upper specimen holder. The tribo-contact occurs inside the cup. The cup is a removable aluminum piece, which serves two important functions. First, it serves as the mounting surface for the lower specimen holder. Secondly, the cup serves as the lubricant reservoir during the test. A Pyrex sleeve, sealed at the bottom by an O-ring, surrounds the cup. This permits the cup to be filled with a lubricant, completely submerging the contact to be tested. The cup contains a small hole (sampling hole) which communicates with the sampling port on the outside of the chamber and provides a means of injecting the lubricant into the cup of the pressure chamber after the chamber has been purged and pressurized.

The removable cup is bolted to a complex force transducer module. The transducer is outfitted with an intricate array of strain gages, which are used to measure the forces during a test. Frictional forces (F_x, F_y), load force (F_z), as well as moment (M_z) are of interest and are relayed to the control panel outside the pressure chamber. There is also a thermal sensor that is used to monitor the temperature of the lower specimen.

The transducer module is firmly mounted to an internal suspension system. This consists of a pair of diaphragm springs, which provide compliance in the Z-direction while maintaining high stiffness in the x, y, and θ directions. These diaphragm springs are used to permit accurate loading while the chamber is pressurized. When the chamber is pressurized to 250 psi, it takes approximately 7000 lbf to hold the two halves closed. Most of this force is taken up by the suspension system, so that with proper strain gage amplifier configuration, tests loads as low as 1 lbf can be accurately applied and monitored.

A.1.3 Thermal Systems

Virtually all internal surfaces of the chamber can be heated. This is required to prevent condensation of refrigerant on these surfaces at high test pressures. Both halves of the pressure chamber are outfitted with cartridge heaters that are used to heat the chamber walls above the condensation temperature. The upper half contains a 400 W cartridge and the lower half has two 500 W cartridges. The temperature of each half can be controlled from the main control panel.

The temperature of the rotary spindle and upper specimen is controlled from -30 °C to 130 °C by an external recirculating unit. Due to the high value of the heat transfer coefficient and the unique design of the passages machined in the spindle, the upper specimen can be maintained within a couple of degrees of the oil/refrigerant mixture temperature.

The last thermal system to be discussed is the chiller. Similar to the recirculator, the chiller is an independent unit with its own controls. This unit pumps a 50/50 mixture of laboratory grade ethylene glycol/distilled water through passages machined into portions of the HPT. It is set at ambient temperature and is used to cool critical parts of the tribometer.

A.1.4 Rotational and Axial Motions

Motion in the tribometer is generated by two independent dc servomotors. A large θ-axis servo motor provides rotational motion for the upper specimen, while a second, somewhat smaller, Z-axis servo motor provides axial motion and loading during the test.

The θ-axis dc servo motor (3 kW) is controlled through a pulse width modulated (PWM) amplifier. The low inertia motor coupled with the high performance amplifier provides excellent response and permits complex motion. The shaft of the motor is attached through a flexible helical coupling to the short shaft entering the chamber. The position of the θ-axis is monitored by a differential optical encoder that is used to control spindle motion.

The Z-axis dc servo motor is controlled separately through its own PWM amplifier. This fast response motor-amplifier combination supplies both Z-axis motion, up to 0.07 in./sec, and test loads, up to 1000 lbf. This motion is transmitted by a lead screw which is driven through a backlash-free 100:1 harmonic drive. An encoder feedback loop supplies a means to monitor the location of the lower half of the chamber, while the transducer acts as the force feedback loop that controls the applied axial load.

A.1.5 Instrumentation and Controls

There are four strain gage amplifiers: F_x, F_y, F_z and M_z. Each consist of an amplifier board that plugs into the control box and interfaces with panel controls and switches as shown in **Figure A.4**. Also shown in **Figure A.4** are the front panels of the two motor circuit boards and the front panel of the temperature control board. The motherboard, with a microprocessor, interfaces with the control circuits and provides a wide variety of functions. It interfaces with a four line LCD display and eight panel switches. In addition to interfacing with the front panel, the microprocessor can perform control, limit and alarm features. It also has the ability to interface with a personal computer, via an RS-232c port, to allow for external configuration and data acquisition.

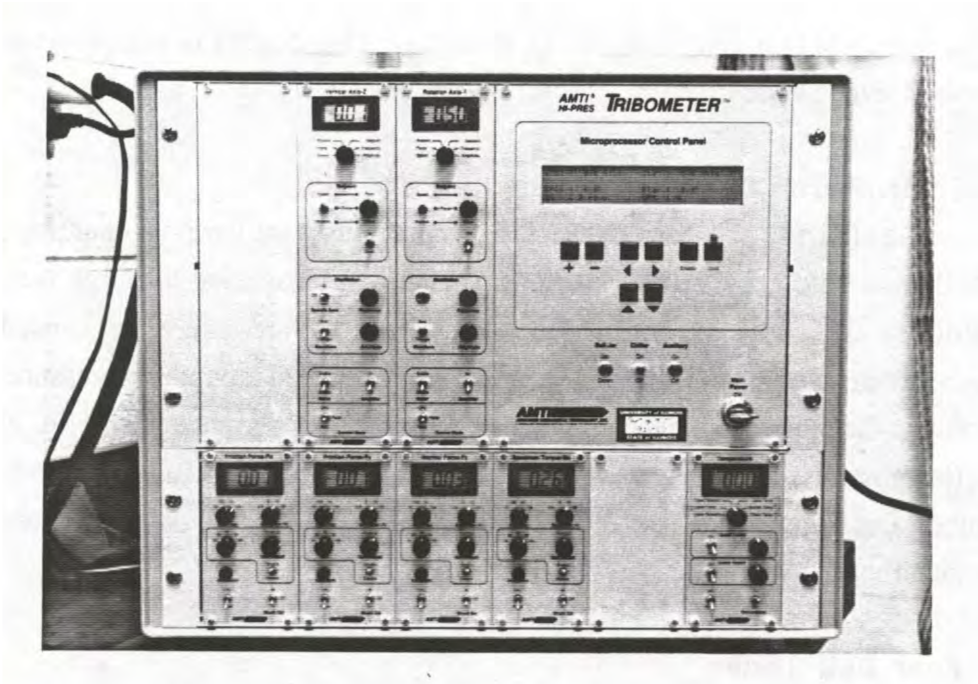


Figure A.4 - HPT Main Control Panel

Feedback for the loads is provided by the transducer in the form of F_x, F_y, F_z and M_z. Each of the force (torque) directions has its own independent amplifier that excites the strain gages and is used to condition the load to equivalent engineering units.

The Z-axis motor control board permits axial loads to be applied during the test, as well as, providing motion to open and close the chamber. The force can be static, up to 1000 lbf, or oscillatory from 0-1000 lbf at frequencies of up to 5 Hz.

The θ-axis motor control loop, with optical encoder feedback, permits the θ-axis servo motor to be precisely controlled. The motor is capable of simple unidirectional rotation (0-2000 rpm) and oscillatory motion with amplitudes of up to 180° and frequencies of up to 5 Hz. Presently, the tribometer controls permit oscillatory motion with either sinusoidal or triangular waveforms. The controls also permit a constant torque to be applied to the test.

The on-board temperature controllers are used to control the temperatures of the cartridge heaters. The upper and lower heaters can be independently set from ambient to 95 °C. Thermal sensor feedbacks from the chamber allow the temperatures to be accurately controlled to ±1 °C.

The last tribometer control is the microprocessor. The Intel 80C188EB 16-bit microprocessor, located on the motherboard, is capable of independently controlling nearly all tribometer functions. It can be used to control the two cartridge heaters as well as θ-axis and Z-axis motions. The interface to the microprocessor consists of a frontal panel keypad with eight switches and a 4 line x 40 character LCD display. This display is used to set test parameters and monitor data values.

A.1.6 Peripheral Equipment

The HPT has also been outfitted with apparatuses for purging, charging and sampling, as well as a data acquisition system. A vacuum pump is used to purge the system, more specifically, the pressure chamber (**Fig A.2**). An external pressure vessel is used to charge the chamber with refrigerant. A silicone heating blanket placed around the pressure vessel is used to generate the required refrigerant pressure to charge the pressure chamber. A 30 lb refrigerant tank (in most cases), attached to the pressure vessel, is used to supply the refrigerant to the chamber. Data acquisition is made possible through the use of a personal computer linked to the motherboard by an RS-232c connector.

A.2 Four Ball Tester

All Four Ball tests were conducted in an air environment. The operating conditions used are described in Section 3.2.5 and are based on ASTM Standard D4172-82. A schematic configuration of the Four Ball tester is shown in **Figure A.5**.

Load

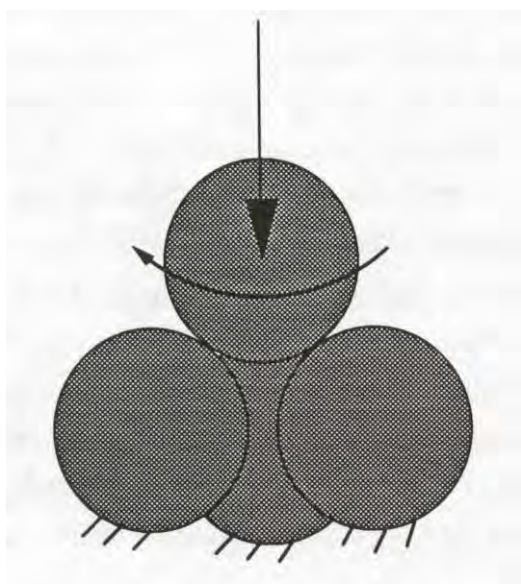
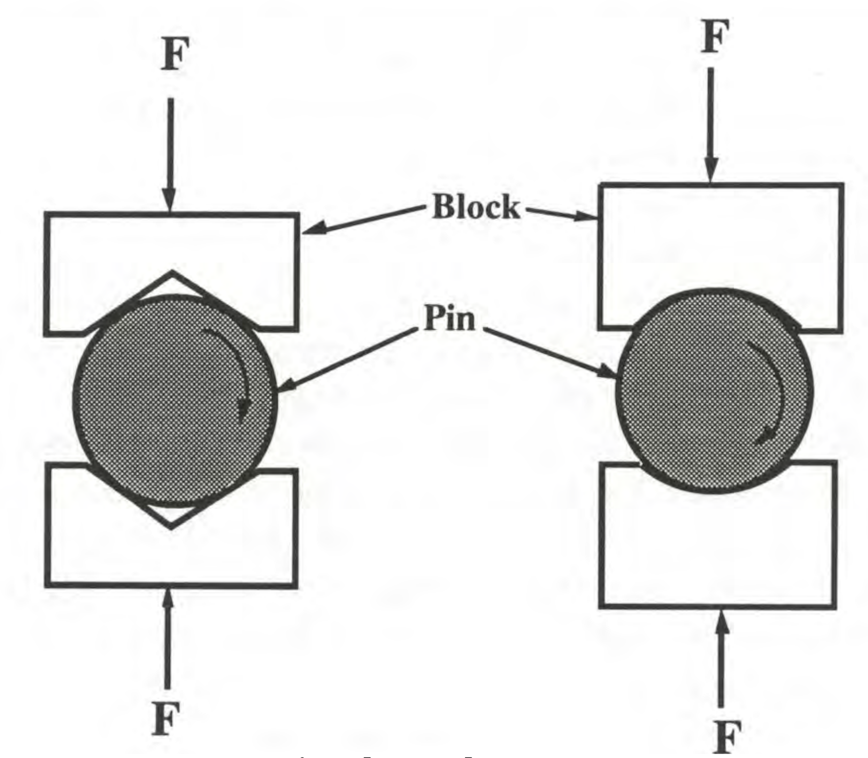


Figure A.5 - A Schematic Configuration of the Four Ball Tester

A.3 Falex™ Tester

All Falex™ data are supplied by companies participating in this research program. A schematic configuration of the Falex™ tester is shown in **Figure A.6**. For the data obtained, a 0.250 in. pin rotates at 290 rpm against a loaded stationary block. The block can have a "V" configuration or a "C" configuration. The latter configuration is usually used when evaluating softer materials.



Rotational Speed = 290 rpm

Pin diameter = 0.250"

Figure A.6 - Schematic Configurations of the Falex™ Tester

APPENDIX B : EXPERIMENTAL PROCEDURE FOR THE HPT TESTS

B.1 Specimen Preparation

Prior to testing and specimen installation, the specimens must be thoroughly cleaned. The specimens, specimen holders, screws, and cup are all ultrasonically cleaned for 10 minutes in a solution of mineral spirits followed by a rinsing with 2-propanol to remove any residues. Gloves are worn, and the specimens are handled with clean forceps during the installation process to prevent the transfer of any contaminants to the specimens, which may result in inaccurate wear measurements. The same procedures are followed upon completion of testing, after which the wear is measured.

B.2 Installation of Specimens

In order to effectively test the equivalent geometries of the critical compressor contacts, specimen holders were designed and made. The upper specimen holder is shown in **Figure B.1**, and the lower specimen holder is shown in **Figure B.2**. The upper specimen holder is used for the oscillatory tests. For the unidirectional tests, the 3" ∅ flat disk specimen is directly mounted onto the spindle. The specimens are mounted into the specimen holders, which in turn are mounted into the cup and onto the spindle. The upper specimen/specimen holder is held directly to the spindle by four 10-32 machine screws, while the lower specimen holder is secured to the cup by two 10-32 machine screws. The hole pattern in the cup permits the specimen holder to be mounted in a variety of orientations, but the most convenient orientation is that in which the specimen is near one of the sight ports.

The cup is then assembled with the glass sleeve, and the sampling hole is plugged with the threaded dowel. The cup is then filled with oil, completely covering the specimen. The cup is installed into the chamber by carefully pressing the unit into position. Care must be taken to align the sampling hole in the cup with the sampling line in the top of the transducer. The six steel locator pins on the top of the transducer assure alignment as well as providing resistance to torsional buckling. The cup is secured to the transducer by tightening three 10-32 machine screws. The threaded dowel is removed and the chamber is closed to the point just before the upper and lower specimens contact each other.

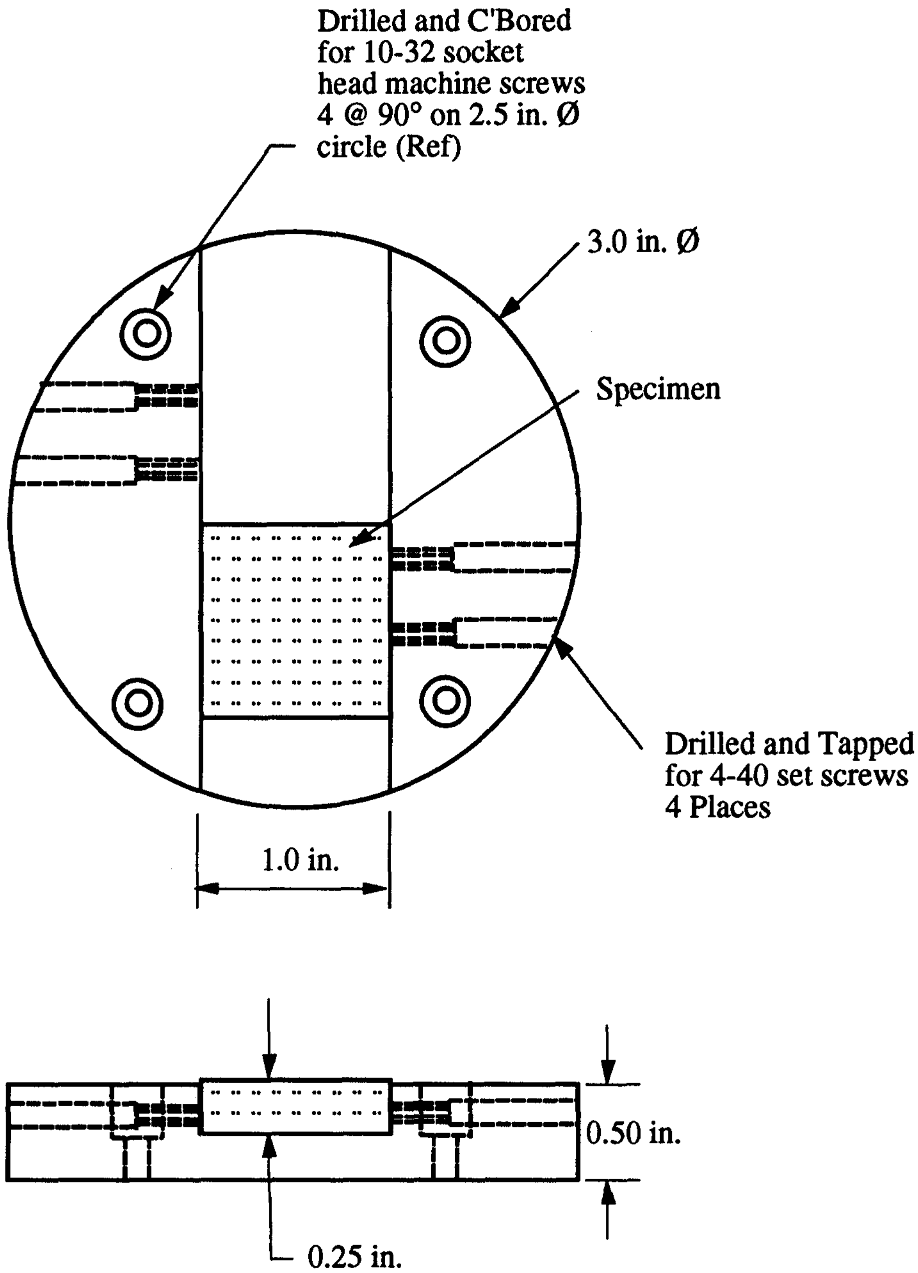
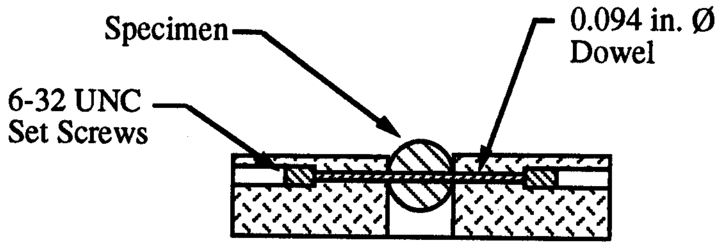


Figure B.1 - Upper Specimen Holder



Section A-A

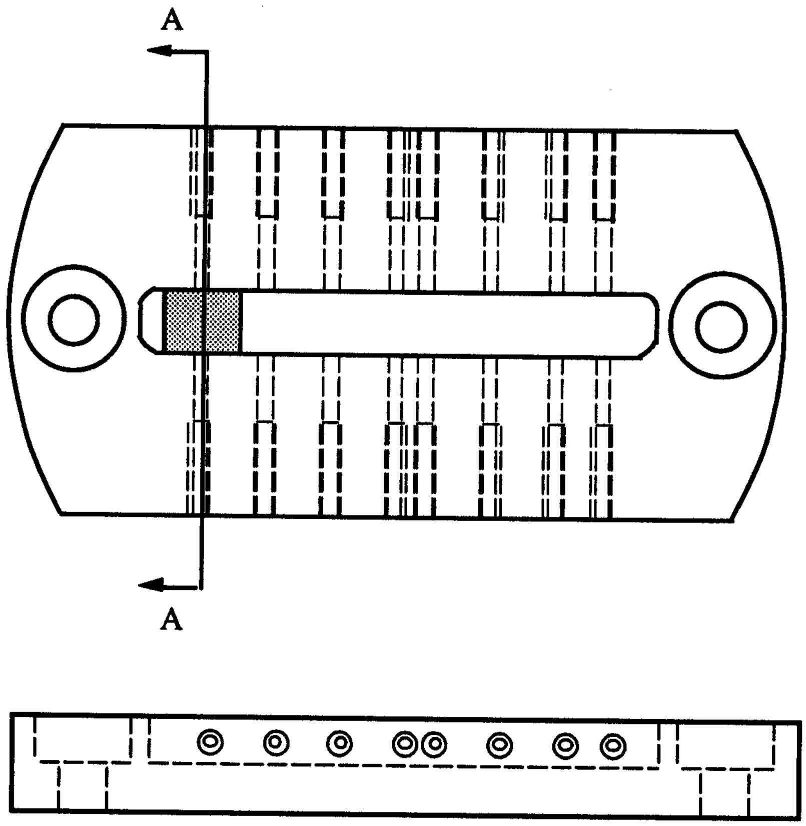


Figure B.2 - Lower Specimen Holder

B.3 Purging Procedure

After the specimens are in close proximity and the desired lubricant, if any, has been added, the system is heated to the test temperature. Both the cartridge heaters and the recirculator are set to the desired test temperature. The cartridge heaters are primarily used to prevent condensation of refrigerant. They help to heat the oil/refrigerant mixture and promote a uniform chamber temperature, thus minimizing thermal gradients.

Before purging, the charging facility and the drain facility are connected to the HPT so that their lines may be evacuated at the same time. By turning on the Welch vacuum pump, the entire chamber can be evacuated (**Figure B.3**). The pressure in the chamber is monitored by the thermal vacuum gage connected to the chamber. For tests involving oils, the chamber is evacuated down to 500 microns; while for tests without oils, the chamber can be purged to below 100 microns. The difference between these purge levels arises from the fact that the oil begins to boil violently at vacuums better than 500 microns. It should be noted that the boiling limit for the oil depends upon the type of oil being tested and its temperature.

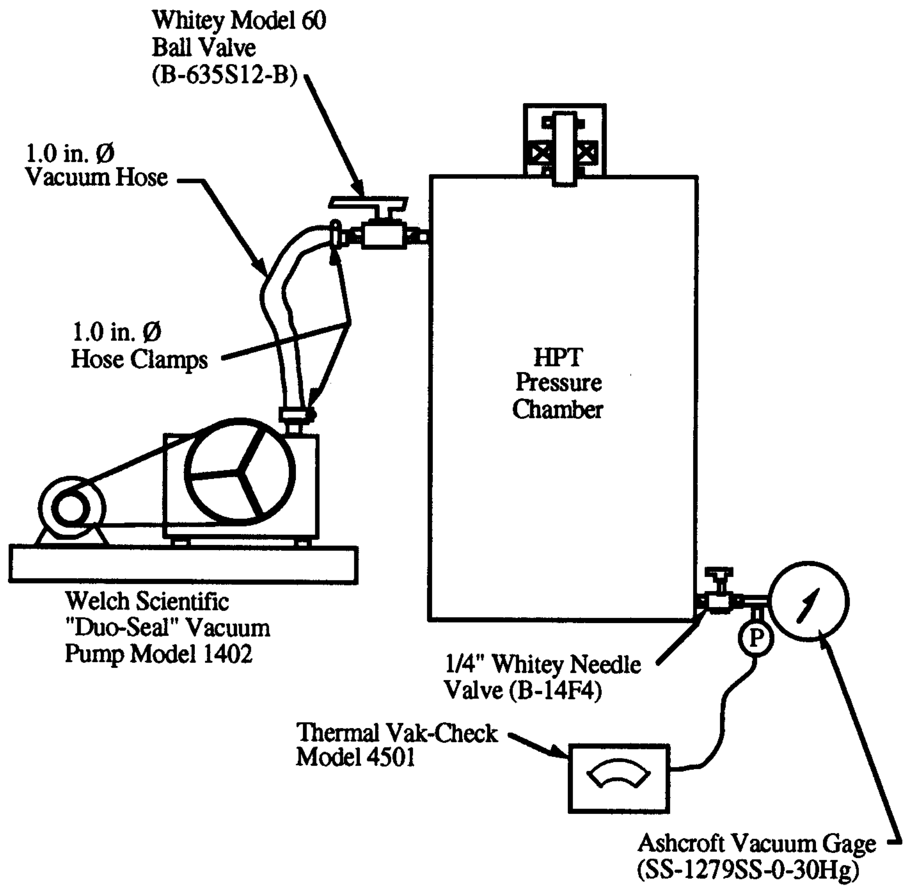


Figure B.3 - HPT Purging Facility

B.4 Charging Procedure

Before the HPT can be charged with a refrigerant vapor, the 8.0 lb pressure vessel (**Figure B.4**) must contain a sufficient amount of the desired refrigerant. The refrigerant in the pressure vessel is transferred to the HPT chamber by proper temperature control of the vessel to generate sufficient pressure. After the chamber has been purged, the chamber is now ready to be charged with refrigerant vapor. By opening valves attached to the pressure vessel and then using the valve attached to the chamber to throttle the flow of refrigerant, the desired pressure is easily obtained. Once this pressure is reached, the chamber is kept at the test pressure for at least one hour to allow the oil/refrigerant mixture to reach equilibrium, a state in which the refrigerant has fully saturated into the oil.

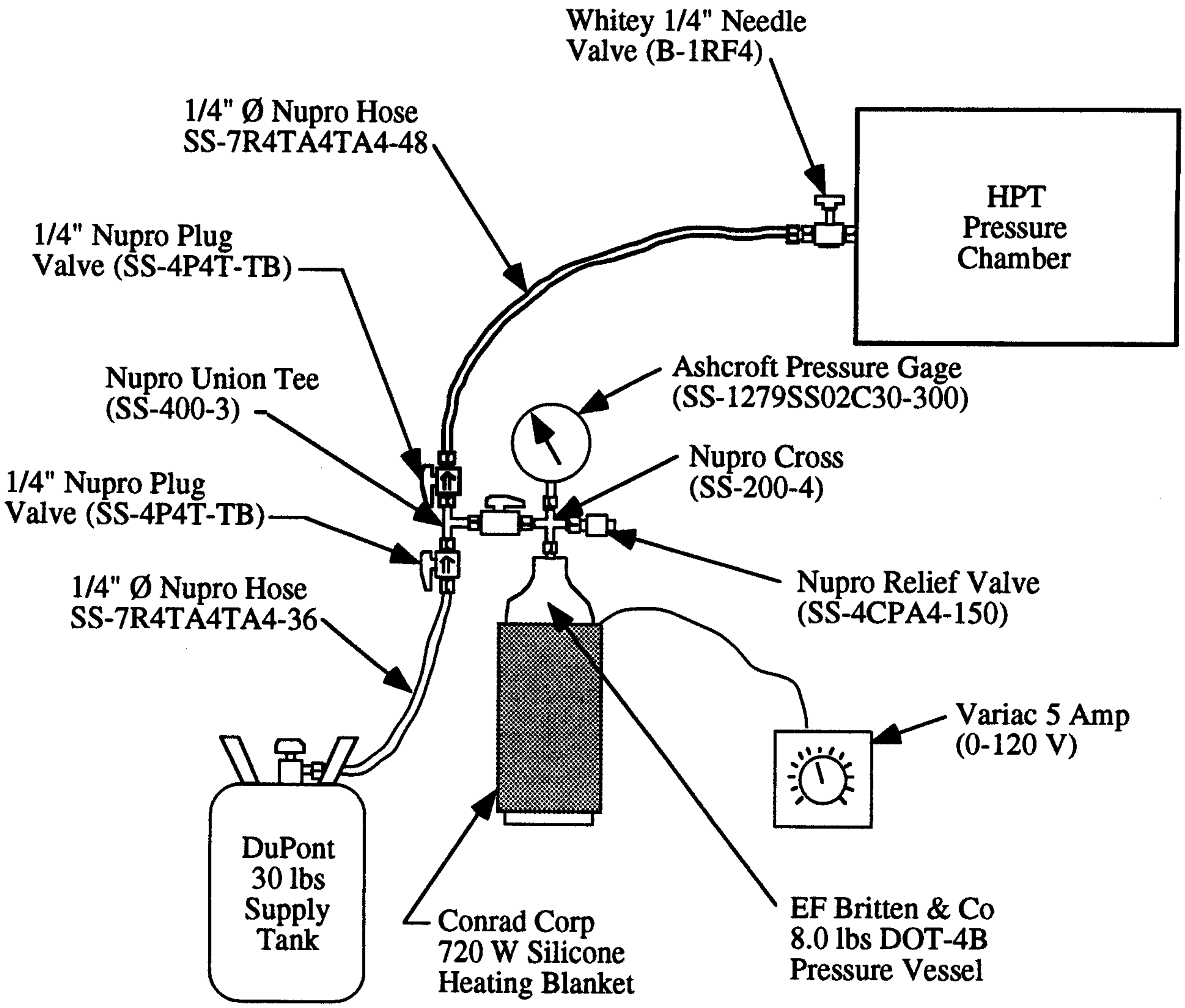


Figure B.4 - HPT Charging Facility

B.5 Running a Test

Once the chamber has reached a steady state condition, the upper and lower specimen are fully engaged and the test can begin. Each test is conducted for one hour. Once the test has begun, the data acquisition system collects instantaneous information from which an average coefficient of friction can be computed. The coefficient of friction (μ) and the upper (T1) and lower (T2) temperatures of both halves of the pressure chamber are displayed graphically as shown in **Figure B.5**.

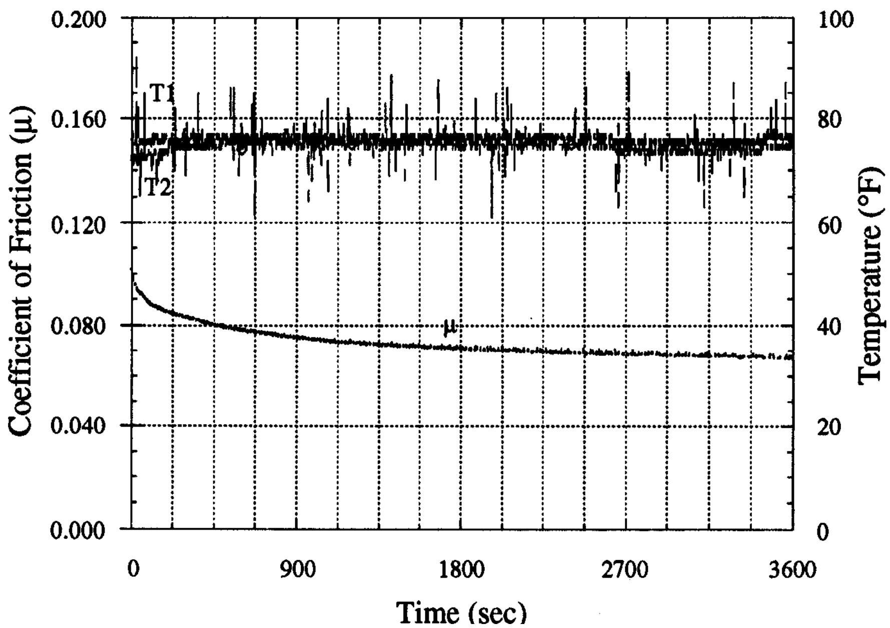


Figure B.5 - Typical Graphic Representation of Computerized Data Collection

APPENDIX C : MEASUREMENTS AND CALCULATIONS FOR THE HPT TESTS

Some common data collection schemes are used throughout this study. These include wear, surface roughness, and hardness measurements. The procedures for collecting these data are subsequently discussed in detail.

C.1 Wear Measurement on Pins

C.1.1 Nikon SMZ-2T Stereoscopic Optical Microscope

The width of the wear scar was measured with a Nikon SMZ-2T stereoscopic optical microscope. The amount of wear was evaluated by measuring the width of the wear scar on the pin. The width can be obtained with an accuracy of 0.0078 mm, which corresponds to a single division on the measuring scale of the microscope at its highest magnification. At least five readings are taken across the length of the specimen and then averaged and recorded as the average wear scar width.

C.1.2 Wear Volume Calculation

The geometry shown in **Figure C.1** is the basis by which the volume removed during a test run is obtained. Due to the high probability of propagation of error, very tight tolerances on the specimen geometry must be maintained as well as very precise measuring techniques.

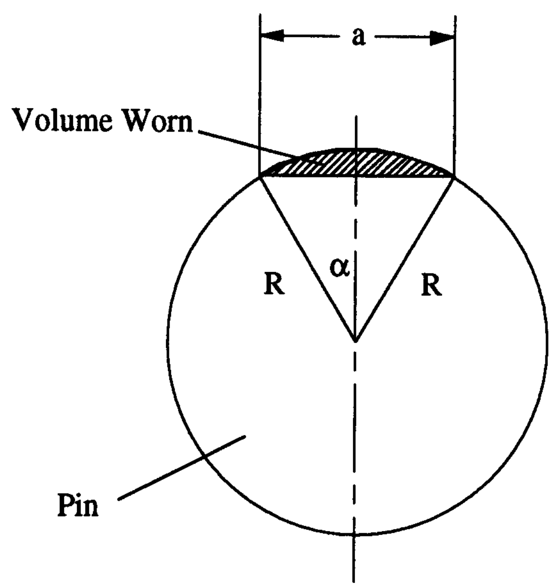


Figure C.1- Geometrical Representation for Calculation of Wear Volume

The volume worn was calculated according to the two formulae:

$$\alpha = 2\arcsin\frac{a}{2R}, \text{ and}$$

$$V = \frac{1}{2} lR\left(\alpha R - a\cos\frac{\alpha}{2}\right)$$

where l is the length of the pin, V is the volume worn, a is the width of the wear scar and R is the pin radius. Since the volume worn is highly contingent upon the dimensions of the pin as well as the wear scar width, great care is taken in the machining of the pin as well as the determination of the wear scar width for consistency of results.

C.2 Wear Measurement on Plates

C.2.1 Dektak 3030 Stylus Surface Profilometer

Upon completion of a test, the wear on the plate specimen is determined quantitatively by analyzing the wear scar depth via the Dektak 3030 stylus surface profiler. The Dektak 3030 has the capability of measuring step height down to a few nanometers. The Dektak stylus surface profilometer can measure wear scar depths of up to 120 µm. A sample printout of the trace of a wear scar from the Dektak is shown in **Figure C.2**.

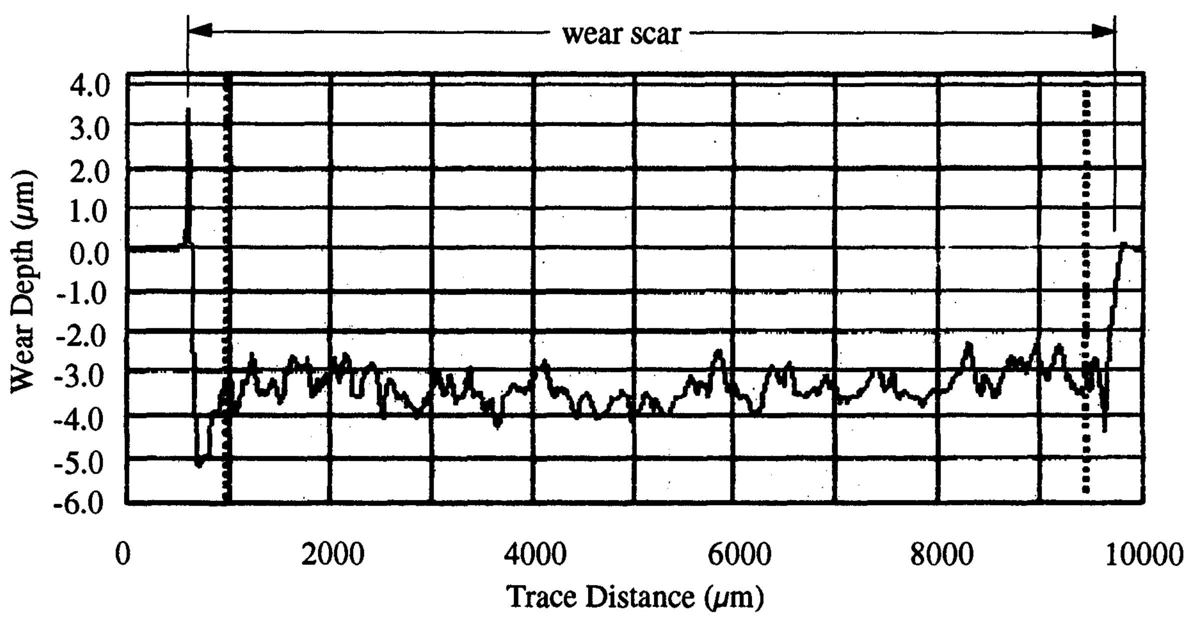


Figure C.2 - Typical Surface Profile of a Plate Wear Scar

The wear depth is determined by computing how far below the virgin (unworn) surface the average wear scar depth falls. The trace of the Dektak begins on the unworn surface, moves on the wear scar in the direction perpendicular to the wear scar, and then returns to the unworn surface on the opposite side of the wear scar. After the trace is made, the plot is leveled by pressing the level key on the control pad. The first and last point of the trace are taken as the reference points and are lined up. Using the Dektak control pad, the reference cursor is moved to the initial point of the wear scar and the measurement cursor is moved to the last point of the wear scar as shown in **Figure C.2**. The wear depth is determined by pressing the average height key on the control pad. The Dektak uses the first and last points of the trace as a bench mark. The distance between the zero reference and each data point located between the reference and measurement cursors (the data points on the wear scar) are determined, and an average value is calculated and displayed as the average height. The value of the average height is negative since the wear scar falls below the reference line.

C.2.2 Surface Profiles

Upon completion of an oscillatory or unidirectional test, the wear on the plate specimen is determined quantitatively by analyzing the wear scar depth via the Dektak stylus surface profiler. The wear scar on the surface of the aluminum pad caused by an oscillatory test is similar to that in **Figure C.3**. The wear scar depth is measured at three different locations on the pad and then an average wear scar depth is computed.

Similarly, the wear scar appearance of the plate specimen caused by a unidirectional test is shown in **Figure C.4**. The wear scar depth is measured at four separate locations and an average wear scar depth is computed.

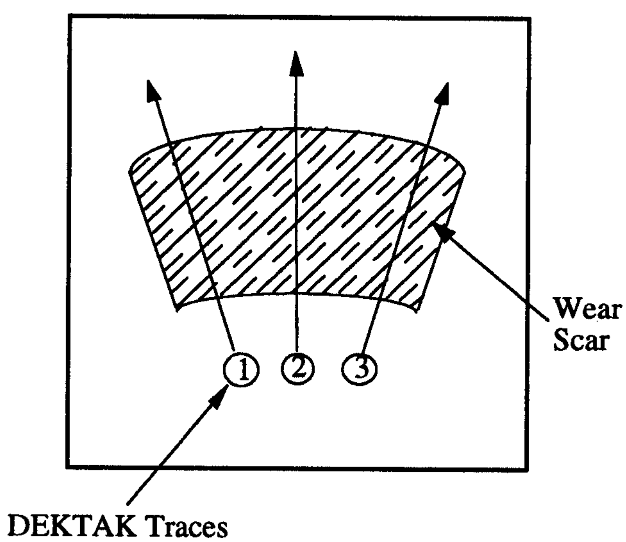


Figure C.3 - Wear Scar on an Aluminum Pad Specimen

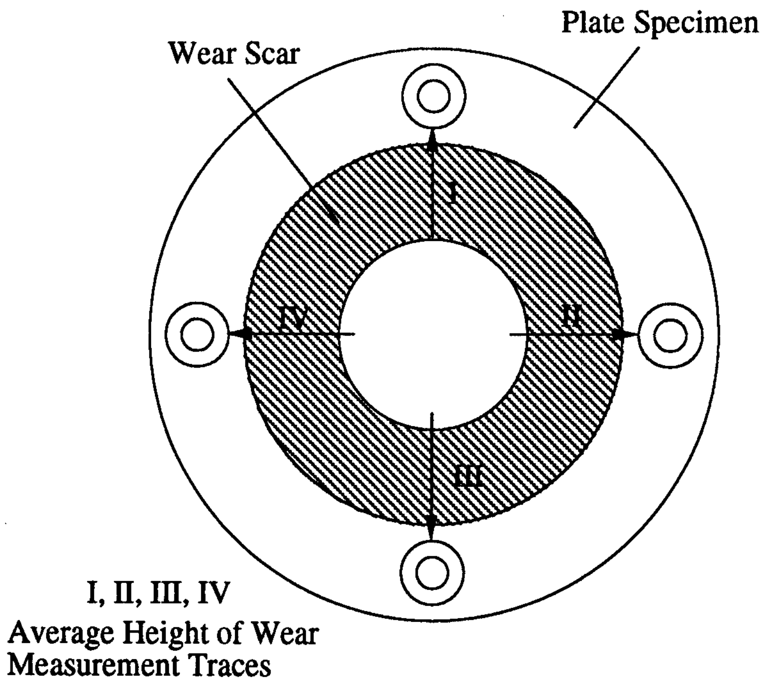


Figure C.4 - Plate Wear Measurements

C.3 Surface Roughness Measurements

Surface roughness is an important parameter in friction and wear tests. Each plate specimen is measured prior to testing. All the measurements were made with the Dektak stylus surface profiler. A load of 40 mg on the stylus was used for all the data obtained.

Traces are always taken perpendicularly to the characteristic machining marks. Each surface roughness value is obtained as an average of four measurements taken at four locations on the plate or pad, approximately equidistant from each other (**Figure C.5**).

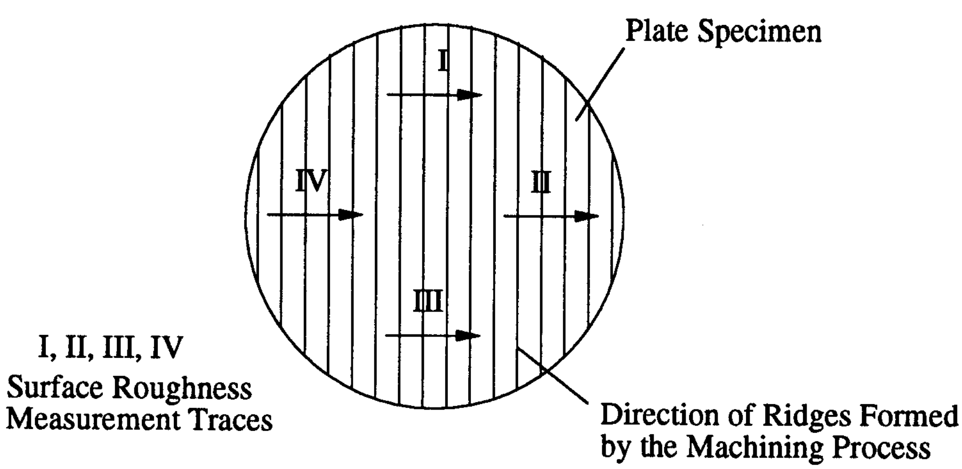


Figure C.5 - Surface Roughness Measurements

One of the most important parameters that has to be defined when measuring the surface roughness is the cut-off length (trace length). The accuracy of the measurement and the repeatability of results depend strongly on this parameter. It is common practice to choose a length one to three times larger than the characteristic peak spacing of the surface in the direction of the traverse. For most cases, the surface of the specimens are ground and had surface roughnesses in the range 0.04-0.10 μm Ra. For ground surfaces, the range of peak spacing varies from 0.08-0.8 mm. Therefore, a cut-off distance of .8 mm (800 μm) was used for the measurements. An example of a trace of the surface roughness of an aluminum pad specimen is given in **Figure C.6**.

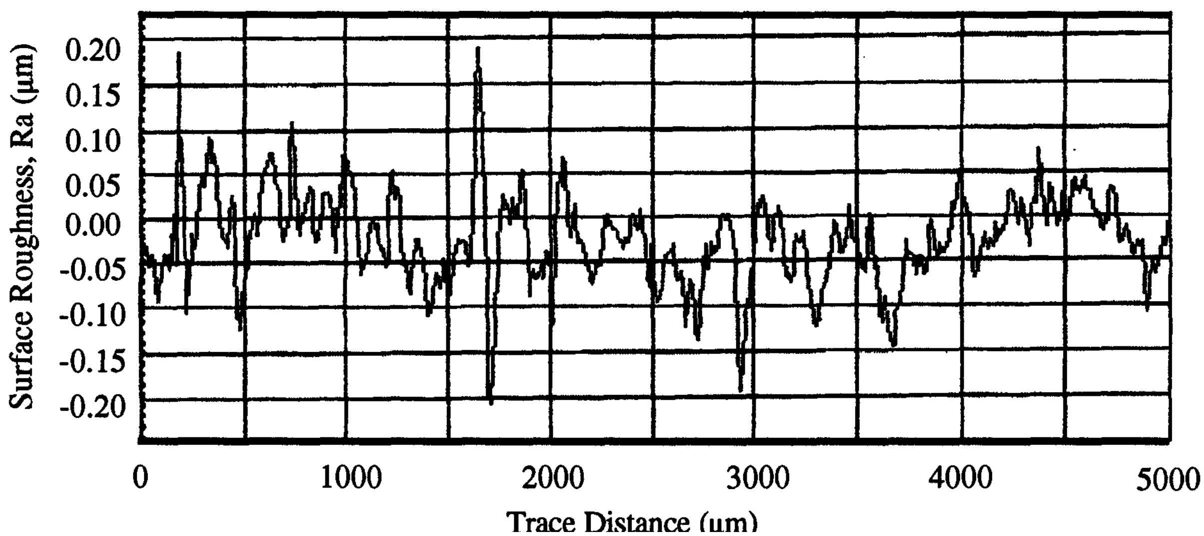


Figure C.6 - Surface Roughness Trace of an Aluminum Pad Specimen

C.4 Surface Hardness Measurements

C.4.1 Vickers Hardness Tester

Surface hardness measurements of all of the specimens except for the 308 die cast aluminum pads were measured using a Buehler Micromet II digital micro-hardness tester (Vickers), according to standard procedures. All surface hardness values used in this study are averages of at least two separate readings per specimen (pads and pins) and four per specimen (plates). The Vickers diamond indentor makes an indentation on the order of microns depending on the load used and the hardness of the material. The hardness measurement may be taken before or after running a test, so long as the indentation is on the unworn surface.

C.4.2 Brinell Hardness Tester

The surface hardness of the 308 die cast aluminum pads was measured using a Brinell hardness tester which uses a 3000 kgf, 10 mm carbide ball. These hardness measurements can only be performed on the pads after the tests have been conducted with the HPT due to the large indentations which are produced on the specimen by the Brinell hardness tester. The indentations are made on the virgin surface of the specimens. The 308 die cast aluminum pads tend to contain some porosity that requires the larger sized indentor of the Brinell apparatus for more accurate results. All surface hardness values used in this study are averages of at least two separate readings per specimen.

C.5 Viscosity Measurements

The viscosity of all of the lubricants used in this study were measured using the Brookfield Digital Viscometer and recorded at both 40°C and 100°C.

The Brookfield Viscometer rotates a sensing element in a fluid and measures the torque necessary to overcome the viscous resistance to the induced movement. This is accomplished by driving the immersed element (spindle) through a beryllium copper spring. The degree to which the spring is wound, indicated by the digital display, is proportional to the viscosity of the fluid.

The viscometer is able to measure over a number of ranges since, for a given spring deflection, the actual viscosity is proportional to the spindle speed and is related to the spindle's size and shape. The temperature controller has a temperature range of up to 300°C.

A typical test procedure consists of first cleaning all of the parts (spindle and test chamber) with mineral spirits followed by 2-propanol. Second, a sample oil volume of 8 cc's is added to the test chamber and brought up to the appropriate temperature (i.e., 40°C or 100°C) by means of the temperature controller. Next, the spindle is attached to the motor-spring and then carefully lowered into the oil sample. The motor is finally turned on causing the spindle to rotate. After a few minutes, equilibrium (temperature and velocity) is reached and a reading can be taken from the digital display. This reading is multiplied by a predetermined factor, which is a function of angular velocity and is directly converted into viscosity.

C.6 Solubility Measurements

When taking an oil/refrigerant sample from the pressure chamber it is important that the following procedure is consistently followed to ensure accurate results. The sampling procedure is based on ASHRAE standards [1].

Before taking a sample, it is important to completely clean the sample cylinder (**Figure C.7**) and fittings with mineral spirits followed by 2-propanol. The sample cylinder is purged for five minutes by the Thermal vacuum pump and immediately weighed (W_1).

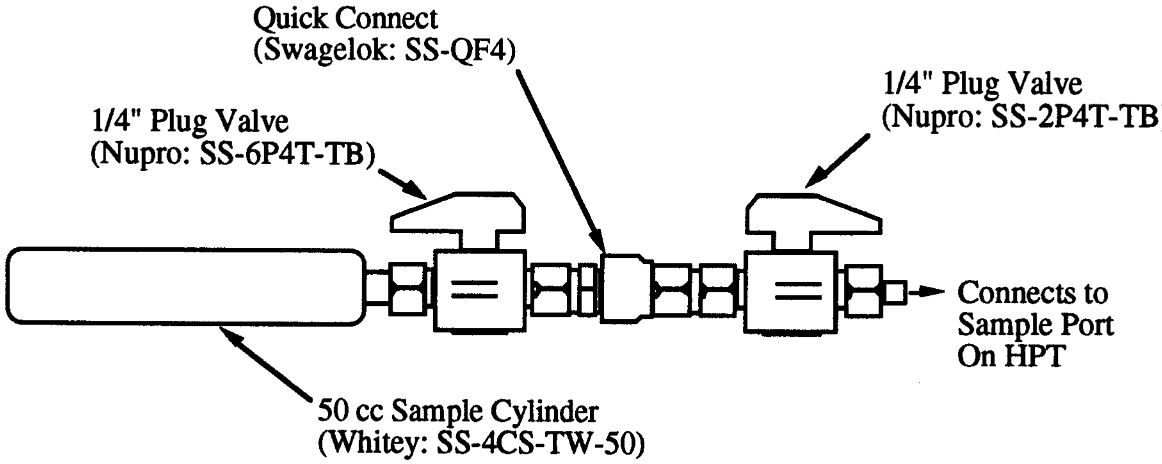


Figure C.7 - Sampling Cylinder

A sample can now be drawn off from the HPT by connecting the sample cylinder to the pressure chamber via the sample port, opening the valves, and waiting five minutes for the sample cylinder to be completely filled. At this point, the sample cylinder is removed from the pressure chamber, completely cleaned with mineral spirits followed by 2-propanol. The mixture of oil and refrigerant is then weighed (W_3).

Finally, the sample cylinder is attached to a capillary tube (3.05 meters long x 0.63 mm ∅) and the refrigerant is bled off from the oil. The process takes approximately two hours. At that time the capillary tube is connected to the Thermal vacuum pump and further purged for one hour with the Thermal vacuum pump. This will remove any additional refrigerant that may be dissolved in the oil. At this point, the sample cylinder is cleaned once again with mineral spirits and 2-propanol and then weighed (W_5).

The weight percent of refrigerant saturated into the oil is calculated using the following relation:

$$W_{\%} = \left[\frac{W_3 - W_5}{W_3 - W_1}\right] x100$$

where W_1 is the weight of the empty sample cylinder, W_3 is the weight of the sample cylinder, oil and refrigerant, and W_5 is the weight of the sample cylinder and oil.

References

1. ASHRAE, 1984. ANSI/ASHRAE 41.4-1984, Standard Procedure for experimentally determining the weight concentration of oil in single phase solutions of oil in refrigerant. Atlanta: American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

APPENDIX D : HPT RAW DATA FOR PART I

Table D.1 - Raw Data for Case 1
(3.8 in./s at 185 lbf)

Test#	Refrig	Oil	Pressure (psi)	Coeff. of Friction	Volume worn (mm^3)	Ave. Co Friction	Ave. Vol. worn (mm^3)
42FT	R-22	Mineral 1	85	0.092	0.0144	0.093	0.0137
43FT	R-22	Mineral 1	85	0.094	0.0130		
63FT	R-134a	Ester 1	85	0.101	0.0165	0.102	0.0155
64FT	R-134a	Ester 1	85	0.102	0.0145		
31FT	R-134a	Ester 2	85	0.082	0.0108	0.088	0.0131
32FT	R-134a	Ester 2	85	0.093	0.0154		
33FT	R-134a	Ester 3	85	0.072	0.0131	0.074	0.0142
34FT	R-134a	Ester 3	85	0.076	0.0152		
39FT	Blend	Ester 1	85	0.077	0.0177	0.075	0.0203
40FT	Blend	Ester 1	85	0.072	0.0229		
37FT	Blend	Ester 2	85	0.067	0.0167	0.068	0.0192
38FT	Blend	Ester 2	85	0.069	0.0218		
35FT	Blend	Ester 3	85	0.083	0.0094	0.083	0.0113
36FT	Blend	Ester 3	85	0.083	0.0131		

Table D.2 - Raw Data for Case 1
(23 in./s at 370 lbf)

Test#	Refrig	Oil	Pressure (psi)	Coeff. of Friction	Volume worn (mm^3)	Ave. Co Friction	Ave. Vol. worn (mm^3)
59FT	R-22	Mineral 1	85	0.076	0.0415	0.078	0.0427
60FT	R-22	Mineral 1	85	0.080	0.0440		
61FT	R-134a	Ester 1	85	0.062	0.0455	0.062	0.0431
62FT	R-134a	Ester 1	85	0.061	0.0408		
51FT	R-134a	Ester 2	85	0.075	0.0463	0.074	0.0486
52FT	R-134a	Ester 2	85	0.072	0.0510		
53FT	R-134a	Ester 3	85	0.053	0.0614	0.056	0.0609
54FT	R-134a	Ester 3	85	0.059	0.0605		
65FT	Blend	Ester 1	85	0.060	0.0560	0.058	0.0543
66FT	Blend	Ester 1	85	0.055	0.0526		
55FT	Blend	Ester 2	85	0.060	0.0556	0.060	0.0517
56FT	Blend	Ester 2	85	0.060	0.0478		
57FT	Blend	Ester 3	85	0.067	0.0522	0.066	0.0456
58FT	Blend	Ester 3	85	0.065	0.0391		

Table D.3 - Raw Data for Case 1
(23 in./s at 185 lbf)

Test#	Refrig	Oil	Pressure (psi)	Coeff. of Friction	Volume worn (mm^3)	Ave. Co Friction	Ave. Vol. worn (mm^3)
3FT	R-22	Mineral 1	85	0.082	0.0115	0.081	0.0113
50FT	R-22	Mineral 1	85	0.080	0.0111		
7FT	R-134a	Ester 1	85	0.060	0.0196	0.058	0.0193
44FT	R-134a	Ester 1	85	0.056	0.0190		
11FT	R-134a	Ester 2	85	0.066	0.0171	0.065	0.0181
45FT	R-134a	Ester 2	85	0.064	0.0191		
16FT	R-134a	Ester 3	85	0.060	0.0198	0.056	0.0202
46FT	R-134a	Ester 3	85	0.052	0.0206		
27FT	Blend	Ester 1	85	0.061	0.0235	0.059	0.0232
47FT	Blend	Ester 1	85	0.057	0.0229		
24FT	Blend	Ester 2	85	0.067	0.0177	0.063	0.0183
49FT	Blend	Ester 2	85	0.059	0.0189		
20FT	Blend	Ester 3	85	0.075	0.0127	0.073	0.0131
48FT	Blend	Ester 3	85	0.071	0.0135		

Table D.4 - Raw Data for Case 2

Test#	Refrig	Oil	Load (lbf)	Pressure (psig)	Speed (in./s)	Co. of Friction	Wear on Disk (g)	Ave. Co. of Friction	Ave. Wear on Disk (g)
31FC	R134a	Ester 4	50	30	23	0.184	0.4024	0.173	0.4019
42FC	R134a	Ester 4	50	30	23	0.161	0.4015		
33FC	R134a	Ester 5	50	30	23	0.168	0.3827	0.168	0.3744
34FC	R134a	Ester 5	50	30	23	0.168	0.3660		
35FC	R134a	Ester 6	50	30	23	0.173	0.3604	0.180	0.3729
36FC	R134a	Ester 6	50	30	23	0.187	0.3853		
39FC	R12	Mineral 2	50	30	23	0.154	0.0853	0.133	0.0858
40FC	R12	Mineral 2	50	30	23	0.112	0.0862		
73FC	R134a	Ester 4	50	30	3.8	0.168	0.0797	0.174	0.0845
74FC	R134a	Ester 4	50	30	3.8	0.179	0.0893		
69FC	R134a	Ester 5	50	30	3.8	0.130	0.0763	0.136	0.0790
70FC	R134a	Ester 5	50	30	3.8	0.142	0.0817		
67FC	R134a	Ester 6	50	30	3.8	0.213	0.0864	0.216	0.0940
68FC	R134a	Ester 6	50	30	3.8	0.218	0.1015		
60FC	R12	Mineral 2	50	30	3.8	0.145	0.0092	0.139	0.0108
61FC	R12	Mineral 2	50	30	3.8	0.132	0.0123		
75FC	R134a	Ester 4	100	30	3.8	0.176	0.0982	0.176	0.1004
76FC	R134a	Ester 4	100	30	3.8	0.175	0.1025		
63FC	R134a	Ester 5	100	30	3.8	0.189	0.1066	0.176	0.1014
64FC	R134a	Ester 5	100	30	3.8	0.162	0.0961		
65FC	R134a	Ester 6	100	30	3.8	0.201	0.1078	0.208	0.1033
66FC	R134a	Ester 6	100	30	3.8	0.214	0.0987		
71FC	R12	Mineral 2	100	30	3.8	0.108	0.0210	0.117	0.0238
72FC	R12	Mineral 2	100	30	3.8	0.125	0.0265		

Table D.5 - Raw Data for Case 3

Test#	Refrig	Oil	Load (lbf)	Pressure (psig)	Speed (in./s)	Coeff. of Friction	Wear Depth (μm)	Ave. Coeff. of Friction	Ave. Wear Depth (μm)
48FC	R 134a	Ester 7	50	7	± 3.8	0.113	30	0.12	29.9
49FC	R 134a	Ester 7	50	7	± 3.8	0.129	29.8		
50FC	R 134a	Ester 8	50	7	± 3.8	0.124	24.2	0.12	26.5
51FC	R 134a	Ester 8	50	7	± 3.8	0.119	29.2		
52FC	R 134a	Ester 9	50	7	± 3.8	0.106	52.5	0.12	50.8
53FC	R 134a	Ester 9	50	7	± 3.8	0.123	49.2		
58FC	R 12	Mineral 2	50	7	± 3.8	0.144	18	0.15	15.6
59FC	R 12	Mineral 2	50	7	± 3.8	0.154	13.2		

Table D.6 - Raw Data for Case 4

Test #	Refrig	Oil	Load (lbf)	Pressure (psig)	Speed (in./s)	Coeff. of Friction	Wear Depth (μm)	Ave. Co Friction	Ave Wear Depth (μm)
32TE	R12	Mineral 2	400	7	15.9	0.116	0.229	0.115	0.2
33TE	R12	Mineral 2	400	7	15.9	0.114	0.171		
18TE	R134a	Ester 7	400	7	15.9	0.070	4.00	0.072	4.63
23TE	R134a	Ester 7	400	7	15.9	0.073	5.26		
36TE	R134a	Ester 8	400	7	15.9	0.096	58.52	0.094	56.11
24TE	R134a	Ester 8	400	7	15.9	0.092	53.70		
17TE	R134a	Ester 9	400	7	15.9	0.079	35.60	0.079	34.50
20TE	R134a	Ester 9	400	7	15.9	0.078	33.40		
34TE	R12	Mineral 2	300	7	15.9	0.120	0.425	0.119	0.425
35TE	R12	Mineral 2	300	7	15.9	0.118			
26TE	R134a	Ester 7	300	7	15.9	0.081	1.79	0.080	1.91
27TE	R134a	Ester 7	300	7	15.9	0.079	2.02		
28TE	R134a	Ester 8	300	7	15.9	0.097	18.8	0.097	18.1
29TE	R134a	Ester 8	300	7	15.9	0.097	17.3		
25TE	R134a	Ester 9	300	7	15.9	0.079	3.59	0.079	3.77
22TE	R134a	Ester 9	300	7	15.9	0.078	3.95		

APPENDIX E : HPT RAW DATA FOR PART II

Table E.1 - Raw Data for Case 5
(±3.8 in./s; 50 lbf; 30 psi; 266°F)

Test No.	Spec. I.D.		Lubricant	Environment	Coefficient of Friction	Scar Depth (μm)
	(lower)	(upper)				
4CO-2	4	5	Ester 5	R-134a	0.113	20.3
5CO-2	5	4	Ester 5	R-134a	0.118	18.4
12CO-2	12	12	Ester 5	R-134a	0.105	17.8
13CO-2	13	13	Ester 5	R-134a	0.131	26.1
6CO-2	6	6	Ester 6	R-134a	0.126	27.1
7CO-2	7	7	Ester 6	R-134a	0.114	26.6
10CO-2	10	10	Ester 6	R-134a	0.111	26.2
11CO-2	11	11	Ester 6	R-134a	0.121	25.5
2CO-2	2	2	Ester 4	R-134a	0.130	27.2
14CO-2	14	14	Ester 4	R-134a	0.135	24.0
15CO-2	15	15	Ester 4	R-134a	0.134	26.7
22CO-2	22	22	Ester 4	R-134a	0.099	26.6
8CO-2	8	8	Mineral 2	R-12	0.138	8.4
9CO-2	9	9	Mineral 2	R-12	0.133	6.4
16CO-2	16	16	Ester 5	air	0.065	16.0
17CO-2	17	17	Ester 5	air	0.090	17.0
18CO-2	18	18	Ester 6	air	0.095	18.7
19CO-2	19	19	Ester 6	air	0.080	23.1
20CO-2	20	20	Ester 4	air	0.087	26.3
21CO-2	21	21	Ester 4	air	0.088	22.0

Table E.2 - Raw Data for Case 6
(±3.8 in./s; 25 lbf; 7 psi; 266°F)

Test No.	Spec. I.D.		Lubricant	Environment	Coefficient of Friction	Scar Depth (μm)
	(lower)	(upper)				
15TE-2	15	15	Ester 8	R-134a	0.080	27.8
16TE-2	16	16	Ester 8	R-134a	0.085	25.0
33TE-2	33	33	Ester 8	R-134a	0.095	24.7
17TE-2	17	17	Ester 7	R-134a	0.091	36.2
18TE-2	18	18	Ester 7	R-134a	0.093	33.3
34TE-2	34	34	Ester 7	R-134a	0.093	41.6
19TE-2	19	19	Ester 9	R-134a	0.076	28.7
20TE-2	20	20	Ester 9	R-134a	0.088	25.5
32TE-2	32	32	Ester 9	R-134a	0.095	32.6
21TE-2	21	21	Mineral 2	R-12	0.111	8.6
22TE-2	22	22	Mineral 2	R-12	0.137	9.6
23TE-2	23	23	Ester 8	air	0.159	12.6
24TE-2	24	24	Ester 8	air	0.145	16.2
29TE-2	29	29	Ester 8	air	0.134	11.3
25TE-2	25	25	Ester 7	air	0.109	24.1
26TE-2	26	26	Ester 7	air	0.150	28.5
30TE-2	30	30	Ester 7	air	0.114	25.0
27TE-2	27	27	Ester 9	air	0.157	21.1
28TE-2	28	28	Ester 9	air	0.142	20.0
31TE-2	31	31	Ester 9	air	0.075	19.3

Table E.3 - Raw Data for Case 7
(±20.6 in./s; 350 lbf; 170 psi; 160°F)

Test No.	Spec. I.D.		Lubricant	Environment	Coefficient of Friction	Scar Depth (μm)
	(lower)	(upper)				
3CAR	2A	2A	Ester 10	R-134a	0.082	0.9466
4CAR	2B	2B	Ester 10	R-134a	0.085	1.0613
18CAR	10A	10A	Ester 10	R-134a	0.082	1.1386
19CAR	10B	10A	Ester 10	R-134a	0.081	0.8673
5CAR	3AC	3AC	Ester 11	R-134a	0.054	0.5272
12CAR	7A	7A	Ester 11	R-134a	0.073	0.5850
13CAR	7B	7B	Ester 11	R-134a	0.087	0.7449
16CAR	9A	9A	Ester 11	R-134a	0.055	0.5689
14CAR	8A	8A	Ester 12	R-134a	0.075	1.3640
2CAR	1B	1B	Ester 12	R-134a	0.082	1.0740
17CAR	9B	9B	Ester 12	R-134a	0.076	1.3790
22CAR	12A	12A	Ester 12	R-134a	0.079	1.0404
23CAR	12B	12B	Ester 12	R-134a	0.082	1.3984
10CAR	6A	6A	Ester 10	air	0.014	0.1361
11CAR	6B	6B	Ester 10	air	0.034	0.1167
6CAR	3AH	3AH	Ester 11	air	0.012	0.1055
7CAR	4AH	4AH	Ester 11	air	0.021	0.1061
8CAR	5A	5A	Ester 12	air	0.031	0.0301
9CAR	5B	5B	Ester 12	air	0.036	0.0247

Table E.4 - Raw Data for Case 8
(±3.8 in./s; 50 lbf; 60 psi; 144°F)

Test No.	Spec. I.D.		Lubricant	Environment	Coefficient of Friction	Scar Depth (μm)
	(lower)	(upper)				
8WIT	8	8	Ester 13	R-134a	0.110	69.3
19WIT	19	19	Ester 13	R-134a	0.105	63.0
11WIT	11	11	Ester 14	R-134a	0.106	34.5
12WIT	12	12	Ester 14	R-134a	0.100	36.1
9WIT	9	9	Ester 15	R-134a	0.069	7.3
10WIT	10	10	Ester 15	R-134a	0.059	5.5
13WIT	13	13	Ester 13	air	0.098	7.2
14WIT	14	14	Ester 13	air	0.091	5.9
15WIT	15	15	Ester 14	air	0.087	5.0
16WIT	16	16	Ester 14	air	0.089	4.6
17WIT	17	17	Ester 15	air	0.059	4.7
18WIT	18	18	Ester 15	air	0.047	4.1

Table E.5 - Raw Data for Case 9
(±20.6 in./s; 250 lbf; 225 psi; 200°F)

Test No.	Spec. I.D. (lower)	Spec. I.D. (upper)	Lubricant	Refrigerant	Coefficient of Friction	Scar Depth (μm)
1TE-3	1A	1A	Alkbenz 1	R-22	0.080	5.24
2TE-3	1B	1B	Alkbenz 1	R-22	0.078	6.61
3TE-3	1C	1C	Alkbenz 1	R-22	0.079	10.19
4TE-3	1D	1D	Alkbenz 1	R-22	0.078	4.92
5TE-3	2A	2A	Alkbenz 2	R-22	0.079	3.35
6TE-3	2B	2B	Alkbenz 2	R-22	0.072	4.04
7TE-3	2C	2C	Alkbenz 2	R-22	0.077	3.75
8TE-3	2D	2D	Alkbenz 2	R-22	0.080	3.90
9TE-3	3A	3A	Mineral 3	R-22	0.076	3.89
10TE-3	3B	3B	Mineral 3	R-22	0.078	4.97
11TE-3	3C	3C	Mineral 3	R-22	0.073	4.02
12TE-3	3D	3D	Mineral 3	R-22	0.069	3.04

APPENDIX F : RAW DATA FOR THE FOUR BALL TESTS

Table F.1- Wear Data Obtained From a Four Ball Tester

Lubricants	Test #1 Wear Scar (mm)	Test #2 Wear Scar (mm)	Ave. Wear Scar (mm)
#Ester 4	0.77	0.79	0.78
#Ester 5*	0.60	0.55	0.58
Ester 6*	0.74	0.80	0.77
#Ester 7*	0.80	0.78	0.79
Ester 8	1.08	1.04	1.06
Ester 9	0.81	0.78	0.80
Ester 10*	1.10	1.08	1.09
Ester 11	1.04	0.96	1.00
Ester 12	1.00	1.05	1.03
Ester 13	0.96	0.96	0.96
Ester 14	0.89	0.87	0.88
Ester 15	0.86	0.85	0.86
Alkbenz 1*	0.68	0.68	0.68
Alkbenz 2*	0.69	0.60	0.65
Mineral 3*	0.57	0.51	0.54
# Same Base Lubricant; * Formulated			

Table F.2 - Statistical Wear Data for the Four Ball Tests (Case 5)

Ester 4			Ester 5			Ester 6		
X_1	σ_1	N_1	X_2	σ_2	N_2	X_3	σ_3	N_3
0.78	0.01414	2	0.58	0.03536	2	0.77	0.04243	2
X_i = Mean; σ_i = Standard Deviation; N_i = Number of Tests								

Table F.3 - Confidence Intervals for Case 5 Using Small Sample Theory

X_1-X_2	0.20	σ_{12}	0.038	t_{12}	5.249	n_{12}	2	$\%_{12}$	99
X_2-X_3	0.19	σ_{23}	0.055	t_{23}	3.442	n_{23}	2	$\%_{23}$	96
X_1-X_3	0.01	σ_{13}	0.045	t_{13}	0.224	n_{13}	2	$\%_{13}$	50

$$\sigma_{ij} = \sqrt{\frac{N_i\sigma_i^2 + N_j\sigma_j^2}{N_i + N_j - 2}}\;;t = \frac{X_i - X_j}{\sigma_{ij}\sqrt{1/N_i + 1/N_j}}$$

$n = N_i + N_j - 2$
% = Percentage Value for the t Distribution (Confidence Level)

Table F.4 - Statistical Wear Data for the Four Ball Tests (Case 6)

Ester 7			Ester 8			Ester 9		
X_1	σ_1	N_1	X_2	σ_2	N_2	X_3	σ_3	N_3
0.79	0.01414	2	1.06	0.02828	2	0.80	0.02121	2
X_i = Mean; σ_i = Standard Deviation; N_i = Number of Tests								

Table F.5 - Confidence Intervals for Case 6 Using Small Sample Theory

X_1-X_2	0.27	σ_{12}	0.032	t_{12}	8.54	n_{12}	2	$\%_{12}$	99
X_2-X_3	0.26	σ_{23}	0.035	t_{23}	7.34	n_{23}	2	$\%_{23}$	99
X_1-X_3	0.01	σ_{13}	0.026	t_{13}	0.392	n_{13}	2	$\%_{13}$	63
$\sigma_{ij}=\sqrt{\frac{N_i\sigma_i^2+N_j\sigma_j^2}{N_i+N_j-2}};t=\frac{X_i-X_j}{\sigma_{ij}\sqrt{1/N_i+1/N_j}}$					$n = N_i + N_j - 2$ % = Percentage Value for the t Distribution (Confidence Level)				

Table F.6 - Statistical Wear Data for the Four Ball Tests (Case 7)

Ester 10			Ester 11			Ester 12		
X_1	σ_1	N_1	X_2	σ_2	N_2	X_3	σ_3	N_3
1.09	0.01414	2	1.00	0.05657	2	1.03	0.03536	2
X_i = Mean; σ_i = Standard Deviation; N_i = Number of Tests								

Table F.7 - Confidence Intervals for Case 7 Using Small Sample Theory

X_1-X_2	0.09	σ_{12}	0.058	t_{12}	1.544	n_{12}	2	$\%_{12}$	85
X_2-X_3	0.03	σ_{23}	0.067	t_{23}	0.450	n_{23}	2	$\%_{23}$	65
X_1-X_3	0.06	σ_{13}	0.038	t_{13}	1.575	n_{13}	2	$\%_{13}$	85
$\sigma_{ij}=\sqrt{\frac{N_i\sigma_i^2+N_j\sigma_j^2}{N_i+N_j-2}};t=\frac{X_i-X_j}{\sigma_{ij}\sqrt{1/N_i+1/N_j}}$					$n = N_i + N_j - 2$ % = Percentage Value for the t Distribution (Confidence Level)				

Table F.8 - Statistical Wear Data for the Four Ball Tests (Case 8)

Ester 13			Ester 14			Ester 15		
X_1	σ_1	N_1	X_2	σ_2	N_2	X_3	σ_3	N_3
0.96	0.0	2	0.88	0.01414	2	0.86	0.00707	2
X_i = Mean; σ_i = Standard Deviation; N_i = Number of Tests								

Table F.9 - Confidence Intervals for Case 8 Using Small Sample Theory

X_1-X_2	0.08	σ_{12}	0.014	t_{12}	5.657	n_{12}	2	$\%_{12}$	99
X_2-X_3	0.02	σ_{23}	0.016	t_{23}	1.266	n_{23}	2	$\%_{23}$	83
X_1-X_3	0.10	σ_{13}	0.007	t_{13}	14.14	n_{13}	2	$\%_{13}$	99

$$\sigma_{ij} = \sqrt{\frac{N_i\sigma_i^2 + N_j\sigma_j^2}{N_i + N_j - 2}}; t = \frac{X_i - X_j}{\sigma_{ij}\sqrt{1/N_i + 1/N_j}}$$

$n = N_i + N_j - 2$
% = Percentage Value for the t Distribution (Confidence Level)

Table F.10 - Statistical Wear Data for the Four Ball Tests (Case 9)

Alkbenz 1			Alkbenz 2			Min-2		
X_1	σ_1	N_1	X_2	σ_2	N_2	X_3	σ_3	N_3
0.68	0.0	2	0.65	0.06364	2	0.54	0.04243	2
X_i = Mean; σ_i = Standard Deviation; N_i = Number of Tests								

Table F.11- Confidence Intervals for Case 9 Using Small Sample Theory

X_1-X_2	0.03	σ_{12}	0.064	t_{12}	0.472	n_{12}	2	$\%_{12}$	65
X_2-X_3	0.11	σ_{23}	0.076	t_{23}	1.440	n_{23}	2	$\%_{23}$	85
X_1-X_3	0.14	σ_{13}	0.042	t_{13}	3.302	n_{13}	2	$\%_{13}$	98

$$\sigma_{ij} = \sqrt{\frac{N_i\sigma_i^2 + N_j\sigma_j^2}{N_i + N_j - 2}}; t = \frac{X_i - X_j}{\sigma_{ij}\sqrt{1/N_i + 1/N_j}}$$

$n = N_i + N_j - 2$
% = Percentage Value for the t Distribution (Confidence Level)

APPENDIX G : SOME REFERENCES RELATED TO REFRIGERANT COMPRESSOR-LUBRICATION

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